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DOUGLAS AIRCRAFT COMPANY

MCDONNELL DOUGLAS CORPORATION

NOTE

This manual is under constant revision. When errors in content are detected, each user has a responsibility to direct a revision request to the manager of the Hydraulics Manual or his staff.

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1.0 ADMINISTRATION AND PROCEDURES

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1.0 ADMINISTRATION AND PROCEDURES

1.1 INTRODUCTION Manuals are published under corporate authorization which recognizes the need for publication of standard, regulatory, or detailed information pertaining to a single function. The need for this revision to the Hydraulics Manual issues from Douglas' obligation to maintain a set of procedural information which will assist the designer in meeting the responsibilities and goals of the Hydromechanical Design Group.

PREFACE While the format and contents of this manual will differ in some respects from prior issues of the Hydraulics Manual, the purpose and usage are traditional. For this reason the introductory remarks from the preface of prior editions are quoted here; this preface is still an excellent guide to use of the Hydraulics Manual:

The Hydraulics Manual has been prepared to make valuable reference information readily available. The material presented in this manual results from the organization and standardization of tested experience and proved data which have been accumulated by the Hydraulics Groups of Douglas.

Outstanding achievements of Douglas attest to the success which is attained by sound design and meticulous workmanship. This fact dictates that the accumulated data on hydraulics systems set forth in this manual must be followed if a high standard of quality is to persist. On the other hand, it is recognized that progress depends upon the flexibility of engineering procedure and design. Recommendations for improvement are invited; but deviation from established practices should not be made without proper authorization.

DEFINITION Hydromechanical Design is the function which provides research, design, test, and production support for aircraft hydraulic power, actuation, and landing gear systems.

GROUP CHARTER The Hydromechanical Group is responsible for the design and installation of hydraulic systems and landing gear systems on all Douglas aircraft. The designers are responsible for the creative and inventive aspects of design and prepare their own engineering drawings, and they are involved in research and development as well as production support. Specifically, their design responsibilities include hydraulic systems and components, landing gears and hardware, carrier aircraft launching and recovery equipment, special related product equipment, and advance developments of directly related equipment. Their product-related research includes static and dynamic loads effects, producibility and cost effectiveness, and environmental considerations such as vibration, shock, thermal effects, and runway condition and contaminants. Specialists are also assigned from this group to Advance Design to provide the hydromechanical design for new aircraft programs.

Piping Development engineers are responsible for developing and then documenting the hydraulic piping systems for production on all types of aircraft, including commercial and military passenger and cargo transports, and military attack and fighter type aircraft. They also develop the piping runs, establish connectors, mountings, special applications, and also handle the piping development for instrumentation, oxygen, and pressurization systems on military aircraft. In cooperation with group test personnel, they conduct research into the development of lightweight and unitary piping modules, including special fitting designs and tooling development.

Civil/Mechanical Technology supports the hydromechanical research and design efforts by providing analysis and synthesis of complex devices, systems, and civil/mechanical structure. Personnel in this group have a thorough knowledge of hydromechanical systems, and are thoroughly experienced in the use of high-speed digital computers, differential analog computers, and direct analog computers. Hydraulic system thermal and dynamic analysis, servo analysis, and component-sizing trade studies are conducted with the aid of computer programs which can be readily modified for specific system characteristics. Aircraft dynamics during landing impact, taxi, arrest, and catapult and landing gear dynamic requirements and capabilities, are proved by analysis and computer study. Other computer programs provide the tool for brake energy requirements analysis, brake thermal studies, and antiskid system dynamics. A major contribution to design, sales, and airport compatibility needs is furnished by these specialists in landing gear/pavement/overpass interaction and strength analysis.

The Hydraulic Test engineers are responsible for the development and qualification testing of hydromechanical systems and components, and for design of the required laboratory facilities. They also formulate specifications covering vendor-supplied hydraulic equipment and evaluate vendor design proposals, test procedures, and qualification test reports. Major contributions in the formulation of military standards and specifications have been made by these personnel, in addition to maintaining the Hydromechanical Design Manual so that it reflects the latest state of the art. These engineers are active in the research and development testing of seals and piping system elements to provide greater reliability in aircraft hydraulic systems.

A Special Projects Group provides the management and implementation of Independent Research and Development (IRAD) and Contract Research and Development (CRAD). These assignments require formulation of plans, fulfillment of the Line Item Description (LID) requirements, periodic reporting to upper-level management, and formalized reporting of task completion or status at the end of each fiscal year. A Principal Engineer (PI) is assigned to guide and accept performance responsibility from inception through final reporting and close-out of each LID. Consultation, staff, and design engineers are provided for support of project-oriented research and design such as antiskid braking systems and interdisciplinary technologies.

Project groups are established as required for the design support of specific airplane programs. Personnel assignments are normally formed around the cadre of specialists assigned to the advance design phase; thus, continuity is assured during production, design, manufacturing, and customer product support.

1.2 APPROVALS The following approvals must be obtained:

By Section Chief	By Design Group Engineer	By Test Group Engineer	
X			Engineering Work Orders
X	X		Layouts, Proposal Drawings, Design Sketches
X	X	X	Installation, Assembly and Detail Drawings
X		X	Specification Control Drawings for hydraulic units
	X		Engineering Orders (normally)
X			EO's (when parts are scrapped)
X			EO's (when DAC production drawings are replaced)
	X		Material Orders (normally)
X			Material Orders (for Purchased parts and major items)
	X		Material Substitutions (normally)
X			Material Substitutions (for major and critical-strength items)
	X		Vendor's Information Requests (normally)
	X	X	VIR's (for vendor-designed hydraulic equipment)
	X	X	VIR's (when tests are involved)
X			Stop Orders
X	X		Time Estimates
X			Test Requests
X			Engineering Laboratory Work Orders
X			Test Time Increases for Engineering and Lab
		X	Detail Followups to Lab WO's for test instructions and material orders
X			Letters
X			Memos addressed outside of section
X	X		Design Engineering Reports
X		X	Test Reports

The Assistant Section Chief will act in the absence of the Section Chief.

1.3 LAYOUT WORK ORDER The Layout Work Order, Form No. 25-402, is used for the purpose of presenting necessary information concerning a job to the designer and as an aid in estimating the time required to complete the job.

a. Layout Work Orders shall be made out for all layout work as soon as the design can be broken down into layouts. The form shall be filled out as completely as possible. The job description shall explain the scope of the layout and all major requirements of the work involved.

b. The space for scheduled time is for the convenience of the designer so that he may properly budget his time. The spaces for hours and percent complete may be blocked in by the designer as the work progresses.

c. The following list of items shall be carefully considered and applied in filling out the order.

   1. Scope and requirements.

   2. Data from other groups for sketch and layout.

   3. Preliminary design drawings.

   4. Reference layout other airplanes.

   5. Specifications.

   6. Problems.

   7. Design Manuals.

   8. Purchased part or GFE drawings.

   9. Other references.

   10. Sketch of proposed design must be approved by the Section Chief, and personnel of other groups when affected.

   11. Finished layout must be approved by the Section Chief and personnel of other groups when affected.

d. The Order must be approved by the Group Engineer before starting the job.

e. It will be the duty of the Group Engineer to keep the draftsman's copy of the Layout Work Order up-to-date, making all corrections and additions to the form as the job progresses.

1.4 SPECIFICATION CONTROL DRAWINGS

a. Purchased parts which affect more than one group, such as motor-driven hydraulic pumps or solenoid valves, must have the approval of all Section Chiefs affected.

b. Purchased assemblies of vendor's own design should be ordered by Douglas specification control drawings number. The drawings must contain the following information:

   1. All dimensions necessary to ensure interchangeability, such as mounting holes, overall body dimensions, location and size of line connections.

   2. Part to be identified by Douglas number or stamped "Spec" and tagged if insufficient room for number.

   3. Notes requiring design and manufacture in accordance with the general Douglas procurement specification, and any additional information necessary to ensure interchangeability of operation and installation, such as maximum handle loads, maximum travel, etc.

4. Test requirements.

5. Identify approved sources per Drafting Manual.

c. To restrict procurement to a particular vendor fill out Form 60-431 (Rev. 5-59) "Engineering Justification for Limited Source Procurement" with signed approval by the Section Chief.

d. For additional information relative to procurement specifications ıgs, and specification control drawings, see Section 20.9 and Drafting Manual section 11.412.

1.5 DESIGN COMPUTATION RECORDS

1.51 PROCEDURE

a. All design computations are to be recorded on 8-1/2 by 11 sheets.

b. All the sheets for each model are to be properly tabulated, indexed and filed in a single design computation book.

1.52 RESPONSIBILITY

a. The Designer of each unit is responsible for the preparation of the computation sheets.

b. The Design Group Engineer is responsible for maintaining the book.

1.53 SCOPE Computation sheets are to be prepared for the following items:

a. Landing gear shock struts.

b. Landing gear retraction mechanisms.

c. Arresting gear mechanisms.

d. Hydraulically actuated mechanisms.

e. Actuating cylinders.

f. All flight control hydraulic units.

g. Reservoirs.

h. Cooling provisions.

i. All power supply selections.

j. Any other calculations that are at all extensive or complex.

1.54 CONTENT The computation sheets shall note the design requirements including a complete, concise, statement of function; show in a step by step procedure that suitable function and strength are obtained in all assemblies and detail parts; include accumulated-tolerance checks as required and a recommended adjustment procedure. Any inter-group liaison items, design features left for development, and/or extraordinary testing required shall be suitably recorded. Helpful related material should be referenced, or if available appended to the computations. The Design Check List (Section 20) should be reviewed for design objectives.

1.55 FUTURE USE These computations are to serve as guides for solving similar design problems encountered in the future and for reference in cases where questions subsequently arise concerning the design.

1.6 REPORTS Engineering information is generally conveyed in two ways, namely, drawings and reports. Reports are usually the product of research or investigation pursued along a given line. To accomplish their purpose effectively, they should be clear, concise, and complete. To be consistent, as much care should be exercised in the preparation of reports as was expended on the research leading to the report.

a. Engineering reports are, where possible, to be written as soon as tests or investigations are completed. In case a test or investigation is partially completed and the work is to be dropped, a report should be written covering the amount of work completed.

b. Generally, reports should be organized approximately in the form shown below. Reference can also be made to ES-20980, Engineering Report Manual. After the report has been prepared, it shall be submitted in rough draft form to the Group Engineer (Test Group or Design Group as applicable). After corrections, the report may be typed in reproducible form and submitted to the Section Chief for approval.

1.61 FORM TO BE FOLLOWED FOR ENGINEERING REPORTS

a. Summary This section should provide the reader with a short digest of the report. Its length should normally not exceed one paragraph. It should normally be the last part of the report to be written.

b. Introduction This section should orient the reader with respect to the problem under consideration. It may contain a statement of the problem investigated, results of previous investigations along similar lines, and the purpose of the present investigation. A definition of symbols and terms employed should also be included.

c. Conditions and Procedures This section should provide enough information to form a clear picture of the means or methods by which the results of the research or investigation were obtained.

Note: Particular care should be exercised in writing this section and the "Results," to enable development of valid conclusions.

d. Results This section should contain the exact answers or results obtained from the research or investigation.

e. Discussion This section should constitute the writer's interpretations of the results obtained from the research or investigation.

f. Conclusions This section should contain a brief statement of the conclusions the writer draws as a result of the investigation or research. Avoid statements which are too broad in their meanings. The conclusions should contain only what may be logically drawn from the results.

g. References While extended bibliographies normally have no place in a report, carefully chosen references can be of value to the reader for further investigation of the subject.

Note: For an expanded treatment on test reports, see Section 90. Also see MDC J0204, "Preparation of Technical Reports."

10. BASIC HYDRAULICS SYSTEMS

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10. BASIC HYDRAULICS SYSTEMS (Cont)

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10.1 GENERAL

10.11 ADVANTAGES OF HYDRAULIC SYSTEMS Why have hydraulics found such wide acceptance for such a variety of aerospace applications? Because hydraulic power transmission offers several definite advantages over mechanical and electrical methods, especially in the higher power ranges of aerospace systems. These advantages may be summarized as follows:

- Most efficient method of power amplification in which a small applied control effort results in a large power output with a minimum of weight and space required.
- Power transmission capabilities that provide more exact output load rate, position and magnitude.
- Smooth, vibrationless power output that is little affected by load variations.
- Infinitely variable rotary or linear motion control within set limits in either direction with minimum manual control and maximum response.
- Ability to handle a number of different loads simultaneously, independently or in sequence with the same power source.
- Hydraulic fluid carries away the heat generated by internal losses and acts as a lubricant to increase component life.

10.12 REDUNDANCY Pilot control force from flight-control system surface loads has increased along with aircraft size and speed until the pilot can no longer provide this force manually. The hydraulically boosted actuator has become the accepted control method for augmenting the pilot force. However, in the event of a power system failure, it is necessary for one or more backup systems to be available. The loss of hydraulic power may result from a leak or loss of an engine that drives the pump. It has become common practice to design these systems in such a manner that the normal hydraulic power demands are distributed among the redundant systems so as to balance out the demand among them. Figure 10-1 illustrates a three-system redundancy. Where four engines are available, a four-system redundancy is common. There must be no fluid connections between each of the redundant hydraulic systems. However, they can be mechanically connected by a motor/pump such that one system-driven motor can mechanically drive a pump on another system, thus providing auxiliary or emergency flow capability to that system as shown in Figure 10-1.

Another method for providing redundancy is by incorporating Reservoir Level Sensing (RLS). This concept is illustrated in Figure 10-1.1. It is currently employed on the MDC F-15 aircraft.

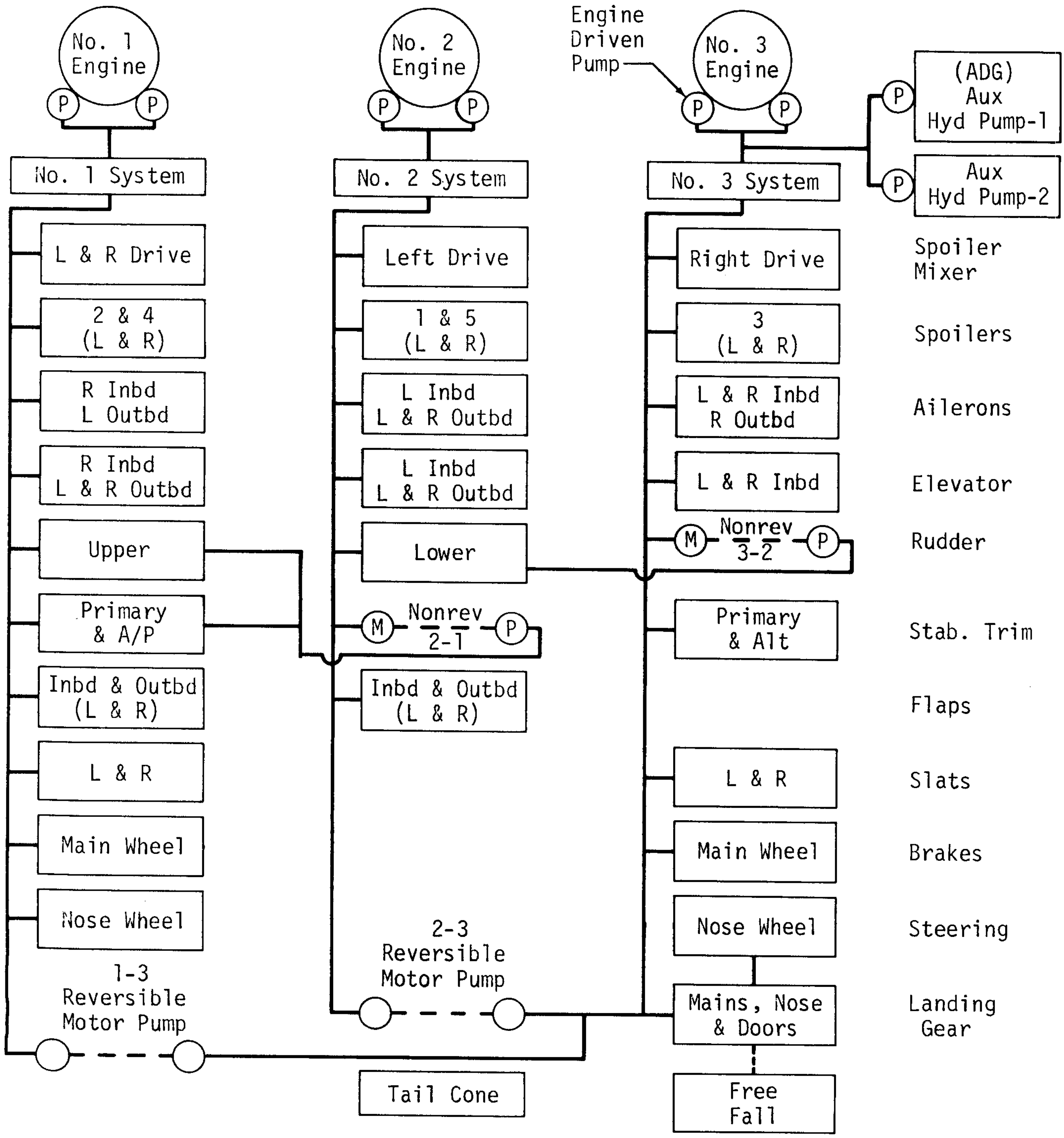

FIGURE 10-1 DC-10 HYDRAULIC SYSTEM POWER DISTRIBUTION DIAGRAM

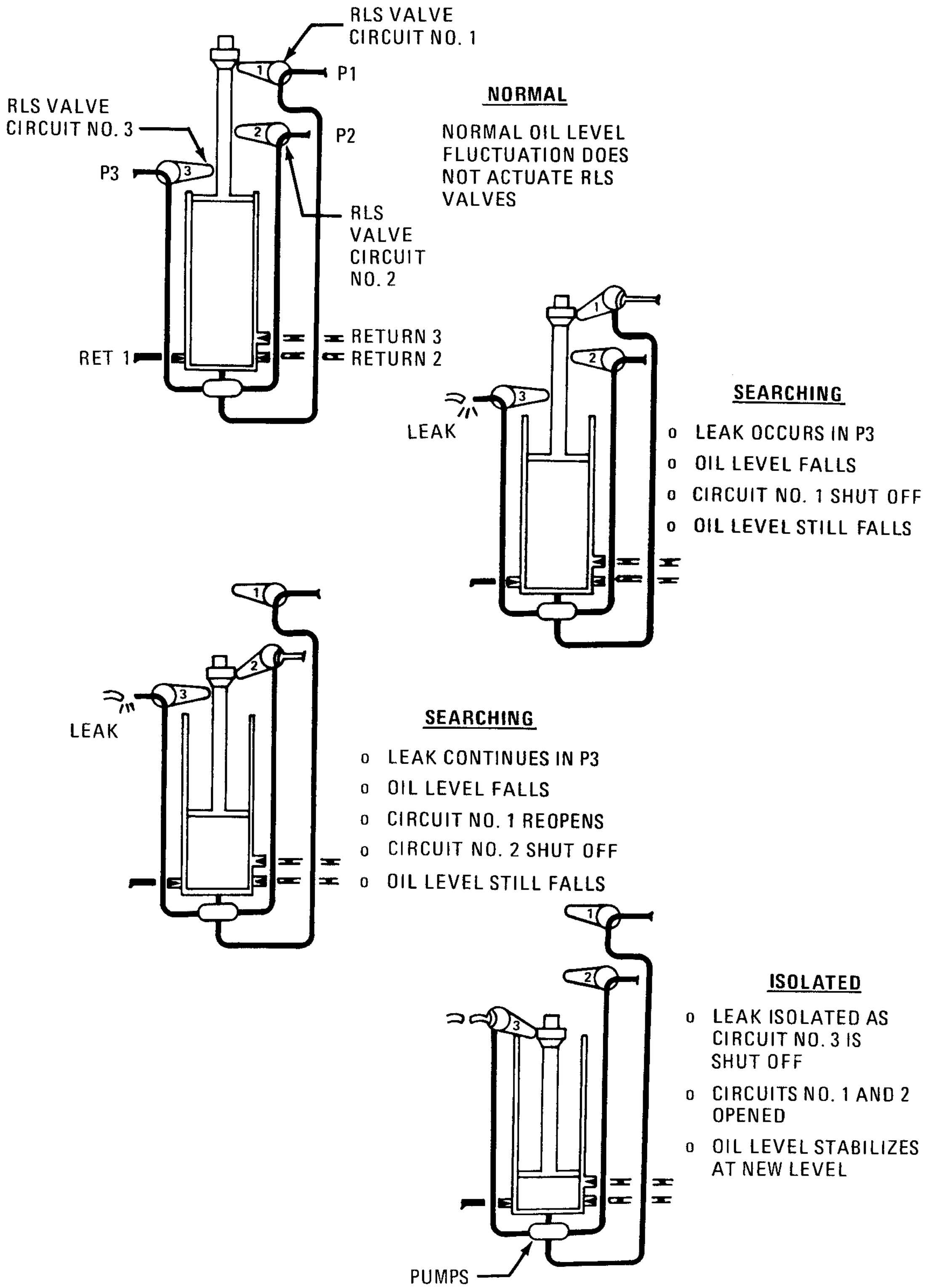

FIGURE 10-1.1 RESERVOIR LEVEL SENSING SCHEMATIC

10.2 SYSTEM DESIGN

10.21 JOB DEFINITION Every aircraft hydraulic system involves transfer of energy to an output device such as an actuating cylinder or a hydraulic motor. The first consideration in system design therefore should be to determine what functions require hydraulic power and how many output devices are required for each function.

10.22 APPLICABLE REGULATIONS AND SAFETY CONSIDERATIONS The applicable regulations and safety considerations of the aircraft are a significant influencing factor in the design of the hydraulic system. These may vary with the type of aircraft and customer involved. Current commercial aircraft are required to satisfy the following:

- FAR 25
- FAA special condition No. 1
- No single failure shall cause loss of more than one hydraulic system
- Geometrically separate systems
- No system interconnects.
- Design Safety Manual, Report No. MDC J7411.

10.23 HORSEPOWER VERSUS FLIGHT PROFILE Major decisions are required prior to sizing the hydraulic system. First, the load history for each of the subsystems must be calculated. Then the nominal system pressure must be selected. (This is currently 3000 psi for most American commercial transport aircraft.) A convenient method for tabulating the flight profile data is illustrated in Figures 10-2.1 and 10-2.2. The hydraulic operations for each condition are identified and the flow demand and horsepower calculated. These are subsequently transferred to a flight profile plot similar to Figure 10-2. A horsepower versus flight profile plot is very useful in system design and provides a visual reference of power demands made on the pump or pumps as the aircraft moves through its operations. Figure 10-2 shows a typical plot. In order to obtain this plot, two parameters concerning the output device must be established:

1. Load
2. Operating Rate

Before further system design can be accomplished, an output device must be tentatively selected. Its displacement, coupled with its operating rate, determines flow rate. The quantity and size of the output device determines the pressure level required. Tentative sizing of the output device (actuating cylinder) is described in Section 40.

PARK – EXTEND SLATS

ENGINE PUMP AVAILABLE FLOW SEE* (ENG IDLE 18.2 GPM)	RUDDER	AILERON	ELEVATOR	SPOILER	WING SLATS	WING FLAPS			NOSE WHEEL STEERING	BRAKES	LANDING GEAR	THRUST REVERSER	CARGO DOORS		LEAKAGE	TOTAL
DEG/SEC					25/3.9										CONST	*18.2
SYSTEM NO. 1					14.0		X								1.0	15.0
SYSTEM NO. 2							X								1.0	1.0
SYSTEM NO. 3							X								1.0	1.0
SYSTEM NO. 4					14.0		X								1.0	15.0

X – FLOW INCLUDED IN LEAKAGE

DEG/SEC																
SYSTEM NO. 1																
SYSTEM NO. 2																
SYSTEM NO. 3																
SYSTEM NO. 4																

DEG/SEC																
SYSTEM NO. 1																
SYSTEM NO. 2																
SYSTEM NO. 3																
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DEG/SEC																
SYSTEM NO. 1																
SYSTEM NO. 2																
SYSTEM NO. 3																
SYSTEM NO. 4																

C15A-S3-003B

FIGURE 10-2.1. PARK – DEMAND FLOW LOADS

CONDITION	OPERATIONS	
1. PARK	– EXTEND SLATS	ENGINE IDLE RPM
2. TAXI	– NW STEERING – SET FLAPS AT X-DEGREE POSITION – BRAKING WITH THRUST REVERSERS – BRAKING WITH THRUST REVERSERS – STEERING AND BRAKING	ENGINE IDLE RPM DEPLOYING STOWING
3. TAKEOFF	– ROLLOUT (RUDDER, AILERON, NG STEERING, SPOILER) – ROTATION (RUDDER, AILERON, ELEVATOR, SPOILER)	ENGINE TAKEOFF RPM
4. TAKEOFF (RTO) (55-PERCENT THRUST)	– RTO (RUDDER, AILERON, SPOILER BRAKES, STEERING, THRUST REVERSER) – AFTER RTO (RUDDER, SPOILER, NG STEERING) – THRUST REVERSER STOW (RUDDER, NG STEERING, THRUST REVERSER)	ENGINE TAKEOFF RPM ENGINE IDLE RPM ENGINE IDLE RPM
5. CLIMB	– GEAR UP, CALM AIR, FLIGHT CONTROLS AT 1/4 RATE	ENGINE CLIMB RPM
6. CLIMB	– GEAR UP, TURBULENT AIR, FLIGHT CONTROLS AT 1/2 RATE	ENGINE CLIMB RPM
7. CRUISE	– CALM AIR, FLIGHT CONTROLS AT 1/4 RATE	ENGINE CRUISE RPM
8. CRUISE	– TURBULENT AIR, FLIGHT CONTROLS AT 1/4 RATE	ENGINE CRUISE RPM
9. DESCENT	– TURBULENT AIR, STABILIZER TRIM, FLIGHT CONTROLS AT 1/2 RATE	ENGINE DESCENT RPM
10. DESCENT (EMERGENCY)	– TURBULENT AIR, STABILIZER TRIM, FLIGHT CONTROLS AT 1/2 RATE	
11. LET DOWN	– TURBULENT AIR, STABILIZER TRIM, FLIGHT CONTROLS AT 1/2 RATE, FLAPS AND SLATS EXTEND, GEAR DOWN	ENGINE DESCENT RPM
12. APPROACH	– TURBULENT AIR, STABILIZER TRIM, FLIGHT CONTROLS AT 100 PERCENT RATE	ENGINE DESCENT RPM
13. LANDING AND ROLLOUT	– FLIGHT CONTROLS AT 100 PERCENT RATE SPOILERS UP, BRAKES, THRUST REVERSER, NOSE WHEEL STEERING	ENGINE IDLE RPM

FIGURE 10-2.2. MISSION PROFILE HYDRAULIC FLOW DEMAND

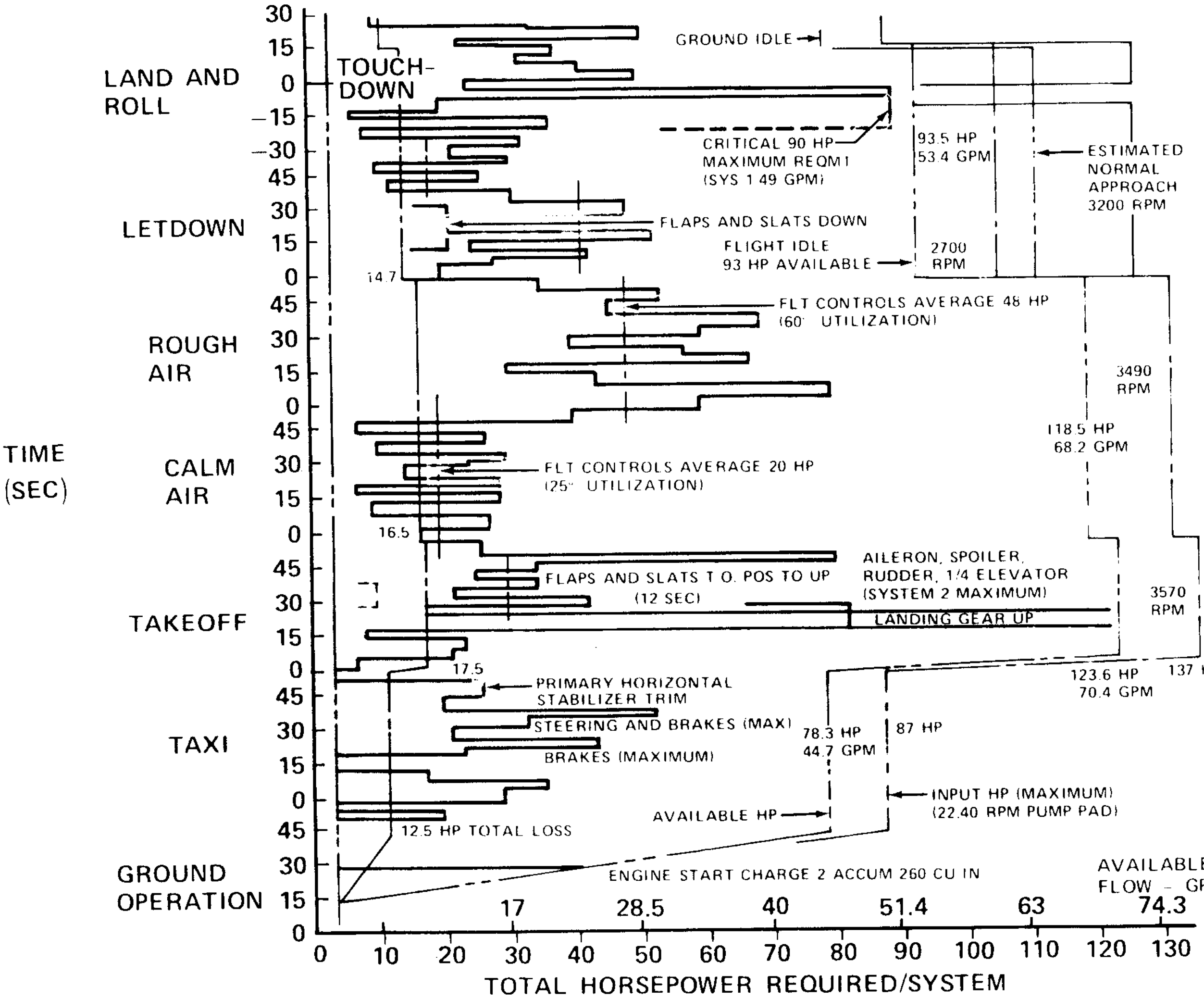


Figure 10-2. Horsepower vs Flight Profile Plot

When the flow rate and pressure level of each circuit have been determined, the power requirement of the circuit can be calculated by the following formula:

$$HP = \frac{QP}{1714} \qquad \text{where} \quad Q = \text{Flow rate, gpm}, \quad P = \text{Pressure, psi}, \quad HP = \text{Horsepower}$$

The power demand of each circuit is calculated, in turn, and plotted on the horsepower vs flight profile plot to determine the peak demand of the system.

Other useful horsepower conversions can be found in Section 120.

10.24 PUMP SIZING Analysis of Figure 10-2 indicates that the hydraulic power supply is critical for takeoff and landing conditions. Forty-nine gpm is required at 2700 pump rpm.

During landing, the formula for determining pump size is as follows:

Minimum Required Pump Size:

$$\frac{Q \times 231}{N \times rpm \times vol\ efficiency} = \frac{49 \times 231}{2 \times 2700 \times 0.90} = 2.34\ in.^3/rev$$

where:
Q = Flow rate, gpm
RPM = Pump revolutions per min
Vol EFF = Pump volumetric efficiency (obtained from pump manufacturer)
N = No. of pumps per system

NOTE: Hydraulic pumps and motors usually have about the following efficiencies:

Volumetric	90 percent
Mechanical	95 percent
Overall	85 percent

In using the above efficiencies, it should be remembered that low volumetric efficiency causes a pump to deliver a low flow or a motor to rotate more slowly. Volumetric efficiency has no effect upon torque. Low mechanical efficiency increases the torque required to drive a pump or decreases the torque output from a motor. Low overall efficiency increases the power required to drive a unit when the power output is known or decreases the power output of a unit when the power input is known.

10.25 LINE SIZING The choice of system line sizes is usually based on trade studies which balance energy loss against cost and weight. Large diameter tubing will conduct the fluid with lower pressure loss than smaller sizes, but will weigh more, be harder to bend, flare, swage, weld, or braze and will cost more.

10.251 BASIC REQUIREMENTS AND RULES OF THUMB

10.2511 Fluid Velocity Limitation A historical rule of thumb is to size the tubing for an average flow velocity of 15 feet per second. Table 10-1 indicates the average rate of flow for tube size between 1/4-inch and 1-1/2 inch OD, based on a maximum wall thickness of standard aluminum-alloy tube conforming to Standard AND 10106, for laminar flow. It is lifted directly from MIL-H-5440 where it has been included in the original issue and all subsequent revisions through Revision D which included the following statement:

"3.5.7 Fluid velocity limitations. It is desired that tube sizes be such that the average velocity of oil in pressure and return lines leading to the directional control valves shall not be in excess of approximately 15 feet per second, except where system analysis shows that proper functioning can be achieved even though

TABLE 10-1

MAXIMUM LAMINAR FLOW FOR NOMINAL TUBE SIZES

MIL-H-5606B or Phosphate Ester at 100°F

Tube Dash No.	Tube O.D.	Flow - GPM
-4	1/4	1.2
-5	5/16	2.3
-6	3/8	3.5
-8	1/2	6.0
-10	5/8	10.5
-12	3/4	16.0
-16	1	29.0
-20	1-1/4	45.0
-24	1-1/2	70.0

the rate be higher. The velocity of flow in the suction lines shall be governed by the pressure requirements for suction lines, as specified herein."

However, it should be noted that in MIL-H-5440E such reference to the 15 fps velocity and the table have both been deleted in favor of the following:

"3.5.7 Fluid velocity limitations. Tubing size and maximum fluid velocity for each system shall be determined considering, but not limited to, the following:

(a) Allowable pressure drop at minimum required operating temperatures.

(b) Pressure surges caused by high fluid velocity and fast response valves.

(c) Back pressure in return lines, as it may affect brakes and pump case drain lines.

(d) Pump inlet pressure, as affected by long suction lines, and a high response rate variable volume pump. Consideration should be given to both pressure surges and cavitation."

The following subparagraph from MIL-H-5440E is also pertinent to the line sizing analysis:

> "3.5.7.1 Fluid flow effects. The systems shall be so designed that malfunctioning of any unit or subsystem shall not occur because of reduced flow, such as created by single-pump operation of a multipump system, or reduced engine speed of a single-pump system. The system shall also be so designed that increased flow will not adversely affect the proper functioning of any unit or subsystems, such as increased flow rate caused by accumulator operation, or units affected by air load operation."

It is a matter of record that a number of high-flow systems in recent aircraft have been designed with lines resulting in fluid velocities of 25-30 fps and even higher in order to avoid excessive weight penalties.

10.2512 Pressure Drop Balance A second rule of thumb that has been commonly used for initial sizing of hydraulic system lines is to allow 1/3 of the available working pressure (1000 psi in a 3000-psi system) for tubing pressure drop at the minimum full-performance design temperature, and size actuators (and valves) to meet the maximum-rate force requirements with the remaining 2/3 pressure available.

In a complex system with a number of load combinations, many of which peak at different portions of the flight regime and thus at different minimum fluid temperatures, the task of choosing line sizes is often tedious and time consuming. Selecting tube diameters that will produce a desired pressure differential available for given loads is relatively simple; but, selecting line sizes that produce the lightest system (while still satisfying the pressure requirements at the loads) requires a high degree of experience. As the number of lines in a system increases, the problem of minimizing the system weight becomes quite formidable. In order to simplify the work, computer programs have been devised and used to give a greater degree of optimization than is feasible to obtain by hand calculations. It is even possible to let the pressure differential across the actuators be a variable in the solution and obtain a truly minimum-weight system for lines and actuators. This approach can produce a worthwhile weight reduction in a high-flow system with long lines where an increase in allowable tubing pressure drop can provide more weight saving than needs to be added for the slightly larger actuator sizes. (See Paragraph 125.07.)

10.2513 Pressure Surge Limits Pressure surges result from both the rapid release of stored energy into a line or actuating cylinder and from the abrupt closure of a valve from a high-flow condition. In the latter case, the magnitude of the pressure rise is a direct function of the original fluid velocity, i.e., for rigid pipes:

$$\text{Pressure rise } \Delta P = V\sqrt{B\frac{w}{g}}\ \text{psi}$$

where V = original fluid velocity in./sec

B = fluid bulk modulus psi

w = fluid density lb/in.^{3}

g = 386 in./sec^2

For MIL-H-5606 fluid at 100°F, with the following physical characteristics:

B = 270,000 psi with any entrained gas removed

B = 200,000 psi in a typical system

w = 0.845 specific gravity = 0.0305 lb/in.^{3}

ΔP = 4.0 x V psi pressure rise

For military aircraft, MIL-H-5440 specifies that "peak pressure (ripple or surge) resulting from any phase of the system operation shall not exceed 135 percent of the main system, subsystem, or return system operating pressure when measured with electronic equipment, or equivalent." In a 3000-psi system, the surge peak should not exceed 4050 psi.

Therefore, for flow velocities above 262 in./sec (22 ft/sec), valve closing times must be limited to values greater than $2\ell/C$ seconds where ℓ = length of pressure line from pump to valve and C = the speed (of sound) at which the fluid pressure wave will travel along the oil filled tube.

$$C = \sqrt{\frac{B}{w/g}}$$

For MIL-H-5606 at 100°F and 3000 psi

$$C = \sqrt{\frac{200{,}000}{0.0305/386}} = 50{,}000 \text{ in./sec}$$

For a 20-foot pressure line, valve closing time must be greater than 0.0096 second.

For high flow velocity conditions, the required valve closure time, T, can be calculated from the following equation:

$$T = \frac{\Delta P}{\Delta P_a} \times T_R$$

where ΔP = the rise in pressure due to instantaneous valve closure

ΔP_a = allowable pressure rise

T_R = time for a return trip of the pressure wave from the pump

$= \ell/C$

For commercial transport airplanes, an even more stringent surge pressure limit is imposed. FAR 25, Volume III, Transmittal I, Amendment 25-23, Effective 5-8-70, specifies that transient pressures will not exceed 125 percent of the design operating pressure.

10.252 MINIMUM FULL-PERFORMANCE DESIGN TEMPERATURES As indicated previously, one of the primary considerations in designing tubing is in meeting the allowable pressure drop at minimum operating temperature. The latter is often specified directly in the airplane performance specification or indirectly in terms of a specified time in which a military aircraft must be ready for takeoff following an alert during cold coak at a specified low ambient temperature. In the latter case, various means of warming up the system fluid can be considered in determining the fluid temperature at the flight condition where full performance must be met. Trade studies of all possible methods are required to arrive at the optimum combination of warmup equipment and tube sizes which result in the minimum (or acceptable) weight and cost penalties. The applicable requirements specified for military aircraft systems, and typical practices for commercial aircraft are as follows:

10.2521 Minimum Operating Temperature for Sizing Lines for Military Aircraft

- Pump Suction Lines

  The following requirements for suction lines are quoted directly from MIL-H-5440:

  "3.10.29.9 Suction line.

  3.10.29.9.1 Suction line flow The suction line from the reservoir to the power pump or pumps shall be so designed as to provide adequate flow and pressure at the pump inlets, with the power pump or pumps operating at the maximum fluid volume output required at the service ceiling of the aircraft, and with the hydraulic fluid at the expected stabilized temperature, but this stabilized fluid temperature shall be not warmer than -20°F, unless provisions are incorporated to control the fluid temperature. Due regard may be given to selecting an altitude less than service ceiling for any system in which the pump does not operate during "in flight" conditions, but this altitude shall be not less than 10,000 feet above sea level. Test data and analysis shall be furnished for the approval of the procuring activity, showing that satisfactory service life of the pump or pumps will be indicated under all operating conditions of the aircraft. This information shall be included in the system design report (see 3.4.1.10).

3.10.29.9.2 Suction line, cold starting Unless otherwise specified by the procuring activity, during cold starting on the ground at all ambient air temperatures down to -65°F, the above conditions of suction-line pressure drop shall not apply until stabilized fluid operating temperatures have been obtained.

3.10.29.9.3 Suction-line filters Filters shall not be installed in the suction line to power-driven pumps, unless such a location for the filter is specifically approved by the procuring activity. If a suction-line filter is contemplated, complete test data shall be submitted showing that the pressure drop through the filter will not interfere with the operation of the hydraulic system, particularly at high altitudes. Suitable relief valves shall be provided in the filter."

- Landing Gear Retraction Systems (Air Force)

The following requirement is lifted from the AFSC Design Handbook 2-1 Airframe, Design Note 4A6:

SUB-NOTE 1 (1) Landing Gear Performance Requirements

	IF	AND	THEN	AND	
RULE	The Landing Gear System is	The Temperature is	The maximum allowable time to extend and lock the gear is 1	The maximum allowable time to retract and lock the gear is 2	AND
1	Power	Above -20°F(-29°C)	15 sec	10 sec	The gear must be retracted and locked before the aircraft reaches 75% of the gear placard speed at the maximum rate of acceleration.
2	Operated	-65°F(-54°C) to -20°F(-29°C)	30 sec 3	10 sec 3	

NOTES:

1. If the landing gear is used as a speed reducing device, the time to extend and lock the gear must be determined by the desired performance.

2. For zero-launch aircraft, the landing gear retraction sequence must be completed one second prior to reaching the gear placard speed.

3. The system must meet these requirements when stabilized at the temperature extremes without allowing warmup time.

4. For multiengine aircraft, the system must meet these requirements during an engine-out condition.

It should be noted that this requirement is more severe than that previously specified in AFSCM 80-1 (HIAD) which allowed the time of operation (for both retraction and extension) between -65 and -20°F to be up to "twice the fastest time obtained at normal temperatures (approximately +70°F)."

- Flight Control Systems (Air Force)

  AFSC DH2-1 specifies the following:

  Ensure that, after the initial breakaway, the increase in force required to operate the control subsystem at -65°F, does not exceed 150 percent of the force required at +70°F (+21°C). Design control subsystems to meet the anticipated temperatures encountered during flight.

- Flap Actuation Systems (Air Force)

  AFSC DH2-1 specifies that, between -65 and -20°F, the time of operation must be not more than 50 percent greater than the normal speed selected with all components of the flap actuating mechanism stabilized at the specified extreme temperature, and without assuming time for warmup of the components.

10.2522 Minimum Operating Temperatures for Sizing Lines for Commercial Transport Airplanes The Federal Aviation Regulations do not specify any low temperature requirements for hydraulic systems in transport category airplanes. However, the following minimum operating temperature capabilities are typical for large commercial jet transport airplanes:

Ambient temperature range in flight: +110 to +130°F
Minimum cold start capability (ambient and fluid): -65°F
Minimum fluid temperature for rated pump suction flow: -20°F
Minimum fluid temperature for full system performance: +50°F

10.3 SYSTEM COMPONENTS

Simply stated, any power control system, and this includes hydraulics, requires a power supply (i.e., pump), some means of transmitting this power (i.e., piping), a device to control the power transmitted (i.e., valves) and some kind of output device (i.e., actuating cylinder or hydraulic motor) to transform the kinetic energy of the fluid into mechanical action. In current aircraft hydraulic systems, all of these components are utilized in addition to many more. Some of these components are covered by the following sections of this manual:

Piping	Section 40
Actuating Cylinders	Section 60
Valves	Section 70

Information regarding the most important remaining components is included in this section.

10.31 PUMPS

10.311 GENERAL In current aircraft hydraulic systems, the pressure compensated variable delivery pump, which automatically adjusts itself according to the pressure in its delivery line, has proven to be the most efficient method of power distribution. A typical pump of this type is shown in Figure 10-3.

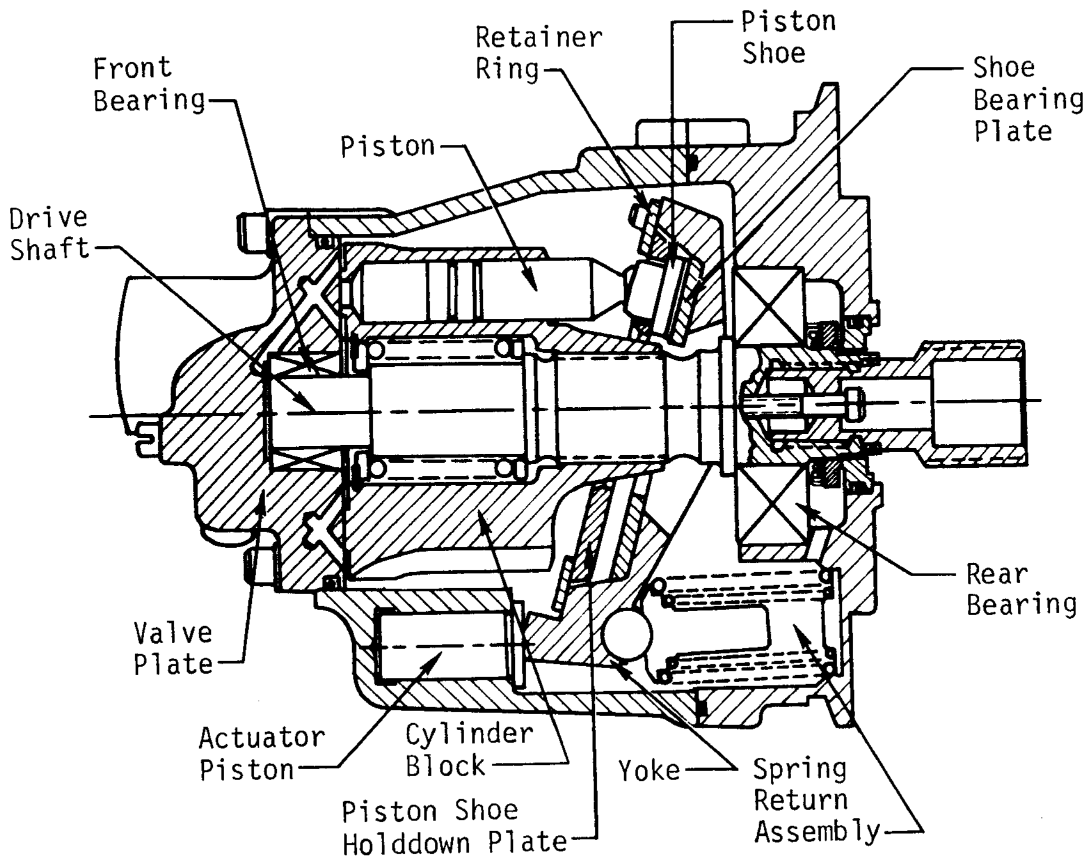


FIGURE 10-3

10.312 PUMP RATING MIL-P-19692 defines the terminology generally used in rating variable delivery pumps. Some of these definitions are listed below:

1. Rated Discharge Pressure The rated discharge pressure of a pump is defined as the maximum pressure against which that pump is designed to operate continuously at rated temperature and rated speed.

2. Maximum Full Flow Pressure The maximum full flow pressure is defined as the maximum discharge pressure at which the pump control will not be acting to reduce pump delivery at rated temperature, speed and inlet pressure. (Also called "cut-off" pressure.)

3. Rated Inlet Pressure The rated inlet pressure of a pump is defined as the indicated pressure at the inlet port of the pump when it is operating at rated speed, maximum full flow pressure and rated temperature.

4. Rated Temperature The rated temperature of a pump is defined as the maximum continuous fluid temperature at the inlet port of the pump.

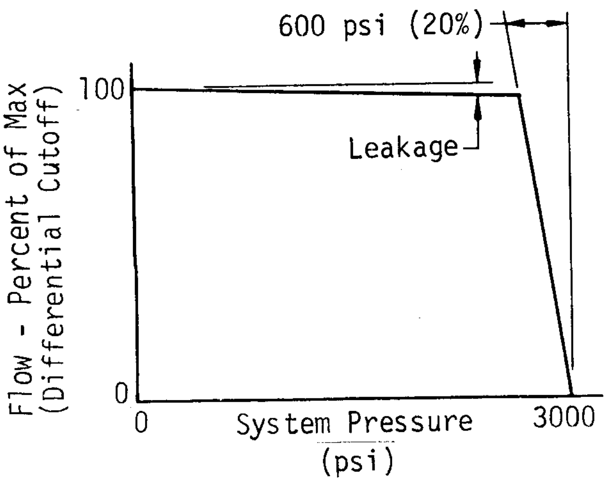
5. Maximum Displacement The maximum displacement of a pump is defined as the maximum theoretical volume of hydraulic fluid delivered in one revolution of its drive shaft.

   The maximum displacement of any pump model shall be determined by calculation from the geometry and dimensions of the pump. The dimensions of the cylinder bore and maximum stroke shall be taken at their nominal values as stated on the applicable standards. The effects of allowable manufacturing tolerances, of deflections of the pump structure, of compressibility of the hydraulic fluid, of internal leakage, and of temperature shall be excluded from the calculation, because the maximum displacement is intended to be an index of the size of the pump rather than of its performance.

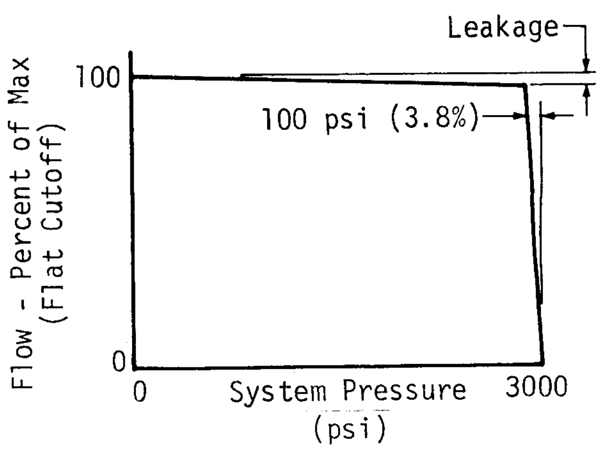
6. Rated Delivery The rated delivery of a pump is defined as the measured output of the pump under conditions of rated temperature, rated speed and maximum full flow pressure.

7. Rated Speed The rated speed of a pump shall be the maximum speed at which the pump is designed to operate continuously at rated temperature, and rated discharge pressure.

10.313 PRESSURE REGULATION Pressure compensated pumps can be supplied with various control arrangements. The pressure regulation characteristics of a pump are determined by the type of control employed. The standard control types are the "flat cutoff" and "differential cutoff" compensators shown in the following figures:



DIFFERENTIAL CUTOFF



FLAT CUTOFF

10.314 <u>TYPICAL ENGINE-DRIVEN PUMP OPERATION</u> Schematic diagrams of the engine-driven operational modes are presented in Figures 10-4 through 10-7. The pump contains nine pistons which reciprocate in a cylinder barrel which is driven by an external power source. The pistons are constrained by a hold-down plate and hydrostatically balanced shoes to run on a face cam. As the barrel rotates the pistons reciprocate within their bores, intaking and discharging fluid through a stationary valving surface on the port cap. The cylinder barrel is supported by a drive shaft or roller bearing so placed as to react against the piston pressure force components which act at right angles to the cylinder barrel polar axis.

The operation of the pump compensating mechanism depressurizer and blocking valve is illustrated schematically in Figure 10-4. The pump is shown delivering a steady flow rate which is less than full demand. The stroking piston is stabilized by the three-way valve in the pressure compensator at a position which will provide a pump displacement consistent with system demand. The three-way valve is controlled by the outlet pressure sensing piston which regulates pressure at a level determined by the pressure sense spring. The depressurizing solenoid is de-energized which commands the blocking valve to remain open. The operation of the pressure compensator system is illustrated in Figure 10-5 for an increasing demand flow rate. Figure 10-6 illustrates its operation for a decreasing demand flow rate. The depressurizer operation is illustrated in Figure 10-7.

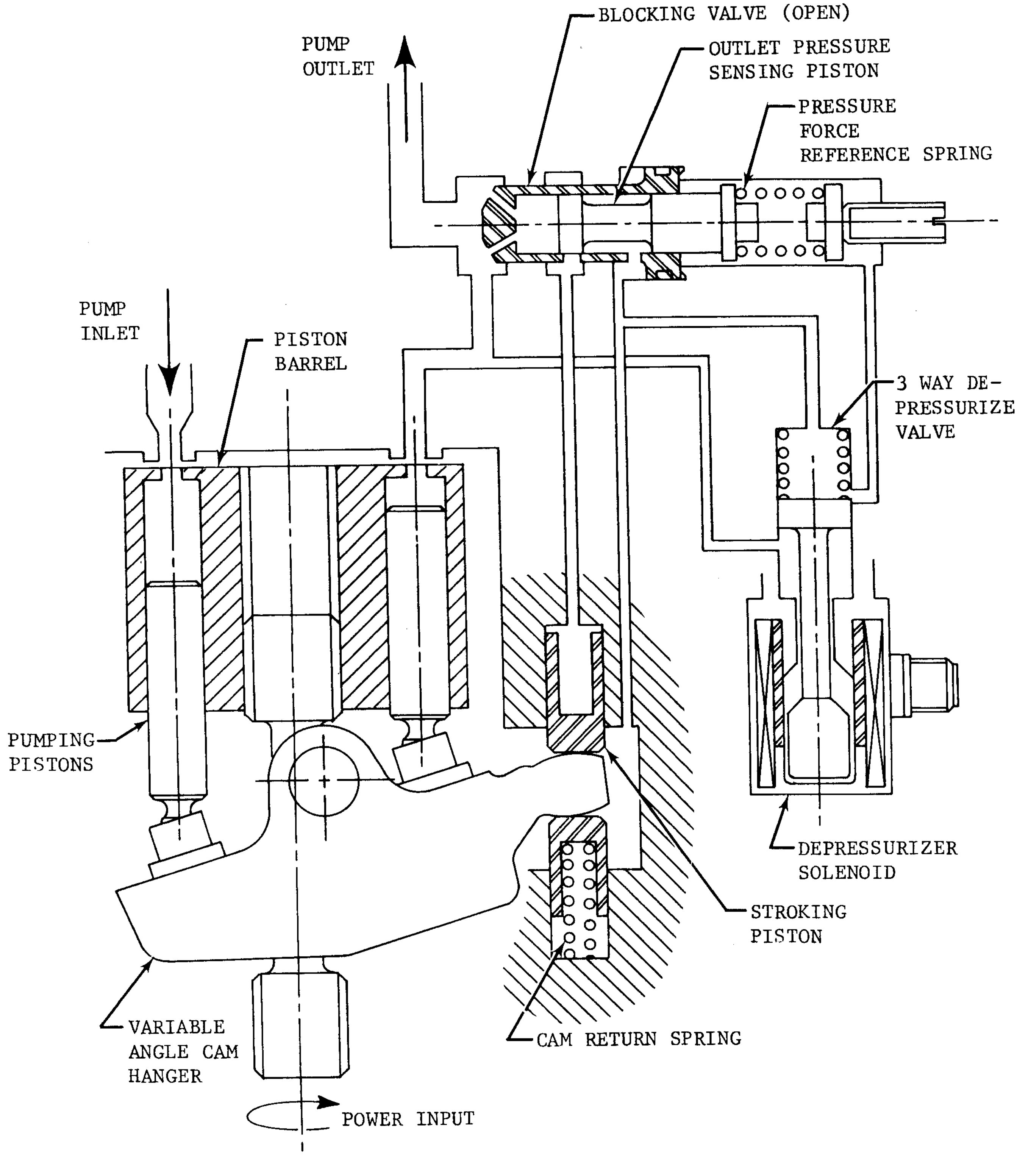
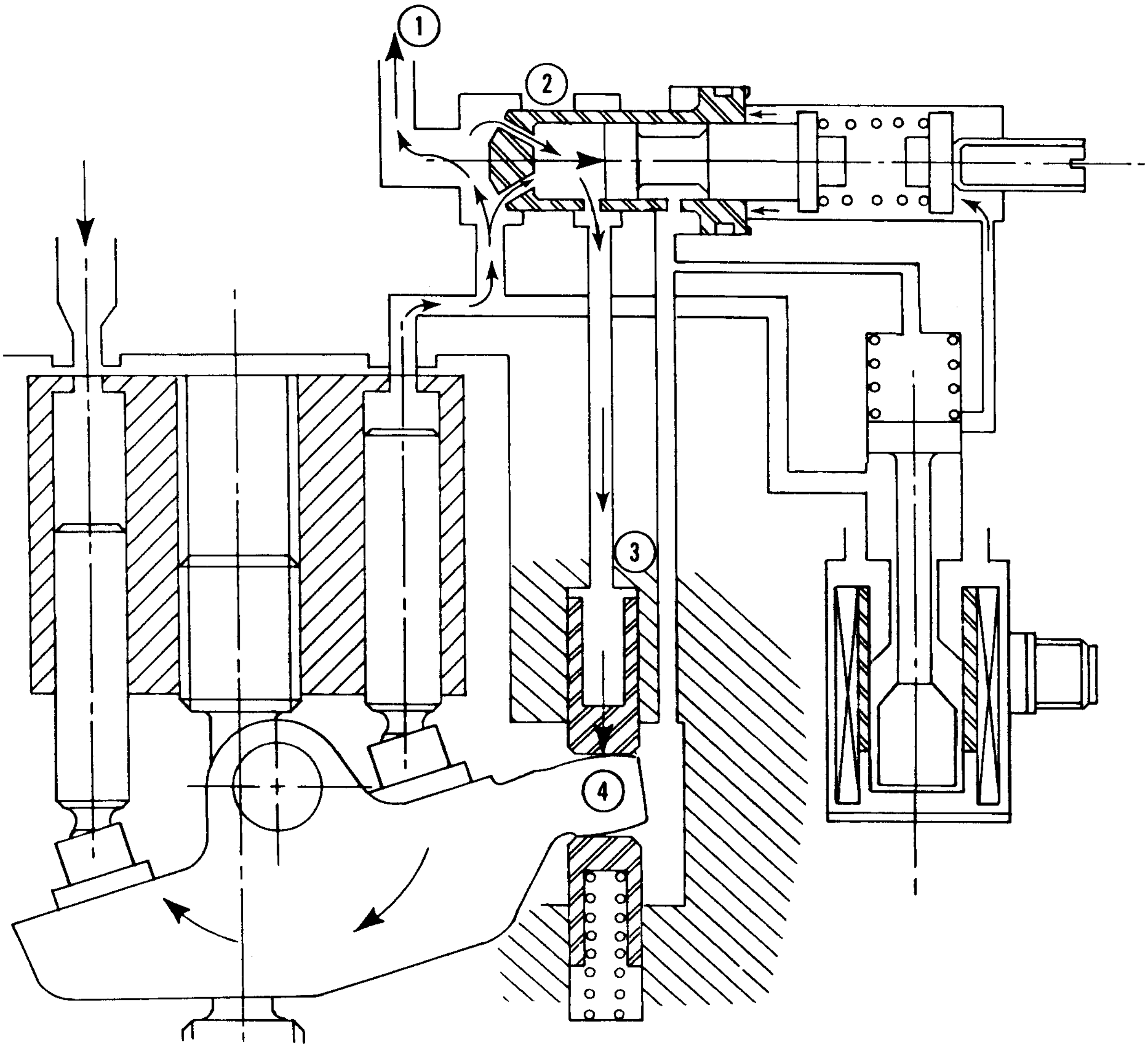


FIGURE 10-4. PUMP COMPENSATOR AND DEPRESSURIZING SYSTEM (NORMAL OPERATION)



1. A small outlet pressure increase results from decreased flow demand.

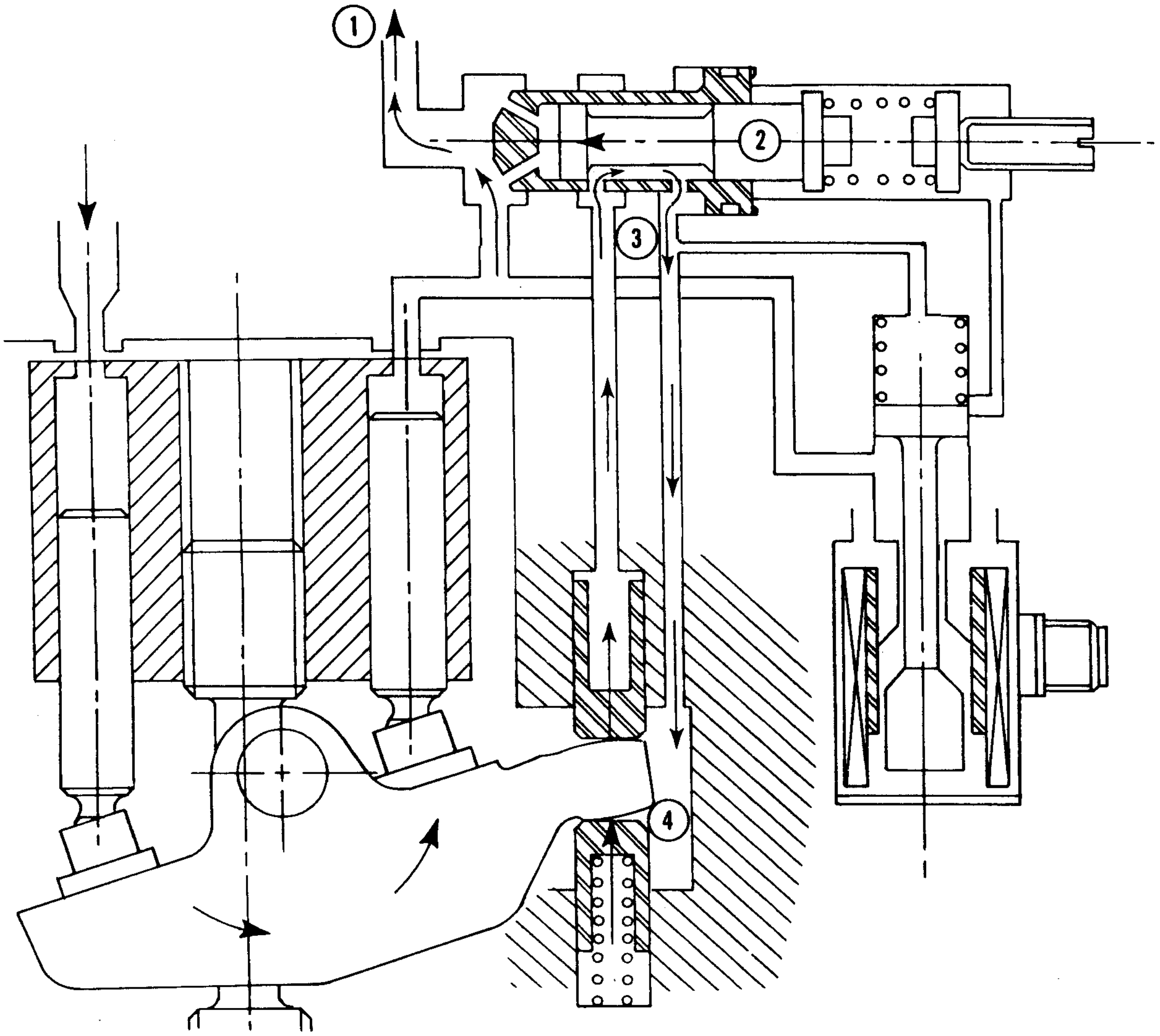
2. Sensing piston detects a pressure increase and moves in direction shown.

3. The spool valve permits pump outlet pressure to enter stroking piston cavity increasing the stroking piston force.

4. Stroking piston rotates hanger and decreases cam angle which results in decreased pump capacity.

5. On reaching desired flow rate and outlet pressure, spool valve returns to no flow position and pump maintains the desired setting.

FIGURE 10-5. COMPENSATOR OPERATION DECREASING FLOW DEMAND



1. A small outlet pressure decay results from increased flow demand.

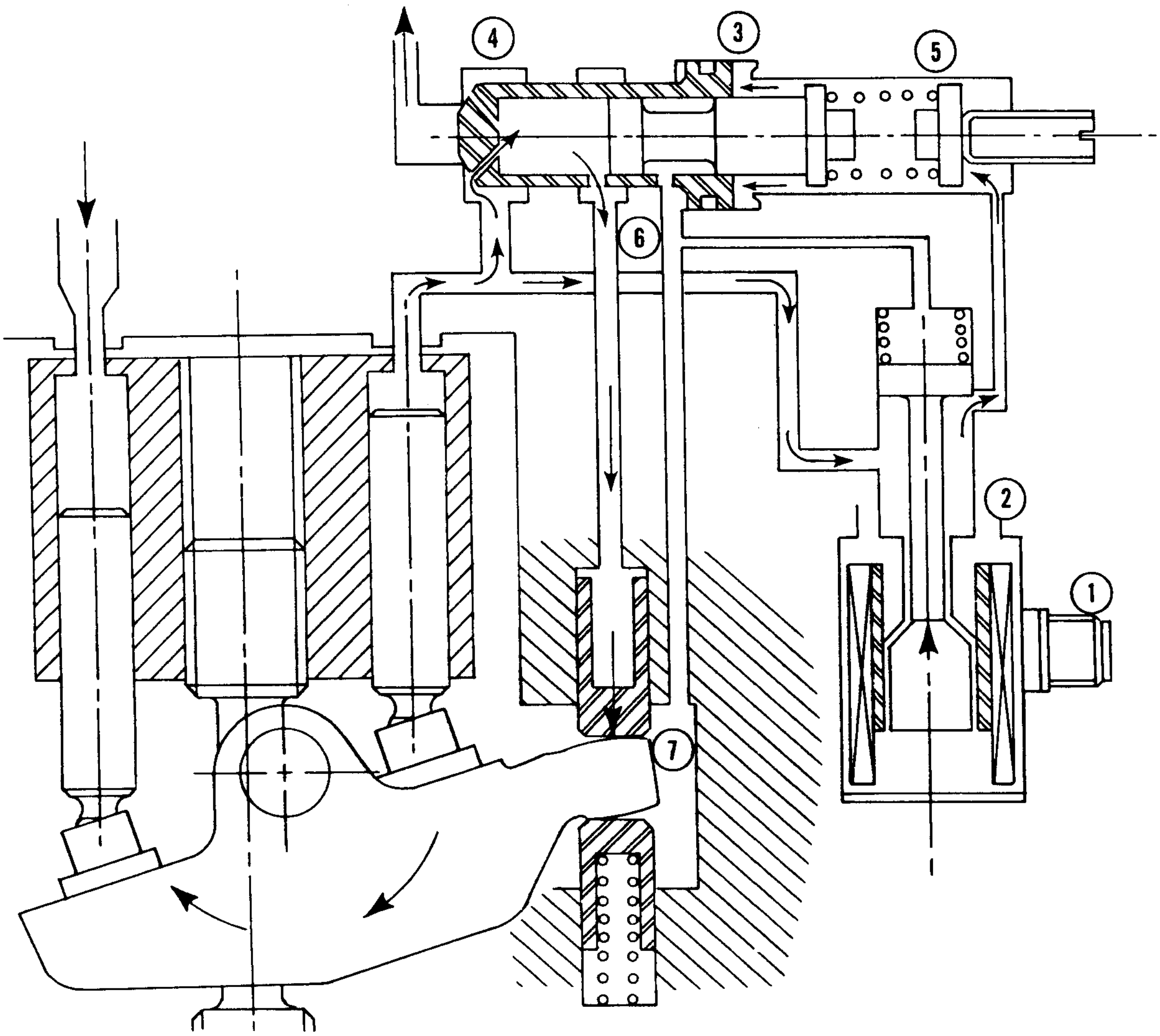
2. The sensing piston detects the pressure decay and moves in direction shown.

3. Fluid flows from the stroking piston cavity through the spool valve. This causes a decreasing stroking piston force.

4. Return spring and pump reaction forces move hanger and increase cam angle which increases pump displacement.

5. On reaching desired flow rate and outlet pressure, spool valve returns to a no flow position and the pump maintains the desired setting.

FIGURE 10-6. COMPENSATOR OPERATION INCREASING FLOW DEMAND



(1.) Energizing the Solenoid displaces the depressurizing valve in the direction shown.

(2.) The depressurizer valve permits pump discharge pressure to flow to the back of the blocking valve (3) and close the blocking valve (4).

(5.) The movement of the blocking valve reduces the pressure reference spring force resulting in a change in compensating pressure.

(6.) Due to the decreased spring force, the compensator valve allows pump outlet pressure to decrease the cam angle (7).

(8.) On reading 500 psi pump discharge pressure, the compensator valve returns to normal operation as described in figures 10-5 and 10-6 but regulates at a reduced pressure.

FIGURE 10-7. DEPRESSURIZER OPERATION

10.315 PUMP POTENTIAL PROBLEMS

1. Pressure Spikes Can be due to compensator valve response as well as downstream control valve closing rates.

2. Pump Ripple Can be due to "tuned" downstream pressure line size and configuration. May be relieved by a resonator at pump outlet or revised piping configuration.

3. Surges in Case Drain Line Can blow out shaft seals or crack pump case. Provide dedicated line directly to reservoir.

4. Excessive Pump Wear Will show up as increased flow and temperature in case drain line. Case drain filter ΔP will increase. May shear shaft if pump internal damage is great. Provide thermal probe in case drain line.

5. Pump Cavitation May show up as pitting in pump cylinder bores and noisy pump operation. Provide adequate suction line and port size. Provide adequate reservoir pressurization.

6. Pump Overheating Too small a case drain line or filter. Too small a reservoir or insufficient exposed piping to dissipate heat.

7. Pump Overspeed Series-wound electric motor-driven pump will overspeed if pump becomes "air bound."

8. Drive Shaft Migration Pump drive shaft must be restrained to prevent it from migrating into the pump or engine splined cavity.

10.32 RESERVOIR DESIGN An aircraft hydraulic system reservoir is a vessel used for storing an extra supply of hydraulic fluid. If it were not for thermal contraction and expansion of the fluid, leakage, differential volumetric displacement of actuating cylinders, and accumulators that are empty at 0 psi and full at system operating pressure, a reservoir would not be needed. Since thermal changes and leakage are inherent, even the simplest system must have a reservoir. Usually it is convenient to incorporate other functions into the reservoir. For instance by pressurizing it, a system can be protected from pulling a vacuum or air into the fluid past the packings; or on more sophisticated systems, inlet pump supply can be maintained at optimum pressure to prevent pump cavitation. A reservoir is a sump for collecting foreign material and a drain valve should be incorporated. A return line filter may be housed in the reservoir. The fluid may be used as a heat sink and a temperature probe installed.

Primarily there are two types of main hydraulic system reservoirs being used in current aircraft. The more recently developed concept is referred to by the military as a "separated type" reservoir and is defined in MIL-R-8931 as "a hydraulic reservoir wherein pressure is generated in the fluid by means whereby no fluid surface is in contact with gas." Analyzing that statement, it means the fluid is separated from and not allowed to come into contact with air and the separator in some way acts to pressurize the enclosed fluid. This type reservoir also has been called "airless bootstrap" and "piston type." Figure 10-8 shows the separated type.

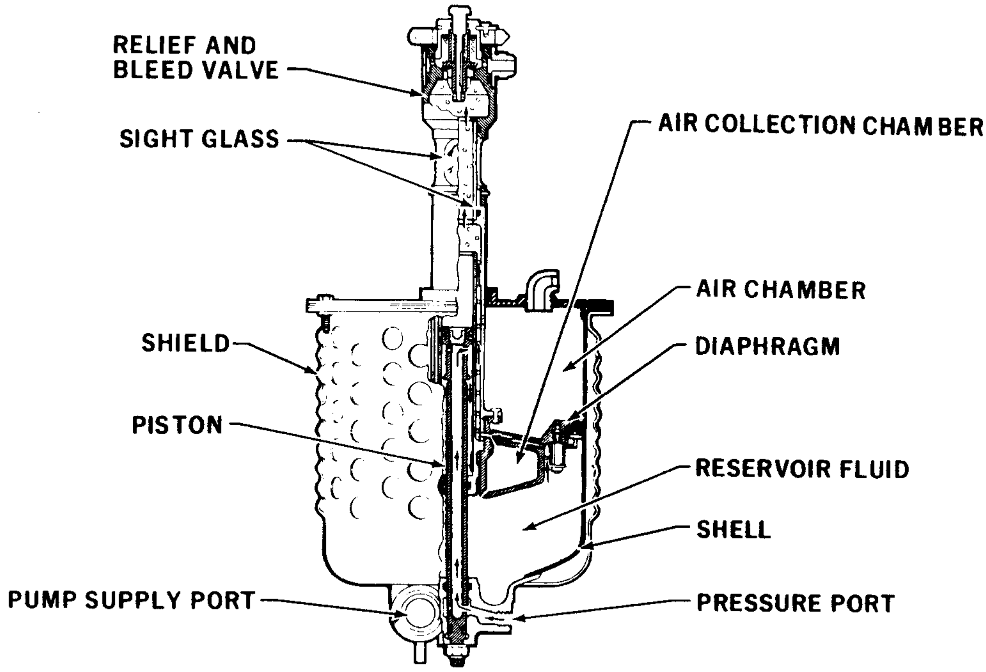


FIGURE 10-8 HYDRAULIC SYSTEM RESERVOIR

An airless bootstrap type reservoir is used in each system to store and supply fluid. During the initial filling and functioning phase, as much of the free air as possible is bled out of the system. System fluid pressure acting on a small piston inside the reservoir applies sufficient force on the large internal piston (100 to 1 mechanical advantage) to pressurize the hydraulic pump inlets to prevent cavitation. Since air is no longer used as the pressurizing media and is deliberately excluded from the system, any potential problems usually associated with foaming, vaporization, contamination or corrosion are either minimized or entirely eliminated. The deletion of air pressure regulators, aspirators, filter, etc., further improves the serviceability and reliability picture.

Reservoir seals include a spring-loaded chevron on the large piston to minimize friction and O-rings with teflon backups on the small piston. Any leakage out of the high-pressure chamber due to dynamic seals only discharges into the low-pressure fluid chamber with no external loss of fluid.

The DC-8 hydraulic system reservoir is a conventional design, i.e., a welded tank partially filled with fluid and pressurized by air pressure in contact with the fluid. The military nomenclature is "nonseparated type" which has merit since it clearly contrasts the two designs currently being used and removes all doubts as to which design is being discussed. A nonseparated reservoir can be made in any shape or size which is advantageous for installation purposes. It is relatively easy to manufacture, minor small deflections do not affect its operation. However, tests must be conducted to develop baffles to eliminate foaming of the returning fluid. Vibrational testing is required on military aircraft and welded fittings are inherently troublesome. Additional components must be installed to accomplish reservoir pressurization.

One of the most difficult aspects of reservoir design is sizing the unit to fit the system. Since hydraulic fluid weighs about 9 pounds/gallon for Skydrol and 7 pounds/gallon for MIL oil, no more fluid than required should be stored in the reservoir. On the other hand, filling the reservoir is a time-consuming maintenance item and the airlines prefer that the interval between required fillings be as long as possible. Two military specifications are available that may be used as a guide for sizing a system reservoir:

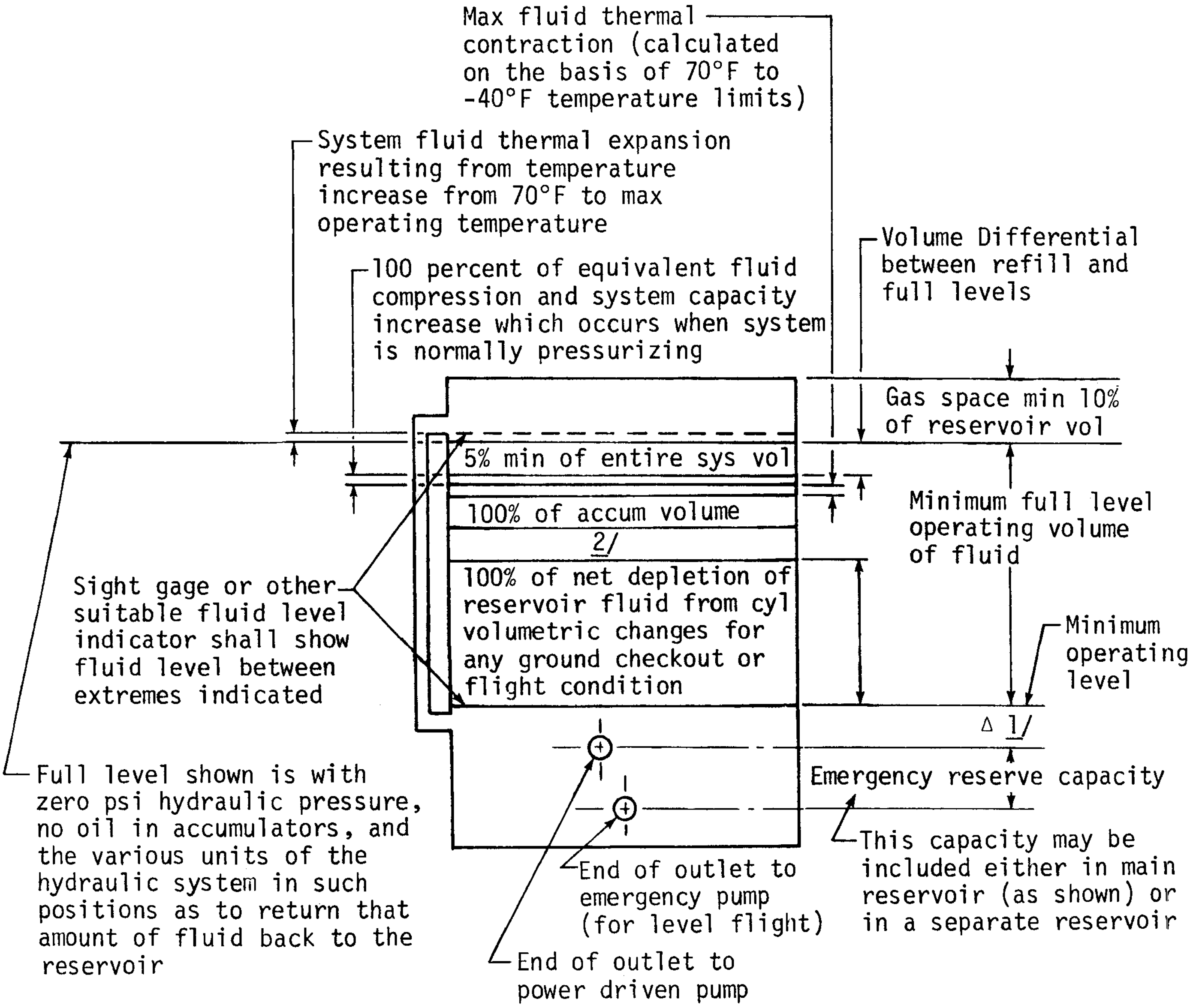
MIL-R-5520C	Reservoirs:	Aircraft, hydraulic, nonseparated type
MIL-R-8931	Reservoirs:	Aircraft and missile hydraulic, separated type

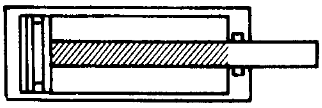
Fortunately these specifications are consistent in stating reservoir capacity requirements and either could be used with a little imagination to design any type of reservoir. Figure 10-9 (which is Figure 1 in MIL-R-5520C) is actually the easiest guide to follow as the fluid requirements are graphically displayed and described on the drawing presented.

At the reservoir outlet that supplies the pump, the design minimum volume of fluid shall be equal to the following:

1. A sufficient amount of fluid to ensure that the hydraulic pump inlet pressurization and satisfactory circulation is maintained for all attitudes and accelerations within the design limitations for which the reservoir is required to function. A nonseparated reservoir is affected by aircraft attitudes and accelerations and usually some additional fluid must be stored to ensure that the outlet port is always covered with fluid, during aircraft maneuvers. Special check valves and baffeling is sometimes required. The separated type reservoir automatically solves this problem; however, the reservoir piston must not be allowed to pass over the outlet port. This can be accomplished by putting the outlet port in the bottom of the reservoir.

2. A fluid volume equivalent to 100 percent of the possible net depletion caused by cylinder volumetric changes which can occur in any ground checkout or flight condition of a recirculating hydraulic system. Reference shall be from the normal reservoir filling attitude and position of the various hydraulic subsystems. The sketch on Figure 10-9 shows the cylinder volume that returns to the reservoir when a cylinder is retracted. It is the rod area times the stroke. The reservoir is to be





Example of volumetric change in an actuating cylinder after acting through the half cycle of operation which returns the most fluid to the reservoir. Exchange volume equals volume of that part of piston rod indicated by shading.

1/ This distance shall be sufficient for both the ground servicing and critical flight conditions (with their corresponding minimum operating levels) to ensure that hydraulic pump suction is maintained.

2/ A volume equivalent to 130 percent of the volumetric capacity of the largest quantity measuring type hydraulic fuse in the system.

Δ These distances may vary between ground and flight conditions.

FIGURE 10-9 RESERVOIR CAPACITY REQUIREMENTS designed with all subsystems in a position that returns the greatest amount of fluid to the reservoir. For instance a landing gear retraction subsystem will either be up or down depending on which position returns the greatest amount of fluid to the reservoir; note that all the cylinders may or may not be in the retracted position depending on the design of the subsystem.

3. A fluid volume equivalent to 100 percent of the reservoir fluid volumetric change caused by charging all accumulators to system operating pressure from a completely discharged position, with no gas precharge in the largest single accumulator, and a minimum design gas precharge in all others in the system.

4. A fluid volume equivalent to 130 percent of the volumetric capacity of the largest quantity-measuring type of hydraulic fuse in the system.

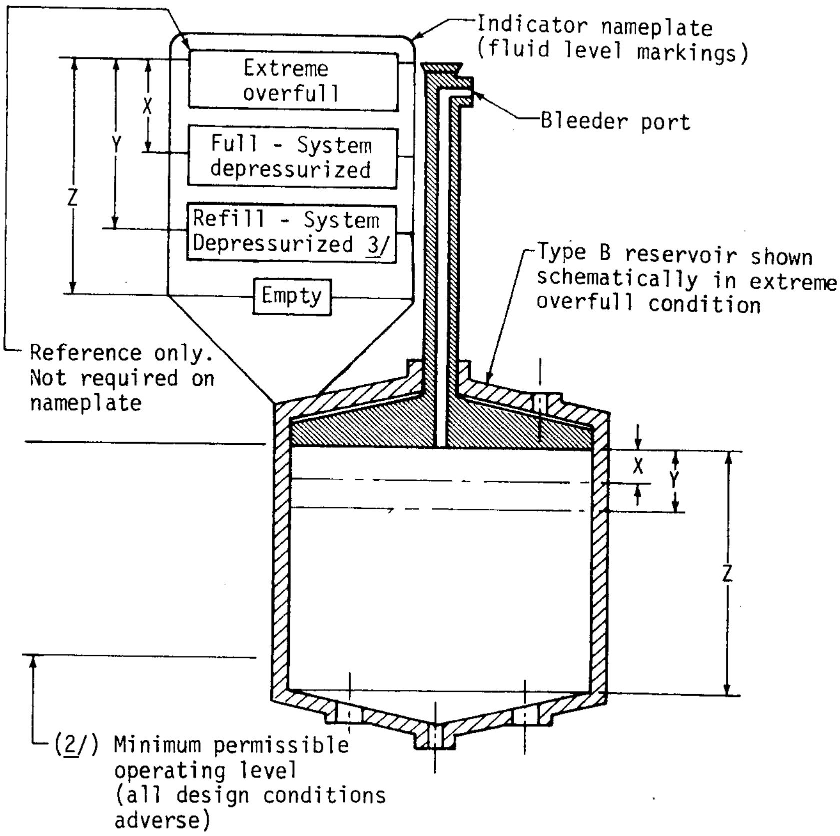
5. A fluid volume equivalent to the maximum thermal contraction which is expected to occur when the entire fluid content of a recirculating system is exposed to a temperature decrease from 70°F down to -40°F.

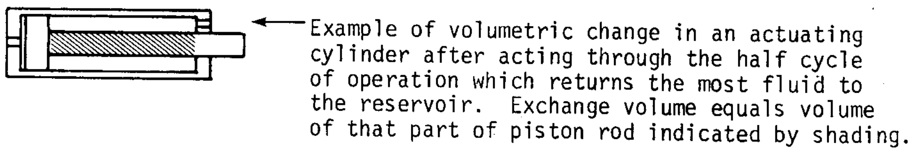
The coefficient of thermal expansion of hydraulic fluid is 0.00045 per °F at 68°F for Skydrol 500A and MIL-H-5606. For reservoir design it is adequate and convenient to use 5-percent increase in volume for 100°F rise in temperature. Since 70°F down to -40°F is 110°F change, use 0.055 times total system volume which must include the fluid in the reservoir to calculate the fluid volume required for Paragraph 5 above. The reservoir volume for this calculation must be approximated and adjusted in subsequent iterations until the estimate and the final results agree.

6. A fluid volume equivalent to not less than 5 percent of the entire system fluid volume, including the reservoir, to minimize the normal frequency of filling. The differential volume between full and refill shall correspond to this leakage allowance.

7. A fluid volume equivalent to that resulting from the effects of fluid compression, line and actuator expansion, and external seal deflection when the hydraulic system which the reservoir is serving is pressurized at normal operating pressure.

In a nonseparated reservoir, a gas space must be provided, which together with other design features, will ensure that there will be no loss of fluid during normal operating conditions. The gas space volume should be at least 10 percent of the reservoir fluid volume. If the vent line (or the reservoir relief valve in the case of a pressurized reservoir) is not capable of accommodating a sufficiently high rate of flow during the most adverse surge condition obtainable without exceeding the reservoir design pressure, then the gas space must be made large enough to limit the maximum pressure to a value less than the design pressure. The surge conditions to be considered include those resulting from the acceleration of hydraulic actuators by aerodynamic loads or gravity, due to actuation by emergency pneumatic systems or negligent service filling from test stands. The condition of an overheated hydraulic system with its accompanying volumetric increase shall also be considered here. A volume equivalent to system thermal expansion resulting from a temperature increase from 70°F to maximum operating temperature must be provided; this is also a design requirement in the determination of the total volume of a separated reservoir, Figure 10-10 (which is Figure 1 in MIL-R-8931).





1/ Per 3.4.1(a); some or all of this volume may be included in the end cavity containing residual oil as pictured, depending upon the particular design situation.

2/ Actual service minimum operating level might be above this level depending upon the most adverse combination of actuator positions obtainable.

3/ Accumulator oil pressure shall be stated if any accumulators are not automatically and rapidly depressurized when hydraulic pumps are stopped.

FIGURE 10-10 RESERVOIR CAPACITY REQUIREMENTS

Potential reservoir problems are as follows:

1. Bootstrap reservoir design should incorporate sufficient piston force, in a static, no-pressure condition, to facilitate reservoir servicing and bleeding.

2. Reservoir filling through the suction line instead of aircraft filler line bypasses the aircraft fill filter. Somehow provide filtration for adding fluid.

3. Reservoirs should have the air-bleed vent high and the suction outlet low. The overboard relief flow capacity should be sufficient to prevent reservoir damage during improper emergency operations, during system operation with an overfilled reservoir, or during reservoir servicing if overfilled.

4. Avoid connecting reservoir vent lines to a common overboard vent, where back pressure can cause backflow into the reservoir from the second vent system.

5. In bootstrap reservoirs, the pressurizing section should have minimum restriction. This allows reverse oil flow into the pressure side of the hydraulic system during reservoir piston motions caused by return oil flow to the reservoir. Reservoir failures have been caused by over-pressurization of pressure chambers of the reservoir.

6. With hydraulic power in one system only and with high rates of motion in large tandem actuators, fluid is pumped from the unused section back to the return system without recovering equal fluid from the pressure side of the unpressurized system. Unless the returned fluid can be dumped at low pressure, the reservoir and other low-pressure components can be damaged. As an alternate to dumping, equip ground test carts with multiple connections to simultaneously pressurize both systems. Overfilling reservoir can make this problem critical.

10.33 ACCUMULATORS

10.331 FUNCTIONS The two primary functions of an accumulator in an aircraft hydraulic system are:

1. Pressure Storage Potential energy is stored in the gas chamber of the accumulator and is used as an auxiliary source of power to supplement the output of the pump during peak demands beyond the capacity of the pump or when the pump is inoperable.

2. Pulsation and Shock Absorption Pulses generated by the pump and sudden pressure rises caused by instantaneous closing of a valve are dampened by the accumulator to minimize the detrimental effect on system components.

10.332 TYPES The two most common types of accumulators used in aircraft hydraulic systems are shown in Figures 10-11 and 10-12. The spherical diaphragm (bladder) type accumulator is used by DAC on commercial aircraft because of its faster response. The cylindrical type (MS 28700) is more frequently used on military aircraft.

10.333 ACCUMULATOR SIZING Accumulators operate on the principle of Boyle's law of gases

$$P_1V_1^n = P_2V_2^n = P_3V_3^n \qquad (1)$$

where

P_1 = Accumulator gas precharge (psi). This pressure must be $\leq P_3$. Usually $P_{2/3}$.

V_1 = Required accumulator size (in.^{3}). This is the maximum volume of gas at precharge pressure.

P_2 = Maximum system design operating pressure (psi).

V_2 = Volume of gas at maximum operature pressure (in.^{3}).

P_3 = Minimum system pressure (psi). The minimum pressure at which the system can operate.

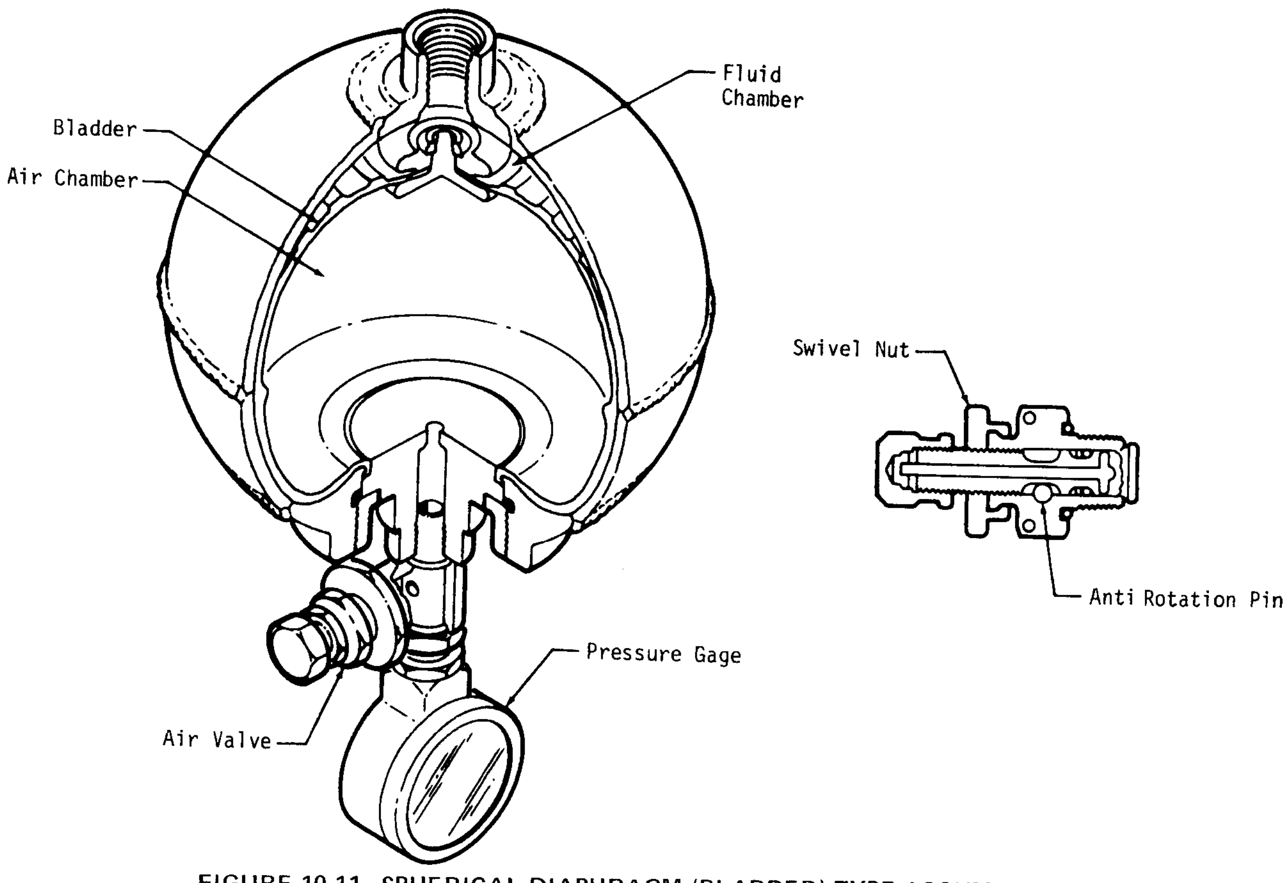


FIGURE 10-11. SPHERICAL DIAPHRAGM (BLADDER) TYPE ACCUMULATOR

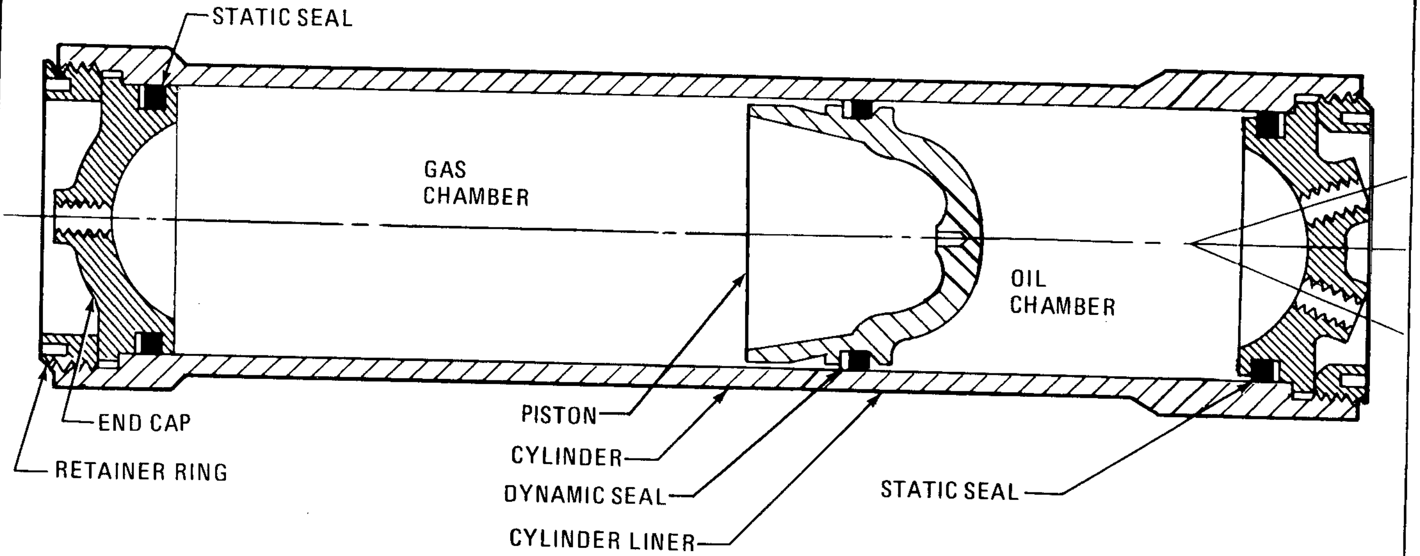
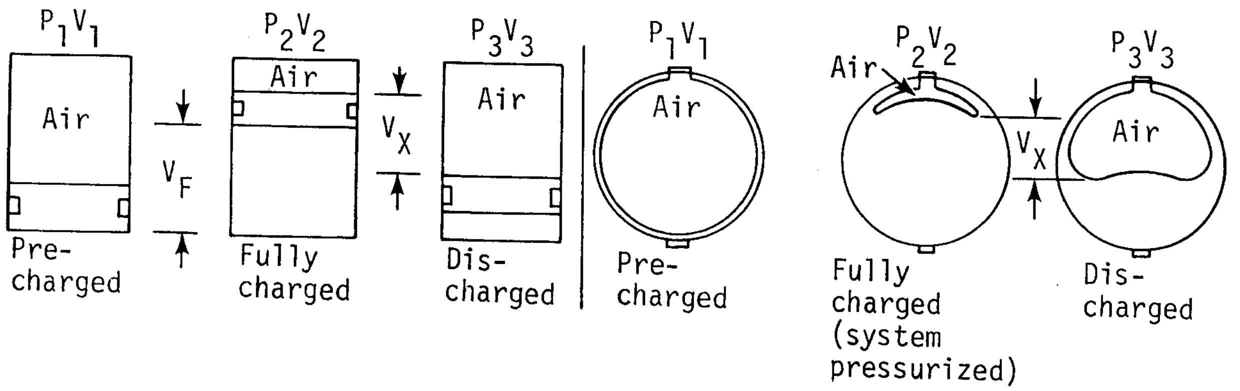


FIGURE 10-12. CYLINDRICAL PISTON TYPE ACCUMULATOR

V_3 = Volume of gas at minimum operating pressure (in.^{3}).

n = 1 (isothermal expansion)
1.4* (adiabatic expansion)
1.25 (polytropic expansion)

* - Usually used in aircraft since most applications require actuations in less than one minute.



The amount of fluid required from the accumulator to meet the needs of the system can be symbolized by V_x

where

$$V_x = V_3 - V_2 \qquad (2)$$

Solving for V_1,

<u>Isothermal Condition</u>:

$$V_3 = V_x + V_2$$

$$P_2V_2 = P_3V_3 \text{ (from Boyle's Law)}$$

$$P_2V_2 = P_3(V_x + V_2)$$

$$P_2V_2 = P_3V_x + P_3V_2$$

$$V_2(P_2 - P_3) = P_3V_x$$

$$V_2 = \frac{P_3V_x}{P_2 - P_3}$$

$$P_1V_1 = P_2V_2 \text{ (from Boyle's Law)}$$

$$V_1 = \frac{P_2V_2}{P_1}$$

$$V_1 = \frac{P_2}{P_1}\,\frac{P_3V_x}{P_2 - P_3}$$

$$V_1 = \frac{V_x(P_3/P_1)}{1 - (P_3/P_2)}$$

<u>Adiabatic Condition</u>:

$$P_1V_1^n = P_2V_2^n = P_3V_3^n$$

or $$V_1(P_1)^{1/n} = V_2(P_2)^{1/n} = V_3(P_3)^{1/n}$$

$$V_x = V_3 - V_2$$

$$V_3 = V_x + V_2$$

$$V_2(P_2)^{1/n} = V_3(P_3)^{1/n}$$

$$V_2(P_2)^{1/n} = (P_3)^{1/n}(V_x + V_2)$$

$$V_2(P_2)^{1/n} = (P_3)^{1/n}V_x + (P_3)^{1/n}V_2$$

$$V_2[(P_2)^{1/n} - (P_3)^{1/n}] = V_x(P_3)^{1/n}$$

$$V_2 = \frac{(V_x)(P_3)^{1/n}}{(P_2)^{1/n} - (P_3)^{1/n}}$$

$$V_1(P_1)^{1/n} = V_2(P_2)^{1/n}$$

$$V_1 = (P_2/P_1)^{1/n}(V_2)$$

$$V_1 = \frac{(P_2/P_1)^{1/n}(P_3)^{1/n}(V_x)}{(P_2)^{1/n} - (P_3)^{1/n}}$$

$$V_1 = \frac{V_x(P_3/P_1)^{1/n}}{1 - (P_3/P_2)^{1/n}}$$

Example:

Problem

What size of accumulator is necessary to supply 30 cubic inches of fluid in a hydraulic system of maximum operating pressure 3000 psia, which drops to minimum 1500 psia? Assuming nitrogen gas precharge of accumulator is 1000 psia.

V_1 = ? (size of accumulator) cubic inches

P_1 = 1000 psia

P_2 = 3000 psia

P_3 = 1500 psia

V_x = 30 cubic inches

Isothermal Solution:

$$V_1 = \frac{V_x(P_3/P_1)}{1 - (P_3/P_2)}$$

$$V_1 = \frac{30(1500/1000)}{1 - (1500/3000)}$$

$$V_1 = (30)(3/2)(2)$$

V_1 = 90 cubic inches

V_1 = 90/231 = 0.39 gallons

Necessary Accumulator Size:
90 cubic inches

Adiabatic Solution:

$$V_1 = \frac{V_x(P_3/P_1)^{1/n}}{1 - (P_3/P_2)^{1/n}}$$

$1/n = 1/1.4 = 0.714$ for N_2

$$V_1 = \frac{30(1500/1000)^{0.714}}{1 - (1500/3000)^{0.714}}$$

$$V_1 = (30)(3.69)$$

V_1 = 110.5 cubic inches

V_1 = 110.5/231 = 0.478 gallons

Necessary Accumulator Size:
111 cubic inches

10.334 ACCUMULATOR FLUID VOLUME CURVES The following curves are useful in determining the fluid volume in a given size accumulator at various pressures.

V_F = Fluid volume, cubic inches

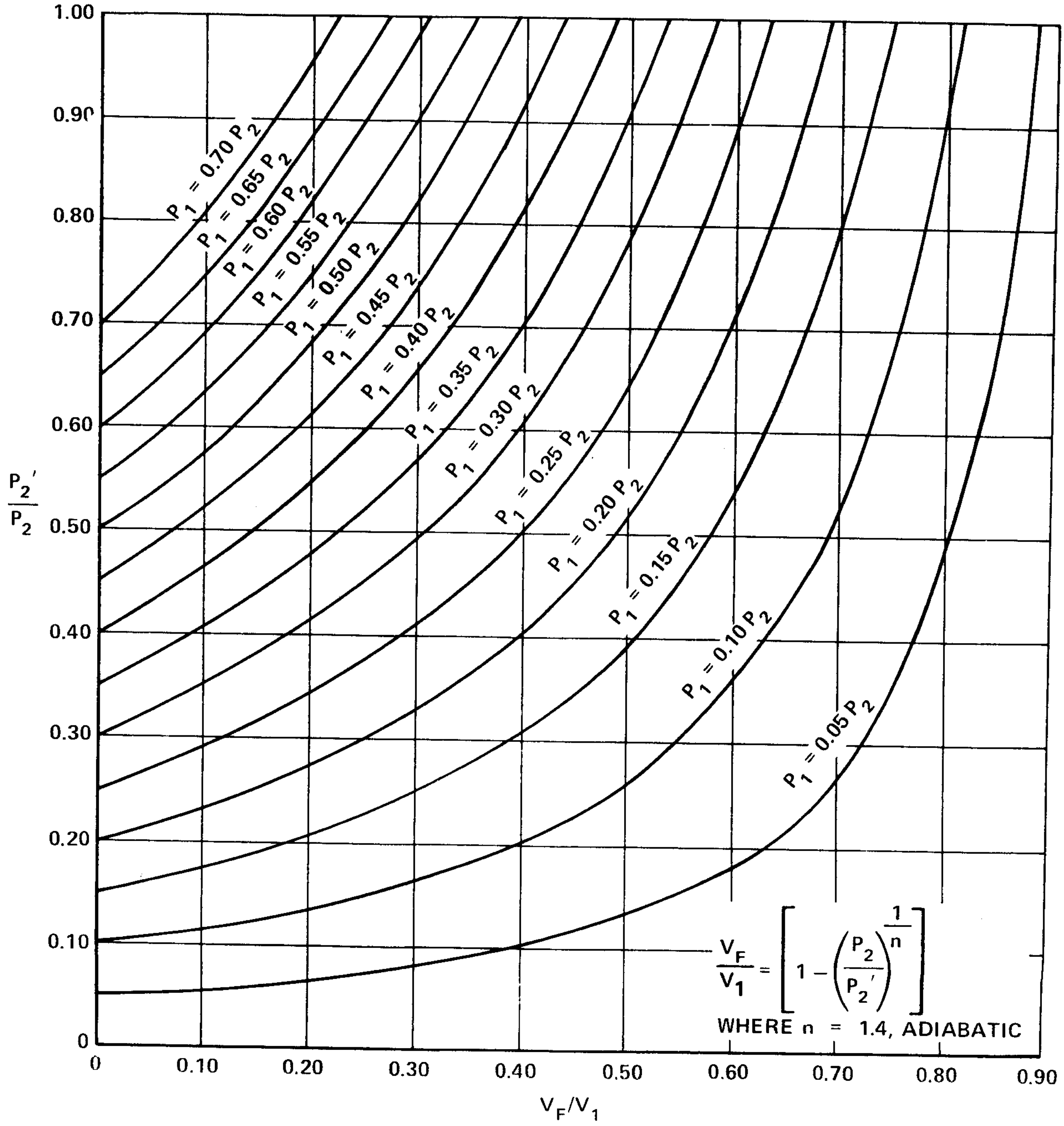
P_2 = Max system operating press., psi

P_2' = Air and oil pressure at V_F, psi

P_1 = Air precharge pressure, psi

V_1 = Volume of air at P_1, cubic inches

V_2 = Volume of air at P_2, cubic inches



10.335 ACCUMULATOR POTENTIAL PROBLEMS

1. Rapid discharging of accumulators into return lines through quick-opening valves can burst reservoirs, pump housing, or other return line or suction line components. Slow valves or restrictors will avoid the problem.

2. Use cushioned clamps on hydraulic cylindrical accumulators if it is necessary to provide a clamping arrangement in the cylindrical area where clamp-induced distortion could cause the piston to bind.

3. Adequate space should be provided to allow use of standard gas charging equipment for charging accumulators without bracket or piping interference.

4. Install accumulator with fluid port up so that air may be bled from the system easily.

5. Heat generated by compression of air in accumulator can act as a catalyst to cause the oxygen to combine with hydraulic fluid or seal materials with explosive force. Always use an inert gas such as nitrogen (N_2) to inflate accumulators.

10.34 FILTERS

10.341 INTRODUCTION Contamination constitutes one of the major causes of hydraulic system component malfunction. Filters are placed in the system to control contamination by trapping particles entrained in the fluid.

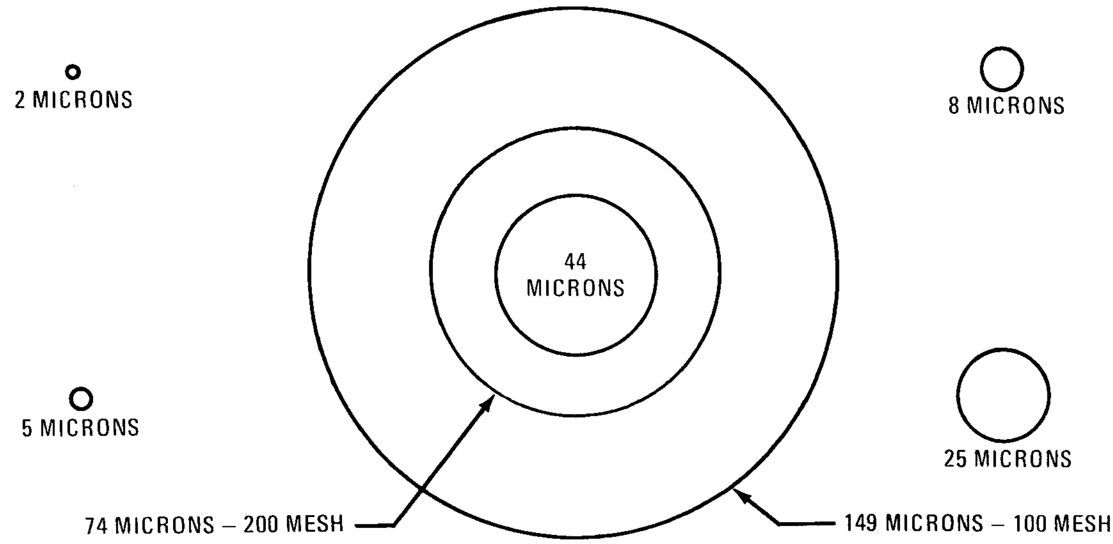
Aircraft hydraulic systems usually contain orifices, servo valves, metering devices, and other parts which have extremely small clearances and openings. As a result, these systems are extremely sensitive to contamination. A typical electrohydraulic servo valve may have a nozzle diameter of 0.032 inch, clearances between nozzle and flapper of 0.00059 inch, and orifices of 0.005 inch. A system containing components with clearances of this size may require filtration to remove accumulations of large numbers of particles smaller than the maximum permissible particle. Removal of these smaller particles (silt) may be required to reduce valve sticking and wear of moving parts.

10.342 FILTER TERMINOLOGY

1. Particle Size The largest dimension of a solid particle of contaminant and quoted in microns as 1 micron, μ, = 10^{-6} meters = 3.94×10^{-5} inches. (See Figure 10-13.)

2. Fiber Size A solid contaminant having a length-to-diameter ratio of 10 to 1 or greater.

3. Filter Rating (Absolute) The micron size corresponding to removal of 100 percent of all hard, spherical particles (i.e., glass beads) larger than a given size.



LINEAR EQUIVALENTS

1 INCH	25.4 MILLIMETERS	25,400 MICRONS
1 MILLIMETER	0.0394 INCH	1,000 MICRONS
1 MICRON	1/25,400 OF AN INCH	0.001 MILLIMETERS
1 MICRON	3.94×10^{-5}	0.000039 INCH

RELATIVE SIZES

LOWER LIMIT OF VISIBILITY (NAKED EYE)	40 MICRONS
WHITE BLOOD CELLS	25 MICRONS
RED BLOOD CELLS	8 MICRONS
BACTERIA (COCCI)	2 MICRONS

SCREEN SIZES

MESHES PER LINEAR INCH	U.S. SIEVE NO.	OPENING IN INCHES	OPENING IN MICRONS
52.36	50	0.0117	297
72.45	70	0.0083	210
101.01	100	0.0059	149
142.86	140	0.0041	105
200.00	200	0.0029	74
270.26	270	0.0021	53
323.00	325	0.0017	44

FIGURE 10-13. RELATIVE SIZE OF PARTICLES – MAGNIFICATION: 500 TIMES

4. Filter Rating (Nominal) The micron size corresponding to removal of 98 percent of all incident particles larger than a given size. "Nominal" is an obsolete, seldom used term.

5. Filter Efficiency The number of particles trapped divided by the number of particles introduced into the filter.

6. Pressure Drop (Clean) The pressure differential across the clean filter and housing under specified conditions of flow rate, temperature, pressure, and fluid medium.

7. Contaminant Capacity (Dirt Holding Capacity) The maximum weight of a contaminant with specified particulate distribution which can be added on the inlet side of a filter without exceeding a specified pressure drop.

8. Media Migration Migration of contaminant originating in the filter material or supporting structure.

9. Collapse Pressure The differential pressure across a filter which will collapse or distort the element in any manner in which the performance of the filter is degraded.

10. Initial Bubble Point The air pressure, in inches of water, at which the first bubble appears in a liquid of known surface tension and temperature in which the element is wetted and pressurized with air. The bubble point is an indication of the maximum pore size of a filter medium. The filter element is immersed in the liquid to a known depth, and air pressure inside the element is gradually increased until the first bubble appears. The bubble point test procedure is presented in detail in MIL-F-8815.

11. Filter Media The material or combination of materials which are used to remove solids from a fluid stream.

10.343 FILTRATION TECHNIQUES

1. Surface Filtration Surface filtration is accomplished by impingement and retention of solid contaminants on a matrix of pores or openings in a single plane or surface. Filtration occurs only at the one surface, and contaminants which are not stopped at this surface pass through the media with no further change in direction. Surface filtration is effective in collection of particles larger than the pore size, but ineffective in collection of fibers and particles smaller than the pore size. Contaminant capacity of surface filters is limited by the amount of surface area which can be provided within a given envelope. Single layer mesh screens are typical surface filtration media.

2. Depth Filtration Depth filtration is accomplished by impingement and retention of solid contaminants in a matrix of pores in series or depth. Sand is a classic example of a depth filter, its filtering action being random absorption an entrapment. Filtration occurs not only at the surface but throughout the thickness of the media. Particles which pass one matrix are subjected to direction changes in a circuitous path. Depth filters are effective in collection of particles larger than the maximum openings and fibers, and will also collect a portion of contaminants smaller than the largest pore size, depending on the media type and thickness. Contaminant capacity of depth type media per square inch of surface area is large because the contaminants may be retained throughout the depth of the material as well as on the surface layer. Examples of depth filtration media are: multiple layers of mesh, wound wire cylinders, stacked paper discs, sintered granulated materials, multiple layers of cloth, compressed or matted organic or inorganic fibers, stacked etched sheet metal discs, open pore plastic foam materials, and stacked membranes.

10.344 FILTER SELECTION The selection of filters for a hydraulic system should include the following steps:

1. Determine the maximum particle size that the critical system can tolerate.

2. Review the system components to determine which items will generate contaminants.

3. Provide filtration at the location or locations in each system which will protect the critical components from self-generated and outside source contamination.

4. Specify a filter or system of filters which will satisfy the system requirements. Particular emphasis should be placed on initial cleanliness of the filter.

10.345 DOUGLAS DESIGN PHILOSOPHY

10.3451 Location and Type Main systems filters are located in the pressure and case drain line downstream of each pump and in the main return line upstream of each system reservoir. In addition to the main system filters, which are depth type, large capacity, and designed to remove all particles over a size such as 15 microns, smaller screen type filter elements are used to protect components such as small orifices (to prevent plugging) and control valves (to prevent jamming). These screens are sized to have openings larger than the main filters, typically 25 to 40 microns. Virtually all contaminants are removed by the main (replaceable) filter elements, while the coarser screens normally encounter few particles larger than their pore size, and are not expected to clog during the time between overhauls of the components in which they are installed.

10.3452 Sizing After system filtration level has been determined, filters are sized for maximum flow conditions, adequate service life, and commonality. The filter housing design, including inlet and outlet ports, and upstream and downstream shutoff devices must not produce excessive pressure drop at rated flow. The filter element must not only pass rated flow when clean, but should be designed to hold enough contaminant to remain in service for an adequate time, typically one year. It has been proven economically advantageous for the airplane operator if all system filters are identical. For the DC-9 and DC-10 this was accomplished, except for customer requested deviations. When uniform sizing is practiced, a low-flow filter such as a pump case drain filter, appears oversize if fluid only is considered. However, in that application, the contaminant flow justifies the increased size.

10.3453 Differential Pressure Indicator The incorporation of a differential pressure indicator (usually a "pop-up" button, but sometimes a pressure switch controlling a light in the cockpit) assures that:

1. A contaminated element will be removed before it becomes completely clogged.

2. The useful life of an element will be obtained before removal.

The choice of pressure rating for the differential device is based on a number of considerations on most aircraft systems, and most locations within those systems, the fluid flow is not steady but varies widely. The indicator must operate when the maximum flow imposed during the flight profile creates a pressure drop selected such that:

1. The element will have a large contaminant capacity.

2. After the indicator is actuated, the system will continue to operate satisfactorily for several more flights, in case immediate replacement of the element is not accomplished.

3. No "side-effects" of higher pressure drop, such as inadvertent unlatching of mechanisms, dragging wheel brakes, excess reduction of net available pressure, are encountered.

In order to avoid premature actuation of the device when the fluid is cold, a thermal lockout device is employed. This device prevents actuation of the ΔP indicator, regardless of pressure drop, until the fluid warms up to the "flat part" of the viscosity curve, typically 70 to 100 degrees F.

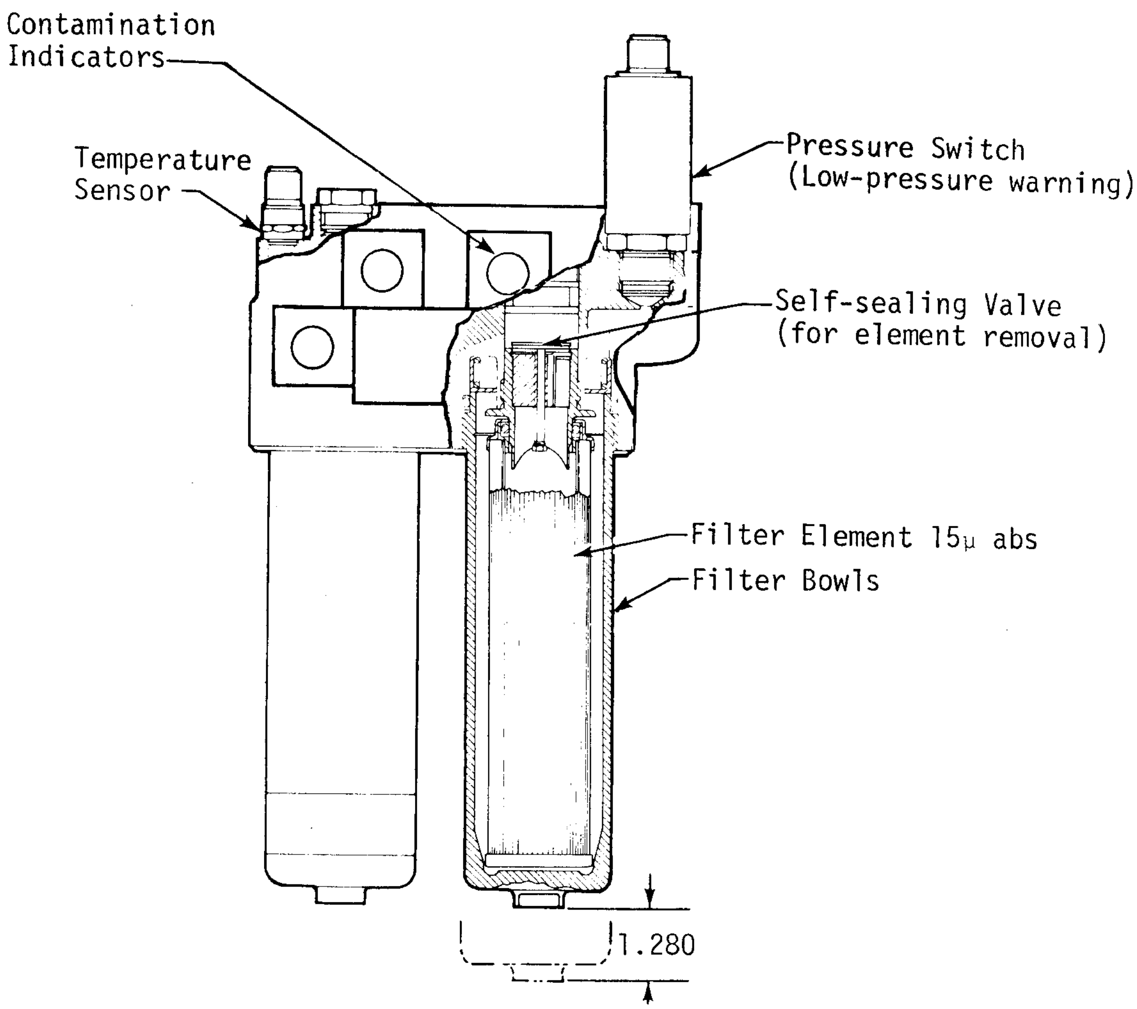
10.3454 By-pass Relief Valves For the main system filters, it is standard practice to use depth-type media, and unless the operating temperature is excessive, an impregnated fiber has been found most satisfactory. Using MIL-F-8815 as a guide, the collapse pressure for a 3000 psi system is specified as 4500 psi. This mechanical strength, combined with the use of redundant systems and differential pressure indicators, has made practical the elimination of by-pass relief valves.

Filters shall be installed in accessible locations and adequate clearance for removal shall be provided in order that the filter elements can be easily replaced when necessary. The differential pressure indicator, if used, shall be so located on the filter assembly that visual detection of a "popped" button can readily be accomplished. In order to reduce weight, provide compactness of design, and reduce the number of connections, it has become standard to integrate the main system filters into a manifold which includes other basic hydraulic system components such as pressure switches, relief valves, temperature indicators, etc. In this case, the manifold is machined to form the filter case.

10.346 SPECIFICATION The specification control drawing (SCD) for a filter should include, but not necessarily be limited to the following:

1. Configuration, size, shape, and weight limitations.

2. Operating fluid.



ENGINE FILTER MANIFOLD

3. Materials for the seals and filter element.
4. Port size and type.
5. Temperature range.
6. Vibration environment.
7. Pressures - operating, proof, and burst.
8. Pressure drop at rated flow (clean).
9. Rated contaminant capacity and press drop.
10. Filter rating.
11. Leakage, internal, and external
12. Differential pressure indicator requirements.
13. Relief or by-pass requirements.

10.347 FILTER POTENTIAL PROBLEMS

1. To avoid clogging, filters incorporated in valves (such as restrictors), should be larger in micron rating than the system filters. The normally accepted size is 25 microns. Filters should be of finger-type design rather than flat disc-type for higher dirt capacity. They should be adequately retained, not pressed in, to avoid blowout.
2. All restrictors with a hole size under 0.070 inch should have filters. (Direct from Specification MIL-H-5440)
3. Qualification test filtration should not be better than the component will see during use.
4. Filter sizing on hydraulic pump case drain lines was formerly based on rated fluid flow. The filters sometimes clogged early and caused pump failures. Oversized filters are now used based on high rate of contamination discharge from pump case ports.
5. In the case of servo valves, the filter should be capable of removing contaminants larger than the smallest passage in the valve.
6. To minimize inventory and service problems, use the same size filter element in as many places as possible.

10.5 FLUIDS

Most of the current aircraft hydraulic fluids can be divided into two general categories: petroleum-based and fire-resistant fluids. At Douglas, the fire-resistant fluid used is a phosphate-ester fluid which is defined in DMS 2014. The petroleum-based fluid is in accordance with MIL-H-5606. MIL-H-83282 is a more fire-resistant petroleum-based fluid.

10.52 PHOSPHATE-ESTER FLUIDS DMS 2014 lists the phosphate-ester fluids according to the following types:

Type 1 - Standard fire-resistant fluid
Type 2 - Long valve life fire-resistant fluid
Type 3 - Low density, long valve life fire-resistant fluid
Type 4 - Long fluid life, long valve life fire-resistant fluid.

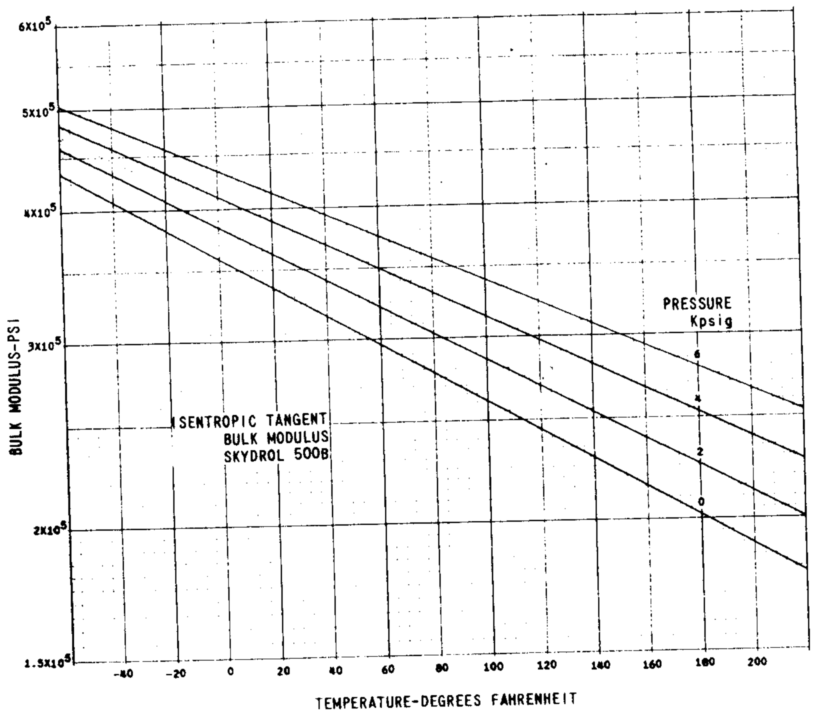
The following is a list of fluids qualified under the requirements of DMS 2014.

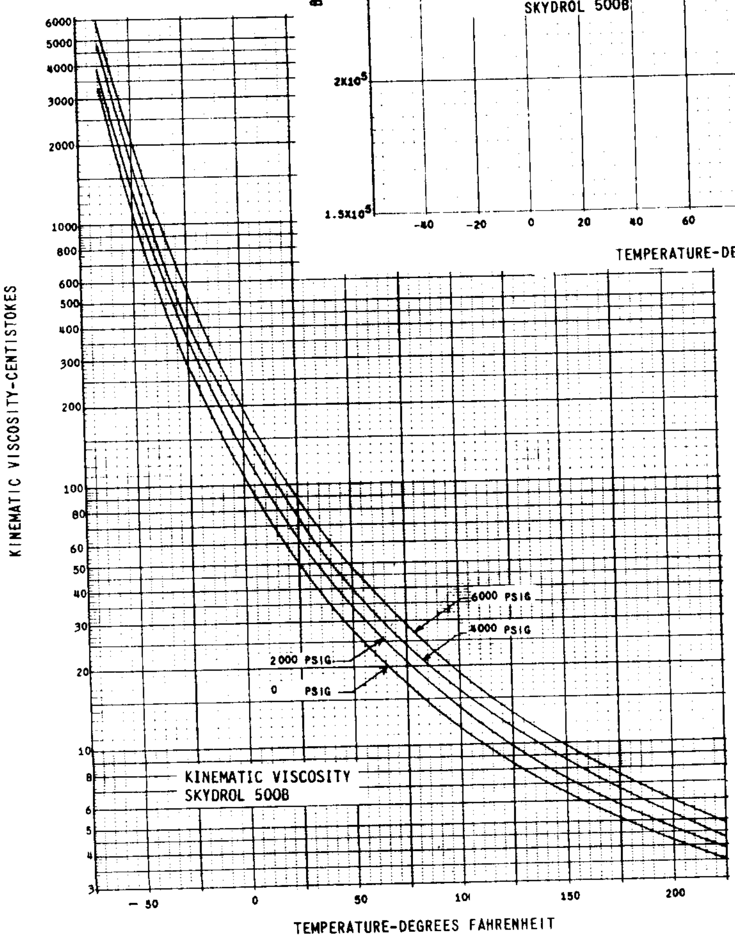
CLASSIFICATION	MANUFACTURER'S DESIGNATION	MANUFACTURER'S NAME AND ADDRESS	REMARKS
TYPE 1 – FLUID	SKYDROL 500-A	MONSANTO CHEM CO., ST. LOUIS, MO	
	HYJET	CHEVRON CHEM CO., SAN FRANCISCO, CA	DOUGLAS REPORT NO. TM-G-FFP-R6407
TYPE 2 – FLUID	SKYDROL 500-B	MONSANTO CHEM CO.	
	HYJET W	CHEVRON CHEM CO.	DOUGLAS REPORT NO. TM-G-FFP-R6407
TYPE 3 – FLUID	SKYDROL-LD	MONSANTO CHEM CO.	DOUGLAS REPORT NO. DAC 59645
	HYJET III	CHEVRON CHEM CO.	DOUGLAS REPORT NO. G2061 10/9/70
TYPE 4 – FLUID	SKYDROL 500-C	MONSANTO CHEM CO.	DOUGLAS REPORT NO. DAC 59645

10.521 PROPERTIES OF PHOSPHATE-ESTER FLUIDS

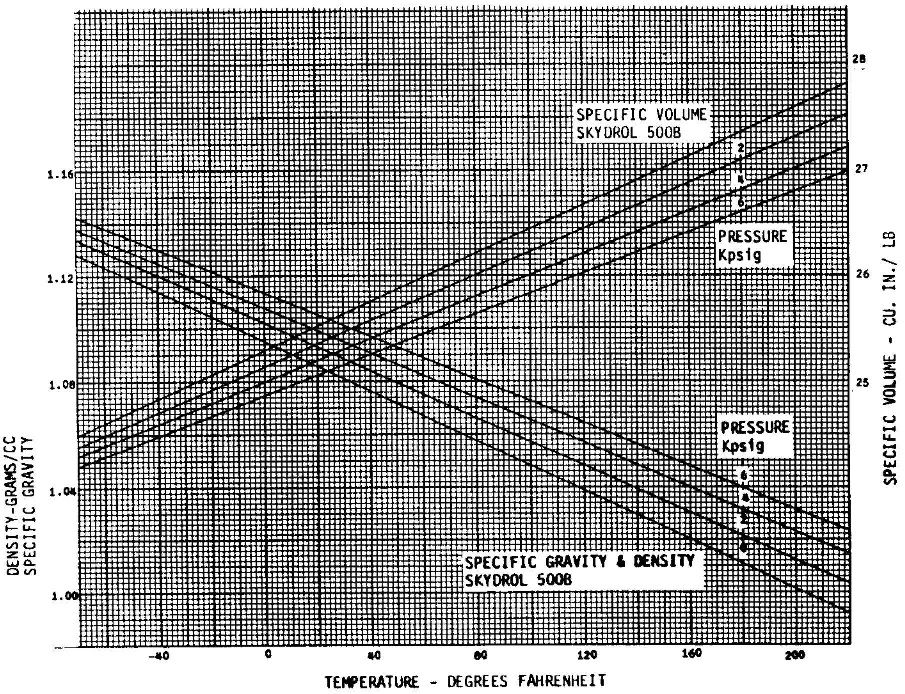
PROPERTY	SKYDROL LD TYPICAL VALUES	SKYDROL 500C TYPICAL VALUES	SKYDROL 500B TYPICAL VALUES	TEST METHODS
VISCOSITY				ASTM D445
−65°F	1130 CS	3600 CS	3500 CS	
−40°F	320 CS	600 CS	600 CS	
100°F	11.21 CS	11.72 CS	11.75 CS	
210°F	3.93 CS	3.77 CS	3.85 CS	
SONIC SHEAR	100°F VISCOSITY / PERCENT SHEAR	100°F VISCOSITY / PERCENT SHEAR	100°F VISCOSITY / PERCENT SHEAR	ASTM VOLUME I OCTOBER 1961 PAGE 1160 APPENDIX XII
TIME: 0 MINUTE	11.21 CS / –	11.78 CS / –	11.76 CS / –	
15 MINUTES	10.83 CS / 3.4%	10.91 CS / 7.4%	10.57 CS / 10.12%	
30 MINUTES	10.40 CS / 7.2%	10.58 CS / 10.2%	9.86 CS / 16.16%	
60 MINUTES	– / –	10.16 CS / 13.8%	8.96 CS / 23.81%	
90 MINUTES	9.80 CS / 12.6%	9.89 CS / 16.0%	8.45 CS / 28.15%	
120 MINUTES	9.69 CS / 13.6%	9.67 CS / 17.9%	8.15 CS / 30.70%	
POUR POINT	<−80°F	<−80°F	<−80°F	ASTM D97
SPECIFIC GRAVITY 25°C/25°C 77°F	0.999	1.066	1.064	MONSANTO 116-B
DENSITY: 100°F	0.9814	1.042	1.052	
150°F	0.9588	1.022	–	
200°F	0.9357	0.998	1.001	
250°F	0.9136	0.974	–	
300°F	0.8900	–	–	
CUBICAL COEFFICIENT OF EXPANSION	4.88×10^{-4}/°F (100 TO 300°F)	4.68×10^{-4}/°F	4.65×10^{-4}/°F	
COEFF OF EXPANSION	4.57×10^{-4}/°F			
WEIGHT PER GALLON 75°F	8.33 LB	8.86 LB	8.87 LB	
MOISTURE	0.50%	0.24%	0.50%	ASTM D1744
BULK MODULUS (ISOTHERMAL SECANT 0 TO 6000 PSI)				ISOTHERMAL SECANT BULK MODULUS BY PVT BLEEDOFF TYPE METHOD
77°F	–	330,000 PSI	340,000 PSI	
100°F	261,000 PSI	–	–	
200°F	207,000 PSI	234,000 PSI	218,000 PSI	
300°F	158,000 PSI	180,000 PSI	168,500 PSI	
400°F	120,000 PSI	135,000 PSI	–	
VAPOR PRESSURE				TRUMP, FOWLER AND VOGLER – JOURNAL OF CHEMICAL AND ENGINEERING DATA 13 209 NO. 2 (1968)
32°F	1.5 MM HG	1.4 MM HG	1.61 MM HG	
72°F	5.6 MM HG	4.8 MM HG	6.80 MM HG	
104°F	14.2 MM HG	10.9 MM HG	–	
167°F	67.0 MM HG	37.7 MM HG	48.60 MM HG	
215°F	180.0 MM HG	54.0 MM HG	160.00 MM HG	
257°F	380.0 MM HG	59.0 MM HG	267.00 MM HG	
FOAM (ML FOAM/SEC COLLAPSE)	ML/SEC	ML/SEC	ML/SEC	ASTM D892-63
SEQUENCE 1 75°F	10/2	20/4	25/10.5	
SEQUENCE 2 200°F	25/5	10/2	20/1	
SEQUENCE 3 75°F	20/5	20/8	55/34	
SPECIFIC HEAT				PERKEN-ELMER – DIFFERENTIAL SCANNING CALORIMETER MODEL DSC-1
100°F	0.446 CAL/G/°C	0.420 CAL/G/°C	0.400 CAL/G/°C	
200°F	0.480 CAL/G/°C	0.465 CAL/G/°C		
300°F	0.521 CAL/G/°C	0.511 CAL/G/°C		
THERMAL CONDUCTIVITY (BTU/HR-FT-°F)	100°F 0.0788 200°F 0.0756 300°F 0.0723	100°F 0.0784 200°F 0.0748 300°F 0.0712	82°F 0.0777 178°F 0.0752	NONSTEADY STATE TRANSIENT METHOD (HOT WIRE)
FOUR-BALL WEAR TEST (600 RPM)(167°F)(1 HR)	SCAR DIAMETER STEEL ON STEEL / STEEL ON BRONZE	SCAR DIAMETER STEEL ON STEEL / STEEL ON BRONZE	SCAR DIAMETER STEEL ON STEEL / STEEL ON BRONZE	ASTM D2266
1 KG LOAD	0.16 MM / 0.25 MM	0.26 MM / 0.30 MM	0.30 MM / 0.50 MM	
10 KG LOAD	0.39 MM / 0.39 MM	0.43 MM / 0.40 MM	0.41 MM / 0.71 MM	
40 KG LOAD	0.58 MM / 0.63 MM	0.56 MM / 0.83 MM	0.68 MM / 1.65 MM	
FLASH POINT	340°F	380°F	360°F	ASTM D92
FIRE POINT	360°F	420°F	420°F	ASTM D92
AIT	800°F	960°F	925°F	ASTM D2155
AIT	>1000°F	>1000°F	>1000°F	ASTM D286
HOT MANIFOLD DRIP				AMS 3150C
MANIFOLD °F	1300°F	1300°F	1300°F	
RESULTS	EQUIVALENT TO HS-1	BETTER THAN HS-1	BETTER THAN HS-1	
HOT MANIFOLD SPRAY				
FLUID °F	167°F	167°F	167°F	
MANIFOLD °F	1100°F	1200°F	1200°F	
FLUID PRESSURE	1000 PSI	1000 PSI	1000 PSI	
RESULTS	NO FLASHING OR BURNING	NO FLASHING OR BURNING	NO FLASHING OR BURNING	
HIGH PRESSURE SPRAY	EQUIVALENT TO HS-1	BETTER THAN HS-1	BETTER THAN HS-1	AMS 3150C
LOW PRESSURE SPRAY	BETTER THAN HS-1	BETTER THAN HS-1	BETTER THAN HS-1	AMS 3150C
WICK FLAMMABILITY	EQUIVALENT TO HS-1	EQUIVALENT TO HS-1	EQUIVALENT TO HS-1	AMS 3150C

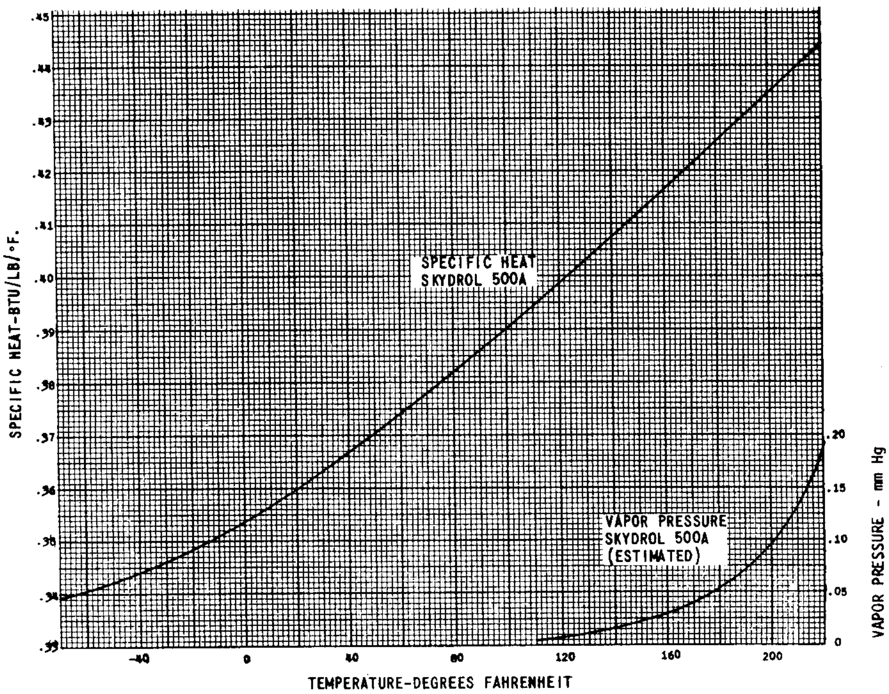
10.522 PROPERTIES OF SKYDROL 500B FLUID





10.522 PROPERTIES OF SKYDROL 500B FLUID (CONT'D)





10.523 TUBING PRESSURE DROP FOR SKYDROL 500 FLUID The following charts plot pressure drop vs flow for straight tubing at various temperatures. The charts are based on fluid properties at atmospheric pressure. Corrections must be made to account for effects of viscosity and density changes with pressure. The correction factors are shown (vs pressure) on the next page.

These pressure drop charts are taken from Douglas Report SM-19405, dated 9-14-55, for Skydrol 500 fluid, quite similar to 500A and 500B Skydrol fluids. The correction factors are based on typical values of kinematic viscosity and density for Skydrol 500B, supplied by Monsanto Co. in 1967.

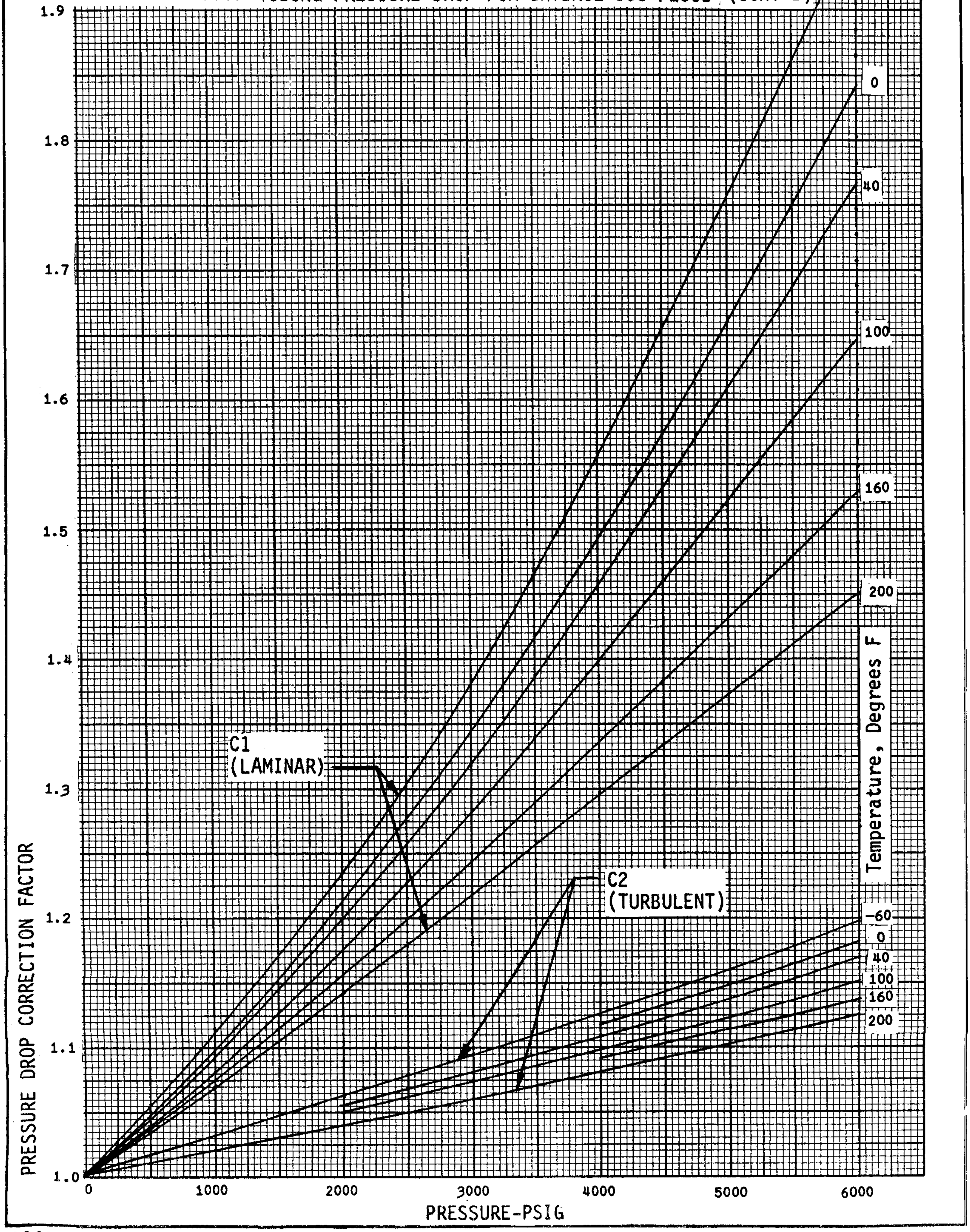
Correction factor C1 is to be applied to pressure drop in the laminar flow region (straight lines at 45 degrees in the charts). The smaller correction factor C2 applies to the turbulent flow region. (straight lines at 60-1/4 degrees). Use conservative judgment in applying a correction factor in the transition region between laminar and turbulent flow.

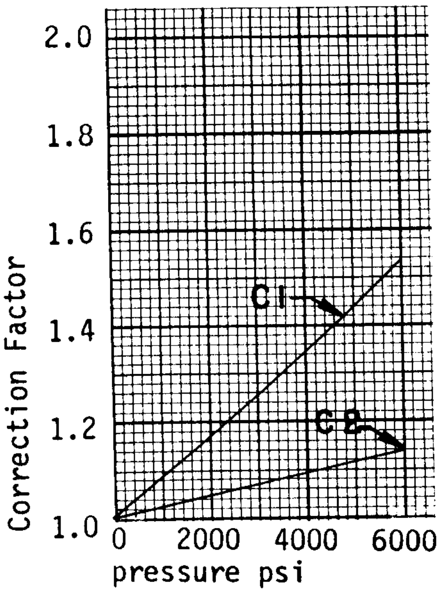
C1 is the ratio of absolute viscosity at pressure to that at atmospheric pressure. C2 is the ratio of density times the fourth root of kinematic viscosity at pressure to that at atmospheric pressure.

Given the flow, correct the pressure loss (psi drop per foot) obtained from the chart by multiplying it by the correction factor.

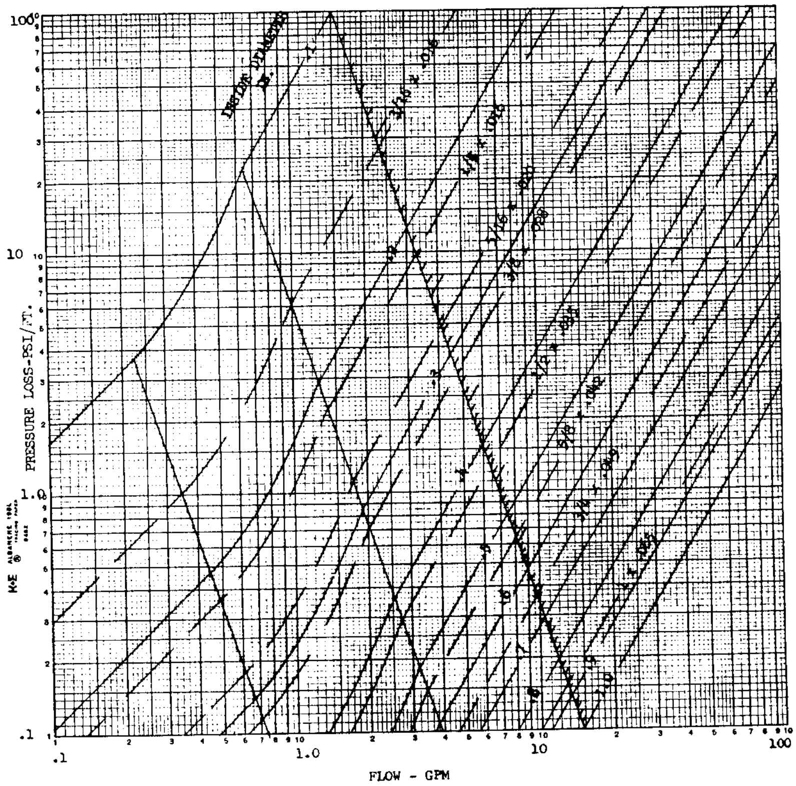
Given the pressure loss, divide it by the correction factor, then determine the flow from the chart. (Alternatively, for laminar flow only, the flow first may be obtained from the chart, then may be corrected by dividing by C1.)

10.523 TUBING PRESSURE DROP FOR SKYDROL 500 FLUID (CONT'D)

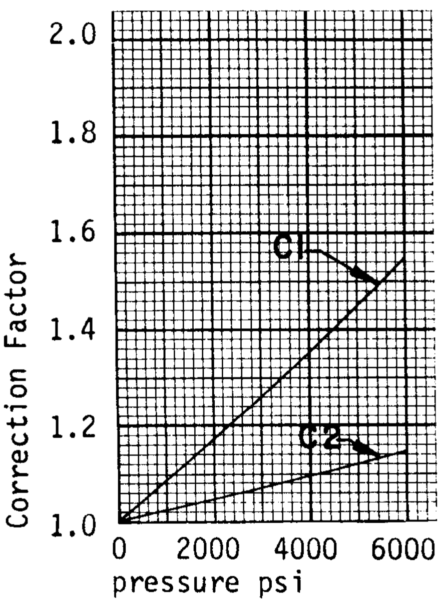




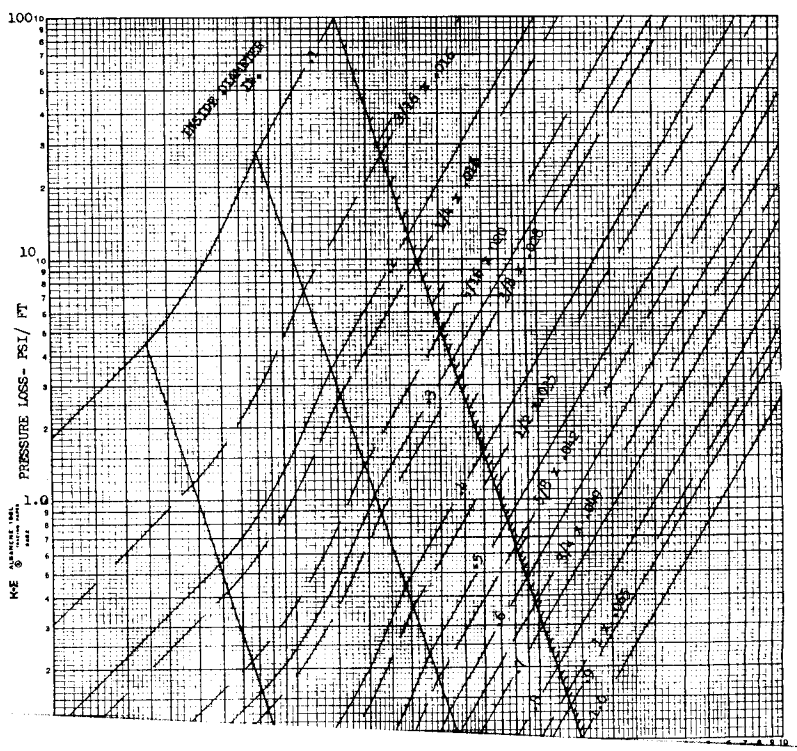
PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 160°F.
FOR SKYDROL -500 FLUID



Use C1 to correct pressure loss for laminar flow, and C2 for turbulent flow.

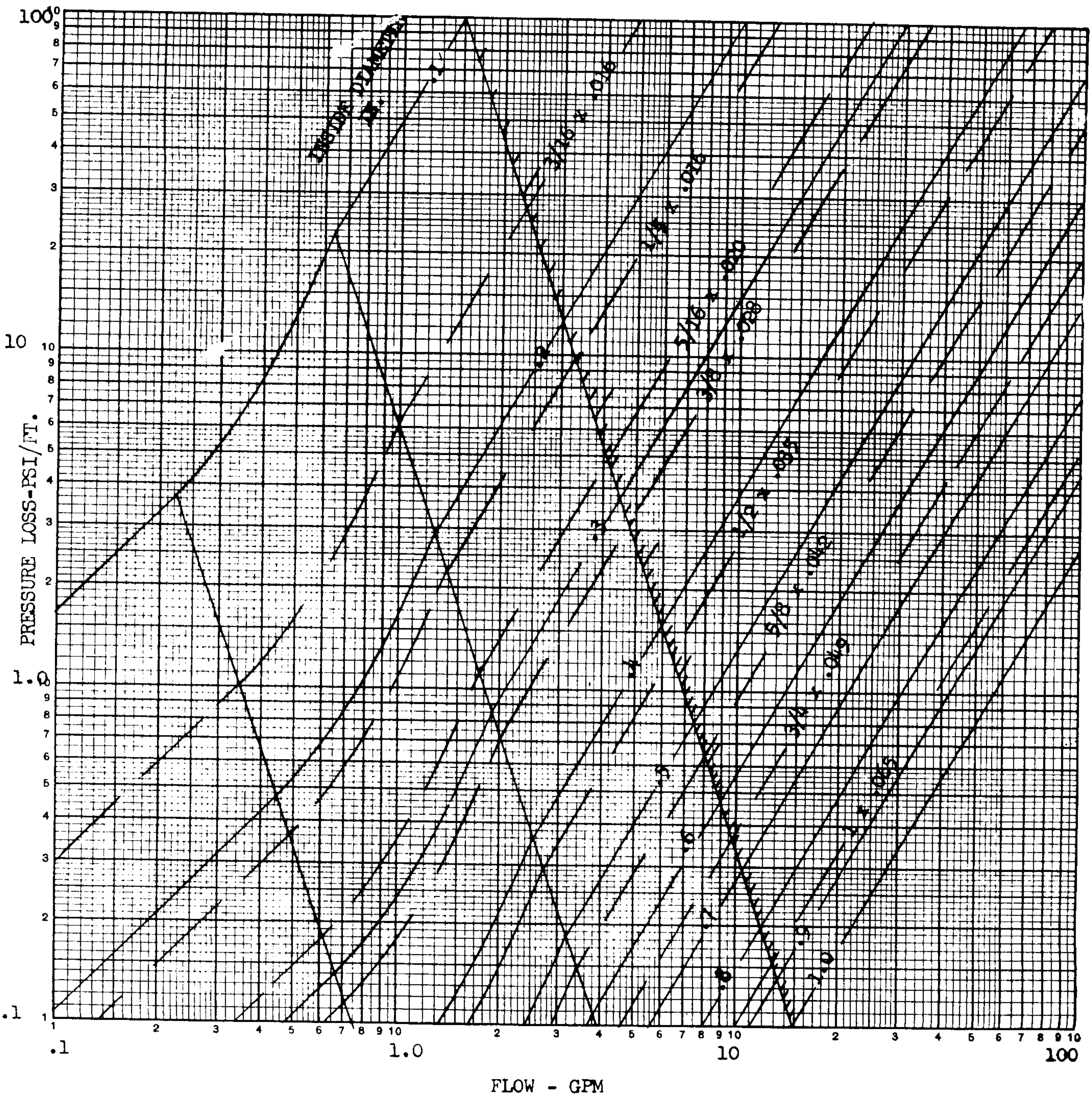


PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 150°F.
FOR SKYDROL -500 FLUID



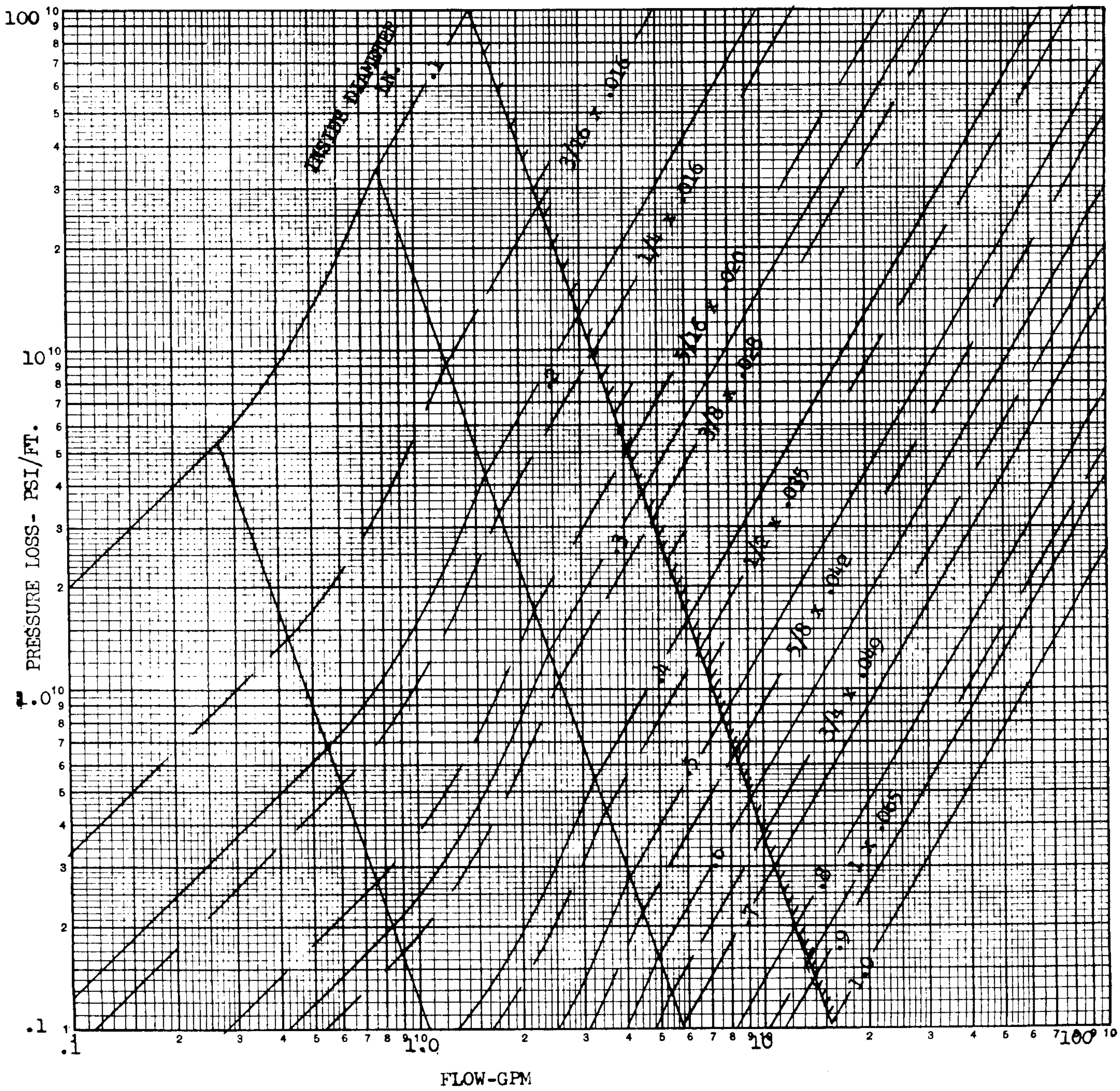
10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 160°F.
FOR SKYDROL -500 FLUID



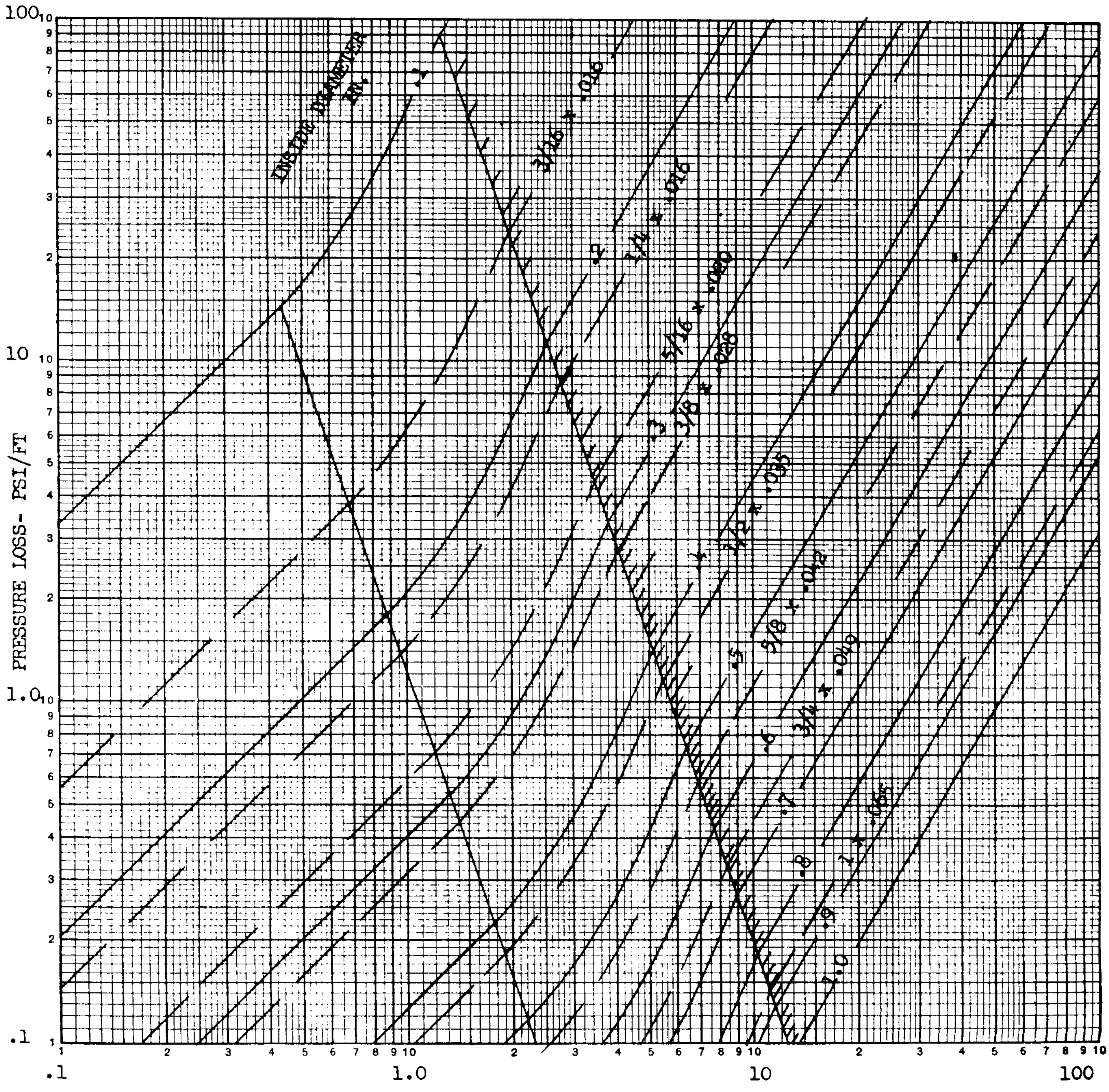
10.523 (CONT'D)

PRSSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 140°F.
FOR SKYDROL-500 FLUID



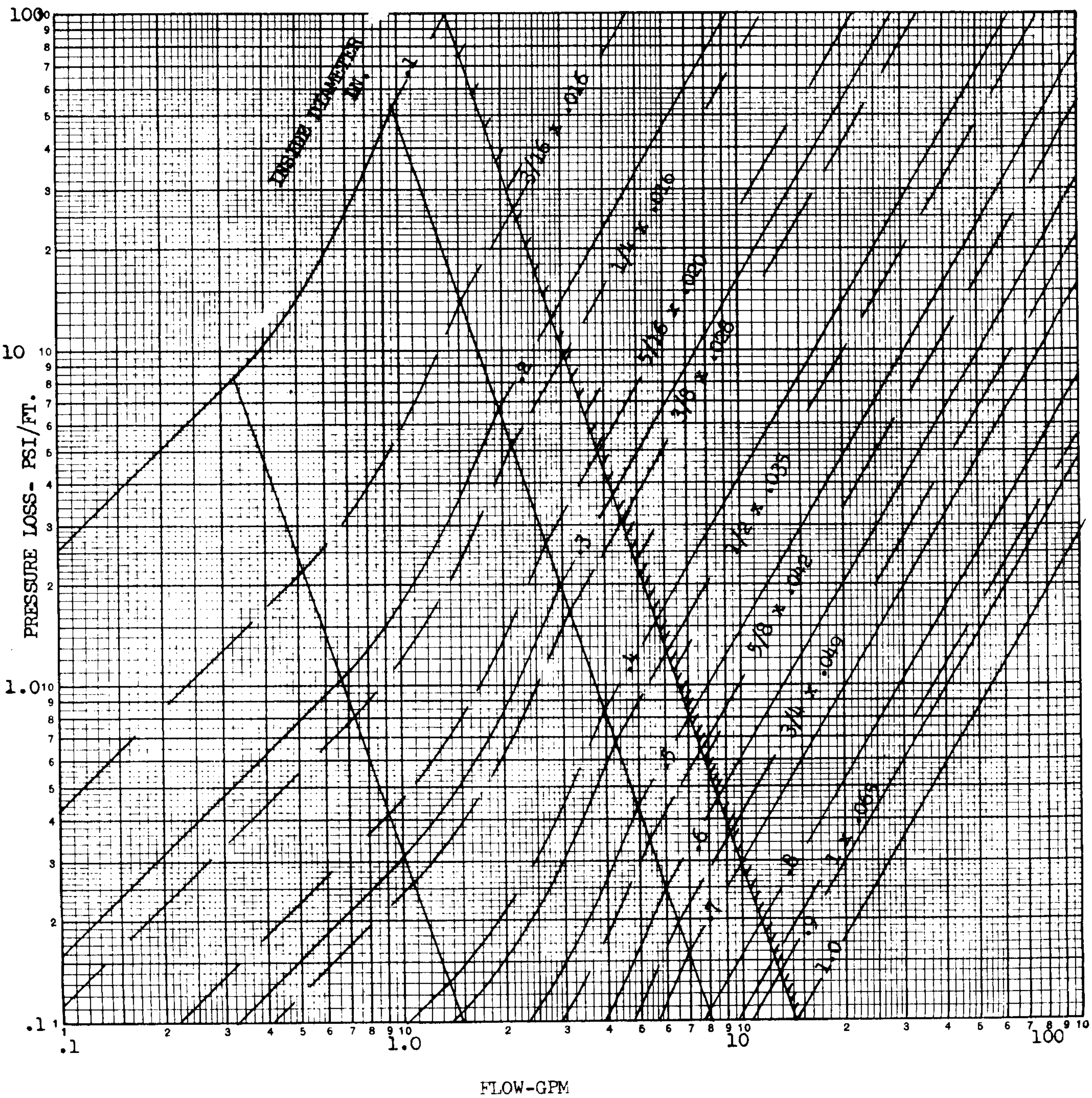
10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 100°F.
FOR SKYDROL-500 FLUID



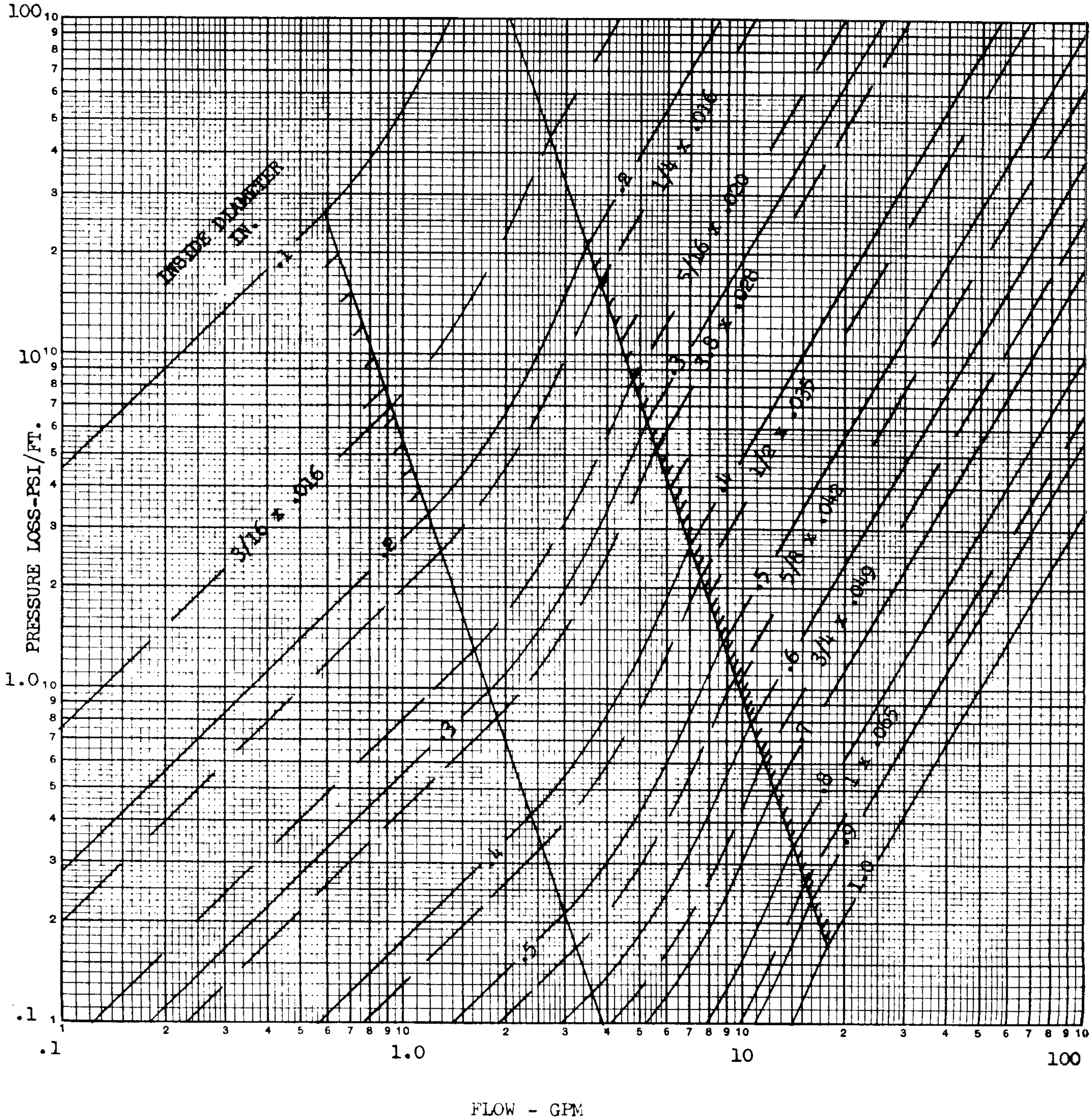
10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 120°F.
FOR SKYDROL-500 FLUID

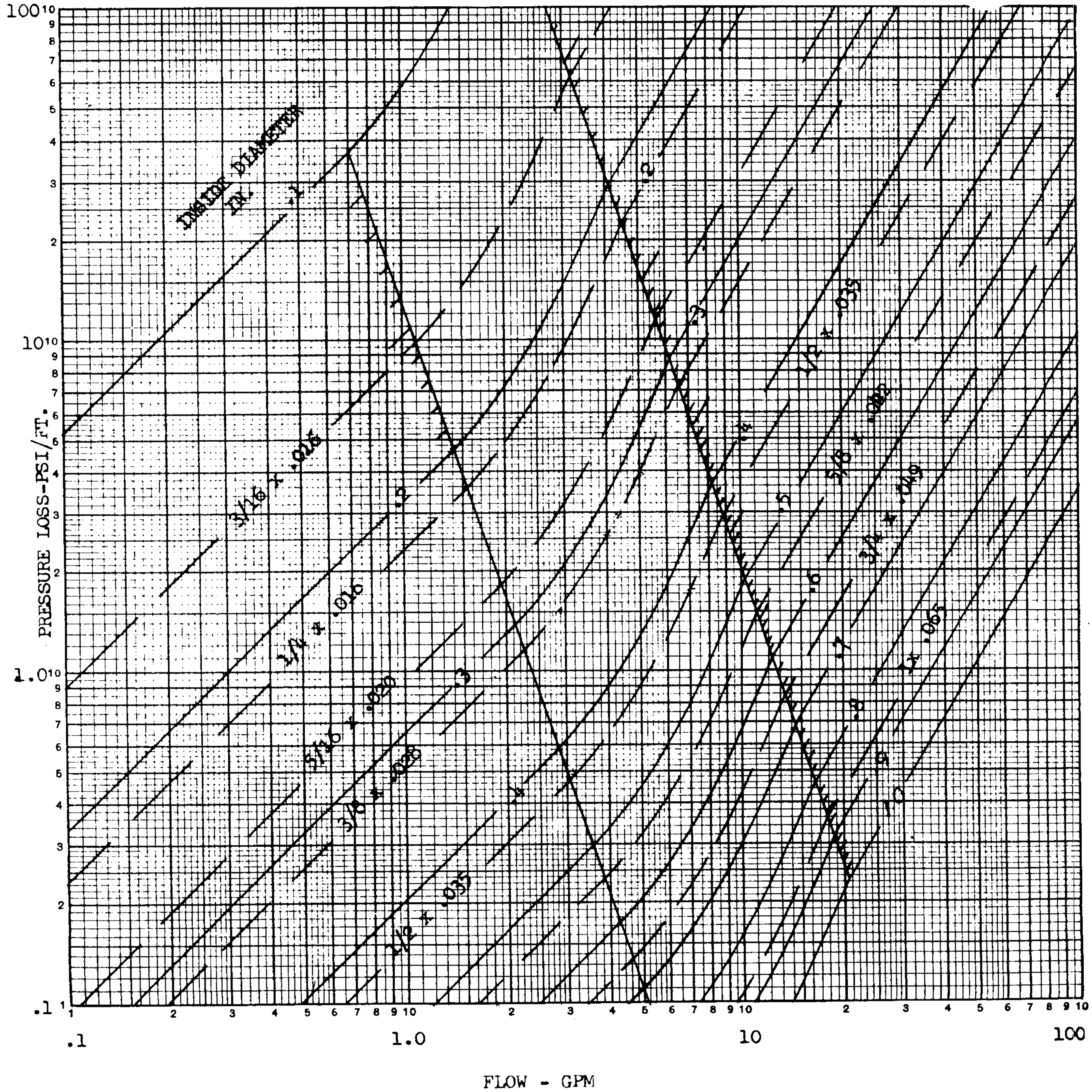


10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 80°F.
FOR SKYDROL-500 FLUID

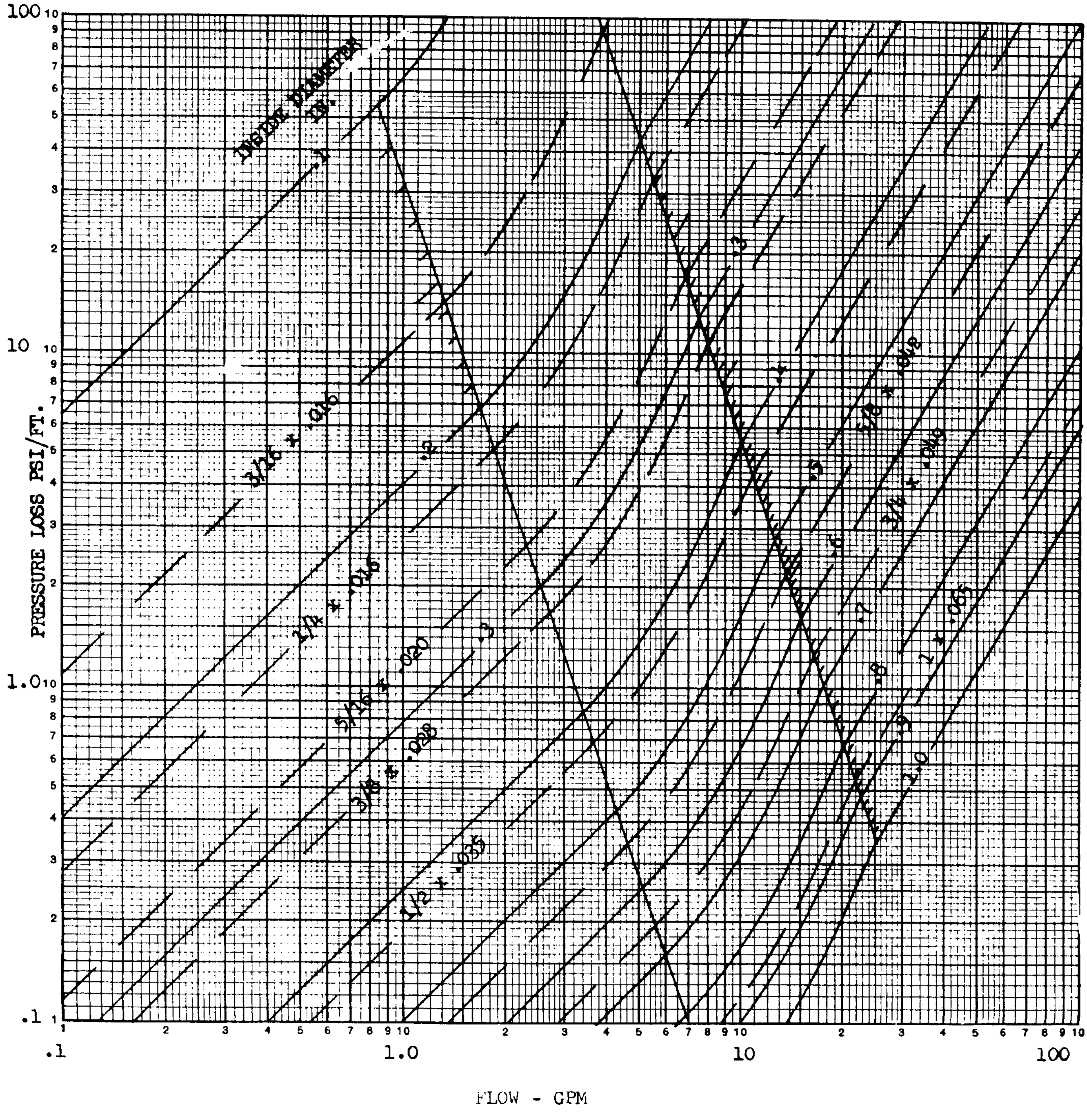


PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 70°F.
FOR SKYDROL-500 FLUID

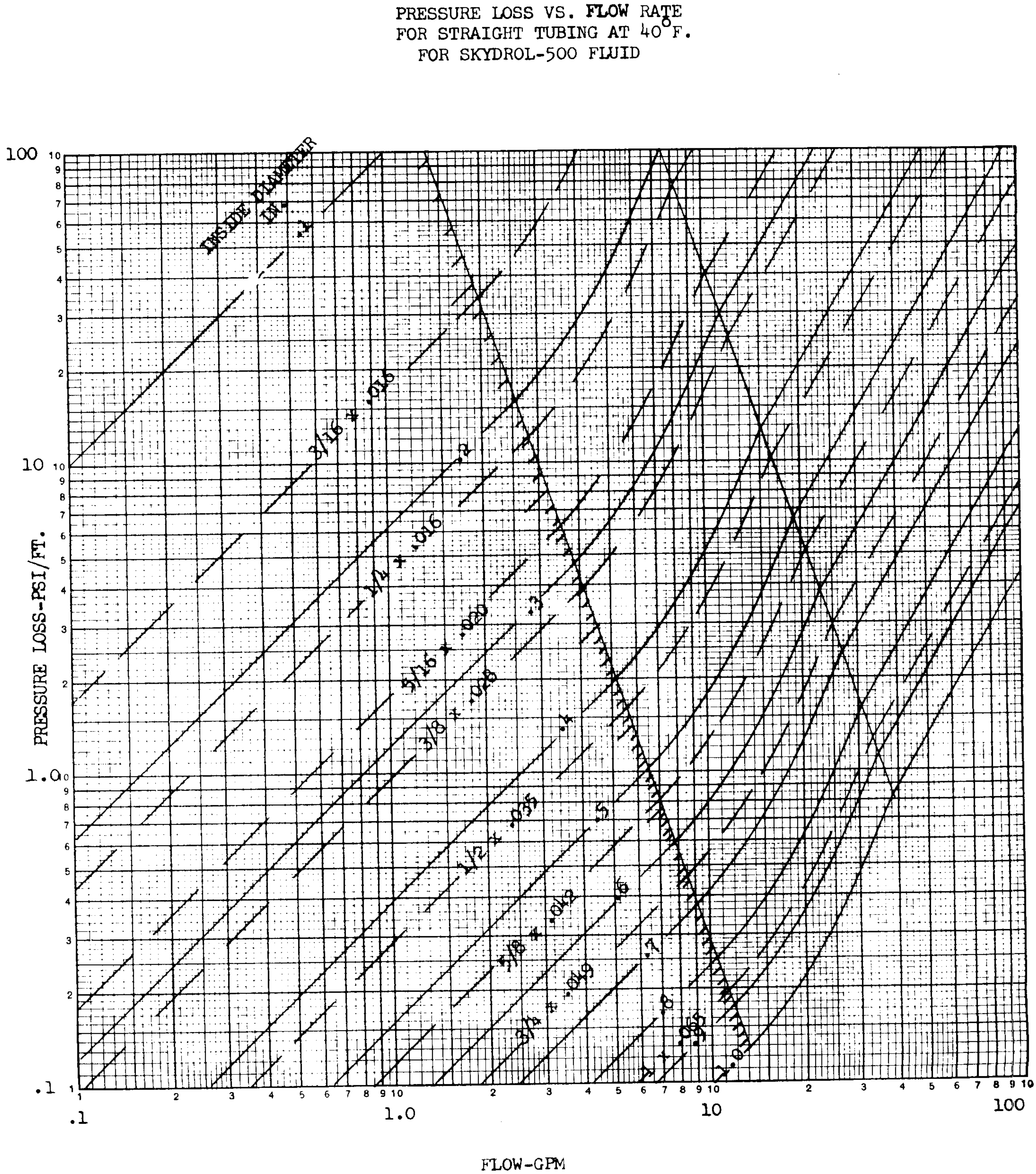


10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 60°F.
FOR SKYDROL-500 FLUID

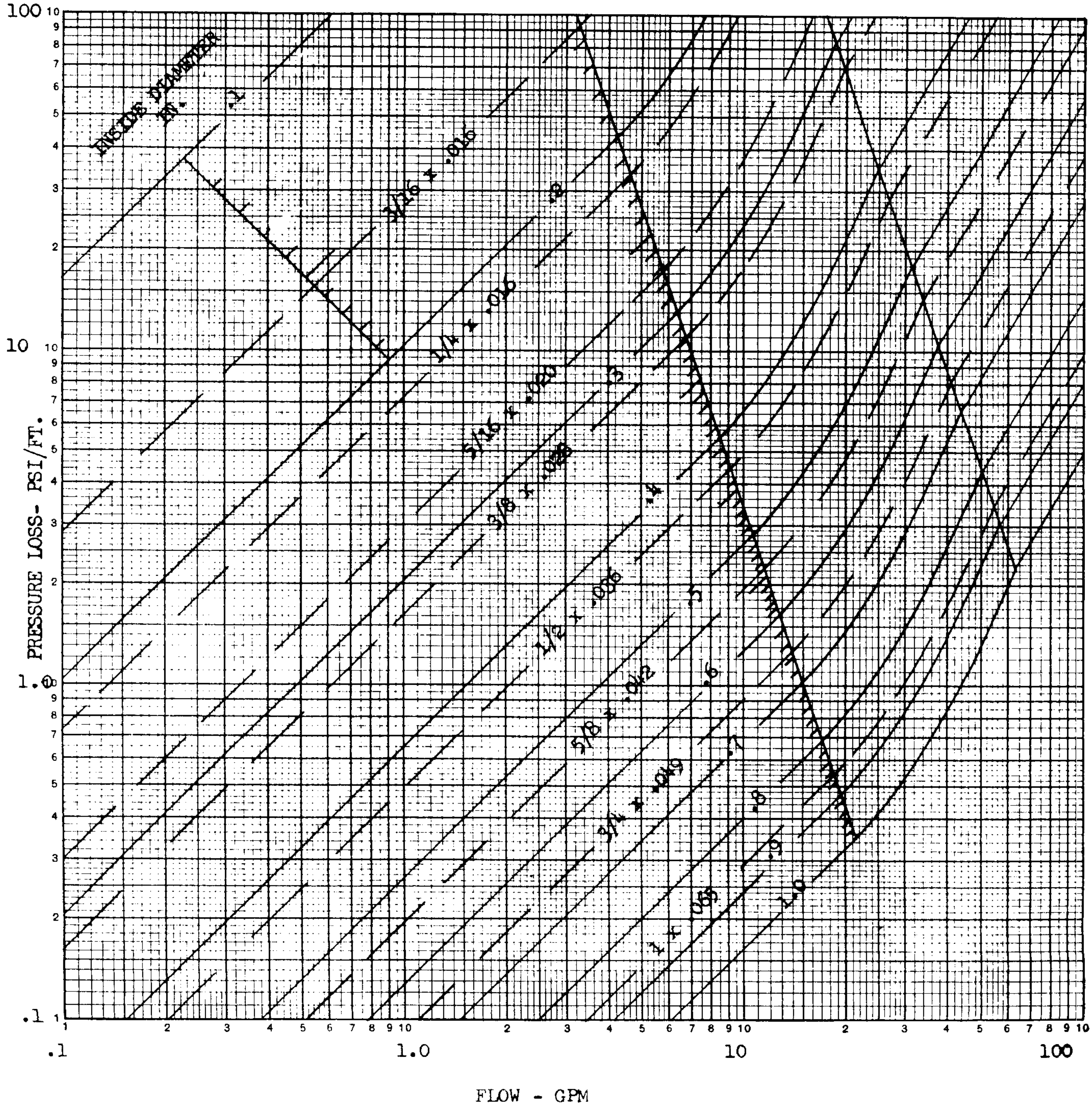


10.523 (CONT'D)


PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 40°F.
FOR SKYDROL-500 FLUID
INSIDE DIAMETER IN.
.1
.2
.3
.4
.5
.6
.7
.8
1.0
3/16 x .016
1/4 x .016
5/16 x .020
3/8 x .028
1/2 x .035
5/8 x .042
3/4 x .049
1 x .065
PRESSURE LOSS-PSI/FT.
100
10
1.0
.1
.1
1.0
10
100
FLOW-GPM

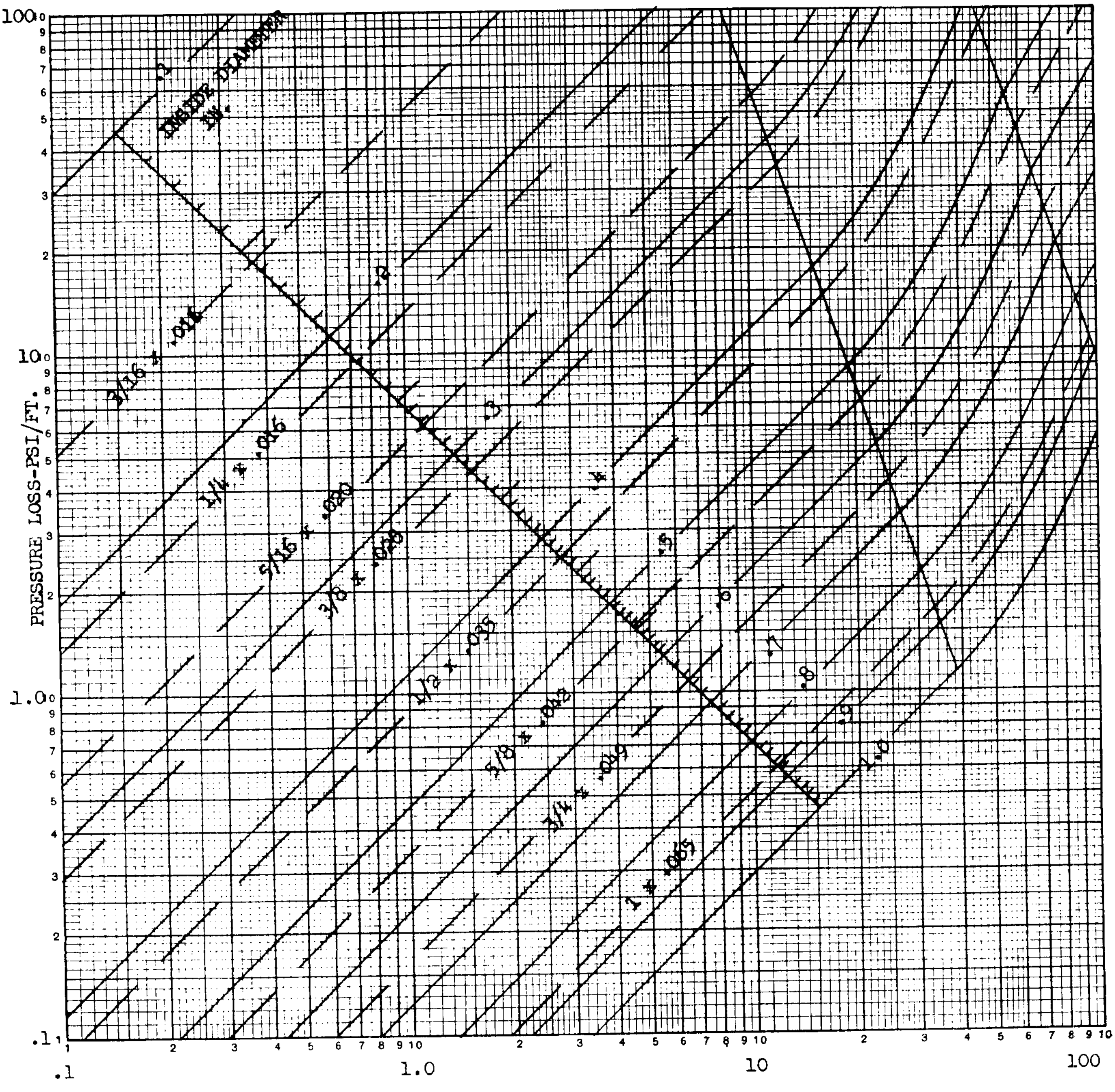
10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 20°F.
FOR SKYDROL-500 FLUID



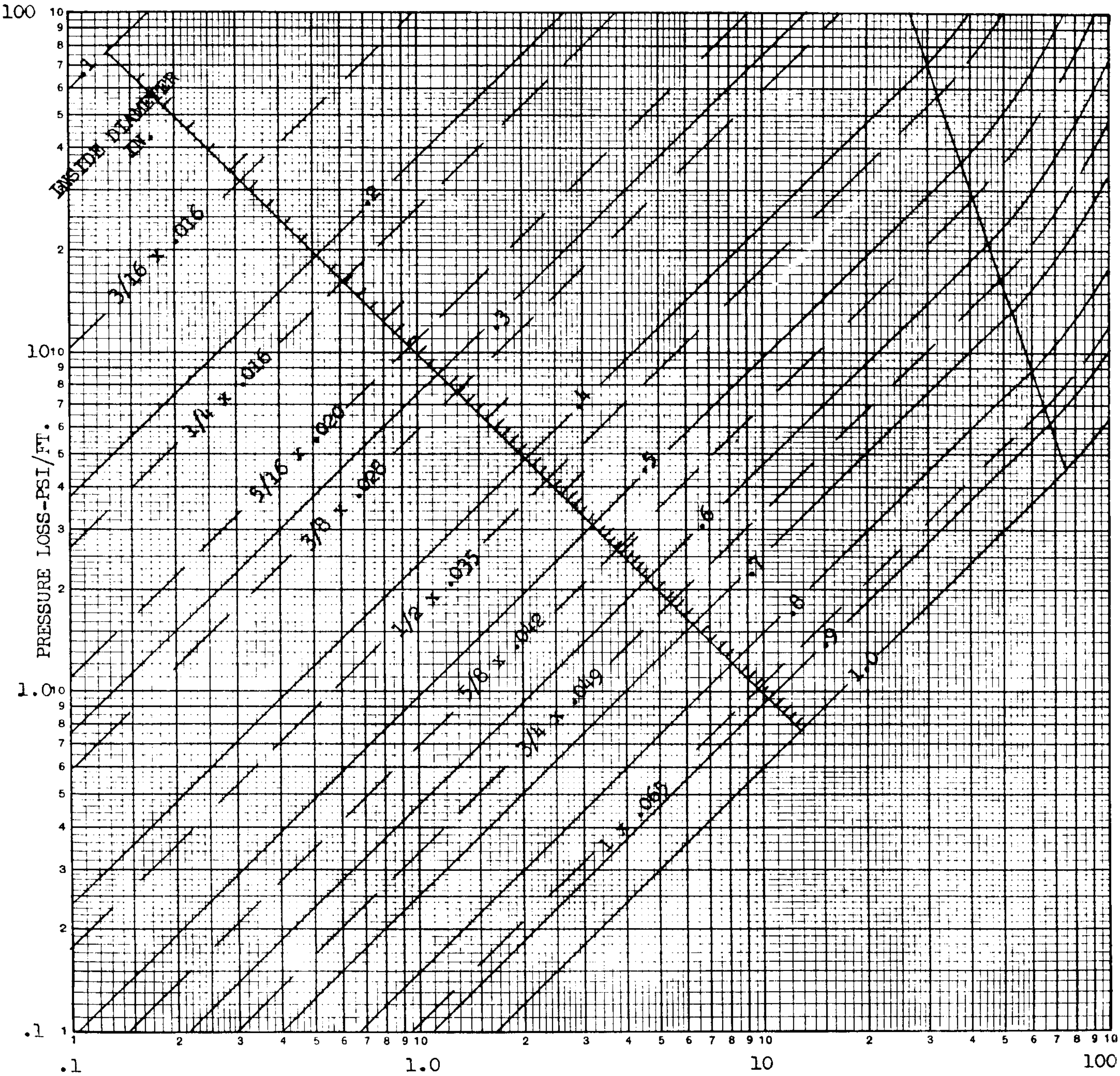
10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT 0°F.
FOR SKYDROL-500 FLUID



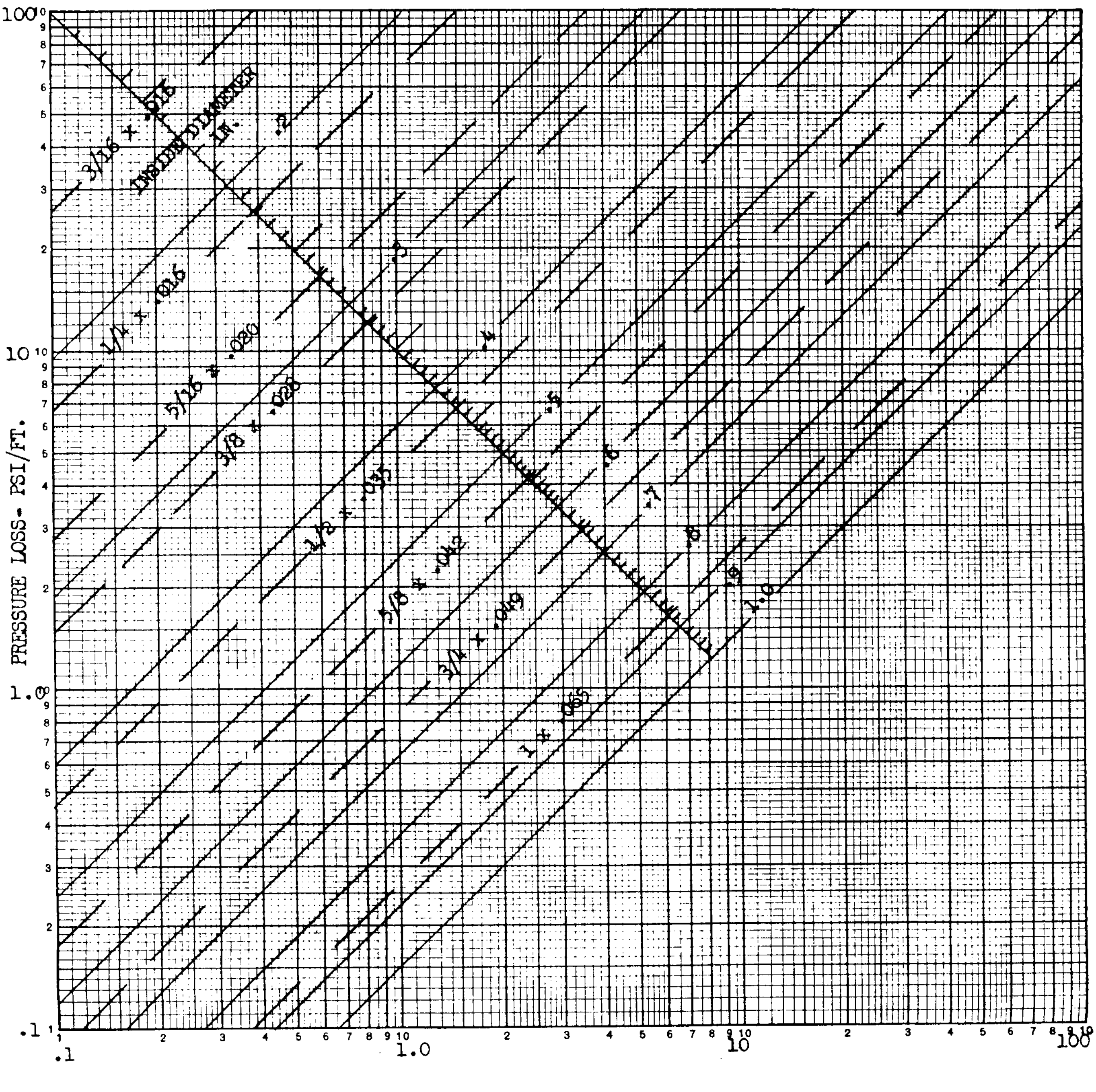
10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT -20°F.
FOR SKYDROL-500 FLUID



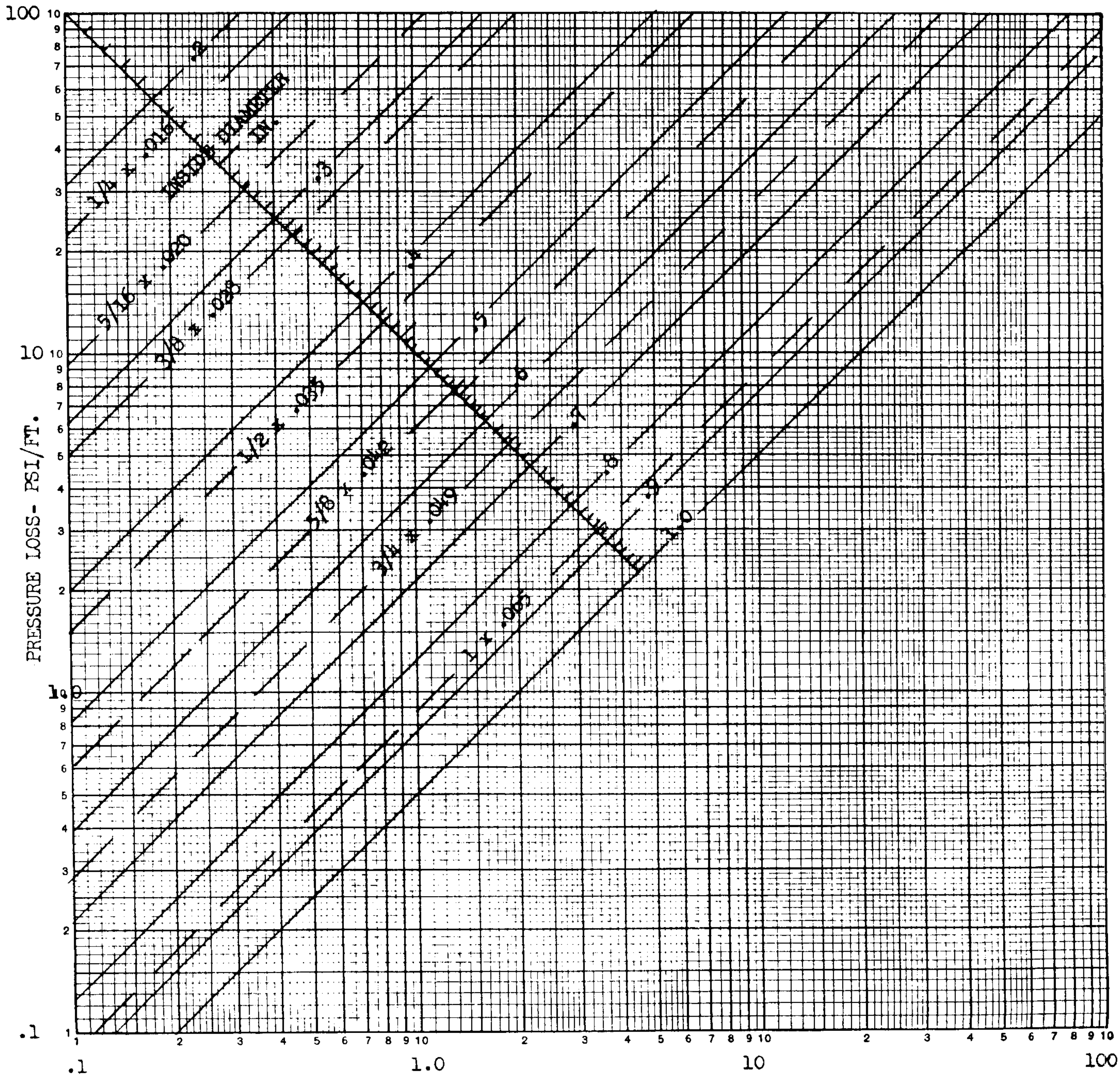
10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT -40°F.
FOR SKYDROL-500 FLUID

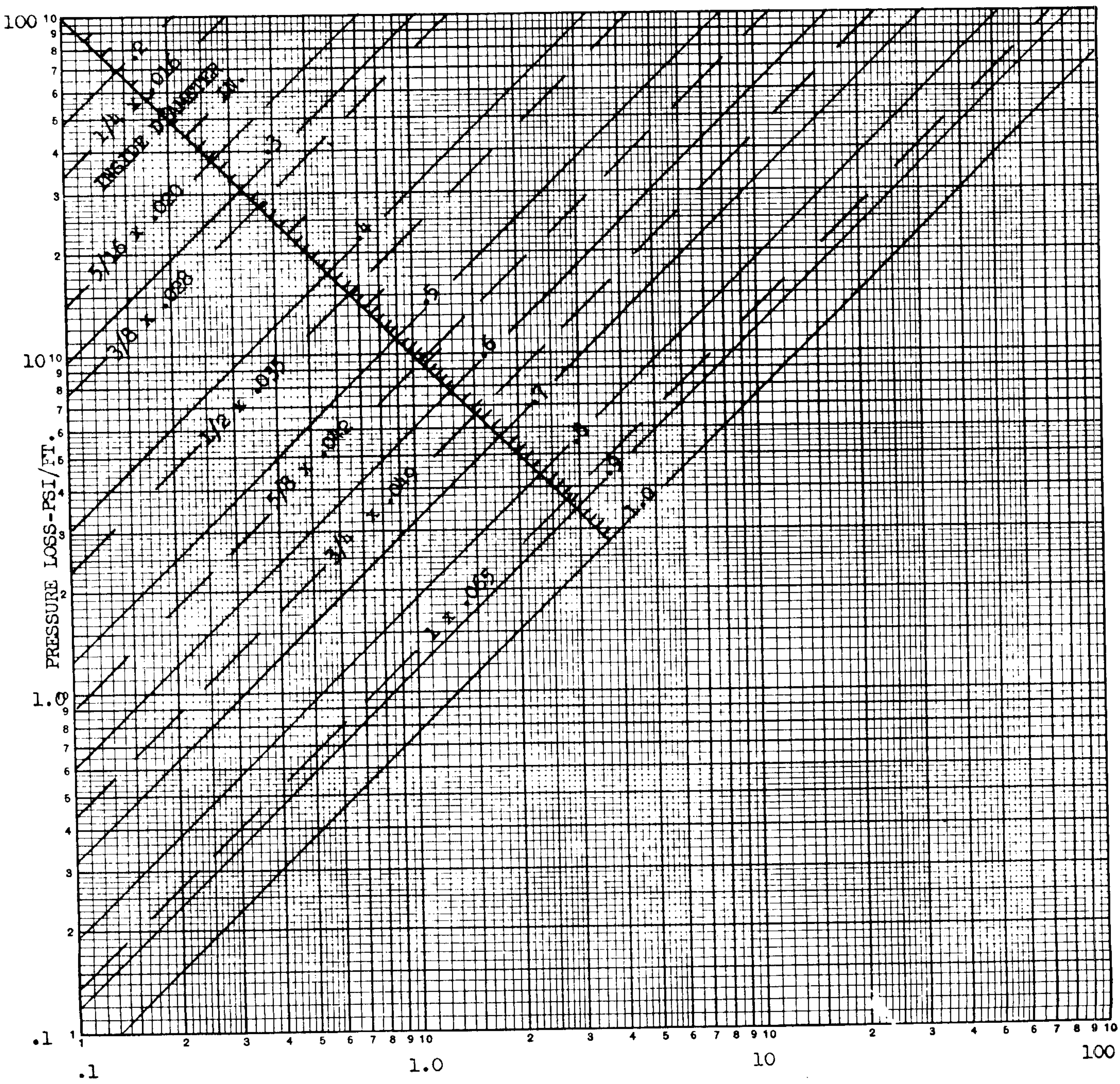


10.523 (CONT'D)

PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT -60°F.
FOR SKYDROL-500 FLUID

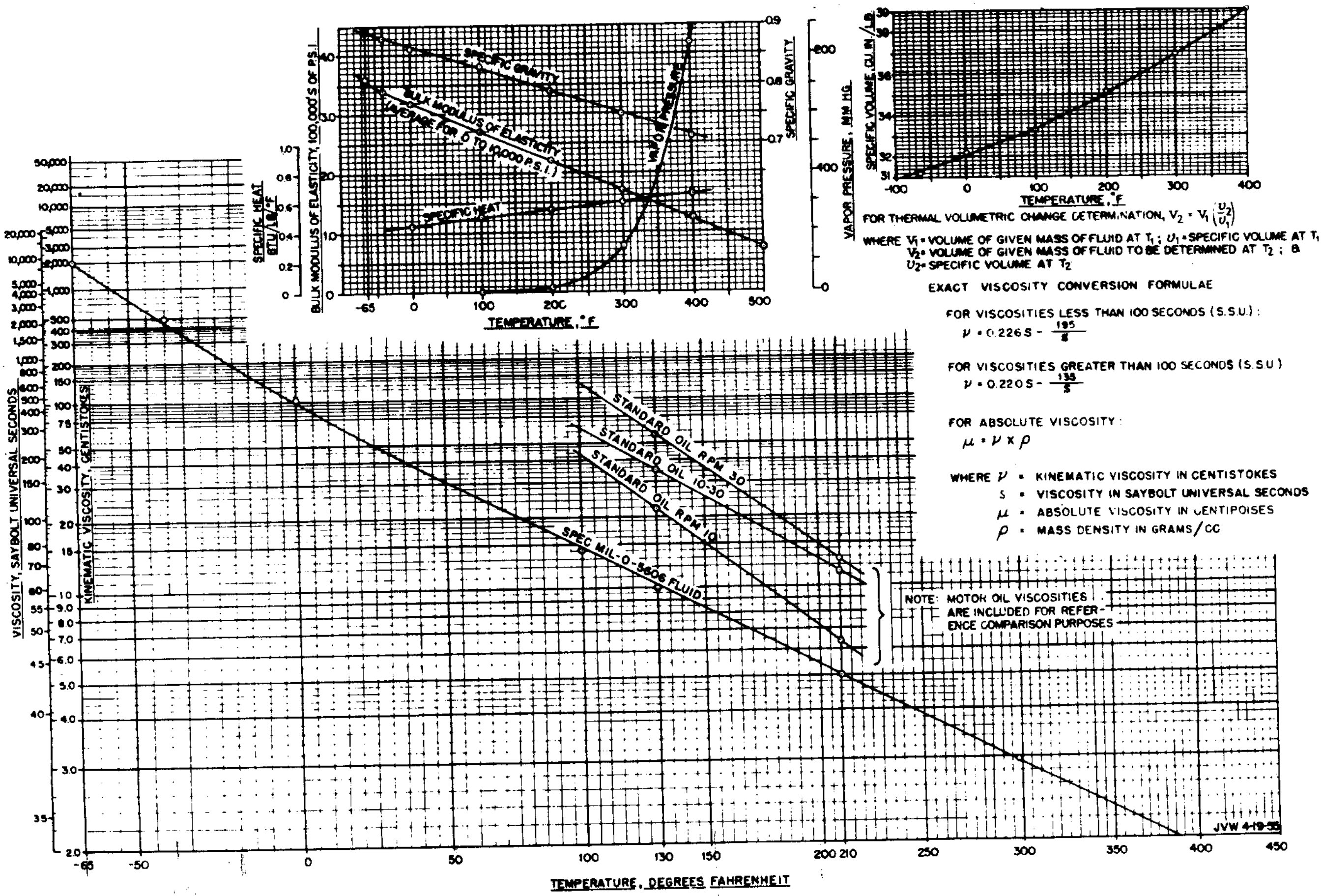


PRESSURE LOSS VS. FLOW RATE
FOR STRAIGHT TUBING AT -65°F.
FOR SKYDROL-500 FLUID



10.53 PETROLEUM-BASED FLUIDS In the following pages of this section, the fluid properties and pressure drop characteristics refer to MIL-0-5606, MIL-H-5606A and MIL-H-5606B. As far as the information presented in this section is concerned, virtually no difference exists between these fluids. The difference between MIL-0-5606 and MIL-H-5606 is one of designation. The difference between MIL-H-5606A and MIL-H-5606B is that MIL-H-5606B has improved shear stability and tighter contamination requirements. A more fire-resistant fluid is defined by MIL-H-83282 and its properties are included in this section.

10.531 PROPERTIES OF MIL-H-5606 FLUID



SOURCE: PENNSYLVANIA STATE UNIVERSITY REPORT NO. PRL 6.3-DEC53

10.532 TUBING PRESSURE DROP FOR MIL-H-5606 FLUID The following charts plot tubing pressure drops for several different temperatures (based on viscosity at atmospheric pressure).

To account for change of kinematic viscosity with pressure (as calculated from tests of similar oils by Pennsylvania State College) a correction factor is plotted on the next page.

These charts are taken from Douglas Report SM-19405 (which also contains similar data for two types of Skydrol fluid). The MIL-0-5606 data are based on USAF Technical Report No. 5997.

TR-5997 indicates that tubing pressure drop charts are derived (for kinematic viscosity at atmospheric pressure) from the following:

1. Laminar flow up to R = 1350: $f = 64/R$

2. Turbulent flow above R = 3400: $f = 0.316/R^{.25}$

3. Transition region between R = 1350 and R = 3400: based on an average curve through test data points

where:

$$f = \text{friction factor} = \frac{2gh}{V^2} \times \frac{\text{inside diameter}}{\text{tube length}}$$

$$R = \text{Reynold's number} = V \times \frac{\text{inside diameter}}{\text{kinematic viscosity}}$$

g = gravity acceleration

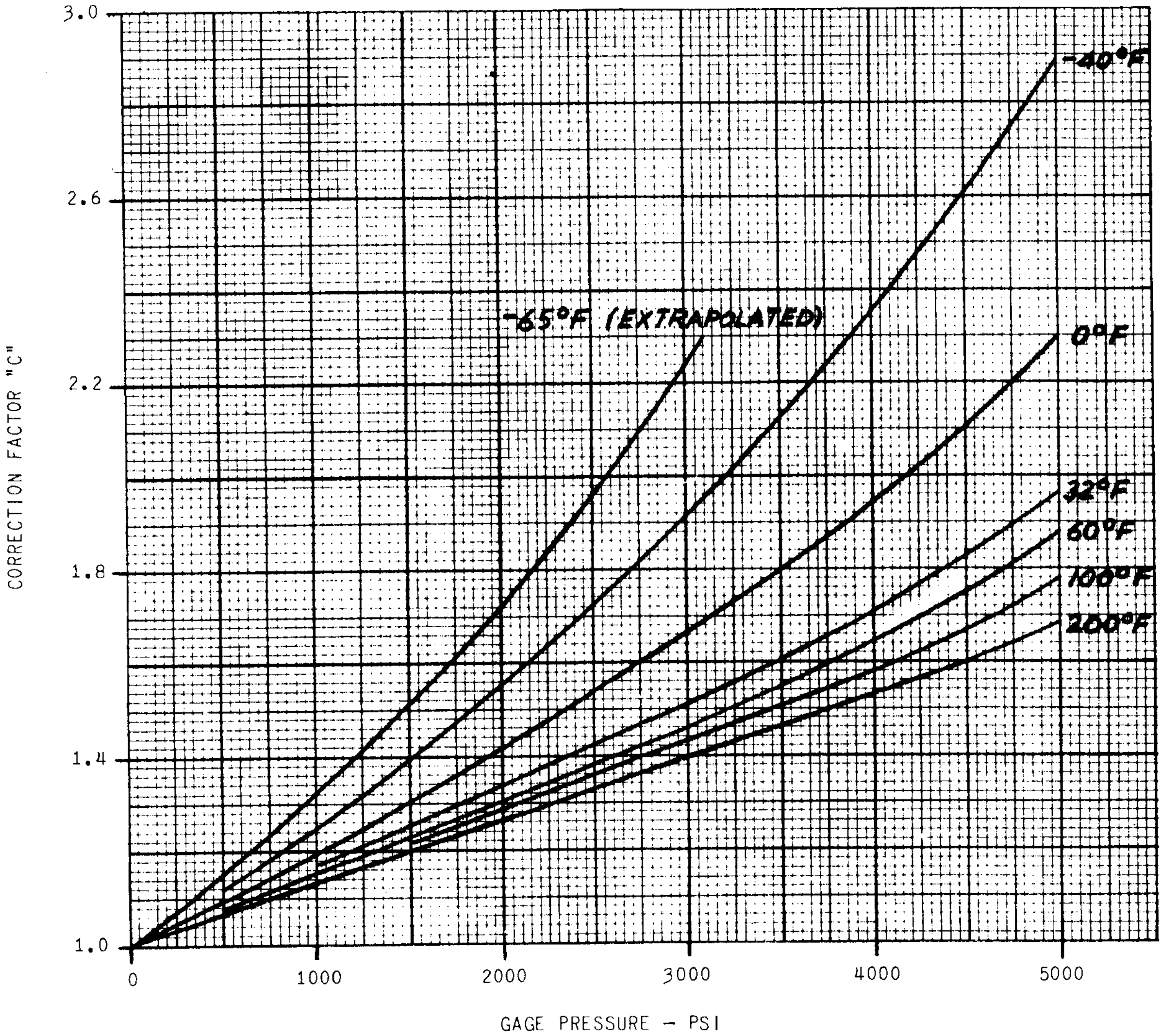
h = pressure drop head

V = average flow velocity

PRESSURE LOSSES IN TUBING

10.532 (CONT.)

KINEMATIC VISCOSITY CORRECTION FACTOR VS. PRESSURE FOR MIL-O-5606 HYDRAULIC FLUID



NOTES:

1. "C" is the ratio of kinematic viscosity at indicated pressure to that at atmospheric pressure.

2. In the (slower) laminar flow region, determine "C" from the estimated average pressure of the fluid for applicable temperature.

   a. Given the pressure drop, correct the flow obtained in the applicable temperature graph of the following pages by dividing by "C".

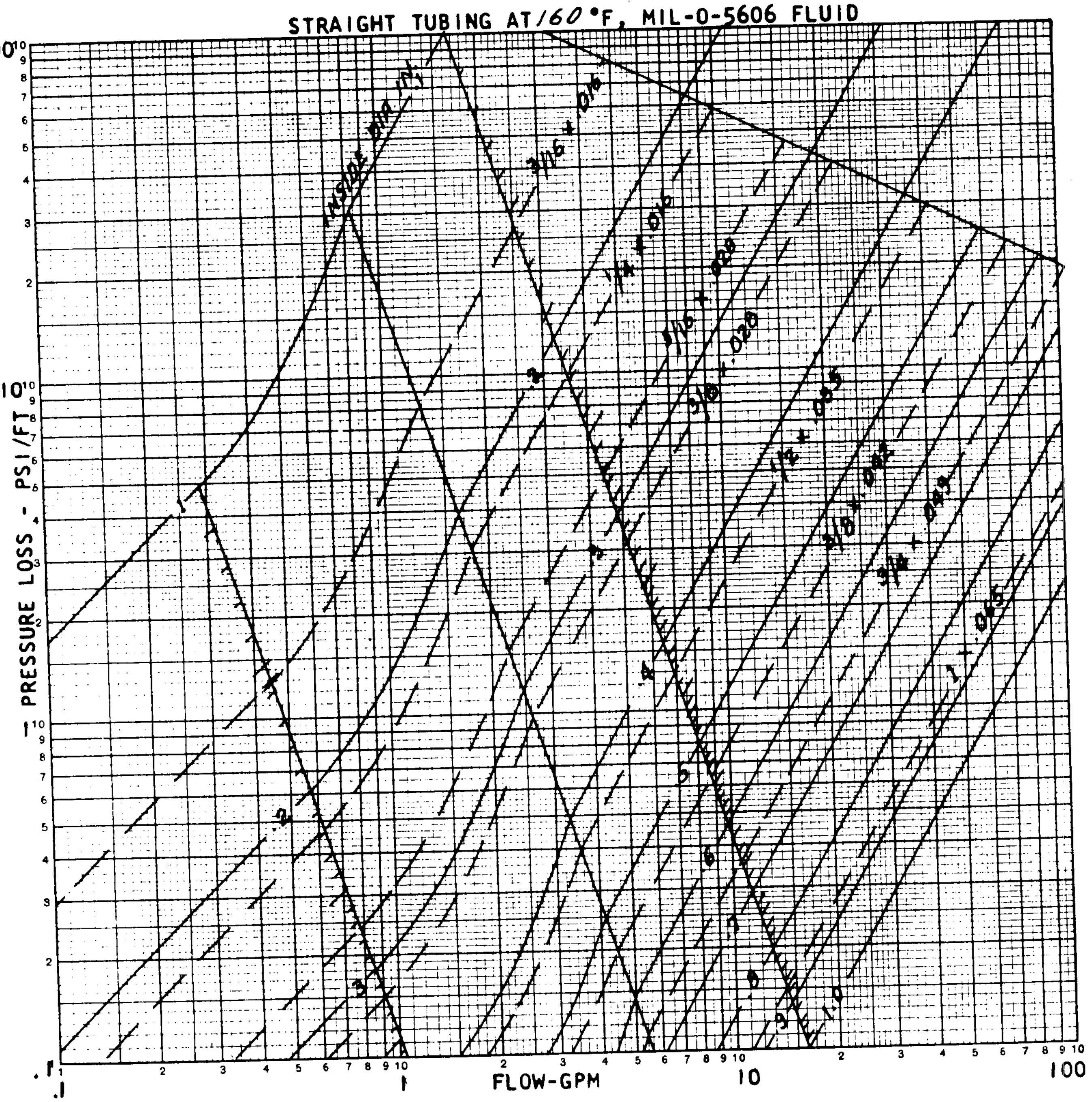
   b. Given the flow, correct the drop obtained in the graph by multiplying by "C".

3. In the turbulent flow region, the effect of "C" usually can be neglected, since pressure drop is proportional only to the fourth root of kinematic viscosity, and flow is inversely proportional only to the seventh root of kinematic viscosity.

4. In the transition region between linear and turbulent flow (curved portion of charts), use conservative judgement in estimating flow or pressure drop corrections.

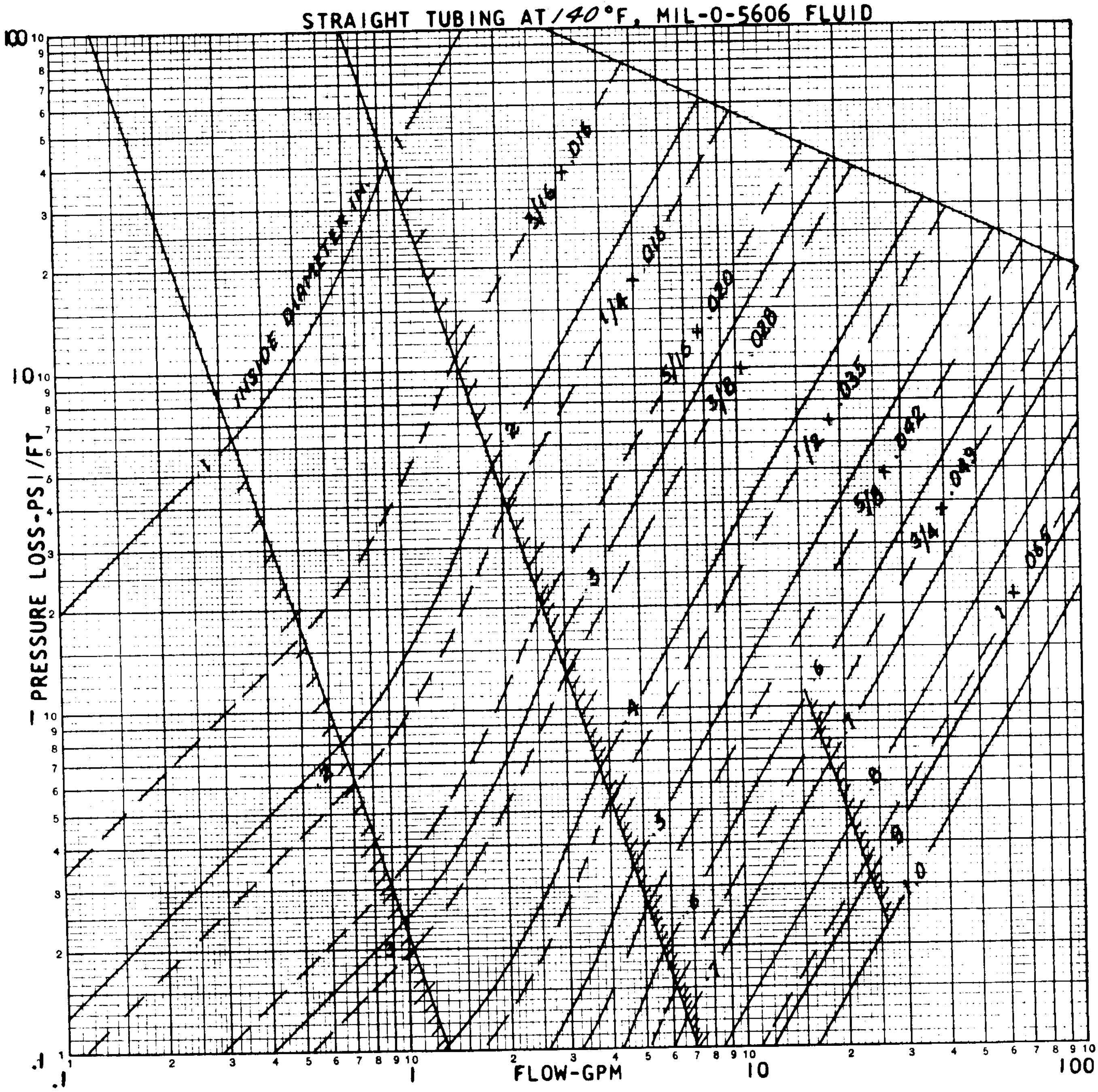
10.532 (CONT'D)

PRESSURE LOSSES IN TUBING



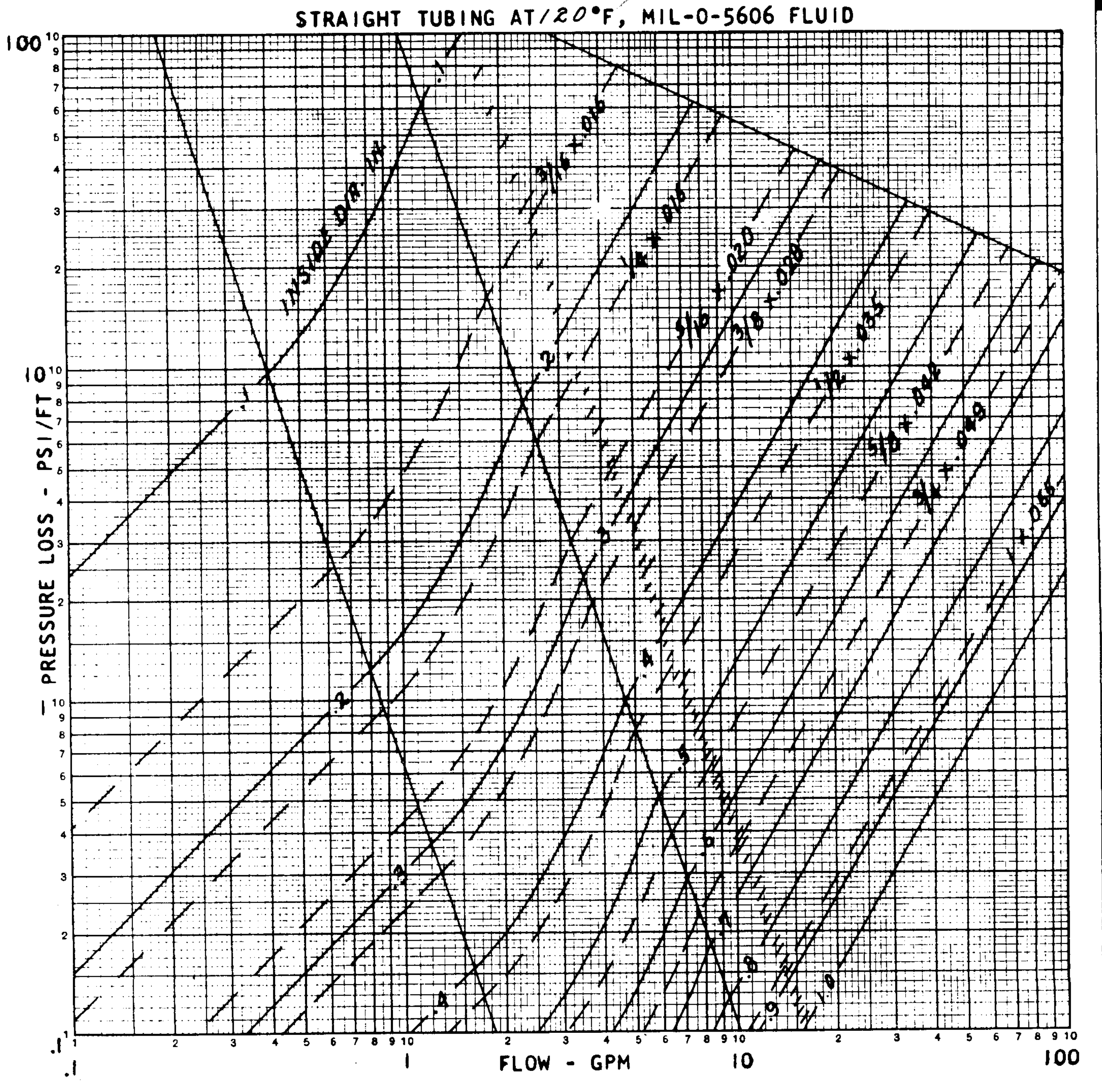
10.532 (CONT'D)

PRESSURE LOSSES IN TUBING



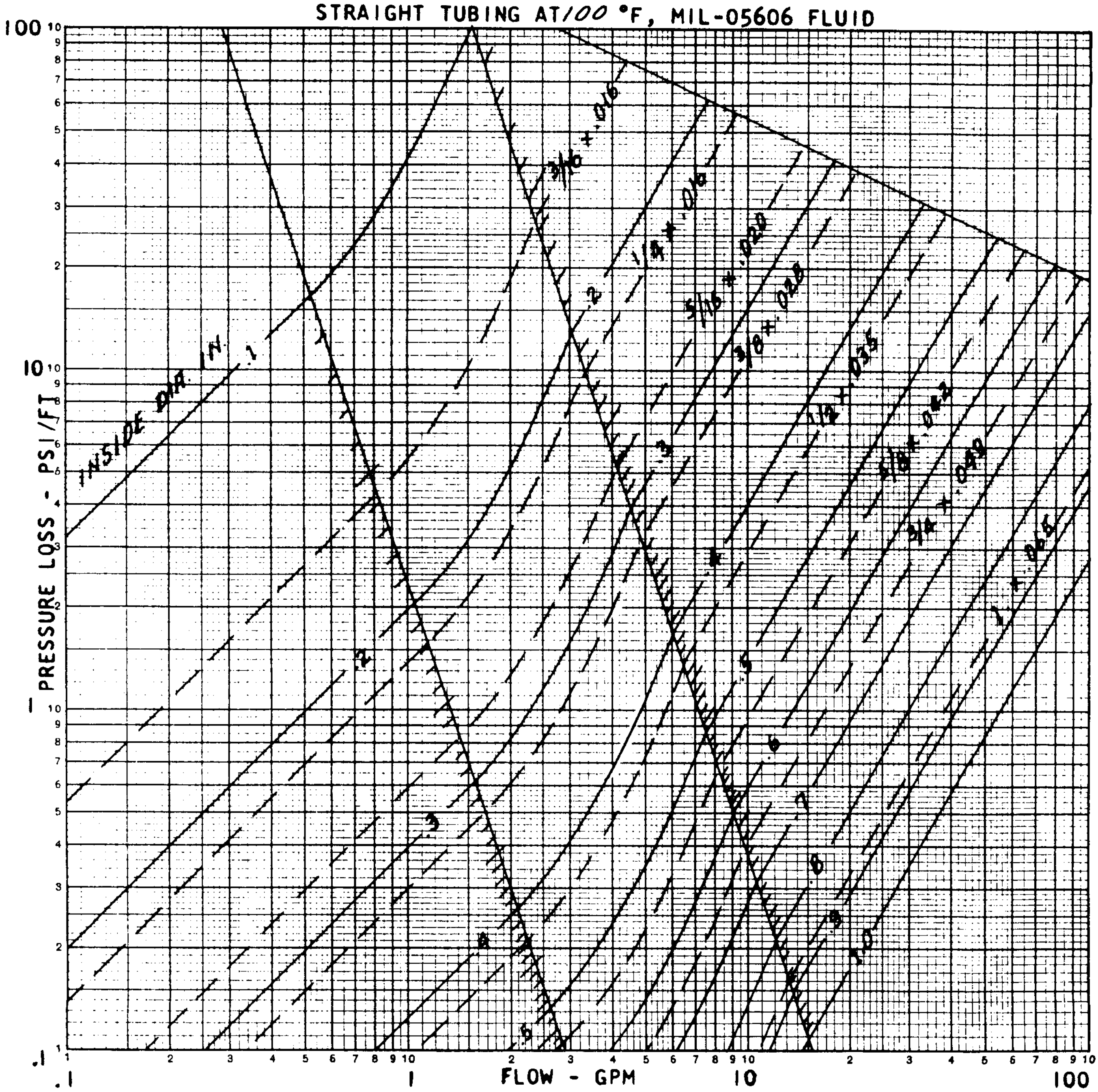
10.532 (CONT'D)

PRESSURE LOSSES IN TUBING



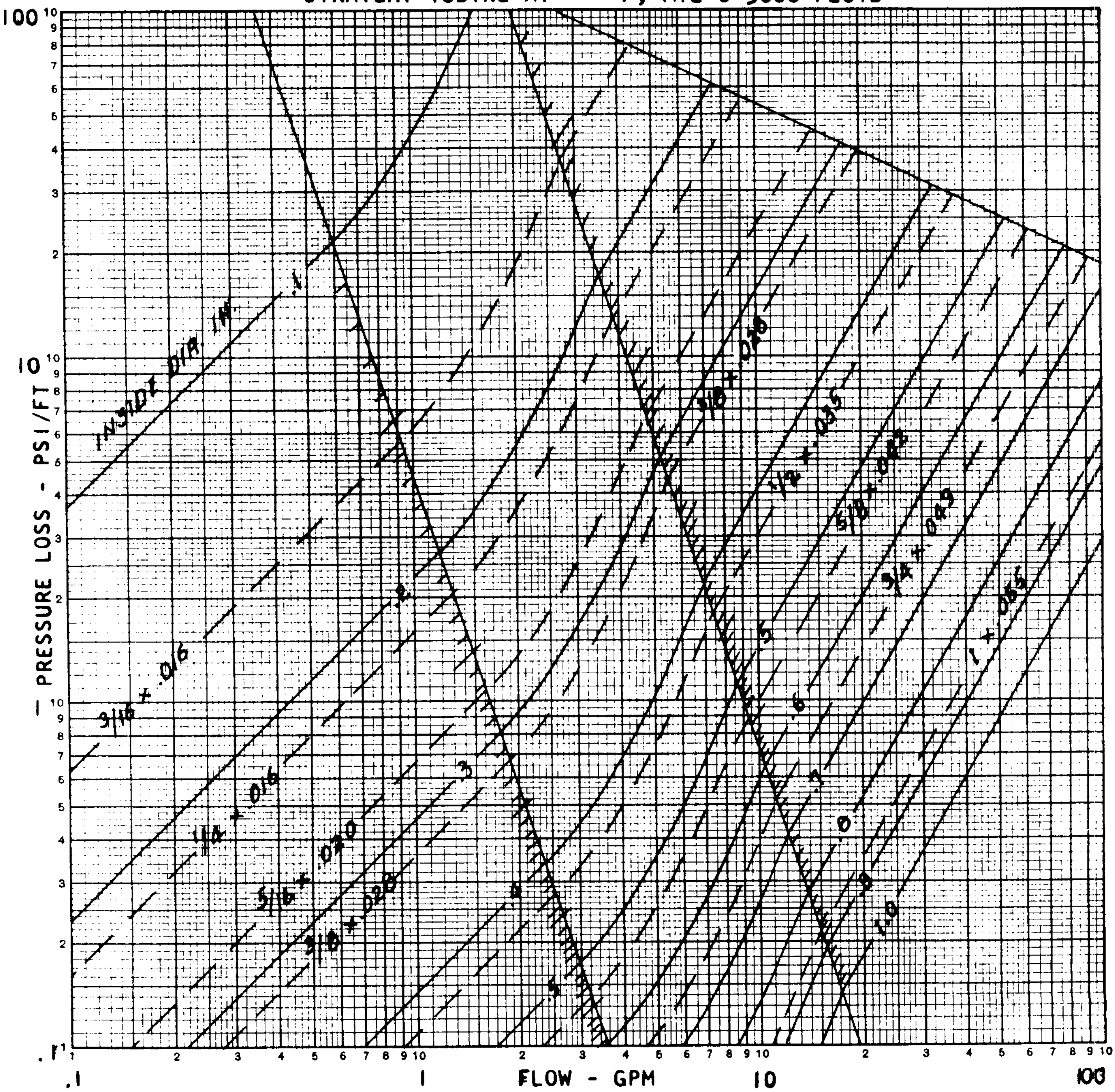
10.532 (CONT'D)

PRESSURE LOSSES IN TUBING



(3)

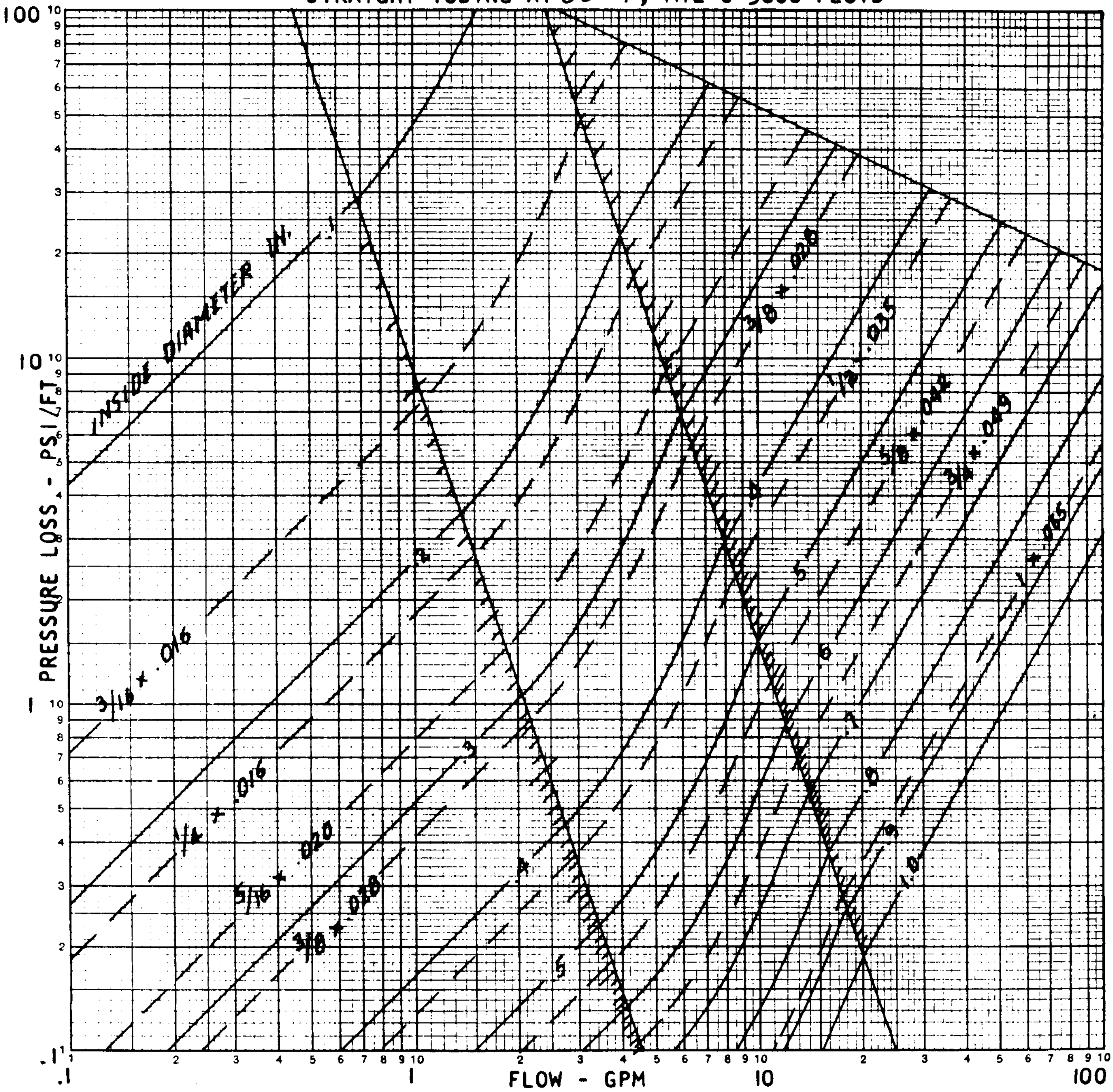
STRAIGHT TUBING AT 90 °F, MIL-O-5606 FLUID



10.532 (CONT'D)

PRESSURE LOSSES IN TUBING

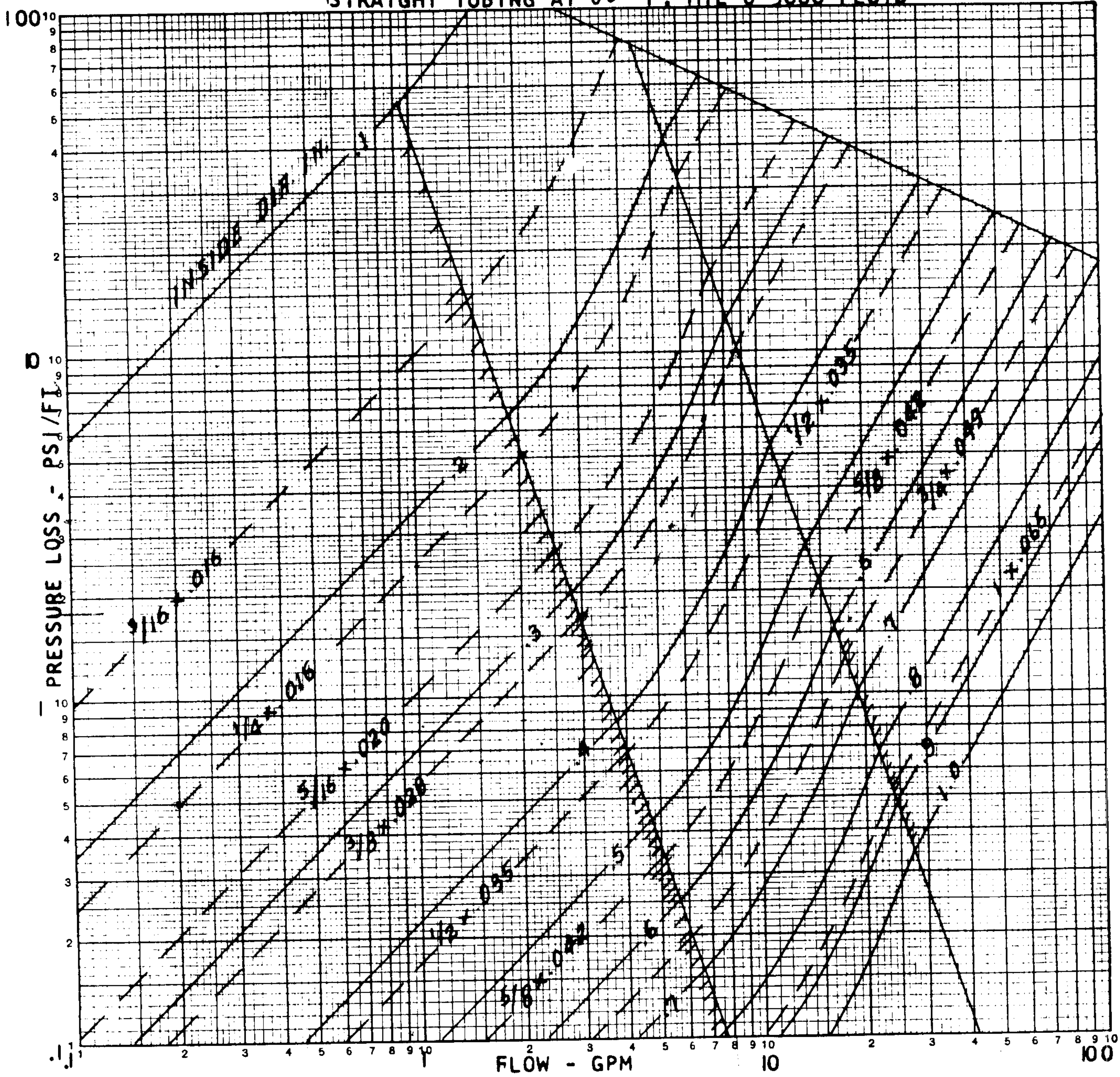
STRAIGHT TUBING AT 80 °F, MIL-0-5606 FLUID



10.532 (CONT'D)

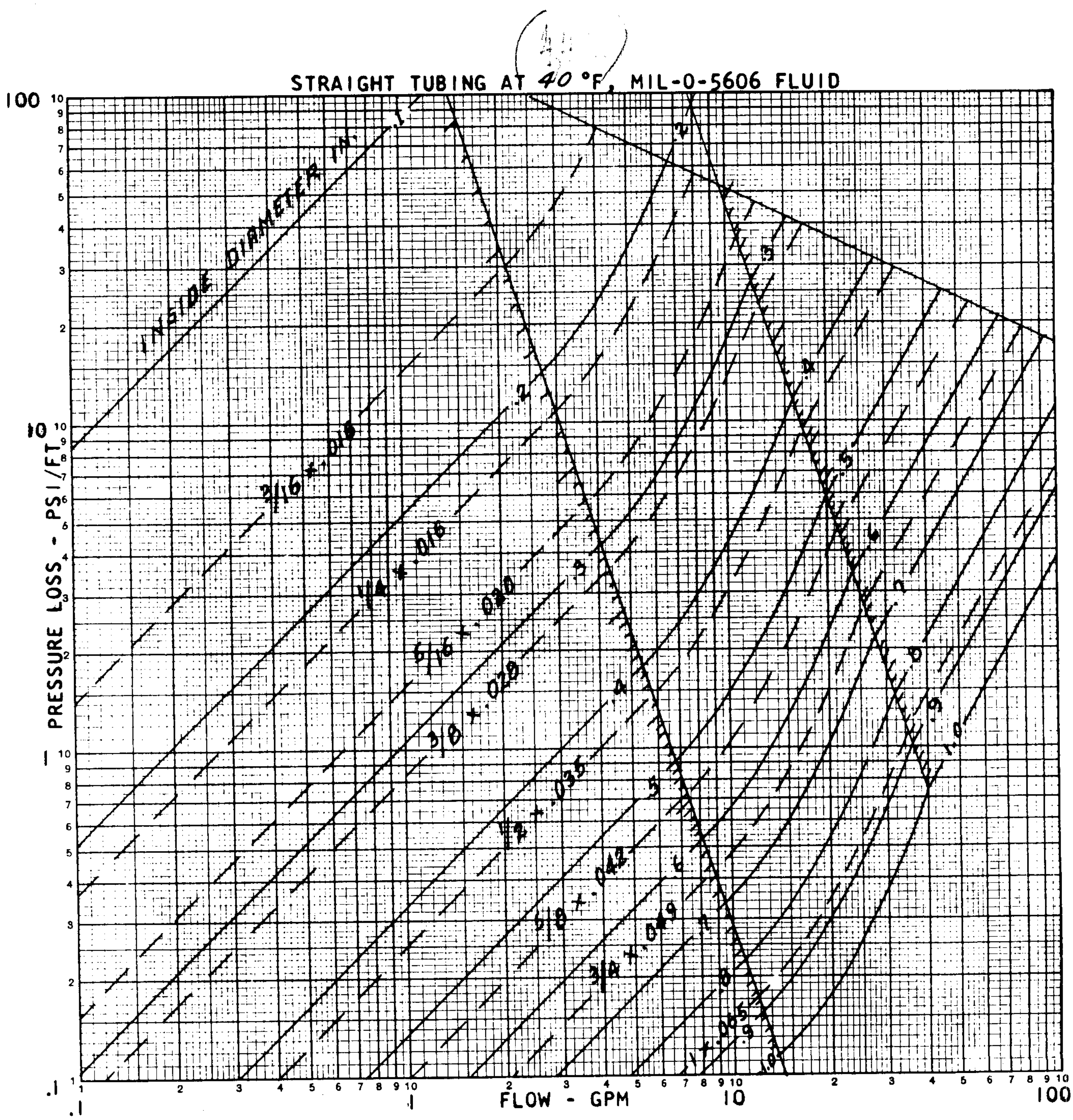
PRESSURE LOSSES IN TUBING

STRAIGHT TUBING AT 60 °F, MIL-0-5606 FLUID



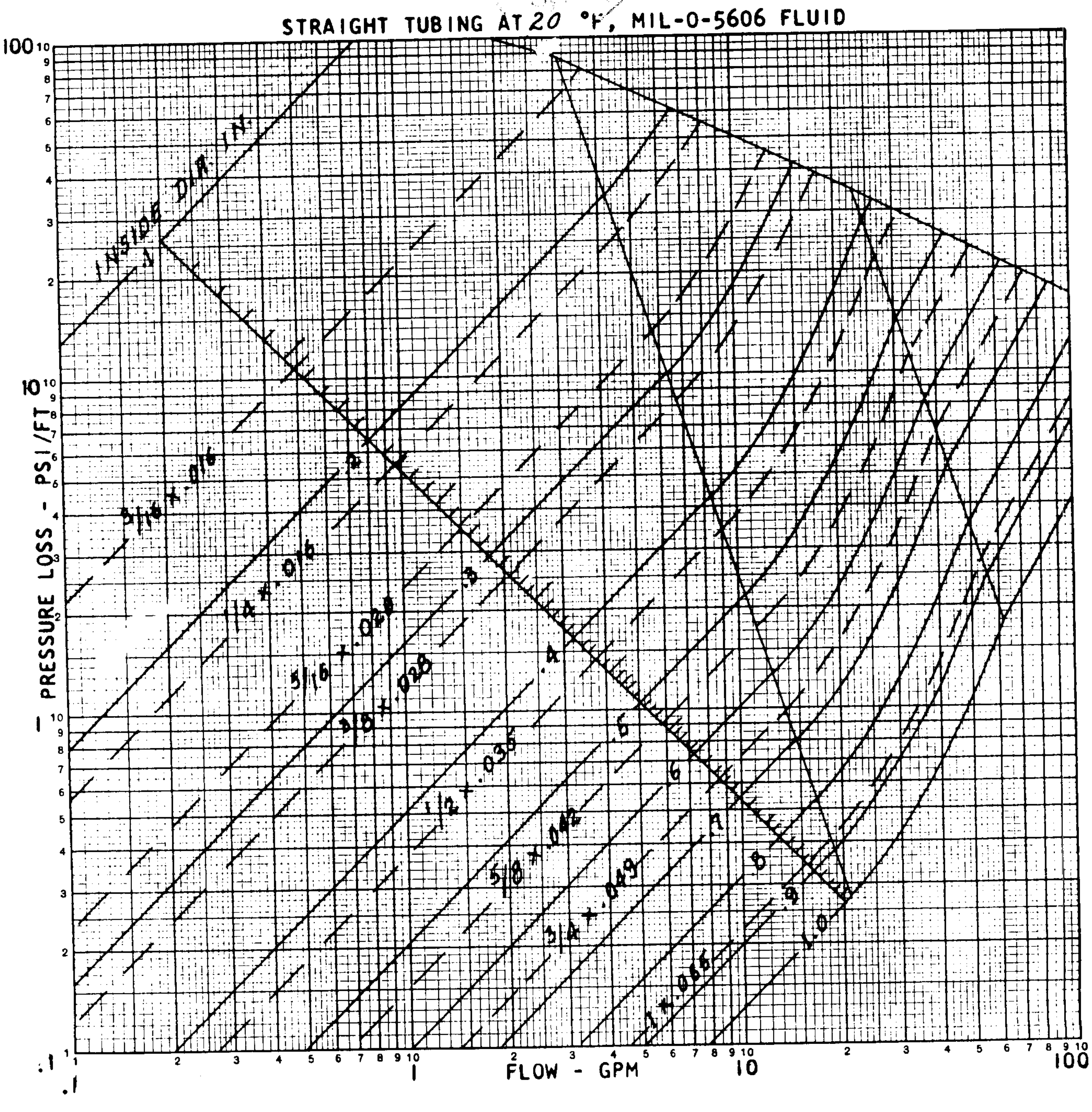
10.532 (CONT'D)

PRESSURE LOSSES IN TUBINE



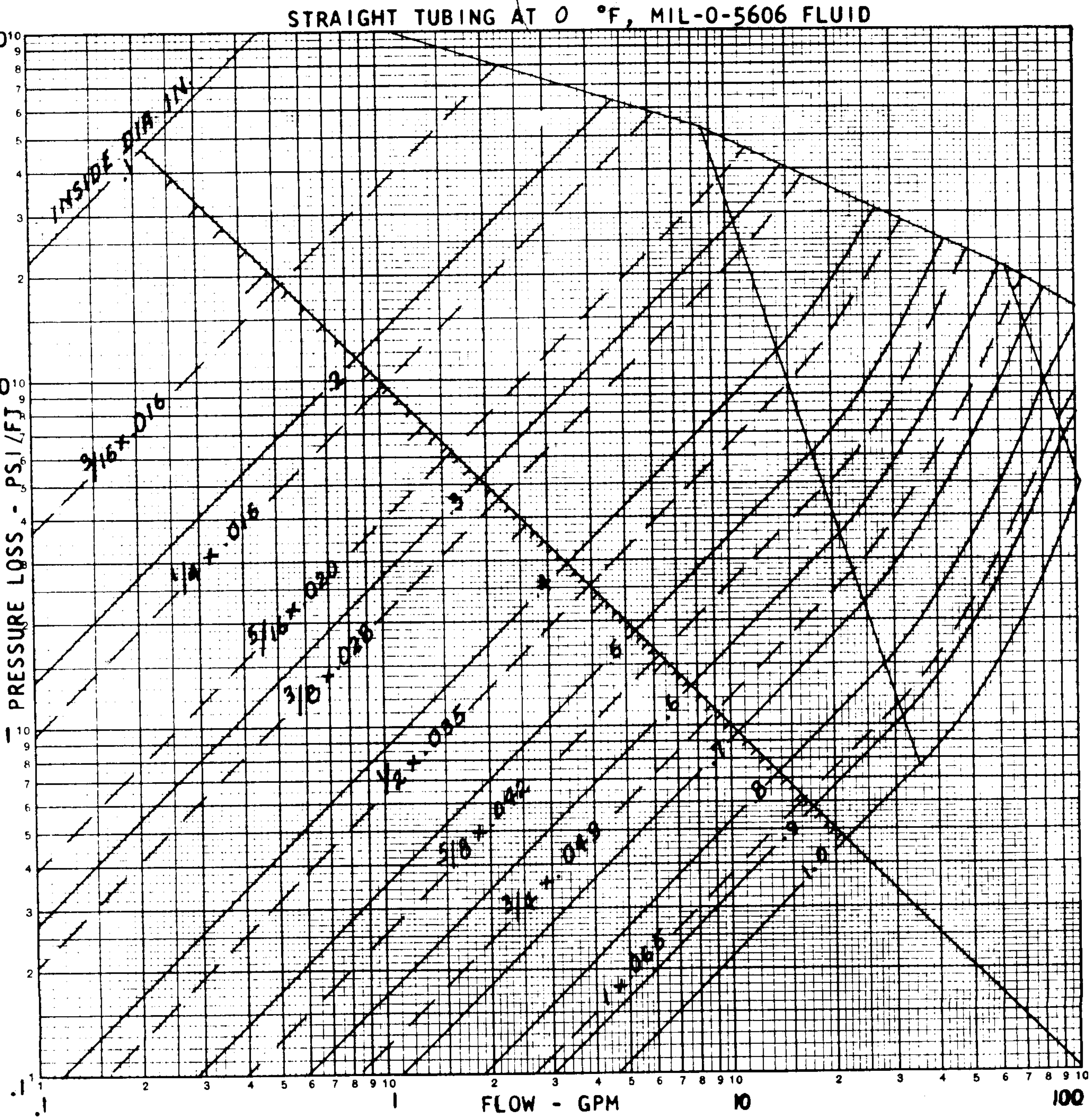
10.532 (CONT'D)

PRESSURE LOSSES IN TUBING



10.532 (CONT'D)

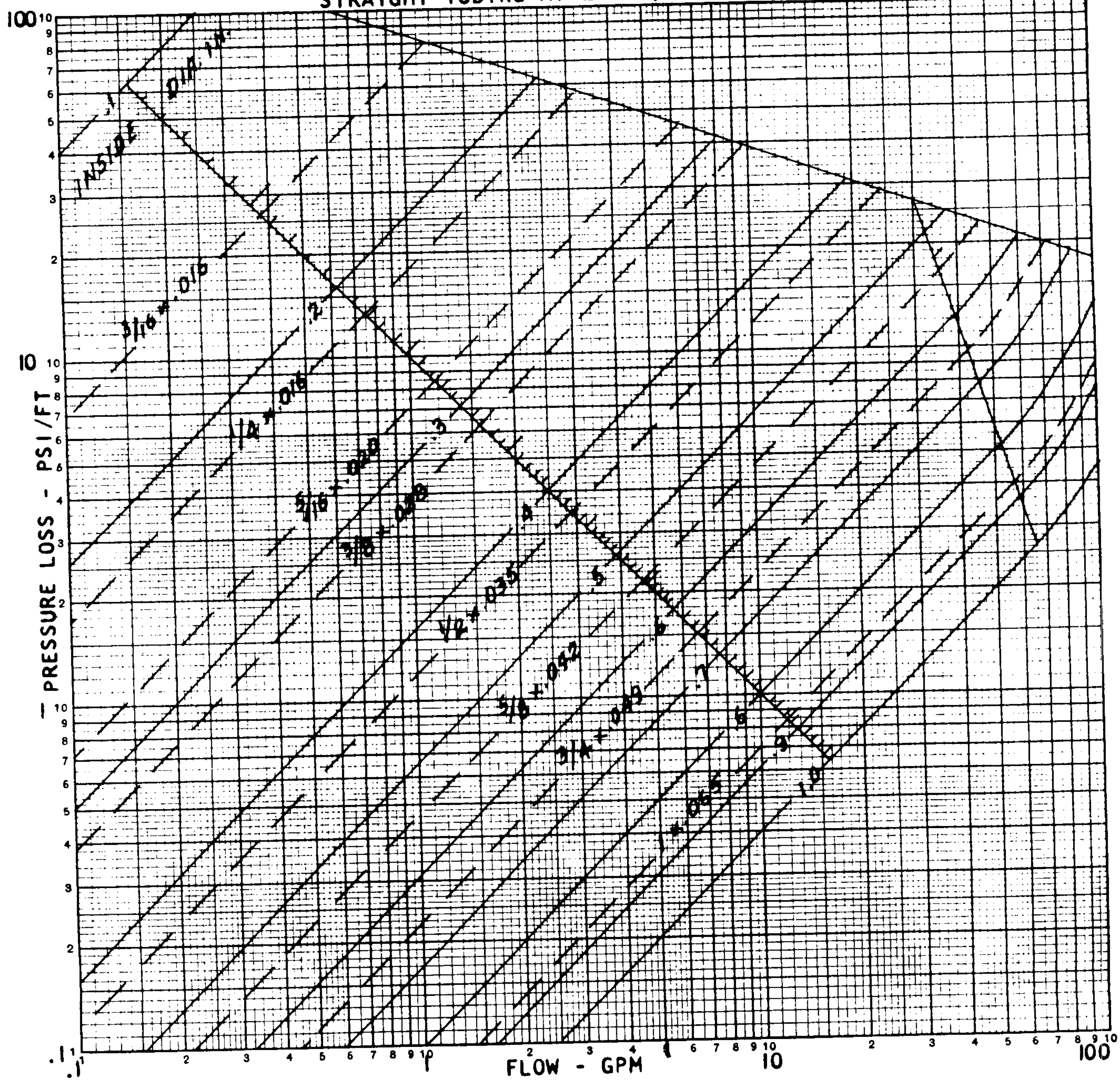
PRESSURE LOSSES IN TUBING



10.532 (CONT'D)

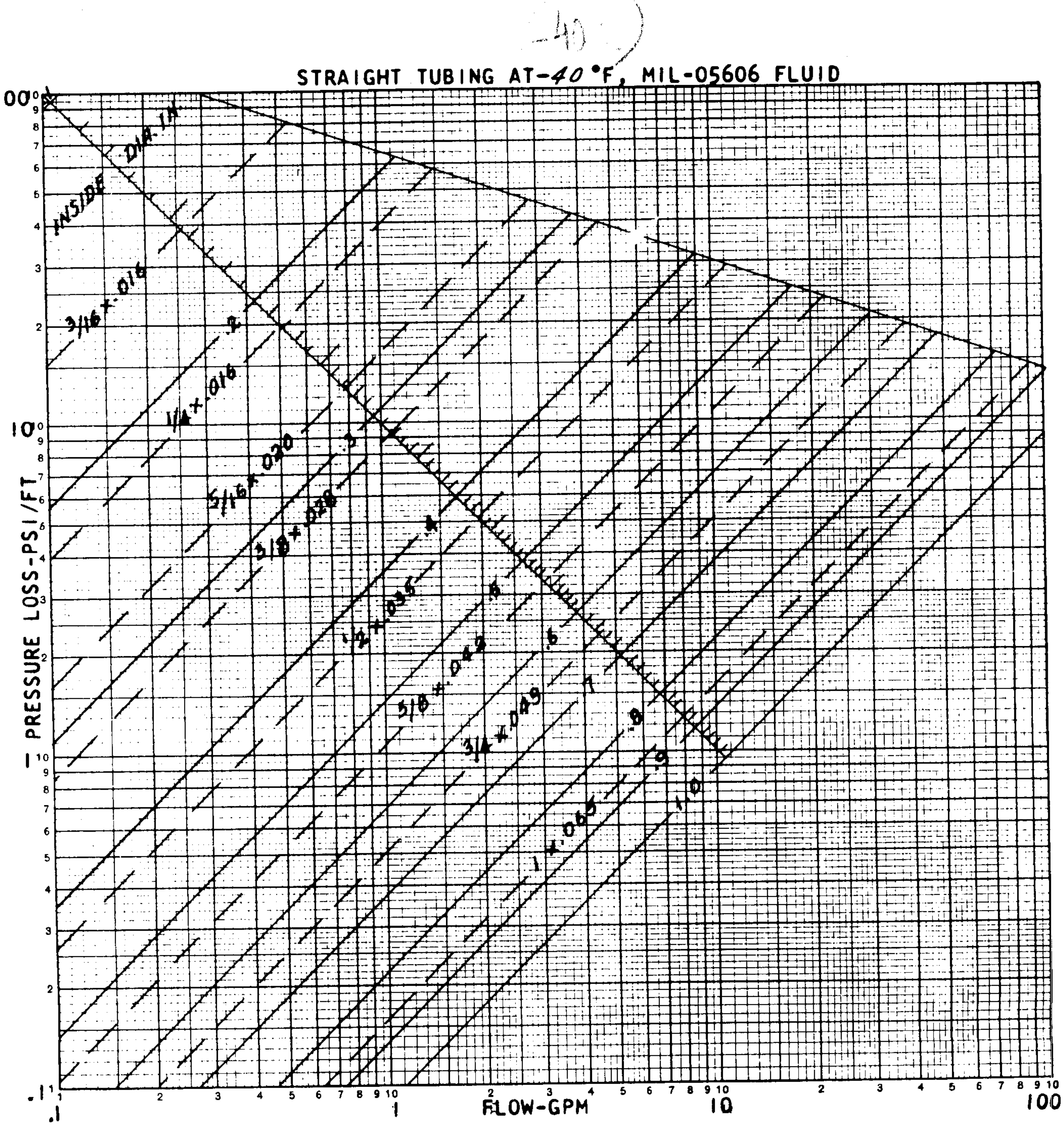
PRESSURE LOSSES IN TUBING

STRAIGHT TUBING AT -20°F, MIL-0-5606 FLUID



10.532 (CONT'D)

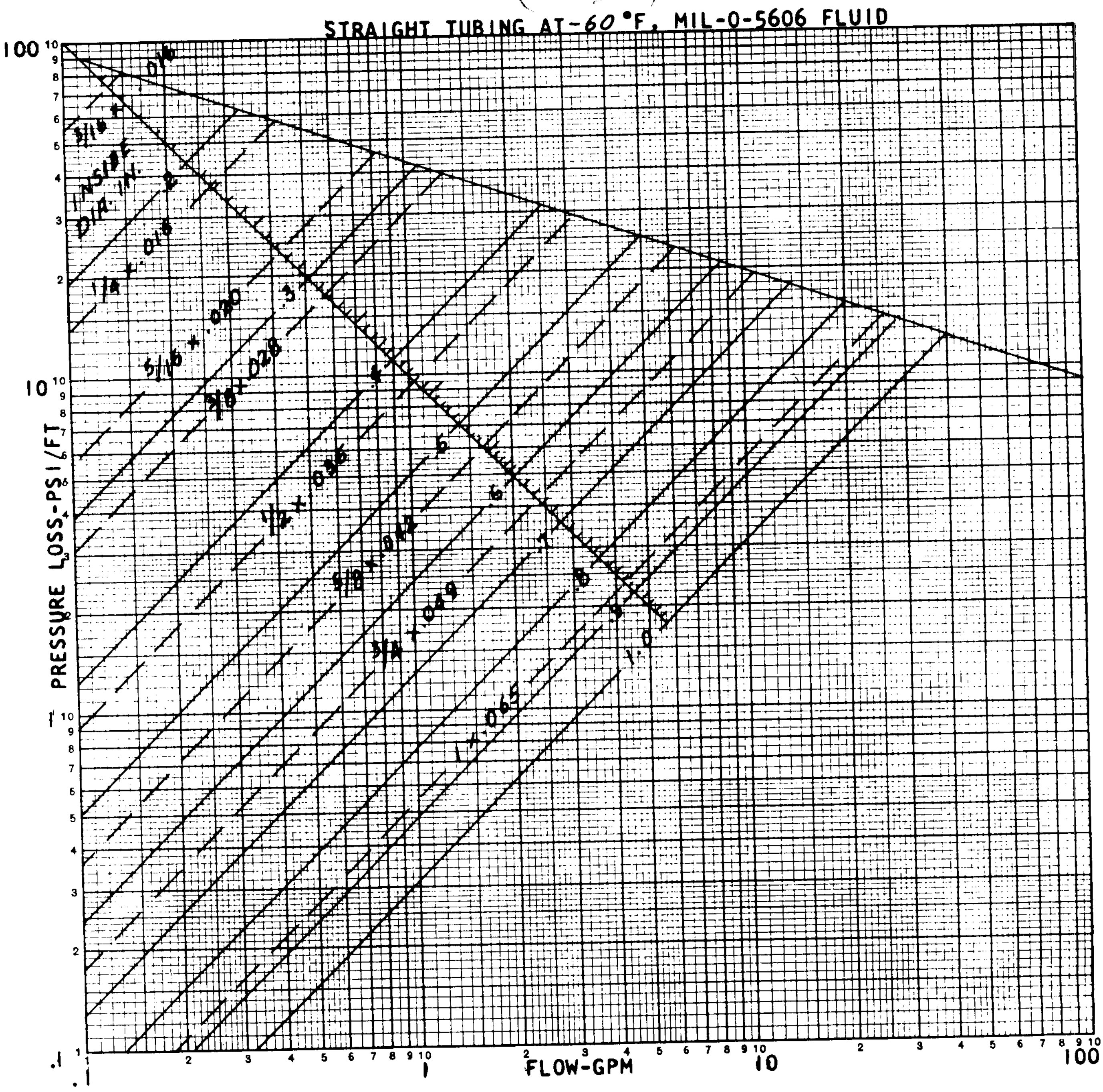
PRESSURE LOSSES IN TUBING

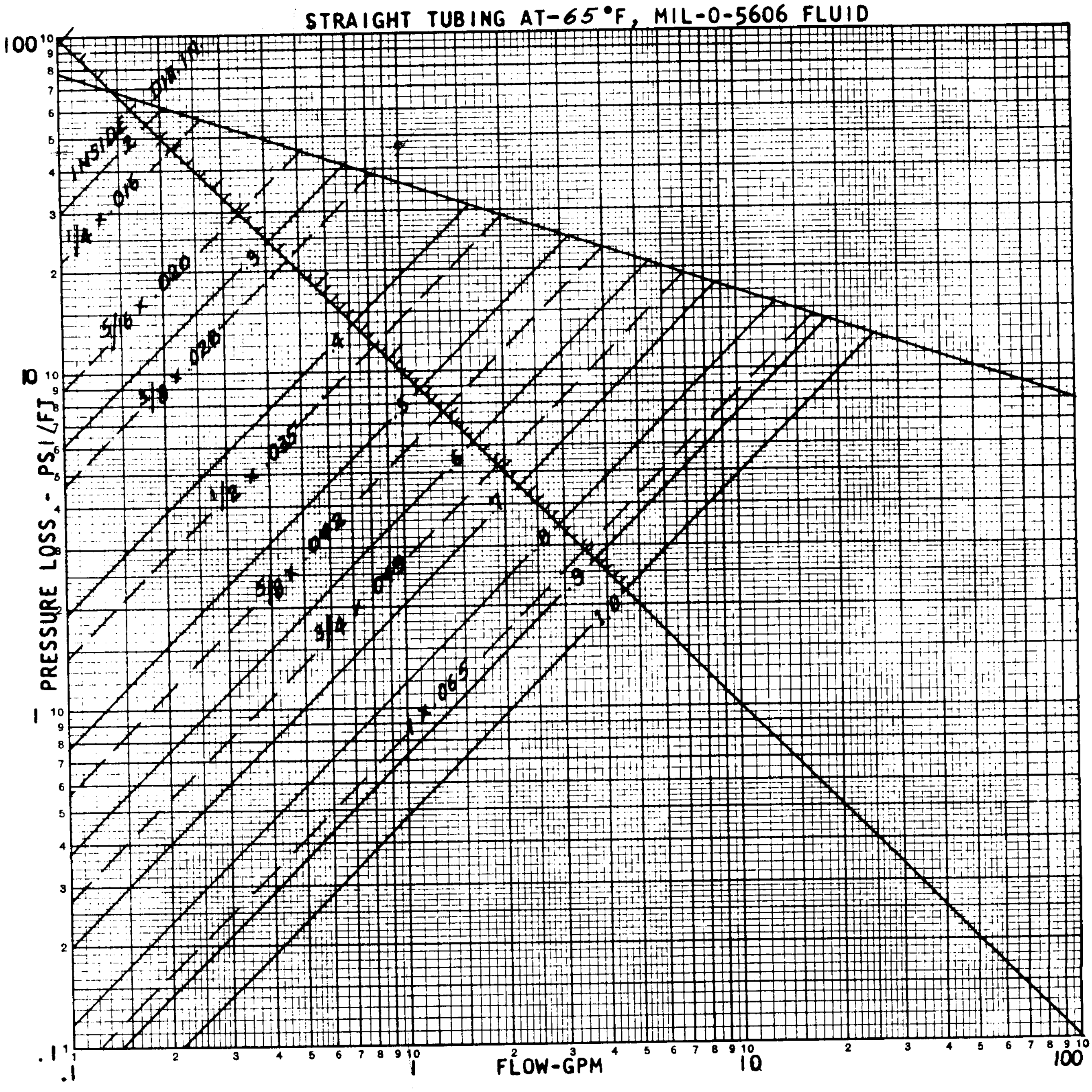


10.532 (CONT'D)

PRESSURE LOSSES IN TUBING

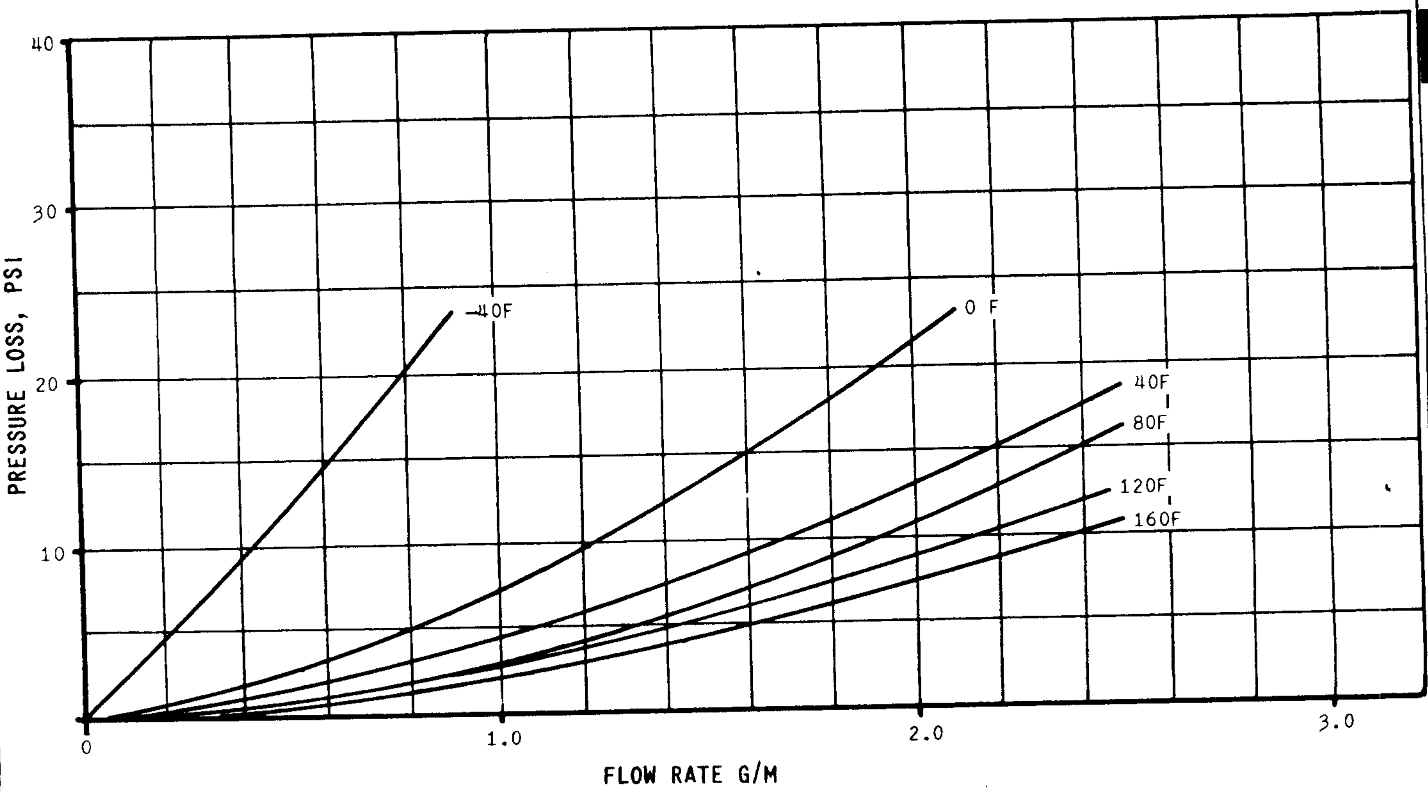
(-60°)



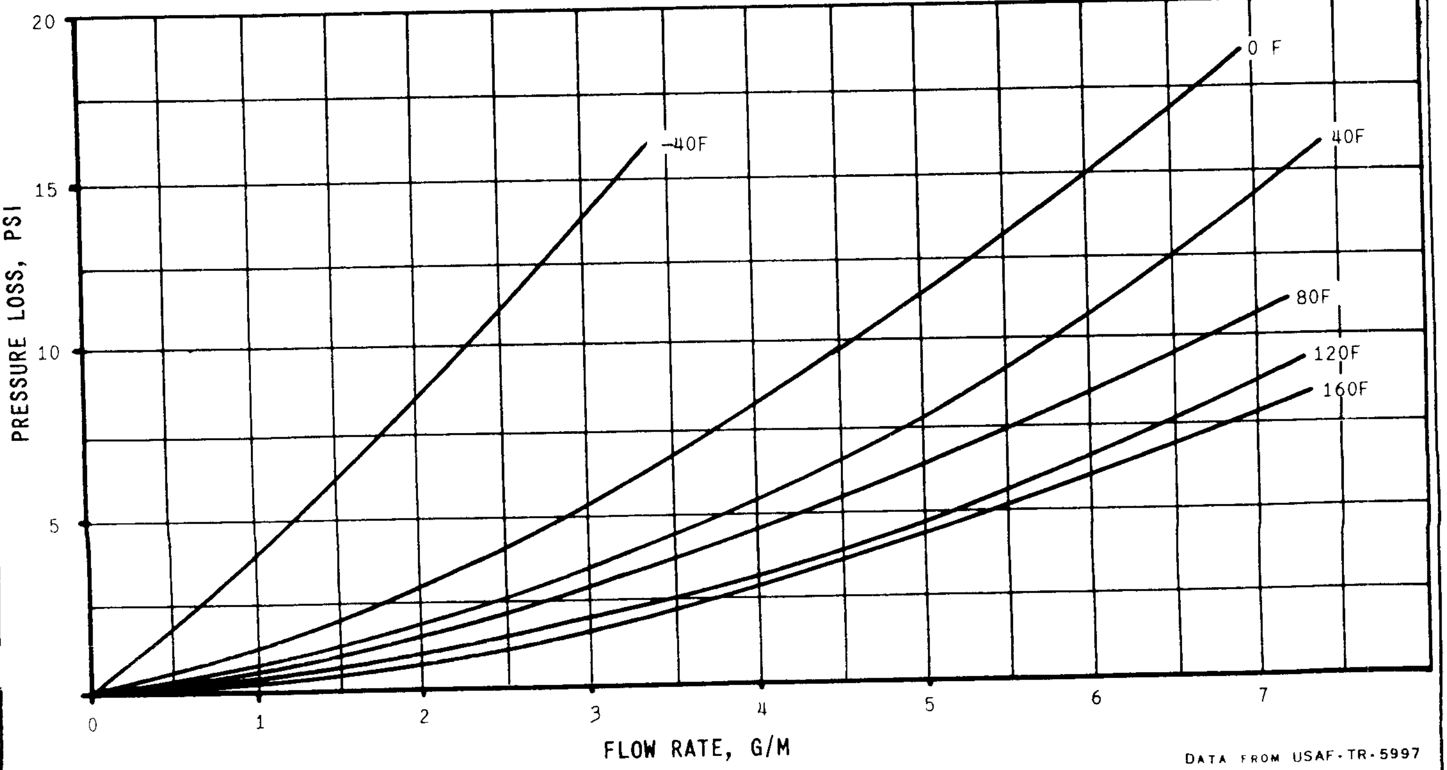

STRAIGHT TUBING AT-65°F, MIL-0-5606 FLUID
PRESSURE LOSS - PSI/FT
FLOW-GPM
INSIDE DIA IN.
1/4 x .016
5/16 x .020
3/8 x .028
1/2 x .035
5/8 x .042
3/4 x .049
1 x .065
.1
1
10
100

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID

AN821-4 ELBOW



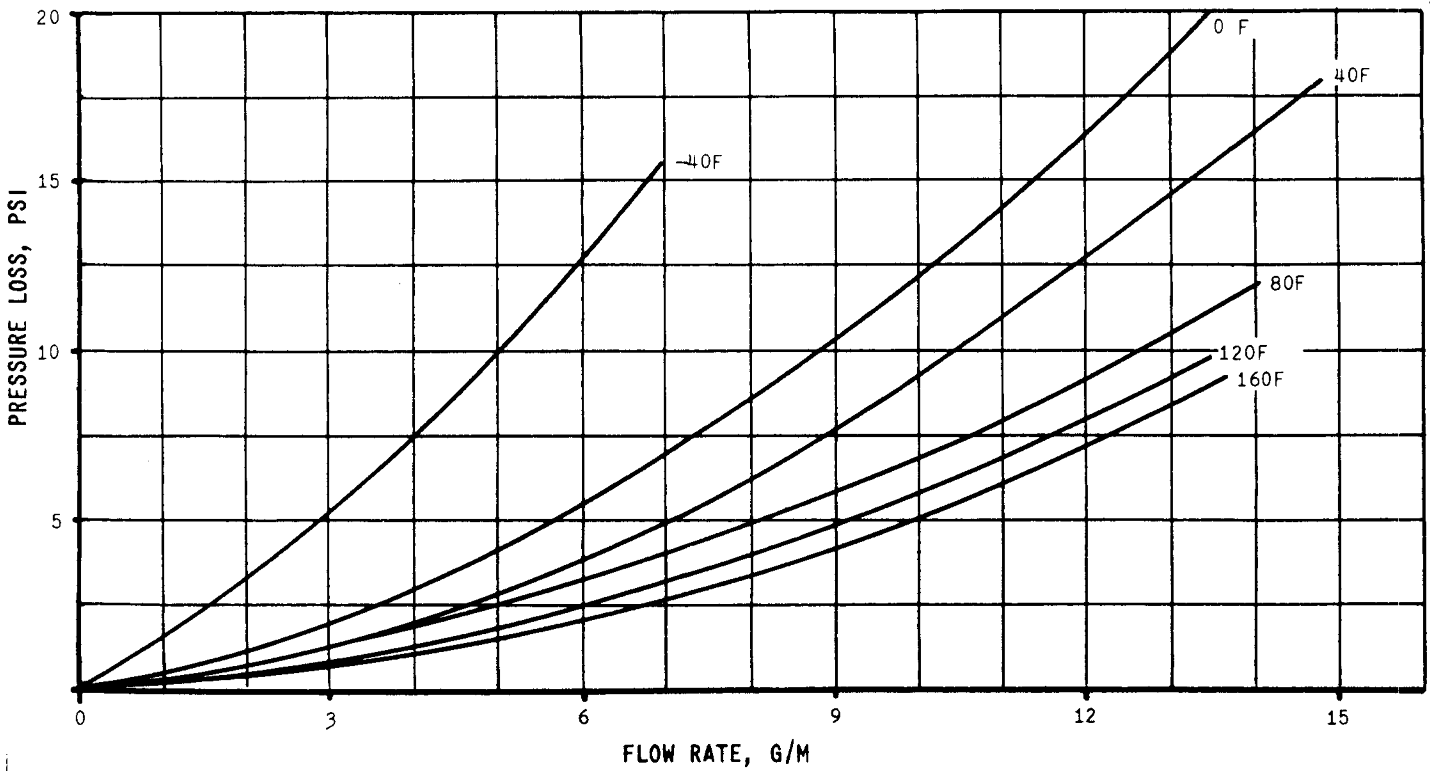
AN821-6 ELBOW



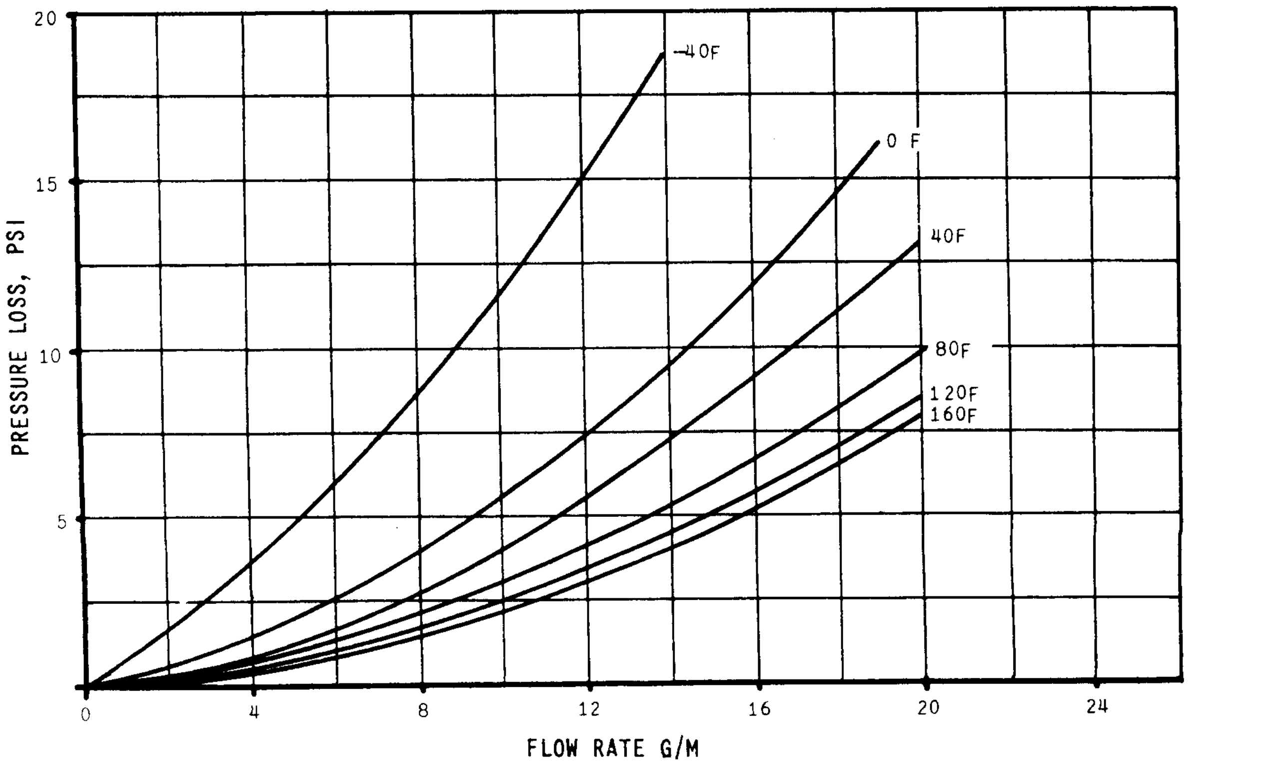
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN821-8 ELBOW



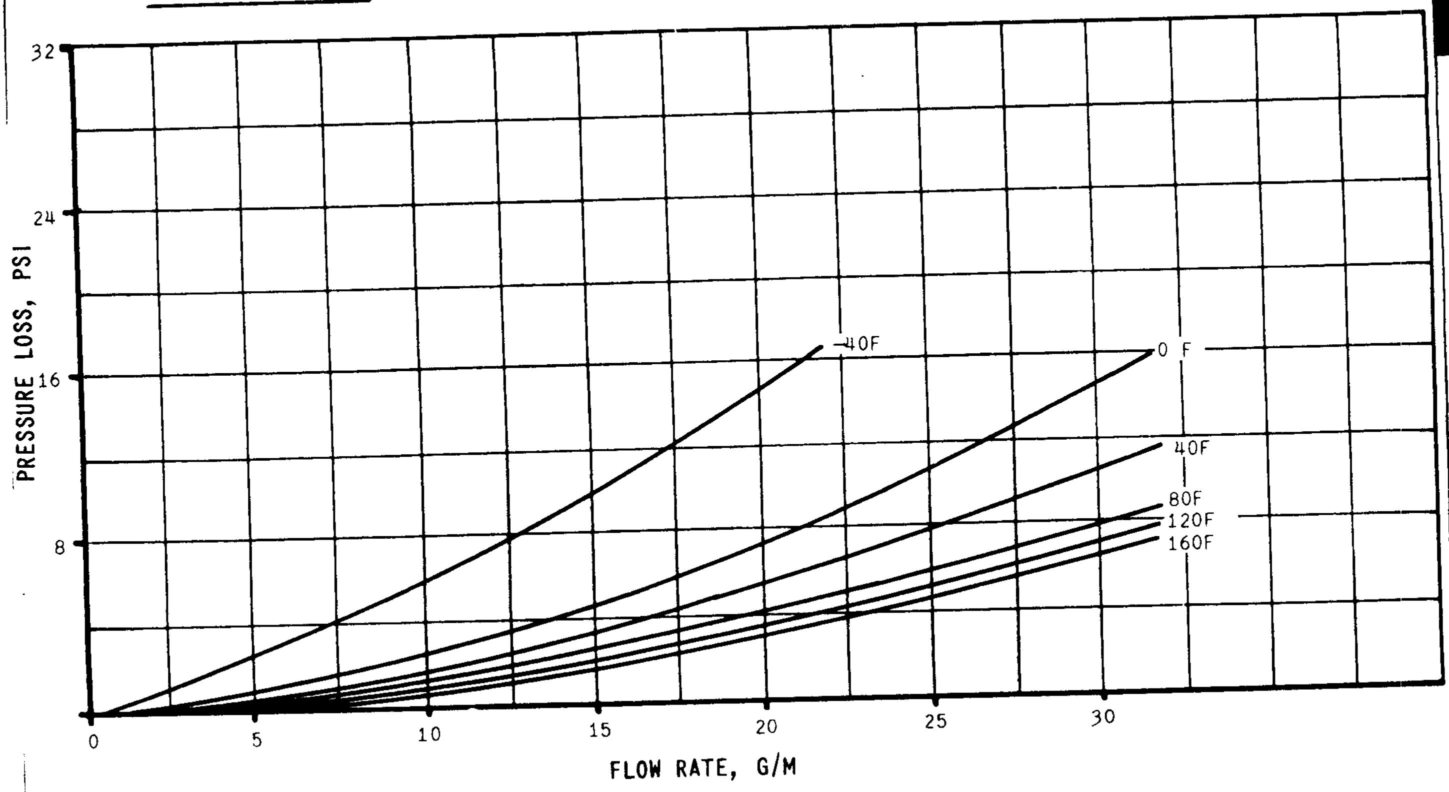
AN821-10 ELBOW



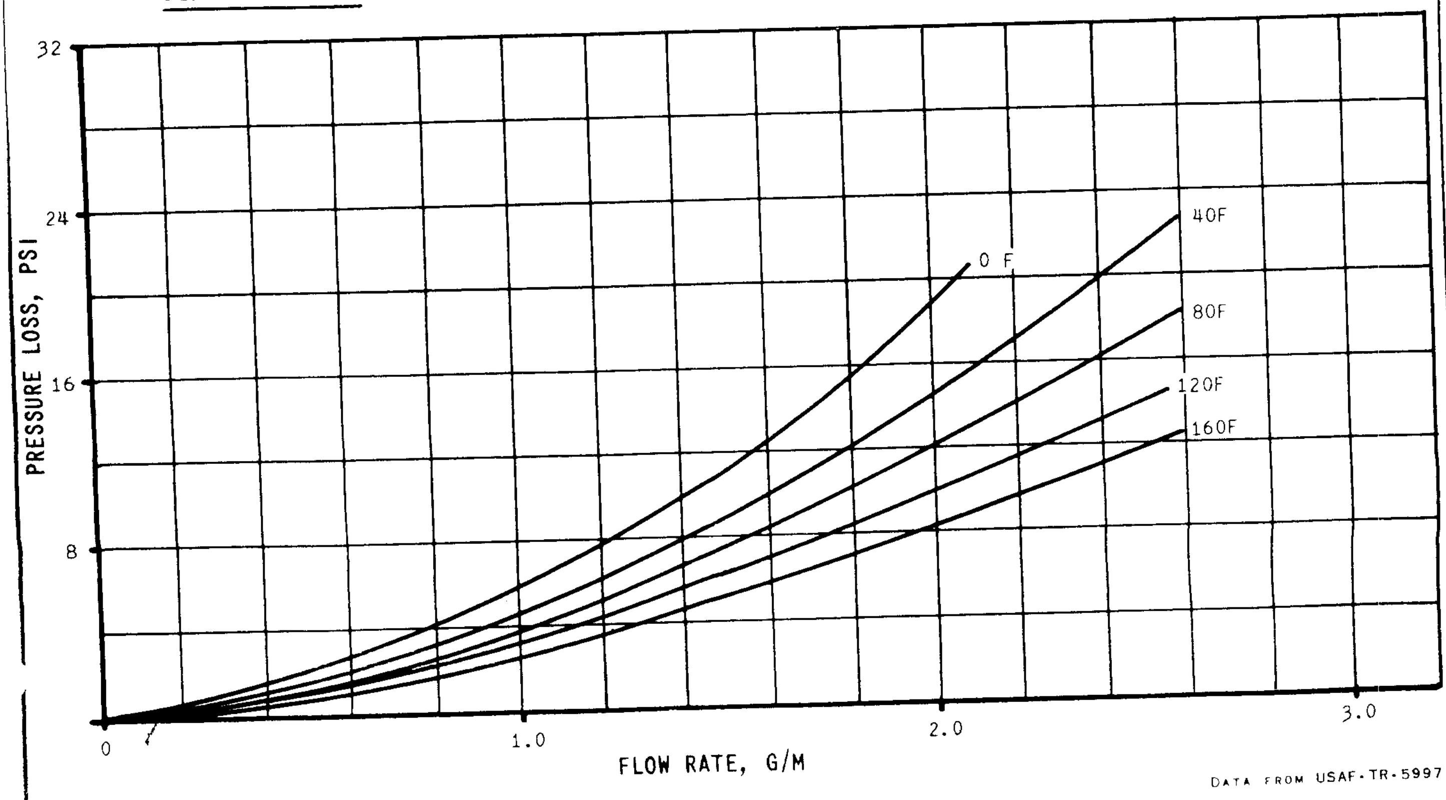
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN821-12 ELBOW



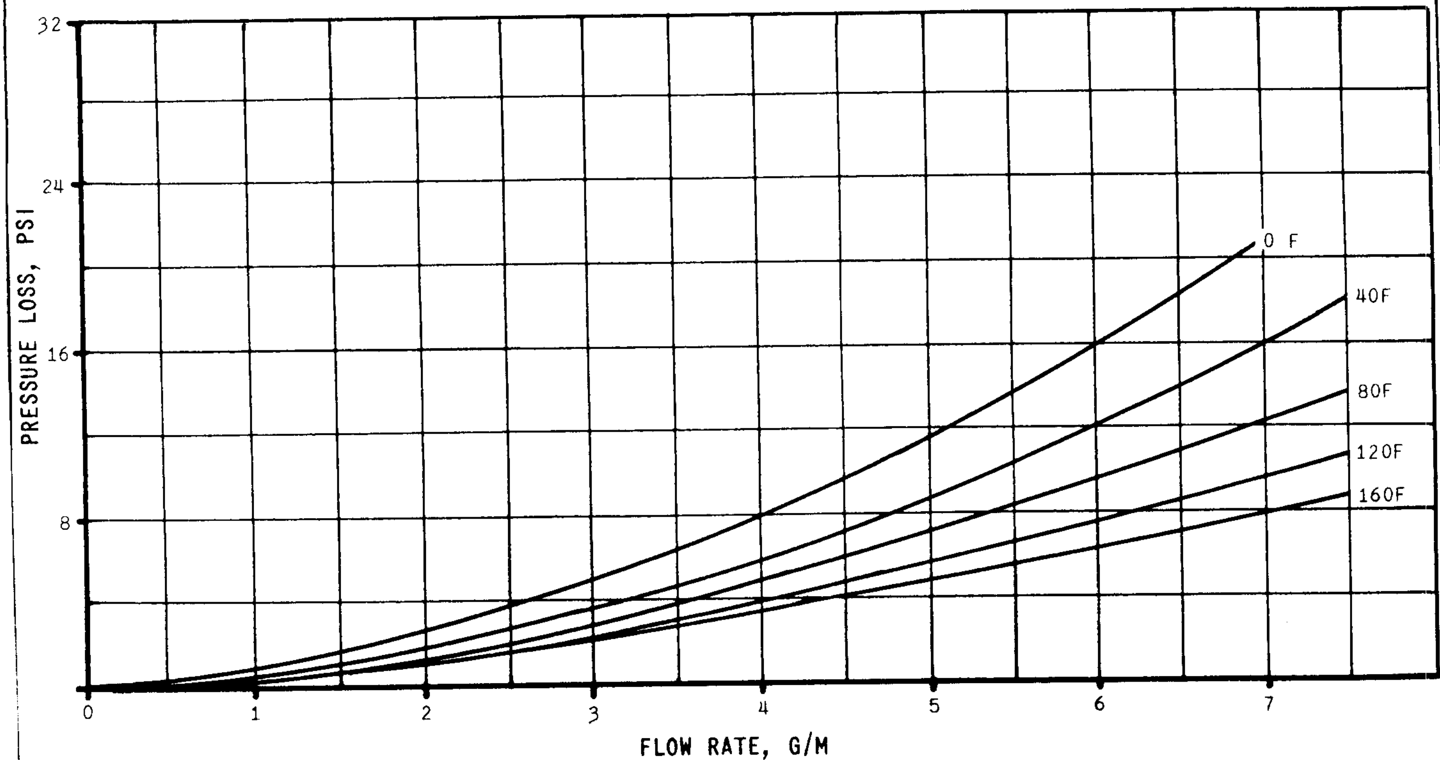
AN824-4 TEE



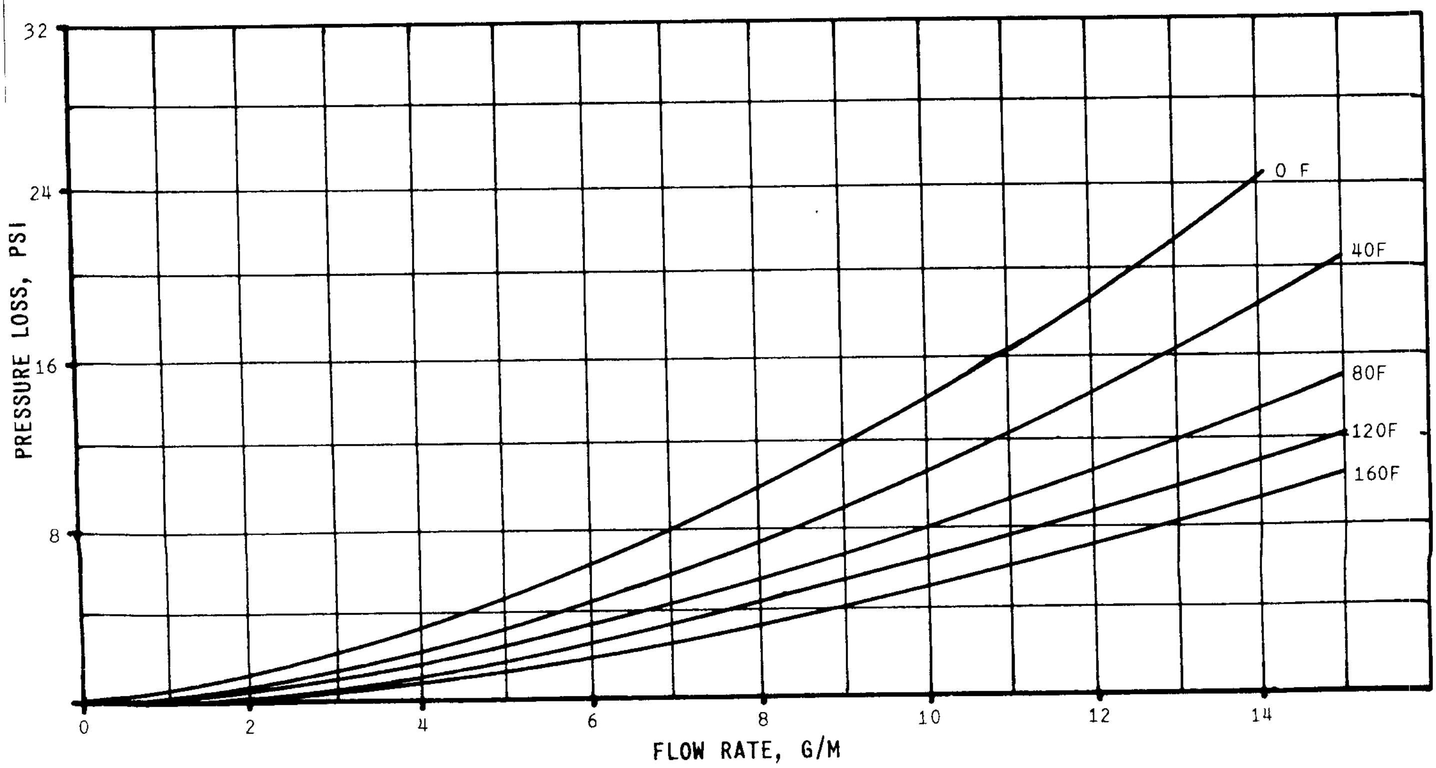
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN824-6 TEE



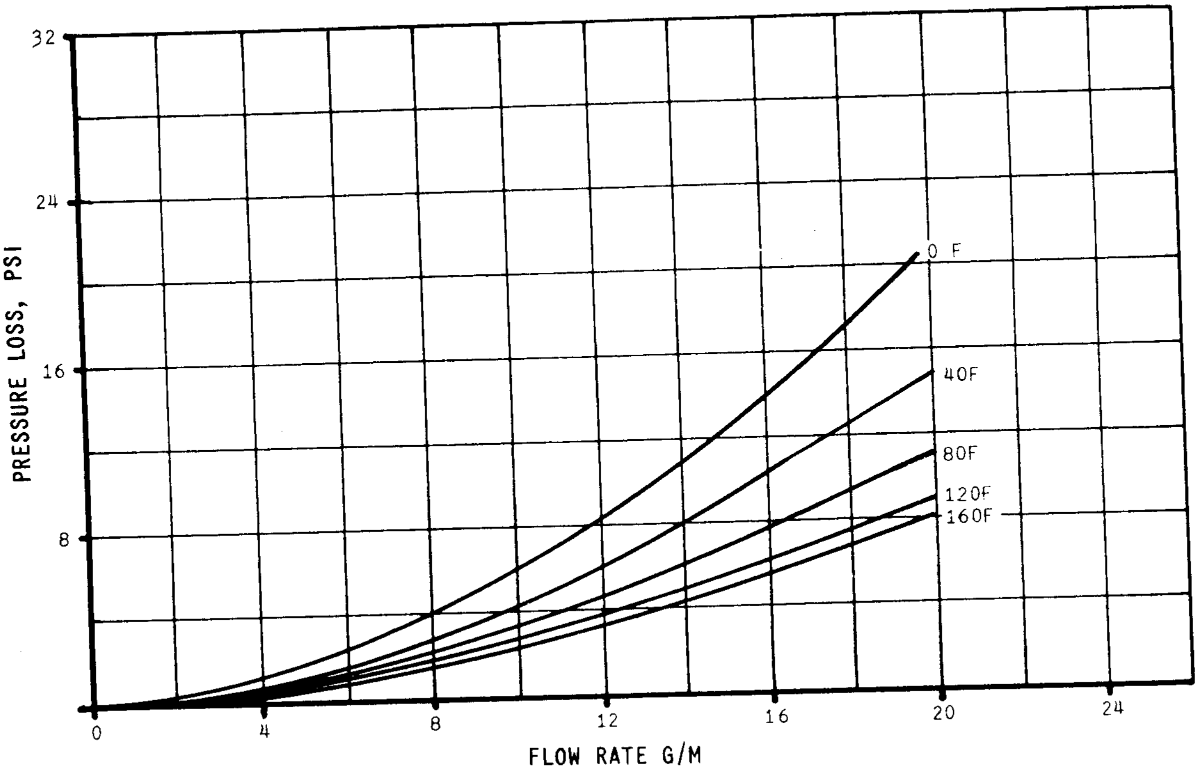
AN824-8 TEE



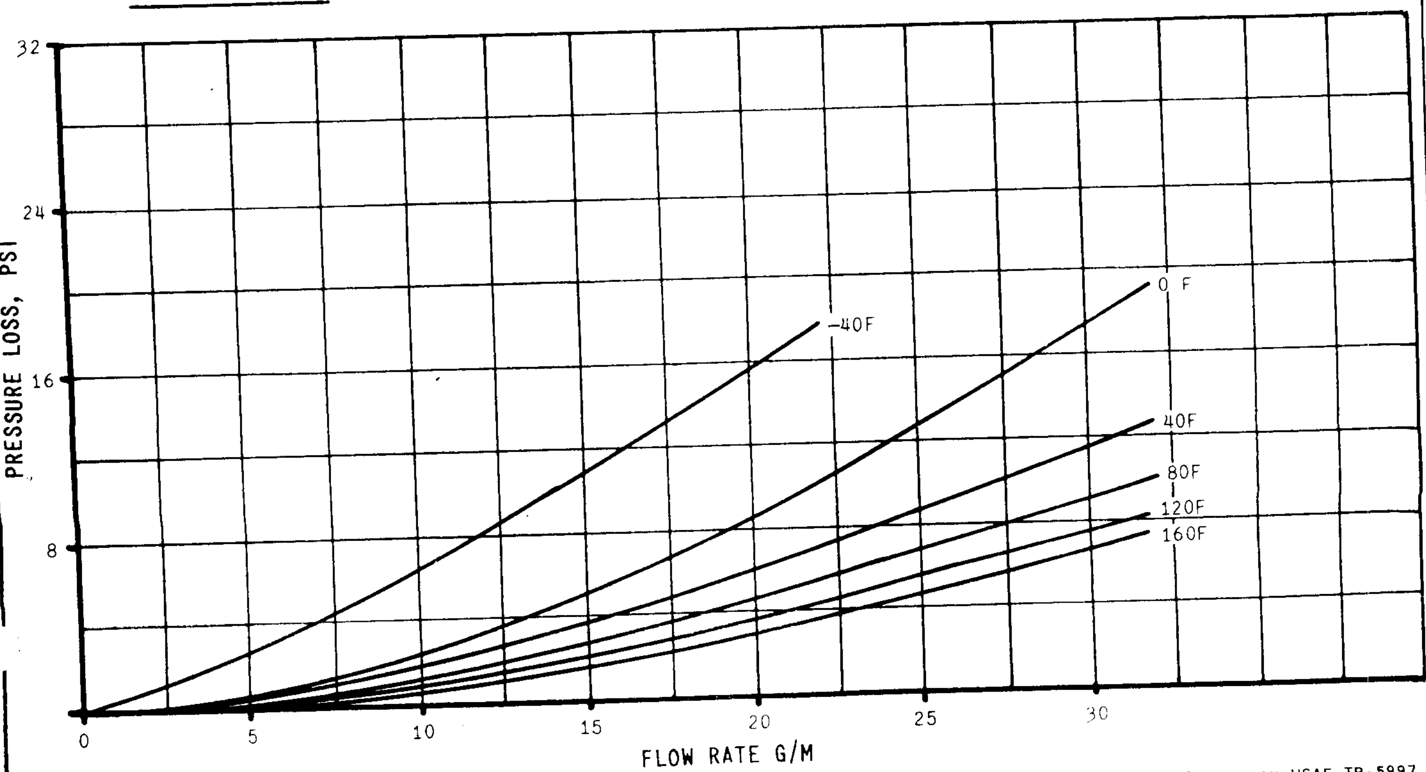
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN824-10 TEE



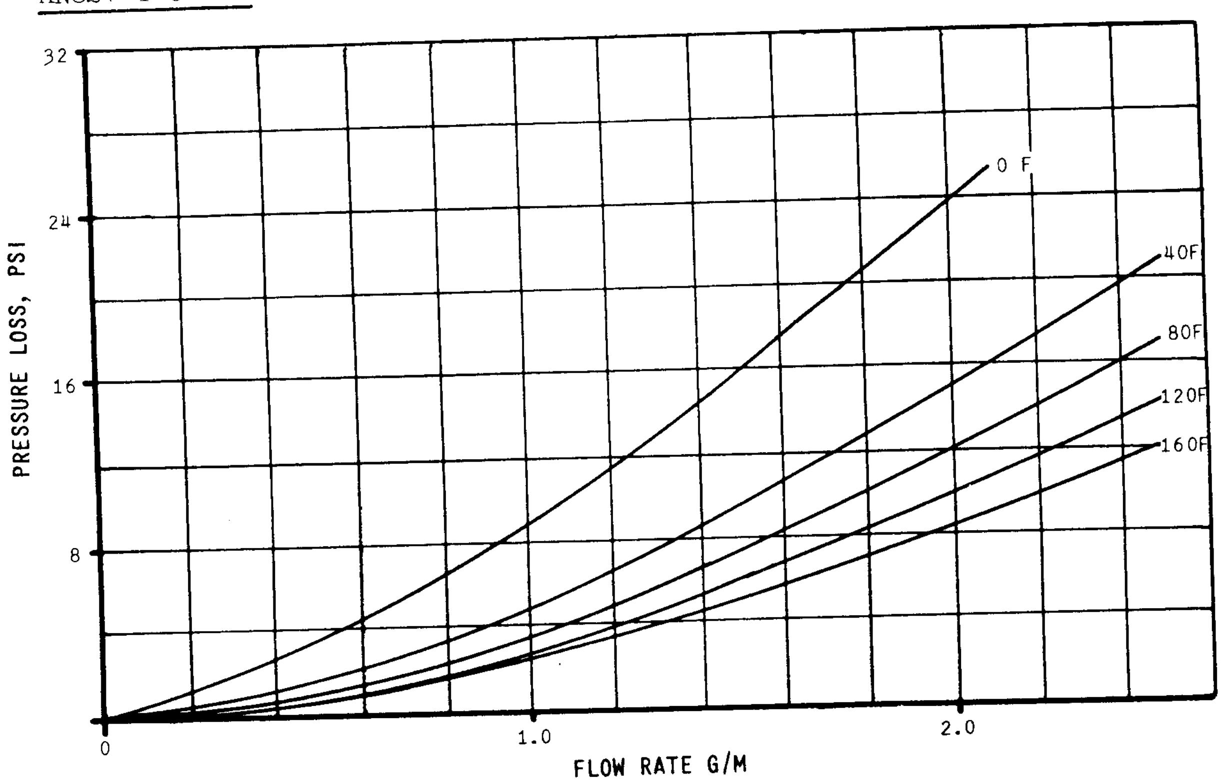
AN824-12 TEE



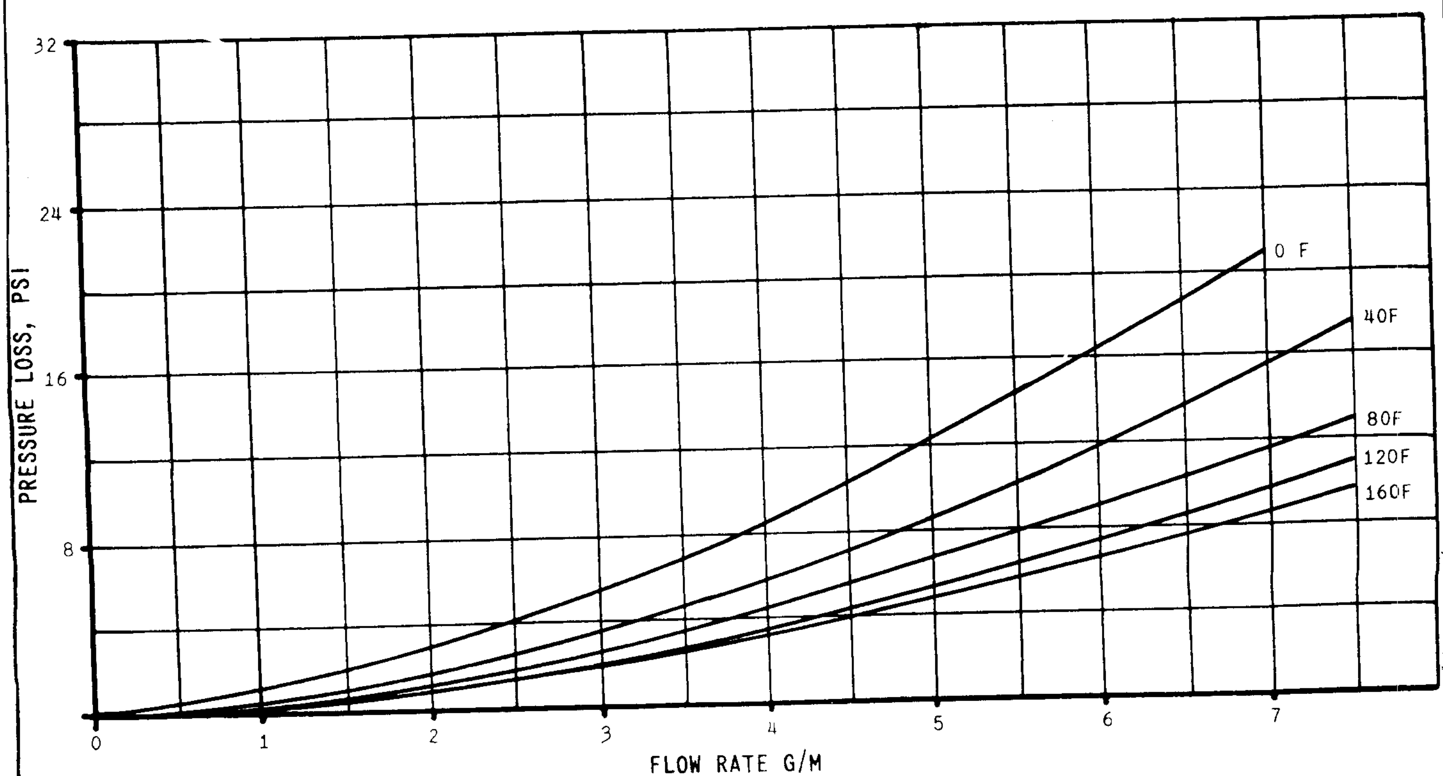
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN827-4 CROSS



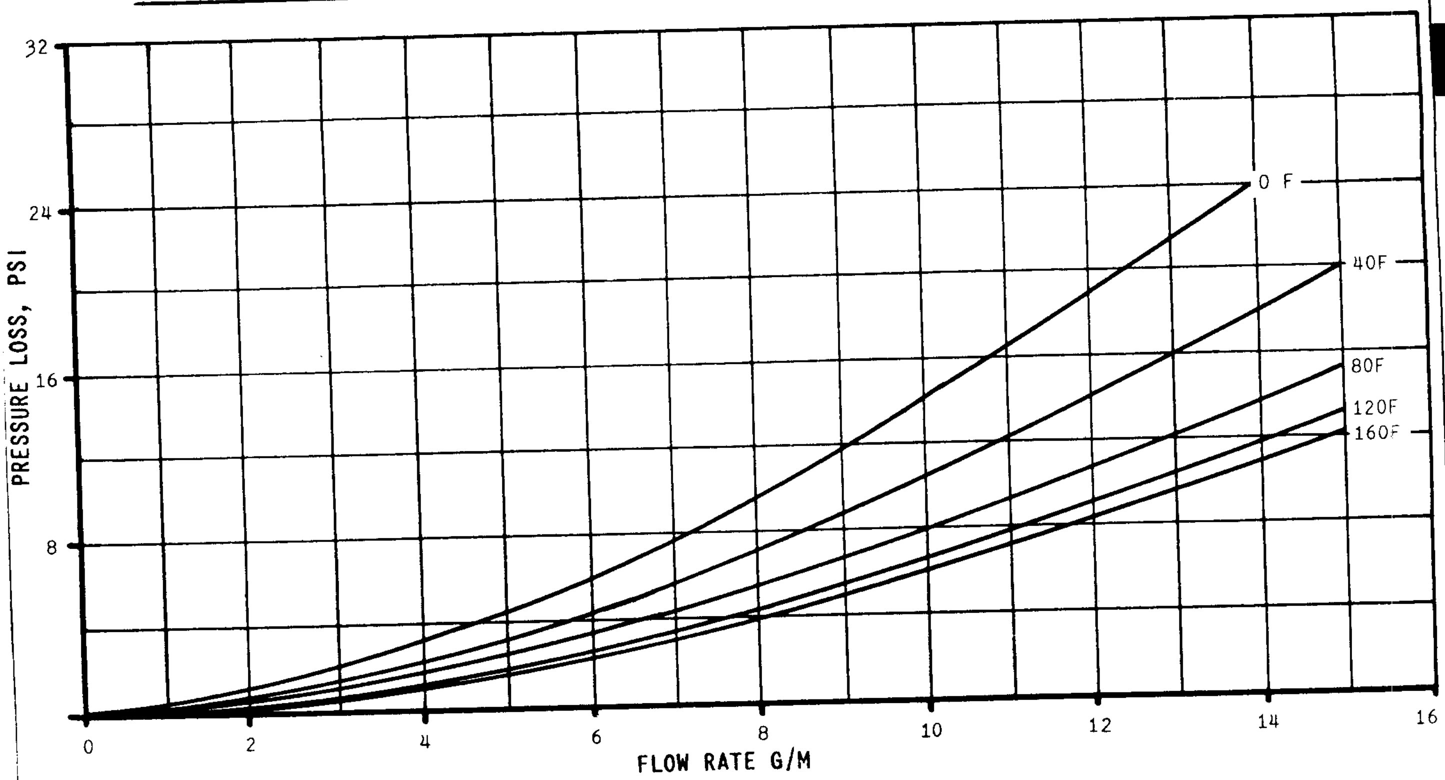
AN827-6 CROSS



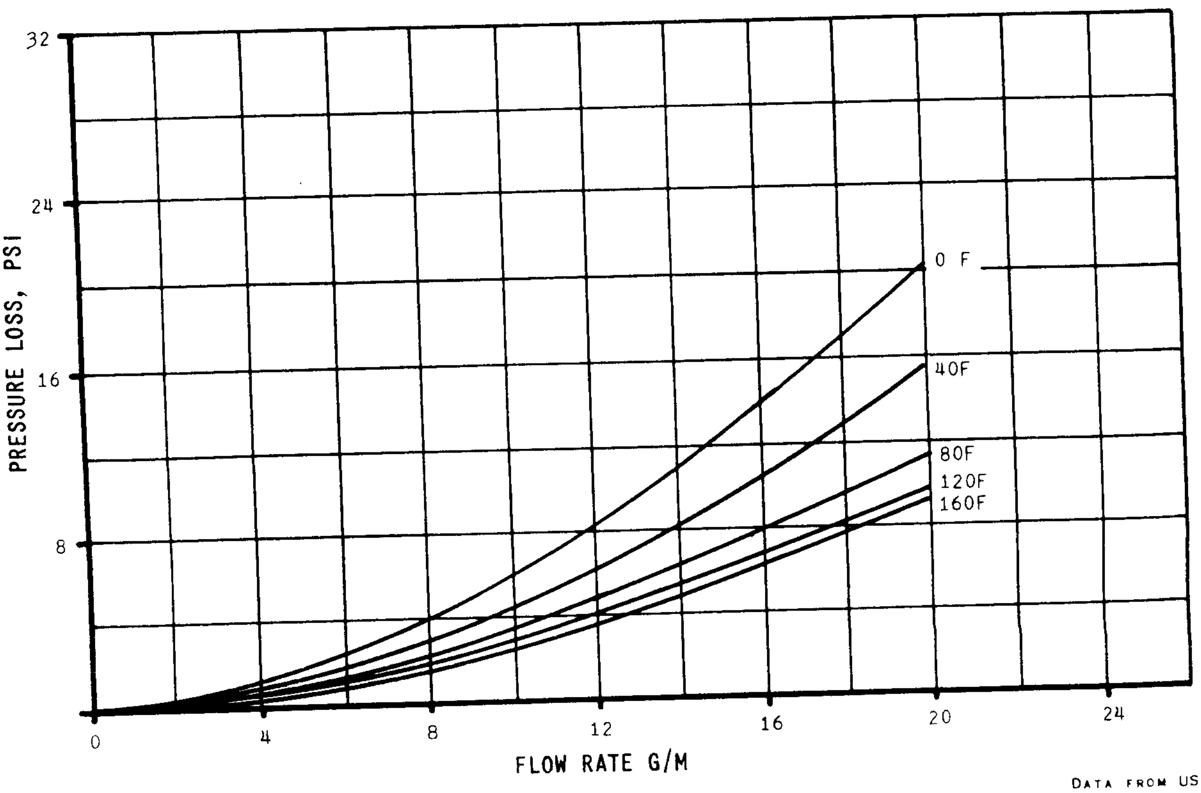
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN827-8 CROSS



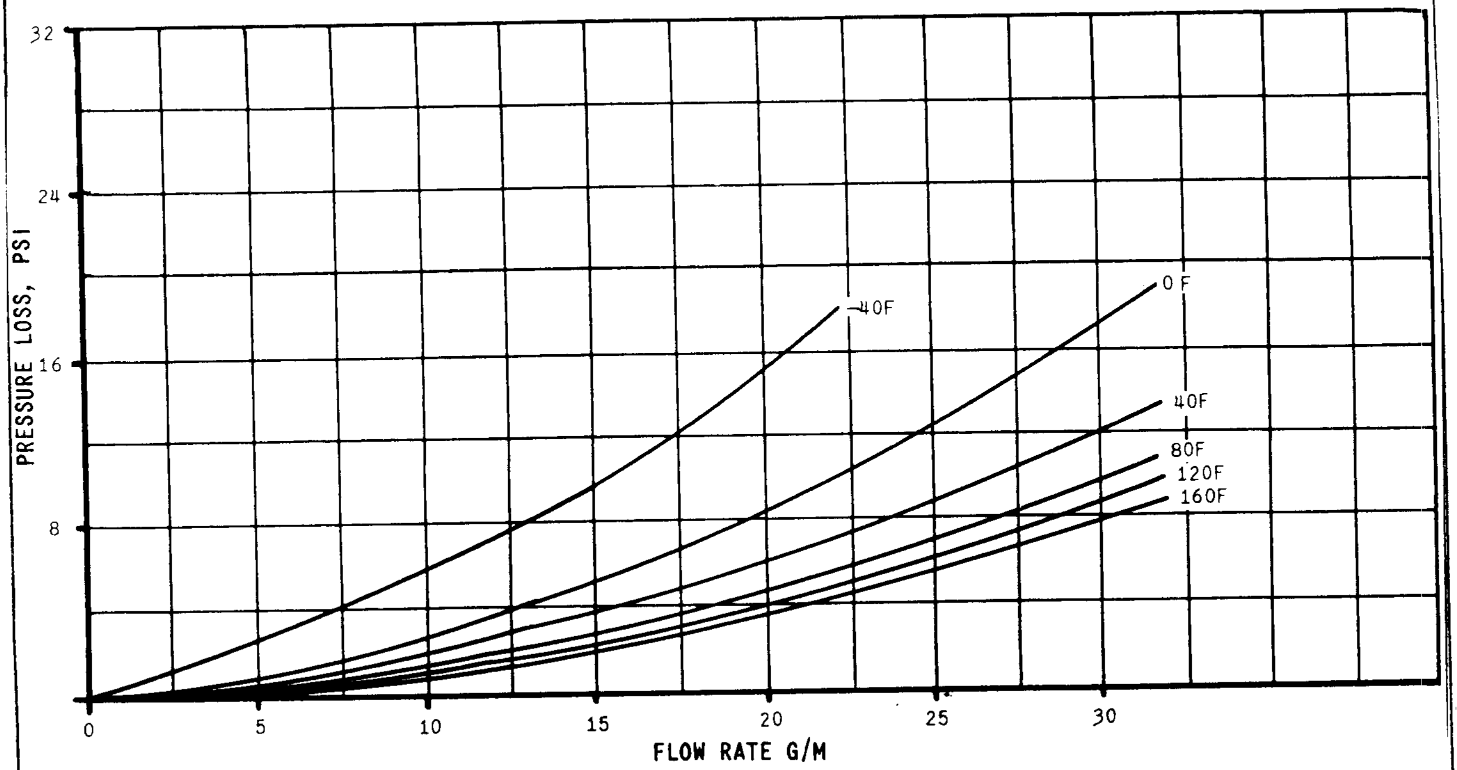
AN827-10 CROSS



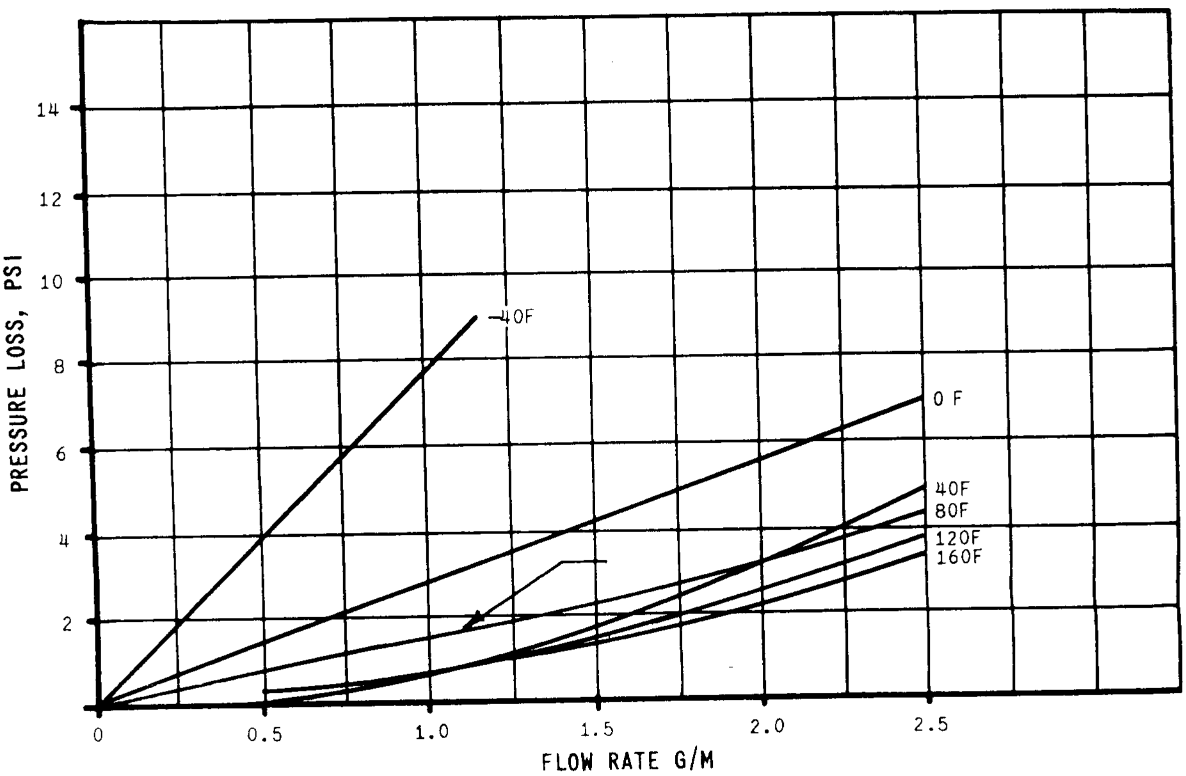
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN827-12 CROSS



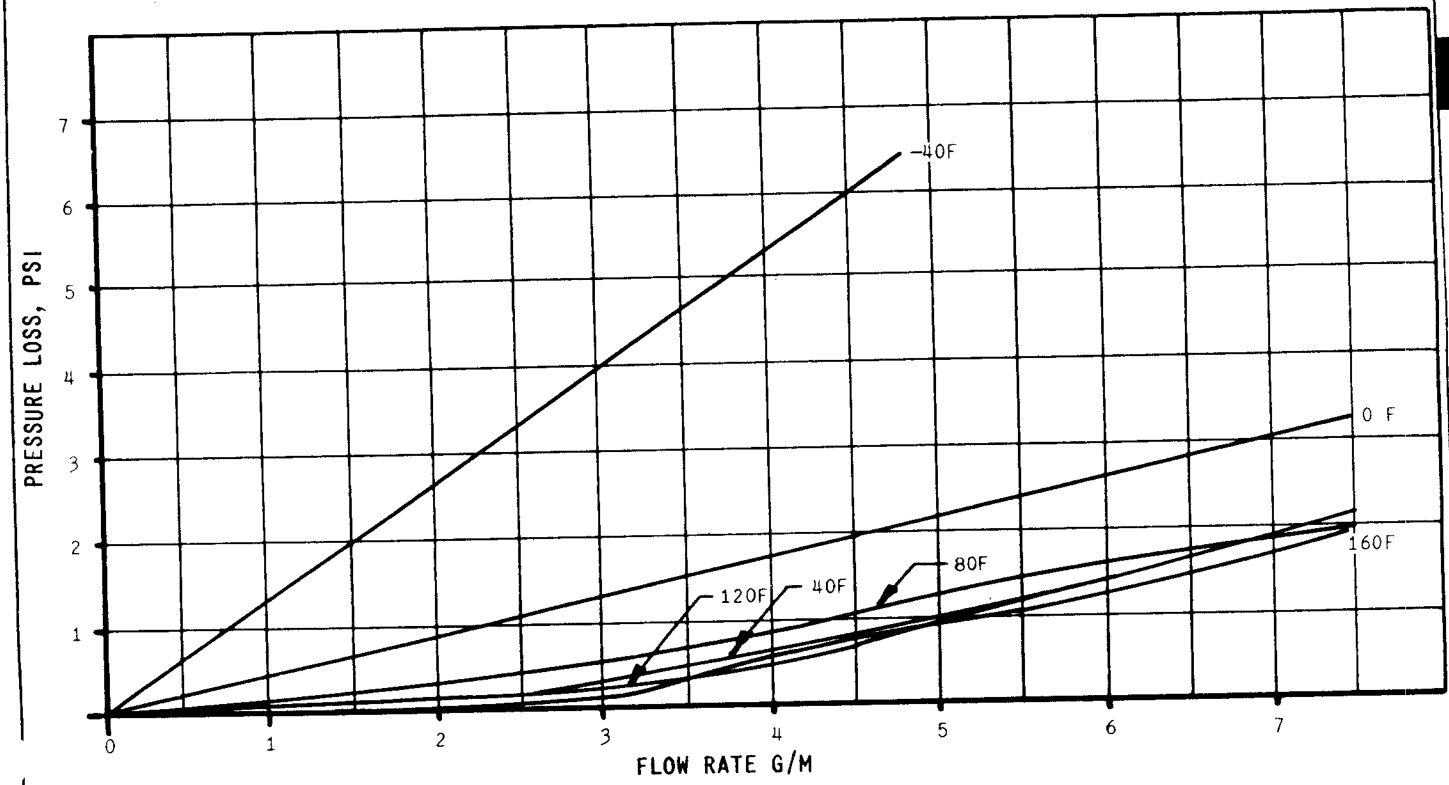
AN815-4 UNION



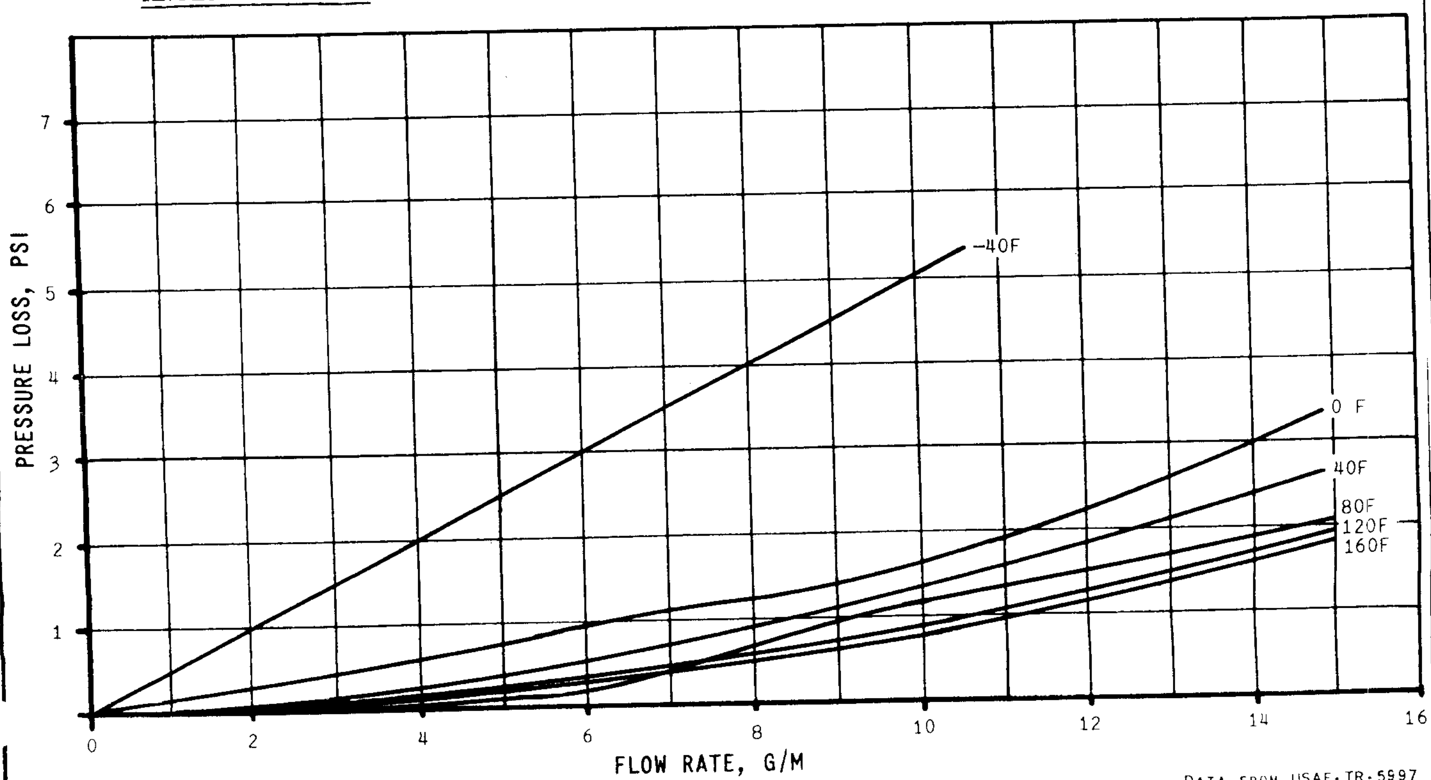
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN815-6 UNION



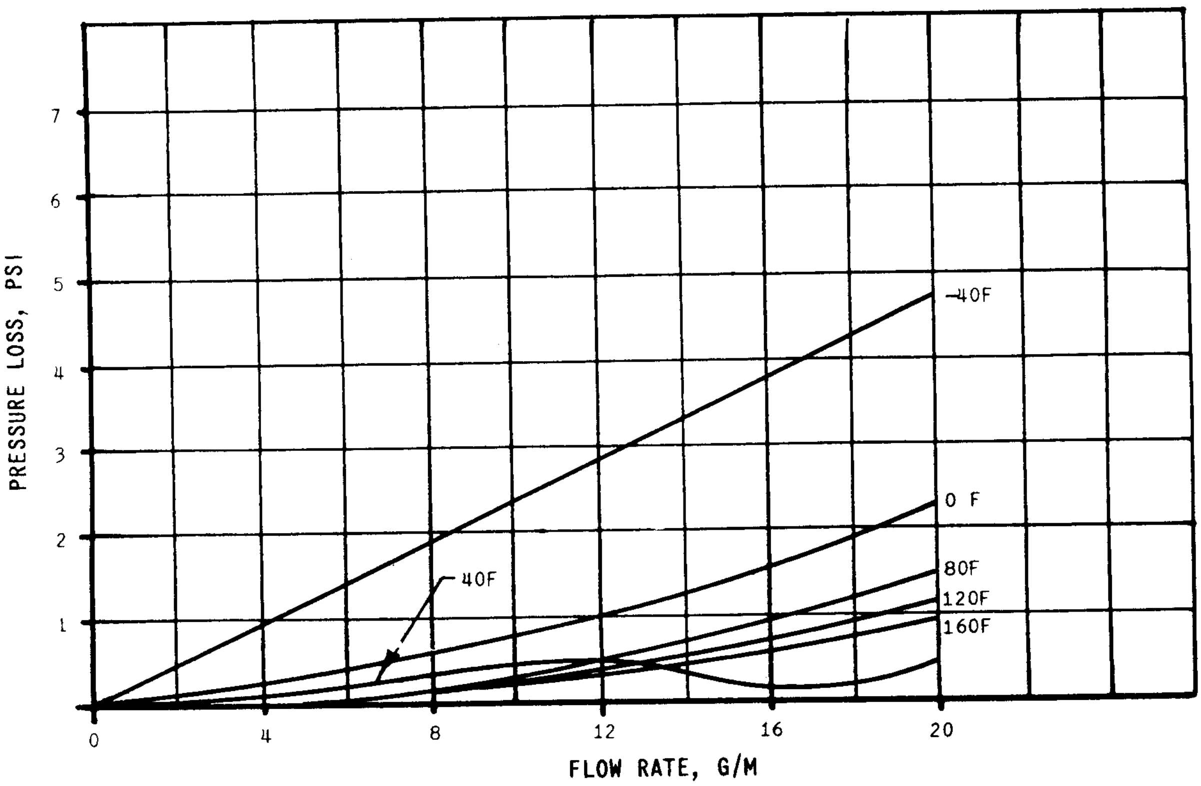
AN815-8 UNION



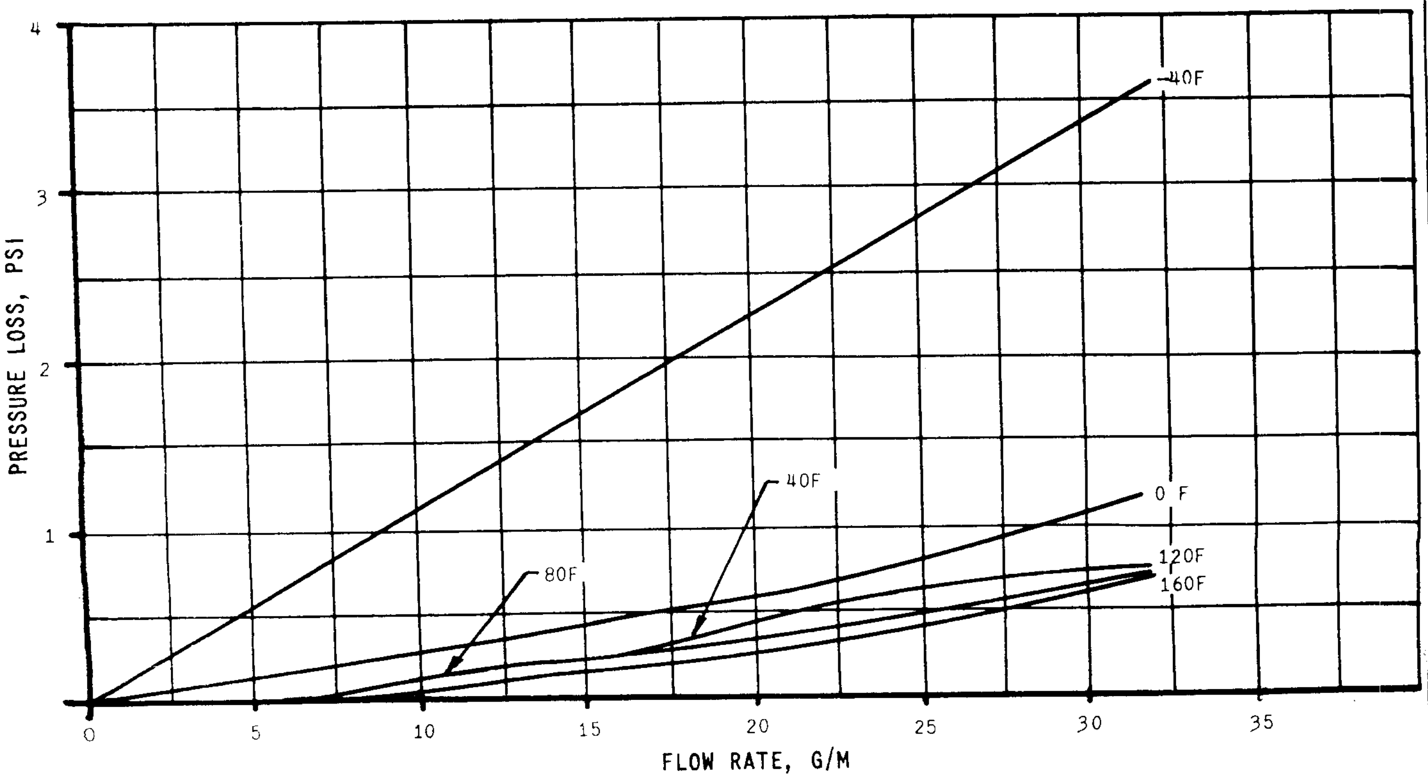
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN815-10 UNION



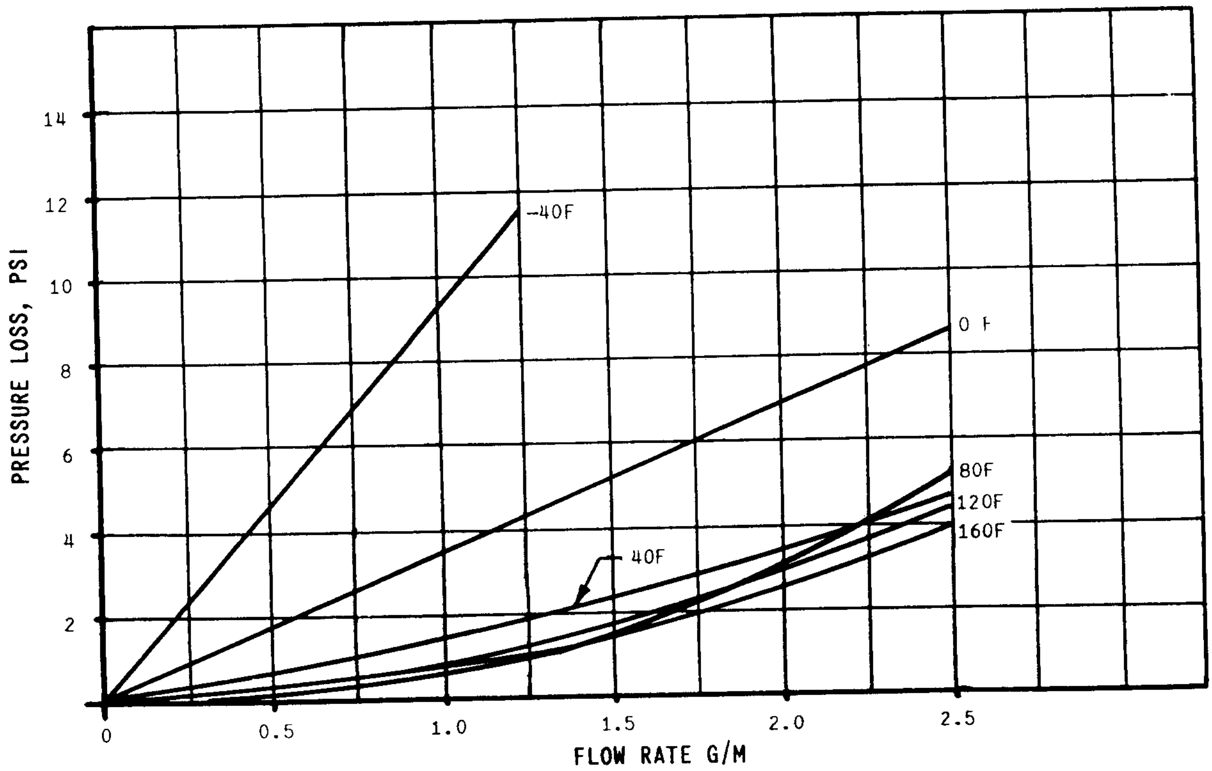
AN815-12 UNION



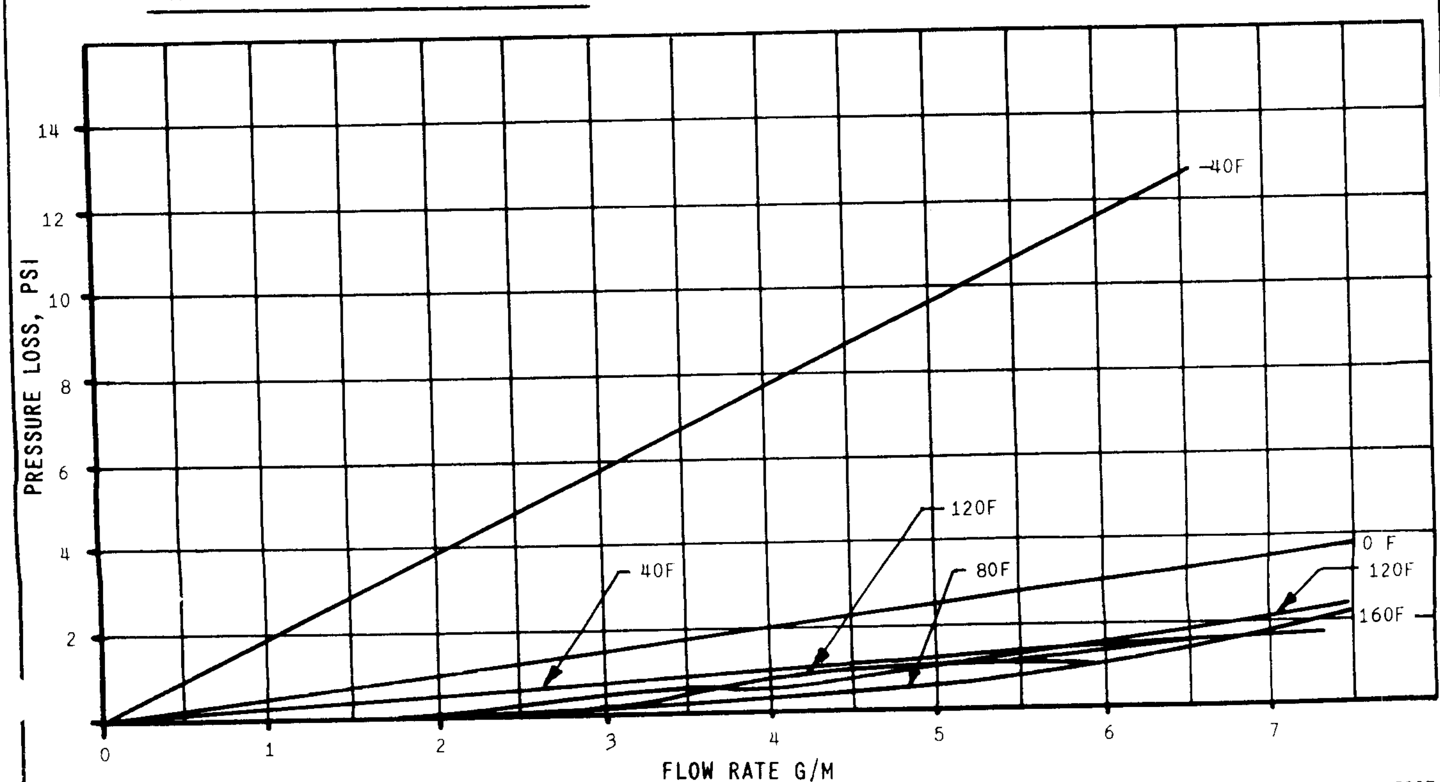
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN824-4 TEE & AN827-4 CROSS



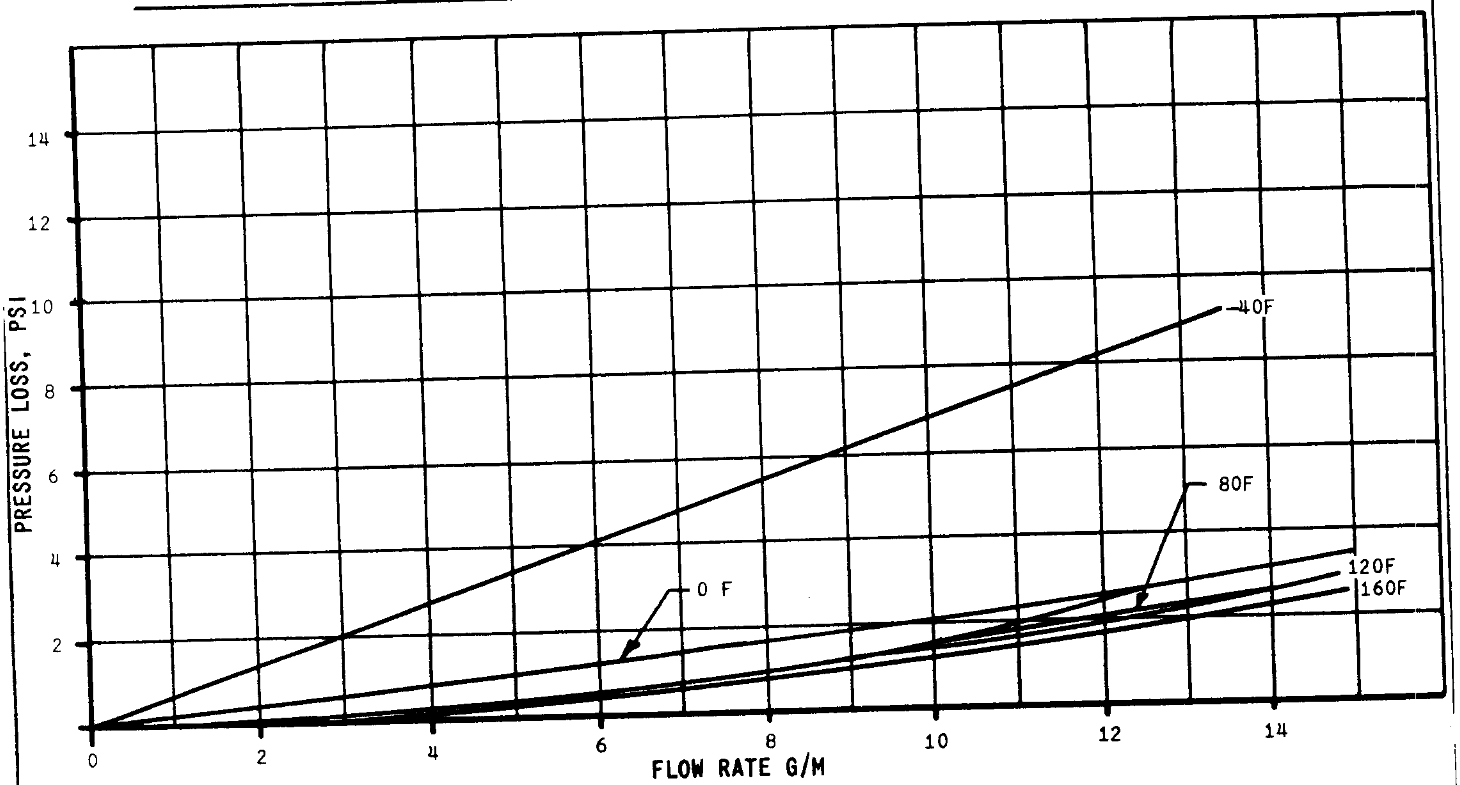
AN824-6 TEE & AN827-6 CROSS



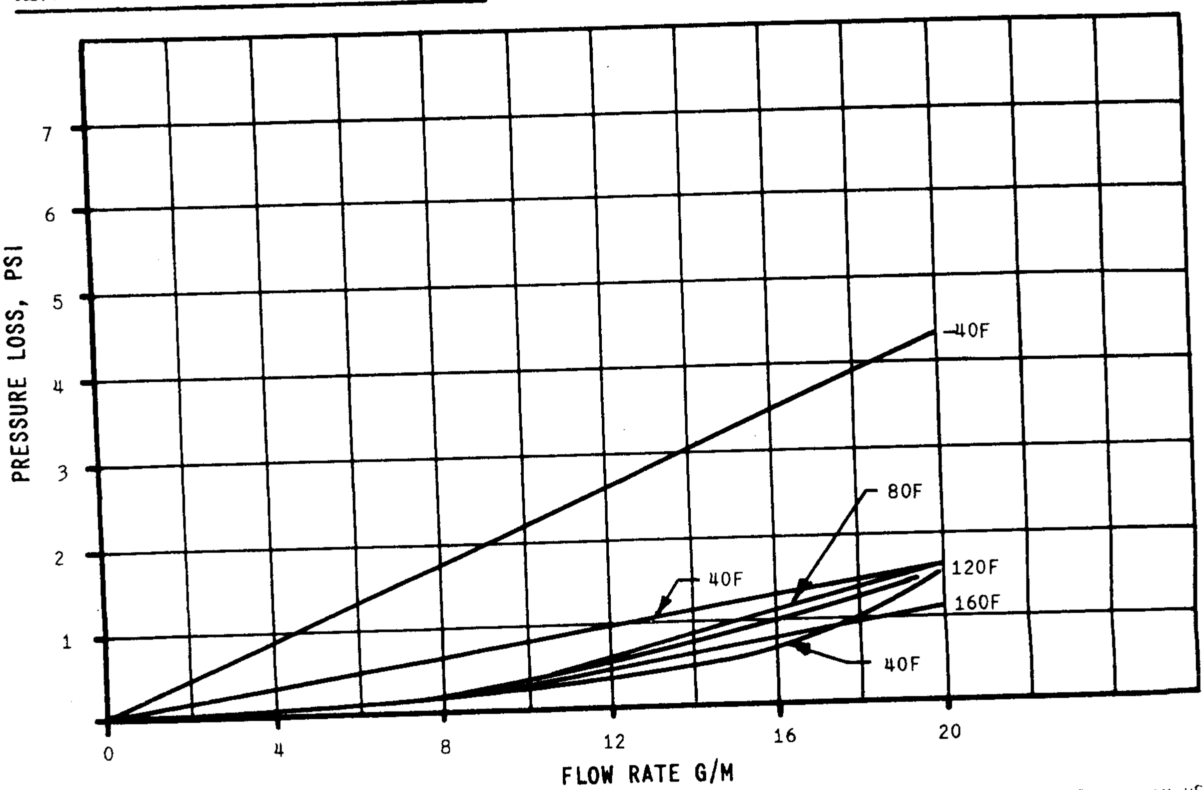
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN824-8 TEE & AN827-8 CROSS



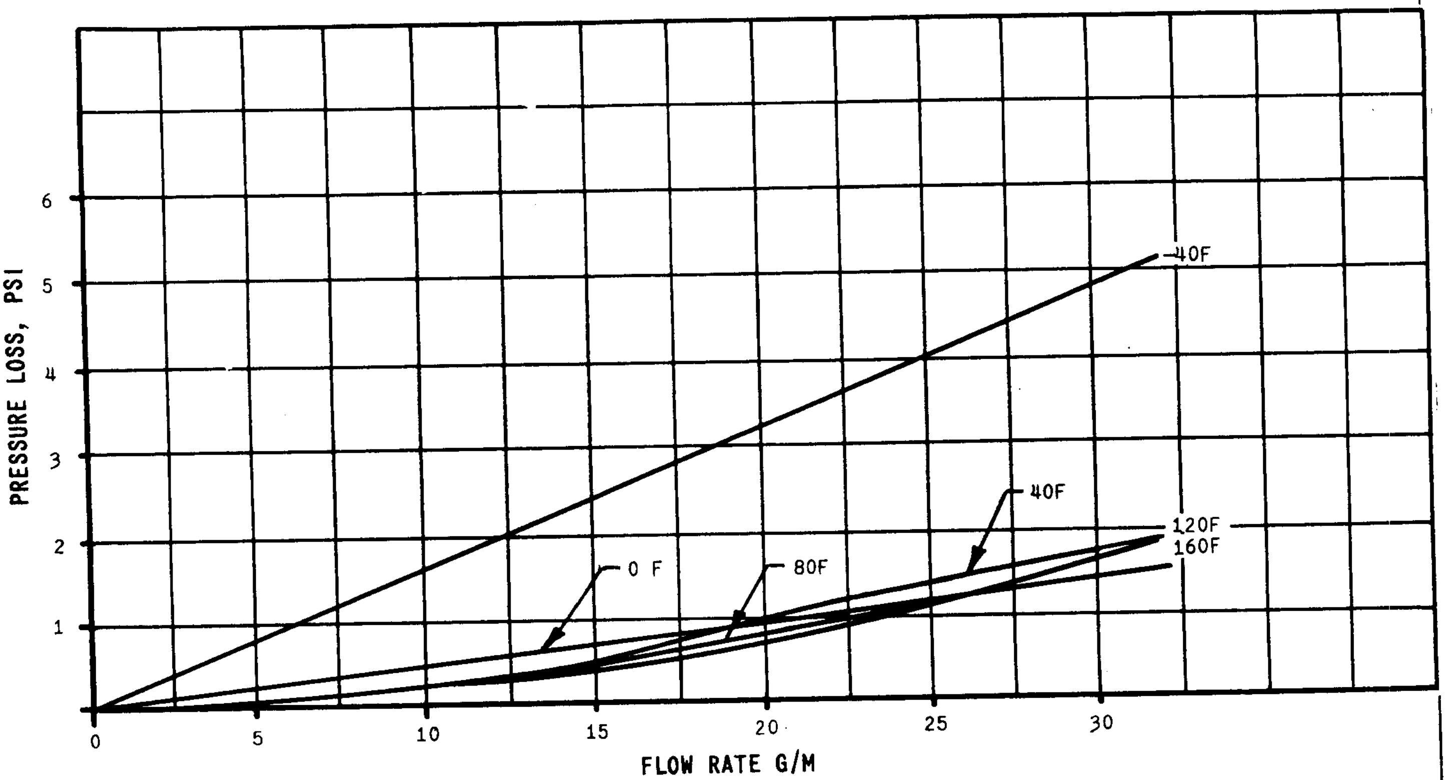
AN824-10 TEE & AN827-10 CROSS



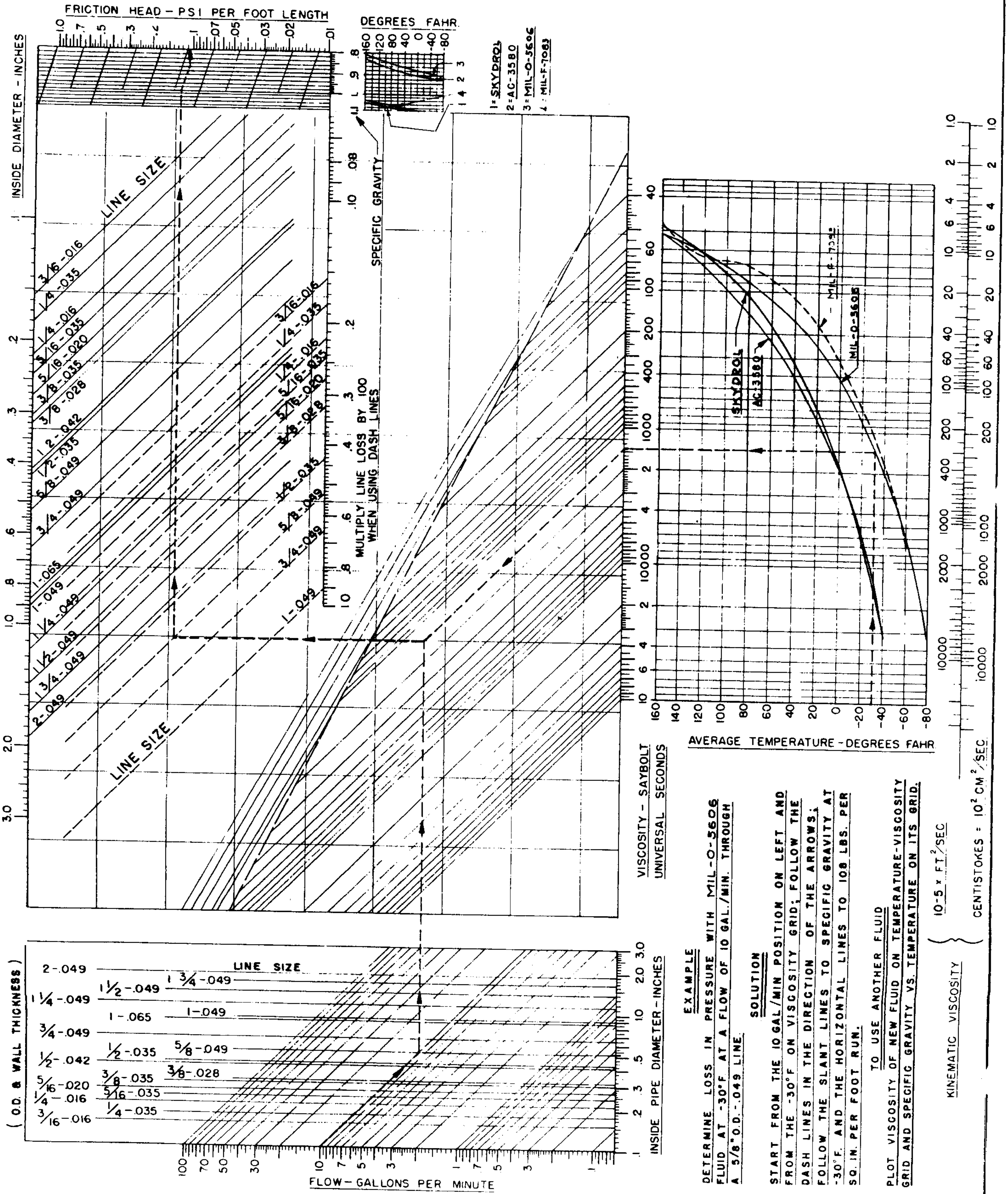
DATA FROM USAF-TR-5997

10.533 FITTING PRESSURE LOSS FOR MIL-H-5606 FLUID (CONT.)

AN824-12 TEE & AN827-12 CROSS



10.534 PRESSURE LOSSES IN TUBING



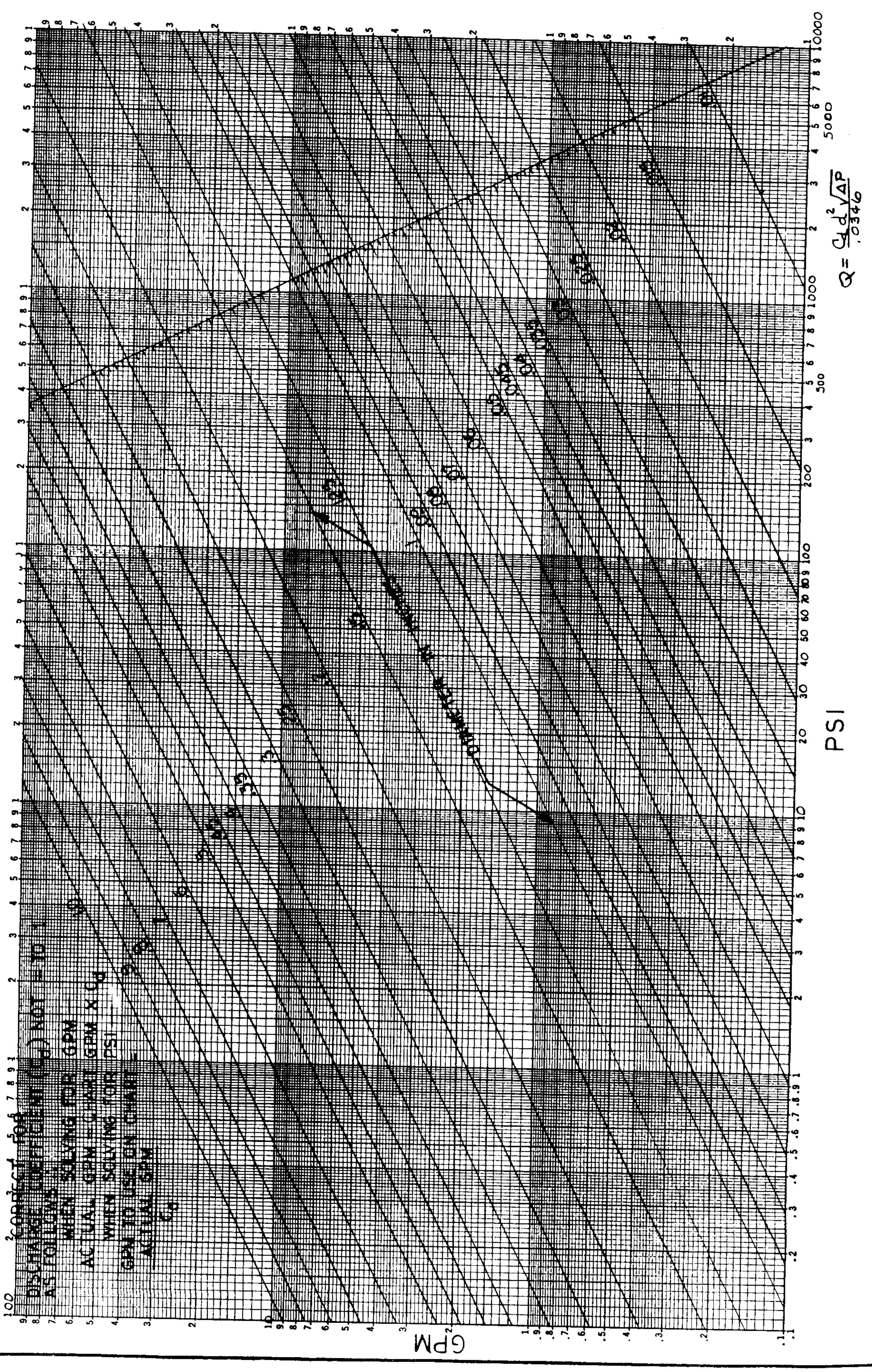
10.535 PROPERTIES OF MINERAL OIL FLUIDS

(REF DAC REPORT G0487)

PROPERTY	MIL-H-5606	MIL-H-83282	TEST METHOD
BASE	PETROLUM	SYNTHETIC PETROLUM	
VISCOSITY			
–65°F	3000 CS	–	
–40°F	–	2978 CS	
130°F	10 CS	–	
210°F	5.2 CS	3.82 CS	
400°F	–	1.10 CS	
POUR POINT SPECIFIC GRAVITY	–90°F	–65°F	
SPECIFIC GRAVITY	0.863 AT 60°F	0.8498 AT 60°F	
DENSITY (LB/IN.^{3} AT 1 ATMOSPHERE)			
60	–	0.0305	
100	0.0305	0.0290	
200	0.0290	–	
300	0.0276	–	
400	–	0.0253	
COEFFICIENT OF EXPANSION (IN.^{3}/IN.^{3}/°F)			
60° TO 160°F	0.00041	–	
WEIGHT PER GALLON			
BULK MODULUS (ADIABATIC TANGENT) PRESSURE			
1000 PSI AT 100°F	220×10^3	–	
1000 PSI AT 250°F	131×10^3	–	
1000 PSI AT 500°F	77×10^3	–	
2000 PSI AT 100°F	–	290×10^3	
2000 PSI AT 200°F	–	215×10^3	
2000 PSI AT 300°F	–	160×10^3	
3000 PSI AT 100°F	246×10^3	–	
3000 PSI AT 250°F	156×10^3	–	
3000 PSI AT 500°F	102×10^3	–	
4000 PSI AT 100°F	–	314×10^3	
4000 PSI AT 200°F	–	238×10^3	
4000 PSI AT 300°F	–	182×10^3	
5000 PSI AT 100°F	270×10^3	–	
5000 PSI AT 250°F	160×10^3	–	
5000 PSI AT 400°F	124×10^3	–	
8000 PSI AT 100°F	–	350×10^3	
8000 PSI AT 200°F	–	280×10^3	
8000 PSI AT 300°F	–	224×10^3	
FLASH POINT	215°F	410°F	ASTM D-92 COC
FIRE POINT	250°F	495°F	ASTM D-92 COC
AUTOIGNITION POINT	470°F	640°F	ASTM D-286
POUR POINT	–90°F	–65°F	ASTM D-97
COMPATIBLE ELASTOMER	BUNA N	BUNA N	ASTM-286
CONTINUOUS OPERATING TEMPERATURE	–65°F/+275°F	–40°F/+275°F	
VAPOR PRESSURE			
LB/IN.^{2} AT 100°F	–	0.0116 PSI	
210°F	8 MM HG	–	
300°F	74 MM HG	0.152 PSI	
500°F	–	0.920 PSI	
TOXICITY	LOW	LOW	

PRESSURE LOSSES THROUGH ORIFICES C_D = 1 FOR SKYDROL "500"

(Based at 70°F, SG = 1.065)



20. DRAFTING DESIGN

CONTENTS

20.1 DESIGN SKETCHES

The basic ideas of a new design are usually developed in a freehand sketch. A meeting of those responsible for approving the work will assure the designer of the acceptability of the design before excessive time is spent on layout. It is usually necessary to locate surrounding structure and clearance points on the layout before starting sketch. Then place a thin piece of paper over it and work freehand approximately to scale. Check with Group Engineers in other groups to ensure the space is available before preceeding with the sketch. Very often a few strength calculations are made at this time on critical points and some scale line work done to ensure the feasibility of the idea. The sketch should be carried far enough to determine materials, methods of fabrication, locking, and adjustment. The time required to make the actual sketch will vary with the type of work and may require only a few hours. At most, sketches should not require more than 25 percent of the scheduled layout time. If a sketch is made accurately to scale and considerable time is spent, the purpose is defeated. It is realized that there are many ramifications in the application of the sketch procedure which will require careful consideration and good judgment. It should be borne in mind that the primary purpose of the sketch procedure is to produce the best possible design in the least amount of time. Sketches must agree with Layout Work Sheet. When a sketch is ready, the Group Engineer will request approval.

Design sketches must be signed by the Section Chief, and any others whose names appear on the Layout Work Order. This constitutes authority to proceed with the layout. There must be no delay in approval. If any person responsible for approving the work is not present at a sketch meeting, the approval shall be made by those who are present.

20.2 DESIGN CHECK LIST

Keep it Simple.

Can it be Eliminated?

Minimize Weight and Size.

Conform with applicable Specifications.

Meet Design Requirements: loads, energy, operating time, sensitivity.

Establish satisfactory operation under Extreme Conditions: temperature, humidity, corrosion, vibration, voltage, wind, dirt, ice.

Provide suitable Life.

Consider Handling Loads and Storage conditions.

Obtain suitable Operating Forces.

Avoid Marginal Performance.

Provide optimum Safety under misuse or failure.

Minimize Stress Concentrations.

Avoid Over-Constraint.

Show Clearances.

Determine the effects of Tolerances on operation, proper assembly and installation, wear, and eccentric loads.

Calculate the effects of Deflections and Friction.

Provide Accessibility for installation and maintenance.

Provide easy, Uncomplicated assembly, disassembly and adjustment. Develop adjustment procedure concurrent with design.

Use Available material.

Use Standard parts and Previously Used parts where possible.

Calculate Inertia effects - parts and fluid.

Does transient Time Available allow proper function and how is kinetic energy dissipated?

Components which receive Limit Loads Each Time they are used (i.e., arresting gear, catapult gear, and parts of the landing gear) must be investigated critically for Stress Concentration and Stress Levels and must demonstrate reasonable service life by Fatigue Testing.

Prevent Incorrect Connections through design configuration. Ensure that designs are "Murphy Proof" by controlling configuration to prevent inadvertent assembly that causes damage or malfunction.

Any essential service (landing gear, arresting hook, flaps, etc.) shall have an Alternate Method of Actuation.

Provide adequate sealing to prevent the entrance of Foreign Materials.

Care should be taken in the Material Selection of faying surfaces with relative motion.

20.3 LAYOUT CHECK LIST

20.31 DRAFTING PRACTICES

20.311 EXTERNAL CLEARANCE

a. Surrounding Clearances Clearances on all sides of the part being laid out must be shown, or reference made to the layouts or motion studies on which these clearances are shown in order to facilitate checking.

b. Size Increase Maximum possible clearance should be provided to allow for future increase in size, additional lines, etc.

c. Attaching Plumbing should be indicated.

1. Avoid Elbow Boss Connections Make every effort to point fluid ports of hydraulic units in the direction of the attaching tubing or hose such that straight fittings may be used instead of the relatively unreliable elbows.

2. All hydraulic connections should be made in one direction to allow easy unit removal.

d. Fasteners Nuts, bolts, rivets, pins, etc., must be shown when there is possibility of critical clearance in installation or operation.

e. Surrounding Parts are to be shown, with the layout or production drawing number and title noted.

f. Tool Clearance and access should be adequately provided for all attachments.

20.312 INTERNAL FITS

a. Ample Overlaps and Clearances should be provided to allow use of reasonable tolerances on the detail drawings.

b. Tolerance Checks are to be made on the layout vellum, or on a print of the layout or assembly drawing filed with the layout or with the design computations. Refer to DM 06.43.

20.313 POSITIONS Extreme positions of any moving parts must be shown, and any intermediate positions critical to function or clearance.

20.314 DIMENSIONS should be limited to the following:

a. Locating dimensions to airplane reference lines.

b. Matching dimensions and tolerances which have been agreed on between sections to ensure matching parts, such as landing gear attachment to structural fittings.

c. Calculated dimensions for which scale accuracy is not acceptable.

d. Special fits and minimum acceptable clearances.

20.315 OTHER SECTIONS affected by the design must be consulted, and approval of their Group Engineers obtained in cases where they are importantly affected.

20.316 REVISED DETAILS If detail parts are changed, the layout should be changed to agree, or, if time is not available, a reference note should be put on the layout.

20.32 NOTES TO INCLUDE WHEN APPLICABLE

20.321 OPERATING LOADS including pressures, velocities, working or actual loads, and maximum loads must be given on the layout for all mechanisms, together with design loads and available forces.

a. Friction Allowance should be made for friction, pressure drop in in lines, etc.

b. References All references used to obtain loads should be listed to facilitate checking.

20.322 OPERATING CONDITIONS All conditions of operation must be shown, by schematic diagrams when necessary, to ensure that other persons will understand.

20.323 ADJUSTMENT All necessary adjustment notes must be given exactly as they should appear on the assembly drawing or adjustment procedure drawing.

a. Bench Adjustments Where possible, adjustments should be made before installation.

20.324 TESTS

a. Special Procedures If special tests appear necessary, they should be roughly outlined and coordinated with the Test Group Engineer.

b. Applicable Data When available, test data, references to test reports, or references to similar tested parts should be given for all designs of mechanisms. Unconventional designs must be tested.

20.325 STANDARDS

a. Standard Designs When standard designs are used, the layout should note STANDARD DESIGN with an arrow to that area.

b. Standard Part numbers and coding should be called out when looked up in the course of design.

20.326 MATERIALS AND PROCESSES

a. Materials and Sizes should be noted unless they can be readily determined from the drawing itself.

b. Treatments Necessary heat treatments, finishes, etc., should be noted, unless conforming to general practice.

c. Stress Inspection Magnaflux or magnetic inspection should be noted for all highly stressed parts. The Group Engineer should be consulted in deciding whether this inspection is necessary.

20.33 STRESS CONSIDERATIONS

20.331 CALCULATIONS Stress calculations, including bearing pressures and approximate design loads, necessary for the design of parts should be made directly on the layout, or on design computation pages.

20.332 CONCENTRATIONS Stress concentrations should be reduced to a minimum by the use of fillets, etc.

20.333 FRICTION MOMENT Links moving under load must be analyzed for the end moment caused by friction plus the column or tension load.

20.334 DEFLECTIONS

a. Limit Load should be used in the calculation of deflections.

b. Stops Parts designed as stops or absorbers of stored energy must not deflect sufficiently to cause malfunction.

c. Eccentricity Control It is frequently desirable to design linkage geometry so that deflection under maximum load produces a minimum eccentricity of load transfer at the most frequent or most critical position.

20.335 DISTORTION of parts due to pressure, thread loads, manufacturing holding loads, gasket loads, interference fits, etc., must be considered.

a. Minimize distortion through design configuration.

b. Allowance If distortion during manufacture is unavoidable, allowance must be made for the necessary finishing operation.

20.336 PRELOADED BOLTS Size for preloaded bolts must be chosen from allowable stress loads in Section 30.

20.34 DESIGN PRACTICES

20.341 GENERAL

a. Crossed Lines and Controls Every effort shall be made to assure that it will be physically impossible to incorrectly install cables, levers, cranks, hydraulic lines, or any other parts that can cause malfunction.

b. Available Tools Provide for use of standard or existing tools where possible, in fabrication and assembly.

c. Standard Parts should be used where possible. The Structures Design Section issues fastener usage policy documents, directly applicable to primary load carrying structure, and to be used as a guide in other applications. These documents describe preferred part combinations, bolt, rivet and lockbolt usage, etc., and include standard part drawings:

Report LB-32938 for DC-8
Report LB-31359 for DC-9
Report DAC-33956 for DC-10

For model DC-10, virtually all the types of parts found in the Standards Manual are controlled by Specification A113250, Approved Parts List, issued by the Reliability and Standards Section. Where applicable, either conform with this list, or submit Parts Approval Request (Form 80-35) for items not covered.

d. Nonstandard Part Shape When nonstandard parts resembling standard parts are required, external proportions should be altered to avoid confusion.

e. Air Bleeding Provision must be made for bleeding brakes and other systems where displaced volume is less than line volume and where the presence of air could cause malfunction.

20.342 STATIC AND MOVING SEALS should conform with Douglas or military standards or appropriate specifications.

a. Gasket Static Seal Installation should be designed in accordance with Section 100.

b. Face Seal O-Rings should be backed with S3929237 rings wherever space permits. See S3929237 or Section 100 for gland dimensions.

c. Face Seal Inner Flange may be omitted, with design supervision approval.

d. Face Seal Deflection Cover plate deflections under maximum operating pressure in combination with out-of-flatness of the mating faces must not expose the face seal to extrusion gapping in excess of 0.004 inch, for an O-ring with S3929237 backup. For unbacked static seat O-rings, extrusion gapping must not exceed 0.0005 inch.

e. V-Ring Packing Installations should conform to Section 100.

f. O-Ring Packing Installations should be designed in accordance with Section 100.

g. Corrosion Protection at Packings All parts which slide across packings should be chrome-plated, hard-anodized, or be of corrosion-resistant material. Packing grooves and static seal glands should be similarly protected or plated or anodized.

20.343 MATERIALS AND PROCESSES

a. Corrosion Resistance Materials, surface coatings, and material combinations must provide suitable corrosion resistance, both inside and outside.

b. Special materials, combinations and applications must have the Section Chief's approval.

c. Aluminum Alloy Mating Parts should not be used in bearings or in frequently used threads. (Example: Reservoir filler plug.)

d. Case Hardening Latches, cams, triggers, and similar parts subject to possible wear with high bearing pressure should be hardened to Rockwell 30-N 76 minimum or equivalent.

20.344 THREADS

a. Standardization Use of threads not in the standard hydraulic thread series, Section 100, should have Group Engineer approval.

b. Preload Torque should be given on the layout in accordance with Section 30.

1. Face Seal bolt preload should conform with Section 100.

c. Avoid Pipe Threads Use of pipe threads in special applications requires Section Chief approval.

1. Male Pipe threads must not cause interference when screwed in 3/32 inch deeper than maximum tight-joint depth listed in Section 100.

d. Internal Pipe Thread Boss OD is listed in Section 100.

1. Steel parts are to conform to the diameter listed.

2. Cast parts to conform to the diameter listed for castings.

3. Wrought Aluminum Alloy parts are to be increased 1/8 inch on diameter.

e. Boss Salvage Allowance On expensive aluminum alloy parts, or on parts permanently joined to such parts, raised boss OD's (for either straight-thread or pipe-thread fittings) shall be increased to the next larger size, to permit retapping, subject to design supervision approval. (Reference Section 100.)

f. Boss Internal Clearance Make it impossible to choke off a fluid flow passage or to produce an internal interference by inadvertent installation of a fitting deeper than normal, including possible substitution of a "bulkhead" fitting for a "straight" fitting.

g. Locking Means should be shown for all threaded joints. Tube assemblies are considered "locking means."

20.345 RETAINER OR SNAP RINGS shall not be used where ring failure could allow blow-apart of the unit, or where end-play could allow failure of seals or other parts.

20.346 SPRINGS Layouts should amply describe all springs.

a. All Critical Features of the design should be shown including means for retention of springs under vibration and fluid-flow effects.

b. Operating Forces and Operating Loads should be shown.

c. Maximum Stress, including Wahl factor, should be shown.

d. Safety Load Tension springs should not be stressed above 80 percent of maximum safe stress at extreme adjustment plus deflection.

e. Spring Rate Variation Plots of load versus deflection showing spring rate variation with spring and wire diameter tolerances, are very useful and are recommended for all layouts.

f. Changes in Spring Diameter due to extension or compression must be considered, and clearance provided.

g. References Drafting Manual, Section 71; the "Manual of Spring Engineering" (American Steel and Wire Company); "Mechanical Spring Design" (Associated Spring Corporation); and Section 30 of this manual have useful information.

h. Spring Load on Adjusting Nuts or Caps, with 4 threads in engagement, should not exceed 10 pounds per inch of thread diameter for personnel safety reasons.

20.347 BEARINGS AND BUSHINGS

a. Antifriction Bearings

1. Multiple-Bearing Fits Only one bearing on a hinge should take the thrust, in both directions. The other bearings should be free to float endwise.

2. Floating-Fit Housings should be steel bushed.

3. Interference-Fit Housings need not have bushings.

4. Shaft Fit All bearings should have the recommended interference fit with the shaft.

5. Manufacturer's Recommendations for allowable loads and fits should be followed. See design supervision if recommendations appear contrary to good practice.

b. Bearing Pressure Allowables See Sections 30 and 60.

c. Nonuniform Bearing Pressure, such as that caused by deflection, must be checked for the maximum value encountered.

d. Threads or Other Irregularities should never contact the bearing surface.

e. Adequate Lubrication must be provided.

1. Lube Fitting must be accessible, and located in a region of low stress level.

2. Omission of Lube Fitting is sometimes permissible for needle bearings, subject to Group Engineer approval.

3. Preferred Lube Fitting is the pressed-in NAS516-1, based on fatigue experience with the MS15001 threaded lube fitting. If a threaded fitting is necessary, special consideration must be made to assure a low stress level at the threaded region.

f. Retention Where a bearing is pressed in, the press fit shall not be relied on for retention. Mechanical retention shall be used.

1. Staking, if used, shall not be relied on to keep parts together.

g. Bushings in Blind Holes should be designed where possible to allow removal.

h. Salvage Allowance For plain unbushed holes, provide extra material for salvage with the next larger OD pin or bushing, in all except inexpensive parts.

1. Exception may be made in nontransport shorter-life applications, subject to Group Engineer approval, where the backing is adequate to assure satisfactory service.

i. Use of Plain Sintered Bearings must have approval of the procuring agency, and shall only be used in applications involving only oscillating or slow motion.

j. Shock Strut design should conform with MIL-S-8552. Allowance must be made for service rework.

20.348 FLEXIBLE HOSE Refer to Section 40 for standards and design considerations.

20.349 FABRICATION

a. Grind and Hone Reliefs OD's and ID's must be provided with relief diameters for all grinding and honing operations.

1. Honed Bores For internal honing operations, the following lengths of reliefs should be provided wherever possible:

1/4-inch relief for bores up to 1-inch diameter.

3/8 inch for 1- to 2-inch bores.

1/2 inch for 2- to 4-inch bores.

5/8 inch for bores over 4 inches.

b. Sheet Metal Stiffening Sheet metal parts should be flanged and braced where possible for maximum rigidity.

20.4 SCHEMATIC DIAGRAM CHECK LIST

The hydraulic system schematic diagram is the authority for the purchase and hookup of a considerable quantity of hydraulic hardware and is the display for system function study. The diagram warrants extraordinary care and accuracy to include all necessary information and to facilitate study. Often liberty is taken in lengths of pressure and return manifolds to simplify display but the sequence of attachment shall be maintained so that line lengths and fittings can be added as required. The following items should be considered before the schematic diagram is approved:

a. Design simplicity and ease of maintenance are synonymous with reliability and cannot be overemphasized. "Gadgets" requiring specialized handling or even design changes requiring reevaluation should be used only if real performance, weight or operating simplicity advantages accrue.

b. Multiple systems working in parallel - for improved reliability - shall have no common dependencies, i.e., common reservoir pressurization, common power supply on multiengine craft. When shuttle valves are used to port emergency power into a common actuator, they shall be located as near the common unit as feasible. If not on the common unit a specification deviation must be secured (for military aircraft).

c. The output from each unit of a multiple pump supply shall be checked off to prevent failure of a pump shaft to allow reverse flow through the inactive pump.

d. A check valve in the selector valve inlet pressure line prevents loss of subsystem pressure in case operation of another subsystem utility momentarily reduces supply pressure. Similar protection can also be realized by using a priority valve (balanced relief). Return line check valves may be employed to prevent system interaction.

e. Adequate provision for thermal volume changes shall be incorporated.

f. Evaluate performance of critical subsystems for all power conditions.

g. Flow surges from one unit or subsystem must not disrupt or seriously disturb function of other units, i.e., flap return plumbed into pump supply or brake return might fail pump housing or apply brakes at inopportune times.

h. For subsystems equipped with emergency actuation the diagram should carry recommissioning instructions and an air bottle drain warning if applicable.

i. Each orifice and check valve installation should be analyzed for pressure boosting (differential piston area + working loads) also for gust and inertia loads. Example: Inadvertent brake application during landing gear retraction affects retract moments.

j. Evaluate synchronization requirements to avoid assymetrical flight loads and to ensure that units operating in parallel will not malfunction or be overloaded by wrong sequence.

k. Micronic filtration should be provided for any servo-controlled fluid metering situation, and fine metallic filters for fixed restrictors of orifice diameter less than 0.090 inch.

l. Line size and material shall be indicated. Steel shall be used in any exposed installation. Hoses shall not be used for high pressure without specific approval.

m. Electromechanical and electrohydraulic control devices must be so designed that loss of a system will not result in hardover signals.

n. Check route of passage of all fluid during each step of normal and standby function.

o. Adequate means for the release of entrapped air from all lines and components shall be provided and the existence of any special equipment and/or instructions shall be noted.* "Bleeding" provisions shall be provided in vibration damping and slave cylinder (wheel brake) applications wherein a small air trap can render the subsystem useless.

p. Where practical external venting (leak passage to atmosphere) must be provided at the juncture of parallel systems to avoid any possible interchange of fluid by seal failure.

q. Normal nameplate data and information essential to service maintenance personnel shall be compiled by the designer and recorded on the schematic diagram near the unit in question.

20.41 CLARITY AND COMPLETENESS OF DESCRIPTION

Units shall be shown schematically in cut-away views. Picturization and adjacent notes shall adequately define function and shall enable flow paths to be readily traced for each operating condition.

Where necessary to clarity, additional views shall be included showing alternate positions of valves or other units.

Alternate positions of cylinders shall be shown in phantom. Actuator position and the corresponding position of the actuated mechanism shall be made clear by notation or by phantomed schematic views of the mechanism.

*In general, system subfunctions which have more actuator displaced volume than line volume are self-bleeding and require no special provisions.

20.5 SUBSYSTEM DESIGN PROCEDURE

20.51 BASIC CONSIDERATIONS

a. Schematic Design The system schematic diagram displays the method of performing the required functions by showing the units of hydraulic machinery and their lines hookup. The designer will think through each step of the operation resulting from all normal and inadvertent conditions and imagine any possible malfunction. The Schematic Diagram Check List Section 20.4 will be helpful in this regard.

b. Capacity Hydraulic systems designed to over-capacity requirements of force or horsepower are wasteful of weight both by way of component installed weight and bypass energy losses. Power supply capacity is based on the specification requirements of the largest subsystem to be operated and at the critical power setting. The duty cycle of any significant subsystem should be inspected to determine advisability of using some type of energy storage means.

c. Line Sizing The proper line size is a compromise between installed weight and allowable pressure loss. Pressure loss considerations include:

1. Acceptable cold temperature operating rates.
2. Orifice sequencing requirements at full range of temperature.
3. Load stroke curve (Supply pressure available - Load pressure required = PSI loss allowable.)
4. Line fluid elasticity and inertia energy and its effect on servomechanism system stability.
5. For estimating purposes a fluid velocity of 15 ft/sec is recommended per MIL-H-5440. Generally this velocity can be considerably exceeded. (See Section 10.)

d. Miscellaneous Considerations Hydraulic efficiency in any but the largest (or design) subsystem is generally insignificant except if a heating problem results. Mechanical efficiency, however, is important since by matching mechanism to load function the structural weight is reduced.

For a given maximum hydraulic pressure and moment acting through a given travel the fluid spring rate and actuator weight are virtually constant whether moment is produced by a small actuator through a long stroke or by a large actuator through a short stroke. Some advantage is generally realized with the more rigid small support bearing in the long-stroke system.

20.52 PRESSURE SURGES Refer to Section 10 in conjunction with the following:

a. Causes Pressure surges are caused by rapid valve operation, bottoming of cylinders suddenly stopping the flow of a long column of fluid in lines, pump feathering mechanism response lag, gust loads on aerodynamic control surfaces and suddenly applied g loads during operation of such services as landing gear retraction mechanisms.

b. Space Provisions If any possibility of surges exists, provision (space) shall be made for the later incorporation of means of preventing or reducing the magnitude of surges.

c. Measurement The hydraulic system will be instrumented and monitored for the presence and magnitude of surges during the hydraulic system ground and flight test programs.

d. Means of Control Means of preventing or controlling surges are:

1. Do not move services any faster than necessary. (Keep flow rates to a minimum.)

2. Keep line lengths between valves and cylinders to a minimum. (Reduce fluid column inertia.)

3. Use dashpots or orifices to slow the motion of the main valving element(s) of solenoid and other selector valves.

4. Use dashpots or metering pins in actuating cylinders to slow the motion at extremes of travel. Such dashpots also serve to reduce mechanical inertia loads on associated mechanisms at the ends of travel.

5. Use surge dampers in valve pressure or cylinder lines. These dampers provide an opening delay which can be varied as required.

6. Use relief valves, if they are sufficiently responsive, to reduce surges.

20.53 EMERGENCY POWER SUPPLY Standby systems for actuating essential functions normally powered such as wheel brakes, landing gear extend, flight controls, etc., shall be provided.

a. Preferred Method Where feasible the standby power shall be air load and/or gravity with manual effort acting through mechanical linkage for control. This is believed to have maximum reliability. For example: Forward retracting landing gear with manual latch release.

b. Manual Control with Power Boost Power boosted systems are sufficiently reliable; however, with loss of power assist, the increase in pilot effort required and the reduced control available would be unacceptable for many functions.

c. Stored Air Pressurized air stored in reservoirs and acting through a shuttle valve at the actuator is commonly used despite the complication and service attention involved. Agreement must be reached on capacity for functions involving repeated applications, i.e., wheel brakes and surface controls.

d. Flight Controls Powered flight controls have developed to the philosophy that two identical or nearly identical systems working in parallel is optimum. Where both are powered from one engine, emergency power is frequently supplied by an air turbine thrust into the wind stream when required.

e. Gunpowder Actuation The lightest power source for large energy standby or other one-shot installation is powder or cartridge actuated but these systems require development. Powder cartridge actuation is not as adaptable to a system requiring an operating period of several seconds, because rate of gas propagation is relatively rapid, and storage or damping delay of gas energy causes great losses due to heat flow and orifice drop.

20.531 SYSTEM SEPARATION In critical functions utilizing standby systems, both basic and standby actuation means and energy supply should be routed along primary structure and should be separated as far as reasonable to avoid similar failure causes. Physical loss of any nonessential components should not significantly affect safety.

20.54 NEW DEVELOPMENTS

20.541 STANDARD COMPONENTS Certain hydraulic system subfunctions are common to all aircraft hydraulic systems regardless of application, i.e., engine shaft to hydraulic energy conversion, pressure control, energy storage, direction control valves, fittings, etc. Standard components to perform these subfunctions have been developed and service tested on numerous airplanes and should be used where applicable.

20.542 SPECIAL COMPONENTS From time to time special design functions or advancements in the state of the art indicate the use of special service units. In general, the Douglas policy and practice regarding development and procurement of these units follow this pattern.

a. The responsible designer shall compile a data sheet setting forth the desired characteristics of the unit and a principle of operation by which these characteristics can be realized. The data sheet need only be adequate for discussion with the Section Chief, but it must reflect sufficient thinking to constitute a serious proposal. (The table in Section 20.62 will be helpful in this regard.)

b. Depending on the significance of the proposal the Group Engineer may recommend a formal specification be prepared followed by a market review and normal procurement.

c. In general, if the desired unit is not a vendor shelf item the schedule deadline will dictate the prototype units be designed, manufactured and tested by Douglas.

Normally the qualification, testing and development of a new unit requires three to four times as long as the engineering, design and fabrication of the prototype unit. This represents a major reason for avoiding the introduction of new or nonqualified components.

20.6 COMPONENT DESIGN PROCEDURE

20.61 STATE THE PROBLEM Briefly record detail function and design philosophy of system or component being designed along with due dates, quantities required and other information required per design data chart (ref. 20.62). Done properly at the outset, this record will ensure that all affected interests are proceeding toward the same end and will also expose areas of sparse knowledge.

This record is useable in liaison with other design and service interests and will save much design time spent in repeated explanations.

The record is to be approved by the Group Engineer and might well be compiled in collaboration with him.

It is mandatory that this record be kept up to date particularly regarding design deficiencies since it is by this means that engineers learn.

GENERAL DESIGN DATA CHART

20.62 NECESSARY AVAILABLE DATA

CHECK LIST OF DESIGN DATA TO BE AVAILABLE AT START OF COMPONENT DESIGN AND SPECIFICATION

		PUMP & MOTORS	RESERVOIR	ACTUATORS	SNUBBERS	FLOW CONTROL VALVES	SELECTOR VALVES	SERVO VALVES	COOLERS	ACCUMULATORS	LANDING GEAR SHOCK STRUTS	LATCHES	ARRESTING AND CATAPULT HOOKS	PRESSURE SWITCHES	WHEELS	BRAKES	TIRES
	TYPE AND SIZE	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
AIRPLANE INFORMATION	**INSTALLATION**																
	SPACE																
	SIZE ENVELOPE	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
	PIPING ARRANGEMENT	X	X	X	X	X	X	X	X	X						X	
	GEOMETRY																
	ATTACHMENT POINTS AND LOADS	X	X	X	X		X	X	X	X	X	X	X	X	X	X	
	MOTION STUDY – (A) STROKE, (B) PRELOAD			AB	A		A	A			AB	AB	A				*
	HYDRAULIC SYSTEM																
	(A) FLUID, (B) OPERATING PSI, (C) BACK PSI	ABC	ABC	ABC	AB	ABC	ABC	ABC	ABC	AB	AB			AB		ABC	
	(D) PEAK FLOW, (E) NORMAL FLOW	DE	D	DE	DE	DE	DE	DE	DE	DE	D					D	
	ENVIRONMENT																
	(A) TEMP RANGE, (B) EXPOSURE TO WEATHER	A	A	AB	AB	A	AB	AB	A	A	AB	AB	AB	AB	AB	AB	A
	(C) VIBRATION, (D) VENTILATION	CD	CD	CD	CD	CD	CD	CD	CD		C			C	CD	CD	D
	PERFORMANCE – MECHANICAL																
COMPONENT OUTPUT OR RESISTANCE	**LOAD ENVELOPE (OPERATING)**																
	(A) GRAVITY, (B) ACCEL, (C) BUNGEE, (D) AIR,	AB	AB	ABCD	ABCD			B			ABCD	ABCD	ABCD	B	B	B	B
	(E) TORQUE, (F) SCHEDULED, (G) HANDLING	EG	G	G	FG		G	G		G	EFG	G	FG	G	G	EG	EG
	STRUCTURAL			X	X						X	X	X		X	X	
	OPERATING FORCES (CONTROL)						X	X				X	X				
	DUTY CYCLE	X		X	X	X			X	X					X	X	X
	RATE – (A) PEAK, (B) CONTINOUS	AB	A	AB	AB	AB	A	AB	AB	A	A	A		A	AB	AB	
	HYDRAULIC																
	PRESSURE LOSS ALLOWABLE																
	(A) VS FLOW, (B) VS FLOW AND CONTROL DEFLECTION				A	A	A	B	A	A	A						
	OPERATING PSI	X		X	X	X	X		X	X		X		X		X	
	LEAKAGE ALLOWABLE	X	X	X	X	X	X	X		X	X			X		X	
	FRICTION OR EFFICIENCY	X	X	X	X	X	X	X	X	X	X	X	X	X			
	EMERGENCY			X	X		X	X			X	X	X			X	
	ENDURANCE	X		X	X		X	X		X	X	X	X	X	X	X	
	SPECIAL DESIGN FEATURES AND REQUIREMENTS																
	(A) STROKE ADJUSTMENT, (B) INTERNAL LATCHES,	B		AB									AB				
	(C) RESPONSE, (D) SPRING RATE,	C		D	D	C		C						C			
	(E) COLD START, (F) BLEED PROVISIONS	EF	EF	F	F			F	E		F					F	
	(G) JACK PADS, (H) TIE DOWN										GH		H				
	SPECIFICATIONS AND REFERENCE																
	(A) AIRPLANE DETAIL, (B) STRENGTH,	A	AB	AB	AB	A	A	A	A	A	AB	AB	AB	A	AB	AB	AB
	(C) DETAIL DESIGN, (D) TEST,	CD	D	CD	CD	CD	C	C			C	C	C		CD	CD	CD
	(E) DOUGLAS DESIGN EXPERIENCE		E	E	E	E	E	E	E	E	E	E	E	E	E	E	E

* Rolling radius - normal -flat tire

20.7 ASSEMBLY DRAWING CHECK LIST

Each assembly and installation drawing must be checked against this list by the draftsman before submitting the drawing for approval.

20.71 DRAFTING PRACTICES

a. Agreement With Layout Drawings must agree exactly with the layout, which should be complete in accordance with the Layout Check List. If the preparation of drawing necessitates a change in the layout, the change must be approved by the layout draftsman and checked against the Layout Check List. If time is not available actually to make the change, such change should be noted on the layout.

Assembly should be drawn from detail parts - this checks for obvious interferences and go-together.

b. Layout Numbers Layout number must be listed on all drawings.

c. Clearances External clearances should always be maximum possible rather than barely sufficient.

d. Identification Marking Nonstandard assemblies which resemble standard parts and might be replaced by standard parts should have their part numbers marked on the part, or the word "SPEC," if there is insufficient room for the number.

e. Purchased Parts Specification control drawings should be made for all vendor items of new design. See Section 20.9 and Standard Practice Bulletin AD2160. Specification drawings normally are not to be prepared for vendor's "shelf items" or minor modifications thereof; call out the vendor part number on the Douglas drawings. (Ref. Standard Practice Bulletin AD2183)

f. Tolerance Check Tolerances of all dimensions must be checked to ensure proper assembly. Assembly drawings provide a convenient means of checking the accumulated tolerances of the detail parts. The method of checking is shown in the Drafting Manual, Section 06.

20.72 STANDARD DESIGNS

a. Torque Notations of Joints Wrench torque for preload should be specified on the following joints:

1. Face seal joints . . . . Section 100

2. Any special joint requiring preload . . Section 30

b. Orifices Refer to Section 10.

c. Flexible Hose Specifications Refer to Section 40.

d. Line Clamps Spacing Line-Clamp maximum spacing is specified in Section 40.

e. Bearings and Bushings

1. Surfaces Threads or other irregularities should never come in contact with the bearing surface. The check is to be made under the conditions of the most unfavorable combinations of tolerances.

2. Alignment Two or more bushings or bearings on a common shaft should be machined (reamed) in line.

3. Oilite Oilite bushings and bearings should be burnished, not reamed.

4. Gasket Specifications Gaskets and mating parts should be as specified in Section 100.

20.727 PACKINGS

a. V-Ring V-ring Chevron packing and mating parts should be as specified in Section 100.

b. O-Ring O-ring packing and mating parts should be as specified in Section 100.

20.728 RETAINER RINGS SPECIFICATIONS Retainer rings and mating parts should be as specified in Standards Manual, Section 08.

a. Retainer Ring Limitations (Mandatory on military aircraft per MIL-H-8775 except where they are positively retained from being dislodged, retainer or snap rings shall not be used where failure of the ring will allow blow-apart of the unit by hydraulic pressure. Neither shall they be used where the buildup of clearances and tolerances will allow destructive end-play contributing toward failure of seals, brinelling, or fatigue.

20.73 MISCELLANEOUS ITEMS

20.731 LAPPED PARTS SPECIFICATIONS Lapped parts should be as specified in Section 70.

20.732 STAKING NOTES When parts are staked to lock, and mating parts are of different materials, a note STAKE TO LOCK IN ________ ONLY should be added. The staking should be done in the softer material to facilitate later disassembly.

20.733 STEEL BALLS Specify MS 19059 Grade 10 chrome alloy, or MS 19060 Grade 10 corrosion-resistant, steel balls for use in hydraulic check valves.

20.734 <u>CROSSED LINES AND CONTROLS</u> Every effort shall be made to assure that it will be physically impossible to incorrectly install cables, levers, bellcranks, hydraulic lines or any other parts that can cause malfunction.

20.735 <u>PIPING</u> All line routing should be sketched and/or mocked up to the satisfaction of the Group Engineer before affected drawings are released.

20.74 <u>ASSEMBLY DRAWING NOTES</u>

a. <u>General Process Standard</u> Call out DPS 3.391 or 3.334 in the general notes. For units containing Skydrol fluid: "THIS UNIT TO BE MANUFACTURED, ASSEMBLED AND TESTED PER DPS 3.391." For MIL-H-5606 mineral oil units: "THIS UNIT TO BE ASSEMBLED AND TESTED PER DPS 3.334."

b. <u>O-ring Control</u> For Skydrol resistant O-rings, the assembly drawing must call out sources and quality control per 7912037, as noted in Section 100.

c. <u>Adjustments</u> Required adjustments must be noted on assembly or installation drawings. If adjustment must be made on the airplane, the procedure shall be incorporated in the hydraulic and landing gear 7-size adjustment and checkout procedure for the airplane. Any such final adjustment information included on the installation drawing would be called out as "reference."

d. <u>Test Notes</u> Each production unit of a hydraulic assembly must be tested. Test notes shall be specified on the assembly drawing according to instructions given in Section 90, and with assistance and approval of the Test Group Engineer.

e. <u>Sequence of Notes</u> Where convenient, notes on the drawing should be in the sequence in which the operations are performed.

20.8 DETAIL DRAWING CHECK LIST

Each detail drawing must be checked against this list by the draftsman before submitting the drawing for approval.

20.81 DRAFTING PRACTICES

a. Detail Drawings Detail drawings must agree exactly with the layout, which should be complete and in accordance with the Layout Check List. If preparation of the detail drawing necessitates a change in the layout, the change must be approved by the layout draftsman and checked against the Layout Check List. If time is not available actually to make the change, such change should be noted on the layout.

b. Layout Numbers Layout numbers must be listed on all drawings.

c. Clearances Clearances should be maximum possible rather than sufficient only to provide clearance.

d. Drilled Holes On hydraulic parts, groups of drilled holes must be dimensioned so that (with all tolerances adverse) parts can be assembled. On other parts, consult the Drafting Manual.

e. Nonstandard Parts Nonstandard parts which resemble standard parts should be marked with the part number, or "SPEC" if there is not sufficient room for the part number. Impression stamping is preferred if function and strength are not impaired. Locate impression stamp in noncritical region.

f. Dimensions and Tolerances Dimensions and tolerances must conform to the standard designs whenever applicable. This includes nonstandard parts which are similar in any respect to standard parts.

20.82 STANDARD DESIGNS

a. Fits and Finishes Fits and finishes should be as specified in the applicable design section of this manual.

b. Lapped Parts Lapped parts should be as specified in Section 70.

c. Bearings and Bushings

1. Cadmium Plated Cadmium plate should not be used on any bearing which has relative movement.

2. Chamfered All bushings should be chamfered on OD and ID.

3. Shouldered Shoulder bushings should be dimensioned to maintain squareness between the shoulder and the bore.

d. Gaskets Gaskets and mating parts should be as specified in Section 100.

e. Packings

1. V-Ring V-ring Chevron packing and mating parts should be as specified in Section 100.

2. O-Ring O-ring packing and mating parts should be as specified in Section 100.

f. Retainer Rings Retainer Rings and mating parts should be as specified in Standards Manual Section 08.

g. Threads Threads should be in accordance with Standard Hydraulic Thread Series in Section 100 of this manual and in Drafting Manual Section 69.

20.83 MATERIALS AND PROCESSES

a. Stock Sizes The stock size listed should be one readily obtainable. (See Drafting Manual, Section 80.)

b. Finish Allowance Material must have sufficient finish allowance for machining with raw stock tolerances at extreme limits. This applies especially to tubing. (See Drafting Manual, Section 80.)

c. Heat Treatment Where steel parts require heat treatment and fatigue is a consideration, the minimum call-out on the drawing shall be 160,000 to 180,000 psi.

1. Vendor Heat Treated Material Material purchased heat treated should be so noted. (See Drafting Manual, Section 80.)

d. Finish Aluminum "ANODIZE per Mil-A-8625 Type II uncolored with dichromate seal (DPS 11.05)" should be specified on all parts which are threaded, serrated, or subject to wear or corrosion, to prevent substitution of less effective surface treatment. Hard anodize may be used with limitations specified in the Drafting Manual, Section 17.

e. Finish (Steel) "GREASE THOROUGHLY Per DPS 3.317" must be specified on all unprotected parts. CHROME PLATE must be specified where the part must resist wear and corrosion simultaneously. CAD PLATE may be used in direct contact with hydraulic fluid providing no wearing contact occurs. Parts not in contact with hydraulic fluid must be protected by CAD PLATE except as noted in 12.823.

f. Stress Inspection Magnaflux or magnetic inspection should be required only on highly stressed parts. See Drafting Manual, 17.34.

g. Fluid Fittings Special tube fittings shall be finished and identified per DPS 3.80.

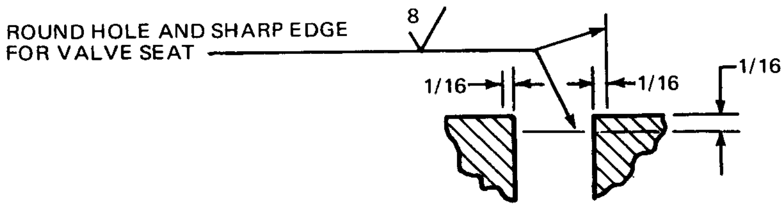
20.84 MISCELLANEOUS NOTES

a. Sequence Where convenient, notes on the drawing should be in the sequence in which the operations are performed.

b. Concentricity Concentricity notes and tolerances must be used on all concentric diameters. (See Drafting Manual 06.91.)

c. Perpendicularity Parallelism Not normally held very close by the general notes.

d. Valve Seats Valve seats shall be specified (ball or poppet type).



e. Flat and Square Surfaces For surfaces which must be straight, square, or flat and square for bearing or alignment, see notes in Drafting Manual 06.9.

20.9 SPECIFICATION CONTROL DRAWINGS

20.91 GENERAL Follow the instructions supplied with Specification Control Drawing formats. Drawing preparation methods are noted in Drafting Manual 11.412.

20.911 DESIGN APPROACH Sections 20.542 and 20.62 are helpful in the design work for specifications.

20.912 REFERENCES The following specifications should be reviewed, to assist specification drawing preparation:

a. Existing Douglas specification drawings.

b. General procurement specification 7912000.

c. Hydraulic component general specification MIL-H-8775.

d. Related military component and system specifications.

e. Environmental test general specification MIL-E-5272.

20.92 APPLICABLE DOCUMENTS The General Procurement Specification (7912000) must be listed under "Applicable Documents," and need not again be referred to under "Requirements" unless deviations from this specification are required. The other items listed under "Applicable Documents" must be referenced in the "Requirements" section (or subsequent sections) to define how they are to be applied.

20.921 GENERAL PROCESS STANDARD One of the following DPS's must be specified: "Douglas Process Standard 3.391, Skydrol Hydraulic Systems for Airplanes"; or "Douglas Process Standard 3.334, Hydraulic Assemblies" for units operating with petroleum base fluid. In the "Requirements" section, include the following wording:

a. "This unit shall be manufactured, assembled and tested per DPS 3.391."

or

b. "This unit shall be assembled and tested per DPS 3.334."

20.922 O-RING CONTROL For any Skydrol unit which may incorporate O-rings, specify: "7912037, NAS1611 and NAS1612 Skydrol O-ring Seal - Procurement Specification," and include under "Requirements": "Skydrol resistant O-ring seals are to be called for on drawings by NAS1611 and NAS1612 numbers. Procurement sources and quality control of NAS1611 and NAS1612 O-rings are to be per Douglas Specification 7912037."

20.923 FLUIDS For specification drawings on commercial models, specify "Fluids Per DMS 2014."

20.93 TESTS Refer to Section 90 in conjunction with the following:

20.931 PRODUCTION ARTICLE TESTS

a. The statement, "Each production article shall be tested by the supplier," must be included.

b. Production tests are usually dependent on the details of the particular vendor's design. In many cases, the vendor should be directed to prepare the production test procedure and submit it for approval at a specific time.

20.932 QUALIFICATION TESTS

a. As indicated in MIL-H-8775, the normal method for qualification of hydraulic units is preparation of one test sample to minimum clearance and, where applicable, maximum packing friction; and a second sample to maximum clearances conducive to possible malfunction due to wear. Neither sample would be subjected to the entire series of tests.

b. Endurance testing often results in appreciable wear and performance degradation (leakage, sensitivity, etc.). Performance changes also are likely to occur during and after environmental testing. These situations must be recognized in design, and the limitations on such changes must be defined in the specification drawing.

c. MIL-E-5272 describes numerous methods for performing environmental tests with varying degrees of severity. Select the appropriate tests to simulate the anticipated service environments.

21.0 MISCELLANEOUS DRAFTING INFORMATION

21.1 NOTES ON DRAWINGS

21.11 PROCESS NOTES The following table lists some new processes not clearly defined elsewhere. For a general list of process notes, see Drafting Manual Section 17.38.

PROCESS	SPEC	DRAWING REQUIREMENTS — DETAIL DRAWING	DRAWING REQUIREMENTS — SPECIFICATION DRAWING
1. AGING – ARTIFICIAL (24 S ALUM)	DPS 7.00-1	AGE PER DPS 7.00-1 (WAS DPS 7.31)	UNIT SHALL BE FINISHED IN ACCORDANCE WITH FINISH SPEC DPS 30.106.
2. ANODIZE – SULFURIC USE ONLY FOR HYDRAULIC OIL CONTACT OR FOR CORROSION RESISTANCE.	MIL-A-8625 TYPE II DPS 11.05	ANODIZE PER MIL-A-8625 TYPE II UNCOLORED WITH DICHROMATE SEAL (DPS 11.05)	UNIT SHALL BE FINISHED IN ACCORDANCE WITH FINISH SPEC DPS 30.106.
3. ANODIZE – HARD (HARD COAT)	DPS 11.04	HARD COAT – THICK PER DPS 11.04	NONE
4. CHROME PLATE	MIL-P-6871 DPS 9.71		NONE
5. CHROME PLATE-THIN (ELECTRO-CHROME)	DPS 9.84	ELECTRO-CHROME PLATE PER DPS 9.84. (THICKNESS IS 0.0001 TO 0.0002 INCH UNLESS SPECIFIED OTHERWISE)	NONE
6. CHEMICAL NICKEL PLATE	DPS 9.67		NONE
7. ELECTRIC ETCH VERSUS IMPRESSION STAMP	DPS 3.02	FOR ALUMINUM ALLOYS: SEE DRAFTING MANUAL 17.352. FOR STEEL HT 180,000 PSI OR ABOVE: IMPRESSION STAMP NOT PERMITTED. MARK HERE PER DPS 3.02. FOR FORGINGS: IMPRESSION STAMPS PERMITTED IN AREAS WHERE MATERIAL IS LATER REMOVED BY MACHINING.	NONE
8. HEAT TREAT-SAE 4340 HI-HT 260,000 TO 280,000 PSI	DPS 4.804	HEAT TREAT 260,000 TO 280,000 PSI AND PROCESS PER DPS 4.804 UNLESS OTHERWISE NOTED.	NONE
9. LACQUER – HYDRAULIC UNIT (HYDROLUBE RESISTANT)	MIL-L-7146 DPS 4.50-47	NONE 1. VENDOR MANUFACTURING OUTLINES SHALL REQUIRE FINISH PER DPS 30.102. 2. DOUGLAS-FABRICATED PARTS TO BE COVERED BY AIRPLANE FINISH SPEC (FS251, ETC.)	UNIT SHALL BE FINISHED IN ACCORDANCE WITH FINISH SPEC DPS 30.106.
10. PRIMER – ZINC CHROMATE	MIL-P-6889 TYPE I DPS 4.50-3	NONE SAME AS 9 ABOVE.	SAME AS 9 ABOVE.
11. PRETREATMENT COATING	MIL-C-15328 DPS 4.50-33	NONE SAME AS 9 ABOVE.	SAME AS 9 ABOVE.
12. SHOT PEENING	DPS 4.999	SHOT-PEEN (ALL OR NOTED) SURFACES PER DPS 4.999. (INTENSITY OF PEENING TO BE ESTABLISHED BY PROCESS ENGINEER).	NONE
13. TIN PLATE	AMS 2408-2	FOR STEEL SPRINGS IN CONTACT WITH HYDRAULIC FLUID: TIN PLATE PER AMS 2408-2. (THICKNESS IS DESIGNATED BY –2 WHICH IS 0.0002 +0.0002/–0.0000 THICK).	ALL SPRINGS USED IN THE UNIT TO BE TIN PLATED IN ACCORDANCE WITH AMS 2408-2.

22.0 INSTALLATION CRITERIA

22.1 COMPONENT ACCESSIBILITY Components requiring frequent maintenance need good accessibility. Items such as reservoirs, accumulators, and filters must have unusually good access for adequate servicing.

22.2 COMPONENT MOUNTING Components which require frequent maintenance such as filters, pneumatic chemical driers, etc., should be rigidly mounted.

22.3 COMPONENT LOCATIONS Consider avoiding natural dirt-collecting areas, such as brake valves under cockpit floor, in system installation planning.

22.4 DUAL-SYSTEM LINES SUBJECT TO THE SAME ADVERSE ENVIRONMENT An example of such a condition is the routing of lines through the wheelwell where leaking air causes strumming of both the normal and emergency lines. Attaching lines to the same panel that may vibrate has a like effect. Proximity to a common fire hazard must also be considered. Consider all environmental hazards in separating lines.

22.5 INSTALLATIONS NEAR HOT-AIR LINES Temperature-aging of elastomers in hydraulic, pneumatic, fuel, or electrical systems can occur when near engine hot air bleed lines. Leaks, pressure loss, or electrical shorts may result. Exercise care in locating these systems and hot air lines, particularly near direct impingement of hot air.

22.6 AVOID ELBOW BOSS CONNECTIONS Make every effort to point fluid ports of hydraulic units in the direction of the attached tubing or hose such that straight fitting may be used instead of the relatively unreliable MS elbow.

22.7 LINE DAMAGE Avoid installing hydraulic components and lines where they may be walked on. However, combat vulnerability must be considered.

22.8 ELECTRICAL AND AIR CONDITIONING EQUIPMENT Locate hydraulic equipment and piping below electrical conduit and air conditioning ducts.

30. STRENGTH ANALYSIS

CONTENTS

31. DEFINITIONS

31.1 LIMIT LOAD (Synonymous with Applied Load) The maximum load anticipated and at which the Yield Point of the material is never to be exceeded. In some cases, it is obtained directly, while in others it is obtained by multiplying the given load by a load factor.

.11 LOAD FACTOR A factor by which the steady forces are multiplied to obtain the equivalent static effect of dynamic forces acting on the airplane, as aerodynamic forces, inertia forces, or ground or water reactions.

.12 YIELD STRENGTH OR YIELD POINT That stress at which a small amount of yielding occurs as defined by MIL-HDBK-5.

.2 WORKING LOAD (Synonymous with Operating Load) The normally encountered load to which a part is subjected.

.3 ULTIMATE LOAD (Ref. MIL-A-8860) Obtained by multiplying the Limit Load by the Ultimate Factor of Safety, except for loading conditions for which specific Ultimate Loads are delineated and except for landing loads (Ref. MIL-A-8862 and MIL-A-8863). Failure shall not occur at the Ultimate Load.

.31 ULTIMATE FACTOR OF SAFETY For the design of structure is usually 1.5. When additional strength is desired, multiplying special factors of safety are specified to provide added safety, rigidity, quality assurance, wear, and to compensate for uncertainties in design.

.4 ALLOWABLE ULTIMATE STRESS Values used are to be those specified in MIL-HDBK-5 or obtained from the Aero-structural Mechanical Group.

.5 DESIGN ULTIMATE STRESS A calculated stress in a part which is obtained by using the Ultimate Load.

.6 MARGIN OF SAFETY That margin by which the actual strength exceeds the required strength. Refer to MIL-HDBK-5, Failure Under Combined Loadings.

$$\text{MARGIN OF SAFETY} = \frac{\text{ALLOWABLE ULTIMATE STRESS}}{\text{DESIGN ULTIMATE STRESS}} - 1$$

.61 REQUIRED MARGIN OF SAFETY

$$\text{REQUIRED M. S.} \geq 0$$

.7 CRITICAL CONDITION The design loading condition for which the Margin of Safety indicates the component is most likely to fail.

31.8 SPECIAL FACTORS Where there is uncertainty concerning the actual strength of a particular part or where the strength is likely to deteriorate in service prior to normal replacement of the part or where the strength is subject to appreciable variability due to uncertainties in manufacturing processes and inspection methods the normal factor of safety is multiplied by a special factor of a value such as to make the probability of the part being understrength from these causes extremely remote.

a. Casting Factor - Generally 2.0 to 1.25, depending on inspection and test methods.

Commercial Transports
See Fed. Aviation Regulation 25.621.

USAF and/or Navy Contracts - Ref. MIL-A-8860
Unless the strength of a casting is substantiated by repeated-load and failure-load structural tests of the casting which simulate the design conditions critical for the casting, or unless the casting is procured to the requirements of specification MIL-C-21180, an analytical positive margin of safety not less than 0.25 is required.

b. Bearing Factors - Ref. DAC Aerostructural Mechanical Group.

Commercial, USAF, and/or Navy Contracts
Bearing stresses based on Ultimate Loads are not to exceed the allowables in MIL-HDBK-5 and are not to be such as to produce excessive wear of the bearing. Bearing factors are to be of sufficient magnitude to provide for the effects of normal relative motion between parts and in joints with clearances (free fit) which are subject to pounding or vibration. A bearing factor need not be employed on a part if another special factor prescribed is of greater magnitude than the bearing factor.

c. Fitting Factor

Commercial Transports - Ref. Fed. Aviation Regulation 25.625.
A fitting factor of at least 1.15 is to be used on all fittings the strength of which is not proven by limit and ultimate load tests in which the actual stress conditions are simulated in the fitting and surrounding structure. This factor is to apply to all portions of the fitting, the means of attachment and the bearings on the members joined. A fitting factor need not be employed with respect to the bearing surface of a part if the bearing factor used is of greater magnitude than the fitting factor.

USAF and/or Navy Contracts - Ref. MIL-A-8860
No fitting factor is required. Douglas Practice is to use a fitting factor as outlined above under commercial transports.

32. GENERAL STRENGTH REQUIREMENTS

GENERAL Hydraulic system components and attaching linkages should be designed to meet the most critical loads or combination of loads. Load factors or design factors have been established for systems in order to insure adequate safety and life of the parts. These load factors, in some cases, must be met to comply with specification requirements, and in other cases are arbitrarily assigned. The following basic criteria have been considered in establishing these design factors:

a. The structure must not fatigue under normal working loads The unit stress under normal working loads must be limited such as not to exceed the fatigue strength of the material for the anticipated life of the structure with stress concentration factors taken into consideration. The allowable working stress to be used should be determined as outlined in Section 32.3 with specific approval of the Group Engineer.

b. The structure must not yield under maximum expected loads The unit stress at the limit load must not exceed the yield strength of the material. It should be realized that, in some cases, test loads may be imposed on the structure which may exceed the maximum limit load encountered on installation. No part of the structure shall take any permanent set or be damaged in any manner when subjected to the applicable test loads.

c. The structure must not fail at the ultimate load The unit stress at the ultimate load must not exceed the ultimate strength of the material. Where applicable, ultimate strength of material should be determined by bending or torsional modulus of rupture.

In addition to the above requirements, consideration should be given to the following:

a. Deflection of parts must not cause malfunctioning Deflection rather than strength often is the criteria for the design of parts. When this is the case, the design should be based on limit loads rather than ultimate loads.

b. Temperature variations shall not cause malfunctioning or excessive stress Consideration should be given to expected temperature variation on the parts so that no binding, sticking, or malfunctioning shall result. Internal stresses, such as those resulting from use of unlike materials in combination, shall not exceed allowable stresses under most adverse conditions. Where parts are expected to operate at extremely high temperatures, allowable unit stresses may be reduced. Consult the Group Engineer for establishing design temperature range and design criteria for hydraulic system and components within the established range.

For cases for which no factors have been established, or it is considered that designated factors are insufficient, load factors may be established or reconsidered with the approval of the Group Engineer and by observing the general rules above.

32.1 DESIGN CONDITIONS Hydraulic system design pressures (operating, proof test, and burst) are to be determined in accordance with the following conditions presented in sections 32.1 and 32.21. Design for the most critical condition.

.11 PRESSURE REQUIREMENTS, MILITARY SPECIFICATION - Ref. MIL-H-5440D, Table I.

CHARACTERISTICS	NOMINAL OPER. PRESS. CLASS 1500	CLASS 3000	PERCENT OPERATING PRESSURE	REMARKS STANDARD SYSTEM PRESSURE
AUTOMATIC PRESSURE REGULATOR – ACCUMULATOR SYSTEM (CLOSED CENTER)				
A. REGULATOR CUTOUT PRESSURE	1500	3000		SYSTEM DESIGN PRESSURE
B. UPPER LIMIT OF OPERATING PRESSURE FOR ALL UNITS (REGULATOR CUT-IN PRESSURE).	1250	2600		
C. MAX. SYSTEM RELIEF VALVE SETTING, AT MAX. SYSTEM FLOW.	1900	3850		SPECIFICATION MIL-V-8813 OR MIL-V-5523 UNITS.
OPEN-CENTER TYPE SYSTEM (CONSTANT DELIVERY PUMP)				
A. SYSTEM DESIGN PRESSURE	1500	3000		RELIEF VALVE FULL-FLOW SETTING
B. UPPER LIMIT FOR FULL FLOW OPERATING PRESSURE.	1350	2700		
C. SYSTEM RELIEF VALVE SETTING. AT MAX. SYSTEM FLOW.	1500	3000		SPECIFICATION MIL-V-5523 OR MIL-V-8813 UNITS.
CLOSED- OR OPEN-CENTER SYSTEM (VARIABLE VOLUME PUMP)				
A. PUMP UNLOADING PRESSURE	1500	3000		SYSTEM DESIGN PRESSURE
B. MAXIMUM LIMIT OF FULL FLOW SYSTEM PRESSURE.	1450	2950		
C. MAXIMUM SYSTEM RELIEF VALVE SETTING AT MAX. SYSTEM FLOW.	1900	3850 *		SPECIFICATION MIL-V-5523 OR MIL-V-8813 UNITS.
THERMAL RELIEF VALVE SETTING				
A. EQUAL TO SYSTEM RELIEF VALVE MAX. SETTING PLUS VALUES NOTED.	150	150		
PROOF PRESSURE (MINIMUM)				
A. LINES AND FITTINGS	3000	6000	200	HOSE PROOF PRESSURE TO BE IN
B. HOSE	3000	6000	200	ACCORDANCE WITH APPLICABLE DETAIL SPECIFICATIONS.
C. UNITS UNDER AIR AND OIL PRESS.	3000	6000	200	SPECIFICATION MIL-A-5498
D. UNITS NORMALLY UNDER OIL PRESSURE ONLY.	2250	4500	150	
E. PARTS OF SYSTEM UNDER ATMOSPHERIC PRESSURE OR SUCTION.	50	50		EXCEPT RESERVOIR AND PUMP SEAL CHAMBERS
F. PARTS OF SYSTEM SUBJECT TO BACK PRESSURE ONLY.	150 % OF ACTUAL MAX. PRESS.	150 % OF ACTUAL MAX. PRESS.		EXCEPT HOSE WHICH SHALL BE 250 % OF ACTUAL MAXIMUM PRESSURE.
BURST PRESSURE (MINIMUM)				
A. LINES, HOSES, AND FITTINGS	6000	12000 ⌖	400 ⌖	HOSE BURST PRESSURE TO BE IN ACCORDANCE WITH APPLICABLE DETAIL SPECIFICATION.
B. UNITS CONTAINING AIR AND OIL UNDER PRESSURE.	6000	12000	400	SPECIBICATION MIL-A-5498
C. UNITS NORMALLY UNDER OIL PRESS. ONLY.	3750	7500	250	
D. PARTS OF SYSTEM SUBJECT TO BACK PRESSURE ONLY.	300 % OF ACTUAL MAX. PRESS.	300 % OF ACTUAL MAX. PRESS.		EXCEPT HOSE WHICH SHALL BE 500 % OF ACTUAL MAXIMUM PRESSURE.
COLLAPSE PRESSURE OF PARTS SUBJECT TO SUCTION.	50 EXTERNAL	50 EXTERNAL		

* For commercial transports, system relief valves are normally set to 3400 ± 50 psi crack with rated flow pressure 3700 psi max.

⌖ For 3000 psi operating pressure, Douglas practice is to use tubing with approximately 15,000 psi min. burst strength which is a factor of 500%.

32.12 STRENGTH REQUIREMENTS, MILITARY SPECIFICATION - Ref. MIL-H-5440D, 3.5.4

"3.5.4.1 Additional loads. All hydraulic systems and components which are subjected, during operation of the aircraft, to structural or other loads which are not of hydraulic origin, shall withstand such loads when applied simultaneously with appropriate proof pressure as specified in table I, without exceeding the yield point at the maximum operating temperature."

"3.5.4.2 Accelerated loads. Actuating cylinders and other components and their attaching lines and fittings, subject to accelerated loads, shall be designed and tested on the basis of a pressure equal to the maximum pressure that will be developed, without exceeding the yield point at the maximum operating temperature."

32.13 STRENGTH REQUIREMENTS, FED. AVIATION REGULATIONS - Ref. FAR 25.1435, (a), (1), 7/27/67

"Each element of the hydraulic system must be designed to withstand the design operating pressure loads in combination with limit structural loads which may be imposed without deformation that would prevent it from performing its intended function, and to withstand, without rupture, the design operating pressure loads multiplied by a factor of 1.5 in combination with ultimate structural loads that can reasonably occur simultaneously. Design operating pressure is maximum normal operating pressure, excluding transient pressure."

32.2 DESIGN LOADS are to be determined in accordance with the following conditions. The most critical are to be used.

32.21 PRESSURE LOADS, DOUGLAS REQUIREMENTS Ultimate loads originating from hydraulic pressure are to be obtained in accordance with the following criteria. Where so noted, an additional 1.15 or larger fitting factor is to be used at stress condentration points.

32.211 HYDRAULIC UNITS

.2111 2.5 x SYSTEM NOMINAL OPERATING PRESSURE

2.5 = 1.67 hyd factor x 1.5 mat'l factor
Consider 1.67 hyd factor = 1.15 fitting factor x 1.45 to account for pressure surges and repeated stress.
Normally hydraulic units are tested to at least 1.5 times system pressure, so Test Margin = 2.5/(1.5 x 1.5 mat'l factor) - 1 = 0.11.

.2112 2.0 x SYSTEM RELIEF VALVE NOMINAL PRESSURE (at applicable relief flow rate)

2.0 = 1.33 hyd factor x 1.5 mat'l factor
Consider 1.33 hyd factor = 1.15 fitting factor x 1.16 to account for pressure surges and tolerance on valve setting.
Normally units are tested to at least 1.25 times nom. relief pressure, so Test Margin = 2.0/(1.25 x 1.5 mat'l factor) - 1 = 0.07.

32.2113 1.875 x THERMAL RELIEF VALVE NOMINAL SETTING (at crack)

1.875 = 1.25 hyd factor x 1.5 mat'l factor
Consider 1.25 hyd factor = 1.15 fitting factor x 1.09 to account for tolerance on valve setting.
Normally applicable units are tested to at least 1.125 times nom. thermal relief pressure, so Test Margin = 1.875/(1.125 x 1.5) - 1 = 0.11.

.2114 1.5 x MAX. TEST PRESSURE (if, for test convenience, test pressure exceeds the above mentioned values)

Use fitting factor; see 32.2117.

.2115 1.67 x MAX. ACTUAL PRESSURE (due to accelerations, pressure boost, etc.)

1.67 = 1.11 hyd factor x 1.5 mat'l factor
Use fitting factor; see 32.2117.
For infrequent applications, this 1.67 factor may be reduced, based on thorough endurance strength evaluation.

.2116 RETURN CHAMBERS

a. 0.8 x SYSTEM NOMINAL OPERATING PRESSURE
Use fitting factor; see 32.2117

b. 3.0 x ACTUAL MAX. STEADY RETURN LINE PRESSURE
3.0 = 2.0 hyd factor x 1.5 mat'l factor

c. 2.0 x ACTUAL PEAK PRESSURE (due to return line surges, accelerations, pressure boost, etc.)
2.0 = 1.33 hyd factor x 1.5 mat'l factor

d. 1.5 x MAX. TEST PRESSURE (if proof pressure is high for test convenience) Use fitting factor; see 32.2117.

.2117 FITTING FACTOR Hydraulic factors of 1.25 or more can be considered to include a 1.15 fitting factor. With hydraulic factors less than 1.25, an appropriate fitting factor must also be used at stress concentration regions.

.212 TUBING AND FITTINGS Use tubing listed in Section 40, which normally is based on the following:

.2121 PRESSURE LINES

a. 7.5 x SYSTEM NOMINAL OPERATING PRESSURE Only for piston-engine commercial aircraft pump pressure lines.

b. APPROX. 5.0 x SYSTEM NOMINAL OPERATING PRESSURE

c. 3.33 x CRITICAL RELIEF PRESSURE

d. 2.0 x MAX. ACTUAL PRESSURE (due to accelerations, pressure boost, etc.)

32.2122 RETURN LINES

a. 1.6 x SYSTEM NOMINAL OPERATING PRESSURE

b. 5.0 x ACTUAL MAX. STEADY BACK PRESSURE

c. Minimum wall size should conform to that tabulated in Section 40, for ruggedness and to prevent instability failure.

.2123 SUCTION LINES

a. 5.0 x MAX. RESERVOIR OPERATING PRESSURE

b. 150 psi internal pressure

c. 100 psi external pressure

d. Minimum wall size should conform with Section 40.

.213 HOSES

.2131 BURST PRESSURE varies with hose size and may be as high as 10 times system pressure in -4 size to as low as 3.33 times in -12 size 3000 psi hoses. The results of impulse testing with water-hammer peaks of 1.5 x base pressure are of more significance in establishing adequate margin of safety than burst margin of safety.

.2132 5.0 x ACTUAL MAX. STEADY BACK PRESSURE in return lines.

.214 EMERGENCY UNITS which are subject to load only during emergency use.

.2141 1.875 x EMERGENCY SYSTEM PRESSURE Where a relief valve is used to limit max. pressure use 1.875 x full flow relief pressure.

.2142 1.5 x MAX. FULL FLOW RELIEF PRESSURE An appropriate fitting factor must also be used at stress concentration regions.

.2143 1.5 x MAX. TEST PRESSURE An appropriate fitting factor must also be used at stress concentration regions.

.215 PNEUMATIC UNITS

.2151 Refer to MIL-C-7905 and MIL-R-8573 for air storage containers subject only to static pressure.

.2152 For other units, refer to general pneumatic specifications MIL-P-5518 and MIL-P-8564, and follow Douglas practice for hydraulic units.

32.216 CONNECTING MECHANISMS for actuating cylinders shall be designed for endurance strength under working loads, yield strength under limit loads, and ultimate strength under ultimate loads, rather than for burst pressure. Where an actuating cylinder is not stopped internally, the connecting mechanism shall be capable of withstanding "harnessed" proof pressure loads carried to the stopping points without permanent deformation.

.2161 THE ACTUATING CYLINDER also must not yield under the "harnessed" proof pressure tension and compression loads, whether or not it stops internally. Under compression, the cylinder yield load usually is also the ultimate load producing collapse of the cylinder as a column. See Section 60 for the cylinder column design load.

.217 JAMMING When an actuating cylinder is attached to a linkage which can possibly jam due to adverse friction, eccentricity, deflection, etc., strength analysis of the mechanism shall be based on limit loads rather than ultimate loads. Points of high stress concentration and end moments occurring at plain bearings must be carefully considered. Loads in the mechanism must not exceed the yield strength under the following conditions:

a. Where the linkage can remain jammed with system pressure in the cylinder: limit load shall be that due to system relief pressure.

b. Where jamming will clear at less than system pressure: limit load shall be that due to the max. pressure to overcome jamming.

c. Where deflections allow the cylinder to bottom: limit load shall be that due to max. pressure causing bottoming.

d. Where load paths are divided in a linkage system, each branch shall be analyzed for its most severe loading.

.22 EXTERNAL FORCE LOADS

Ultimate Loads originating from external forces shall be obtained as follows. Where so noted, an additional 1.15 or larger fitting factor shall be used at stress concentration points. Also refer to the MIL-A-8860 series of specifications and Fed. Aviation Regulations, Part 25, Subpart C.

Fatigue must always be considered and can be more critical than the load factors listed below. For example, where the normal working load is frequently applied and is close to the maximum limit load, as often occurs with flap actuators, fatigue strength is critical; and, in effect, the general flight control factor of 2.5 must be multiplied by an additional factor related to stress concentration and the required endurance life.

32.221 HYDRAULIC BUNGEE LOADS

1.75 x maximum hydraulic bungee limit load. (Use fitting factor.) The 1.75 factor consists of a 1.5 material factor multiplied by a 1.17 hydraulic factor.

.222 FLIGHT CONTROL LOADS (Incl. wing flap and dive brake)

2.5 x normal working flight control load.
1.875 x max. limit load. (Consists of 1.5 x 1.25 control system factor.)

.223 LANDING OR FLIGHT LOADS OF +1G

2.5 x max. limit load.

.224 AIR LOADS ON DOORS, COWL FLAPS, ETC.

2.5 x normal working load.
1.5 x max. limit load. (Use fitting factor.)

.225 ACCELERATED FLIGHT LOADS

1.5 x max. limit load. (Use fitting factor.)
1.67 x max. pressure that accelerated load will develop, regardless of relief valve setting, for component and cylinder burst pressure. (Use fitting factor.) For infrequent applications, this 1.67 factor may be reduced, based on thorough endurance strength evaluation.

.226 TOWING LOADS

Refer to MIL-A-8862 and Fed Aviation Regulations 25.509 and 25.511.

.227 HANDLE LOADS

a. Hand Pumps Hand operated pumps shall be designed for the following ultimate handle loads, and in each case the load given is for each man pumping: 300 pounds in the direction of the motion of the pump handle and 150 pounds load applied axially and sideways. The handle, pivot, links, piston, body, stops, supporting structure, etc., should be analyzed for the above loadings. The loads should be considered as being applied separately.

b. Other See MIL-A-8865 and Fed. Aviation Regulation 25.405 for pilot applied loads.

32.3 <u>DESIGN FACTORS FOR REPEATED LOADING</u> The following data is to be used in determining design factors where none have been established.

When a part is loaded repeatedly to a high stress, fatigue failure may result. Fatigue failures tend to increase with increase in stress at cycling load, with increase in number of cycles, and with number of stress-raisers, i.e., notches, etc., in the part.

Figure 30-1 gives a curve of stress (in percent ultimate) plotted against cycles to failure (at "working" or cycling load). The curve "typical aircraft fittings" is an average curve from test results of a number of fittings having an average degree of stress concentration. The scatter of these tested parts from the plotted line was on the order of ±25 percent stress. Two extreme lines have also been included for parts having higher than average stress concentrations, and for parts having lower than average stress concentrations. These curves cannot be used to predict the fatigue life of a part because, in airplane service, values of "working load", number of cycles at working load, and degree of stress concentration are only known very approximately.

However, these curves can be used in design to guide the designer in the amount of margin he should allow over the usual 1.5 material factor and 1.15 fitting factor, in the case of parts subject to repeated loads which are a high percentage of their ultimate strength or for parts subjected to high frequency vibratory loads.

In general, parts loaded by pressure, such as flaps subject to air loads, fuselages subjected to pressure loads, and parts in hydraulic systems subjected to hydraulic pressure, where the part works to its maximum (applied) load on each cycle, or parts subjected to vibratory loads, are the only parts which should have extra margins in accordance with page 32.321.

32.31 <u>USE OF FIGURE 30-1</u> Parts loaded repeatedly to a high percentage of their ultimate strength or subjected to vibratory loads shall have a total factor between "working load" and ultimate strength as shown in Figure 30-1.

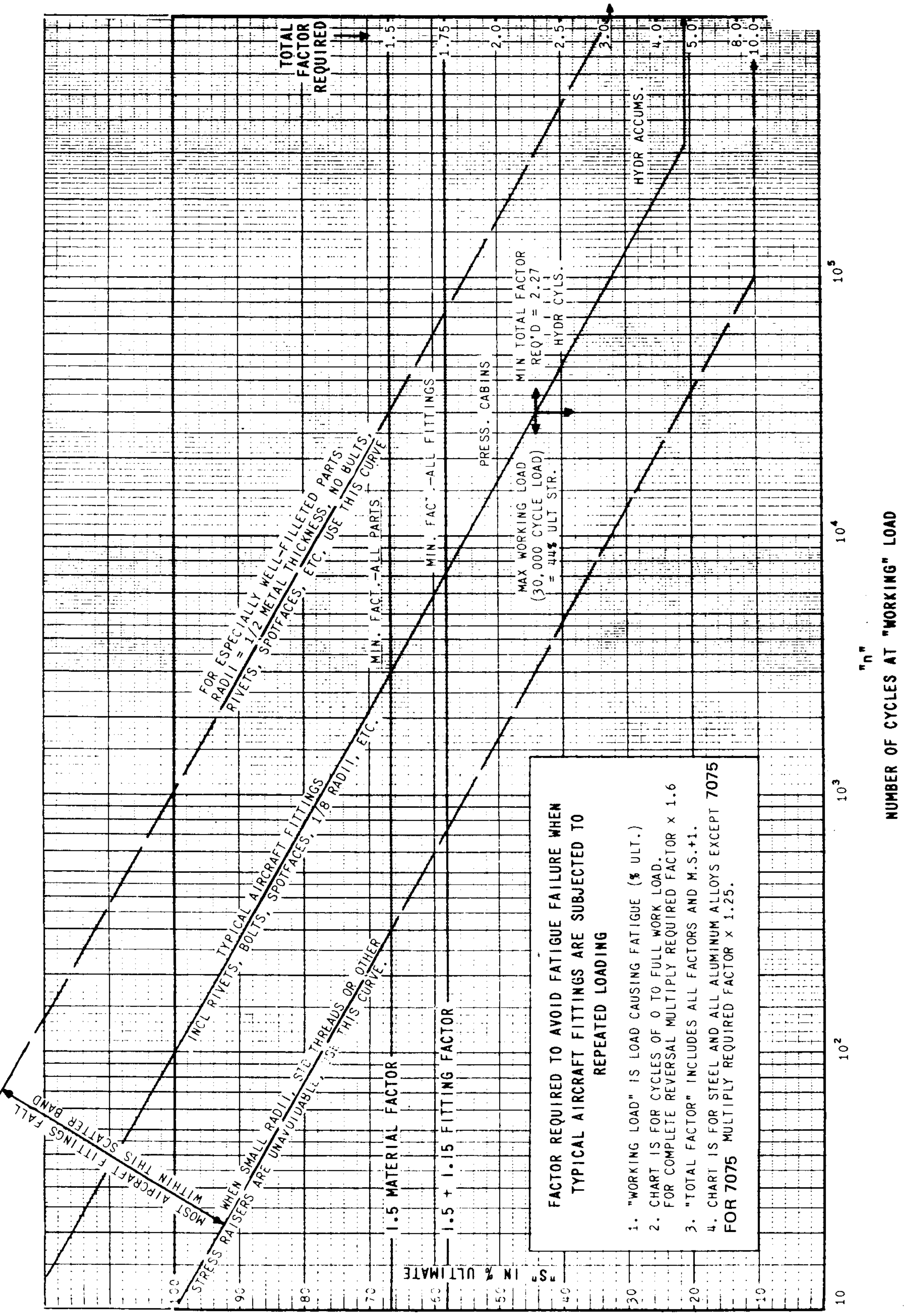
NOTE: This is usually not applicable to parts loaded by aerodynamic gust or by accelerated flight loads because these parts usually have enough factor.

The stress used shall be the stress (or load) at "working load" in percent of ultimate strength.

The number of cycles shall be computed approximately for the desired life of the part.

Working load is the normal operating load, or the load at which the part will be subjected to "n" cycles.

FIGURE 30-1


TOTAL FACTOR REQUIRED
1.5
1.75
2.0
2.5
3.0
4.0
5.0
8.0
10.0
FOR ESPECIALLY WELL-FILLETED PARTS, RADII = 1/2 METAL THICKNESS, NO BOLTS, RIVETS, SPOTFACES, ETC. USE THIS CURVE
TYPICAL AIRCRAFT FITTINGS INCL RIVETS, BOLTS, SPOTFACES, 1/8 RADII, ETC.
WHEN SMALL RADII, STD THREADS OR OTHER STRESS RAISERS ARE UNAVOIDABLE, USE THIS CURVE.
MOST AIRCRAFT FITTINGS FALL WITHIN THIS SCATTER BAND
MIN. FACT.-ALL PARTS
MIN. FACT.-ALL FITTINGS
1.5 MATERIAL FACTOR
1.5 + 1.15 FITTING FACTOR
PRESS. CABINS
MAX WORKING LOAD (30,000 CYCLE LOAD) = 44% ULT STR.
MIN TOTAL FACTOR REQ'D = 2.27 HYDR CYLS.
HYDR ACCUMS.
FACTOR REQUIRED TO AVOID FATIGUE FAILURE WHEN TYPICAL AIRCRAFT FITTINGS ARE SUBJECTED TO REPEATED LOADING
1. "WORKING LOAD" IS LOAD CAUSING FATIGUE (% ULT.)
2. CHART IS FOR CYCLES OF 0 TO FULL WORK LOAD. FOR COMPLETE REVERSAL MULTIPLY REQUIRED FACTOR x 1.6
3. "TOTAL FACTOR" INCLUDES ALL FACTORS AND M.S.+1.
4. CHART IS FOR STEEL AND ALL ALUMINUM ALLOYS EXCEPT 7075
FOR 7075 MULTIPLY REQUIRED FACTOR x 1.25.
"S" IN % ULTIMATE
100
90
80
70
60
50
40
30
20
10
10
10^2
10^3
10^4
10^5
"n"
NUMBER OF CYCLES AT "WORKING" LOAD

The approximate curve for "typical" fittings, poor fittings, or especially good fittings shall be used. Points may be interpolated if desired. Note that the chart is for cycles of 0 to full (working) load. For complete reversal (including vibratory loads), the required factor must be multiplied by 1.6.

32.32 PROCEDURE

a. Find the intersection of the number of cycles desired life with the line representing the type of part (typical, poor, good).

b. Read horizontally from this point the highest permissible working stress (or load) in percent ultimate and the minimum total factor required.

c. Multiply the factor by 1.25 if 7075; multiply the factor by 1.6 if load completely reverses in each cycle.

NOTE: The "total factor" includes the 1.5 material factor, fitting factor, control system factor, etc., if used, and the M.S. + 1, i.e., a 25 percent M.S. is treated as an additional 1.25 factor. (Casting factor of 1.25 is not included since this represents the reduced strength of the actual casting from the test bar value. Casting factors above 1.25 may be included in the "total factor".)

33. DESIGN CRITERIA FOR ASSEMBLIES

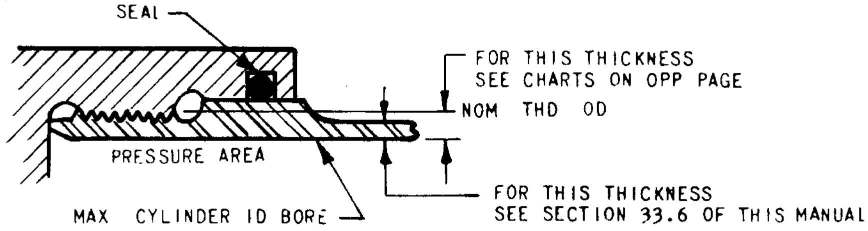
33.1 AXIALLY LOADED THREAD JOINTS (END CAP TO CYLINDER)

.11 CYLINDER WALL THICKNESS AT JOINT

.111 PRESSURE ON THREADS

ILLUSTRATION:

Δ



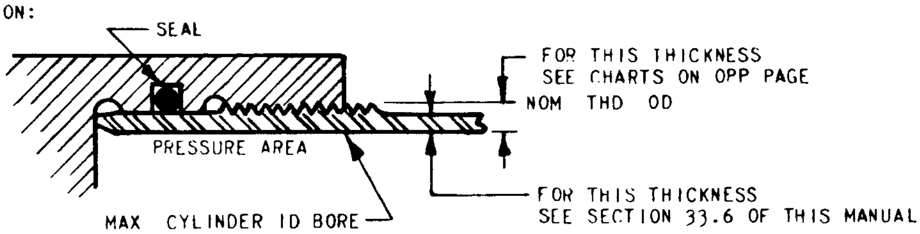
METHOD: (REFER TO THREAD CHARTS ON OPPOSITE PAGE - USE SOLID LINES)
SELECT PROPER CHART. ENTER CHART AT CYLINDER BORE SIZE. PROJECT A LINE UP TO DESIRED DESIGN PRESSURE LINE. THEN READ ACROSS FOR WALL THICKNESS.

EXAMPLE: A 4 INCH CYLINDER BORE WITH A DESIGN PRESSURE OF 12,500 PSI WILL REQUIRE A 95,000 PSI STEEL WALL CYLINDER THICKNESS OF 7/16 INCH.

.112 PRESSURE NOT ON THREADS

ILLUSTRATION:

Δ



METHOD: (REFER TO THREAD CHARTS ON OPPOSITE PAGE - USE DASH LINES)
SELECT PROPER CHART. ENTER CHART AT CYLINDER BORE SIZE. PROJECT A LINE UP TO DESIRED DESIGN PRESSURE LINE. THEN READ ACROSS FOR WALL THICKNESS.

EXAMPLE: A 3 INCH CYLINDER BORE WITH A DESIGN PRESSURE OF 12,500 PSI WILL REQUIRE A 125,000 PSI STEEL WALL CYLINDER THICKNESS OF 7/32 INCH.

.113 MISCELLANEOUS DATA

a. All design pressures are based on ultimate strength and are obtained as described in Section 32 of this manual.

b. Do not use values of thickness less than those derived from the lowest design pressure curve on the thread chart (opposite page).

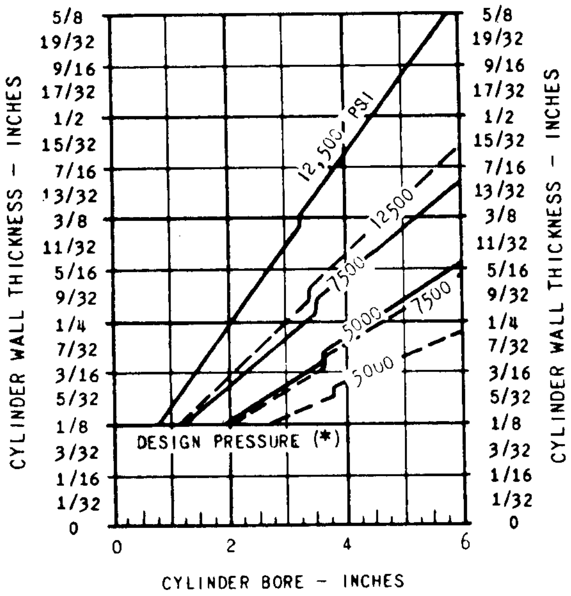
c. Use standard cylinder bore sizes in accordance with Section 60 of this manual.

d. Use standard thread series in accordance with Section 120 of this manual and Drafting Manual Section 69.

33.114 THREAD CHARTS

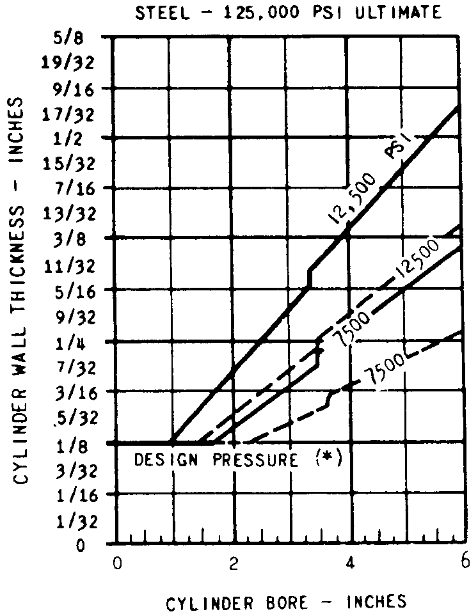
(1)

STEEL - 95,000 PSI ULTIMATE



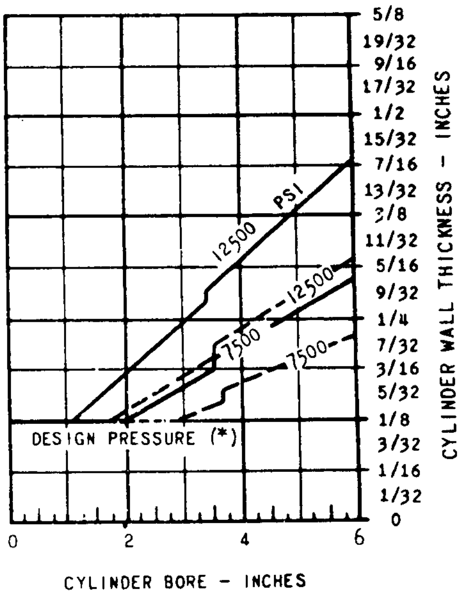
(2)

STEEL - 125,000 PSI ULTIMATE



(3)

STEEL - 150,000 PSI ULTIMATE

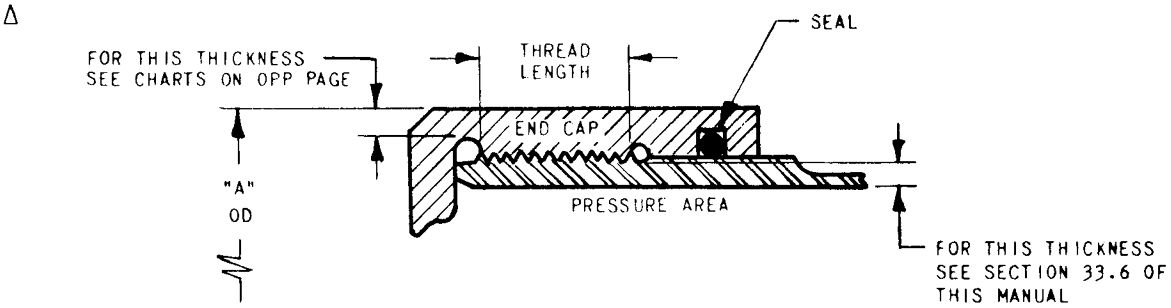


* BASED ON ULTIMATE STRENGTH

33.12 END CAP THICKNESS AT JOINT

.121 PRESSURE ON THREADS

ILLUSTRATION:



METHOD: (SEE THREAD CHARTS ON OPPOSITE PAGE - USE DASH LINES FOR DESIGN PRESSURES AND BROKEN SOLID LINES FOR THREAD LENGTHS)
SELECT PROPER CHART. ENTER CHART AT NOMINAL THREAD DIAMETER. PROJECT A LINE UP TO DESIRED PRESSURE LINE AND ALSO BEYOND TO MINIMUM BODY THREAD LENGTH LINE. THEN READ ACROSS FOR CAP THICKNESS AND THREAD LENGTH RESPECTIVELY.

FOR AD AT "A"; TAKE NOMINAL THREAD DIAMETER AND ADD TWICE THE CAP THICKNESS.

EXAMPLE: FOR AN END CAP OF 17S-T BAR WITH A 3 INCH NOMINAL THREAD DIAMETER AND A DESIGN PRESSURE OF 3,750 PSI, THE END CAP THICKNESS IS 9/32 INCH, THE LENGTH OF THREAD IS 5/8 INCH, AND THE OD AT "A" IS 3.562.

.122 MISCELLANEOUS DATA

a. The 150,000 psi end cap chart (5) is based on a 150,000 psi ultimate minimum cylinder; all other end cap charts (1, 2, 3, 4) are based on a 95,000 psi ultimate cylinder.

b. When possible, use the dimensions as shown on the end cap thickness thread charts on opposite page.

c. The thickness may be reduced to the minimum machinable wall (2,500 psi design pressure-steel curve). To do this, increase length in proportion to decrease in thickness. If greater than 20 percent change, see Table XI, SM Report No. 2581.

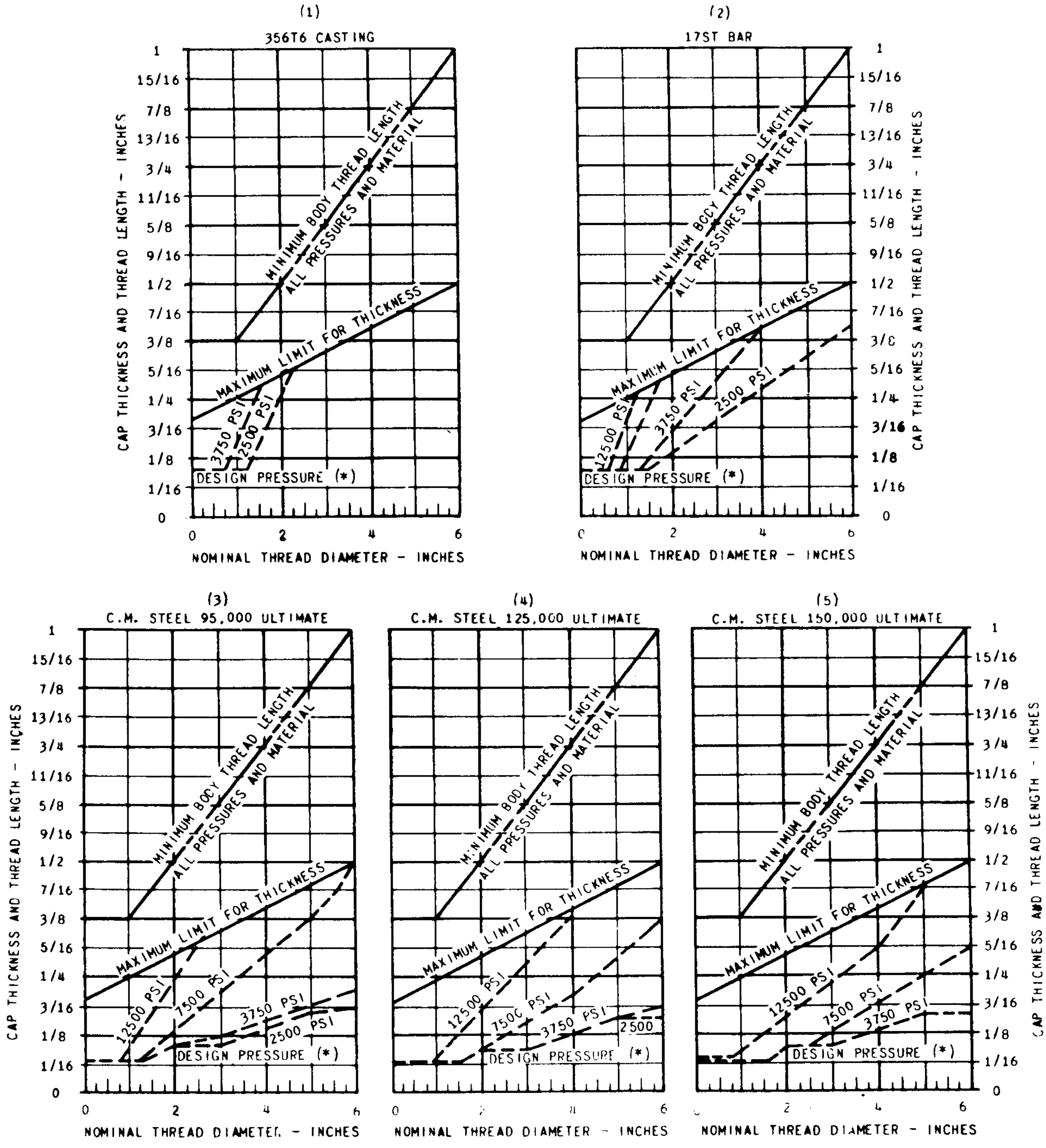
d. For design pressures, see Section 32 of this manual.

e. See that necessary margins are included in all castings.

f. Disregard effect of piston rods.

g. Use standard thread series, Section 120 in this manual and Drafting Manual Section 69.

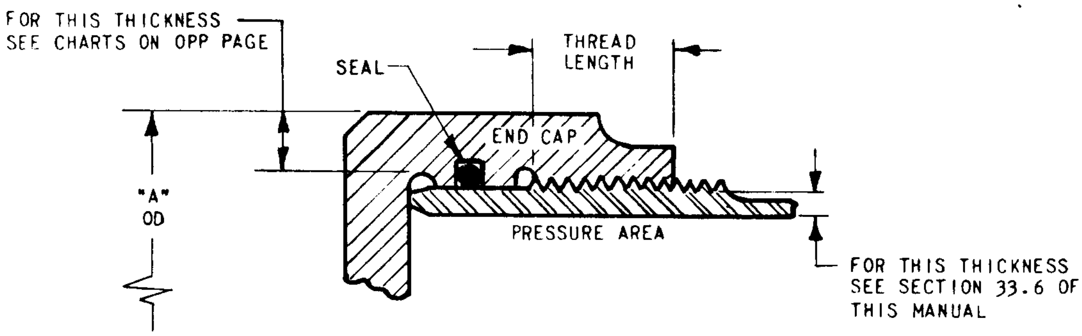
33.123 THREAD CHARTS



* BASED ON ULTIMATE STRENGTH

33.124 PRESSURE NOT ON THREADS

ILLUSTRATION:



METHOD: (SEE THREAD CHARTS ON OPPOSITE PAGE - USE DASH LINES FOR DESIGN PRESSURES AND BROKEN LINES FOR THREAD LENGTHS)
SELECT PROPER CHART. ENTER CHART AT NOMINAL THREAD DIAMETER. PROJECT LINE UP TO DESIRED DESIGN PRESSURE LINE AND BEYOND TO MINIMUM BODY THREAD LENGTH LINE. THEN READ ACROSS FOR CAP THICKNESS AND THREAD LENGTH FROM EACH INTERSECTION POINT.

FOR OD AT "A" TAKE NOMINAL THREAD DIAMETER AND ADD TWICE THE CAP THICKNESS.

EXAMPLE: FOR AN END CAP OF 17S-T BAR WITH A 3 INCH NOMINAL THREAD DIAMETER AND A DESIGN PRESSURE OF 7,500 PSI, THE END CAP THICKNESS IS 3/16 INCH, THE LENGTH OF THREAD IS 5/8 INCH, THE OD AT "A" IS 3.375 INCH.

.125 MISCELLANEOUS DATA

a. The 150,000 psi end cap chart (5) is based on a 150,000 psi ultimate minimum cylinder; all other end cap charts (1, 2, 3, 4) are based on a 95,000 psi ultimate minimum cylinder.

b. When possible, use the dimensions as shown for the end cap thickness by the thread charts on opposite page.

c. The thickness may be reduced to the minimum machinable wall (2,500 psi design pressure-steel curve). To do this, increase length in proportion to decrease in thickness. If greater than 20 percent change, see Table XII, SM Report 2581.

d. For design pressures, see Section 32 of this manual.

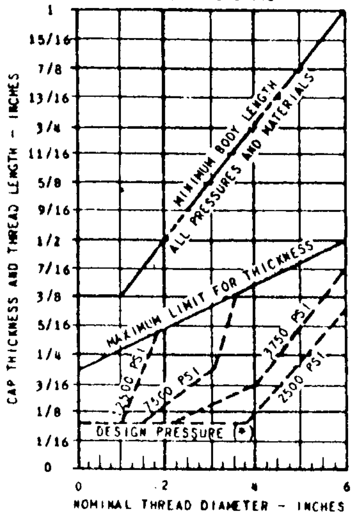
e. See that necessary margins are included in all castings.

f. Disregard effect of piston rods.

g. Use standard thread series, Section 120 in this manual and Drafting Manual Section 69.

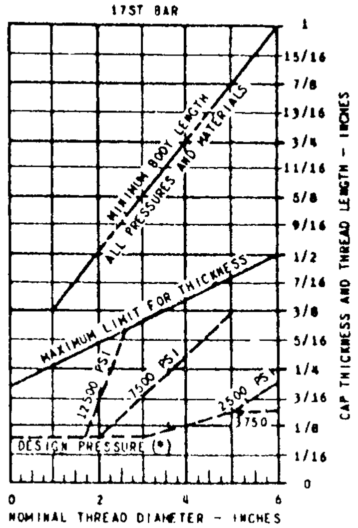
33.126 THREAD CHARTS

(1)
356T6 CASTING



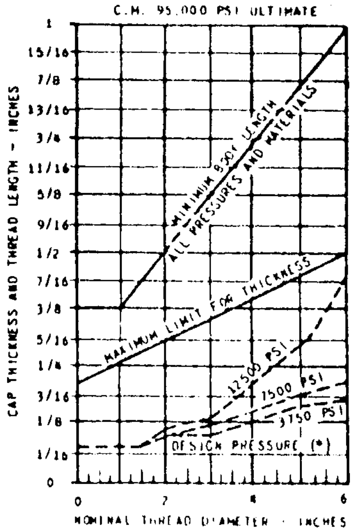
NOTE: THIS CHART IS BASED ON A 95,000 PSI ULTIMATE MINIMUM CYLINDER

(2)
17ST BAR



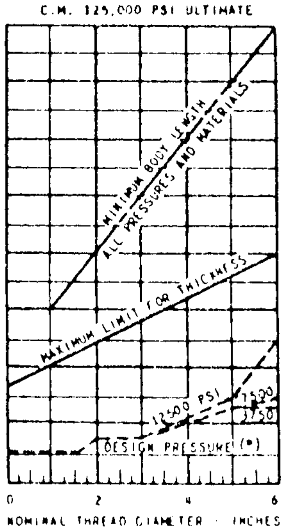
NOTE: THIS CHART IS BASED ON A 95,000 PSI ULTIMATE MINIMUM CYLINDER

(3)
C.M. 95,000 PSI ULTIMATE



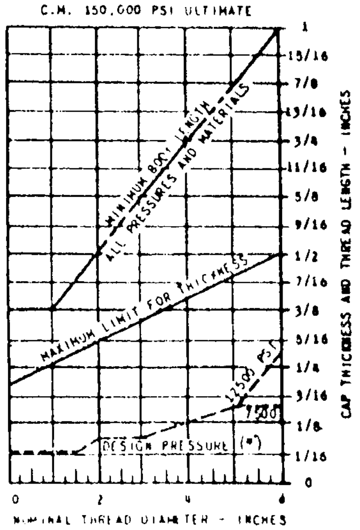
NOTE: THIS CHART IS BASED ON A 95,000 PSI ULTIMATE MINIMUM CYLINDER

(4)
C.M. 125,000 PSI ULTIMATE



NOTE: THIS CHART IS BASED ON A 95,000 PSI ULTIMATE MINIMUM CYLINDER

(5)
C.M. 150,000 PSI ULTIMATE

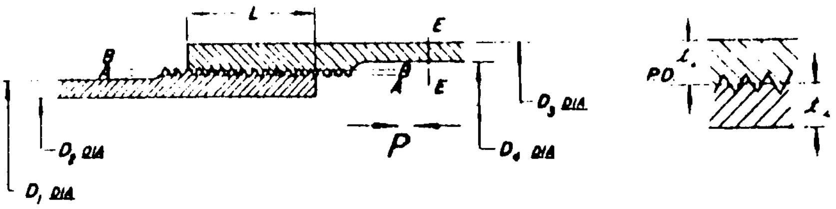


NOTE: THIS CHART IS BASED ON A 150,000 PSI ULTIMATE MINIMUM CYLINDER

* BASED ON ULTIMATE STRENGTH

33.2 STRESS ANALYSIS OF THREADED JOINTS

ILLUSTRATION:



.21 UNIT TENSION OR COMPRESSION STRESS

FIND: INNER AND OUTER MEMBER STRESS (f_t OR f_c)

WHEN: f_t = UNIT TENSILE STRESS (PSI)

f_c = UNIT COMPRESSIVE STRESS (PSI)

P = DESIGN AXIAL LOAD (LBS)

D_1 = OD TUBE (INCHES)

D_2 = ID TUBE (INCHES)

D_3 = OD OUTER MEMBER (INCHES)

D_4 = ID OUTER MEMBER (INCHES)

SOLUTION:

INNER MEMBER

$$f_t \text{ OR } f_c = \frac{P}{\frac{\pi}{4}(D_1^2 - D_2^2)}$$

OUTER MEMBER

$$f_t \text{ OR } f_c = \frac{P}{\frac{\pi}{4}(D_3^2 - D_4^2)}$$

NOTE: $83\% = \dfrac{1 \text{ X } 100}{1.2 \text{ FITTING FACTOR}}$

F_t ALLOWABLE = 83% ULTIMATE TENSILE STRESS OR 50% IF CASTING

F_c ALLOWABLE = 83% ULTIMATE COMPRESSIVE STRESS OR 50% IF CASTING

33.22 RADIAL UNIT TENSION OR COMPRESSION STRESS (HOOP STRESS)

FIND: INNER AND OUTER MEMBER STRESS (f_t OR f_c)

WHEN: f_c = RADIAL UNIT COMPRESSIVE STRESS (PSI)

f_t = RADIAL UNIT TENSILE STRESS (PSI)

P = DESIGN AXIAL LOAD (LBS)

L = LENGTH THREADS ENGAGED (INCHES)

t_i = THICKNESS ID TO PD INNER MEMBER (INCHES)

t_o = THICKNESS OD TO PD OUTER MEMBER (INCHES)

SOLUTION:

INNER MEMBER

60° THREAD

$$f_c = .0919 \frac{P}{Lt_i}$$

45° THREAD

$$f_c = .066 \frac{P}{Lt_i}$$

OUTER MEMBER

60° THREAD

$$f_t = .0919 \frac{P}{Lt_o}$$

45° THREAD

$$f_t = .066 \frac{P}{Lt_o}$$

NOTE: $83\% = \frac{1 \text{ X } 100}{1.2 \text{ FITTING FACTOR}}$

F_t ALLOWABLE = 83% ULTIMATE TENSILE STRESS OR 50% IF CASTING

F_c ALLOWABLE = 83% ULTIMATE COMPRESSIVE STRESS OR 50% IF CASTING

60° THREAD AMERICAN NATIONAL STANDARD

45° THREAD DOUGLAS SPECIAL

33.23 UNIT SHEAR STRESS

FIND: INNER AND OUTER MEMBER STRESS (f_s)

WHEN:

n = .577 (A CONSTANT)
m = .0919 (A CONSTANT)
q = .650 (A CONSTANT)
R = .758 (A CONSTANT)
S = .866 (A CONSTANT)
U = .50 (A CONSTANT)
D_F = THREAD PITCH DIAMETER
D_M = THREAD PITCH DIAMETER

} 60° THREAD

n = .414 (A CONSTANT)
m = .066 (A CONSTANT)
q = .848 (A CONSTANT)
R = .848 (A CONSTANT)
S = 1.207 (A CONSTANT)
U = .60 (A CONSTANT)
D_F = FEMALE THREAD DIA
D_M = MALE THREAD DIA

} 45° THREAD

f_s = UNIT SHEAR STRESS (PSI)
p = PITCH = 1 DIVIDED BY NO. OF THREADS PER INCH
E_i = MODULUS OF ELASTICITY - INNER MEMBER
E_o = MODULUS OF ELASTICITY - OUTER MEMBER
ρ = HYDRAULIC DESIGN PRESSURE (PSI)
f_c = UNIT COMPRESSIVE STRESS (PSI)
f_t = UNIT TENSILE STRESS (PSI)
t_i = THICKNESS ID TO PD INNER MEMBER (INCHES)
t_o = THICKNESS OD TO PD OUTER MEMBER (INCHES)
L = LENGTH OF THREADS ENGAGED (INCHES)
PD = PITCH DIAMETER
P = DESIGN AXIAL LOAD (LBS)

SOLUTION:

a. INNER MEMBER (MALE THREAD – SECTION "A–A")

$$f_s\ (MAX) = \sqrt{f_s^{\,2} + \left(\frac{f_c}{2}\right)^2}$$

WHEN: $f_s = \dfrac{P}{AREA\ A\text{–}A}$

(Shear)

$f_c = \dfrac{mP}{Lt_i}$ OR $\dfrac{nf_s}{1}$ (WHICHEVER IS GREATER)

(Hoop Compression or Radial Compression)

This term to be deleted if no hyd pressure on thd

$$\text{MIN SHEAR AREA "A–A"} = \left[\frac{qp}{1} - \frac{1}{2}\left(\frac{m\,P\,PD}{Lt_iE_i}\right) - \frac{1}{2}\left(\frac{m\,P\,PD}{Lt_oE_o}\right) - \overbrace{\frac{1}{4}\left(\frac{\rho\ (PD)^2}{E_ot_o}\right)} - \frac{PD\ TOL\ (MALE)}{2} - \frac{MINOR\ DIA\ TOL\ (FEMALE)}{2}\right] \frac{\pi L\ (MAX\ MINOR\ DIA\ (FEMALE)}{Sp}$$

APPROX AREA "A–A" = $U\ \pi\ D_F{}^*L$ (ROUGH ASSUMPTION ONLY)

b. OUTER MEMBER (FEMALE THREAD – SECTION "B–B")

$$f_s\ (MAX) = \sqrt{f_s^{\,2} + \left(\frac{f_t + f_c' + f_t'}{2}\right)^2}$$

WHEN: $f_s = \dfrac{P}{AREA\ B\text{–}B}$ $f_c' = nf_s$

(Shear) *(Radial Tension)*

$f_t = \dfrac{m\,P}{Lt_o}$ $f_t' = \dfrac{\rho\ PD}{2t_o}$

(Hoop Tension) *(Hoop Tension, Pressure on Thd)*

This term to be deleted if no hyd pressure on thd

$$\text{MIN SHEAR AREA "B–B"} = \left[\frac{Rp}{1} - \frac{1}{2}\left(\frac{m\,P\,PD}{Lt_iE_i}\right) - \frac{1}{2}\left(\frac{m\,P\,PD}{Lt_oE_o}\right) - \overbrace{\frac{1}{4}\left(\frac{\rho\ (PD)^2}{E_ot_o}\right)} - \frac{PD\ TOL\ (FEMALE)}{2} - \frac{MAJOR\ DIA\ TOL\ (MALE)}{2}\right] \frac{\pi L\ (MIN\ MAJOR\ DIA\ (MALE)}{Sp}$$

APPROX AREA "B–B" = $U\ \pi\ D_M{}^*L$ (ROUGH ASSUMPTION ONLY)

NOTE: F_S ALLOWABLE = 83% ULTIMATE SHEAR STRESS OR 50% IF CASTING.

* FOR THE 60° THREAD, D_M MAY BE SUBSTITUTED FOR D_F AS EACH HAS THE SAME VALUE. FOR THE 45° THREAD, D_M AND D_F CANNOT BE SUBSTITUTED ONE FOR THE OTHER BECAUSE GREATER ACCURACY IS NECESSARY IN DETERMINING THE DIMENSION IN QUESTION.

33.24 PRELOADED BOLTS

.241 SOLID BOLTS (LUBRICATED)* The table shown below applies to lubricated bolts, nuts, studs, and cap screws. The values given are for steel bolts heat treated to 125,000 psi ultimate tensile strength. The nuts must be equal to or stronger than the bolts. When a material with an ultimate strength other than 125,000 psi is used, the allowable axial load and torque from the table should be multiplied by the ration: ultimate strength/125,000 psi.

*Lubricant should be used in all preloaded bolt applications where the lubricant will not be determental (such as self-locking nuts).

a. Drawings for preloaded bolts must specify:

APPLY ANTI-SEIZE LUBRICANT PER DPS 1.22

b. Drawings for a cap screw must specify:

LUBRICATE UNDER HEAD

c. The size of the bolt may be determined from the allowable axial load (Column 2 below). Bolts must never be loaded beyond the value given in the table below when they are subject to test pressure or applied loads.

THREAD SIZE	LUBRICATED	
	ALLOWABLE AXIAL LOAD LBS	FT LBS TORQUE
1/4 - 28	1,770	5.5
5/16 - 24	2,900	11.5
3/8 - 24	4,300	20
7/16 - 20	6,000	32
1/2 - 20	8,120	48.5
9/16 - 20	10,200	68
5/8 - 20	13,000	96
11/16 - 18	16,000	128
3/4 - 18	19,000	167
13/16 - 18	23,000	215
7/8 - 18	27,000	272
15/16 - 18	30,000	322
1 - 18	35,000	403
1 1/16 - 14	41,000	488
1 1/8 - 18	46,000	588
1 3/16 - 14	51,000	676
1 1/4 - 18	56,000	775
1 5/16 - 14	65,000	933
1 3/8 - 18	71,000	1060
1 7/16 - 18	77,000	1180
1 1/2 - 18	86,000	1380

33.2411 WRENCH TORQUES FOR CASTELLATED AND SELF-LOCKING STEEL NUTS (NONLUBRICATED)

NUT & BOLT	STANDARD WRENCH TORQUES		
	BOLT SIZE	RECOMMENDED TORQUE (IN #)	MAXIMUM ALLOWABLE (COTTER PIN) TORQUE (IN#)
AN 365 & AN 310 NUTS WITH AN BOLTS OR EQUIVALENT	#10	20-25	40
	1/4	50-70	100
	5/16	100-140	225
	3/8	160-190	390
	7/16	450-500	840
	1/2	480-690	1100
	9/16	800-1000	1600
	5/8	1100-1300	2400
	3/4	2300-2500	5000
	7/8	2500-3000	7000
	1	3700-5500	10000
	1-1/8	5000-7000	15000
HIGH TENSION NUTS WITH NAS-144 SERIES HIGH TENSION BOLTS	1/4	73-100	
	5/16	145-200	
	3/8	230-280	
	7/16	650-720	
	1/2	700-1000	
	9/16	1200-1440	
	5/8	1620-1860	
	3/4	3360-3600	
	7/8	3600-4320	
	1	5400-7980	
	1-1/8	7260-10140	

APPROXIMATE AXIAL PRELOAD IN BOLTS WITH AN365 NUTS	
BOLT SIZE	AXIAL PRELOAD IN # PER IN # TORQUE
#10	27.0
1/4	21.5
5/16	17.1
3/8	14.9
7/16	12.6
1/2	11.4
9/16	10.3
5/8	9.2
3/4	7.6
7/8	6.8
1	6.0
1-1/8	4.8

1. VALUES FROM HANDBOOK OF INSTRUCTIONS FOR AIRCRAFT DESIGNERS, VOL I, PT B, p 4-4 (7-55).
2. VALUES ARE BASED ON FINE THREAD SERIES, CADMIUM PLATED AND NON LUBRICATED.
3. INSTALLATION WRENCH TORQUES FOR GENERAL STRUCTURE ARE COVERED BY EXISTING INSPECTION PROCEDURES AND SHOULD NOT BE CALLED FOR ON DRAWINGS. VALUES ARE FURNISHED MAINLY FOR INFORMATION. IN A FEW SPECIAL CASES, HOWEVER, SUCCESS OF THE DESIGN MAY BE PARTICULARLY DEPENDENT UPON CONTROL OF WRENCH TORQUES. IN SUCH CASES, WRENCH TORQUES SHALL BE CALLED FOR.
4. SPECIFICATION OF WRENCH TORQUE IS NOT AN ACCURATE METHOD OF ESTABLISHING PRELOADS IN BOLTS, OR TIGHTNESS OF NUTS BECAUSE OF THE LARGE VARIATIONS IN FRICTION OF CONTACTING SURFACES. A 300% OR GREATER DIFFERENCE IN BOLT STRETCH DUE TO A CONSTANT TORQUE CAN BE EXPECTED WTTH DRY NUTS. SOME METHOD OF ESTABLISHING ACTUAL LIMITS OF BOLT STRETCH IS NECESSARY IF GREAT ACCURACY IS REQUIRED.
5. SHEAR NUTS ARE NOT TORQUED TO SPECIFIED VALUES AS STANDARD SHOP PRACTICE. AIR CORPS RECOMMENDED TORQUE VALUES FOR SHEAR NUTS, AN364 & AN320, ARE 60% OF CORRESPONDING VALUES FOR AN365 & AN310 NUTS
6. DRAWINGS CALLING FOR SELF-LOCKING NUTS SHOULD NEVER SPECIFY LUBRICATION.

.242 HOLLOW BOLTS (LUBRICATED)

The charts on the opposite page apply to bolts, studs, and cap screws. The values given are for steel bolts, heat treated to 125,000 psi ultimate tensile strength. The nuts or housing must be equal to or stronger than the bolts. When a material with an ultimate strength other than 125,000 psi is used, the allowable axial load and torque from Section 33.31 should be multiplied by the ratio: ultimate strength/ 125,000 psi.

a. Drawings for preloaded bolts must specify:

Apply anti-seize lubricant per DPS 1.22

b. Drawings for a cap screw must specify:

Lubricate under head

Note: Drawings calling for elastic stop nuts should never specify lubrication.

33.3 MODULUS OF RUPTURE

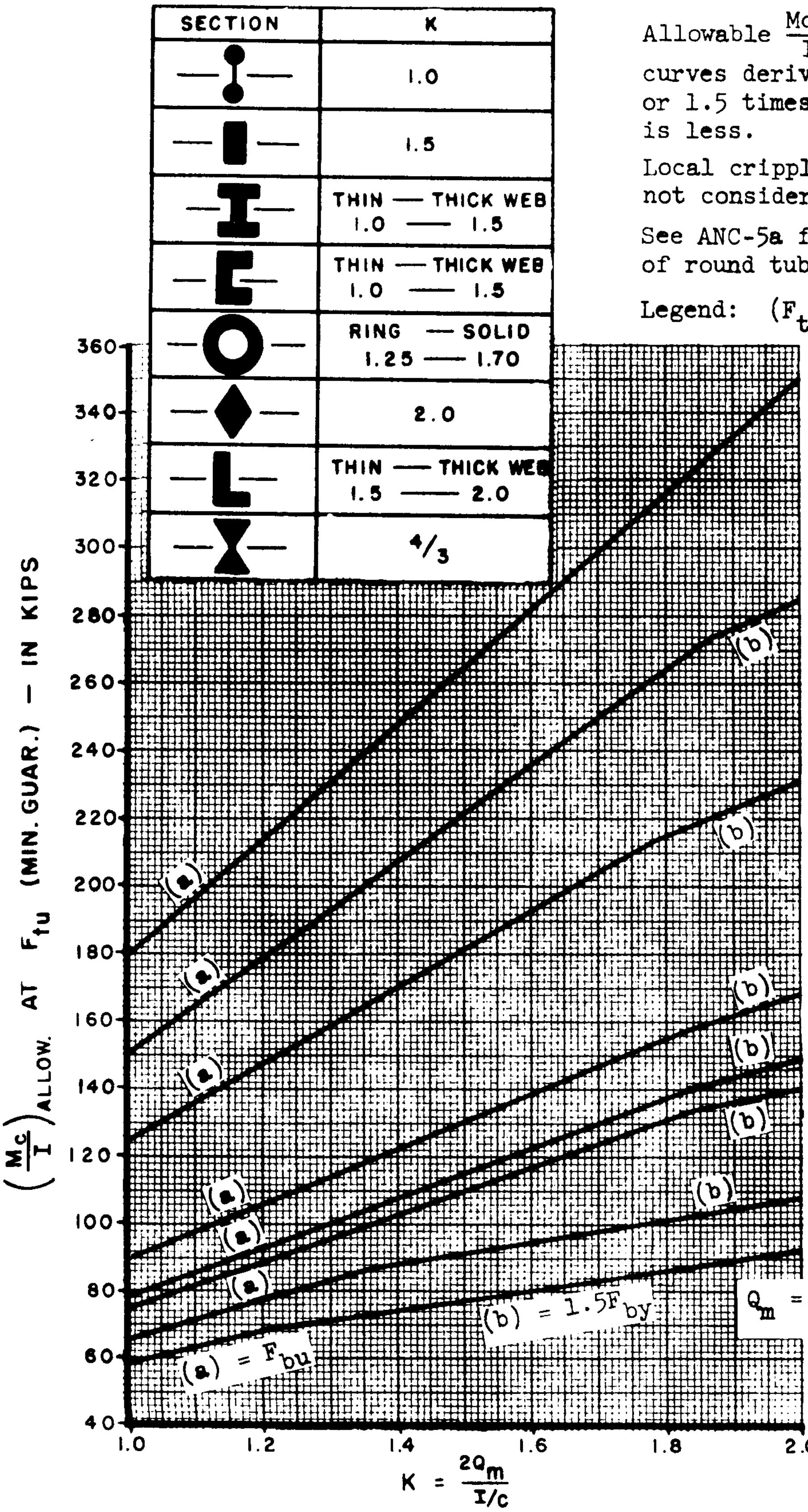
33.31 MODULUS OF RUPTURE OF STEEL AND DURAL SHAPES

Allowable $\frac{Mc}{I}$ vs. K curve is envelope of curves derived from ultimate allowable or 1.5 times yield allowable, whichever is less.

Local crippling may be critical but is not considered in this chart.

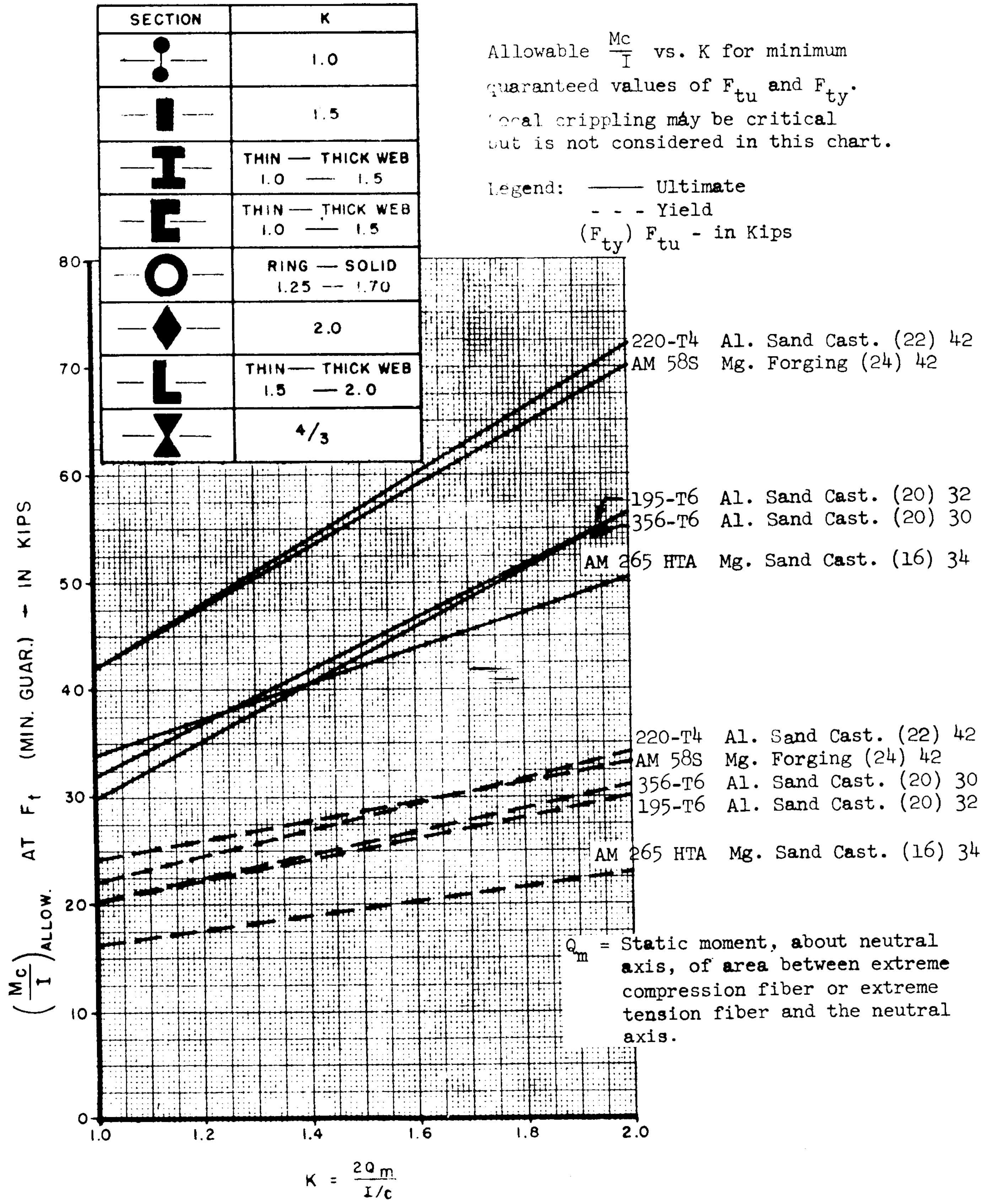
See ANC-5a for modulus of rupture value of round tubing.

Legend: (F_{ty}) F_{tu} - Minimum guaranteed value in Kips

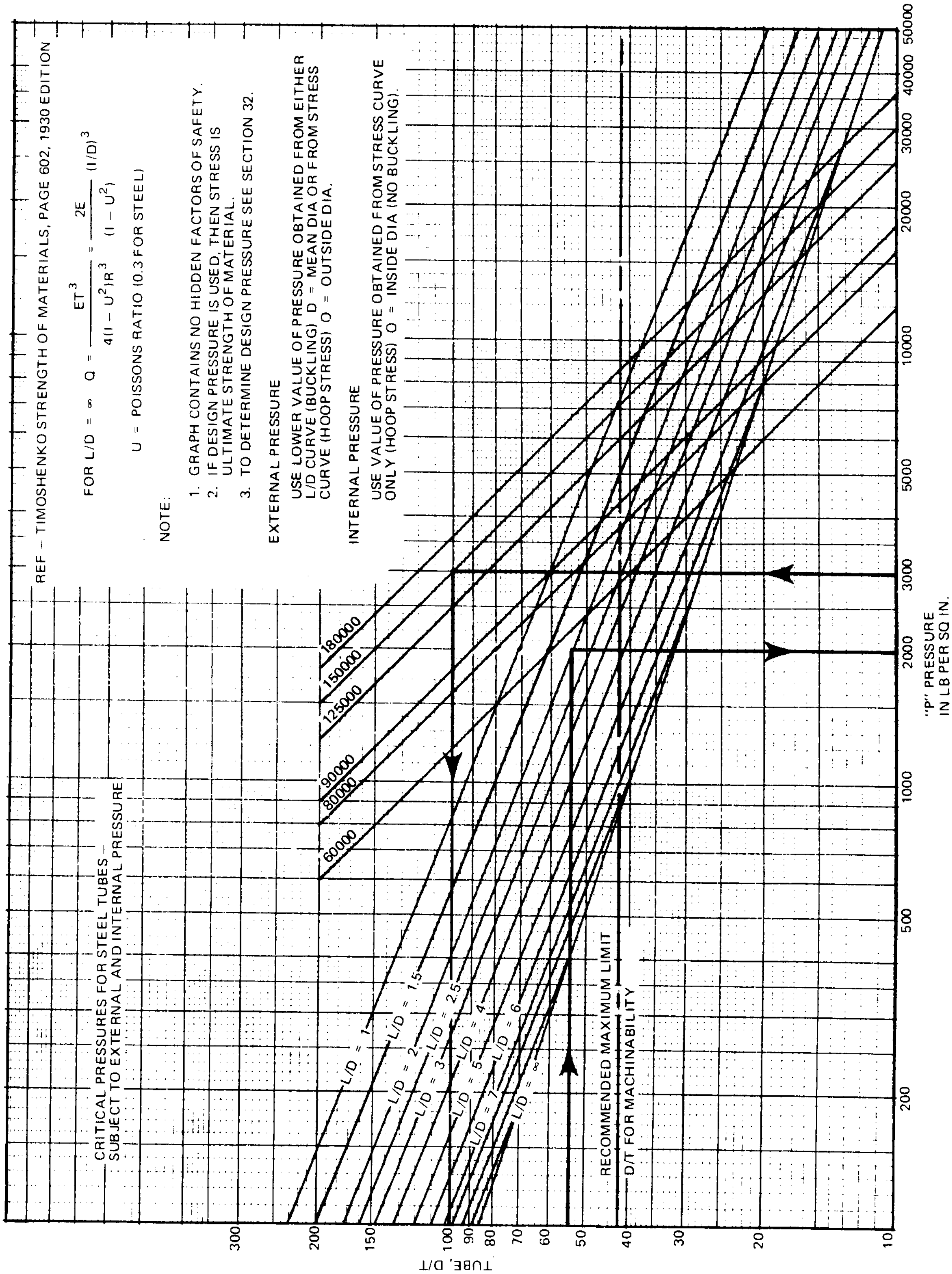


Q_m = Static moment, about neutral axis, of area between extreme compression fiber or extreme tension fiber and the neutral axis.

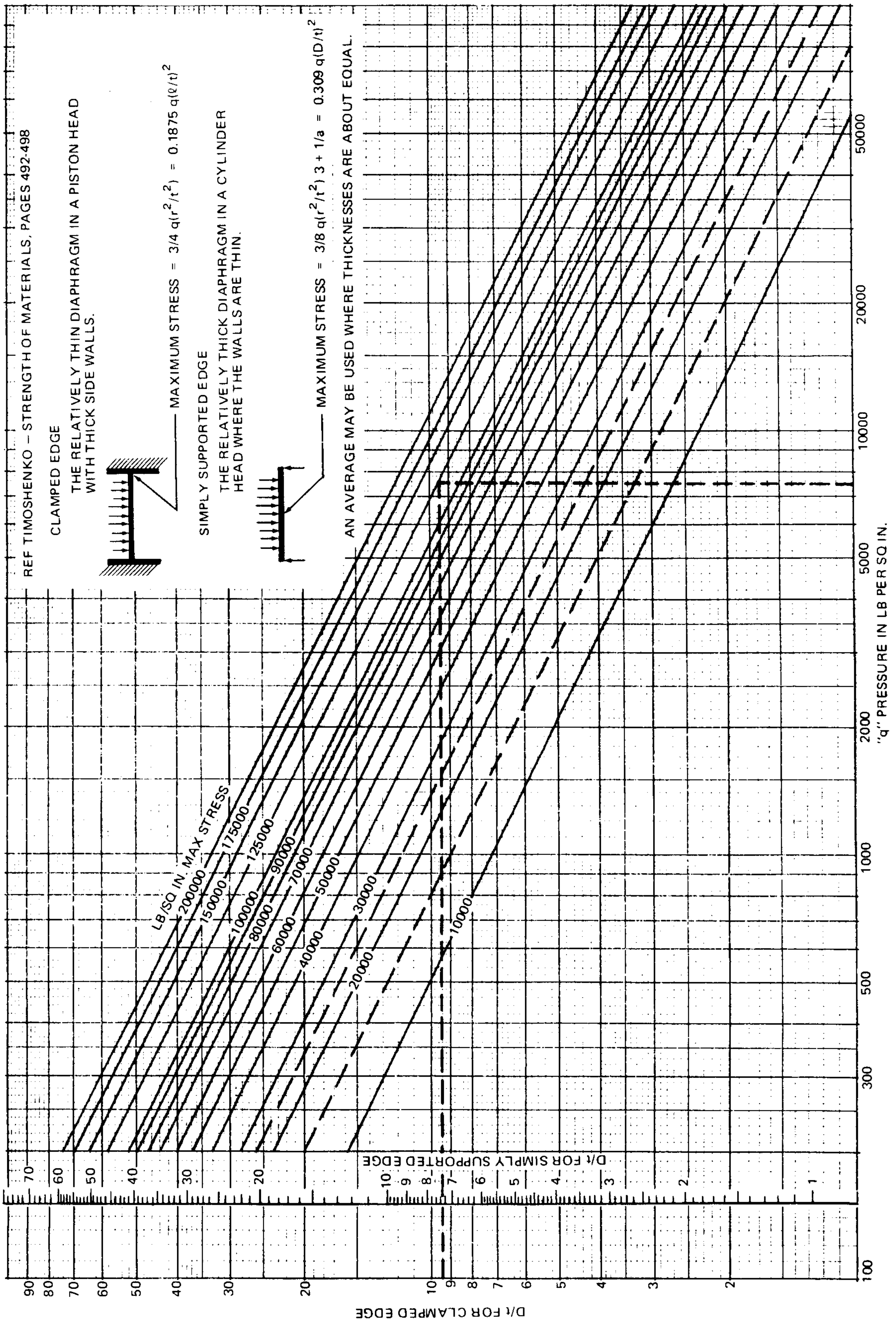
33.32 MODULUS OF RUPTURE OF DURAL AND MAGNESIUM SHAPES



33.4 CRITICAL PRESSURES FOR STEEL TUBES



33.5 STRESS IN UNIFORMLY LOADED CIRCULAR FLAT PLATES



33.6 BEARINGS

33.61 ALLOWABLE BEARING PRESSURE

The values given in the chart below under "Moving Joints" or "Non-moving Joints" are for joints with loose fits. Bearing factors are from MIL-HDBK-5. For interference fits use the values given in the table under "Non-moving Joints - No Shock Load," which include a fitting factor of 1.15 on wrought and sintered materials and a casting factor of 1.25 on castings.

	MATERIAL	ULTIMATE BEARING STRENGTH PSI	ALLOWABLE ULTIMATE BEARING STRENGTH				REMARKS
			NON-MOVING JOINTS		MOVING JOINTS		
			NO SHOCK LOAD[2]	SHOCK LOAD	NO SHOCK LOAD	SHOCK LOAD	
ALUMINUM	2014 FORGINGS	90,000	78,200	45,000	45,000	36,000	
ALUMINUM	2017 SHEET OR BAR	67,000	58,200	33,500	33,500	26,800	
ALUMINUM	2024 SHEET OR BAR	84,000	73,000	42,000	42,000	33,600	
ALUMINUM	356-T6 ALUMINUM ALL ALLOY CASTING	48,000	38,400	24,000	NOT REC	NOT REC	NO FLUOROSCOPIC INSPECTION REQUIRED BY CAA FOR BEARING STRESSES
BRONZE	CENTRIFUGALLY CAST ALUMINUM BRONZE	120,000	96,000	60,000	60,000	48,000	
BRONZE	TOBIN BRONZE (HARD)	60,000	52,100	30,000	30,000	24,000	
MAGNESIUM	DOW H-T4 MAGNESIUM CASTING (HT)	36,000	28,800	18,000	NOT REC	NOT REC	NO FLUOROSCOPIC INSPECTION REQUIRED BY CAA FOR BEARING STRESSES
MAGNESIUM	DOW H-T6 MAGNESIUM CASTING (HTA)	50,000	40,000	25,000	NOT REC	NOT REC	
STEEL	STEEL (ON STEEL)	SEE MIL-HDBK-5	87% OF ULT BRG STRENGTH	50% OF ULT BRG STRENGTH	6,500*	5,000*	CONSULT DESIGN SUPERVISION
(3)	BEARIUM B-4 B-10	21,650 25,500	17,340 20,400	10,825 12,750	10,825 12,750	8,660 10,200	
	COMPO AND OILITE	7,500	6,520[4]	NOT REC	3,750[5]	NOT REC	
	XB COMP AND SUPER OILITE	30,000	26,100[6]	NOT REC	15,000[7]	NOT REC	
	NEEDLE BEARING (INNER RACE TYPE)						CONSULT DESIGN SUPERVISION

NOTES: (1) USE MANUFACTURER'S STATIC NON-BRINNEL RATING OR LIMIT LOAD RATING TIMES 1.5 FOR ULTIMATE ALLOWABLE ON BEARINGS, WITH OR WITHOUT SMALL MOTION.

(2) WHERE BEARING FACTORS ARE REQUIRED WHICH ARE GREATER THAN THE FITTING OR CASTING FACTOR QUOTED ABOVE, THE ULTIMATE BEARING STRENGTH OF THE MATERIAL SHOULD BE USED.

(3) BEARIUM IS FOR USE IN APPLICATIONS REQUIRING A PLAIN BEARING OR BUSHING COMBINED WITH A LOW COEFFICIENT OF FRICTION. ("BEARIUM" IS A TRADE NAME SIMILAR TO "OILITE" OR "COMPO" AND IS NOT TO BE CONFUSED WITH THE ELEMENT BARIUM.)

(4) MAXIMUM STATIC LIMIT STRESS – 8500 PSI

(5) MAXIMUM LIMIT STRESS UNDER ROTATION – 4000 PSI

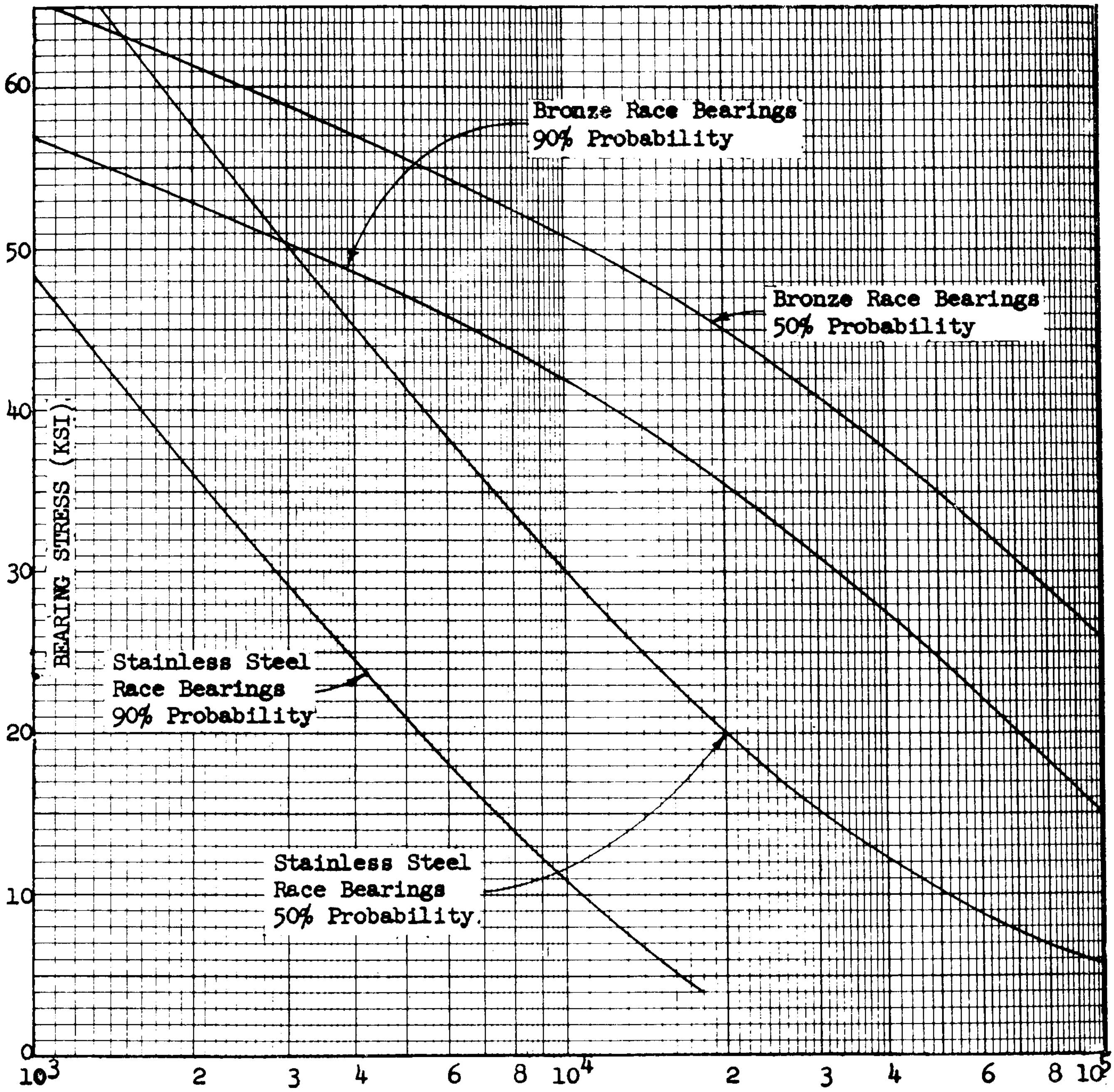
(6) MAXIMUM STATIC LIMIT STRESS – 15,000 PSI

(7) MAXIMUM LIMIT STRESS UNDER ROTATION – 8000 PSI

REF: H.I.A.A. p. 8-8 AND MIL-B-5687.

*AT LONG BEACH USE VALUES FROM MIL-HDBK-5.

33.62 SPHERICAL BEARING LOAD-LIFE CURVES



CYCLES TO FAILURE
(Failure is defined as occurring when the bearing clearance ≧ 1% of the ball dia.)

NOTES:
1. The above life prediction curves are suitable for safe design use whenever the bearing application is in strict conformance with the stipulations of this section.
2. Each cycle consists of an oscillation back and forth through any angle between 5° and 90° with the bearing stress on the spherical surface varying from the peak stress (shown above) at one travel extreme to the same stress in the opposite direction at the other extreme.

33.7 SPRING ANALYSIS AND DESIGN

.71 GENERAL This section presents design formulas for tension and compression springs as well as allowable stress curves for commonly used spring materials. In addition, a chart is included to facilitate the selection of the basic spring dimensions.

The basic stress formula for round wire tension and compression springs is:

$$f_s \; - \; 2.55\ KPD/d^3 \text{ (including Wahl factor)*}$$

f_s = Torsion stress in pounds per square inch

P = Load on spring in pounds

D = Mean diameter of coil in inches (outside diameter minus wire diameter)

d = Diameter of wire in inches.

The basic deflection formula is:

$$\delta = \frac{8PD^3N}{Gd^4}$$

δ = Deflection in inches

G = Torsional modulus of rigidity

N = Number of active coils.

In addition to the basic stress as given by the above formulae, the wire curvature contributes a higher localized stress which becomes important for repeated loadings but does not appreciably affect overall spring yielding. Tension springs have another stress concentration at the bend for the end loop. This too is important only for predicting spring life. The first factor is plotted in Figure 30-2, and the second factor is shown in Figure 30-3.

Spring designs should be checked for both permanent set and for allowable life. Unsupported compression springs should also be checked for buckling. It is common practice to allow some yielding of compression springs as they can be easily closed solid by the spring manufacturer to produce the necessary internal residual stresses and still possess the required free length dimensions. It is

*For design purpose use P load at solid spring height to obtain $f_{s\ max}$.

much less common to design for "set" in tension springs as they are usually wound closed or with initial tension. If they are then "set", the free length dimension or the initial tension will be altered considerably. This can be compensated for in the design but it is not generally done. An approximate curve of permanent set as a function of the ratio of working stress to torsional yield stress is shown in Figure 30-4.

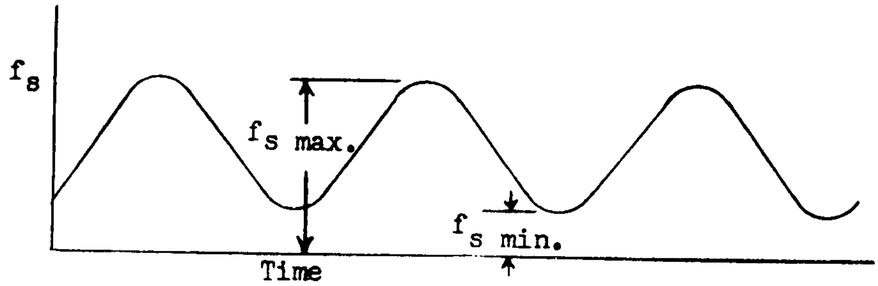
33.72 PERMANENT SET CHECK To check a given spring design for permanent set, the basic stress formula shown above should be used without any stress concentation factors. The ratio of calculated stress to the torsional yield stress (Figure 30-5) should then be computed. The permanent set may then be read from the curve in Figure 30-4. In general, it is recommended that the ratio of $f_s/F_{s\ yield}$ be kept under 1.00 for tension springs and under 1.25 for compression springs. If these limits are observed, no design compensation for permanent set is needed.

33.73 ALLOWABLE LIFE CHECK In addition to the permanent set check required above, the spring design should be checked for allowable life. This check must include all stress concentration factors. For compression springs, only the Wahl factor need be considered. For tension springs both the Wahl factor and the stress concentration factor due to the end loop must be considered in the last coil adjacent to the end loop so that the maximum stress at that point including both stress concentration factors will not exceed the maximum stress in the body of the spring when only the Wahl factor is considered. It is recommended that end loop radii be at least equal to twice the wire diameter and even larger if possible to reduce the probability of service failure.

The required life of springs used in aircraft varies with the application. For a spring which is cycled only several times a flight (for example an arresting gear dashpot spring) a life of 5000 cycles would be satisfactory. If it is cycled more often (for example a "feel" system spring) the required life should be around 1,000,000 cycles.

The maximum stress is computed by multiplying the basic stress by each of the stress concentration factors given in Figures 30-2 and 30-4. The ratio of $f_{s_{max}}/F_{s_{yield}}$ is then computed. The stress range is then determined as the ratio:

$$\frac{f_{s_{max}} - f_{s_{min}}}{f_{s_{max}}}$$



The allowable life curves in Figure 30-6 are then used to determine the number of cycles to failure. It is pointed out that these fatigue curves are the best available average data and that proper allowance for scatter should be made. It is estimated that 99 percent of the time the actual life will be greater than:

$$\log_{10} n_{min} \geq 0.90 \log_{10} n_{actual} \quad \text{(n is number of cycles to failure)}$$

33.74 BUCKLING OF COMPRESSION SPRINGS Unsupported compression springs should be checked for lateral buckling. The curve in Figure 30-7 may be used for this check.

33.75 NUMBER OF ACTIVE COILS The number of active coils for a given type of spring end is shown on the chart, Figure 30-8.

33.76 TOLERANCES Most spring designs result in a fairly large stress tolerance when the adverse effects of the different variables are considered. The effect of this stress variation on the permanent set and the spring life should be considered. If the accumulated tolerance effect is judged to be excessive, test load limits on the drawing which reflect only a portion of the possible tolerance may be used. It should be borne in mind, however, that this procedure will result in a larger number of spring rejections.

33.77 USE OF THE SPRING DESIGN CHART Figure 30-9 has been prepared to facilitate the selection of wire diameter and coil diameter for given load and deflection requirements. The spring chart relates the four quantities: load, total deflection divided by solid height of active turns, outside diameter and wire diameter. Any two of these quantities can be selected arbitrarily and the other two read from the chart. The usual problem is, given the load, total deflections and solid height, to find the wire size and outside diameter. The chart is entered with the known quantities which are plotted along the abscissa and ordinate of the graph. The intersection of the known values determines the closest standard wire size and outside diameter of the coils.

Direct application of the chart is limited to music wire, MIL-W-6101, for wire diameter up to 0.170 inches and pre-tempered 6150 wire, ASTM-A232-47 for wire diameters greater than 0.170 inches. Use of the chart with no correction will result in the elastic limit stresses per Figure 30-5 with no permanent set. The average life of a spring may be determined from Figure 30-8. For springs designed from the spring chart, Figure 30-9, it is only necessary to multiply the stress concentration factors together to obtain K, the ordinate of the S-N curves since

$$\frac{Smax}{\text{torsional elastic limit}} = 1$$

Due to such considerations as fatigue or permanent set, it may be necessary to design a spring for a stress other than that used in the chart. If design load and deflection are each multiplied by a factor equal to the torsional elastic limit stress divided by the design stress, the chart will give a spring that will be stressed to the design stress when design load is applied. A factor less than one increases the stress above those used in the chart. A factor greater than one decreases stress below those used in the chart. Use of the spring design chart is further explained in Paragraph 33.79.

33.78 SAMPLE CALCULATIONS

Given: D = 0.750 inch

d = 0.072 inch

P = 30 pounds

Elastic deflection = 1 inch

R = 0.50

Material - Music wire MIL-W-6101

Compute:

a. Permanent set

b. Allowable life as compression spring and as a tension spring with full diameter end coils and $R_o/d = 3$.

c. Determine the required reduced diameter of the end coils to allow the same life as a compression spring.

d. Check the compression spring for buckling.

Solution:

a. The spring chart gives a load of 23.6 pounds and a Δ/d of 3.57 at the intersection of $D_o = 0.822$ and d = 0.72. A 30-pound load will cause stresses exceeding the elastic torsional limit F_{sy} = 121,000 (Figure 30-5). f_s = 30/23.6 (121,000) = 154,000 psi. f_s could also have been obtained by substitution in the formula,

$$f_s = \frac{2.55\ PD}{d^3} = \frac{2.55\ (30)\ (0.75)}{(0.072)^3} = 154{,}000 \text{ psi}$$

$$\frac{f_s}{F_s \text{ yield}} = \frac{154{,}000}{121{,}000} \text{ or } \frac{30}{23.6} = 1.27$$

$$\frac{\text{permanent set}}{\text{elastic deformation}} = 0.32 \text{ (from Figure 30-4)}$$

permanent set = 0.32(1) = 0.32 inch.

b. Allowable life as compression spring

$$\frac{D}{d} = \frac{0.75}{0.072} = 10.4 \quad K = 1.13 \text{ (Wahl factor from Figure 30-2)}$$

$R = 0.5$

$$\frac{KS_{max}}{\text{torsional eleastic limit}} = \frac{1.13\ (154{,}000)}{121{,}000} = 1.44$$

No. of cycles = 4400 (from Figure 30-6)

Allowable life as tension spring

$$\frac{R_o}{d} = 3, \quad C = \frac{R_o}{R_1} = \frac{R_o}{R_o - d/2} = \frac{0.216}{0.180} = 1.20$$

$$\frac{CKS_{max}}{\text{torsional elastic limit}} = \frac{1.20\ (1.13)\ (154{,}000)}{121{,}000} = 1.71$$

The curve in Figure 30-6 for R = 0.50 does not go beyond K = 1.6.

Extrapolating, the number of cycles equals 348.

c. Reduced coil diameter of tension spring to give same life as compression spring. The problem is to adjust the coil diameter so as to make

$$\frac{CKS_{max}}{\text{torsional elastic limit}} = 1.44.$$

By trial and error it was found that a value of D = 0.61 gives the following:

$$S_{max} = \frac{2.55\ PD}{(d)^3} = \frac{2.55\ (30)\ (0.61)}{(0.072)^3} = 125{,}000 \text{ psi}$$

$$C = 1.20$$

$$\frac{D}{d} = \frac{0.61}{0.072} = 8.48 \quad K = 1.163 \text{ (Figure 30-2)}$$

Torsional elastic limit 121,000 (Figure 30-5)

$$\frac{CKS_{max}}{\text{torsional elastic limit}} = \frac{(1.163)\ (1.20)\ (125{,}000)}{121{,}000} = 1.44$$

d. Buckling of compression springs as a column. Solid height of actual turns may be found from the modified Δ/d from the chart or from the deflection equation (Paragraph 33.71).

From the chart, solid height of active turns $= \dfrac{(1)}{3.57\ \dfrac{30}{23.6}} = 0.22$

From the deflection equation, solid height of active turns equals

$$(d)\ (n) = \frac{FGd^5}{8PD^3} = \frac{(1)\ (11.5 \times 10^6)\ (0.072)}{8\ (30)\ (0.75)^3} = 0.22$$

Solid height = 0.22 + 0.072 (1.5) = 0.328

(1-1/2 inactive turns, Figure 30-8)

Free height = 0.328 + 1.00 + 0.25 = 1.58

(0.25 inch allowed so spring does not close solid)

$$\frac{\text{Free length}}{D} = \frac{1.58}{0.75} = 2.11$$

From the chart (Figure 30-7), the critical value of

$\dfrac{\text{Deflection}}{\text{Full length}}$ is 0.71

For the spring being checked

$$\frac{\text{Deflection}}{\text{Free length}} = \frac{1}{1.58} = 0.633$$

The spring will not buckle.

FIGURE 30-5

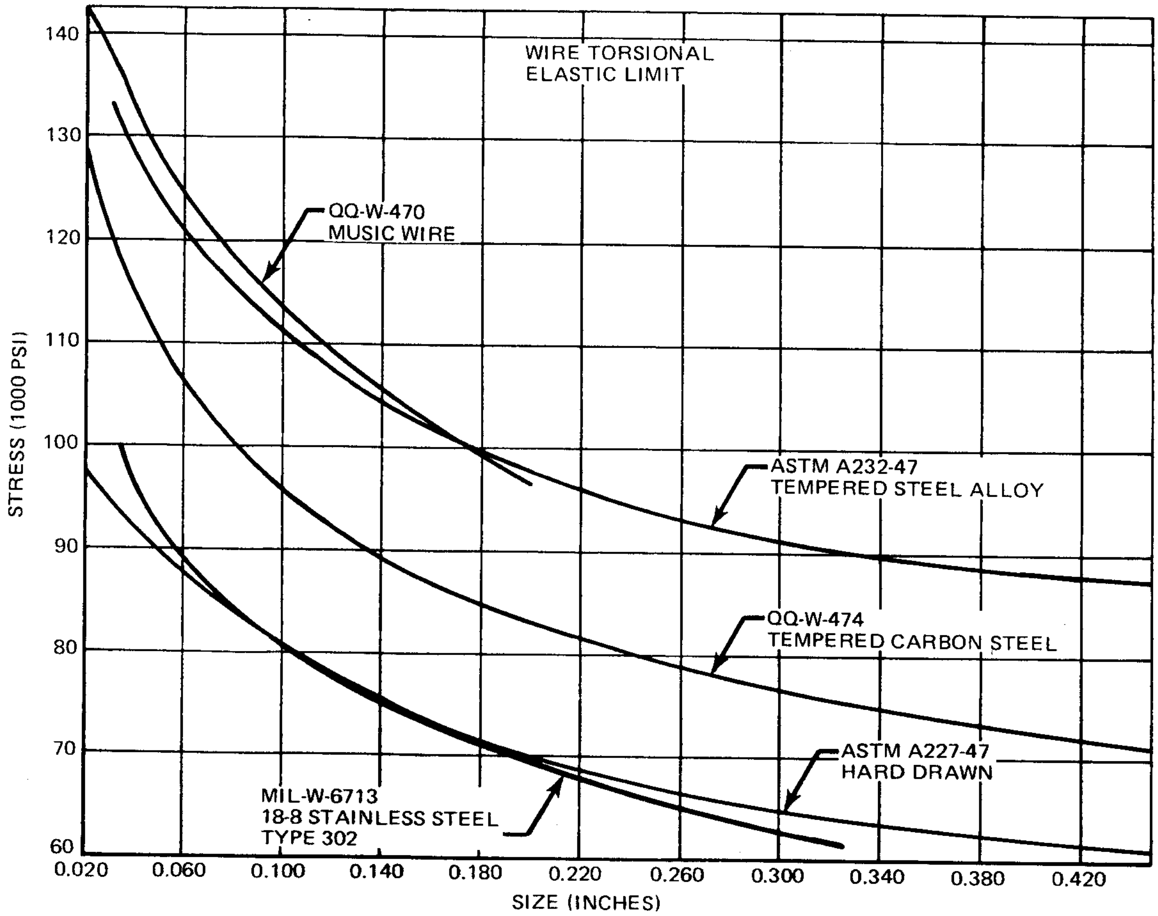
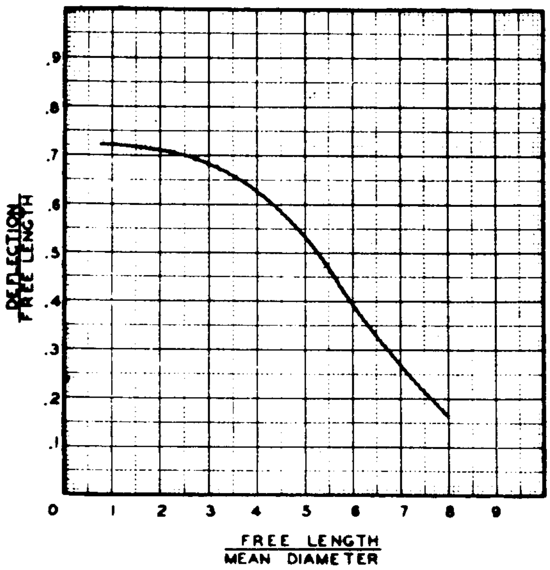


FIGURE 30-7.

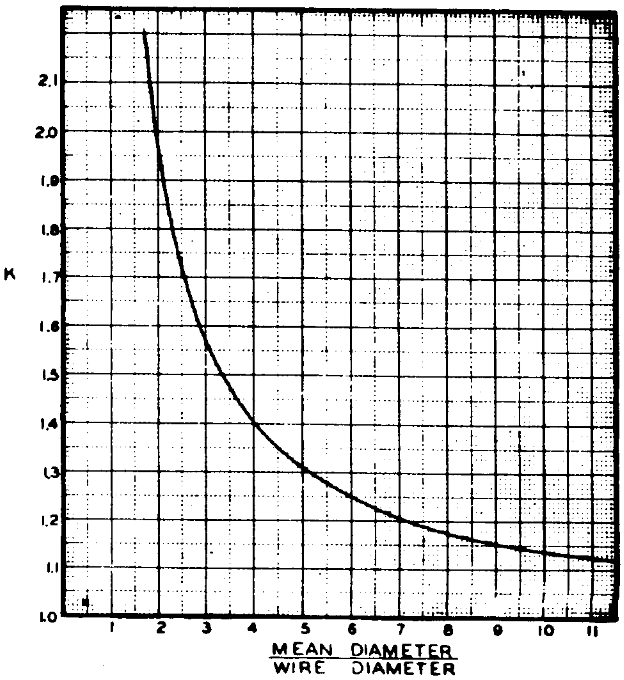
BUCKLING CHART FOR COMPRESSION SPRINGS



BUCKLING WILL OCCUR IF DEFLECTION IS GREATER THAN VALUES SHOWN.

FIGURE 30-2.

WAHL CORRECTION FACTOR FOR SPRINGS



$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C} \qquad C = \frac{D}{d}$$

FIGURE 30-8.

COMPRESSION SPRINGS, TYPES OF ENDS

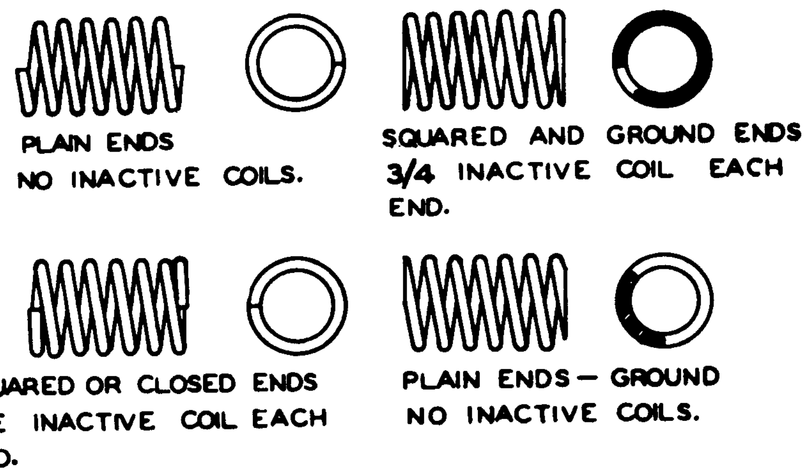
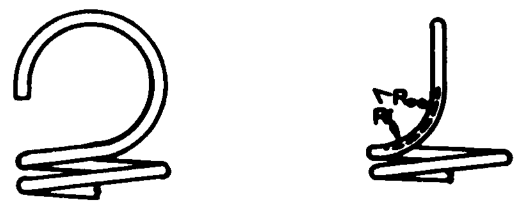


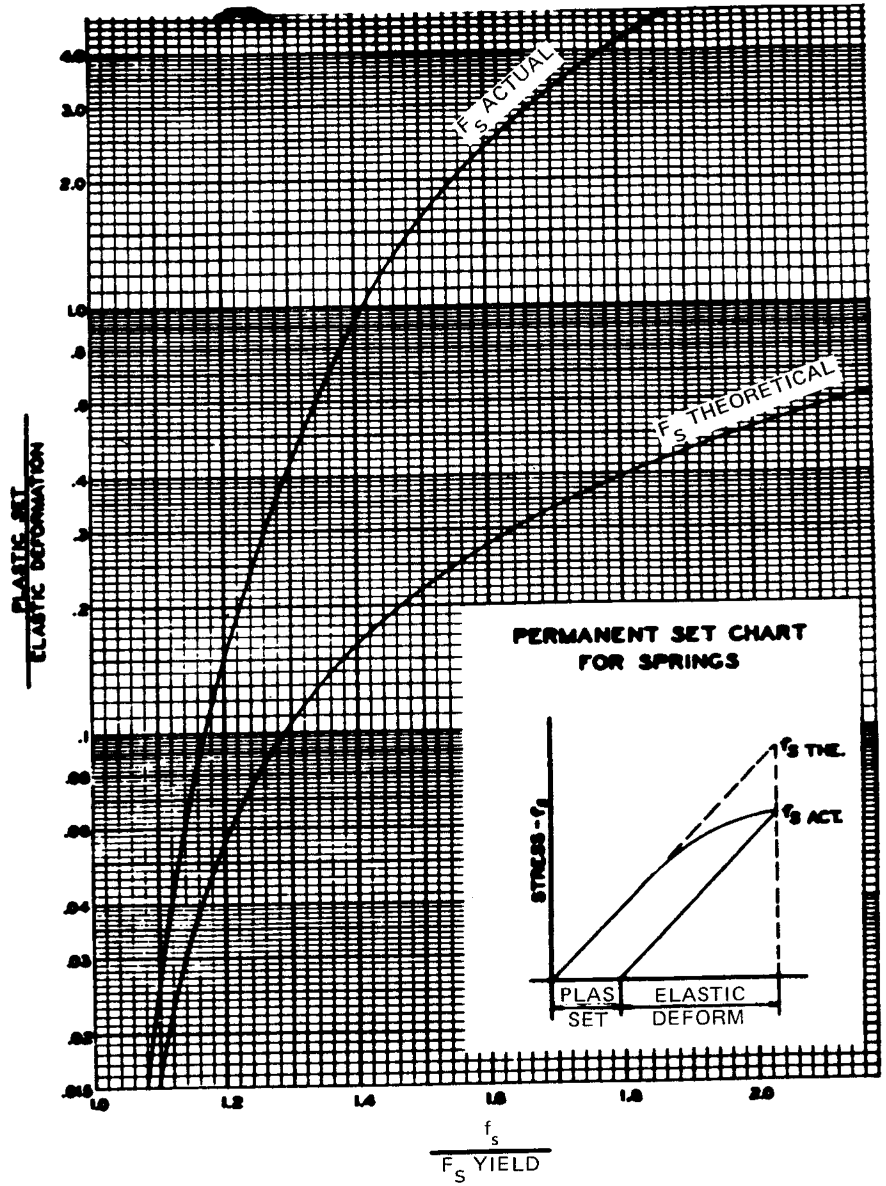
FIGURE 30-3

STRESS CONCENTRATION IN HOOKS ON EXTENSION SPRINGS



C = STRESS CONCENTRATION FACTOR = $\frac{R_o}{R_i}$

FIGURE 30-4



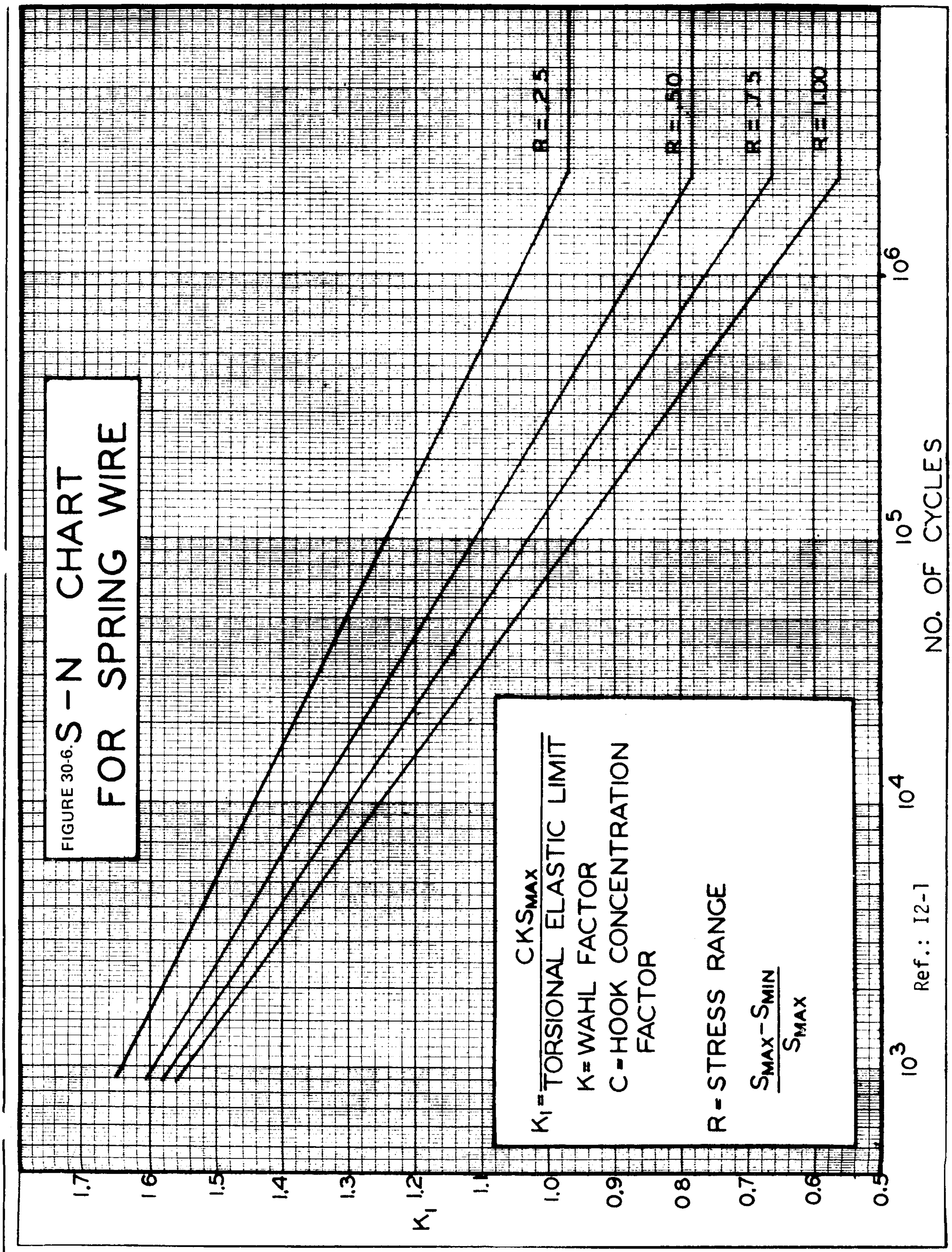


FIGURE 30-6. S-N CHART FOR SPRING WIRE

REFERENCES

1. Wallace Barnes Co. "Mechanical Springs".
2. S.A.E. War Engineering Board "Manual on Design and Application of Helical and Spiral Springs for Ordnance".
3. The Colorado Fuel and Iron Corp. "Springs and Formed Wires".
4. The Spring Manufacturer's Association, Inc. "Standards for Mechanical Springs".
5. American Steel and Wire Co. "Manual of Spring Engineering".
6. A.S.T.M. Standards - 1949 Part I
7. A.S.T.M. Standards - 1942 Part I
8. A.S.T.M. "Tool Engineers' Handbook"
9. S.A.E. Standards "SAE Handbook - 1948"
10. Republic Aviation Corp. "Springs; Specification for"
11. S.A.E. "Proper Use of Spring Materials"
12. NA-S-58
13. MIL-W-6713
14. QQ-W-474a
15. MIL-W-6101
17. Product Engineering December 1950 "Simplified Helical Spring Design"
18. Spring Computer Program D9271 (Available in the McAuto Sigma 9 Computer)

33.79 USE OF SPRING DESIGN CHART

a. Direct Application To find spring for given load and installation.

1.1 Use Δ/d as required by geometry for installation, or otherwise use lowest practical value to get least weight spring.

1.2 Enter chart with Δ/d from 1.1 and maximum load P. Pick nearest smaller wire diameter, and interpolate for coil diameter if necessary. Stresses for maximum load P and wire diameter d are read from Figure 30-5.

1.3 Compute number of active coils from

$$N = \frac{\text{maximum required deflection}}{d(\Delta/d) \text{ from chart}}$$

b. Design of Springs for Other Allowable Stress This is required in order to account for a specific fatigue life, to use a different material than used in the chart, or to work heavy, infrequently operated springs to a higher stress.

2.1 Compute the ratio, $\phi = \dfrac{F_s \text{ (proportional limit)}}{f_s \text{ (desired operating stress)}}$

2.2 Multiply P and Δ/d as used in above example by:

$$P' = \phi P \qquad \Delta'd = \Delta/d$$

Select spring from chart using Δ'/d and P'

c. Analysis of Specific Springs

3.1 Locate the intersection of D_o and d applicable to spring.

3.2 Read Δ/d and P from chart, the coordinates of the 3.1 intersection.

3.3 Obtain $F_{s(\text{elas. limit})}$ for specified wire diameter from Figure 30-5.

3.4 f_s for spring $= F_s$ (from 3.3) $\times \dfrac{P_{\text{applied}}}{P_{\text{chart}}}$

3.5 Spring constant $K = \dfrac{P_{\text{chart}}}{\Delta/d \text{ (chart)} \times d \times N_{\text{coils}}}$

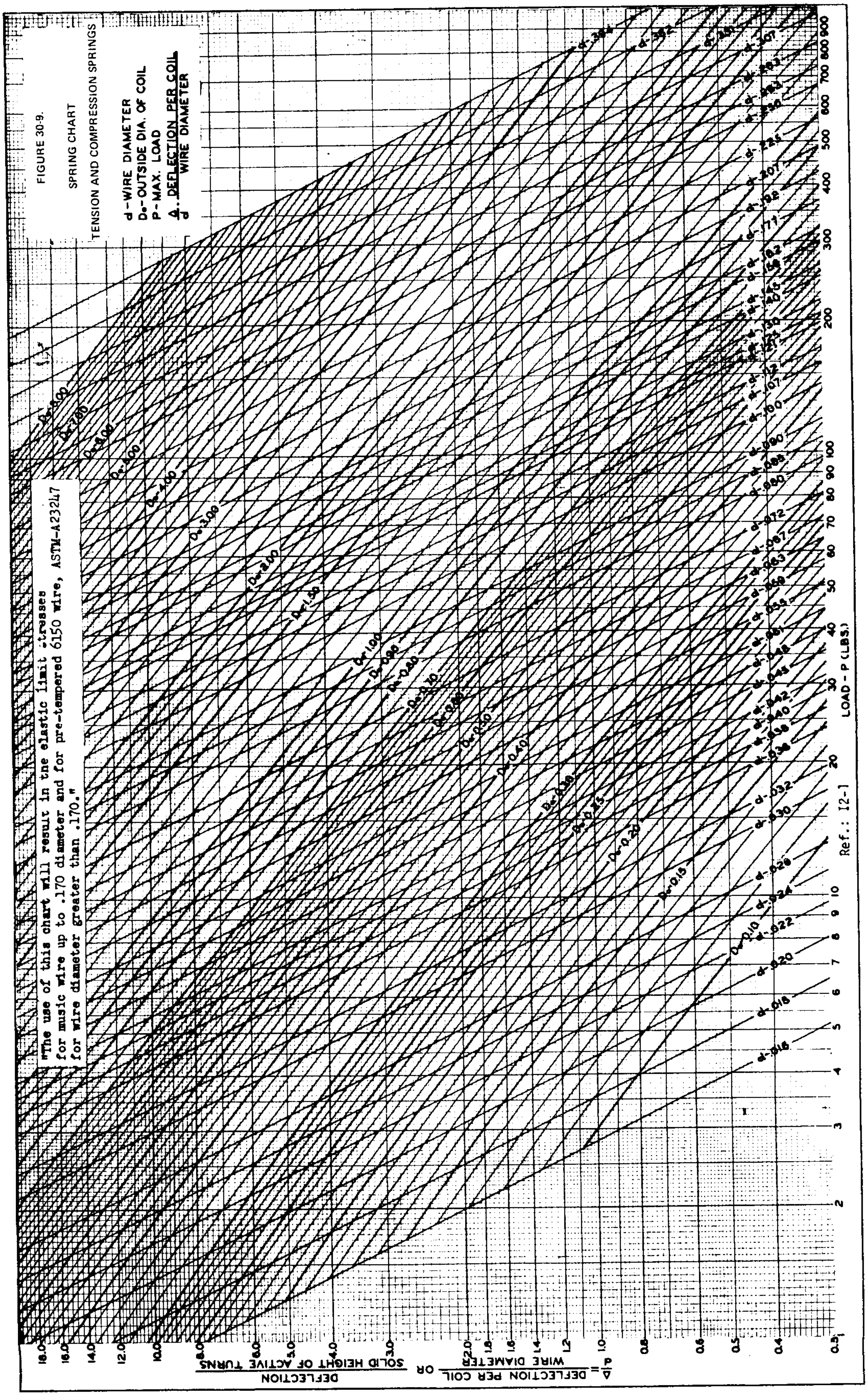


FIGURE 30-9.
SPRING CHART
TENSION AND COMPRESSION SPRINGS
d - WIRE DIAMETER
D - OUTSIDE DIA. OF COIL
P - MAX. LOAD
Δ/d - DEFLECTION PER COIL / WIRE DIAMETER
"The use of this chart will result in the elastic limit stresses for music wire up to .170 diameter and for pre-tempered 6150 wire, ASTM-A23247 for wire diameter greater than .170."
LOAD - P (LBS.)
Δ/d = DEFLECTION PER COIL / WIRE DIAMETER OR DEFLECTION / SOLID HEIGHT OF ACTIVE TURNS
Ref.: I2-1

33.8 JOINT DESIGN

33.81 INTRODUCTORY REMARKS

a. Analysis of Lugs and Shear Pins The method described in this section is semi-empirical and is considered applicable to aluminum or steel alloy lugs. The analysis considers loads in the axial, transverse and oblique directions. Each of these loads is treated in Sections 33.83, 33.84, and 33.85 respectively. See Figure 30-10 for description of directions.

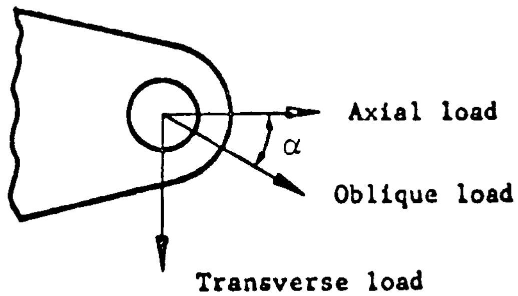


FIGURE 30-10

33.82 REFERENCES

Anon.: "Structures Manual," NASA

Melcoln, M. A. and Hoblit, F. M., "Developments in the Analysis of Method for Determining the Strength of Lugs Loaded Obliquely or Transversely," Product Engineering, June 1953.

Cozzone, F. P., Melcoln, M.A., and Hoblit, F. M., "Analysis of Lugs and Shear Pins Made of Aluminum or Steel Alloys," Product Engineering, May 1950.

Anon.: "Metallic Materials and Elements for Aerospace Vehicle Structures, MIL-HDBK-5A, Department of Defense, 8 February 1966.

Anon.: "Analysis of Lugs and Shear Pins and Comparison to Tests," DACO Report S.M. 22913, 12 September 1957.

33.83 ANALYSIS OF LUGS WITH AXIAL LOADING

A lug-pin combination under tension load can fail in any of the following ways, each of which must be investigated by the methods presented in this section:

a. Tension across the net section. Stress concentration must be considered.

b. Shear tear-out or bearing. These two are closely related and are covered by a single calculation based on empirical curves.

c. Shear of the pin. This is analyzed in the usual manner.

d. Bending of the pin. The ultimate strength of the pin is based on the modulus of rupture.

e. Excessive yielding of bushing (if used).

f. Yielding of the lug is considered to be excessive at a permanent set equal to 0.02 times pin diameter. This condition must always be checked as it is frequently reached at a lower load than would be anticipated from the ratio of the yield stress, F_{ty} to the ultimate stress, F_{tu} for the material.

Notes: Hoop Tension at tip of lug is not a critical condition, as the shear-bearing condition precludes a hoop tension failure.

The lug should be checked for side loads (due to misalignment, etc.) by conventional beam formulas (Figure 30-10).

Analysis procedure to obtain ultimate axial load.

a. Compute (see Figure 30-11 for nomenclature)

$$e/D,\ W/D,\ D/t;\quad A_{br} = Dt,\quad A_t = (W-D)t$$

b. Ultimate load for shearing bearing failure:

Note: In addition to the limitations provided by curves "A" and "B" of Figure 30-13, K_{br} greater than 2.00 shall not be used for lugs made from 0.5-inch-thick or thicker aluminum alloy plate, bar or hand-forged billet.

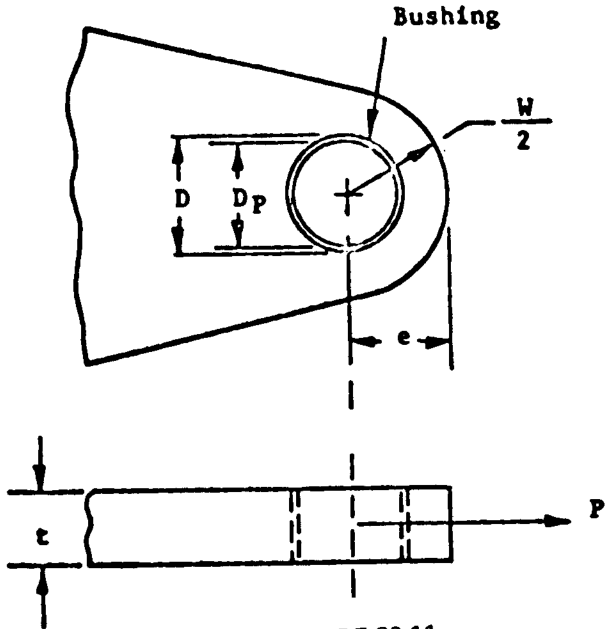


FIGURE 30-11

Enter Figure 30-13 with e/D and D/t to obtain K_{br}

The ultimate load for shear bearing failure, P'_{bru} is

$$P'_{bru} = K_{br} F_{tux} A_{br} \tag{1}$$

where

F_{tux} = ultimate tensile strength of lug material in transverse direction.

c. Ultimate load for tension failure:

Enter Figure 30-14 with W/D to obtain K_t for proper material

The ultimate load for tension failure P'_{tu} is

$$P'_{tu} = K_t F_{tu} A_t \tag{2}$$

where

F_{tu} = ultimate tensile strength of lug material

d. Load for yielding of the lug

Enter Figure 30-15 with e/D to obtain K_{bry}

The yield load, P'_y is

$$P'_y = K_{bry} A_{br} F_{ty} \tag{3}$$

where

F_{ty} = tensile yield stress of the lug material.

e. Load for yielding of the bushing in bearing (if used):

$$P'_{bry} = 1.85 F_{cy} A_{brb} \tag{4}$$

where

F_{cy} = Compressive yield stress of bushing material

$A_{brb} = D_p t$ (Figure 30-11)

f. Pin bending stress

(1) Obtain moment arm "b" as follows:

(See Figure 30-12a) compute for the inner lug

$$r = \left[\left(\frac{e}{D}\right) - \frac{1}{2}\ \frac{D}{t_2}\right] \tag{5}$$

Take the smaller of P'_{bru} and P'_{tu} for the inner lug as $(P'_u)_{min}$ and compute $(P'_u)_{min}/A_{br}F_{tux}$.

Enter Figure 30-16 with $(P'_u)_{min}/A_{br}F_{tux}$ and "r" to obtain the reduction factor, "γ" which compensates for the "peaking" of the distributed pin bearing load near the shear plane. Calculate the moment arm "b" from

$$b = \left(\frac{t_1}{2}\right) + g + \gamma \left(\frac{t_2}{4}\right) \tag{6}$$

where "g" is the gap between lugs as in Figure 30-12a and may be zero. Note that the peaking reduction factor applies only to the inner lugs.

(2) Calculate maximum pin bending moment M, from the equation

$$M = P \left(\frac{b}{2}\right) \tag{7}$$

(3) Calculate bending stress resulting from "M", assuming an My/I distribution.

(4) Obtain ultimate strength of the pin in bending by use of Section 33.82, Reference 1. If the analysis should show inadequate pin bending strength, it may be possible to take advantage of any excess lug strength to show adequate strength for the pin by continuing the analysis as follows:

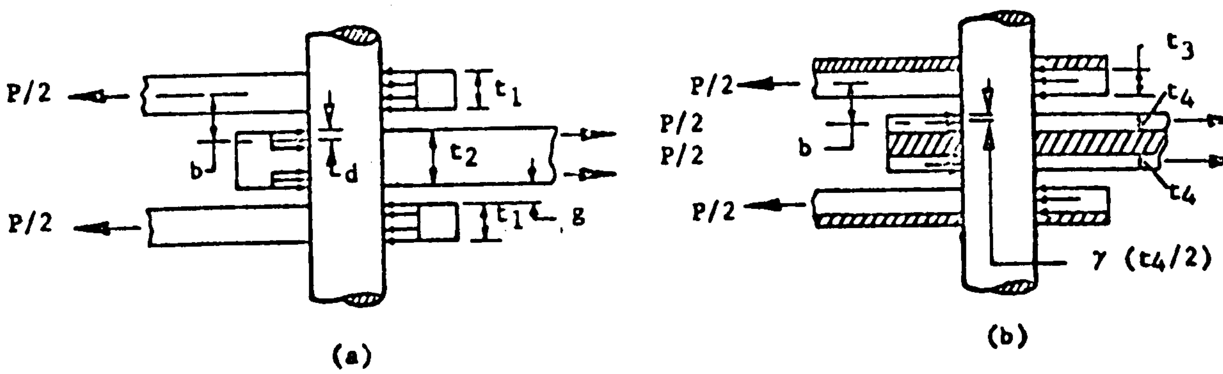


FIGURE 30-12

(5) Consider a portion of the lugs to be inactive as indicated by the shaded area of Figure 30-12b. The portion of the thickness to be considered active may have any desired value sufficient to carry the load and should be chosen by trial and error to give approximately equal margins of safety for the lugs and pin.

(6) Recalculate all lug margins of safety, with ultimate loads reduced in the ratio of active thickness to actual thicknesses.

(7) Recalculate pin bending moment, $M = P\ (b/2)$, and margin of safety using a reduced value of "b" which is obtained as follows:

Compute for the inner lug, Figure 30-12b

$$r = \left[\left(\frac{e}{D}\right) - \frac{1}{2}\right] \frac{D}{2t_4} \tag{8}$$

Take the smaller of P'_{bru} and P'_{tu} for the inner lug, based upon the active thickness, as $(P'_u)_{min}$ and compute $(P'_u)_{min}/A_{br}\ F_{tux}$, where $A_{br} = 2t_4\ D$. Enter Figure 30-16 with $(P'_u)_{min}/A_{br}\ F_{tux}$ and "r" to obtain the reduction factor "γ" for peaking. Then the moment arm is

$$b = \frac{t_3}{2} + g + \gamma\left(\frac{t_4}{2}\right) \tag{9}$$

This reduced value of "b" should not be used if the resulting eccentricity of load on the outer lugs introduces excessive bending stresses in the adjacent structure. In such cases the pin must be strong enough to distribute the load uniformly across the entire lug.

Curve A of Figure 30-16 is a cutoff to be used for all aluminum alloy hand-forged billet when the long transverse grain direction has the general direction C in the sketch.

Curve B is a cutoff to be used for all aluminum alloy plate, bar and hand-forged billet when the short transverse grain direction has the general direction C in the sketch, and for die forging when the lug contains the parting plane in a direction approximately normal to the direction C.

Legend - Figure 30-14 - L, T, N, indicate grain in direction F in sketch.

L = longitudinal
T = long transverse
N = short transverse (normal)

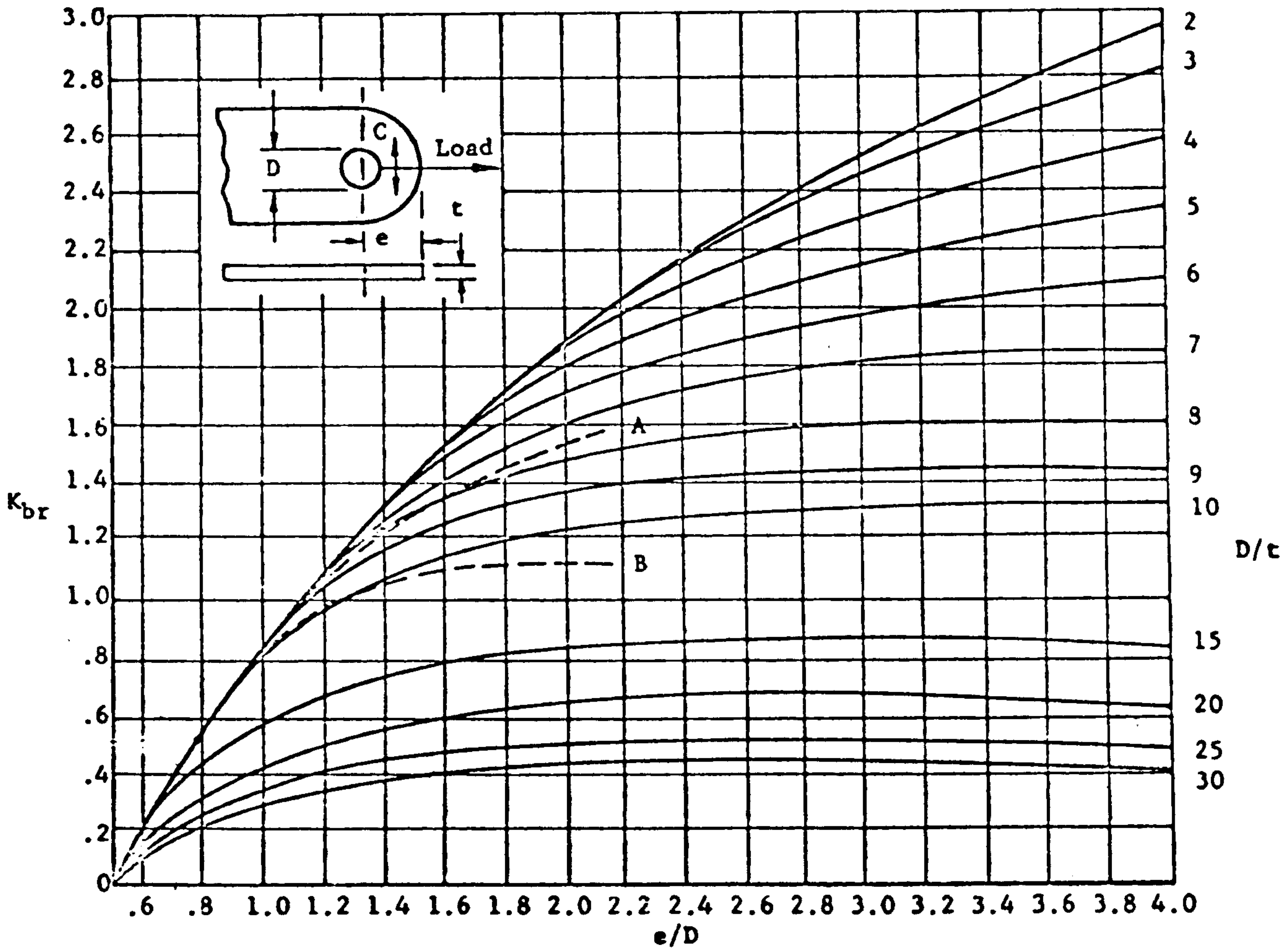


FIGURE 30-13

Curve 1

4130, 4140, 4340 and 8630 steel
2014-T6 and 7075-T6 plate $\leq$ 0.5 inch (L, T)
7075-T6 bar and extrusion (L)
2014-T6 hand-forged billet $\leq$ 144 square inch (L)
2014-T6 and 7075-T6 die forgings (L)

Curve 2

2014-T6 and 7075-T6 plate > 0.5 inch, $\leq$ 1 inch
7075-T6 extrusion (T, N)
7075-T6 hand-forged billet $\leq$ 36 square inch (L)
2014-T6 hand-forged billet > 144 square inch (L)
2014-T6 hand-forged billet $\leq$ 36 square inch (T)
2014-T6 and 7075-T6 die forgings (T)
17-4 PH
17-7 PH-THD

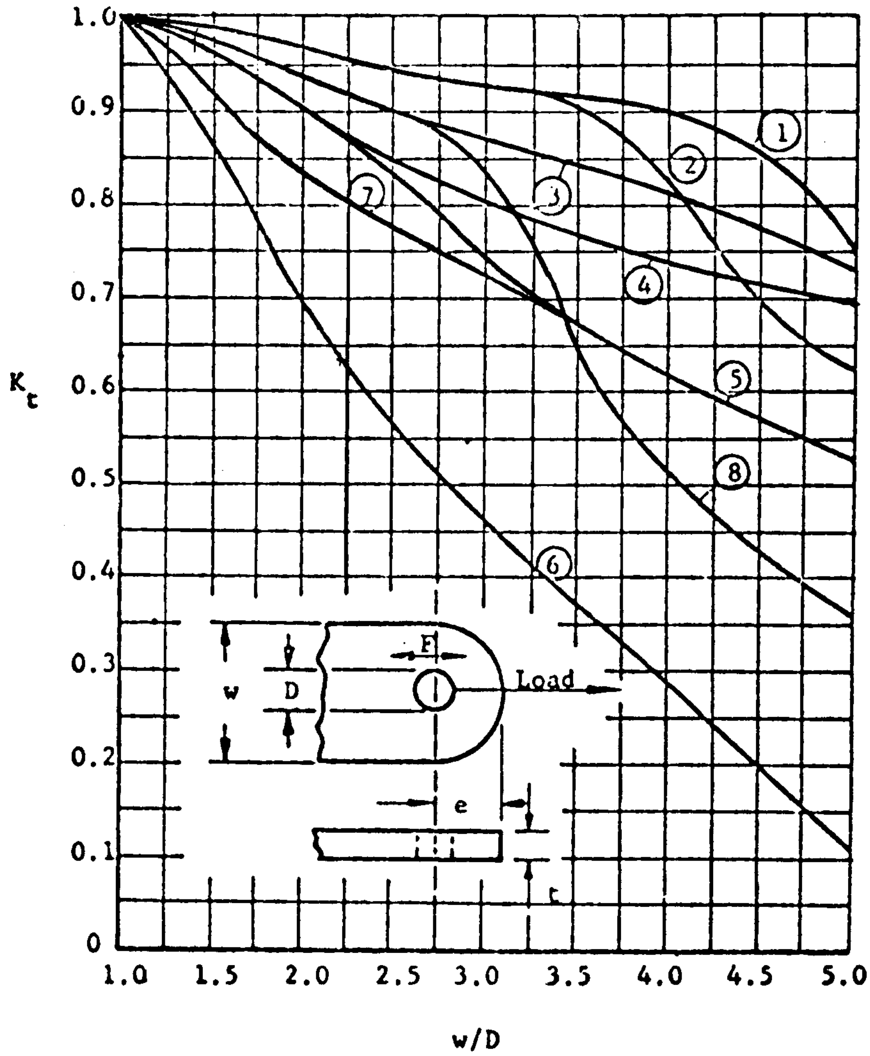


FIGURE 30-14

Curve 3

2024-T6 plate (L, T)
2024-T4 and 2024-T42 extrusion (L, T, N)

Curve 4

2024-T4 plate (L, T)
2024-T3 plate (L, T)
2014-T6 and 7075-T6 plate > 1 inch (L, T)
2024-T4 bar (L, T)
7075-T6 hand-forged billet > 36 square inch (L)
7075-T6 hand-forged billet $\leq$ 16 square inch (T)

Curve 5

195T6, 220T4, and 356T6 aluminum alloy casting
7075-T6 hand-forged billet > 16 square inch (T)
2014-T6 hand-forged billet > 36 square inch (T)

Curve 6

Aluminum alloy plate, bar, hand-forged billet, and die forging (N)

Note: For die forgings, N direction exists only at the parting plane. 7075-T6 bar (T)

Curve 7

18-8 stainless steel, annealed

Curve 8

18-8 stainless steel, full hard

Note: For 1/4, 1/2 and 3/4 hard, interpolate between Curves 7 and 8.

Special Applications

a. Irregular lug section - bearing load distributed over entire thickness.

For lugs of irregular section having bearing stress distributed over the entire thickness, an analysis is made based on an equivalent lug with rectangular sections having an area equal to the original section.

b. Critical bearing stress - MIL-HDBK-5A lists the values of the ultimate and yield bearing stress of materials for e/D values of 2.0 and 1.5, these are valid for values of D/t to 5.5. The ultimate and yield bearing stress for geometrical conditions outside of the above range may be determined in the following manner:

(1) Ultimate bearing stress: For the particular D/t and e/D, obtain K_{br} from Figure 30-13 then:

$$F_{bru} = K_{br} F_{tux}$$

where

F_{tux} = Ultimate strength of lug material in transverse direction

(2) Yield bearing stress: With the particular e/D obtain K_{bry} from Figure 30-15. Then

$$F_{bry} = K_{bry} F_{tyx}$$

where

F_{tyx} = Tensile yield stress of lug material in transverse direction.

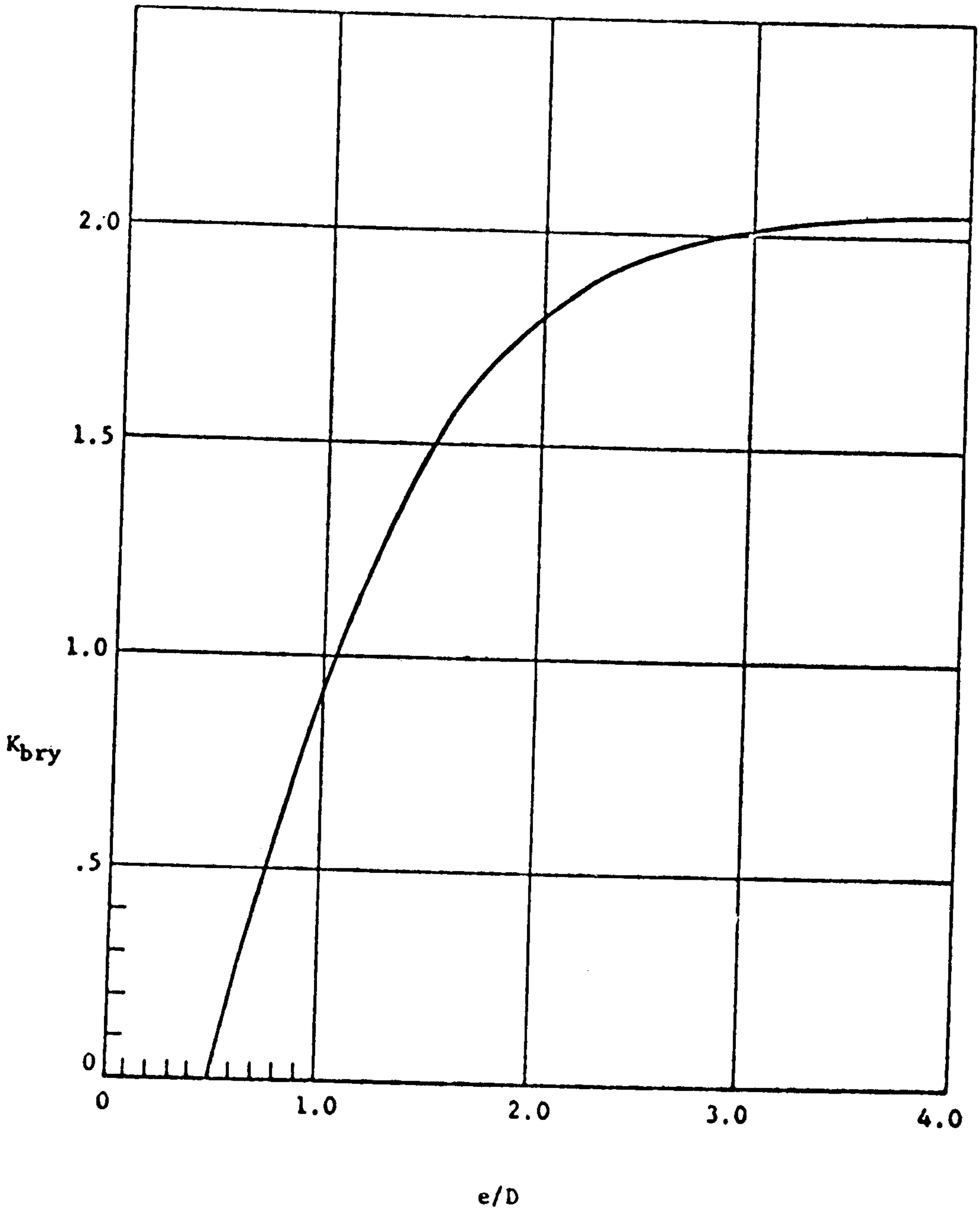


FIGURE 30-15

Peaking Factors for Pin Bending. Dash Lines Indicate Region Where These Theoretical Curves are not Substantiated By Test Data.

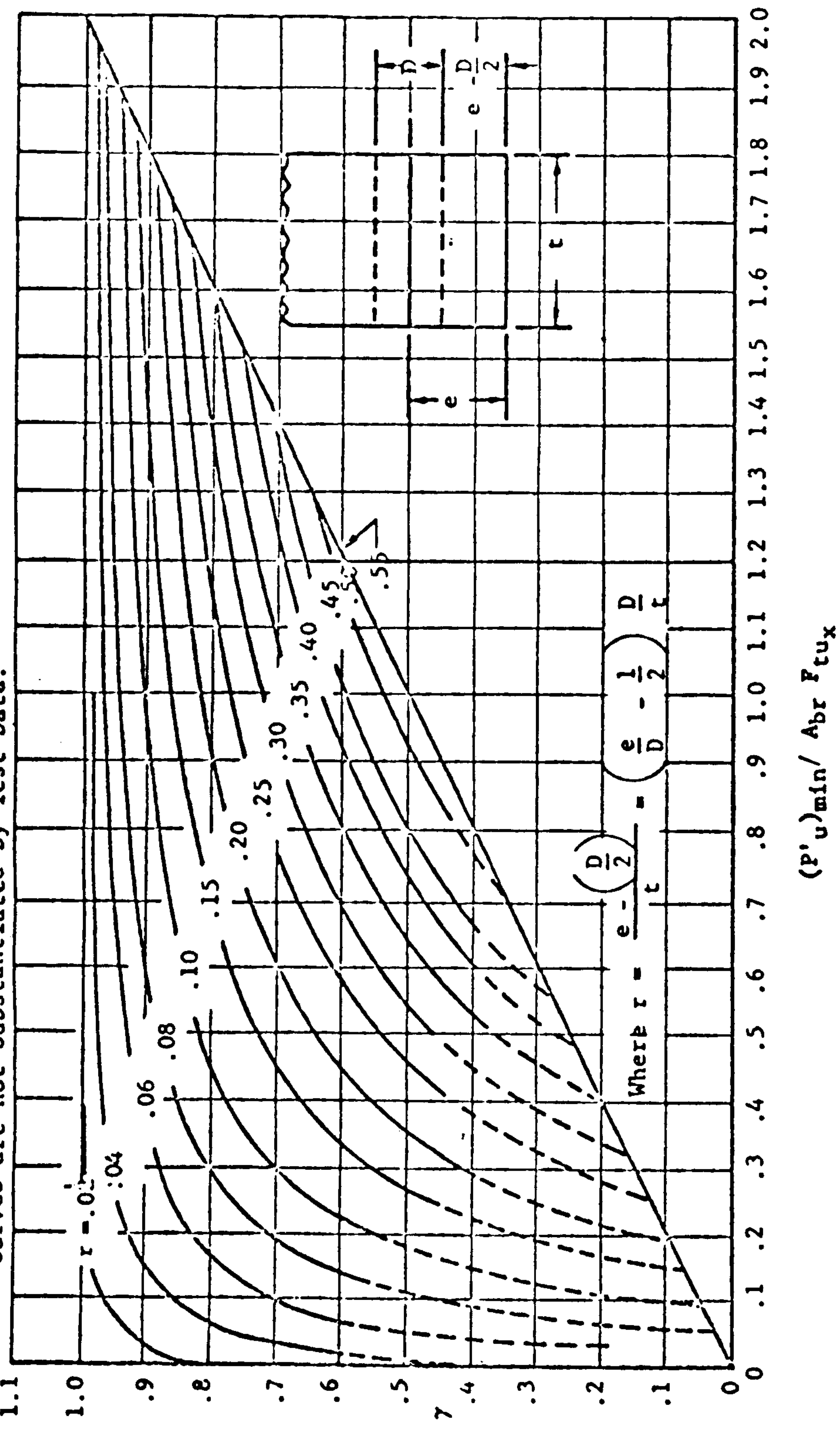


FIGURE 30-16

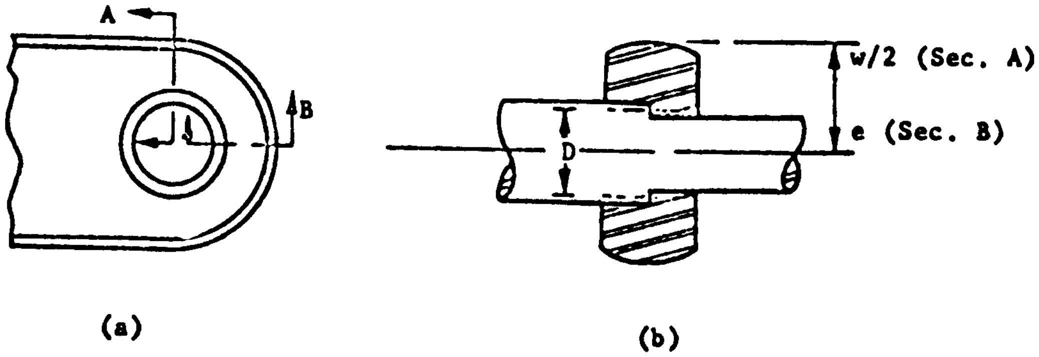


FIGURE 30-17

c. Eccentrically located hole - If the hole is located as in Figure 30-18 (e_1 less than e_2), the ultimate and yielded lug loads are determined by obtaining P'_{bru}, P'_{tu} and P'_y for the equivalent lug shown and multiplying by the factor

$$\text{factor} = \frac{e_1 + e_2 + 2D}{2e_2 + 2D}$$

d. Multiple shear connections - Lug-pin combinations having the geometry shown in Figure 30-19 are analyzed according to the following criteria:

(1) The load carried by each lug is determined by distributing the total applied load "P" among the lugs as shown on Figure 30-19 and the value of "C" is obtained from Table 30-1.

(2) The maximum shear load on the pin is given in Table 30-1.

(3) The maximum bending moment in the pin is given by the formula, $M = P_1 b/2$ where "b" is given in Table 30-1.

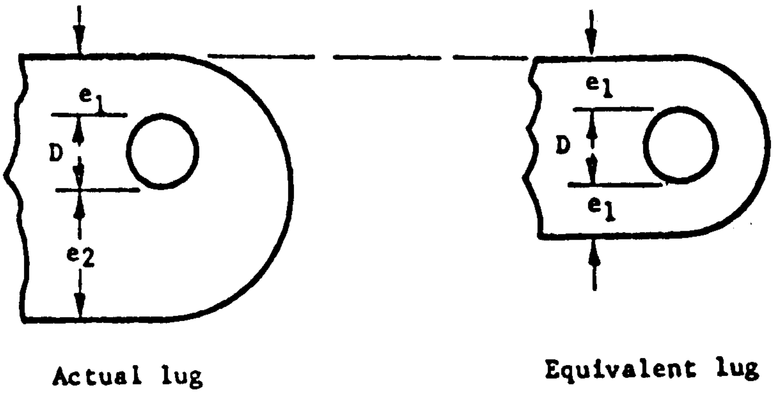


FIGURE 30-18

33.84 ANALYSIS OF LUGS WITH TRANSVERSE LOADING

Shape Parameter In order to determine the ultimate and yield loads for lugs with transverse loading, the shape of the lug must be taken into account. This is accomplished by use of a shape parameter given by:

$$\text{Shape parameter} = \frac{A_{av}}{A_{br}}$$

where

A_{br} is the bearing area = Dt

A_{av} is the weighted average area given by

$$A_{av} = \frac{6}{(3/A_1) + (1/A_2) + (1/A_3) + (1/A_4)}$$

A_1, A_2, A_3 and A_4 are areas of the lug sections indicated in Figure 30-20.

a. Obtain the areas A_1, A_2, A_3, and A_4 as follows:

(1) A1, A2, and A4 are measured on the planes indicated in Figure 30-20a (perpendicular to the axial center line), except that in a necked lug, as shown in Figure 30-20b, A1 and A4 should be measured perpendicular to the local wall centerline.

(2) A3 is the least area on any radial section around the hole.

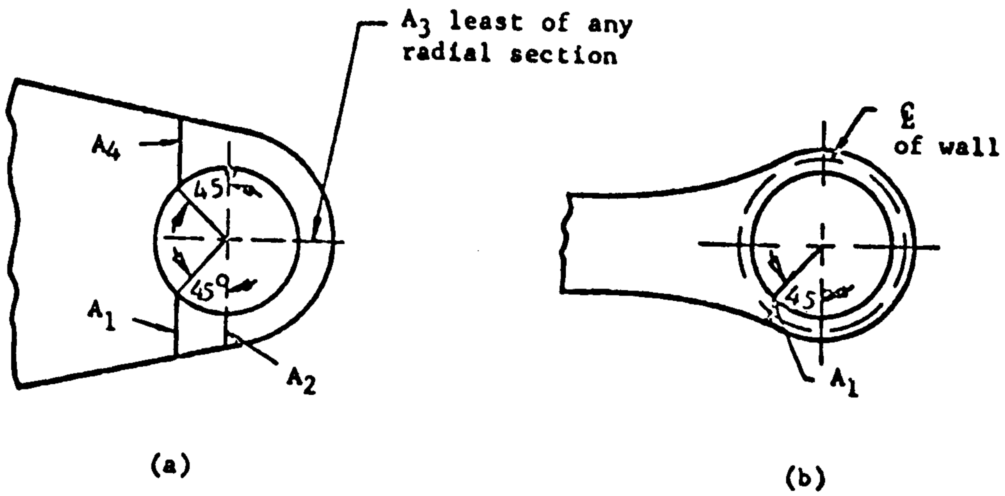


FIGURE 30-20

(3) Thought should always be given to assure that the areas A_1, A_2, A_3, and A_4 adequately reflect the strength of the lug. For lugs of unusual shape (e.g., with sudden changes of cross section), an equivalent lug should be sketched as shown in Figure 30-21 and used in the analysis.

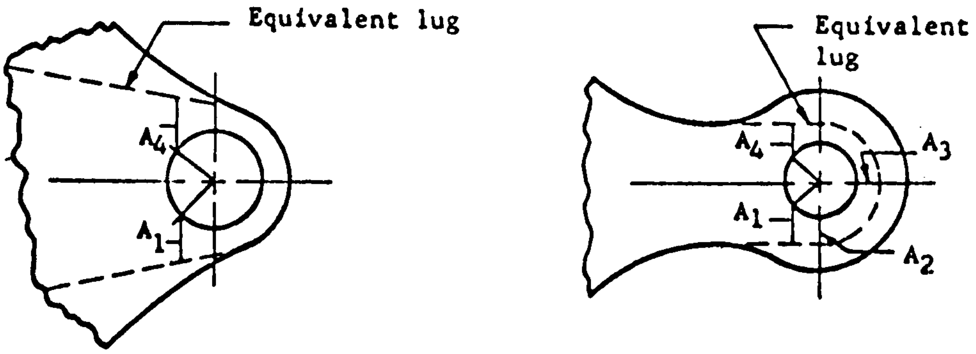


FIGURE 30-21

b. Obtain the weighted average

$$A_{av} = \frac{6}{(3/A_1) + (1/A_2) + (1/A_3) + (1/A_4)}$$

c. Compute $A_{br} = Dt$ and A_{av}/A_{br}

d. Ultimate load P'_{tru} for lug failure:

(a) Obtain K_{tru} from Figure 30-23

(b) $P'_{tru} = K_{tru} A_{br} F_{tux}$

e. Yield load P'_y of the lug:

(1) Obtain K_{try} from Figure 30-23

(2) $P'_y = K_{try} A_{br} F_{tyx}$

f. Check bushing yield and pin shear as outlined previously.

g. Investigate pin bending as for axial load with following modifications: Take $(P'_u)_{min} = P'_{tru}$. In the equation

$r = [e - (D/2)]/t$ use for the $[e - (D/2)]$

term the edge distance at $\alpha = 90$ degrees.

Curve 10

2014-T6 and 7075-T6 plate > 1 inch
7075-T6 hand-forged billet ≤ 16 square inch

Curve 11

7075-T6 hand-forged billet > 16 square inch
2014-T6 hand-forged billet > 36 square inch

All curves are for K_{tru} except the one noted as K_{try}

Note: The curve for 125,000 HT steel in Figure 30-23 agrees closely with test data. Curves for all other materials have been obtained by the best available means of correcting for material properties and may possibly be very conservative to some places.

In no case should the ultimate transverse load be taken as less than that which could be carried by cantilever beam action of the portion of the lug under the load (Figure 30-22). The load that can be carried by cantilever beam action is indicated very approximately by curve (A) in Figure 30-23, should K_{tru} be below curve (A), separate calculation as a cantilever beam is warranted.

33.85 ANALYSIS OF LUGS WITH OBLIQUE LOADING

Interaction Relation In analyzing lugs subject to oblique loading it is convenient to resolve the loading into axial and transverse components (denoted by subscripts "a" and "tr" respectively), analyze the two cases separately and utilize the results by means of an interaction equation. The interaction equation $R_a^{1.6} + R_{tr}^{1.6} = 1$, where R_a and R_{tr} are ratios of applied to critical loads in the indicated directions, is to be used for both ultimate and yield loads for both aluminum and steel alloys.

Where, for ultimate loads

R_a = (Axial component of applied load) divided by (smaller of P'_{bru} and P'_{tu} from Equation 1 and Equation 2)

R_{tr} = (Transverse component of applied load) divided by (P'_{tru} from analysis procedure for α = 90 degrees)

and for yield load:

R_a = (Axial component of applied load) divided by (P'_y from Equation 3)

R_{tr} = (Transverse component of applied load) divided by (P'_{try} from Analysis Procedure for α = 90 degrees)

Analysis Procedure

a. Resolve the applied load into axial and transverse components and obtain the lug ultimate and yield margins of safety from the interaction equation:

$$M.S. = \frac{1}{\left[R_a^{\,1.6} + R_{tr}^{\,1.6}\right]^{0.625}} - 1$$

b. Check pin shear and bushing yield as in Section 30.83.

c. Investigate pin bending using the procedures for axial load modified as follows:

$$\text{Take } (P'_u)_{min} = \frac{P}{\left(R_a^{\,1.6} + R_{tr}^{\,1.6}\right)^{0.625}}$$

In the equation $r = [e - (D/2)]/t$ use for the $[e - (D/2)]$ term the edge distance at the value of "α" corresponding to the direction of load on the lug.

33.9 STRESSES DUE TO PRESS FIT BUSHINGS* Pressure between a lug and bushing assembly having negative clearance can be determined from consideration of the radial displacements. After assembly, the increase in inner radius of the ring (lug) plus the decrease in outer radius of the bushing equals the difference between the radii of the bushing and ring before assembly.

$$\delta = u_{ring} - u_{bushing} \tag{1}$$

δ - difference between outer radius of bushing and inner radius of the ring

u - radial displacement, positive away from the axis of ring or bushing

Radial displacement at the inner surface of a ring subjected to internal pressure p is:

$$u = \frac{D_p}{E_{ring}}\left[\frac{C^2 + D^2}{C^2 - D^2} + \nu_{ring}\right] \tag{2}$$

*Timoshenko, Strength of Materials, Volume 2, 1941, p241.

Radial displacement at the outer surface of a bushing subjected to external pressure p is:

$$u = -\frac{D_p}{E_{bush.}}\left[\frac{B^2 + A^2}{B^2 - A^2} - \nu_{bush.}\right] \tag{3}$$

A - Inner radius of bushing
B - Outer radius of bushing
C - Outer radius of ring (lug)
D - Inner radius of ring (lug)
E - Modulus of elasticity
ν - Poisson's ratio

Substitute Equations (2) and (3) into Equation (1) and solve for p;

$$p = \frac{\delta}{\frac{D}{E_{ring}}\left(\frac{C^2 + D^2}{C^2 - D^2} + \nu_{ring}\right) + \frac{B}{E_{bush.}}\left(\frac{B^2 + A^2}{B^2 - A^2} - \nu_{bush.}\right)}$$

Maximum radial and tangential stresses for a ring subjected to internal pressure occur at the inner surface of the ring (lug).

$$f_r = -p \qquad f_t = p\left[\frac{C^2 + D^2}{C^2 - D^2}\right]$$

Positive sign indicates tension. The maximum shear stress at this point is

$$f_s = \frac{f_t - f_r}{2}$$

Maximum radial stress for a bushing subjected to external pressure occurs at the outer surface of the bushing.

$$f_r = -p$$

Maximum tangential stress for a bushing subjected to external pressure occurs at the inner surface of the bushing.

$$f_t = -\frac{2pB^2}{B^2 - A^2}$$

The allowable press fit stress may be based on:

a. Stress Corrosion The maximum allowable press fit stress in magnesium alloys should not exceed 8000 psi. For all aluminum alloys the maximum press fit stress should not exceed 0.50 F_{ty}.

b. Static Fatigue Static fatigue is the brittle fracture of metals under sustained loading, and in steel may result from several different phenomena, the most familiar of which is hydrogen embrittlement. Steel parts heat treated above 200 ksi, which by nature of their function or other considerations are exposed to hydrogen embrittlement, should be designed to an allowable press fit stress of 25 percent F_{tu}.

c. Ultimate Strength Ultimate strength cannot be exceeded, but is not usually critical in a press fit application.

d. Fatigue Life The hoop tension stresses resulting from the press fit of a bushing in a lug will reduce the stress range for oscillating loads thereby improving fatigue life.

The presence of hard brittle coatings in holes that contain a press fit bushing or bearing can cause premature failure by cracking of the coating or by high press fit stresses caused by build-up of coating. Therefore, Hardcoat or HAE Coatings should not be used in holes that will subsequently contain a press fit bushing or bearing.

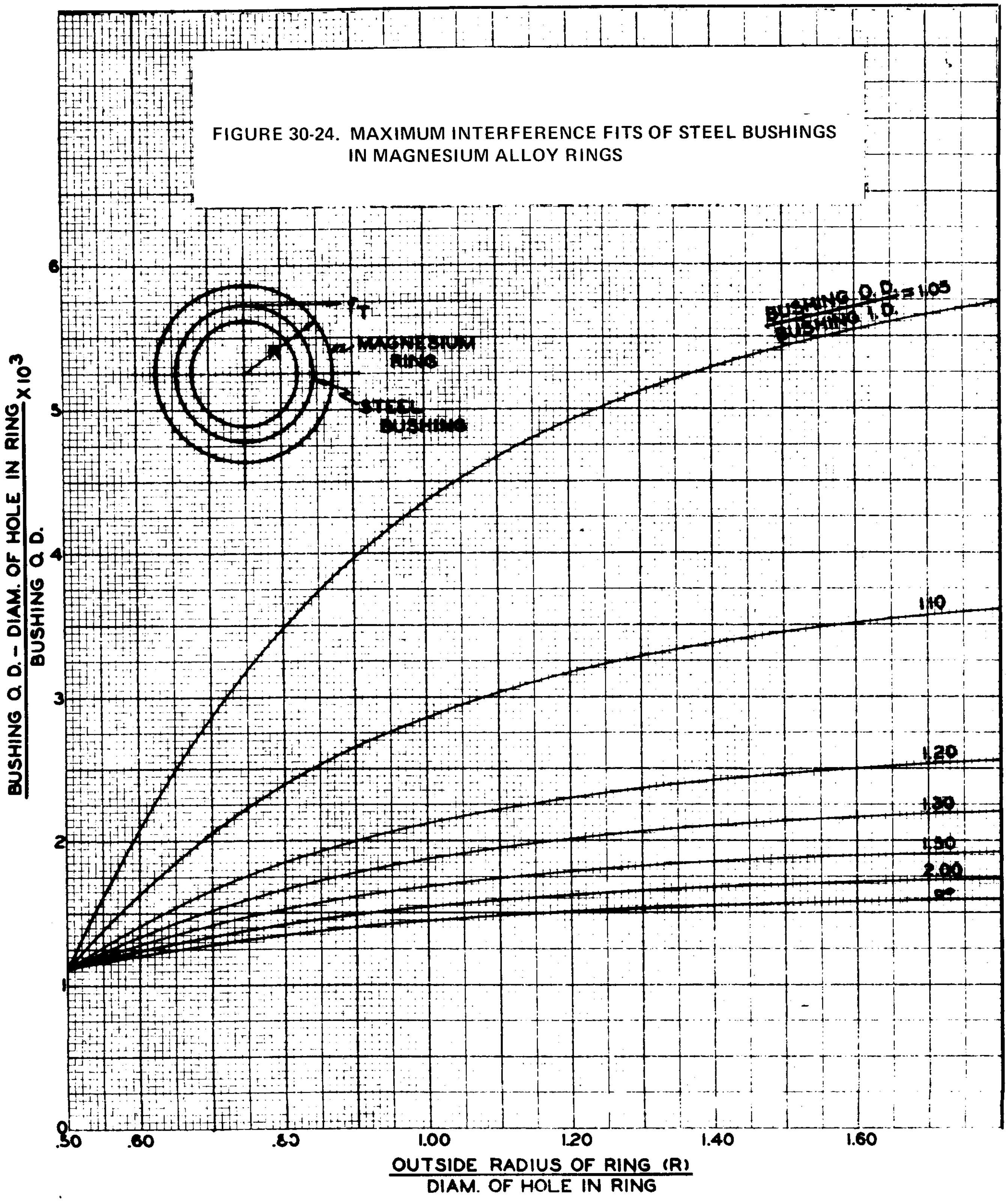


FIGURE 30-24. MAXIMUM INTERFERENCE FITS OF STEEL BUSHINGS IN MAGNESIUM ALLOY RINGS

O. D. of the bushing is the after-plating diameter of the bushing.

The curves are based upon a maximum allowable interference tangential stress of 8000 psi.

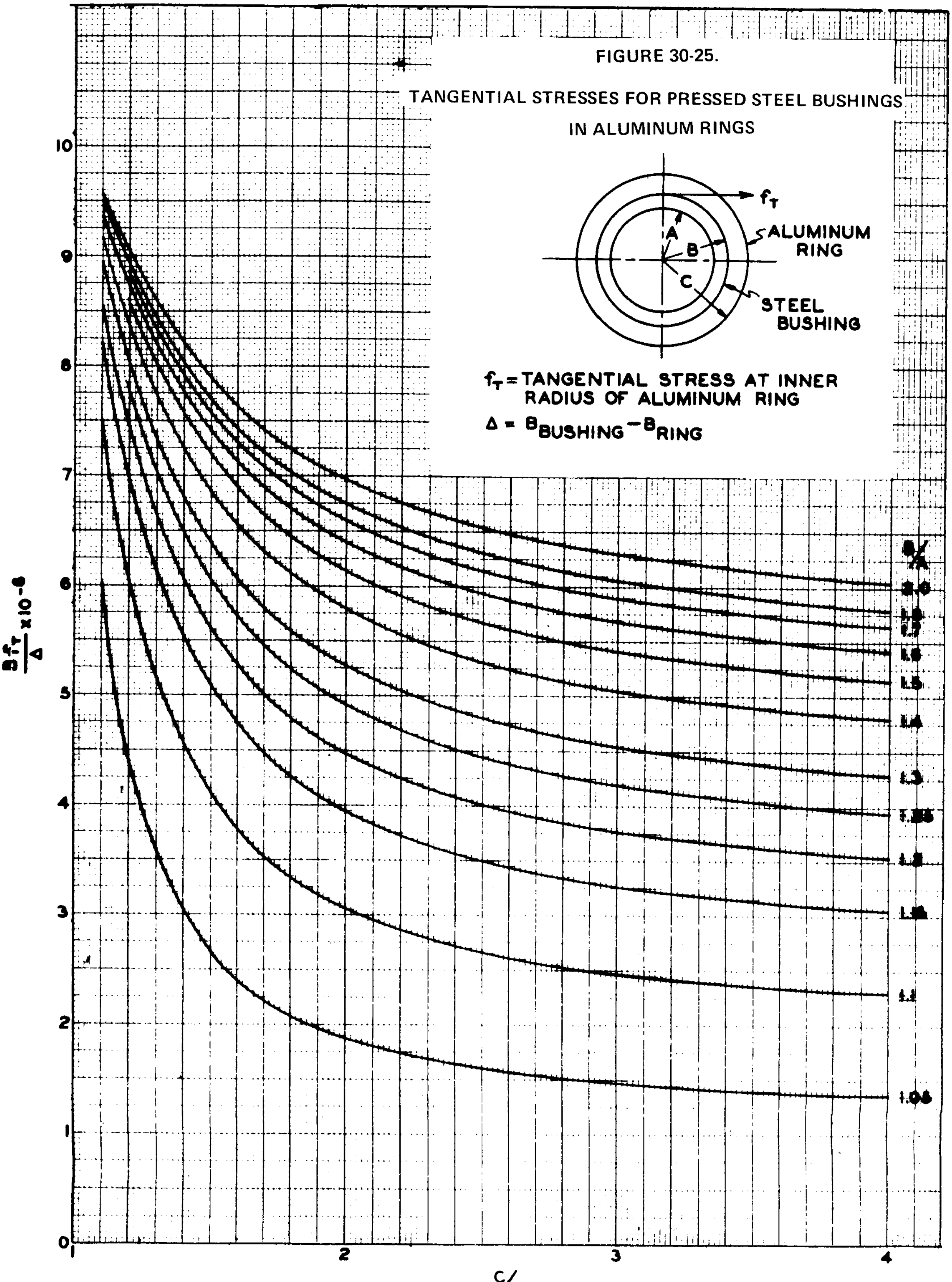


FIGURE 30-25.

TANGENTIAL STRESSES FOR PRESSED STEEL BUSHINGS IN ALUMINUM RINGS

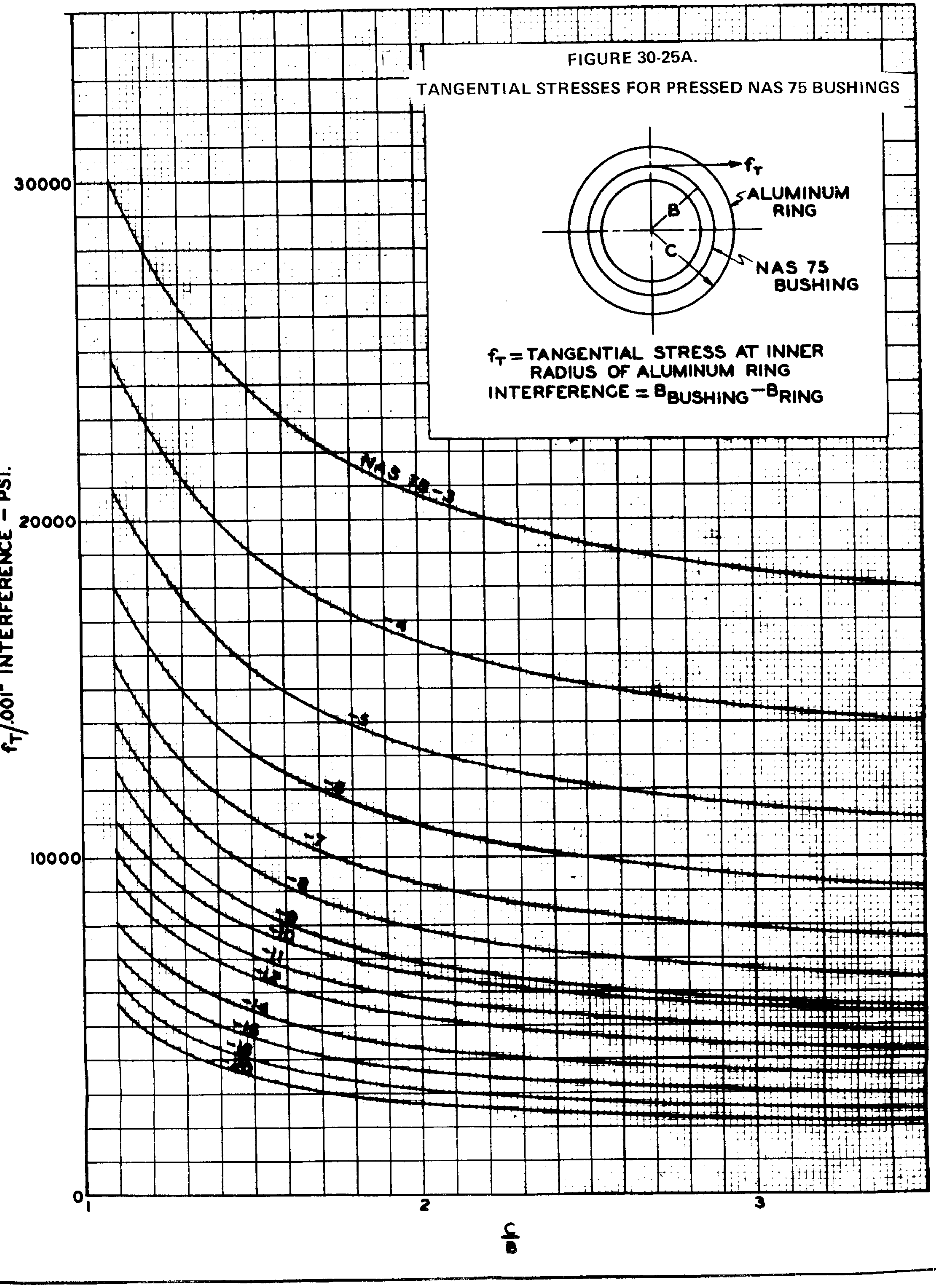


FIGURE 30-25A.
TANGENTIAL STRESSES FOR PRESSED NAS 75 BUSHINGS
f_T
ALUMINUM RING
B
C
NAS 75 BUSHING
f_T = TANGENTIAL STRESS AT INNER RADIUS OF ALUMINUM RING
INTERFERENCE = $B_{BUSHING} - B_{RING}$
30000
20000
10000
0
1
2
3
f_T/.001" INTERFERENCE – PSI.
$\frac{C}{B}$
NAS 75-3
-4
-5
-6
-7
-8
-11
-12

34.0 THE EFFECTS OF FRICTION ON THE POWER REQUIREMENTS OF MECHANISMS

34.1 INTRODUCTION In an ideal, frictionless system the mechanism efficiency is unity and the work put into the system is equal to the useful work received from the system. Due to friction, however, the ideal system never exists, and the mechanism efficiency is always less than unity. It is the purpose of this article to present and discuss methods of analysis of mechanisms that take into account the effects of friction.

34.2 SUMMARY Two types of friction are considered in this article. They are friction in rotating joints and friction in sliding parts.

The evaluation of the effects of friction in rotating joints by the method of friction circles is presented in Section 34.3, and the effects of friction in slides are evaluated by a conventional graphical method in Section 34.4.

Examples are presented in Section 34.5. An example of the application of the recommended procedure for the determination of the power requirements of an actuator operating a landing gear retraction mechanism is presented in Section 34.51. Section 34.52 discusses effects of sliding friction on the functioning of the mechanism in the desired direction.

34.3 THE EFFECTS OF FRICTION IN ROTATING JOINTS In an ideal, frictionless mechanism the internal loads between elements of the system pass through the centers of the connecting joints. This is assumed to be true if anti-friction bearings are used in the joints.

When bushings, or plain bearings are used, this is no longer true. Due to the effects of friction, the load lines displace themselves off the centers of the joints creating moments that oppose the impending rotation of the joints. In plain bearings an overall coefficient of friction of $\mu = 0.20$ is recommended as usually conservative for design. In individual cases, higher (up to 0.60) values for μ may have to be used.

The method of friction circles permits the calculation of the magnitude of these load line displacements, thus establishing the correct force system which is used to determine the power requirements of the mechanism.

The basic theory of the method of friction circles is now briefly discussed.

Consider the simple system shown in Figure 30.26. The mechanism is statically in balance with the loads and reactions passing through the centers of all the joints.

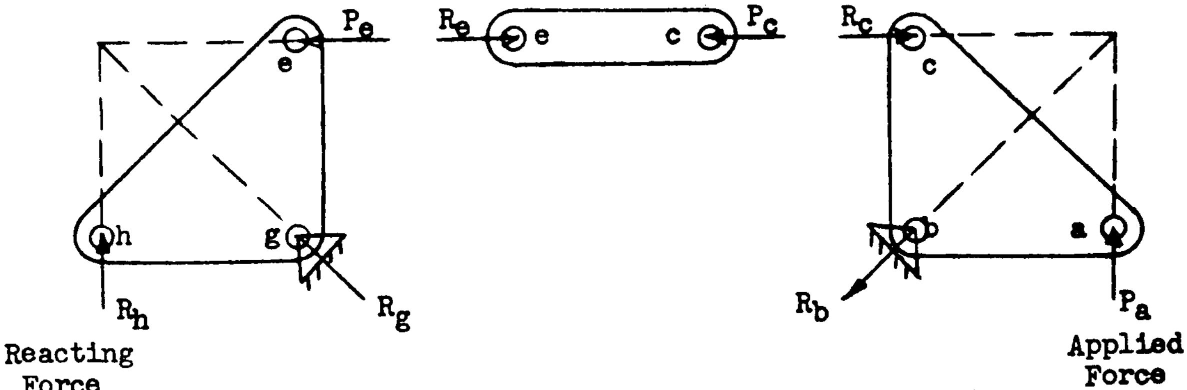
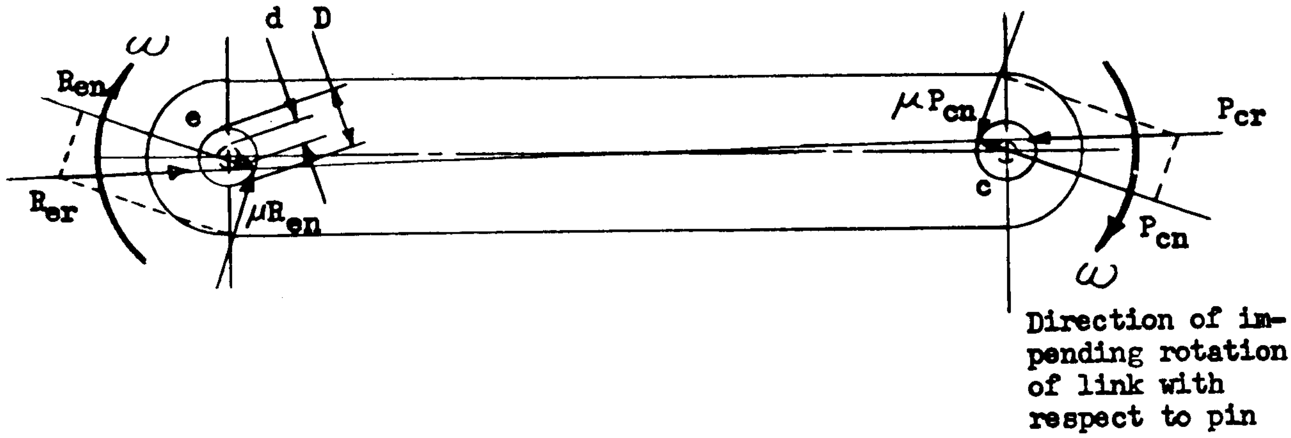
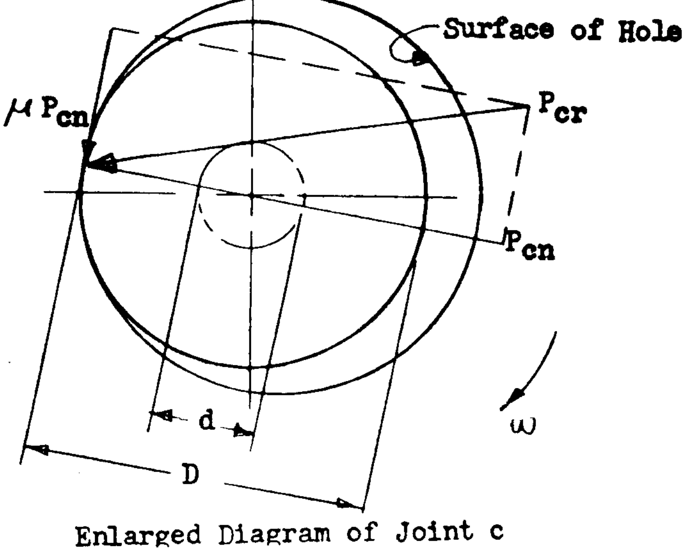


FIGURE 30-26

Now suppose the system is allowed to move under load. Friction forces are set up in the joints, due to the sliding of the parts relative to their connecting pins. This introduces moments about the centers of the joints opposing the impending rotation. This is illustrated with a free body of the link ce.



Consider now the forces acting at joint c of the link ce as shown to an exaggerated scale.



Enlarged Diagram of Joint c

P_{cn} corresponds to P_c of Figure 30-26. Due to the pin rolling along the surface of the hole as rotation takes place, P_{cn} does not act on the line connecting the centers of joints c and e as P_c in Figure 30-26, but acts normal to the surface on which the pin bears against the wall of the hole. P_{cn} does, however, pass through the center of the pin. As sliding takes place between the pin and the hole, friction forces are introduced normal to P_{cn} and tangent to the surface on which the pin bears. These friction forces oppose the impending rotation of the joint.

The normal and the friction forces combine to form the resultant P_{cr} which acts at a distance d/2 from the center of the pin.

It is seen from the free body of link ce that P_{cr} and R_{er} must be equal and collinear for the link to be in balance.

The distance d/2 is known as the radius of the friction circle and may be established in terms of known parameters by equating moments about the center of the pin.

$$\mu P_{cn}(D/2) = P_{cr}(d/2)$$

But

$$P_{cr} = P_{cn}\sqrt{1 + \mu^2}$$

Therefore

$$d = \frac{\mu}{\sqrt{1 + \mu^2}} D$$

Or for all practical purposes when μ is small

$$d = \mu D$$

It is important that the resultant load be applied so that the direction of the friction moment it creates about the center of the joint be such as to oppose the impending rotation of the link about the pin. In other words, the statics of the free body as shown in Figure 30-26 must be satisfied.

Figure 30-27 illustrates graphically, by the use of the friction circle concept, how the mechanical advantage of the simple system of Figure 30-26 is reduced because of friction in the joints.

Note that on each of the bellcranks the lever arms for the applied forces become smaller while those for the reacting forces became larger; therefore, in order to keep the reacting force at joint j the same as for a frictionless system, the applied force must be increased.

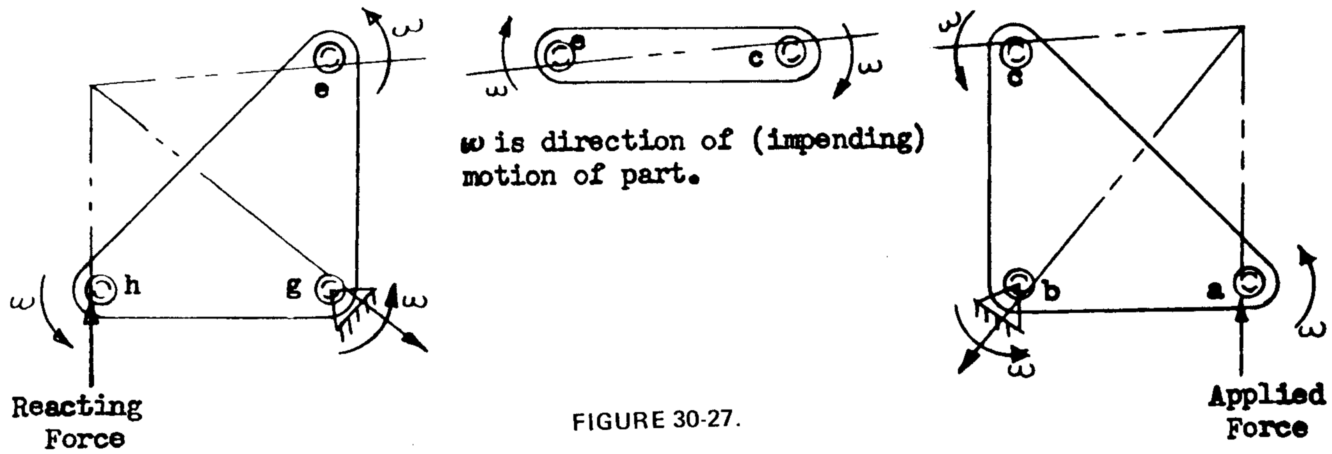


FIGURE 30-27.

Particular care should be taken for mechanisms with links operating near dead center, i.e., when the joints of two connecting links all lie on the same line. When the links are loaded, the effects of friction greatly increase the requirements for moving the linkage over dead center, and if the available power is not sufficient to overcome these friction effects, the mechanism will not function properly.

Some illustrations of mechanisms operating near dead center are shown in Figures 30-28 and 30-29.

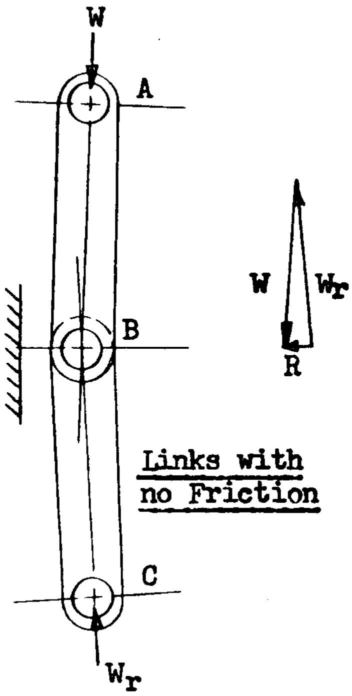


FIGURE 30-28

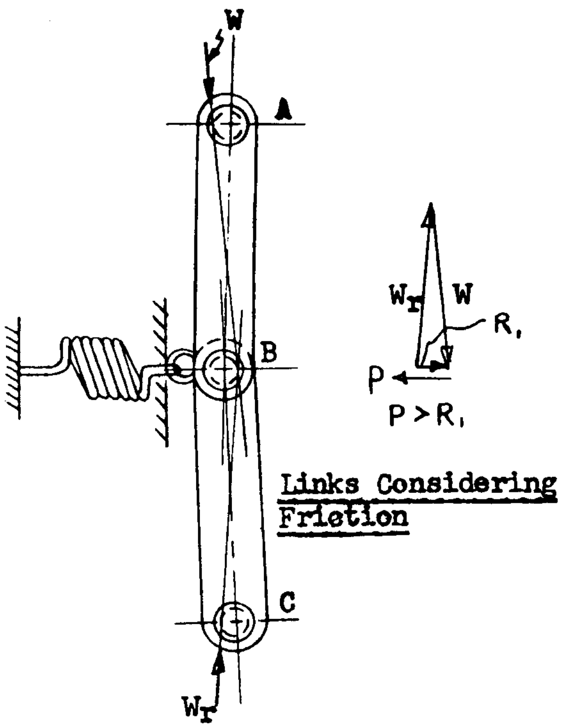
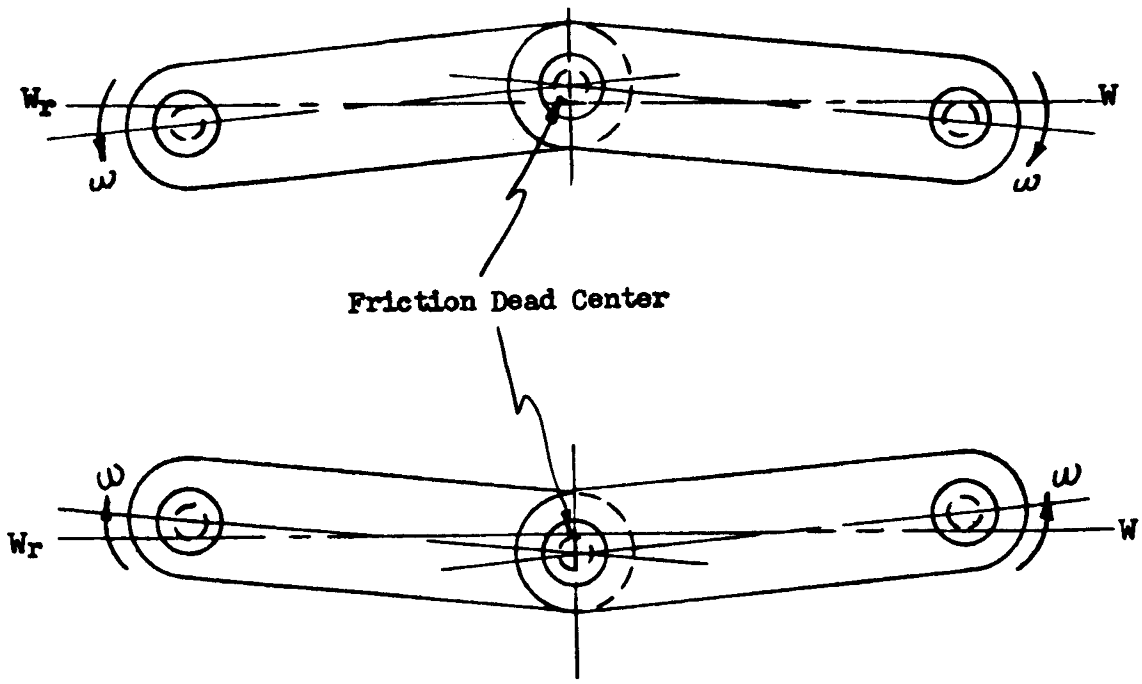


FIGURE 30-29

Figure 30-28 shows the force diagram of two links near dead center with the effects of friction neglected. W and W_R have a resultant R acting to the left and joint B will move to the left until it hits the stop.

Figure 30-29 shows the force diagram of the same links in the same position as Figure 30-28 but with friction effects considered. Due to friction, a spring force $P > R_1$ is required to produce motion of point B to the left.

When the two forces W and W_R, including the effects of friction line up as shown in the sketch below, the linkage is at its "friction dead center."



The linkage has a friction dead center on either side of its geometrical dead center and will remain at balance in any position between these friction dead centers with no restraining force. Consequently, a force is required to move the linkage past its friction dead centers.

The importance of careful design of mechanisms operating near dead center is emphasized by the following example.

An idealized landing gear retraction mechanism is shown in Figure 30-30 with the gear in its full down position and the drag brace links lined up on dead center. The effects of friction have been neglected in calculating the power requirements for moving the links off their dead center position. (This mechanism is not intended to represent a good design and is meant for illustration purposes only.)

Summing moments about the center of joint A (Figure 30-30), the only force the actuator must initially overcome is that of the spring.

$$P_{\text{actuator req.}} = \frac{25(4/12)(12 + 14)}{1.3} = 167 \text{ pounds}$$

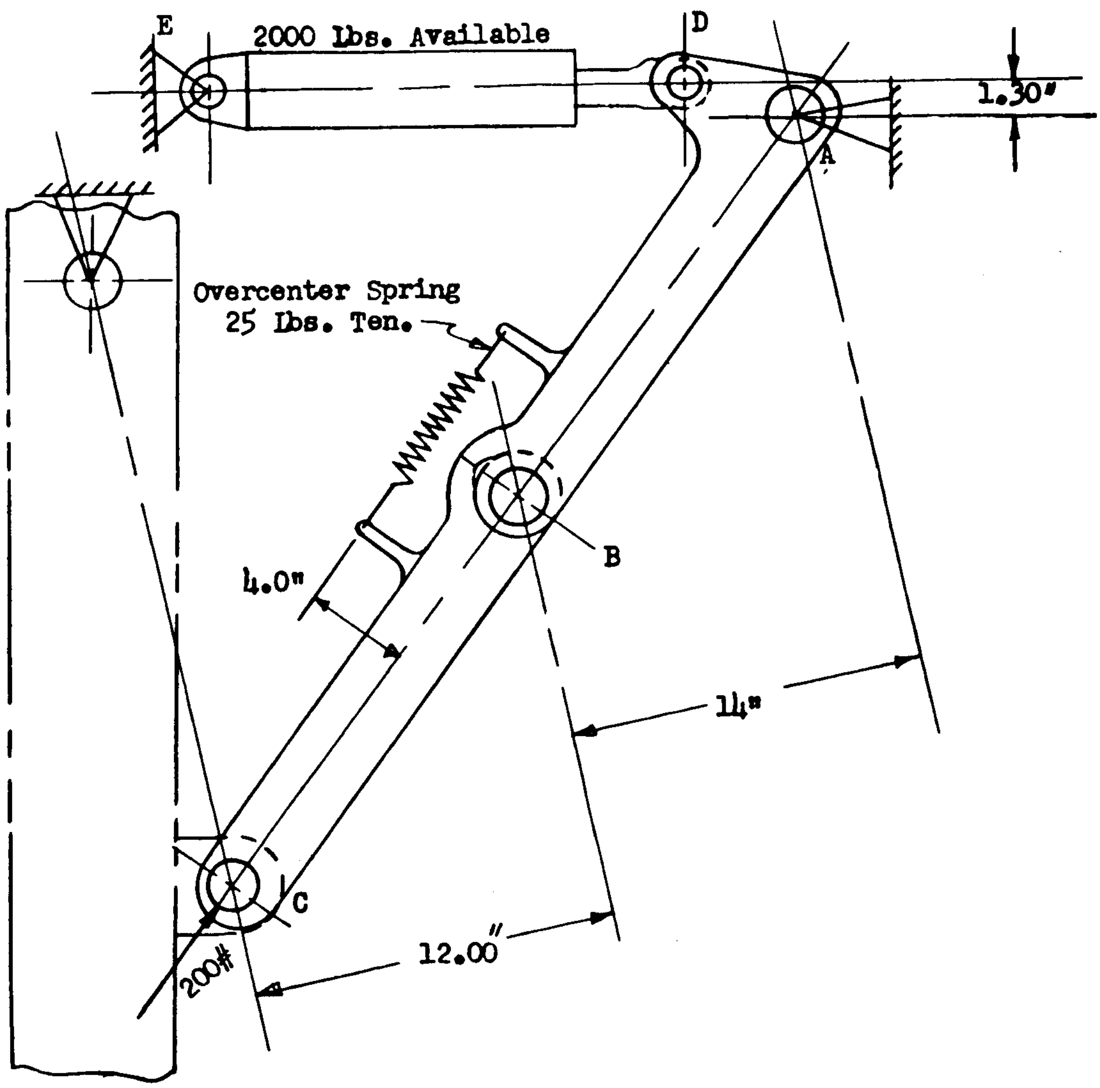


FIGURE 30-30

This is only about 8.4 percent of the available force.

Figure 30-31 shows the same mechanism, but with the effects of friction taken into account.

With friction considered, the requirements are considerably different than those of the frictionless system. Summing moments about the reference point 0 of joint A in Figure 30-31:

$$P_{\text{actuator req.}} = \frac{200(1.0)}{0.13} = 1540 \text{ pounds}$$

Note that this is the actuator force required only to move the links off their dead center position.

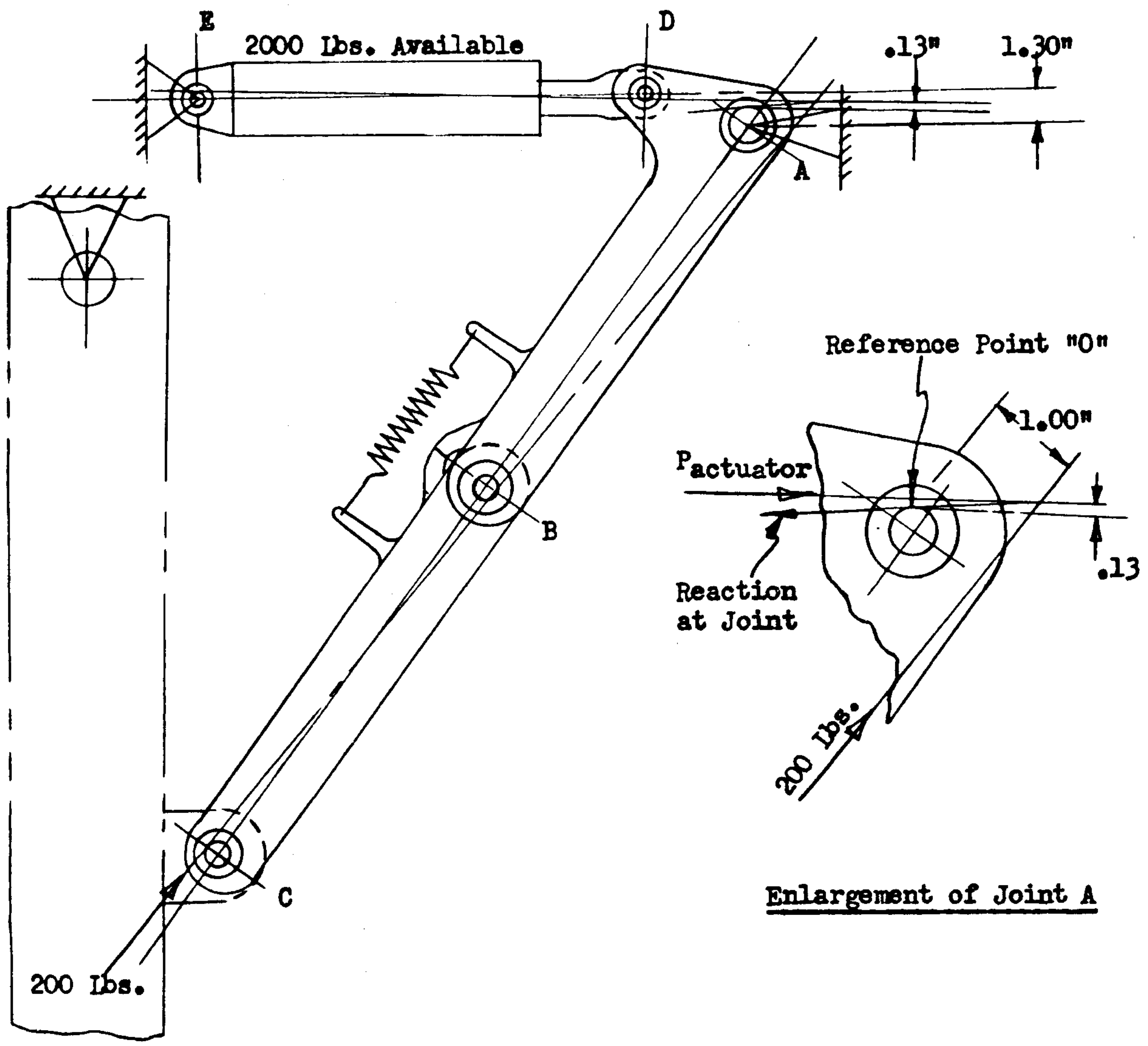


FIGURE 30-31.

The spring has been neglected in this example for simplification. It is noted that with friction effects taken into account the required actuator force has increased about 9 times and is almost equal to the available force of the actuator. A very small decrease in the alignment of the actuator would cause the system to be totally inoperative.

Often, of necessity, mechanisms must be designed with the actuator having very little mechanical advantage at the beginning of its stroke. (This is usually done so as to obtain greater mechanical advantage at other positions of the mechanism.)

Also the frictionless power requirements are generally small at this position and because of this the potentially harmful effects of the low mechanical advantage of the power source may easily be overlooked. As illustrated by the previous example, the effects of friction coupled with a low frictionless mechanical advantage may cause the mechanism to be inoperative.

Often a mechanism is designed so that a portion of its system reverses its direction of motion after some position is reached while the rest of the system continues to move in the direction of initial motion. A clear understanding of the motion of such mechanisms is important when locating the positions of the load lines with friction in the joints. It is easy to make errors in the location of these load lines with the result that friction increases rather than decreases the efficiency of the mechanism. This is obviously impossible.

Figure 30-32 shows two bellcranks, the driver GED and the driven ABC connected by the link CD. Both bellcranks rotate in the same direction at this position of the mechanism. The load lines with friction effects considered are shown for bellcrank ABC.

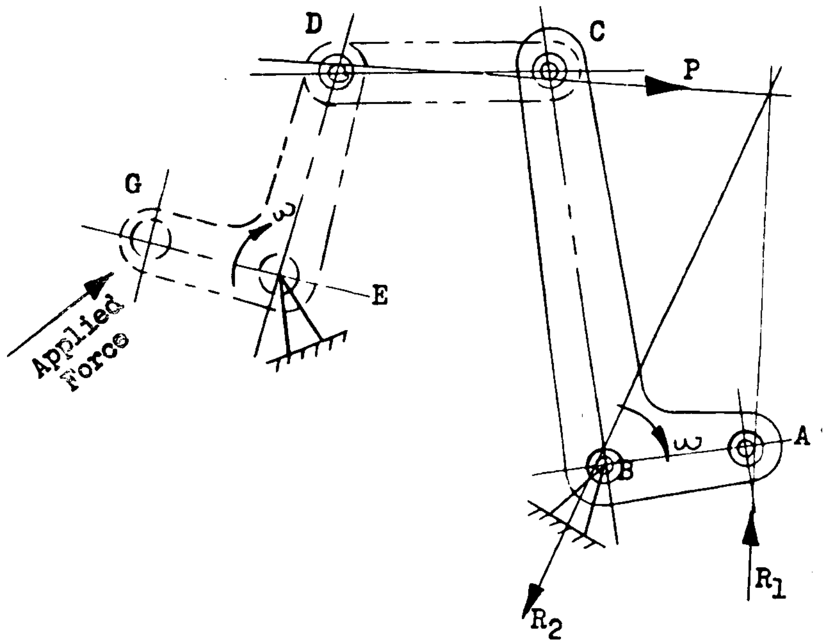


FIGURE 30-32

In Figure 30-33 the same linkage has moved to a different position. The driver bellcrank GED is moving in the same direction as in Figure 30-32, but the driven bellcrank ABC has reversed its direction of rotation. The loads and reactions of ABC have changed directions and the location of the load line for the load P has also changed at this position of the mechanism.

As pointed out in the example, one must have a complete knowledge of the motion of the mechanism before attempting to locate the load lines with friction in the joints.

Since the diameter of the friction circle is proportional to the diameter of the connection about which the joint rotates, it would seem beneficial to the functioning of the mechanism to design joints with connections of small diameter. One must, however, consider the bending deflection of the connecting bolt since this may result in binding of the joint which could cause a greater friction loss than if a rigid bolt of larger diameter were used. This is especially true of single shear joints. It is recommended that single shear applications be avoided whenever possible.

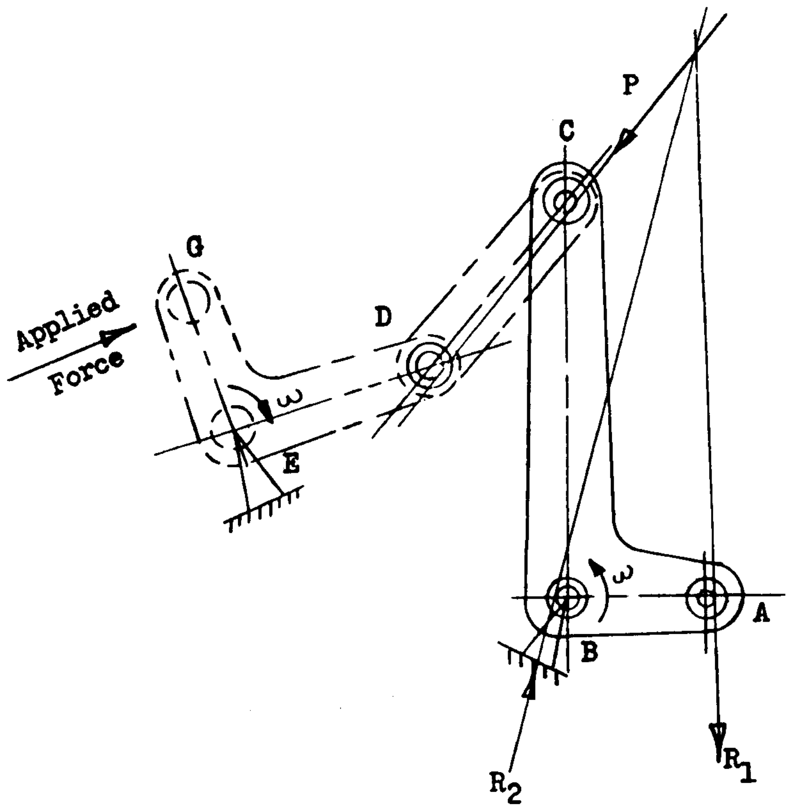
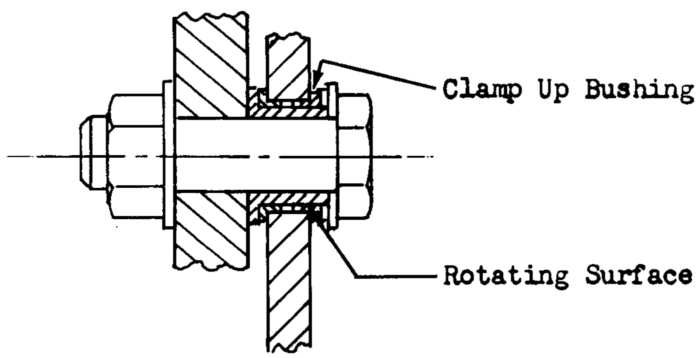


FIGURE 30-33

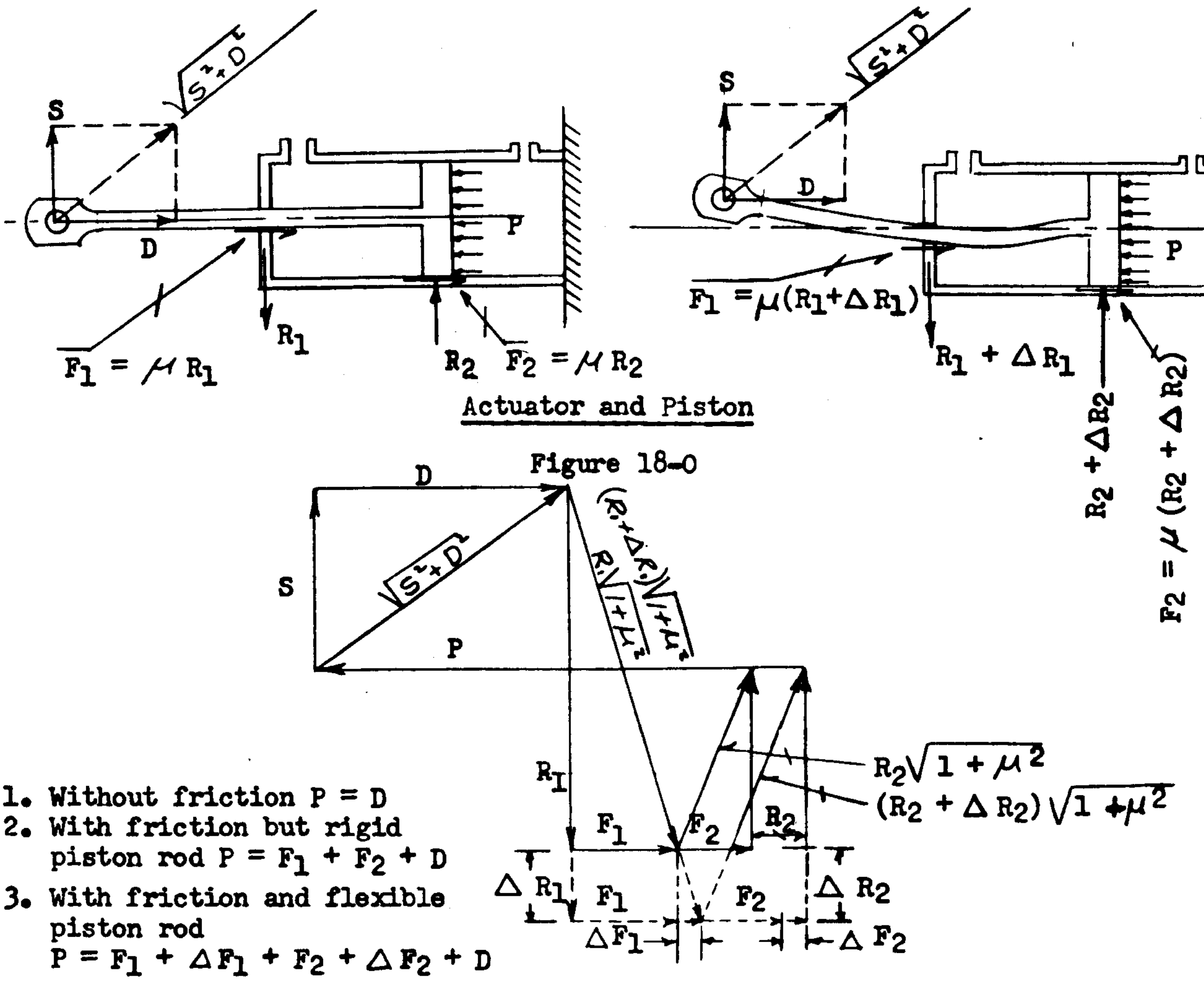
If single shear joints cannot be avoided, the connecting bolt is to be held from rotating by clamping it in one of the links so that all the rotation is done by the other link. A clamp-up bushing should be used in the installation. The sketch below shows the proper installation of a single shear joint.

Single Shear Joint



34.4 THE EFFECTS OF FRICTION IN SLIDING PARTS A friction force will oppose the movement of one surface relative to the surface with which it is in contact. It is exerted tangentially to the surfaces and opposite the direction of motion.

In some cases it is necessary to consider the flexibility effects of mechanism components.



1. Without friction $P = D$
2. With friction but rigid piston rod $P = F_1 + F_2 + D$
3. With friction and flexible piston rod $P = F_1 + \Delta F_1 + F_2 + \Delta F_2 + D$

FIGURE 30-35

It is seen from the example that an increase in the magnitudes of the reactions R_1 and R_2, either due to a further extension of the piston, an increase in the side load L, or a deflection of the piston, or all three, will cause an increase in the friction forces F_1 and F_2. This would require an increase in the actuator force P to balance the piston. It is possible that the force required to overcome the friction may be greater than that required to move the primary load D.

34.5 EXAMPLES

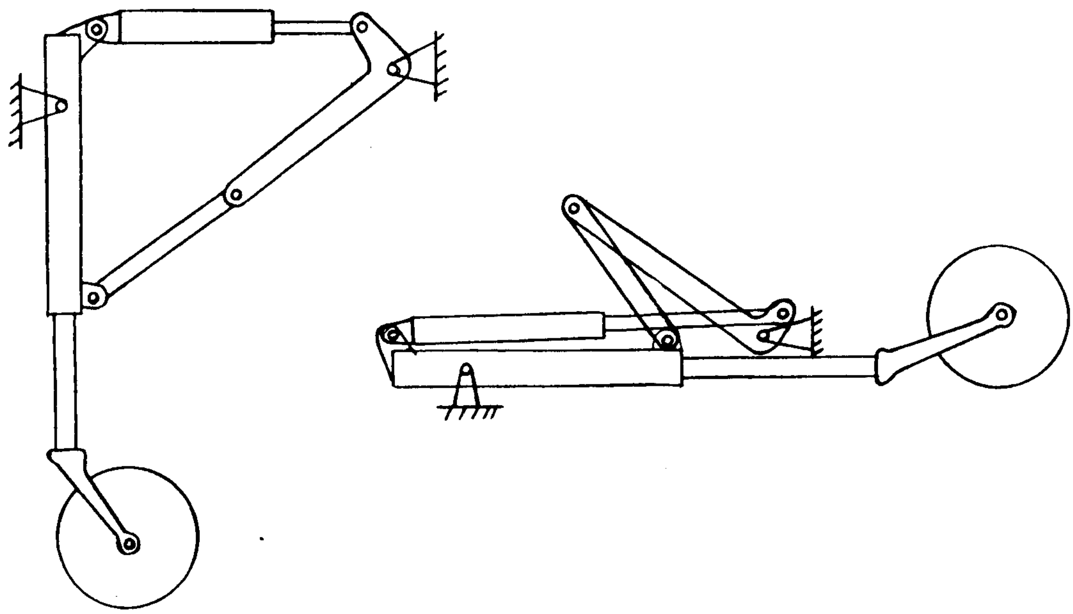
34.51 LANDING GEAR RETRACTION MECHANISM POWERED BY A HYDRAULIC ACTUATOR

A typical nose landing gear is shown in its fully extended and fully retracted positions in Figure 30-36.

In order to retract the gear the actuator must overcome inertia load of the gear and airloads acting on the gear and the wheel well doors.

The superposition of loads is advantageous in the determination of the power requirements of a mechanism such as this, and the following procedure is recommended for obtaining the load stroke envelope for the hydraulic actuator.

a. First a scale layout of the mechanism is required. This layout should be large enough to obtain accurate measurements, and the mechanism should be shown on the layout at a number of positions of retraction sufficient to obtain the actuator load at enough points along its stroke to define accurately the load-stroke curve.



Fully Extended Gear **Fully Retracted Gear**

FIGURE 30-36

b. Now with the geometry of the layout, determine the actuator load at each position of retraction required to balance a unit load applied at the location and in the direction of each external load. (Each load is done separately.) Calculate both the "no-friction" actuator load where the load lines pass through the centers of the joints, and the actuator load with friction in the joints. The load lines for the latter calculations are determined through the application of the method of friction circles as presented in 34.3. The power requirement for each externally applied load may be determined by multiplying the requirement for the unit load by the external load. The total requirement is then obtained by summing the requirements of all individual loads. It should be noted that this procedure includes only friction due to joint rotation.

In the cases involving sliding friction such as in landing gears with shrink links the resulting effects on the power requirements of the mechanism may be determined as follows:

For each position of the mechanism the normal loads on the sliding surfaces and the corresponding friction loads are found.

A combination of external loads resulting in the highest total normal loads at the sliding surfaces and hence in the highest friction loads should be used for the first check.

However, it should be determined whether the above combination of external loads results in the highest required actuator load.

Such is not necessarily the case. In other words, some combination of external loads not causing the highest sliding friction loads could result in the highest power requirements. The correct answer can be found only by actual analysis. It should be pointed out that in the above analysis the proper weight breakdown of the moving parts is important. For example, in the case of a landing gear with a shrink mechanism, the unsprung weight of the gear, rather than the total gear weight should be used in determining the sliding friction.

The application of the above method may cause difficulties in trying to keep the effects of individual applied loads separated as recommended above, and in problems of this type individual judgment must be exercised, before an elaborate analysis is carried out.

c. Sometimes a curve of the efficiency of the actuator vs stroke is desired. The efficiency may be expressed as the ratio of the "no-friction" actuator load to the "friction" actuator load.

$$\eta = \frac{P_{\text{no-friction}}}{P_{\text{friction}}}$$

Efficiency may also be expressed as the ratio of the product of the ratios of the moment arms of the applied forces to the moment arms of the reacting forces with friction in the joints to that without friction in the joints.

$$\eta = \frac{\left(\dfrac{A_{A_F}}{A_{R_F}}\right)_1 \left(\dfrac{A_{A_F}}{A_{R_F}}\right)_2 \cdots\cdots \left(\dfrac{A_{A_F}}{A_{R_F}}\right)_h}{\left(\dfrac{A_A}{A_R}\right)_1 \left(\dfrac{A_A}{A_R}\right)_2 \cdots\cdots \left(\dfrac{A_A}{A_R}\right)_n}$$

where:

η = efficiency of the system

A_{A_F} = Moment arm of the applied force with friction in the joints.

A_{R_F} = Moment arm of the reacting force with friction in the joints.

A_A = Moment arm of the applied force without friction in the joints.

A_R = Moment arm of the reacting force without friction in the joints.

1, 2 . . . . n = Subscripts designating pivot points of bellcranks in the system.

d. Load-stroke curves for the actuator with and without friction may now be constructed as shown in Figure 30-37.

It is again pointed out here that the accuracy of the layout is very important in obtaining good results.

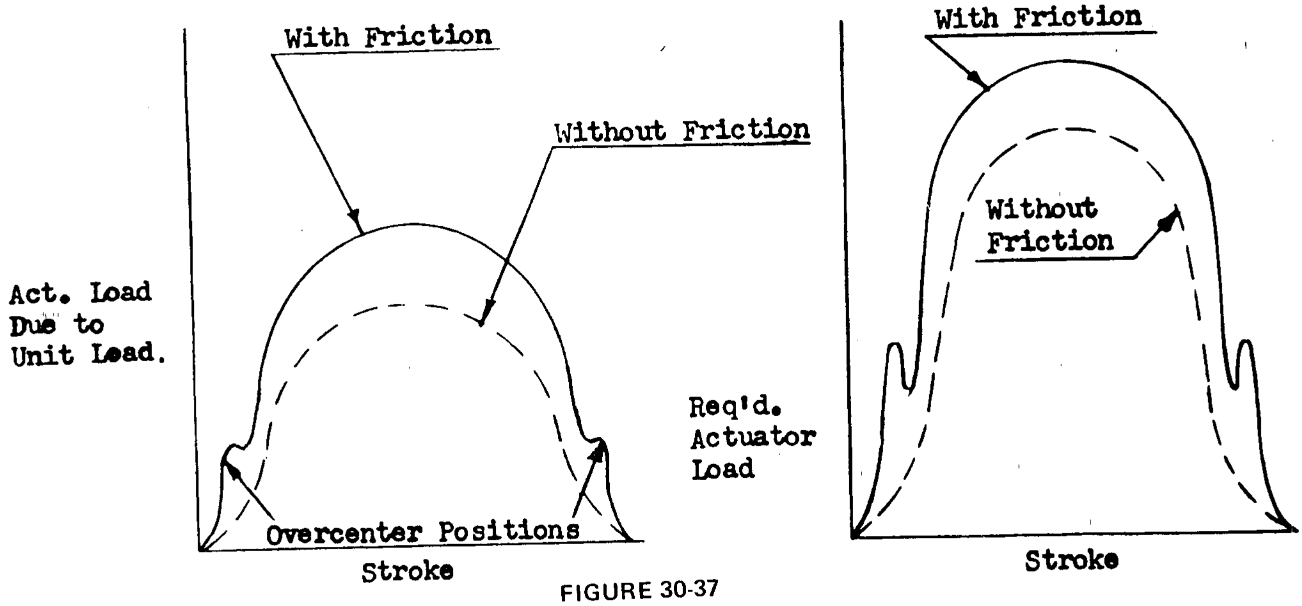


FIGURE 30-37

Load Stroke Curve vs. Unit Load

Load Stroke Curve vs. Required Actuator Load

Either the Load Stroke Curve vs. Unit Load or the Load Stroke Curve vs. Required Actuator Load may be used to calculate the efficiency-stroke curve shown in Figure 30-38.

NOTE: The final load stroke curve should be obtained by superposition of load stroke curves separately calculated for each externally applied load.

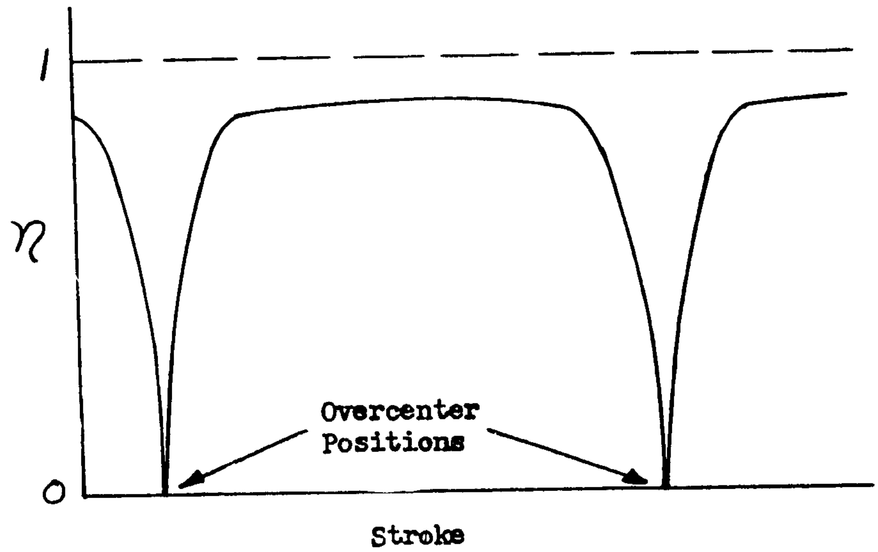


FIGURE 30-38

It is recommended that the data be compiled in tabular form. Good bookkeeping is of primordial importance.

A modification of the method just outlined is to construct the no-friction load-stroke curve and then, with a few significant points, construct an efficiency-stroke curve. With data from these curves the friction load-stroke curve may be constructed. This method is less time consuming than the preceding method, but only at a sacrifice of accuracy. This method is acceptable for use in the preliminary stages of a design, but before any final decisions are made its results should be verified by checks using the first method. Care must be exercised, however, in the establishment of the efficiency-stroke curve when using the simplified method. The considerable loss of efficiency when parts of the mechanism are near dead center has been pointed out in the example of 34.3. In some mechanisms the moment arms of the bellcranks with friction effects considered may be obtained with sufficient accuracy by adding or subtracting the radius of the friction circle from the no-friction moment arms, but in other mechanisms this will result in large unconservative errors. This is illustrated with Figure 30-39, positions 1 and 2.

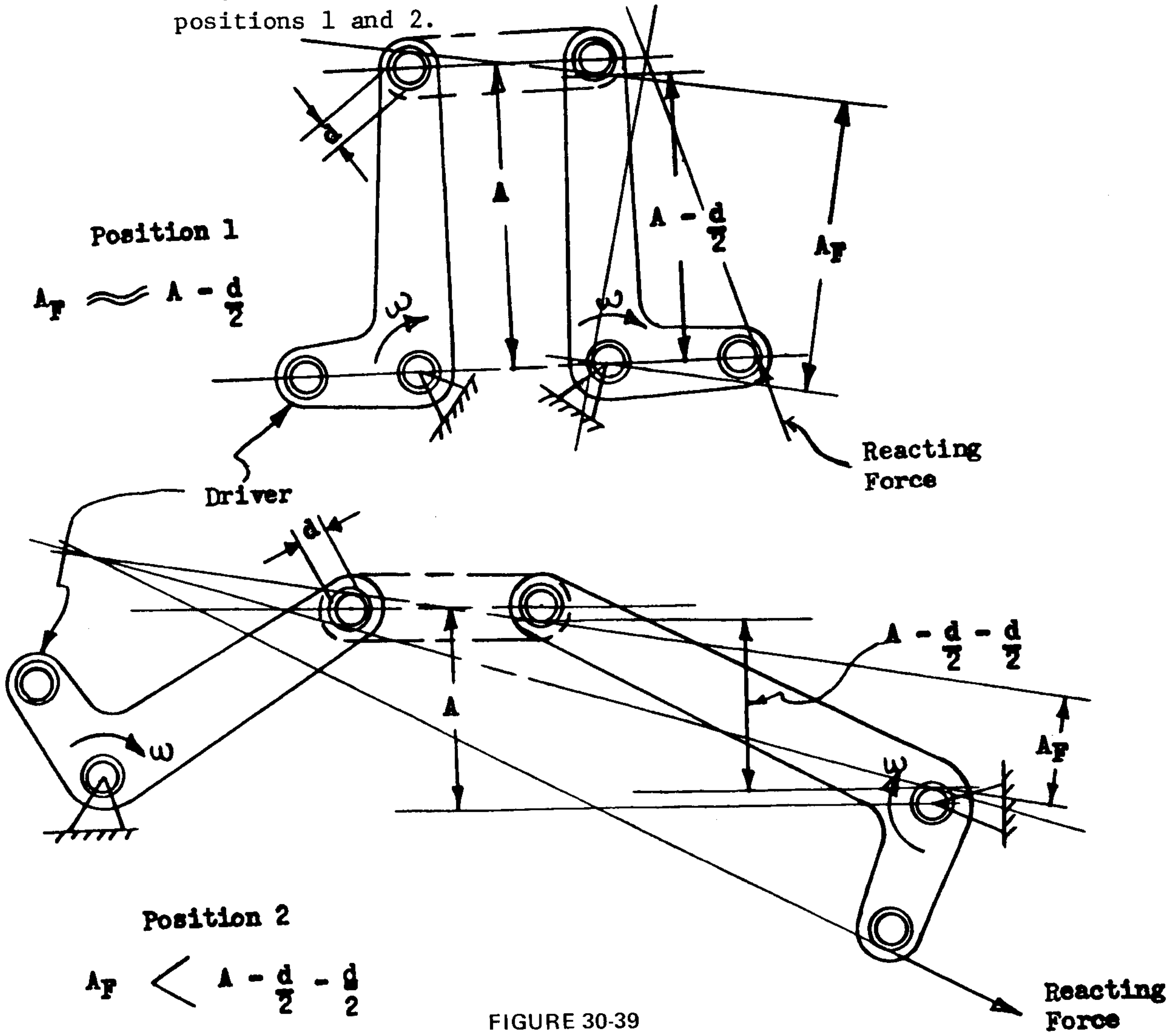


FIGURE 30-39

34.52 <u>THE EFFECTS OF SLIDING FRICTION ON FUNCTIONING OF A LANDING GEAR SHOCK STRUT</u> Friction forces at the bearings of landing gear shock struts opposing the motion of the inner strut during landing may be of consequence and should be considered in the design of such assemblies. These friction forces may alter the load-stroke characteristics of the gear and the resulting thrust loads on the bearings. The lower bearing is especially critical and requires consideration when designing the mounting arrangement.

In extreme cases the friction forces may be of sufficient magnitude to completely prevent the strut from compressing. A graphical representation of this condition is shown in Figure 30-40.

To ensure compression of the strut, the resultant of the vertical and horizontal components of the applied landing gear load must fall inside the "friction triangle" ABC of Figure 30-40.

In addition to loads resulting from the side and drag components applied to the gear, loads caused by the deflections of the struts may also be imposed on the bearings.

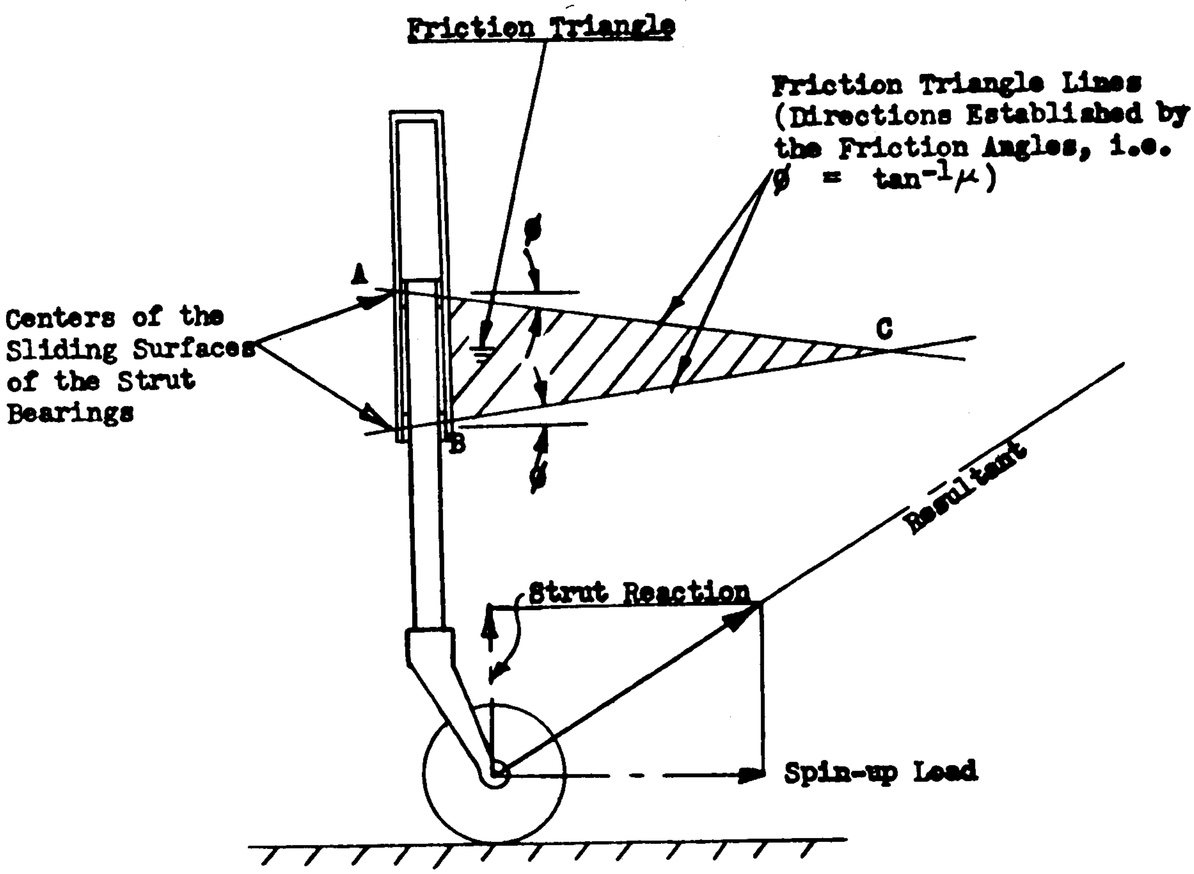


FIGURE 30-40

A discussion of the loads applied to the lower bearing follows. Figure 30-41 shows a bearing that is fully backed up by the wall of the outer cylinder.

When landing loads are applied there is generally a difference in the slopes of the deflection curves of the inner and outer cylinders at the bearings as illustrated in Figure 30-42. A moment at the bearings is required to make the slopes equal. This moment must be transferred from the inner cylinders against the inner and outer walls of the bearing. Friction forces acting normal to the couple loads will also tend to oppose the motion of the strut, at the surface of motion.

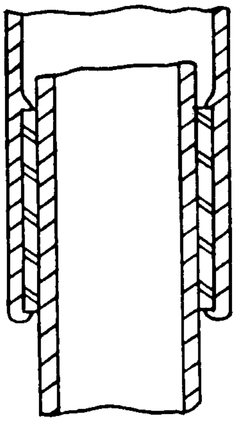


FIGURE 30-41

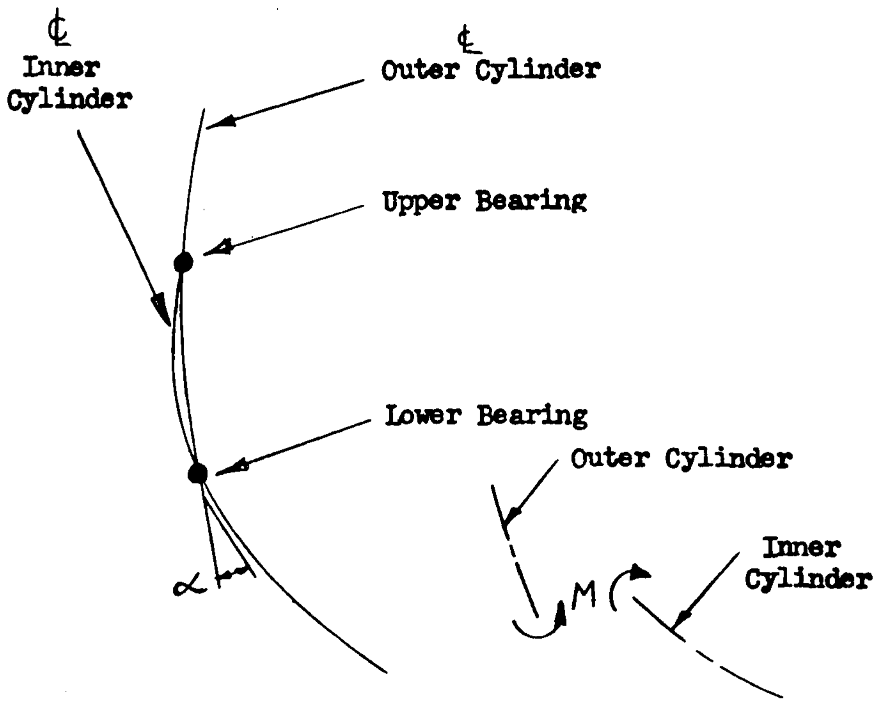


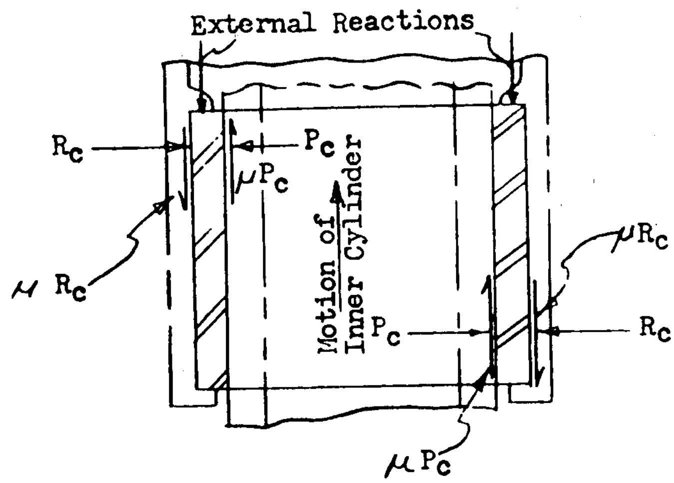
FIGURE 30-42

An approximate method of determining the forces acting on the bearing is as follows:

a. Balance free bodies of the inner and outer struts.

b. Find the slopes of the deflection curves at the lower bearing for inner and outer struts. If these slopes are equal, there will be no binding. If the slopes are not equal determine the required moment at the lower bearing to make the slopes equal. This moment must be transferred through the bearing between the inner and the outer cylinders.

c. A free body of the bearing is shown on the following page.



Free Body of Lower Bearing

The condition just discussed may be relieved by allowing the bearing to pivot in the outer cylinder as the struts deflect.

34.6 CHECK OF ANALYSIS BY ENERGY METHOD A quick and convenient check of the analysis described in Section 34.5 is afforded by the application of the principle of virtual displacements commonly known at MDC as the "Energy Method." Briefly, the principle states that if a frictionless system in equilibrium be given an infinitesimal displacement consistent with the constraints of the system, the total work done by external forces is equal to zero.

To apply the principle:

a. Assume a small displacement of the external force consistent with the geometry.

b. Determine the corresponding displacement of the actuator end.

c. Compute the total work done.

d. Equate total work to zero to find the load in the actuator.

e. Repeat for as many positions as desired to obtain a reliable load stroke curve.

f. Compare with results obtained by method described in Section 34.5.

This check should be done in all cases to ensure against gross errors.

35. MECHANISM OPERATIONAL SUMMARY AND CHECK LIST

Improper functioning of mechanisms has been a major source of trouble in the development of MDC airplanes. The proper design and analysis of mechanisms is a major responsibility which the Strength Department shares with the Design Department.

The factors which must be considered in the design and analysis of mechanisms are numerous and are frequently complex. The preparation of a "Mechanism Operational Summary and Check List" is required for each mechanism. It is intended to assist the analyst in properly considering all pertinent factors which enter into the design of the mechanism. In addition, the "Mechanism Operational Summary and Check List" is intended to supply a concise, convenient summary of the important facts regarding each mechanism for insertion in the "Operating Devices and Mechanisms" report which is prepared for each airplane. An additional function is to supply a convenient basis for review of the mechanism by Strength Department supervision.

Preparation of this summary should begin in the initial layout stage of a mechanism. The Check List items which have not as yet formulated should be noted "Not Yet Resolved". Additional items and detail should be added to this summary as the design progresses. At the time of drawing release, the summary should be checked over in detail to see that each item has been properly considered.

The "Mechanism Operation Summary and Check List" consists of three primary categories, each dealing with a major aspect of the mechanism as follows:

a. Description of Mechanism
b. Operation and Design Criteria
c. Structural Criteria

Each of the three categories has a number of sub-items listed below the above categories. For each category, start with the first sub-item and make a notation or comment for each, even if the notation is "not applicable". Appropriate drawing numbers, layouts, MAC reports, etc., should be referenced.

In addition to the list of sub-items in each category, additional items may be appropriate for particular mechanisms. These should be listed following the basic sub-items and should be treated in the same manner.

35.1 DESCRIPTION OF MECHANISM

35.11 LIST OF REFERENCES Pertinent data such as the Design Data Book drawing number for geometry and the Installation drawing number for component parts; also applicable Test Request numbers.

35.12 SKETCH (Attached to the Check List)

35.13 LIST OF COMPONENTS INVOLVED Primary power source, components driven by the mechanism, primary supporting structure components; discuss the primary parts (failure of which would cause loss of mechanism) and the effects of the mechanism on surrounding structure clearancewise, loadwise and deflectionwise.

35.14 SEQUENCE OF OPERATION Discuss operation of the mechanism in the order in which it functions including, when applicable, its operational sequence with other mechanisms; information presented here should cover such things as relative timing with related mechanisms and the effect of combined mechanism operation on the power source. (hydraulic or pneumatic)

35.15 INSTALLATION AND RIGGING PROCEDURE Discussion of the adjustments made upon installation of the mechanism to achieve proper functioning; this includes travel adjustments such as link and actuator length adjustments and stop adjustments (overcenter positioning); also limit switch adjustments; refer to the Rigging Procedure drawing when applicable.

35.16 PRELOAD INFORMATION Discuss preload requirements such as limits of door gap and surface deflections under flight conditions; discuss provisions to achieve proper preloading of the mechanism and the components operated by the mechanism such as warp provisions in surfaces and/or "built in" foreshortening of links.

35.17 TESTS: PROOF, FUNCTIONAL AND STRENGTH List and discuss the various tests proposed to demonstrate the adequacy of the mechanism installation and also list the tests conducted to determine rigging or preloading data; include the Test Request number and identify the test vehicle (production airplane number, "iron bird," static test airplane) and the relative times at which the tests are scheduled to be run.

35.18 EMERGENCY PROVISIONS AND SAFETY FEATURES Discuss secondary power source such as pneumatic system for emergency operation; note the differences between emergency and normal operation (airplane conditions, speed of operation). List safety features such as overcenter springs, manual release, pressure relief valves, various sensing devices to relieve the loading, pilot warning devices.

35.19 FAILURE ANALYSIS Discuss the effects of power reduction or loss of power upon the mechanism (for example failure of power as the mechanism is operating); also the effects of structural and power failures of the mechanism upon the safety of flight of the airplane as a whole.

35.2 OPERATIONAL AND DESIGN CRITERIA

35.21 OPERATING CONDITIONS List airplane speeds, load factors and attitudes at time of mechanism operation; discuss required surface deflections and times of operation; refer to sources of information for operating conditions (Airplane detail specification, MIL specifications, DAC reports, memos); discuss various requirements when pertinent; attach curves of load information such as hinge moment vs deflection. This data should be listed with each of the following conditions:

a. Nominal Operating Conditions

b. Emergency Operating Conditions

35.22 OPERATING CRITERIA Discuss the effects of the following criteria upon the requirements of the operating conditions.

35.221 FRICTIONAL CRITERIA Use a friction coefficient, $\mu = 0.20$ for plain bearings with lubrication (for ball, or roller bearings $\mu = 0$) when checking normal operation of the mechanism including calculation of time to operate.

35.222 EXTREME OPERATIONAL CRITERIA Variations of mechanism configuration which affect operation such as mislocation of joints due to manufacturing tolerances, misrigging, or maladjustment, poor lubrication (mechanism should be able to function with $\mu = 0.60$ at any one joint with $\mu = 0.20$ at others), temperature effects (expansion and contraction due to temperature changes affect preload, clearances and travel; elastic property changes with temperature); discuss the limit of operation with various combinations of these extreme conditions; state the extent to which these factors have been considered in the design of the mechanism.

35.23 EFFECTS OF DEFLECTIONS Discuss the effects of deflection under mechanism operation loads (mechanism and support structure component deflections affecting power, travel and clearance); also when mechanism is in locked position and surrounding structure deflects under various flight conditions (possibility of unlocking mechanism under adverse conditions); the effects of surface deflection upon the applied loads (including aerodynamic drag); discuss the precautions which have been taken in the design of the mechanism to prevent improper functioning due to deflections.

35.24 POWER SOURCE Describe system including such data as hydraulic and pneumatic line sizes and lengths from the power source to the mechanism actuator, list for components such as restrictors and check valves; discuss various pressure drops such as line losses and valve losses used in calculating pressure-flow relationships; present pressure-flow curves and related data (line factors, temperature considerations, air expansion characteristics, hydraulic fluid characteristics); include data covering the effects of simultaneous operation with other mechanisms upon the power source; refer to applicable drawings and specifications (Lines Installation drawings, pump manufacturer's specifications): For electrically powered systems reference source of speed-low characteristics, discuss clutches, stops and rotational inertia absorbers. This data should be presented with each of the power sources:

a. Primary Power Source (Normal operation)
b. Secondary, or Emergency Power Source

35.25 POWER REQUIREMENTS Include the required load-stroke curves for the various conditions including the effects of the various operating criteria; consider here the effects of assisting loads during operation (affecting actuator pressure, stop loads, speed of operation, sequence of operation with other mechanisms); refer back to other pertinent paragraphs of the Mechanism Operational Summary and Check List.

35.3 STRUCTURAL CRITERIA

35.31 STRENGTH REQUIREMENTS Discuss strength conditions in a manner similar to Paragraph 35.21, Operating Conditions; include applicable data and references; discuss loading conditions in which the design loads are affected by relief valve pressures for hydraulic, or pneumatic powered mechanisms; also loads occurring when full power output is applied with mechanism against its stops.

35.32 FATIGUE CONSIDERATIONS Discuss considerations given to loading spectrums and fatigue life requirements; list applicable detail design considerations such as the moment effects of bearing friction in rod ends during rotation and fatigue considerations given to lug and bolt design.

35.33 DYNAMIC LOADING CONSIDERATIONS Discuss any inertia type loadings such as may occur at the ends of the mechanism travel; discuss methods of reacting such loadings (snubbing type shock absorbers, energy absorbed by structural deflections, resulting stop loads); frequency of application.

35.34 EFFECTS OF STRUCTURAL DEFLECTIONS ON LOADS These effects should be considered from a strength standpoint in much the same manner in which they are considered in Paragraph 35.23 for operation.

35.35 ACTUATOR STRUCTURAL CONSIDERATIONS When establishing the actuator space envelope, consideration should be given early in the mechanism design to such details as adequate lug radii to meet the fatigue requirements, the selection of the proper size bearings to prevent galling, adequate diameter threaded rod ends to prevent combined bending-tension fatigue failure (See Paragraph 35.32), adequate beam-column strength to prevent failure when subjected to the frictional end moments with the column load, adequate travel and adjustment; the actuator required outputs and strength requirements should be presented along with the cycling load-stroke spectrum for the functional qualification test (the cylinder should be cycled in a test setup duplicating its actual motions when installed in the mechanism); these considerations are especially important when the actuator is purchased by Specification Control Drawing (if actuator is a Specification Control Item, identify it here by its Specification Control drawing number); the above data should be tabulated and discussed here for reference.

35.36 BEARINGS Discuss the types of bearings used, including their materials and configurations; the operating and static stresses to which they are subjected; their lubrication provisions; list of references for the selection of the bearings.

36. MECHANICAL AND PHYSICAL PROPERTIES

This section of the hydraulic manual is intended to supplement MIL-HDBK-5A data on mechanical properties of materials at both room and elevated temperature. All metallic materials customarily used for airframe structural elements are listed in the current issue of MIL-HDBK-5A. As mechanical properties of newer alloys are evaluated and approved for design use by the FAA, the data will be included in this section.

36.1 MATERIAL SPECIFICATIONS Aircraft structural materials are usually defined by a Federal or Military specification. Where there is no such specification, or when it is desired to specify requirements additional to or more stringent then such specifications, a DAC Material Specification (DMS) is prepared for procurement of the material and is referred to in the material call-outs on the drawings.

Since DMS materials often are in limited supply they should be required only where the peculiar properties they specify are actually needed for design requirements. Examples of this are: 4340 steel DMS 1555, and 17-4PH Cres bar DMS 1764 in which the DMS specifies minimum mechanical properties and ductility in the transverse grain direction not specified by the Military specifications.

36.2 LIMITATION ON USE OF ALLOWABLE STRENGTHS HIGHER THAN MINIMUM GUARANTEED VALUES FAR 25.615 specifies that guaranteed minimum design mechanical properties ("A" values) must be used for analysis of single members of assemblies, the failure of which would result in loss of the structural integrity of the component. The "90 percent probability ("B" values)" may be used for analysis of redundant structures in which loads are safely distributed to other load carrying members after the failure of an individual element.

DAC policy permits used of "B" values for both the ultimate strength analysis and the fail safe analysis for those assemblies which meet the fail safe strength criteria of FAR 25.571(c).

36.3 FRACTURE TOUGHNESS AND STRESS CORROSION RESISTANCE CONSIDERATIONS
Fracture toughness refers to the ability of the material to resist the enlarging of existing cracks under load. This characteristic is highly desirable for important structural members which may be expected to encounter crack initiation by fatigue or as a result of impact damage while under flight loads. Normally the higher the ductility of an alloy, the greater is its fracture toughness.

Closely related and often confused with fracture toughness is stress corrosion resistance. Stress corrosion failure is a phenomenon occurring in susceptible materials under sustained stress (well below yield stress) in the presence of a corrosive environment. The level of the sustained stress required for stress corrosion to take place is called the threshold stress. In many important structural materials the threshold stress in the transverse grain direction is substantially lower than in the longitudinal direction.

Three alternative approaches exist for stress corrosion prevention: 1) Keep sustained stresses below the threshold stress by careful detail design and close manufacturing quality control; 2) choose materials which have high threshold stress characteristics; 3) protect the structure from corrosive environment.

Certain characteristics of commonly used alloys are listed here for reference. The characterization of other alloys in MIL-HDBK-5A is highly useful.

2024-T3	- Excellent fracture toughness strength. The higher strength -T81 and -T86 heat treats result in lesser crack resistance.
7075-T73	- Excellent stress corrosion resistance. Has up to 15 percent lower mechanical properties than -T6 heat treat.
7075-T6 or -T651	- Has highest strength. However, the stress corrosion threshold stress value in the short transverse grain direction is only 7,000 psi.
17-7PH Steel	- Has excellent stress corrosion resistance.
17-4PH Steel Bar	- Has excellent stress corrosion resistance. Specify DMS 1764 to obtain guaranteed transverse grain minimum mechanical properties and ductility.
4340 Steel	- For adequate stress corrosion resistance the heat treat must be kept below 220,000 psi ultimate tensile stress. If heat treated above 260,000 psi the steel surface must be protected from the atmosphere. Specify DMS 1555 to obtain adequate ductility and guaranteed mechanical properties in the transverse grain direction.
6AL-4V Titanium	- Has good fracture toughness. This alloy is susceptible to stress corrosion in contact with dry salt above 600°F. In contact with red fuming nitric acid and a small amount of water, less than 1.5 percent, the reaction may be violent.
6AL-6V-2SN Titanium	- High strength alloy with good ductility.

The following alloy is being currently evaluated. Preliminary indications are as follows;

7080-T7 - It appears to be an excellent forging alloy. Hot water quenching after the solution heat treatment results in minimum quenching stresses and minimum machining warping. This alloy appears especially desirable for forgings with equivalent thicknesses in the range of 3 to 6 inches. It has excellent stress corrosion resistance. There are preliminary indications of notch fatigue sensitivity.

36.4 ELEVATED TEMPERATURE PROPERTIES OF METALS The metals used for construction of aircraft suffer a considerable loss of strength at elevated temperatures. For the analysis of all structures which are normally at temperatures above room temperature while subjected to design loads, the strength allowables must be corrected for such temperature. MIL-HDBK-5A presents the curves of correction factors by which room temperature allowables are to be multiplied. These data are provided for all alloys commonly used in the airframe. For information on newer alloys contact the Douglas Process Engineering metallurgists.

In addition to the changes in properties at temperature listed above, exposure to temperatures above certain levels results in changes in crystalline structure. This occurs whether or not the part is subjected to load at the time the elevated temperature is endured. In certain alloys the changes in crystalline structure are accelerated by stressing. Alloy steels should not be subjected to temperatures in excess of those listed in Table 2.3.0.1(b) of MIL-HDBK-5A for periods greater than 1 hour per inch of thickness.

Aluminum alloys are partially annealed by exposures above approximately 250°F. Since this annealing process occurs over a long period of time, the correction curves for these alloys normally are given as a family of curves with 1/2-, 10-, 100-, 1000-, and 10,000-hour exposure periods. Additional curves are given to show the reduction of strength at room temperature as the result of prior elevated temperature exposure. During this partial annealing process, the alloying elements congregate at grain boundaries and also diffuse through cladding. As a result, significant changes in susceptibility to corrosion can occur. For important alloys, curves have been developed to evaluate complex time temperature exposure histories. Paragraph 3.2.7.11 of MIL-HDBK-5A illustrates the use of these curves. See Northrop Report WADC TR 56-585, Part II, "Effects of Temperature-Time-Stress Histories on the Mechanical Properties of Aircraft Structural Metallic Materials," for the effects of stress levels on the annealing rate.

No steel part which is subjected to temperatures above 550°F may be cadmium plated. The cadminum diffuses into and between the grain boundaries of the steel. The result is a brittle surface with poor fatigue and stress corrosion cracking properties. Since this phenomenon occurs in a very short period of exposure, extreme care should be used in salvaging cad plated steel parts after fires or local overheating for even very short periods of time.

Titanium alloys experience phase changes between 550°F and 1000°F. The nature and temperature for these changes varies with the alloy; therefore, careful heed should be given the discussion in MIL-HDBK-5A, entitled "Environmental Considerations," for each alloy.

For higher temperature applications, discuss appropriate materials with the Douglas Process Engineering matallurgists.

37. MISCELLANEOUS DATA

37.1 INTRODUCTORY REMARKS This section provides ready access to information quite often required by the designer. The information contained herein does not readily conform to that in other parts of the manual. For this reason, it is felt that a section entitled "Miscellaneous Data" is necessary. A feature of this section that can prove to be of considerable value is entitled "Data References" (Subsection 37.2). Its purpose is to maintain a reference for locating data which are not included in this manual.

Data can be quickly added here without interfering with the sequential arrangement and page numbering of other sections, thereby becoming usable much sooner. It is intended that this part of the manual contain items that fall into the category of general data rather than methods of analysis.

37.2 REFERENCES

1. Anon: "Process Standards," Douglas Aircraft Division.

2. Anon: "Stress Corrosion," DAC Structural Mechanics Section Analysis Guideline No. 1966-8, December 20.

3. Simpson, R. A., and Stone, M.: "Permissible Wall Thickness for Interference Fit Bushings to Limit Stress Corrosion," DAC Memo C1-250-SM-160, 1 July 1963.

4. Anon: "Standards Manual," Douglas Aircraft Division (1 September, 1967 Rev.).

5. Anon: "Controls Manual," Douglas Aircraft Division (1 August, 1967 Rev.).

6. Anon: "Drafting Manual," Douglas Aircraft Division (1 August, 1966 Rev.).

7. Anon: "Airworthiness Standards, Transport Category Airplanes," Federal Aviation Regulations, Part 25, Federal Aviation Agency, 3 November 1964.

8. Anon: "Airplane Strength and Rigidity - Miscellaneous Loads," MIL-A-8865, Military Specification, 18 May 1960.

9. Anon: "Metallic Materials and Elements for Aerospace Vehicle Structures," MIL-HDBK-5A, Department of Defense, 8 February 1966.

10. Grimes, R. H.: "Spherical Staking Development - Ball Bearings, Model - General," SM 13791, DACO Report, 23 September 1950.

11. Anon: "Castings, Classification and Inspection of," MIL-C-6021F, Military Specification, 4 November 1964.

12. Dager, W. E.: "Casting Classification and Inspection Standards, Model - General," LB 31215, DACO Report, Revised 15 August 1963.

13. Anon: "Airplane Strength and Rigidity, General Inspection for," MIL-A-8860, Military Specification, 18 May 1960.

14. Anon: "Material Specifications," Douglas Aircraft Division

15. Anon: "Airplane Airworthiness, Transport Categories," CAM 4b, Federal Aviation Agency, September 1962.

16. Anon: "Aluminum Alloy Castings, High Strength," MIL-A-21180C - Amendment 1, Military Specification, 26 February 1965.

37.3 FORGINGS Section 30 of the drafting manual can be used for general information pertaining to forgings. The following data are presented to aid the strength analyst in analyzing forgings. Additional data will be presented in future revisions of this manual.

The design of die forgings dictates the direction of grain flow in the forged part. The designer strives to adjust the design of the raw forging so that inherent forging characteristics are used to the best advantage. The Loads/Stress Group may be consulted in regard to the design of forgings, especially in the case of expensive parts. The designer indicates the desired grain direction on all forging drawings. The parting plane must also be shown and noted. Inspection of forged parts is controlled by notes on the drawings.

Because of the time required to manufacture dies for die-forged parts, it may be necessary to use substitute parts on the earlier production aircraft. It should be noted that the final configuration of the die forging is obtained from a series of dies of varying configurations. Bar stock is used for substitute parts, where feasible; otherwise, hand forgings are used. For example, bar stock would be used in cases where the amount of machining required would not result in excessive cost. When analyzing a substitute part, the designer should be aware of the fact that substitute parts have different material properties than the die forgings.

Hand forgings are made by "blacksmithing" bars or billets to approximate size and shape with flat dies and then machining to the final configuration. Section 30 of the drafting manual enumerates the advantages of hand forgings, principally from the manufacturing standpoint. The allowable stresses for hand-forged aluminum stock are lower than those for die forgings and decrease with an increase in size of hand-forged billets. MIL-HDBK-5A gives allowables for hand-forged billets of varying sizes. Both 7075 and 2014 hand forgings have very low cross-grain elongation. Consequently, in analyzing parts which are highly stressed in the transverse direction, benefical yielding effects (bending moduli) should be carefully evaluated. Die forgings have an advantage over hand forgings in that grain flow contours follow the natural outlines of the part. It is advantageous to have the local grain flow along the principal load lines.

Material properties of die forgings are vitally affected in the vicinity of the parting plane. Reductions in tensile stress allowables should be used when analyzing parts with tensile loads acting across the parting plane. Aluminum and magnesium die forgings are frequently subject to unhealed porosity in this location. Steel parts are also subject to reduced tension allowables across the parting plane. The actual reductions to be used have not been determined for the Model DC-10 aircraft as of this date. The strength analyst should refer to his leadman for the design policy to be used. Page 14a of Douglas Report No. LB 31430 gives factors recommended previously for the analysis of forged lugs for DC-9 landing gear parts.

The stress man should watch for situations where a sustained stress exists across the parting plane. When this occurs, parts may be susceptible to stress corrosion failures. See 37.2, References 2 and 3 for stress corrosion data. Parts with interference fit bushings are subjected to a constant stress. Section 33.9 discusses stresses caused by press fit bushings.

During the forging process, steel parts accumulate a layer of decarburized material (decarb) on exposed surfaces. This material is neglected when making a strength analysis. The thickness of the decarb layer on a raw forging depends upon the number of heats required during the forging process. Heat treating of parts also results in a layer of decarb. The thickness of the layer is negligible if the part is copper plated before being heat treated.

38. FATIGUE STRENGTHS OF AIRCRAFT MATERIALS

Fatigue failures come from imperfect features in detail design, improper installation, unintentional injury, errors of the machine shop and lack of adequate inspection. Any change in cross section will cause a local increase in stress, the intensity of which depends on how abruptly the cross section changes. If the local stress is above the endurance limit, failure will occur if the load is sufficiently repeated. The following characteristics of manufactured parts contribute to early fatigue failure and should be avoided or adequate allowance in the design be made to keep the stress below the critical point:

a. Surface finish
b. Fillet radii
c. Square and V-notches, threads
d. Keyways and splines
e. Collars, clamping and press-fits
f. Welds
g. Riveted joints
h. Oil holes
i. Roughening due to corrosion
j. Inclusions
k. Grain direction and grain size
l. Internal stress
m. Hydrogen embrittlement
n. Decarburization.

The endurance limit is the highest stress which, regardless of the number of times it is repeated, will not cause failure. The theoretical stress-concentration factor K_t is the ratio of actual maximum stress to the nominal stress. The nominal stress is usually based on the net area. The actual maximum stress is obtained by the theories of elasticity and so apply only when the maximum stress is within the elastic limit of the material. Any plastic yielding that occurs as a result of over-stressing greatly relieves stress concentrations, Reference 2.

38.1 STRESS CONCENTRATION FACTORS In determining the stress concentration factor to be used in a specific design, several characteristics must be considered. Susceptibility to stress concentrations varies widely with materials. The stress concentration factor varies with size and approaches the theoretical stress concentration factors as the size increases. Care must be exercised not to apply factors determined experimentally on small specimens to large parts. This would give an unsafe design. If theoretical stress concentration factors (K_t) are used in design, the design will be conservative for normal sized parts. See References 2 and 4.

The severity of the effect of stress raisers may be decreased by putting in additional discontinuities such as several notches adjacent to the original notch or relieving grooves at radial hole edges. The designer must allow for the fact that test specimens are invariably smoothly radiused and polished whereas the part that is made from his drawings will be rougher and have local nicks and gouges.

See references at end of this section for specific information on stress concentration factors.

38.2 REQUIRED LIFE In the normal airplane, the designer is not usually required to design for an infinite life but uses a cycle life based on a reasonable airplane service. In fact, it is not desirable to design for greater than the required life because of the extra weight which would result. 4000 total flight hours are considered reasonable for the fatigue design of fighter, attack, and similar aircraft types. The magnitude and frequency of loads imposed during the life of the airplane can best be determined by a statistical analysis of flight and landing loads encountered on similar airplanes in service. If the data are not available to determine loading versus frequency, an estimate may be made of the frequency of loading per flight and the desired service life computed. Parts such as hydraulic units have the required life cycles established by Government specification.

38.3 AXIAL LOAD FATIGUE BEHAVIOR The following S-N curves are presented to serve as a guide in the design of parts for fatigue strength. The actual test points have been faired and the lines represent probable behavior of the materials. Scatter has not been evaluated and therefore these diagrams should not be used for safe design values; see Reference 3. K_t is the theoretical stress concentration factor calculated for the specimens as shown in Figure 30-43. It should be noted that no advantage is realized in using high heat treat steels, if long fatigue life is required and stress raisers are present.

FATIGUE LIFE RESULTING FROM MAXIMUM STRESS EQUAL TO TWO-THIRDS ULTIMATE TENSILE STRESS FROM NACA TN 3866

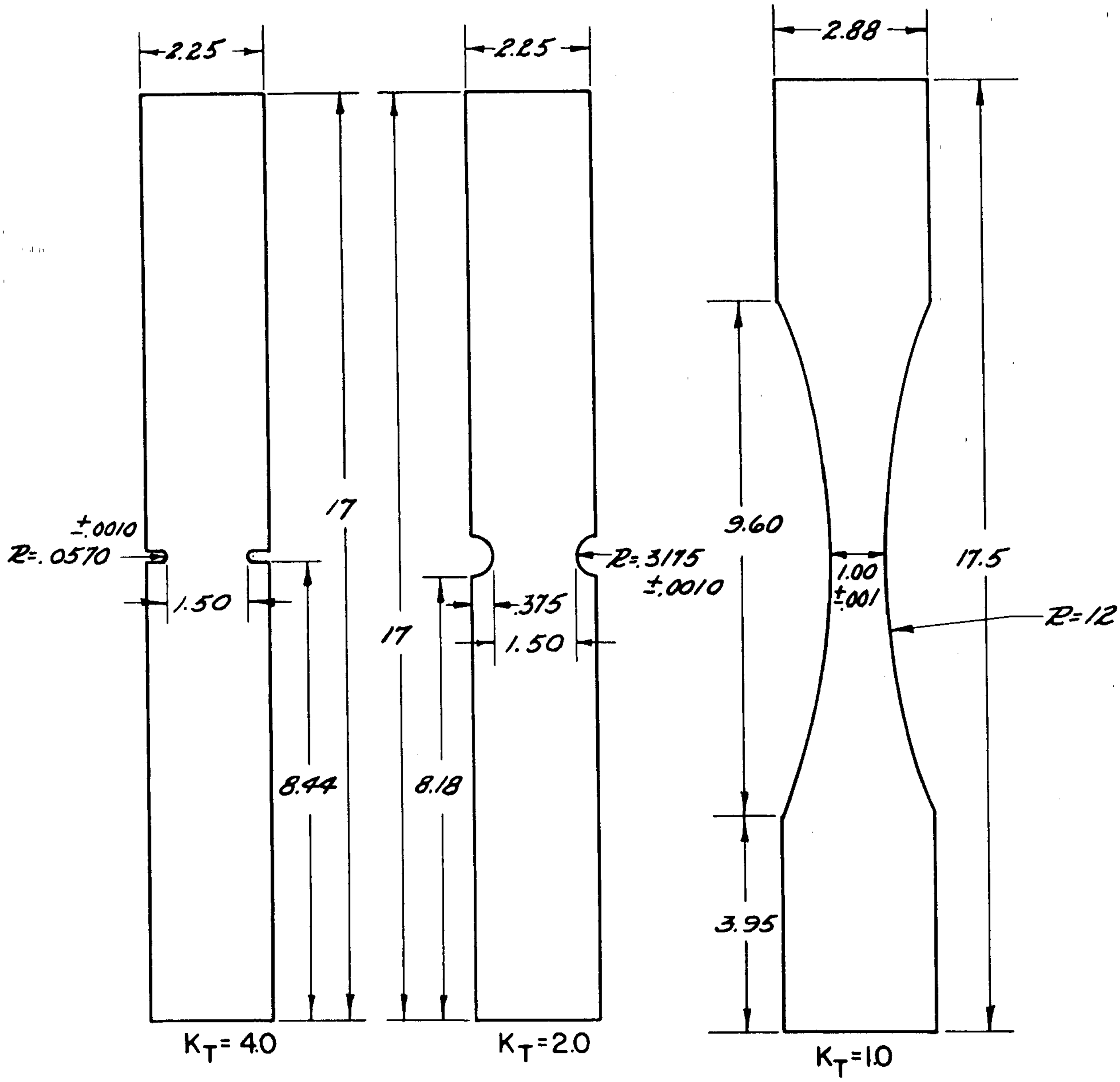
Material	Mean Nominal Stress (ksi)	Fatigue Life, N. Cycles, for $K_T = 1.0$	$K_T = 2.0$	$K_T = 4.0$
2024-T3 Aluminum Alloy	0 20	9,000 -	200 3,000	22 200
7075-T6 Aluminum Alloy	0 20	3,500 -	200 1,400	22 130
Normalized SAE 4130 Steel	0 20	3,900 -	610 3,200	130 420
Hardened SAE 4130 Steel	0 50	80,000 105,000	300 1,900	80 500

"Maximum Stress" is the maximum tensile stress applied in a particular endurance test.

"Minimum Stress" is the minimum stress applied, considering tension to be positive, and compression negative.

"Mean Nominal Stress" is that stress half-way between the "Maximum Stress" and the "Minimum Stress."

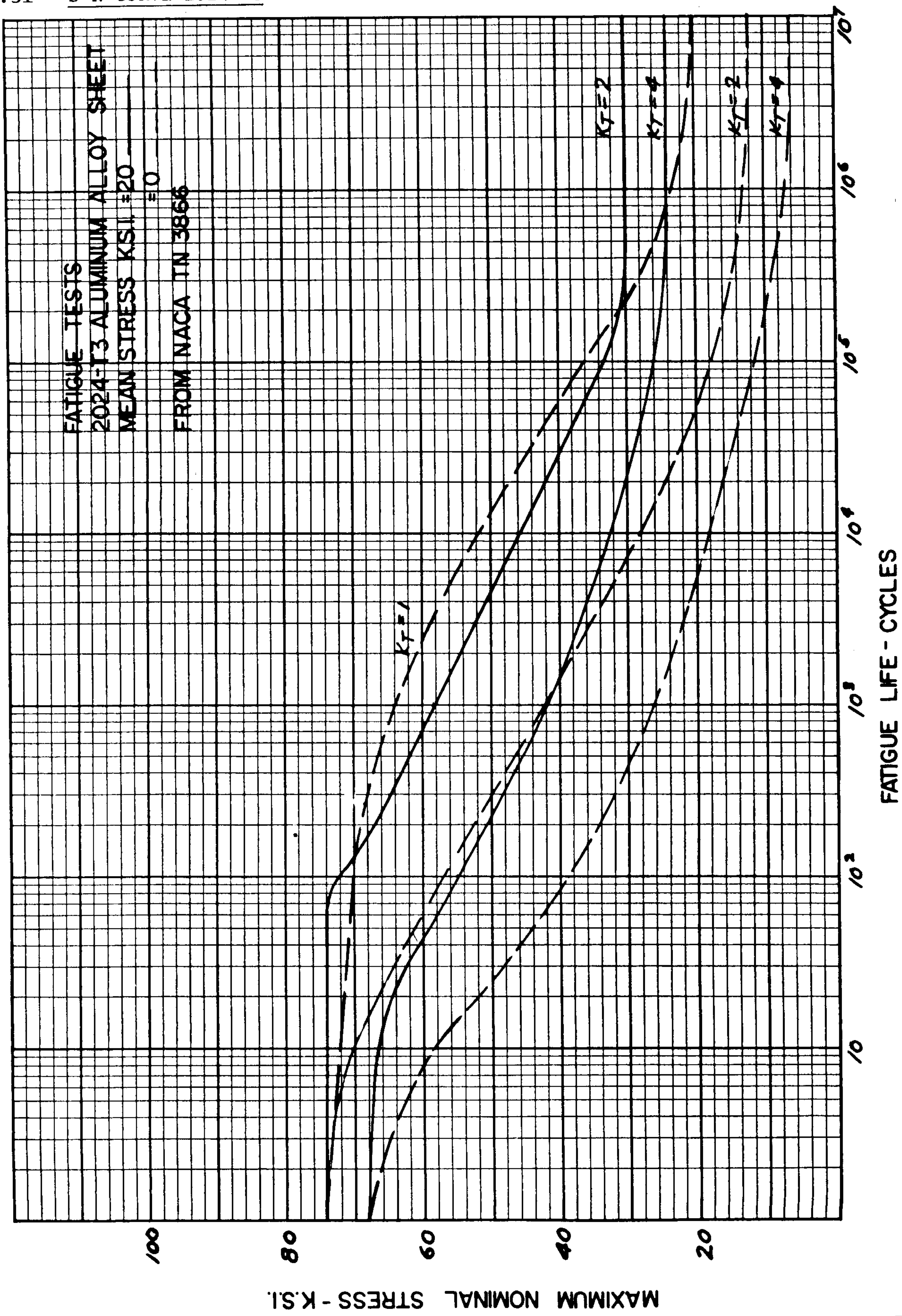
Thus a specimen subjected to 50 ksi maximum stress and 0 ksi mean nominal stress would be cycled through a range between 50 ksi tension and 50 ksi compression, which would be more severe than for a specimen under 50 ksi maximum stress and 20 ksi mean nominal stress (a range of 50 ksi tension to 10 ksi compression).



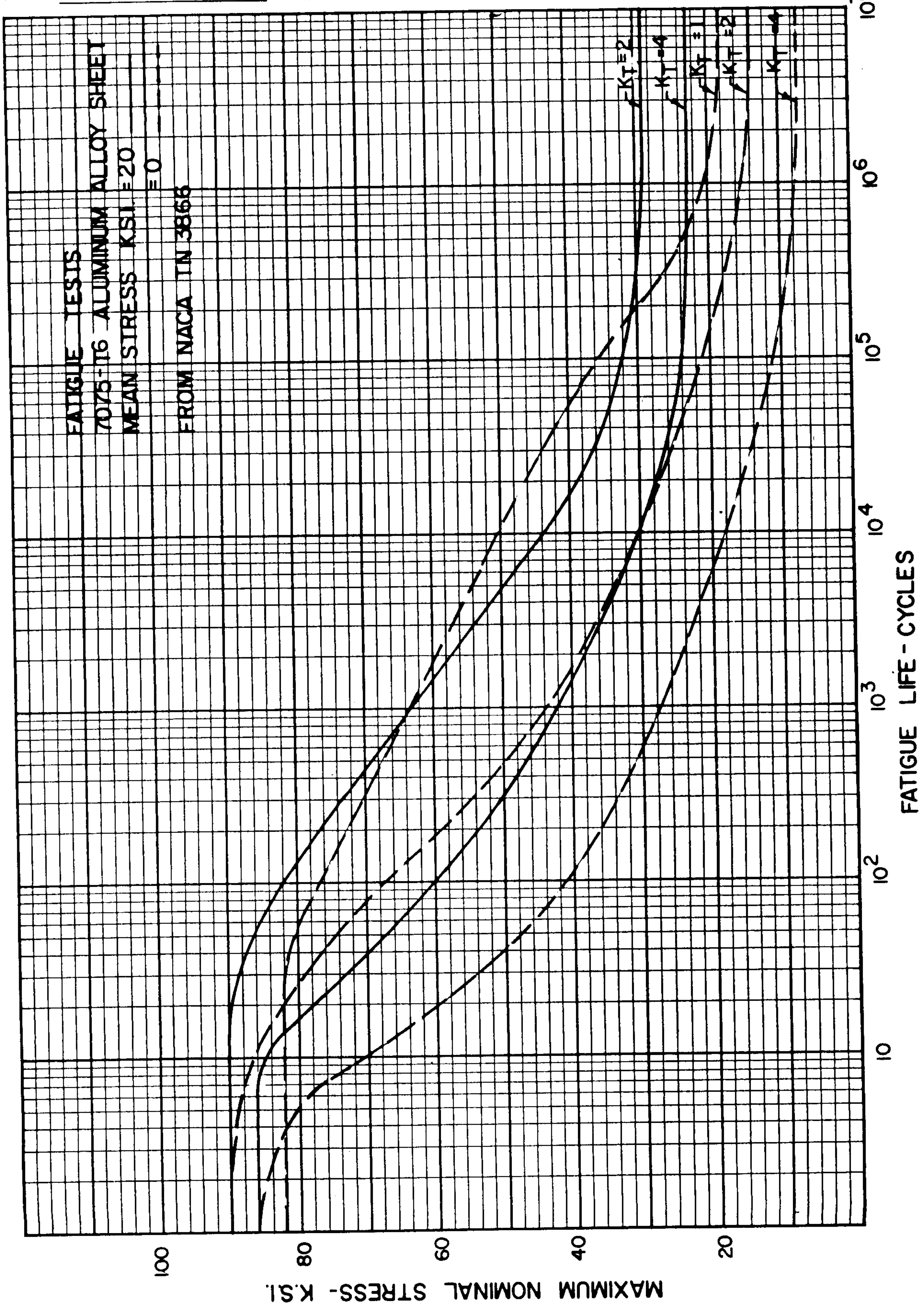
CONFIGURATION OF TEST SPECIMENS
ALUMINUM -.090 THICK
STEEL -.075 THICK

FIGURE 30-43

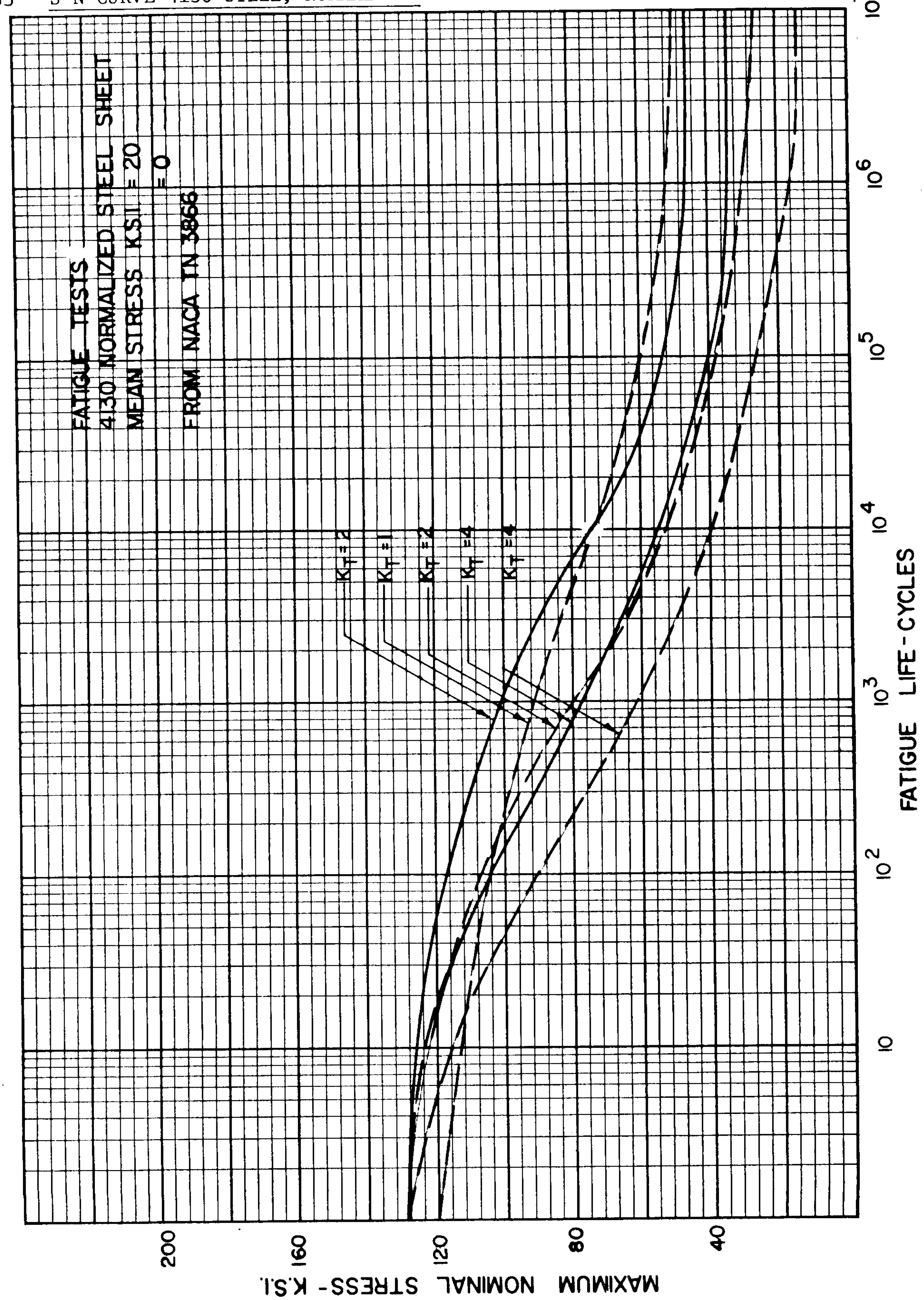
38.31 S-N CURVE 2024-T3



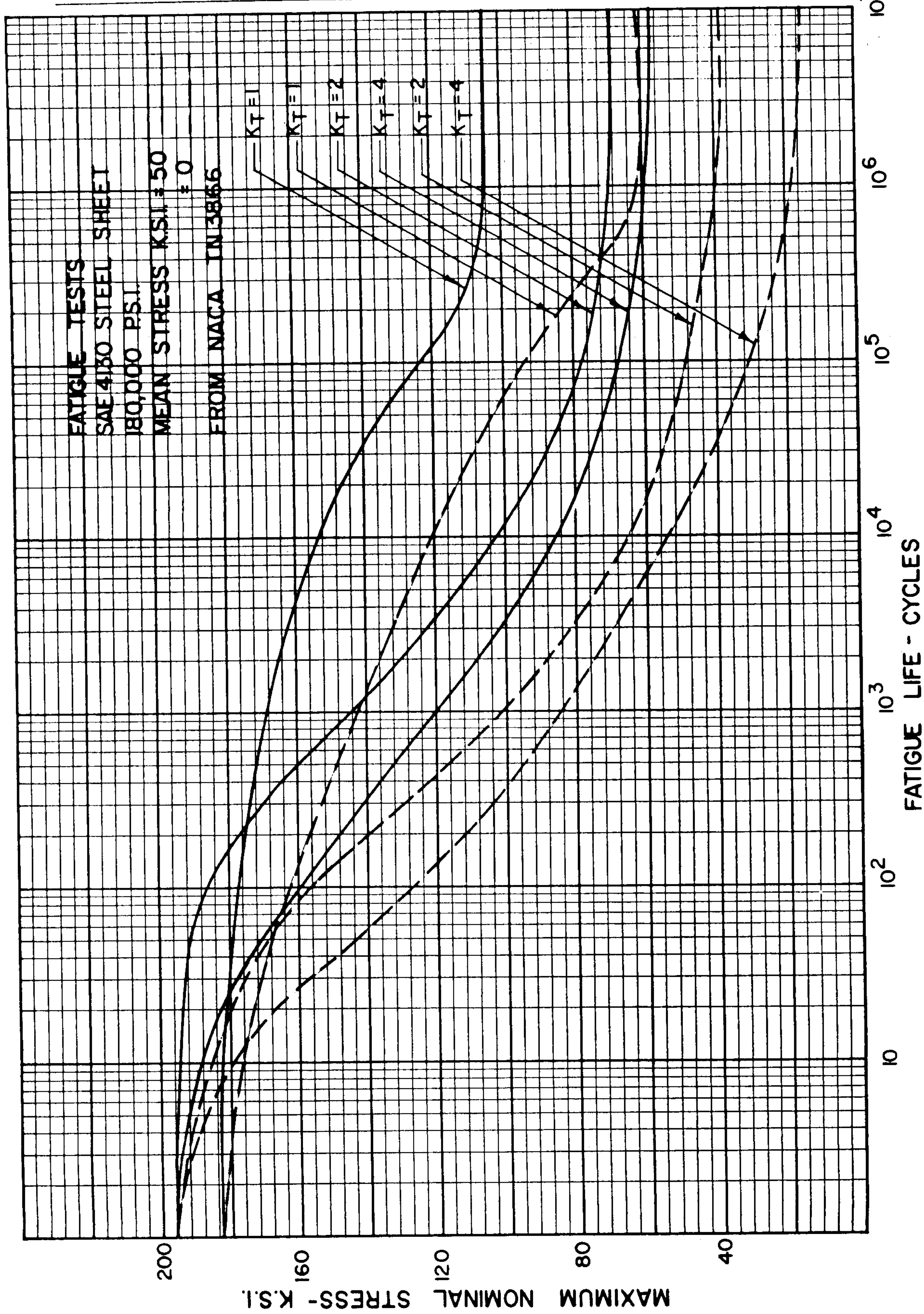
38.32 S-N CURVE 7075-T6



38.33 S-N CURVE 4130 STEEL, NORMALIZED



38.34 S-N CURVE 4130 STEEL, 180 KSI



38.4 REFERENCES

1. "Prevention of the Failure of Metals Under Repeated Stress" - Battelle Memorial Institute

2. NACA TN 2389 Fatigue Strengths of Aircraft Materials - Axial - Load Fatigue Tests on Notched Sheet Specimens of 24S-T3 and 75S-T6 Aluminum Alloys and of SAE 4130 Steel with Stress Concentration Factors of 2.0 and 4.0 by H. J. Grover, S. M. Bishop and L. R. Jackson.

3. NACA TN 2324 Fatigue Strengths of Aircraft Materials - Axial - Load Fatigue Tests on Unnotched Sheet Specimen of 24S-T3 and 75S-T6 Aluminum Alloys and of SAE 4130 Steel by H. J. Grover, S. M. Bishop and L. R. Jackson

4. NACA TN 3866 Fatigue Tests on Notched and Unnotched Sheet Specimens of 2024-T3 and 7075-T6 Aluminum Alloys and of SAE 4130 Steel with Special Consideration of the Life Range from 2 to 10,000 Cycles, by Walter Illg.

5. Stress Concentration Factors and Their Effect on Design by G. H. Neugebauer - Product Engineering, February 1943 and March 1943.

39. DESIGNING FOR REPEATED LOADS

by

R. L. Templin and E. C. Hartmann

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Aluminum Company of America

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DESIGNING FOR REPEATED LOADS

by

R. L. Templin and E. C. Hartmann

It is quite well known that metals may fail under the action of repeated loads, even when the loads are of considerably lower magnitude than those required to cause failure under a single application. When the stress at some location is high enough and is applied often enough, a minute crack appears and gradually extends during succeeding cycles until the strength of the part is so reduced that the part ruptures completely. The mechanism by which the initial fracture occurs and progressively enlarges is not completely understood. Some investigational work has been done on this problem and is continuing, with definite indications that good progress toward a satisfactory explanation of the mechanism of fatigue failure is being made. Meanwhile, the effects of fatigue failure are well known and are encountered often enough to have thoroughly impressed the engineering profession with the need for care in design to avoid premature failures in service.

Many factors influence the time interval between the formation of the first minute crack and the final failure of a part. On one extreme are parts in which this interval is very short compared to the over-all life of the part. On the other extreme are frequently encountered cases in which the fatigue crack seems to be self-stopping because of the fact that as the crack progresses it renders the part less stiff, with a resulting transfer of load to some other adjacent part capable of sustaining the added loading. Generally speaking, however, there is little comfort for the designer in the thought that some fatigue cracks are self-stopping or that some can readily be detected before they become dangerous. In too many instances these desirable conditions are not present and rapid propagation of the crack can lead to serious trouble. The only sensible attitude on the part of the designer is to try to proportion all parts so that fatigue cracks will not form during the lifetime which is considered adequate for the case in hand. This approach becomes more urgent, of course, for those designs where failure can jeopardize human life.

Limitations of Fatigue Analyses

Unfortunately, present-day knowledge of fatigue is not sufficiently advanced to permit design for a specified life within close limits even in the simplest parts. This is in marked contrast to present-day knowledge in the field of static design where, in many cases, the designer can work to specified loadings within close limits even on fairly complex assemblies. Even under carefully controlled laboratory conditions, using duplicate test specimens, testing machine, specimen preparation, and testing technique, it is not unusual to encounter ratios of 5 to 1 or 10 to 1 of fatigue life of specimens of the same material. Consideration of the many variables that occur in practical applications makes it clear why designing for a precise specified life is not a practical procedure. On the other hand, small decreases in actual maximum stress level for a given set of repeated-load conditions are usually conducive to large improvements in fatigue life, so that if the engineer focuses his attention on controlling maximum stresses the problem becomes less discouraging and the question of the exact life of the part frequently becomes of only academic interest.

The fact that it is not possible today to design for a specific life or to predict the life of an existing part within close limits should not discourage the engineer from trying to obtain satisfactory resistance to fatigue action. It is not a difficult matter to review a design and to modify it to give better performance under repeated loadings. Most of the outstanding cases of fatigue failure in actual service are traceable to conditions that are today recognizable in advance as poor from the fatigue viewpoint. Furthermore, many such cases are susceptible to fairly simple remedies once they have been brought to the attention of the designer. This indicates that proper attention given to such situations during the drafting-board and fabrication stages might avoid many service failures. It is this thought which has prompted the preparation of this paper, in which no attempt will be made to present design procedures of a mathematical nature for proportioning parts to resist fatigue action. Instead, attention will be focused on principles and practices of design, fabrication, and maintenance, which have proven to be of value in avoiding premature fatigue failures.

Laboratory tests have indicated that fatigue curves are usually substantially flat for the first 100 to 1000 or perhaps more cycles of load application. In other words, most parts will withstand 100 to 1000 repetitions of a load almost equal to their static ultimate load. This means that where the number of cycles of loading applied to a part is relatively small the part can safely be designed for static action alone without regard to the effects of repeated loadings, thus greatly limiting the field in which it is necessary to apply the principles and practices discussed in this paper. Other designs, though subjected to many cycles of very small loadings, are subjected to only a few cycles of loadings large enough to cause fatigue action. Here again it is unnecessary to give much attention to possible fatigue action. There still remain, however, many engineering assemblies and parts that do not fall into either of these categories and for which some attention to fatigue is necessary. Generally speaking, if loadings are present that produce tensile stress peaks (nominal stress times stress concentration factor) approaching or exceeding the fatigue strength of the material for the stress ratio and number of cycles under consideration, it is well worth-while to make some further fatigue analyses. In such cases, and especially where human life may be jeopardized, the principles and practices discussed herein should be given careful attention.

Selection of Material

Little attention will be given in this paper to the question of selection of material. Material selection is often dictated by considerations other than maximum fatigue strength. Furthermore, tests indicate that selection of material, within rather wide limits, is not nearly so important in achieving satisfactory fatigue life as is the exercise of care in design, fabrication, and maintenance. While minor improvements in fatigue life may be accomplished merely by changing material, few serious fatigue difficulties have been completely corrected merely by changing materials. Serious fatigue difficulties are almost always traceable to improper design, fabrication, and maintenance. Until design, fabrication, and maintenance are improved it is futile to seek relief through mere substitution of one material for another.

It should be emphasized that no satisfactory relationships seem to exist between fatigue properties and static properties of metals. Many attempts have been made to correlate tensile strength, hardness, and ductility with fatigue properties but any such relationships must be considered quite approximate and applicable to only a very limited range of conditions. Even for a group of alloys having the same base metal, static tensile properties can be changed appreciably by heat treatment and cold working without a corresponding change in fatigue properties.

Published fatigue strengths of materials must be used with caution in selecting materials for resistance to fatigue in engineering applications. Differences in fatigue strength values are obtained depending on the kind of fatigue test (direct-stress or repeated flexure), the stress ratio (ratio of minimum stress to maximum stress in each cycle), the shape and size of test specimen, the preparation of specimen, and the testing techniques used. Sometimes the samples used for the fatigue tests are not truly representative of the material to be used in the actual structure. It is recognized that factors such as internal discontinuities and defects may have deleterious effects on the fatigue properties. Smooth-specimen fatigue results often rate materials in a different order from notched fatigue specimens and no one style of conventionalized notch can adequately represent conditions in engineering structures and parts. Numerical differences in smooth-specimen fatigue results are often greater for a given group of related materials than the differences obtained on notched specimens.

Most of the fatigue data which have been published for various metals were obtained from tests conducted at room temperature. Some data are available, however, from tests made at other temperatures and these show that there is generally a reduction of fatigue strengths at elevated temperatures. It is important, therefore, that the operating temperature of the material be taken into account. Some parts of structures or machines which normally operate at or near room temperature may occasionally be subjected to higher temperatures for a short time. The effect of such temporary over-heating on fatigue properties should be considered.

Importance of Stress Concentrations

As has already been implied, fatigue fractures start at some location where the peak stress exceeds the fatigue strength of the material for the stress ratio and number of cycles under consideration. Designing for fatigue would be simpler if the determination of the peak stresses were not so difficult. Most engineering stress analyses deal only with nominal stresses and make no attempt to establish the actual peak stresses, an omission justified by the fact that localized peak stresses have little effect on successful static behavior of structures and machines using ductile materials. In fatigue, however, the peak stresses govern the life of the assembly. When the peak stress is within the elastic range and when the geometry of the part is one for which the theoretical stress concentration factor has been established by analytical or experimental techniques, it is relatively easy to calculate the peak stress. In practical design, however, such cases are conspicuous by their infrequent occurrence. In most cases the designer does not know all the theoretical stress concentration factors and has no practical way of establishing them with any certainty. Furthermore, in many cases where the theoretical stress concentration factors are known, the computed peak stresses are above the elastic range and some plastic action takes place on the first cycle, leaving a small amount of localized permanent set and a resulting modification not only of the peak stress but also of the stress range. In many other cases two or more stress concentrating features are superimposed so that the determination of the peak stress is much more complicated. Cases of the latter type arise, for example, when fabrication marks are superimposed on holes or re-entrant corners; punched rivet holes with rough surface on the inside and lubrication holes in highly stressed areas are other examples.

Even though the mathematical determination of peak stresses is usually not a practical design procedure, an understanding of this subject is the key to good design for fatigue. Almost without exception it can be stated that any modification of the form or dimensions of a part which alleviates the peak tensile stresses will improve fatigue life. Thus, by studying stress concentration factors, much can be learned about how to produce designs that are superior from the standpoint of resistance to repeated loads and how to evaluate approximately the influence of various geometric features. With this approach it is easy to understand why the experienced designer of parts subjected to repeated loads avoids sharp re-entrant corners, notches, and unnecessary open holes, and why he seeks smooth transitions and generous fillets. A little time spent studying information on stress concentration factors such as that given in various published articles and textbooks is worth a great deal to the designer who wishes to produce parts having superior fatigue life.

A frequent source of high stresses and consequent poor fatigue life is found in unsymmetrical and eccentrically loaded parts that must necessarily flex with each application of load. In many such instances, the secondary flexing of the part may safely be ignored in designing for static loads, but this procedure cannot be defended in the case of designing for fatigue. The best procedure is to eliminate the undesirable secondary flexing of the member but where this is impossible the extra stress produced should by all means be evaluated and taken into account in designing for fatigue.

Provisions of additional material to increase the stiffness of the part exhibiting secondary bendings will generally result in fatigue strength improvement, except where the additional material introduces severe stress concentrations as a result of abrupt changes in area or where the degree of flexing is practically constant regardless of the amount of stiffening material added. In the latter two cases the added material usually does more harm than good.

Joints and Connections

The most important source of fatigue difficulty is, no doubt, the joints and connections in an assembly. Few indeed are the joints and connections that do not involve stresses higher than those in the remainder of the assembly. Joints carefully bonded with adhesives and free from the more obvious discontinuities found in riveted, bolted, and welded connections, still involve stress concentrations that make the joints more susceptible to fatigue failure than the surrounding material.

Redundant Structures

A redundant structure, with its multiple load paths, when inspected adequately and regularly, may prove better suited to early detection and repair of fatigue cracks than a structure of comparable static strengths in which cracks in any one of several key members might precipitate sudden collapse of the whole. The existence of even a sizeable and plainly visible fatigue crack in a redundant structure does not necessarily mean that complete collapse is imminent. A crack of the same magnitude in a nonredundant structure, on the other hand, may be more likely to produce complete collapse before there has been an opportunity to detect and repair the damage. Neither type of structure is likely to survive severe service loadings after a fatigue crack is well developed in some important member, but a redundant structure is probably more likely to survive moderate service loadings under such conditions than is a nonredundant structure.

Influence of Fabrication Practice

While the design of a structure is probably the most important single item in assuring freedom from fatigue difficulties, the importance of fabrication should not be overlooked. The care with which the designer's instructions are carried out can influence to a considerable extent the final life of the structure. If the fabricators recognize the importance of smooth transitions and generous fillets and the harmful effects of sharp re-entrant corners and notches, they will be particularly careful not to spoil a good design by unintelligent exercise of their prerogatives. Even in those cases where the designer has unwittingly provided insufficient clearances and other difficult situations, the careful shop will attempt to remedy these matters in some manner which will not result in poor resistance to fatigue. Careful and conscientious attention to surface finishes, freedom from severe notches, looseness, and similar features will go a long way toward insuring good performance of the finished assembly.

Finish

It is difficult to draw a sharp distinction between good practice and poor practice regarding such items as surface finish, freedom from scratches, nicks, dents, gouges, and similar blemishes. There is little doubt that inspectors have sometimes gone to ridiculous extremes in their demands on the fabricator under the illusion that nothing but the finest quality is acceptable where fatigue is a possibility. While it can be shown academically that such extreme precautions have some slight beneficial effect, these benefits often do not carry over to practical cases, especially in structures. In the presence of all the usual stress raisers such as rivet holes, welds, and re-entrant corners, any beneficial effects from extreme care in surface finish and freedom from minor blemishes are completely overshadowed. Exceptions might be made in the case of highly critical machine parts and structural parts of marginal strength, but there is little reason to go to extremes on most parts of a riveted or welded assembly.

Special Techniques

Considerable attention has been given to special techniques that can be used to improve fatigue resistance through some form of cold working which sets up compressive residual stresses in the surface layers. Shot peening and localized cold rolling are two procedures used. There is no question of the beneficial effects of such treatments when properly carried out, but care should always be taken to avoid an excessive amount of such localized cold working. If the operation should be carried too far, incipient fractures may result. Anyone considering the use of these special treatments would do well to study the literature on the subject before proceeding with practical shop operations.

Other techniques are available for improving the fatigue characteristics of parts by setting up favorable residual stresses or otherwise altering the stress conditions. Generally speaking, anything that reduces the variable stress, without raising the peak tensile stress, will have a beneficial influence on fatigue life. If the peak tensile stress is lowered at the same time the variable stress is reduced the benefits are even greater. Sometimes an extreme reduction in variable stress can prove beneficial even though the peak tensile stress is slightly increased, but it should be remembered in such cases that it is the reduction in variable stress and not the increase in peak stress that is giving the benefit. One example in this general field is the proper tightening of bolts carrying tension so that repeated loadings produce only a small fluctuation in tensile stress in the bolt.

Residual Stresses

The question of the harmful effects of residual stresses is often raised, especially when the residual stress in question is one induced accidentally or otherwise by the fabricator. In this connection it must not be forgotten that most metal products have some degree of residual stress over which the designer has no control. These initial residual stresses come about by virtue of operations performed by the material producer. Each subsequent operation merely modifies the residual stresses already present so that without a special investigation it is impossible to evaluate accurately the final residual stress resulting after any specific operation performed on a given piece of metal. This whole subject involves many questions that have not yet been answered but what little information is available seems to indicate that residual stresses of the type induced by the normal heat-treating and forming operations in an ordinary fabricating shop are not extremely important in governing the fatigue life of a part. There is considerable evidence to indicate that the design of a part is much more important in influencing its fatigue life than are any peculiarities of residual stress that may be present in the metal as a result of normal fabricating procedures.

Maintenance

Once an assembly has been completed and placed in service, it becomes the responsibility of the maintenance personnel to see that the life of the part is not shortened through gross neglect. If the various parts are well designed and carefully built, the assembly should perform satisfactorily throughout its intended life without serious fatigue difficulties. If properly maintained it will do so. On the other hand, however, such items as corrosion, wear, improper lubrication, overheating, and marks left by accidents, if allowed to take their toll without proper corrective measures may increase the stress peaks or introduce new ones and thus definitely shorten the fatigue life of the assembly. Proper cleaning, application of protective coatings, lubrication, and careful repair of damage will go far toward insuring for any given assembly the life built into it by the designer and fabricator.

Rules for Improving Fatigue Life

1. Give major attention to actual stresses, including stress concentrations, rather than to the nominal average stresses.

2. Visualize how load is transferred from one part or section to another in a structure or machine and the distortions that occur during loading, in order to help locate the points of high stress.

3. Avoid adding or attaching secondary brackets, fittings, handles, steps, bosses, grooves, and openings at locations of high stress.

4. Use gradual changes in section and symmetry of design.

5. Give careful attention to location of joints and types of joints used. Joints are one of the most frequent sources of fatigue weakness.

6. Use symmetrical joints wherever possible. A symmetrical joint free from secondary flexing is usually superior to an unsymmetrical joint of equal static strength. Thus double-shear riveted joints are better than simple-lap joints.

7. Use suitable means to stiffen unsymmetrical joints. Where a symmetrical joint cannot be achieved, it usually helps to stiffen the joint so that secondary flexing is reduced to a minimum.

8. Design joints so that all parts will participate equally. Such joints are usually superior to those where some parts are carrying more than their share of the load. This is true even though the poor distribution applies only at moderate load levels and disappears in the event of serious overload. Simple rivet patterns are often as good or better in fatigue than the more elaborate patterns sometimes used to increase the calculated static joint efficiency.

9. Do not use rivets for carrying repeated tensile loads. Rivets are not well suited for carrying such loads, particularly if there is any prying action against the head of the rivet. Rivets which perform in excellent fashion under static tensile loadings may give a very poor performance under repeated tensile loadings. Bolts are usually superior to rivets under repeated tensile loadings.

10. Give more attention to possible tensile failure of the plates rather than the shear failure of the rivets under repeated loads. Riveted joints are much more prone to fail by tensile fracture of the member plates themselves through the rivet holes than by shear of the rivets. The possibility of rivet shear fatigue can almost always be dismissed in any joint that is reasonably well balanced from the static viewpoint.

11. Avoid open holes and loosely filled ones. Open holes and loosely filled holes are a more likely source of fatigue difficulty than holes containing well driven rivets or tight fitting bolts and pins.

12. Give more attention to holes containing stressed rivets or bolts than to those containing idle or unstressed rivets or bolts even though the nominal stress in the adjacent plate material is the same in both cases. The peak stress at the edge of the hole is higher when part of that stress is introduced by a load on the rivet or bolt in question.

13. Do not expect maximum fatigue strength of parts in which machine countersunk rivet holes are used. Machine countersunk rivet holes seem to be more prone to produce fatigue failures than other types. Dimpled rivet holes are poor if any fracturing of the metal is caused by the dimpling operation.

14. Use tightly drawn-up bolts, because they are less disturbing to fatigue life than those less tightly drawn up.

15. Use simple butt joints in preference to other types of welded joints. Carefully-prepared welded butt joints seem to be the equal of riveted joints in resistance of repeated loads, but poorly-prepared welded joints are definitely inferior to riveted joints. Spot-welded joints are usually inferior to riveted joints in fatigue.

16. Give attention to the geometry of a welded joint, including such factors as smoothness, undercutting, cracks, excessive porosity, spatter, and symmetry. These factors are often more important in affecting the resistance to fatigue than filler metal, welding process, or flux, except in so far as these latter items affect the others noted.

17. Dress weld beads flush with the adjacent plates for welded butt joints to have maximum fatigue strength. Such procedure may reduce somewhat the static strength of the joint but will be beneficial in fatigue, especially where the weld beads, before dressing, are not very smooth.

18. Alleviate possible fretting between faying surfaces. Fretting may be a cause of early fatigue failure and is often prevented by decreasing the spacing of the bolts or rivets used to hold the parts together. In some cases suitable gaskets have been found helpful in minimizing frettings.

19. Do not depend upon arresting the growth or propagation of fatigue cracks by drilling holes at or near the ends of active cracks or by removing the crack and the adjacent material. The latter procedure may be helpful in arresting fatigue failure at a specific place but may only shift slightly the location of a new fatigue crack. Neither method has been found thoroughly reliable.

20. Give preference to redundant-type structures where this type of structure is permissible. A redundant structure, with its multiple load paths may prove better suited to early detection and repair of fatigue cracks than a nonredundant structure in which a crack in any one of several key members might be more likely to precipitate sudden collapse of the whole.

21. Give careful attention to fabrication details in order to improve fatigue life. Assuming that the designer has done a good job in specifying such items as the size of fillets, location of oil holes and cutouts, machining finishes, and workmanship, the fabricating shop should adhere strictly to these details which are so important in obtaining maximum fatigue strength. Beyond compliance with drawings and specifications, it is important that high-quality workmanship be used in making and assembling the parts. Any needed departures from the drawings or specifications should be reviewed by the fabricator with the designer.

22. Choose the proper surface finishes, as these may have an important effect on the fatigue strength of machines and structures. In numerous instances, however, these effects may be overshadowed by those resulting from the presence of unavoidable stress raisers such as rivet holes, welds, re-entrant corners, openings, and changes in section.

23. Provide suitable protection against corrosion.

24. Avoid types of metallic plating that have widely different characteristics from the underlying material and therefore may be conducive of early fatigue failure.

25. Watch for opportunities to prestress members or parts of members in such a way as to reduce the range of stress without appreciably increasing the maximum stress in the cycle.

26. Consider the benefits of shot peening and localized cold working, which set up beneficial residual stresses in the surface layers.

27. Give attention to proper maintenance of machines and structures subjected to fatigue action in order to obtain maximum service life. Suitable maintenance includes adequate protection against corrosion, wear, abuse, overheating, improper lubrication, and repeated overloading.

28. Avoid the use of machines and structures under conditions involving vibration at the critical or fundamental frequency of either individual parts or of the structure as a whole. Such vibrations usually involve many cycles of relatively high stresses. However, vibration does not invariably connote stresses that will cause fatigue failures in machines and structures. The mean stress, amplitude of variable stress, total number of repetitions, together with the fatigue properties of the material, are the important items to be considered.

29. Give consideration to the effects of temperature on the fatigue properties of metals used in machines or structures which are expected to withstand repeated loads at high or low temperatures.

References for Further Information

The designer of machines or structures intended to withstand repeated loads should become reasonably well acquainted with the technical literature now available on fatigue. One of the outstanding books on the subject is, "Prevention of the Failure of Metals Under Repeated Stress," by Battelle Memorial Institute (1941). Committee E-9 of the American Society for Testing Materials has prepared a "Manual on Fatigue Testing," 1949, (Special Technical Publication No. 91), which is very informative. This committee is also compiling an extensive bibliography on the subject. In the American Society for Metals publication, "Metals Review," will be found references to many articles on fatigue. Still other references will be found in the American Society of Mechanical Engineers' "Applied Mechanics Review," and in the references given in CADO "Technical Data Digest," issued by the U.S. Air Force, Air Materiel Command. These references include the majority of papers on fatigue currently published.

40. PIPING DESIGN DATA

NOTE: Piping design data, which were included as Section 40 of the previous Hydraulics and Landing Gear Manual, are now available in a separate volume entitled Hydraulics Piping Manual.

50. MECHANISMS

CONTENTS

50. MECHANISMS

50.1 GENERAL As aircraft become more specialized and compact, their satisfactory performance relates more closely to the reliability of their operating machinery. Assuming an acceptable design approach has been selected, adequate reliability can be developed by trial and error methods but both time and expense can be saved by meticulous attention to details in the design stage and by observing certain basic principles which appear obvious yet are often ignored.

Mechanisms must be simple. The use of complicated mechanisms and "gadgets" must be avoided. The fancied or small real gain in operating efficiency or weight is seldom worth the additional cost and time of manufacturing adjusting and maintaining the more complex mechanism.

Mechanisms must be easy to adjust and service. Adjustment must be a simple operation and access, both for personnel and tools, for this adjustment must be adequate. The ease of access should be directly proportional to the frequency of adjustment to the mechanism. Adjustment should be a one man job wherever possible.

The designer must bear in mind that often military maintenance crews may consist of relatively inexperienced men working with only the most common of tools. They must frequently make repairs under adverse weather conditions and other forms of duress.

50.2 IMPORTANT CONSIDERATIONS

50.21 CONSTRAINT Probably the most important concept the mechanisms designer must acquire is to recognize the degree of constraint in a system of parts. Particularly in aircraft where significant accelerations are experienced in any direction he must walk the narrow path between under constraint which will allow the movement of parts into unacceptable positions or areas and over constraint which costs (a) close manufacturing tolerances, (b) low design efficiency (assumed load distribution in redundant supports must be conservative), and since nothing in an airplane is rigid, (c) a significant deflection problem.

On any part, stationary or moving, there are three possible moments and three possible forces. Any unit or part must be secured or constrained to resist these moments and forces. Ideally the constraint should be limited to resisting each force at only one point and each moment by only one couple. Over constraint is troublesome in moving parts where failure to meet the dimensional characteristics of the two or more restraining points for any particular force will cause binding on the moving parts.

To help visualize - in any mechanism not over constrained any mounting point may be moved in any direction and still have a satisfactory operation. Conversely, a mechanism is over constrained when any joint can be made freer, i.e., change a pin to a universal, an ordinary bearing to a self-aligning bearing, etc., and still have an operating mechanism. Visualizing these two quick checks for each point in a mechanism for each significant position for any possible cycle is a worthwhile expenditure of design time.

50.22 FRICTION The sliding friction exhibited by materials in bearing is extremely variable. The probable range of friction coefficient ($\frac{\text{sliding force}}{\text{normal force}} = \mu$) for most bearing situations in aircraft machinery is from $\mu = 0$ to $\mu = 0.25$ but occasionally much higher.

IF FRICTION HELPS NEGLECT IT.
IF FRICTION HURTS PROVIDE FOR IT.

Depending on the importance of the mechanism, the operating conditions and the compromise cost of weight and space, the designer frequently ensures satisfactory function at $\mu = 0.25$. Any mechanism which becomes "self locking" at μ less than 0.25 must be brought to the attention of Design Supervision.

50.221 SELF LOCKING A mechanism becomes self locking at that value of coefficient of friction which when multiplied by the sliding support bearing forces reacting the applied force, added to any other resisting motion (air pressure, air load, weight) just equals the applied force tending to cause motion. Examples of the application of this principle appear in Sections 80 and 40.

50.23 ENVIRONMENT Whether or not it is possible to tolerate all conditions of heat, dust, vibration and other forms of environmental hazard, it is important that the designer consider these hazards and provide for them insofar as is expedient. Any significant default must be brought to the attention of Design Supervision.

50.24 DURABILITY The ability of a mechanism to perform the required duty for the required life with a minimum of service is a measure of durability. If required duty and life are available with reduced tolerance control, finish quality, materials quality, weight and cost then as time is available such reductions should be incorporated. Methods and consequence of failure and ease of inspection should be weighed heavily in any quality reduction analysis. It should be noted here in places of anticipated wear or other forms of deterioration the designer would do well to decide on and record acceptable limits and ensure adequate function under these limits.

50.3 MECHANISM SCHEMATICS AND MOTION STUDIES

50.31 PLANE OF DRAWING

a. The plane in which motion studies are made should be the plane in which the motions take place.

b. If all motions take place in a series of parallel planes, the motion study should be made on the projection of the parallel planes onto one plane.

c. Full effect of three dimensional motion must be considered.

50.32 PATH OF MOTION

a. Mechanism studies must show a sufficient number of intermediate points to be able to completely visualize the motion of all levers and links.

b. The path of travel of all pivot points must be plotted.

c. A generous allowance for overtravel must be included in moving mechanisms.

50.33 MEMBER SIZES

a. Approximate member sizes should be determined early in the motion study in order to check clearances within the mechanism and clearances with external parts.

b. Member sizes should be sketched on the motion study at points where the clearances might be critical.

c. Detail design must be worked out and sketched at points where links fold back on one another past 90 degrees to ensure that satisfactory members can be made.

50.34 MOMENT ARMS

a. For determination of moment arm loads in motion studies, it must be remembered that, neglecting friction, loads act along the center lines of pin-ended members. Hence, the moment arm is the perpendicular distance from the projected center line of the link to the pivot point regardless of the angular position of the layer.

b. Short moment arms should be avoided because of the high deflection relative to movement and the high friction loads. This includes short moment arms due to short levers and links almost in alignment.

c. The moment arms, loads, and moments should be plotted in tabular form for each position of the mechanism which is analyzed.

50.35 LOADS

a. External loads and the resultant loads at the other end of the system can best be shown graphically, plotted against travel or angular displacement.

b. Loads may be calculated by the following two methods:

1. Using moments arms and loads.

2. Equating system input and output work, neglecting friction.

The first method is preferred, since it may be more readily checked. The second method may be used as a check on the first method by applying it to one or two points in the path of motion. Allowance for friction in the first method may be done by use of the friction circle as explained in 50.38.

c. Strength Calculations on short members should be calculated with the load eccentric and operating tangent to the friction circle. In Section 50.38, Figure 50-1, the link 2-3 is subjected to the eccentric load acting along the line MM or NN - hence introducing bending conditions at the ends of the link.

d. An allowance for preload must be included in motion studies. Consideration must be given to the fact that in a system of less than infinite rigidity a preloaded member does not start to move at the same time as an unpreloaded member. As an example, during the spread cycle a wing fold system may preload the wing to spread before inserting the lock pin. The fold cycle releases this preload before the lock pin is withdrawn thus depending on the preload deflection the lock pin may be loaded and resist withdrawal.

50.36 DEFLECTIONS

a. The effect of deflection of supporting structure must be considered when links and levers approach alignment. This deflection must be considered to ensure that the mechanism will not go over dead center due to deflection increasing the distance between the pivot points.

b. The effect of deflections on friction loads must be considered.

c. The effect of deflections in causing actual mechanical interferences must be checked.

50.37 TOLERANCES The effect of tolerances which affect the center distance between pivot points and thus affect angles, motions, etc., must be considered. This must especially be checked with non-adjustable links or when links and levers approach alignment.

50.38 FRICTION The friction circle method as explained below may be used in compensating for friction in a mechanism when a moment arm-load analysis is run on the mechanism.

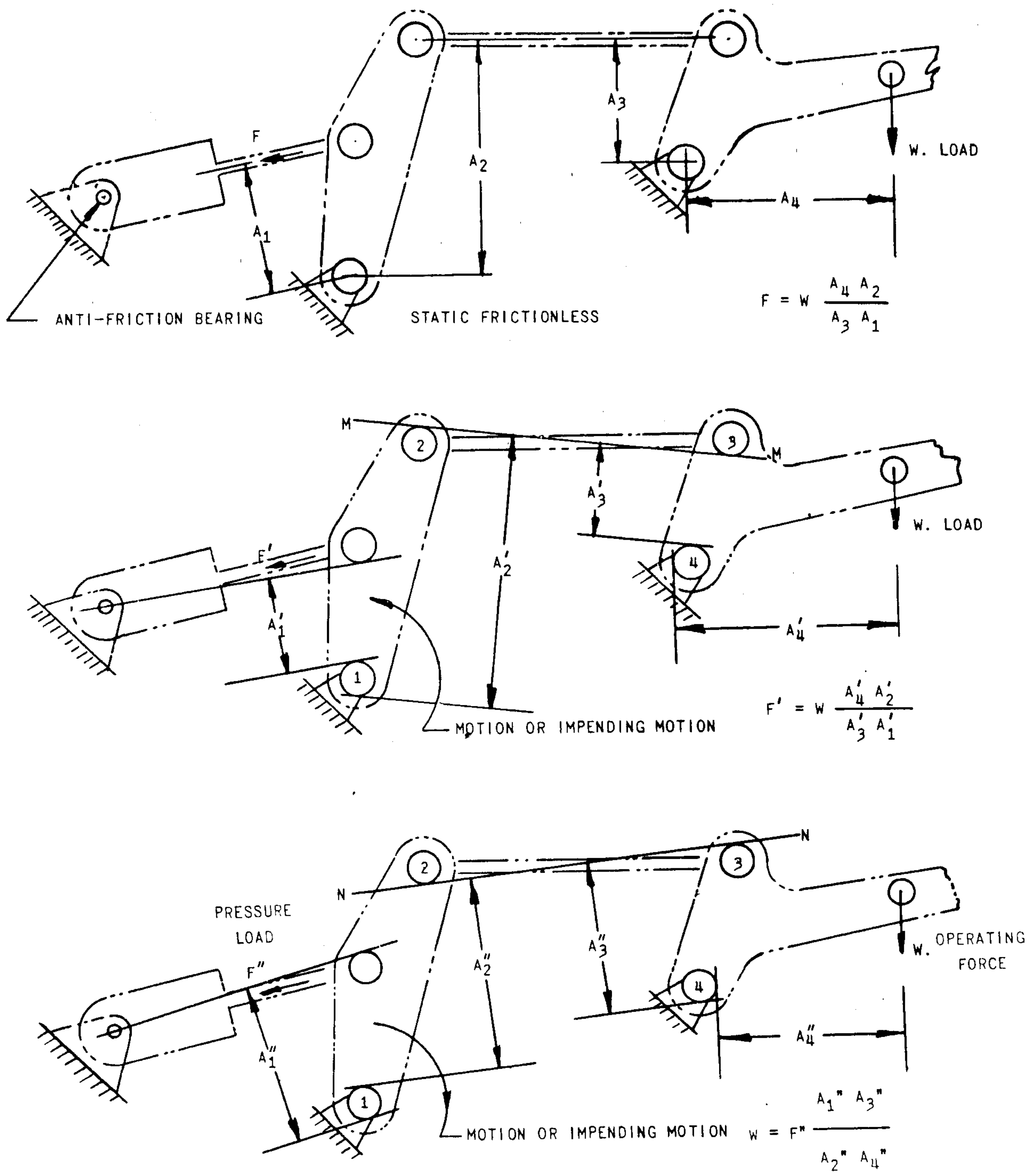
a. Each of the pivot points in the mechanism is replaced by a friction circle. The diameter of this friction circle is equal to the diameter of the bushing, or bearing, times the coefficient of friction. For all plain bearings, the coefficient of friction shall be assumed to be 0.25; hence, the diameter of the friction circle is equal to 0.25 times the diameter of the plain bearing. For all anti-friction bearings; i.e., roller, ball, or needle, the coefficient of friction shall be assumed to be zero - hence, no friction circle shall be used at such bearings.

b. In measuring moment arms, the length of the arm is not measured to the pivot point but is measured to the actual line of action of the force which is tangent to the friction circle. This graphical method automatically takes into account the retarding friction force. Thus, in the example Figure 1, 50.38, the lines of force action are shown. The size of the friction circle has been exaggerated in order to clarify the lines of force action.

50.38 FRICTION (CONT)

EXAMPLE:

FIGURE 50-1



c. In order to determine the direction of the force lines, i.e., determine whether MM or NN is the proper line of force in the link 2-3 and whether A' or A" is the proper lever arm for the operating force, the following procedure is used.

1. The system shall be assumed frictionless; i.e., all forces acting along the links and all moment arms being measured from the pivot point to the center line of the link.

2. A general equation shall be worked out for the operating force in terms of the load.

3. Since the friction tends to increase the operating force, all moment arms in the numerator of the general equation shall be increased while all moment arms in the denominator shall be decreased.

50.39 MISCELLANEOUS

a. Neglecting friction, all roller loads act at 90 degrees to the surface on which the roller bears.

b. The effect of out-of-plane rotations on loads, friction, etc., must be considered.

50.4 DESIGN FEATURES EMPHASIZED BY DOUGLAS EXPERIENCE

a. Extreme changes in section are to be avoided even if calculated loads are nil.

b. Sequenced operations depending on a check valve seating (sequence valve) will malfunction and are undesirable; depending on the consequence of the malfunction they are unacceptable.

c. Mechanisms performing locking functions by changing geometry over center rather than by disconnecting are more reliable particularly when environment is not completely controllable.

d. On-center and near on-center linkages often require a special stop to avoid over center travel allowed by deflection. Stops must be arranged so that primary load deflection has no influence on geometry.

e. Latches releasing under load must be examined for component inertia during the release cycle.

f. Latches that vary bearing area during the unlatch cycle and that unlatch under load must be carefully designed to avoid galling during life test.

g. Avoid latching moving parts to moving parts, experience shows this frequently leads to impossible tolerance situations.

h. Mechanism is more expensive than structure; 2-3 pounds of structure weight can be used to eliminate 1 pound of functioning machinery.

NOTE: A wealth of test and development data is readily available to the designer in the form of hydraulic and landing gear test reports. Responsibility for becoming sufficiently aware of this information for proper use rests squarely on the designer.

i. If an item can be used as a handhold or step by personnel, assume it will be used as such and protect or strengthen accordingly.

j. Provide ample adjustment provisions in areas where structural tolerance or manufacturing buildups can occur; e.g., latches, stops, switches, braces from gear to structure, etc. Don't believe anybody that says that structural attach points will be located by first-class tooling. They always say this, but they never do it. Structural points will float in any direction at least 0.50 inch per 100 inches with respect to any point on a different sub-assembly. Allow sufficient adjustment for gear in up position.

k. Attachments for hydraulic actuators must be designed for fatigue cycles because of pressure cycles in the system. Don't depend on proper adjustment of cylinder to avoid these loads.

l. Good fatigue life depends absolutely on generous blending of intersections, large fillet radii, and especially external edge radii. All joints subject to relative motion shall be designed so that variations in torque on the pivot bolt will not cause binding or jamming.

50.41 LUBRICATION

a. Don't use graphite-loaded grease in high-temperature antifriction bearing applications because it dries up, leaving hard graphite collection to interfere with proper bearing function.

b. Lubricate and protect all critical joints. Dry lube must be compatible with oiling.

c. Lube fittings must be accessible and located in a region of low stress level.

d. Lubricated mechanisms located in a high-temperature area can cause jamming from burned oil carbon. Consider the possibility of lubrication occurring when not allowed.

e. Left- and right-hand part lubrication fittings should be multiple or located on the part so as to be accessible when installed. Ensure that unused passage is lubricated periodically to prevent corrosion.

f. Long lubrication passages result in frozen grease and blocked fittings.

g. Unclamped monoball bearings must have two grease fittings to ensure proper lubrication on ball (ID and OD).

h. Put grease distribution grooves in large bushings that will be loaded while being lubricated, and make sure groove gets grease to the area that is under load.

i. Don't use plasma-sprayed bronze coating for bearing surfaces. It cannot be applied reliably enough to keep it from flaking off.

j. Provide lubrication in highly loaded nonrotating landing gear mechanism joints to prevent stress corrosion.

k. Provide $\sqrt[64]{}$ surface finish in lube holes of high-strength steel parts and carefully blend intersection at large bore.

50.42 <u>BUSHINGS</u>

a. Make bushing out of Al-Ni Bronze; centrifugal castings are permissible, but ring forgings are cheap and require no casting factor in stress analysis.

b. Avoid beryllium-copper bushings, especially where there is a shock load or much distortion of the parts under load.

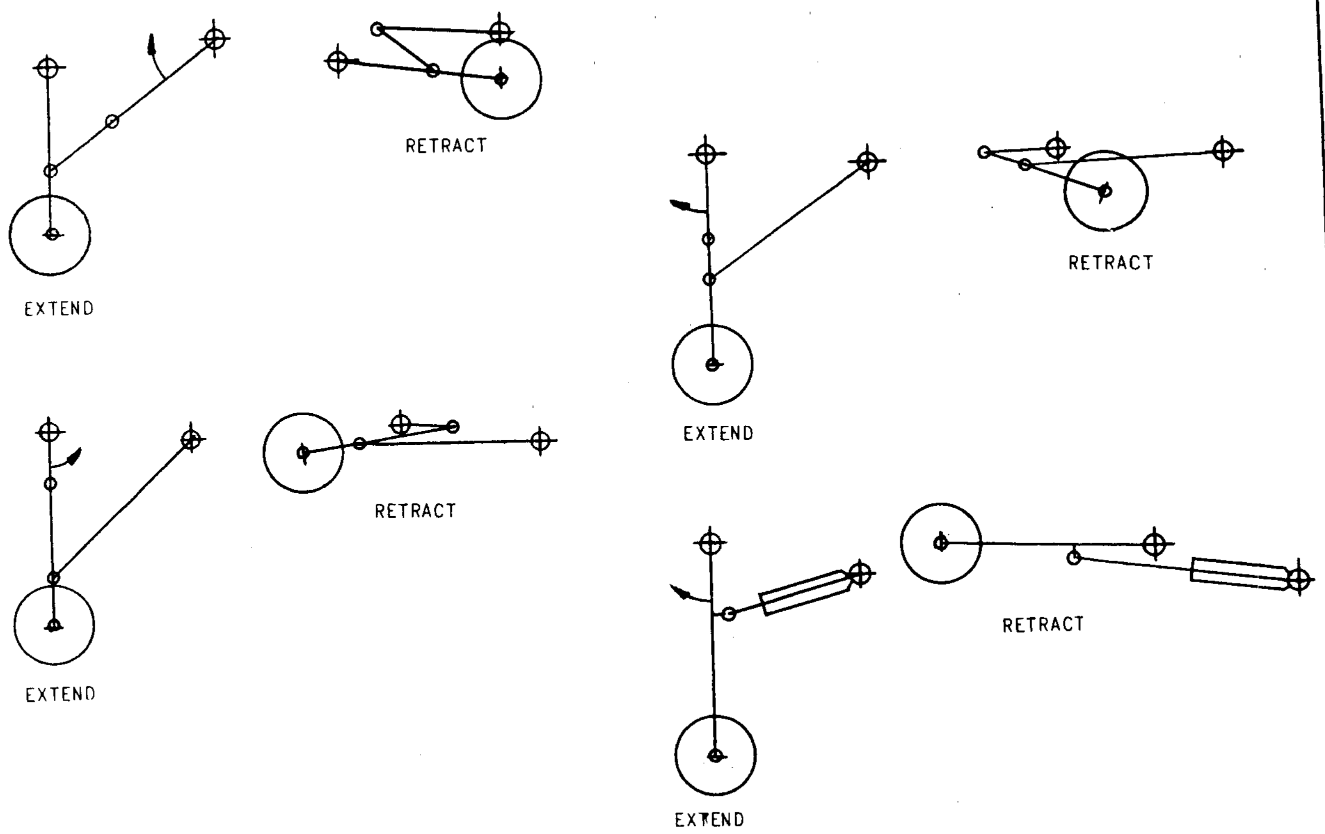
c. A machining relief is not necessary on the OD of shoulder bushings.

d. Make fail-safe provisions for flanged bushings if failure of the flange can allow the bushings to migrate.

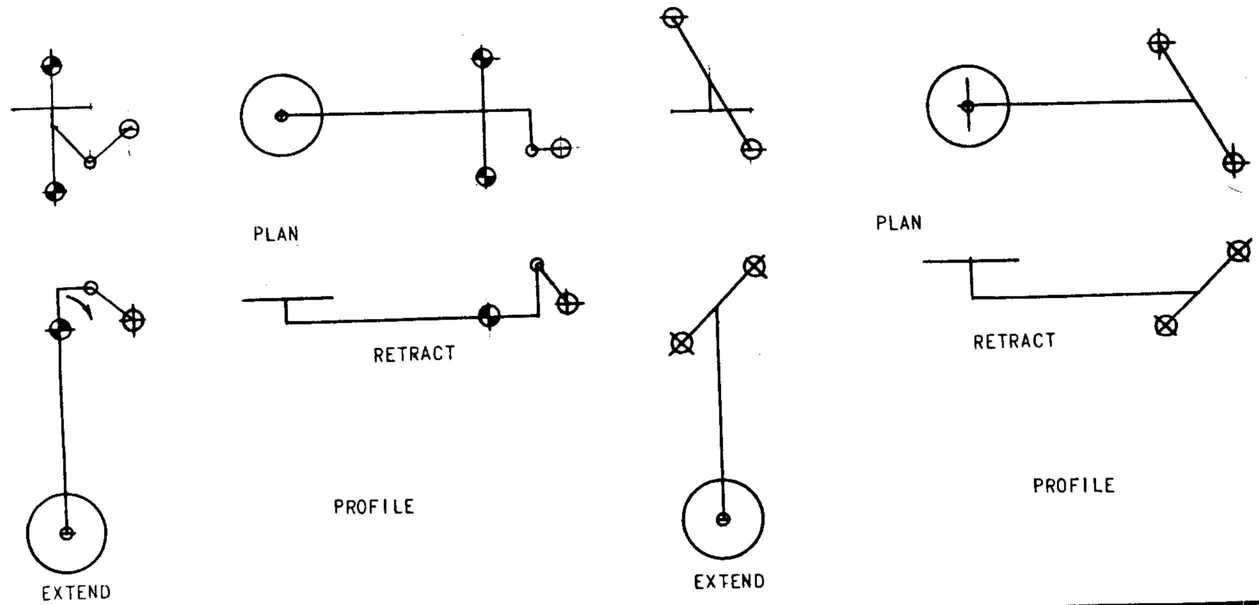
e. Provide means for extracting bushings easily without damage to basic part. Don't make the walls of large bushings too thin. Don't use two bushing flanges in moving contact. Put a CRS washer between them or chrome-plate one of the flanges.

50.5 TYPICAL HYDRAULIC AND LANDING GEAR MECHANISM

50.51 LANDING GEAR ACTUATING LINKAGE

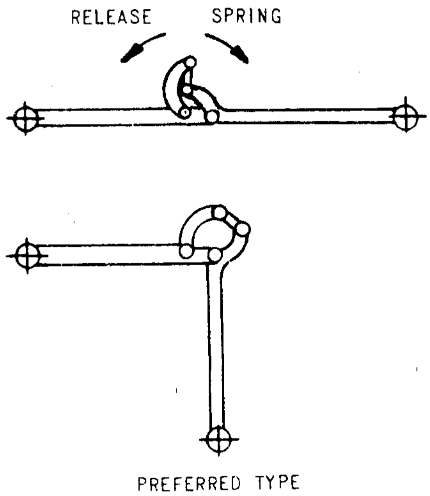


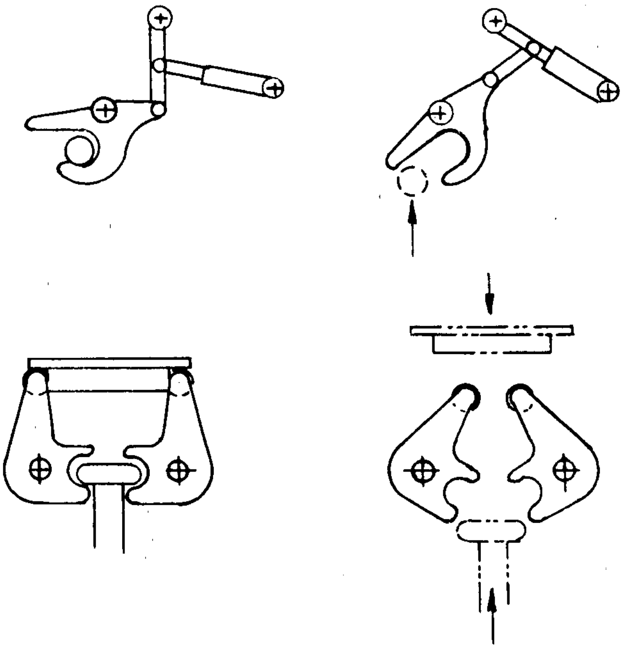
STRUT ROTATING



50.5 TYPICAL HYDRAULIC AND LANDING GEAR MECHANISMS (CONT)

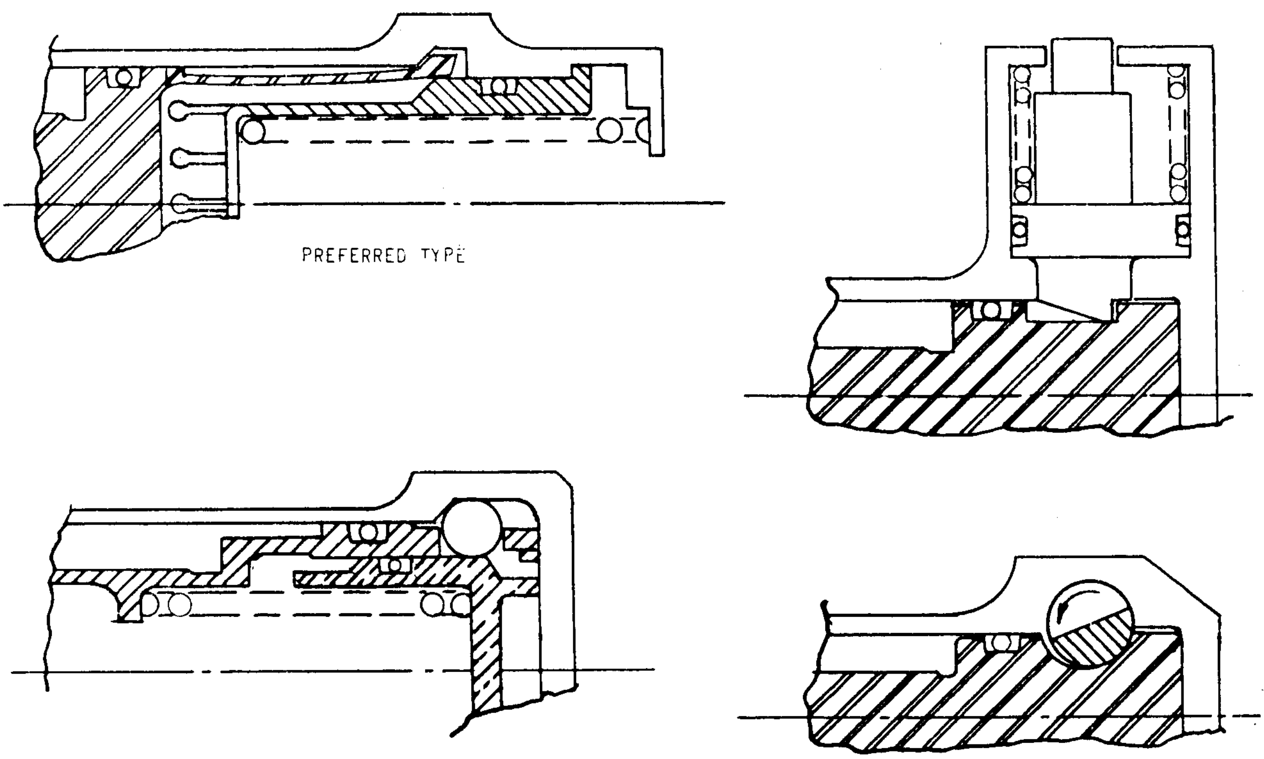
50.52 EXTERNAL LATCHES





50.53 INTERNAL LATCHES

Use when required



60. ACTUATING CYLINDER DESIGN

CONTENTS

60.1 GENERAL

60.11 PURPOSE The purpose of the hydraulic actuating cylinder is to convert fluid power to mechanical power. Actuating cylinder design is best accomplished by maintaining simplicity and by the use of proven, reliable methods in keeping with good design practice. Available space and design requirements will greatly influence the simplicity of the design and must be considered early in the event "tradeoffs" with other design sections are required.

60.12 TYPES OF ACTUATING CYLINDERS A few of the more commonly used actuating cylinders are shown schematically in Figure 60-1 and tabulated on the following page.

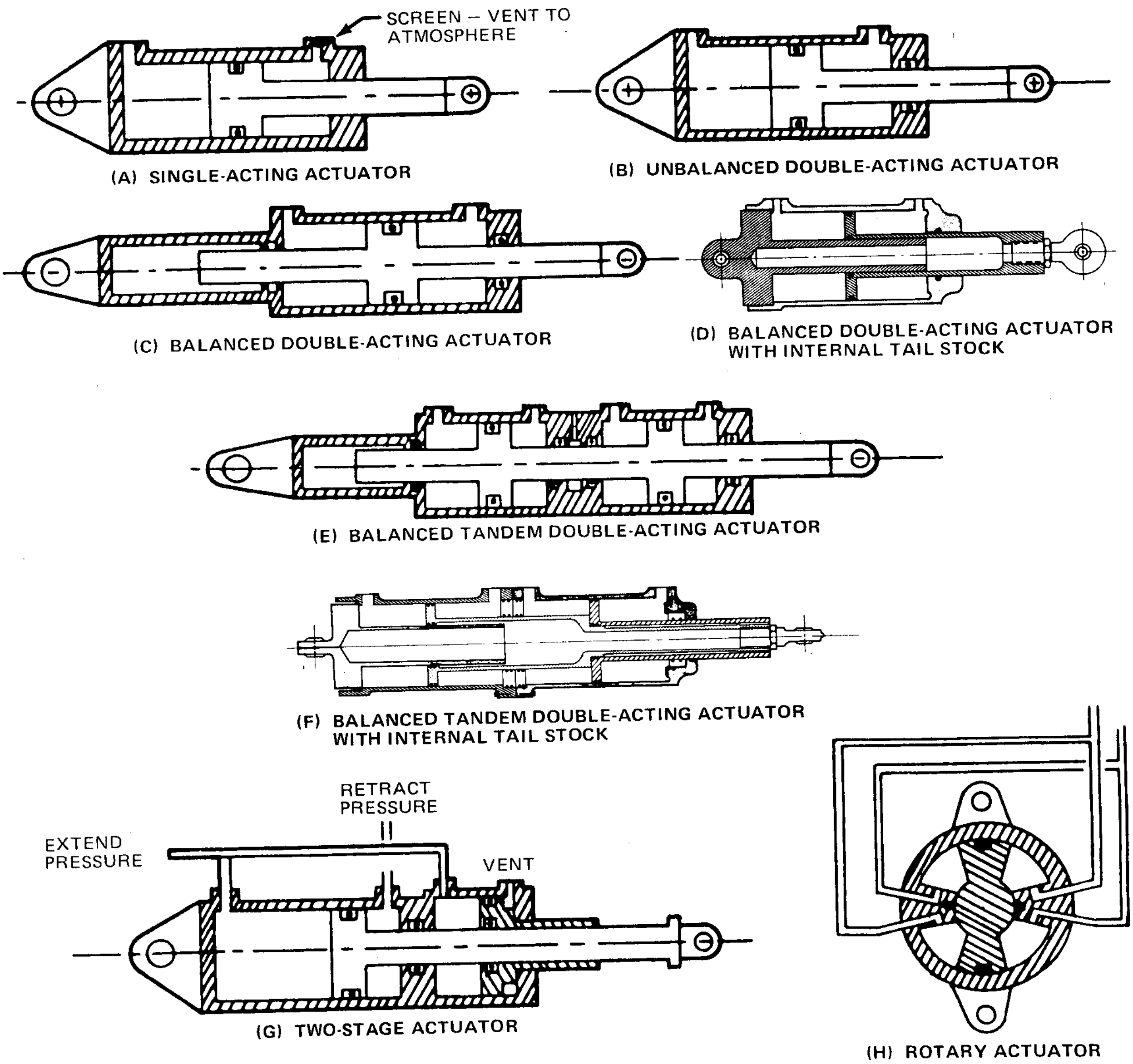


FIGURE 60-1. TYPES OF ACTUATORS

FIGURE	TYPE	USE	ADVANTAGES	DISADVANTAGES	REMARKS
60-1A	SINGLE ACTING	FORCE REQUIRED IN ONE DIRECTION ONLY	FEWER SUPPLY LINES, PACKINGS, ETC.	ATMOSPHERE SIDE REQUIRES CORROSION PROTECTION	USUALLY SPRING ACTUATED IN OPPOSITE DIRECTION
60-1B	UNBALANCED DOUBLE ACTING	DIFFERENT FORCE REQUIRED IN EACH DIRECTION	SHORTER LENGTH, LIGHTER WEIGHT, AND FEWER PACKINGS THAN BALANCED	DIFFERENTIAL FLUID DISPLACEMENT AFFECTS RESERVOIR. PRESSURE BOOST PROBLEMS	PROBABLY THE MOST COMMON TYPE
60-1C	BALANCED, DOUBLE ACTING	SAME FORCE REQUIRED IN EACH DIRECTION	NO DIFFERENTIAL DISPLACEMENT OR PRESSURE BOOST PROBLEMS	LONGER AND HEAVIER WITH MORE PACKINGS THAN 60-1B	PERMITS SMALLER RESERVOIR
60-1D	BALANCED, DOUBLE ACTING INTERNAL TAIL STOCK	SAME FORCE REQUIRED IN EACH DIRECTION	SHORTER LENGTH THAN 60-1C	COMPLICATES MACHINING	GOOD FOR CONFINED SPACE
60-1E	TANDEM	PROVIDE REDUNDANT MEANS OF ACTUATION	CONTINUES TO OPERATE WITH ONE SYSTEM LOSS	INCREASED COMPLEXITY	USED IN PRIMARY FLIGHT CONTROL SYSTEMS
60-1F	TANDEM WITH INTERNAL TAIL STOCK	PROVIDE REDUNDANT MEANS OF ACTUATION	SHORTER LENGTH THAN 60-1E	COMPLICATES MACHINING. SLIGHT UNBALANCE WITH ONE SYSTEM OUT	GOOD FOR CONFINED SPACE
60-1G	TWO-STAGE	DIFFERENT FORCE REQUIRED FOR PORTION OF STROKE	PROVIDE SOLUTION FOR UNIQUE ACTUATION PROBLEM	INCREASED COMPLEXITY	CAN ALSO PROVIDE FINITE STROKE POSITIONS
60-1H	ROTARY	PROVIDE DIRECT ROTARY OUTPUT	OPERATE IN LIMITED ENVELOPE. SAME AS 60-1C	CERTAIN APPLICATIONS INTOLERANT TO SEALING CHARACTERISTICS	DOUGLAS HAS NOT USED THIS TYPE

FIGURE 60-1. TYPES OF ACTUATORS (CONTINUED)

60.2 ACTUATING CYLINDER DESIGN REQUIREMENTS

Before the design can be initiated, a complete and thorough understanding of the design requirements must be attained. In most cases there will be many requirements from various sources that will dictate the design. A checklist of requirements and sources is as shown in Table 60-1:

TABLE 60-1

ACTUATING CYLINDER DESIGN REQUIREMENTS AND SOURCES CHECKLIST

Requirement	Source	Remarks
Hinge Moment versus deflection or load versus stroke curves.	Struct Dynamics or Aerodynamics Group	If hinge moment or load is based on wind tunnel models, it may be necessary to compute hinge moments or load using scale factors.
Load Factors or G Loads	Aerodynamics, FAA, MIL Specs or customer specifications.	Load factor may vary with the airplane type (military, commercial, carrier based, V/STOL).
Customer Requirements	MIL Specs or FAA Requirements	If the regulating documents do not specify load or strength requirements, supervision and/or strength group should be consulted.
Snubbing	Aerodynamics or customer specifications.	Assume snubbing is required for all actuating cylinders on commercial airplanes.
Actuating Time	Aerodynamics or Customer Specifications.	
Number of Drive Stations and Adjacent Structure	Tradeoff Studies	Space and clearance for piping should be considered.
Ambient Temperature	Thermodynamics Group	Will affect type of system used, especially regarding fluid medium and packings.
Vibration Other Environmental Conditions	Structural Dynamics Usually in report form along with temperature and vibration data.	
System Pressure	Hydro-Mech	Assume 3000 psi system unless notified otherwise.
Fatigue Criteria	Struct Dyn	

Other design requirements may become evident as the design progresses. Discussions of these requirements may be found in the following text.

60.3 TRADEOFF STUDIES AND PRELIMINARY SIZING

The next step in actuating cylinder design is to study the kinematics of the item (gear, flap, etc.) to be actuated and make reduced-scale sketches of alternate actuating cylinder configurations. Each configuration should include adjacent structure, and the assumed location of associated linkage and piping. A possible configuration may be an integrated servo-actuator where space is required for the integral valving and electrical components. The preliminary sizing of each actuating cylinder configuration should be accomplished to provide tradeoff factors.

60.31 LOAD STROKE CURVE

a. Determine actuator moment arm (R) or mechanical advantage (MA) for several positions of stroke.

b. Divide the limit hinge moment (MH_L) by the product of the moment arm (R) times the number of actuators to determine the load (P).

$$P \text{ (LB) Limit} = \frac{MH_L \text{ (IN.-LB)}}{R\text{(IN.)} \times \text{No. of Actuators}} \qquad (1)$$

NOTE: Limit load is defined as the actual operating load encountered (ultimate load ÷ 1.5)

c. Plot load versus stroke curve.

d. Compare available load versus stroke with required load versus stroke to determine actuator efficiency. Determine if efficiency can be improved by revising kinematics.

60.32 BORE AND ROD SIZE

The method for determining the bore and rod size for an unbalanced, double acting actuating cylinder is shown. For a balanced, double action actuator, the rod size must be assumed in preliminary sizing.

a. The area required to extend the actuating cylinder can be calculated by dividing the maximum limit load (P_L) by the hydraulic system press (P_H)

$$A_{EXT} \text{ (IN.}^2\text{)} = \frac{P_L \text{ Ext Max (Limit) (Lb)}}{P_H \text{ (PSI)}} \qquad (2)$$

NOTE: Some allowance must be made for friction, reservoir, and minimum pump pressure. For a 3000-psi hydraulic system, 2700-psi cylinder pressure is normally used for preliminary sizing.

b. From this the bore diameter may be calculated

$$\text{Bore Dia (IN.)} = \left(\frac{A_{EXT}}{0.785}\right)^{1/2} \qquad (3)$$

c. The actual bore diameter is determined by using the diameter required for the next larger available packing (Reference Section 100).

d. Calculate the area required to retract the actuator in a similar manner.

$$A_{RET}\ (\text{IN.}^2) = \frac{P_{RET\ MAX\ (LIMIT)}\ (\text{LB})}{P\ (\text{PSI})} \qquad (4)$$

e. The actuator rod diameter can then be calculated.

$$\text{ROD DIA (IN.)} = \left(\frac{A_{EXT} - A_{RET}}{0.785}\right)^{1/2} \qquad (5)$$

f. The actual rod diameter to use will be the diameter required for the next smaller available packing. (Reference Section 100).

NOTE: It should be recognized that the preliminary rod size may not satisfy strength requirements. It is recommended that, if the rod size appears unacceptable, a preliminary column analysis be made (Reference Section 60.49).

60.33 PIPING AND PORT SIZE Actuator flow (Q) requirements can be determined when the stroke (S), area (A) and actuating time (t) are known. The following equations are for an unbalanced, double acting actuator. It should be obvious that balance area actuators simplify the calculations.

a. Flow into actuator head port when extending is:

$$Q_{EXT_{IN}}\ (\text{GPM}) = \frac{A_{EXT}(\text{In.}^2) \times S\ (\text{In.}) \times 60\ (\text{Sec/Min})}{t_{EXT}\ (\text{Sec}) \times 231\ (\text{In.}^3/\text{Gal})}$$

$$= 0.26 \times \frac{A \times S}{t} \qquad (6)$$

b. Flow out actuator rod port for known flow into head port is just the ratio of the areas times the known head port flow ($Q_{EXT_{IN}}$).

$$Q_{EXT_{OUT}} = \frac{A_{RET}}{A_{EXT}} \times Q_{EXT_{IN}} \tag{7}$$

c. If retract time differs from extend time the actuator flow when retracting is:

$$Q_{RET_{OUT}} = \frac{t_{RET}}{t_{EXT}} \times Q_{EXT_{IN}} \tag{8}$$

$$Q_{RET_{IN}} = \frac{t_{RET}}{t_{EXT}} \times Q_{EXT_{OUT}} \tag{9}$$

d. The preliminary size of pipes and ports can now be determined. Recommended flows for tubing, per MIL-H-5440, is the flow corresponding to a fluid velocity of 15 ft/sec. Table 60-2 is based on this recommendation. The recommended flow rate can be varied, depending on how critical the pressure drop is.

TABLE 60-2
TUBE SIZE vs TUBE FLOW

Non Tube O.D. (Inches)	Dash No. (O.D. In. 1/16)	Recommended Flow (GPM)
1/4	-4	1.2
5/16	-5	2.3
3/8	-6	3.5
1/2	-8	6.0
5/8	-10	10.5
3/4	-12	16.0
1	-16	29.0
1-1/4	-20	45.0
1-1/2	-24	70.0

60.34 RETRACTED LENGTH For preliminary sizing, if the actuator stroke is between 3 inches and 11 inches, it is reasonably accurate to make the following assumptions:

a. For single acting and unbalanced, double acting actuators, the retracted length is equal to two times the stroke.

b. For balanced, double acting actuators, the retracted length is equal to three times the stroke.

c. For balanced tandem double acting actuators, the retracted length is equal to four times the stroke.

60.35 <u>PRELIMINARY WEIGHT</u> The actuator weight should be estimated by using the method described below or by using the weight of an existing actuator having approximately the same diameters, length and stroke as the actuator being designed. The following method is used by the Weights Group in preliminary design to estimate the dry weight of simple hydraulic actuator assemblies. Actuators with high length to bore ratio, all steel construction, special features, etc., will have greater weight.

Actuator dry weight W_1 is estimated as follows:

$$W_1 = 0.065 + 0.131\ L_R^{0.8} \times B^{2.0} \tag{10}$$

where B = Actuator Bore Diameter (inches)

L_R = Actuator Retracted Length (inches)

NOTE: If W_1 is 2.5 pounds or more, W_1 is the estimated weight.

If W_1 is less than 2.5 pounds, estimate actuator dry weight as follows:

$$W_2 = -0.249 + 0.531\ L_R^{0.5} \times B^{1.2} \tag{11}$$

NOTE: If W_2 is less than 2.5 pounds, W_2 is the estimated weight.

If W_2 is 2.5 pounds or more, the estimated weight is the average of W_1 and W_2.

60.36 <u>PRELIMINARY DESIGN APPROVAL</u> Each actuator configuration should be evaluated and compared to the other possible configurations with respect to size, weight, and compliance with load requirements, flow demand, clearance with adjacent structure and other components, and accessibility for servicing. After each location has been evaluated by the designer, his choice made, and reasons noted, a meeting should be held with those responsible for design approval before proceeding any further with the design.

60.4 DESIGN SKETCH AND PRELIMINARY STRESS ANALYSIS

After design approval for the actuator configuration has been obtained, it is recommended that the designer take the following action:

a. Coordinate the configuration with affected sections.

b. Assure design requirements are still valid.

c. Review and incorporate any changes required by design approval meeting.

d. Check actuator stroke to ensure sufficient overtravel is provided to allow for structural deflections. These deflections may be due to actuator operating loads, load factors encountered when the actuator is static, or other items such as wing deflection.

e. If the actuator does not bottom in the extended or retracted position, the load imposed on structure may be greater than the external load (air load, etc.). Structure section should be notified when such a condition exists.

f. A full-size layout showing motion of the item to be actuated and adjacent structure should be started. The cylinder should be sketched on an overlay to this layout.

g. Prior to continuing the design sketch, the designer should review Douglas methods and layout procedure in Section 60.6 and 60.5, respectively.

h. As the design sketch proceeds, it will be necessary to conduct a preliminary stress analysis. The actuator must be designed not only to meet ultimate load requirements but also the fatigue load requirements.

60.41 <u>DUTY CYCLE</u> Knowing the expected life of the airplane is only part of the fatigue requirements. The designer must determine the number of times the actuator will be pressurized, number of load applications other than loads due to hydraulic pressure, and whether the loads are tension or compression. The loads used in determining the fatigue life are the actual working loads the actuator will encounter in day-to-day operation.

a. To determine the number of pressure or load applications per flight, plot a pressure-duty cycle for a typical flight.

b. Figure 60-2 is an example of such a plot, showing the pressure-duty cycle of a DC-9 wing flap actuator.

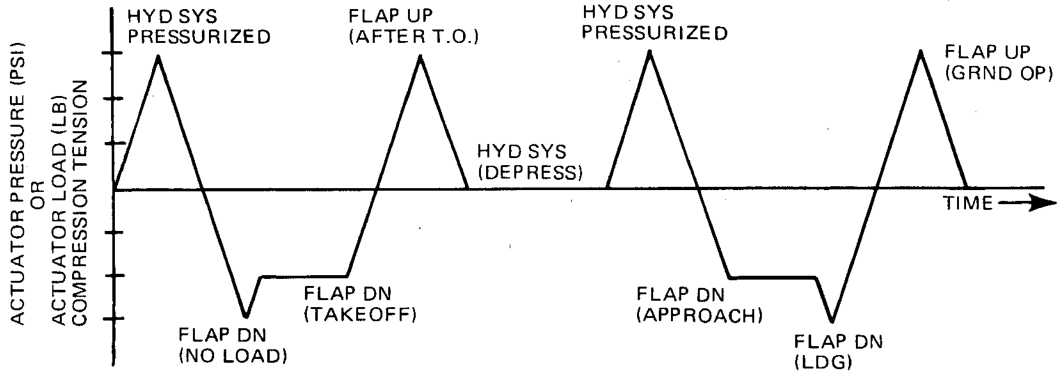


FIGURE 60-2. DC-9 WING FLAP ACTUATOR PRESSURE DUTY CYCLE/FLIGHT

c. From the pressure-duty (or load-duty) plot, the number of tension and compression load applications per flight can be determined. Knowing the life of the airplane, the fatigue life of the actuator can be calculated.

Airplane life = 40,000 flights
Tension load applications = 4/flt x 40,000 = 160,000
Compression load applications = 2/flt x 40,000 = 80,000

d. When determining the number of actuator cycles or load applications, it may be necessary to use a factor to account for miscellaneous ground operation.

e. Use constant life fatigue curves from MIL-HDBK-5 or S-N curves to determine acceptable stress levels for materials to be used.

60.42 DESIGN COMPRESSION LOAD

a. Effective area to extend x operating pressure = limit compression load. (12)

b. Effective area to extend x hydraulic system relief pressure x ultimate safety factor = ultimate compression load (13)

NOTE: Ultimate safety factor is normally 1.5.

60.43 DESIGN COMPRESSION LOAD FOR COLUMN ANALYSIS

a. Effective area to extend x operating pressure x safety factor = design compression load. (14)

NOTE: Safety factor = 1.25

b. Effective area to extend x critical pressure x safety factor = design compression load. (15)

NOTE: Safety factor = 1.15
Critical pressure = system relief pressure or the critical pressure the cylinder must withstand without failure.

c. Use whichever load is greater.

60.44 DESIGN TENSION LOAD

a. Effective area to retract x operating pressure x surge factor = limit tension load. (16)

NOTE: Surge factor is normally 1.5 ∴ operating pressure x 1.5 = proof pressure. Proof pressure should be used unless a surge factor >1.5 is required.

b. Limit tension load x 1.5 = ultimate tension load.

The hydraulic pressures, ultimate safety factors and surge factors will vary where actuator design is influenced by external loads and where additional requirements dictate.

60.45 BEARING STRESS

If the actuator piston bottoms, the bearing design load may be the applied load (air load, etc.) instead of the actuator load at operating pressure. If the actuator does not bottom, the bearing load is the critical actuator laod.

a. To determine the applied bearing pressure (fb_r), divide the actuator design limit load (P) by the projected bearing area (A).

$$fb_r = \frac{P}{A} \tag{17}$$

NOTE: The projected area (A) for journal bushings is the bushing bore x the length of bushing or width of lug supporting bushing, whichever is shorter. For spherical bearings, approximate projected bearing area is the ball diameter x outer race width or bore diameter x bore length, whichever sees the most rotation.

b. The allowable bearing stress depends on whether shock loads exist and if the joint is moving or not. To determine the allowable bearing stresses, divide the ultimate bearing strength (fb_r) of the material used by the bearing factor specified in Table 60-3 below.

Table 60-3
Bearing Factors for Joints with Infrequent Relative Motion

*Infrequent Relative Motion Under Design Loads	Shock or Vibration	Bearing Factor
None	None	1.0
Yes	None	2.0
None	Yes	2.0
Yes	Yes	2.5

*Infrequent rotation is considered to be rotation of less than 100 revolutions per hour.

$$F_{br}\text{ (allowable)} = \frac{F_{br}}{\text{Bearing Factor}} \quad (18)$$

c. Allowable bearing pressures for some materials are given in Table 60-4.

ALLOWABLE BEARING PRESSURE The values given in the chart below under "Moving Joints" or "Non-moving Joints" are for joints with loose fits. Bearing factors are from MIL-HDBK-5. For interference fits use the values given in the table under "Non-moving Joints - No Shock Load," which include a fitting factor of 1.15 on wrought and sintered materials and a casting factor of 1.25 on castings.

	MATERIAL	ULTIMATE BEARING STRENGTH PSI	ALLOWABLE ULTIMATE BEARING STRENGTH				REMARKS
			NON-MOVING JOINTS		MOVING JOINTS		
			NO SHOCK LOAD(2)	SHOCK LOAD	NO SHOCK LOAD	SHOCK LOAD	
ALUMINUM	2014 FORGINGS	90,000	78,200	45,000	45,000	36,000	
ALUMINUM	2017 SHEET OR BAR	67,000	58,200	33,500	33,500	26,800	
ALUMINUM	2024 SHEET OR BAR	84,000	73,000	42,000	42,000	33,600	
ALUMINUM	356-T6 ALUMINUM ALL ALLOY CASTING	48,000	38,400	24,000	NOT REC	NOT REC	NO FLUOROSCOPIC INSPECTION REQUIRED BY CAA FOR BEARING STRESSES
BRONZE	CENTRIFUGALLY CAST ALUMINUM BRONZE	120,000	96,000	60,000	60,000	48,000	
BRONZE	TOBIN BRONZE (HARD)	60,000	52,100	30,000	30,000	24,000	
MAGNESIUM	DOW H-T4 MAGNESIUM CASTING (HT)	36,000	28,800	18,000	NOT REC	NOT REC	NO FLUOROSCOPIC INSPECTION REQUIRED BY CAA FOR BEARING STRESSES
MAGNESIUM	DOW H-T6 MAGNESIUM CASTING (HTA)	50,000	40,000	25,000	NOT REC	NOT REC	
STEEL	STEEL (ON STEEL)	SEE MIL-HDBK-5	87% OF ULT BRG STRENGTH	50% OF ULT BRG STRENGTH	6,500*	5,000*	CONSULT DESIGN SUPERVISION
(3)	BEARIUM B-4 B-10	21,650 25,500	17,340 20,400	10,825 12,750	10,825 12,750	8,660 10,200	
	COMPO AND OILITE	7,500	6,520(4)	NOT REC	3,750(5)	NOT REC	
	XB COMP AND SUPER OILITE	30,000	26,100(6)	NOT REC	15,000(7)	NOT REC	
	NEEDLE BEARING (INNER RACE TYPE)						CONSULT DESIGN SUPERVISION

NOTES: (1) USE MANUFACTURER'S STATIC NON-BRINNEL RATING OR LIMIT LOAD RATING TIMES 1.5 FOR ULTIMATE ALLOWABLE ON BEARINGS, WITH OR WITHOUT SMALL MOTION.

(2) WHERE BEARING FACTORS ARE REQUIRED WHICH ARE GREATER THAN THE FITTING OR CASTING FACTOR QUOTED ABOVE, THE ULTIMATE BEARING STRENGTH OF THE MATERIAL SHOULD BE USED.

(3) BEARIUM IS FOR USE IN APPLICATIONS REQUIRING A PLAIN BEARING OR BUSHING COMBINED WITH A LOW COEFFICIENT OF FRICTION. ("BEARIUM" IS A TRADE NAME SIMILAR TO "OILITE" OR "COMPO" AND IS NOT TO BE CONFUSED WITH THE ELEMENT BARIUM.)

(4) MAXIMUM STATIC LIMIT STRESS – 8500 PSI

(5) MAXIMUM LIMIT STRESS UNDER ROTATION – 4000 PSI

(6) MAXIMUM STATIC LIMIT STRESS – 15,000 PSI

(7) MAXIMUM LIMIT STRESS UNDER ROTATION – 8000 PSI

REF: H.I.A.A. p. 8-8 AND MIL-B-5687.

*AT LONG BEACH USE VALUES FROM MIL-HDBK-5.

60.46 BARREL WALL THICKNESS Barrel wall thickness can be determined using Figure 60-3.

a. Enter figure at actuator burst pressure.

b. Move vertically up the pressure line until it intersects the curve having the Ft_u of the material used.

c. Move horizontally to the left from this intersection and read D/t value.

d. Knowing the barrel bore diameter (D), the wall thickness (t) may be calculated

$$t_{min_{barrel}} = \frac{\text{Barrel bore dia}}{\text{Value of D/t from Figure 60-3}} \quad (19)$$

60.47 PISTON AND PISTON ROD WALL THICKNESS The piston wall thickness can be determined by using both the L/D and Ft_u curves of Figure 60-3.

a. Enter figure at actuator burst pressure.

b. Move up the pressure line until it intersects the L/D curve.

NOTE: L/D is determined by dividing the piston rod length (L) by the rod outside diameter (D).

c. Move horizontally and find D/t.

d. The D/t value should also be determined by the method used for determining the barrel D/t.

e. The lower D/t value should be used in computing t.

$$t_{min_{rod}} = \frac{\text{Rod outside diameter}}{\text{Min value of D/t from Figure 60-3}} \quad (20)$$

60.48 BARREL DIAPHRAGM THICKNESS The barrel diaphragm thickness (t) is determined using Figure 60-4.

$$t_{min} = \frac{\text{Grind relief diameter}}{\text{Value of D/t from Figure 60-4}} \quad (21)$$

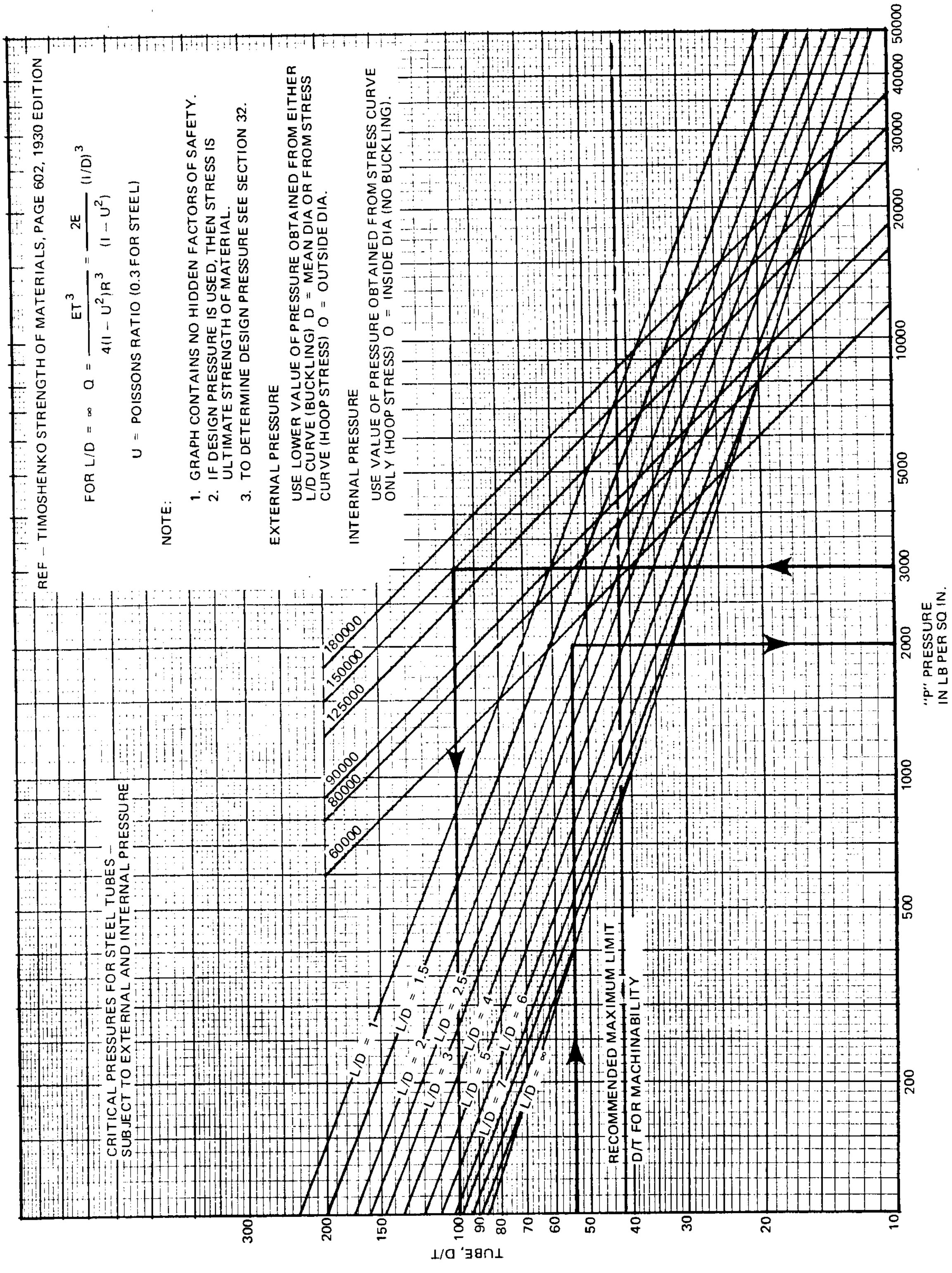


FIGURE 60-3. CRITICAL PRESSURES FOR STEEL TUBES

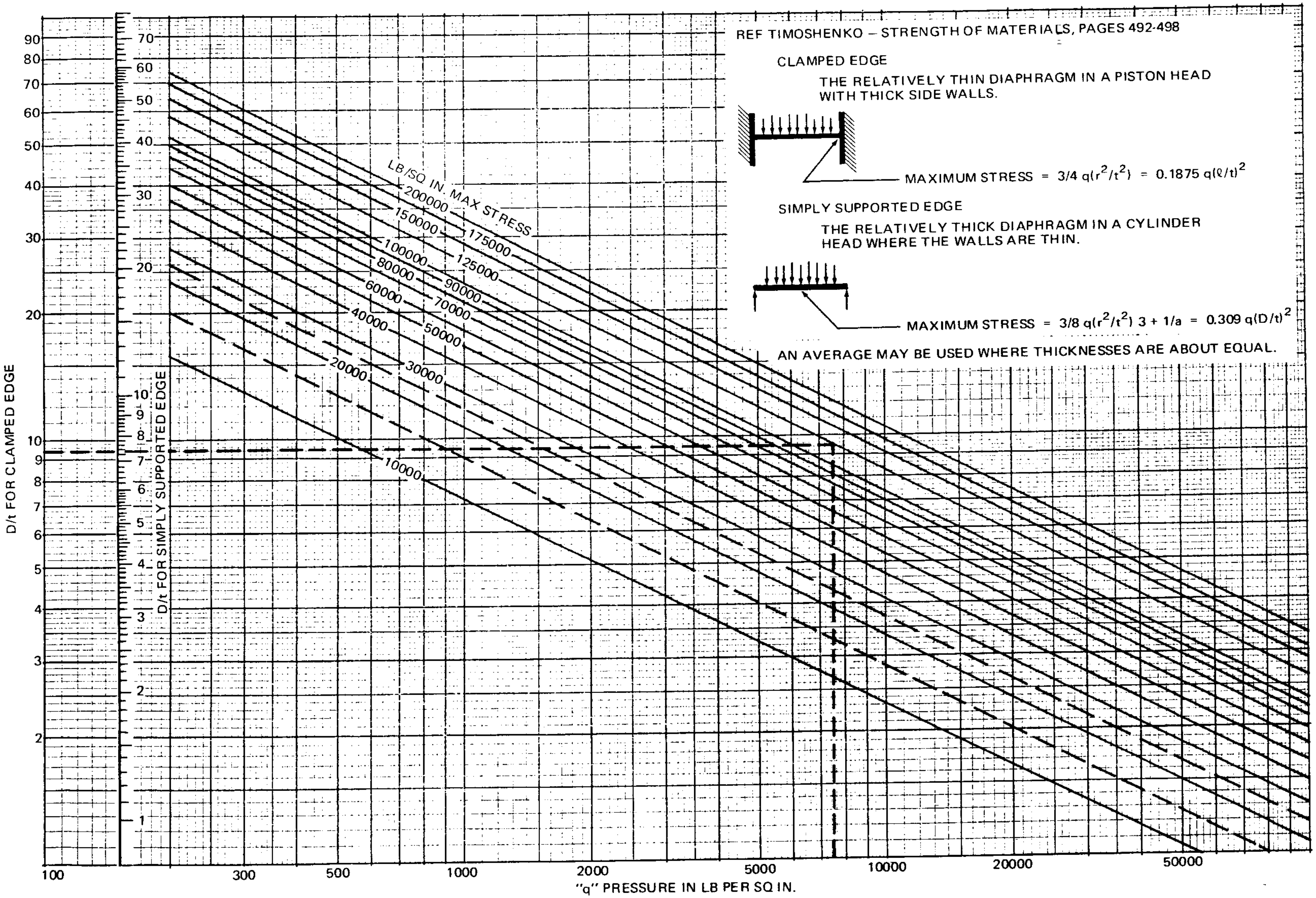
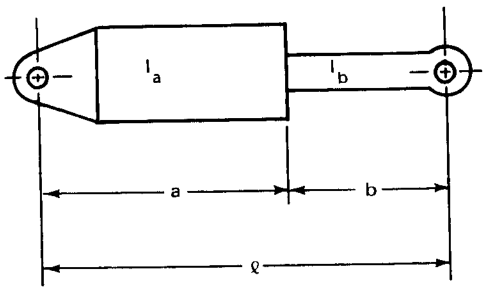


FIGURE 60-4. STRESS IN UNIFORMLY LOADED CIRCULAR FLAT PLATES

60.49 COLUMN ANALYSIS The actuator critical column load can be preliminarily checked using the stepped column curves shown in Santa Monica Report 1547. For example, the critical column load for actuators with two different basic sections can be determined as follows:



ℓ = Actuator Extended Length

a = Barrel Length

OD_a = Barrel Outside Diameter

ID_a = Barrel Inside Diameter

$I_a = \frac{\pi}{64}\left(OD_a^4 - ID_a^4\right)$

b = Rod Length Beyond Barrel

OD_b = Rod Outside Diameter

ID_b = Rod Inside Diameter

$I_b = \frac{\pi}{64}\left(OD_b^4 - ID_b^4\right)$

a. Calculate the value of I_a/I_b and a/ℓ and using these values find K from the curves of Figure 60-5.

b. Determine the critical column load and margin of safety by solving the following equations:

$$P_{CR} = K \times \frac{EI_a}{\ell^2} \tag{22}$$

$$M.S. = \left(\frac{P_{CR}}{P_{actual}} - 1\right) \times 100$$

Critical column load for columns of more that two different sections, tapered columns, etc., can be determined by referring to Figures 60-5 and 60-6 for the appropriate formula and curves.

NOTE: Using the formula and curves Figures 60-5 and 60-6 are restricted to actuators having piston rod and barrel sections with the same modulus of elasticity (E). If the stress in either section of the stepped column exceeds the proportional limit, the calculated M.S. will be negative and the piston and/or barrel wall thickness should be increased and the critical column load recalculated. Experience has shown that this method is satisfactory for preliminary analysis.

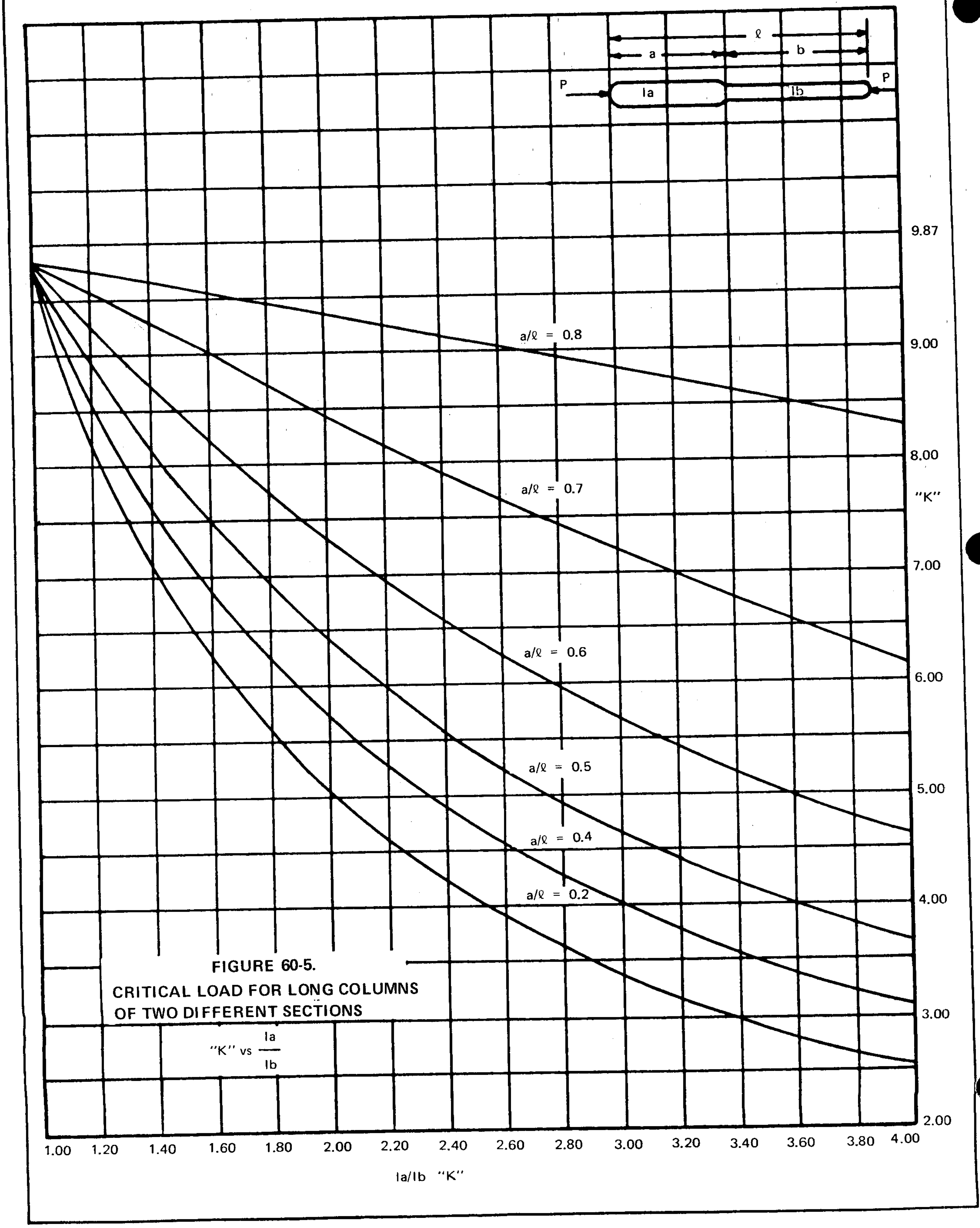


FIGURE 60-5.
CRITICAL LOAD FOR LONG COLUMNS OF TWO DIFFERENT SECTIONS

"K" vs $\frac{Ia}{Ib}$

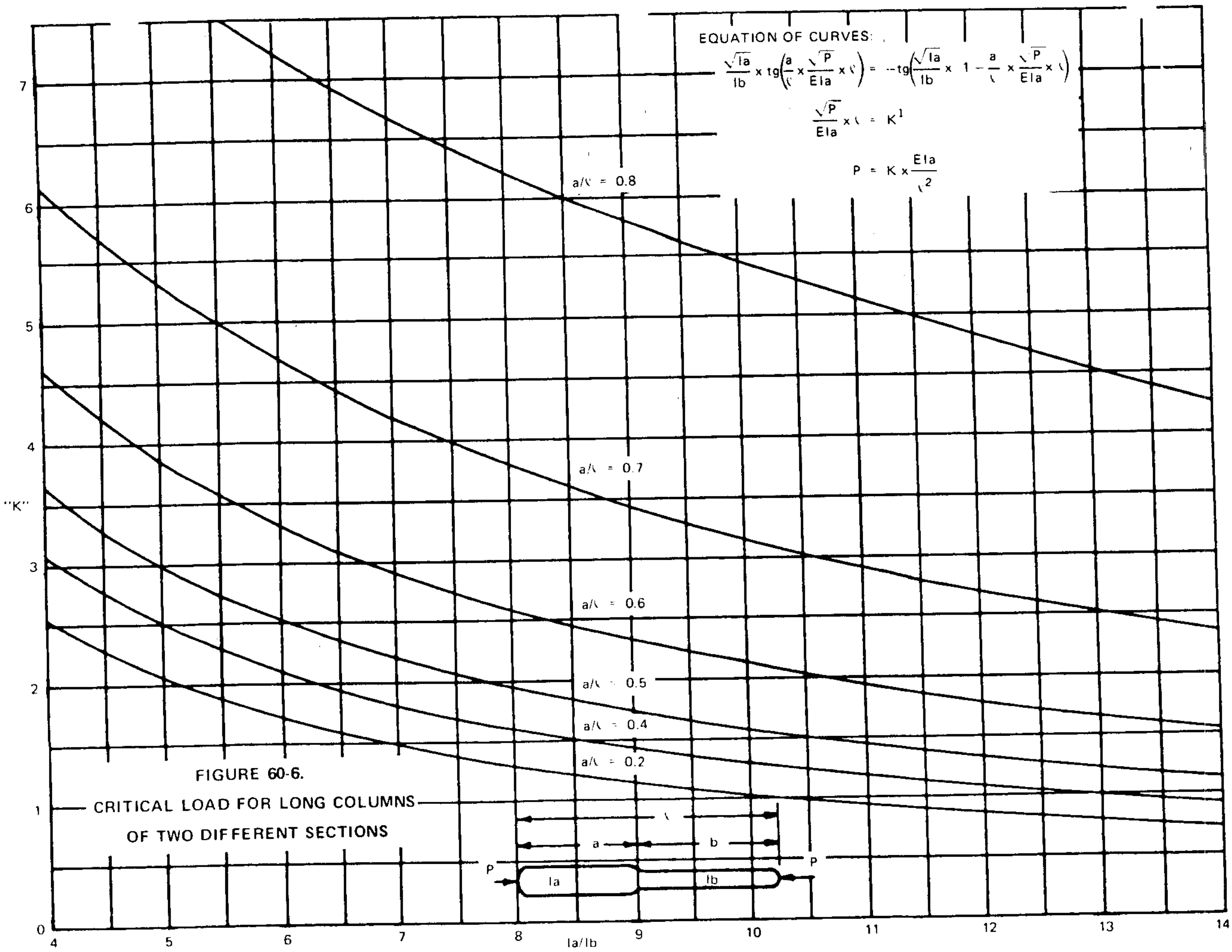


FIGURE 60-6.
CRITICAL LOAD FOR LONG COLUMNS OF TWO DIFFERENT SECTIONS

60.491 SKETCH COMPLETION AND APPROVAL

- The actuator sketch should be completed by showing the following items:
  - Static and dynamic packings
  - Grind and thread reliefs
  - Selected materials
  - Heat treat and surface protection
- When the designer is confident the sketch is complete, supervision should be notified and approval requested.

60.5 FINAL LAYOUT AND DETAIL STRESS ANALYSIS

When design approval of the sketch has been obtained, the final actuator layout can be accomplished. The recommended layout procedure is as follows:

a. The layout is to show (1) the cylinder assembly in the retracted position, (2) the motion study with clearances for the cylinder envelope and plumbing indicated. This information may all be put on the cylinder assembly layout drawing, or the final envelope may be transferred back to the motion study.

b. Actuating cylinder specification MIL-C-5503 (for military aircraft) and Section 60.6 are to be reviewed for requirements and methods.

c. Starting Point: Barrel ID Rod OD, Retracted Length, Stroke.

d. Layout barrel and rod thickness with allowance for tolerance, eccentricity and salvage. These allowances can be found in Section 60.6.

e. Layout eyebolt in center of adjustment. Show bearing attach points, rod threads and locking features. See Section 60.6.

f. Layout head side cylinder end. General proportions should be similar to previously tested designs. Show bearing attachment parts.

g. Layout piston head. Allow for overtravel in stroke if required.

h. Layout rod side cylinder end. General proportions should be similar to previously qualified designs.

i. Allow for piston snubbing. See Section 60.66.

60.51 <u>DETAILED STRESS ANALYSIS</u> Although structural dynamics approval of the actuator details must be obtained, the designer must still be responsible for the initial stress analysis of the actuator. There are times when details must be released with minimum stress analysis by the structural dynamics. The detail stress analysis by the structural dynamics will be made, but actuator details may be in various stages of fabrication by that time. If the designer has made a thorough stress analysis, there should be no major revisions required due to stress deficiencies. If a thorough stress analysis has not been made, it is possible that major revisions to the actuator details would be required. This type of revision not only increases cost but delays production and installation of the actuator and possibly airplane delivery.

The detail stress analysis computations should be neatly recorded in sequence by part. Operating and strength requirements for the actuator should also be recorded along with these computations. Calculations made during the sketch phase of the design (sizing, flow, wall thickness, etc.) should be recorded, checked and corrected if necessary.

60.511 PISTON ROD CRITICAL ANALYSIS The piston rod should be checked for fatigue stress in tension and compression at a section near the threaded end and should include bending stress due to bearing friction and eccentricities. Bearing friction coefficient usually used is 0.1. Eccentricity usually is 0.010. The actuator loads used should be the actuator limit loads.

a. For tension:

$$f_t = \frac{P}{A} + \frac{M_y}{I} \tag{23}$$

$F_t = 0.5\ F_{ty}$ or allowable stress level obtained from constant life or SN curves.

$$MS = \left(\frac{F_t}{f_t} - 1\right) \times 100$$

b. For Compression:

$$f_c = \frac{P}{A} + \frac{M_y}{I} \tag{24}$$

$$F_c = 0.5\ F_{cy}$$

$$MS = \left(\frac{Fc}{f_c} - 1\right) \times 100$$

c. The rod should be checked for fatigue stress in tension at a section through the threaded end. A fatigue factor of 6.5 is required at this section because threads are not rolled. Use equation 23 to find f_t and

$$F_t = F_{tu}$$

$$MS = \left(\frac{F_{tu}}{f_t \times 6.5} - 1\right) \times 100$$

d. Rod threads should be checked in shear with minimum eyebolt engagement.

$$f_s = \frac{P}{A} \tag{25}$$

where: P = Ult. compression load

A = $\mu\pi$PD L (approx area)

μ = 0.5 (for 60° threads)

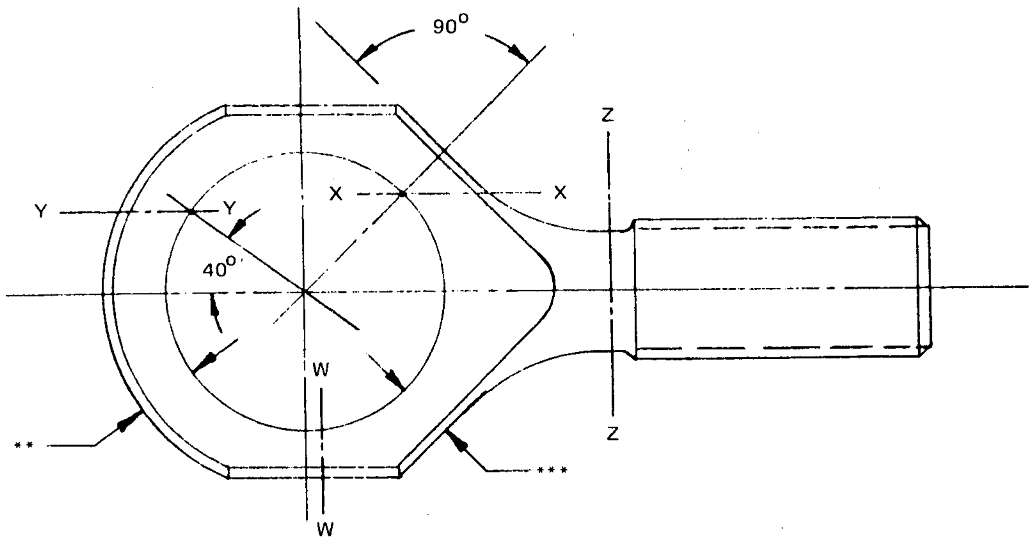
PD = Thread Pitch Diameter

L = Length of thread engagement

$$F_s = 83\%\ F_{su}$$

$$MS = \left(\frac{F_s}{f_s} - 1 \right) x\ 100$$

60.512 EYEBOLT FATIGUE STRENGTH To allow for stress concentrations, repeated loading, press fit stress, and bending loads, analyze the four indicated sections of the eyebolt for stress from maximum operating tension or compression load, as applicable. The tabulated factors modifying the governing yield allowables have been determined from cylinder life tests of 20,000 cycles duration. Longer life applications may require greater margins. See Figure 60-7.



SECTION	LOAD	APPLIED STRESS*	REFERENCE
W-W	TENSION	$f_t = P/2A_{te} \leq 0.6\ F_{ty}$	60.5121
X-X	TENSION	$f_s = P/A_{sa} \leq 0.4\ F_{sy} = 0.22\ F_{ty}$	60.5122
Y-Y	TENSION	$f_s = P/A_{se} \leq 0.6\ F_{sy} = 0.33\ F_{ty}$	60.5123
Z-Z	COMPRESSION	$f_c = P/A_z + MC/I \leq 0.5^{**}\ F_{cy}$	60.5124
	TENSION	$f_t = P/A_z + MC/I \leq 0.5^{**}\ F_{ty}$	

*REDUCE ALLOWABLE STRESS AT THE SECTION WHERE THE LUBE FITTING IS LOCATED BY 33-1/3 PERCENT TO COMPENSATE FOR STRESS CONCENTRATIONS.

**USE 0.375 FOR FATIGUE LIFE OF APPROXIMATELY 200,000 CYCLES.

***ANY RADIAL SECTION SMALLER THAN SECTION W-W MUST MEET THE TENSION STRESS REQUIREMENT OF W-W.

FIGURE 60-7. EYEBOLT ENDURANCE STRENGTH DATA

60.5121 EYEBOLT SHEAR SECTION AREA VS DIMENSIONS CONTROLLING THESE AREAS

A_{sc} = min shear-out area assuming perfect concentricity (includes both sections)

A_{se} = min shear-out area after correcting for that eccentricity which produces min shear-out area

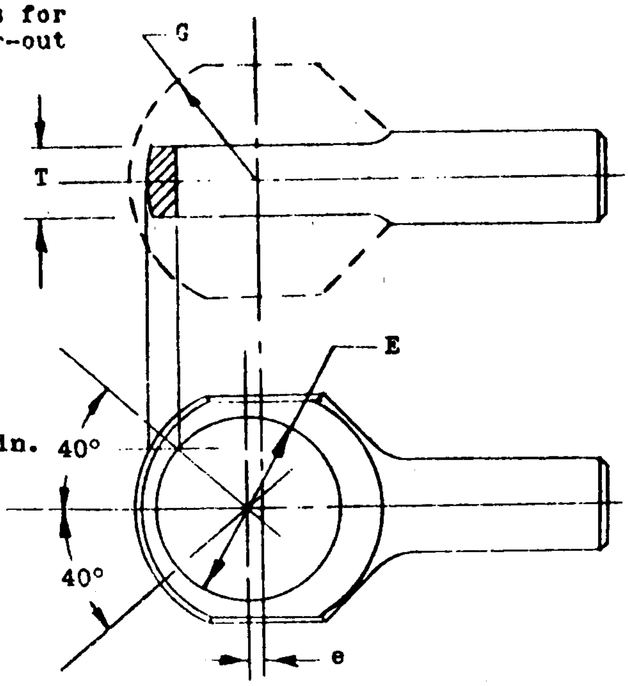
G = min spherical radius of eyebolt end

E = max bearing hole dia

T = min eyebolt eye thickness

e = relative eccentricity of centers for E and G which produces min shear-out area

$K''_s = A_{sc}/G^2$



EXAMPLE:

G = 0.750
E = 1.093
T = 0.437
e = 0.010

$\frac{E}{G} = 1.46$

$\frac{T}{G} = 0.583$

K''_s = 0.357 (from chart)

A_{sc} = 0.357 × 0.750² = 0.201 sq in.

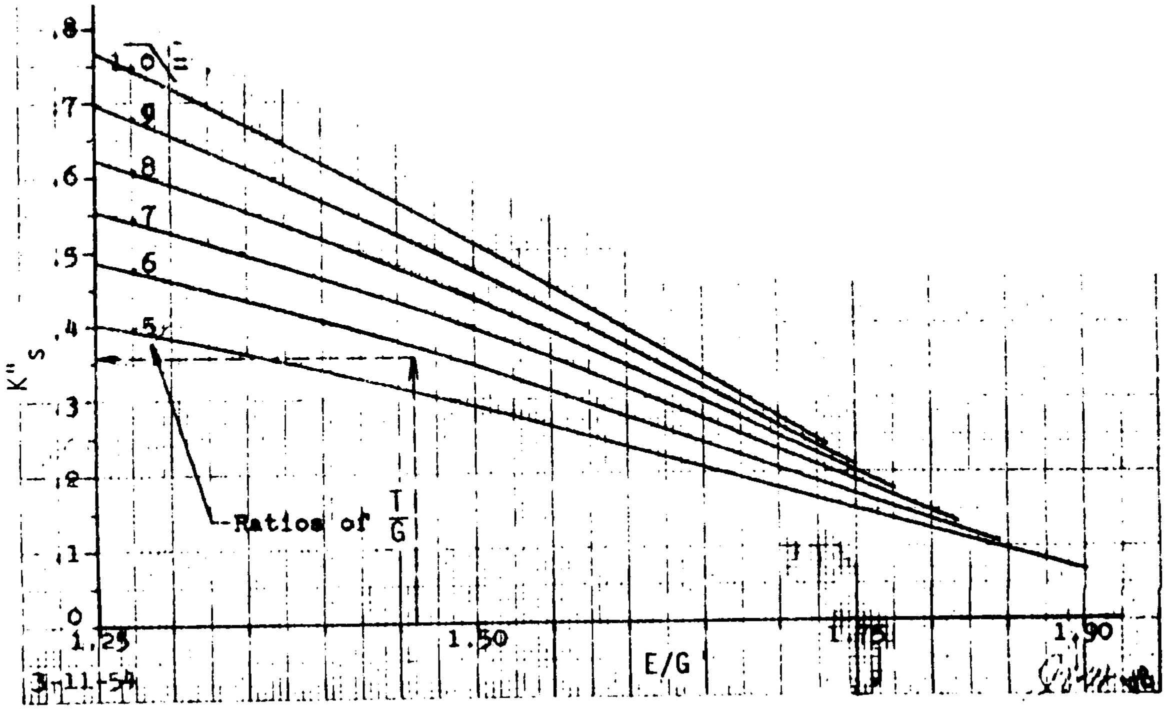
ECCENTRICITY CORRECTION:

$*A_{se} = A_{sc} \pm 2(e \times T)$
$= 0.201 - 2(.010 \times .437)$

A_{se} = 0.192 sq in.

$f_s = P/A_{se}$

*In this example the eccentricity correction is negative.

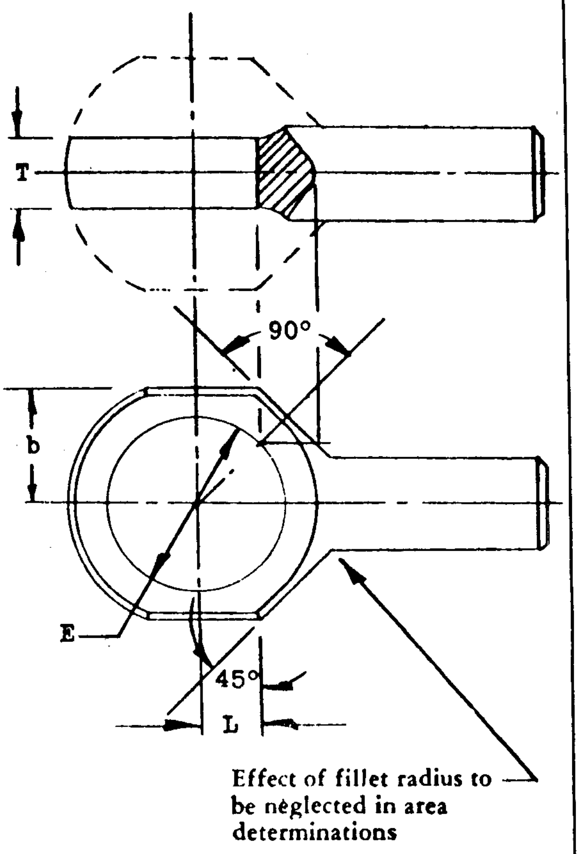


60.5122 EYEBOLT SHEAR SECTION AREAS VS DIMENSIONS CONTROLLING THESE AREAS

This chart applies to eyebolts with 45° angle as shown and with a shank dia less than 0.707E. For larger relative shank dias, the shear-out area is to be regarded as that area defined by the intersection of the shank dia (extended) with the adjoining portion of the eyebolt eye.

A_s = min shear-out area (includes both sections) uncorrected
A_{sa} = actual min shear-out area (after correcting A_s for condition of $L \neq \frac{b}{2}$)*
b = min radius of eyebolt eye O D
T = min eyebolt eye thickness
L = min axial distance from ℄ of bearing hole to the intersection of the eyebolt eye O D and the 45° conical surface

$K'_s = A_s/b^2$ E = bearing hole dia (max)



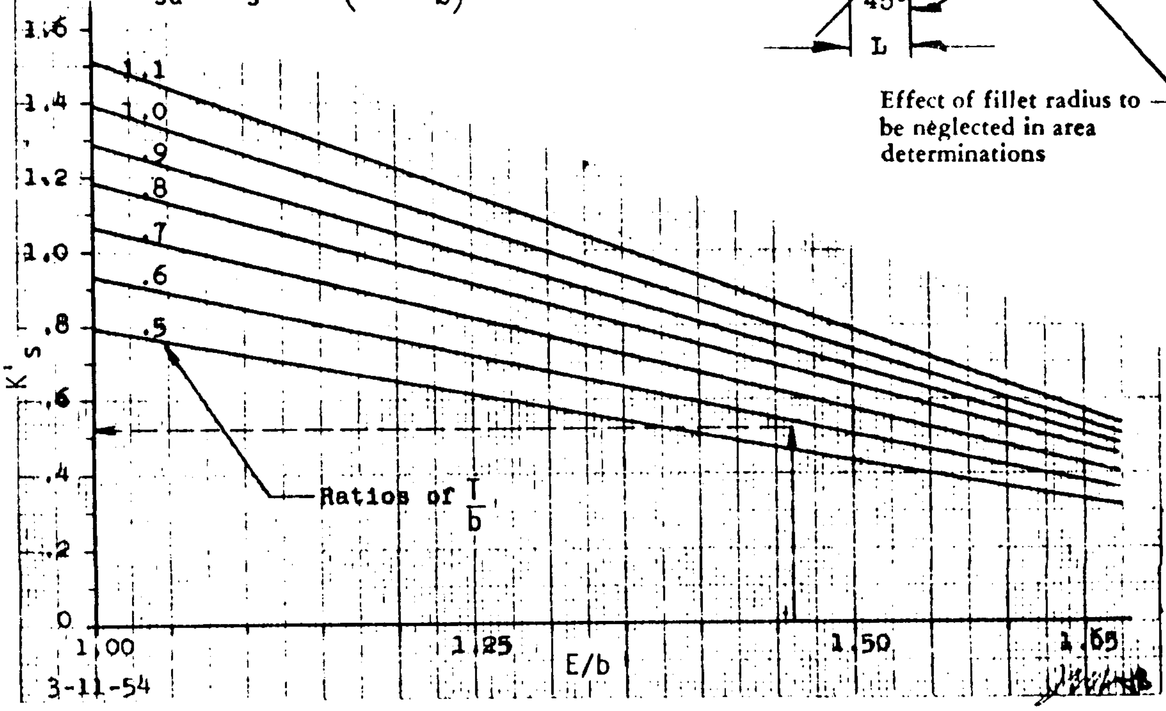
EXAMPLE:

b = 0.750 T = 0.437 $\frac{E}{b} = 1.46$
E = 1.093 L = 0.500 $\frac{T}{b} = 0.583$

$K'_s = 0.52$ (from chart)
$A_s = 0.52 \times 0.750^2 = 0.293$ sq in.
$A_{sa} = 0.293 + 2 \times 0.437 \left(.500 - \frac{.750}{2}\right)$
$= 0.401$ sq in. $f_s = P/A_{sa}$

*This chart developed for the case in which L = b/2. For cases where L ≠ b/2, correct as follows:

$$A_{sa} = A_s + 2T\left(L - \frac{b}{2}\right)$$



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60.5123 EYEBOLT TENSION SECTION AREAS VS DIMENSIONS CONTROLLING THESE AREAS

A_{tc} = min tension area/side of eye assuming perfect concentricity

A_{te} = min tension area/side of eye after correcting for that eccentricity which produces min tension area

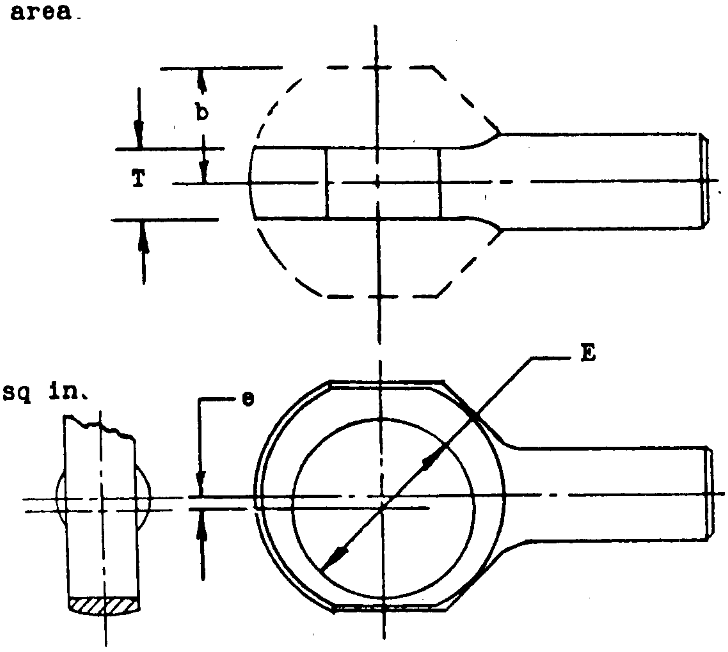
b = min radius of eyebolt eye O D

E = max bearing hole dia

T = min eyebolt eye thickness

e = relative eccentricity of centers for E and eyebolt eye O D which produces min tension area.

$K_t = A_{tc}/b^2$



EXAMPLE:

b = 0.750 $\quad \frac{E}{b} = 1.46$

E = 1.093

T = 0.437 $\quad \frac{T}{b} = 0.583$

e = 0.010

K_t = 0.148 (from chart)

A_{tc} = 0.148 × 0.750² = 0.083 sq in.

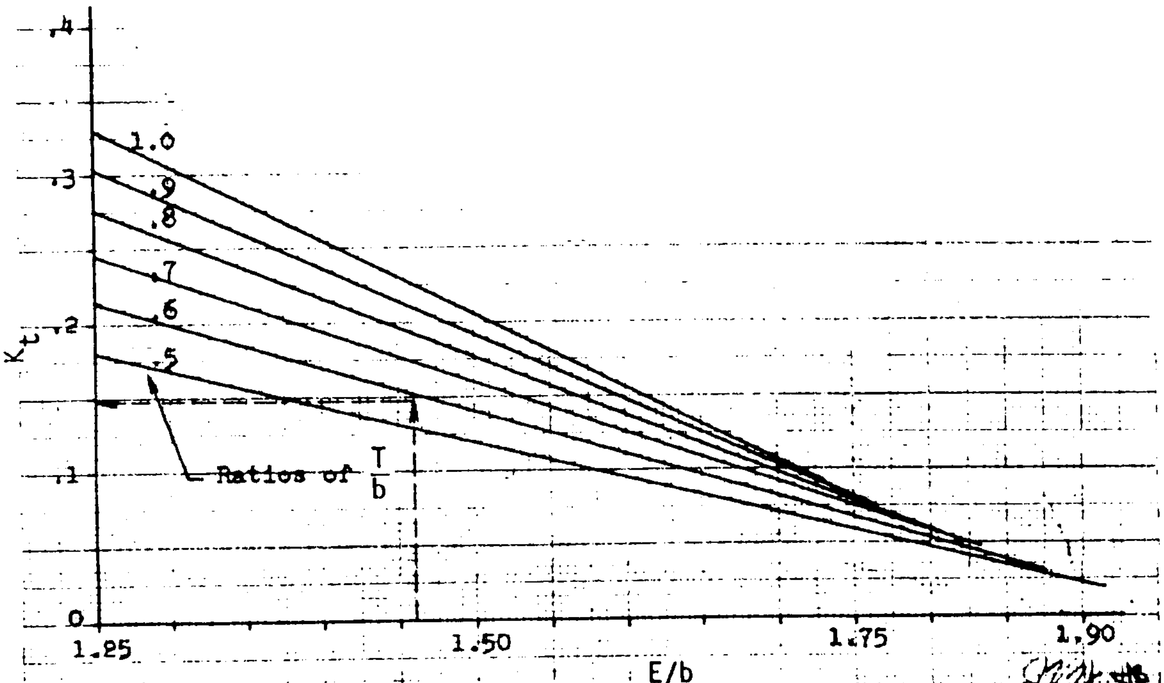
ECCENTRICITY CORRECTION:

$A_{te} = A_{tc} - (e \times T)$

$= 0.083 - (.010 \times .437)$

A_{te} = 0.079 sq in.

$f_t = \frac{P/2}{A_{te}}$



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60.5124 EYEBOLT SHANK SECTION PROPERTIES

Minimum area A_Z and minimum section modulus I/C are tabulated for the minimum shank diameters ("C" dia) listed in 41.17, and for the minimum root diameters of rolled eyebolt threads.

Rolled Thread Shank Dia per 41.172

MIN DIA	MIN A_Z	MIN I/C
.223	.03906	.001089
.280	.0616	.002155
.343	.0924	.003962
.399	.1250	.00624
.462	.1676	.00968
.520	.2124	.01380
.583	.2669	.01945
.702	.3870	.03396
.821	.5294	.0543
.946	.703	.0831
1.062	.886	.1176
1.187	1.107	.1642
1.312	1.352	.2217
1.437	1.622	.2913
1.562	1.916	.3741
1.687	2.235	.4714
1.812	2.579	.584
1.937	2.947	.713
2.062	3.339	.861
2.187	3.757	1.027

Rolled Thread Relief Dia per 41.173 & 41.174

MIN DIA	MIN A_Z	MIN I/C
.189	.02806	.000663
.244	.04676	.001426
.306	.0735	.002813
.358	.1007	.004505
.421	.1392	.00733
.476	.1780	.01059
.539	.2282	.01537
.655	.3370	.02759
.769	.4645	.04465
.893	.626	.0699
1.004	.792	.0994
1.129	1.001	.1413
1.254	1.235	.1936
1.379	1.494	.2574
1.504	1.777	.3340
1.629	2.084	.4244
1.754	2.416	.530
1.879	2.773	.651
2.004	3.154	.790
2.129	3.560	.947

Cut Thread Relief Dia per 41.175

MIN DIA	MIN A_Z	MIN I/C
.189	.02806	.000663
.243	.04638	.001409
.306	.0735	.002813
.358	.1007	.004505
.420	.1385	.00727
.475	.1772	.01052
.538	.2273	.01529
.654	.3359	.02746
.767	.462	.0443
.892	.625	.0697
1.002	.789	.0988
1.127	.998	.1405
1.252	1.231	.1927
1.377	1.489	.2563
1.502	1.772	.3327
1.627	2.079	.423
1.752	2.411	.528
1.877	2.767	.649
2.002	3.148	.788
2.127	3.553	.945

Rolled Thread Root (Minor) Diameter

THREAD	MIN DIA	MIN A_Z	MIN I/C	THREAD	MIN DIA	MIN A_Z	MIN I/C
.2500-28	.2041	.03272	.000835	1.1250-12	1.0192	.816	.1039
.3125-24	.2591	.05273	.001708	1.2500-12	1.1442	1.028	.1471
.3750-24	.3214	.0811	.003259	1.3750-12	1.2690	1.265	.2006
.4375-20	.3736	.1096	.005119	1.5000-12	1.3940	1.526	.2659
.5000-20	.4360	.1493	.00814	1.6250-12	1.5194	1.813	.3444
.5625-18	.4916	.1898	.01166	1.7500-12	1.6442	2.123	.4364
.6250-18	.5540	.2411	.01669	1.8750-12	1.7692	2.458	.544
.7500-16	.6702	.3528	.02955	2.0000-12	1.8942	2.818	.667
.8750-14	.7841	.4829	.04733	2.1250-12	2.0192	3.202	.808
1.0000-14	.9088	.649	.0737	2.2500-12	2.1442	3.611	.968

60.513 BARREL THREADS The shear stress on the barrel threads should be checked with minimum engagement of gland. Use Equation 83 to calculate the minimum shear area.

This equation accounts for the deflection of both inner and outer members due to hydraulic pressure.

$$F_s = \frac{P}{A} \tag{26}$$

where A is calculated using equations noted and P is actuator ultimate load.

$$MS = \left(\frac{F_s}{f_s} - 1\right) x\ 100$$

60.514 ROD JAM NUT TORQUE Use the following formula:

$$T = P\left(\frac{A_r}{A_r + A_b}\right)\left[\frac{1}{2\pi N} + \mu\left(\frac{R_p}{0.866} + G\right)\right] \tag{27}$$

where: P = max piston operating tension load

μ = 0.12 (coefficient of friction)

A_b = net bolt area (at pitch dia)

A_r = net rod (bolted abutment) area at P.D.

E_b = bolt Young's modulus

E_r = rod Young's modulus

N = number of threads per inch

R_p = thread pitch radius

G = mean friction radius of rod and nut bearing surfaces

Note: Specify torque on drawing ± 5 percent.

Analyze stress in bolt, nut, and rod end taking into account relative elasticity of bolt and rod. For stress analysis, use preload resulting from above torque applied with μ = 0.06.

$$\text{Maximum bolt tension} = \text{preload} + \text{applied tension load}$$

$$x\left(\frac{A_b E_b}{A_r E_r + A_b E_b}\right) \tag{28}$$

Maximum bearing on nut = preload + applied compression load

$$\times \left(\frac{A_r E_r}{A_r E_r + A_b E_b} \right) \tag{29}$$

Minimum bolt tension = preload - applied compression load

$$\times \left(\frac{A_b E_b}{A_r E_r + A_b E_b} \right) \tag{30}$$

Minimum bearing on nut = preload - applied tension load

$$\times \left(\frac{A_r E_r}{A_r E_r + A_b E_b} \right) \tag{31}$$

60.515 PISTON BEARING STRESS The bearing stress due to the piston head bearing against the gland and also against the barrel diaphragm should also be checked.

60.516 FINAL COLUMN ANALYSIS Another method of computing critical column loads is the twenty equal increment method. This method is applicable to any pin ended column.

a. Divide the column length (ℓ) into twenty equal increments.

b. Compute the area at the midpoint of each increment.

c. Compute the compressive stress (f_c) for each increment.

$$f_c = \frac{P}{A}$$

where P = design compression load (equation 14 or 15)

d. Determine the modulus of elasticity (E) or the tangent modulus (E_t), depending on whether f_c is elastic or plastic range for the material chosen. Compressive tangent modulus curves can be found in MIL-STD-5A. (Figure 60-8 is an example of compressive tangent modulus curve for 300M var steel).

e. Compute the moment of inertia at the midpoint of each increment.

f. Evaluate K/E_tI or K/EI for each increment. The value of K for each increment being obtained from the following table. The values of K hold for any pin ended column and are independent of ℓ or I.

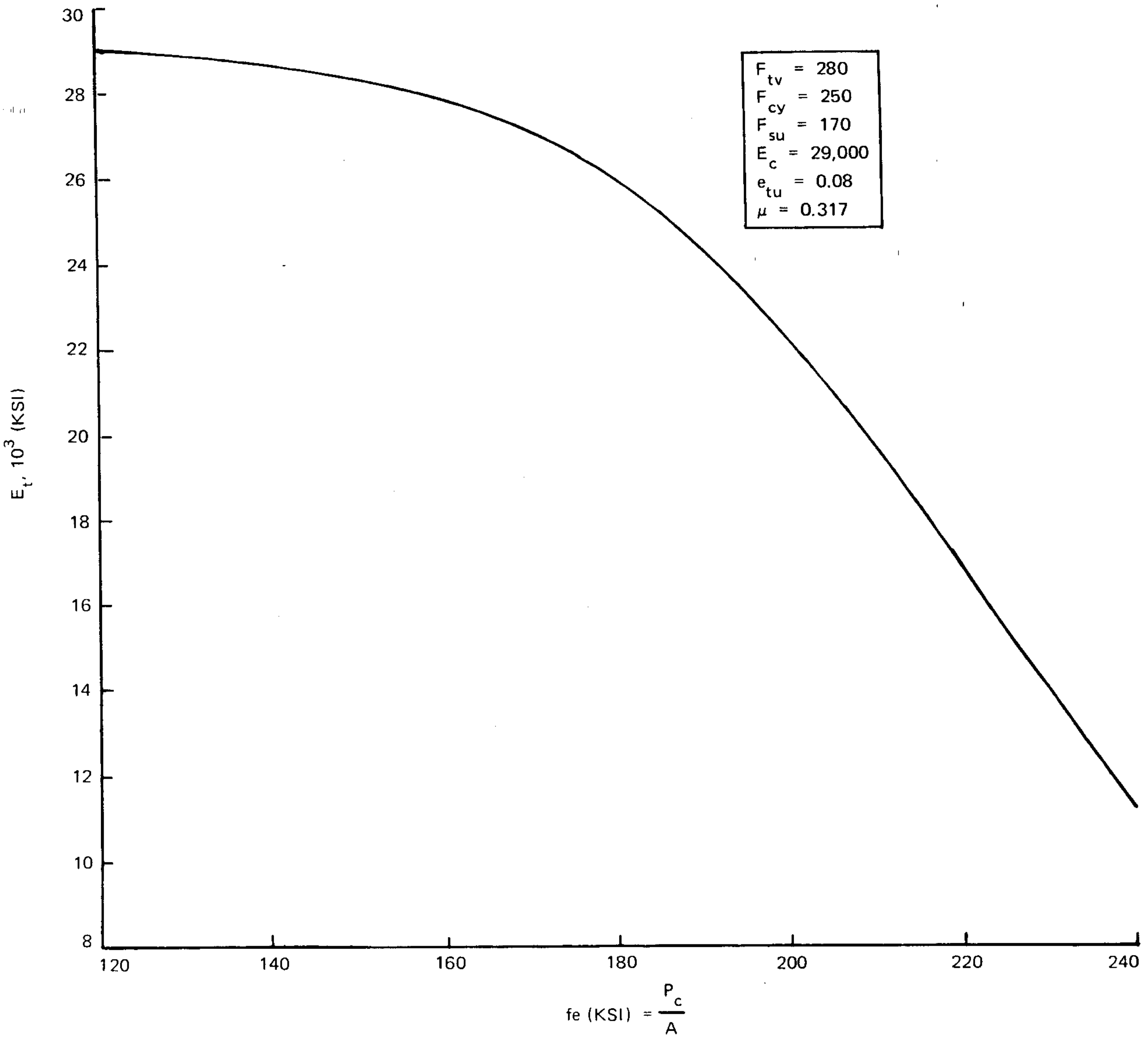


FIGURE 60-8. COMPRESSIVE TANGENT MODULES 280-300 KSI UTS 300M VAR STEEL

INCREMENT	K
1	0.020
2	0.177
3	0.476
4	0.887
5	1.370
6	1.879
7	2.362
8	2.773
9	3.072
10	3.229

INCREMENT	K
11	3.229
12	3.072
13	2.773
14	2.362
15	1.879
16	1.370
17	0.887
18	0.476
19	0.177
20	0.020

g. The critical column load may now be obtained using the equation

$$P_{CR} = \frac{(320)\quad(0.9)}{\ell^2 \sum_0^{\ell} \frac{K}{EI}} \tag{32}$$

Several other methods of column analysis are shown in Section 60.7.

60.6 DOUGLAS METHODS

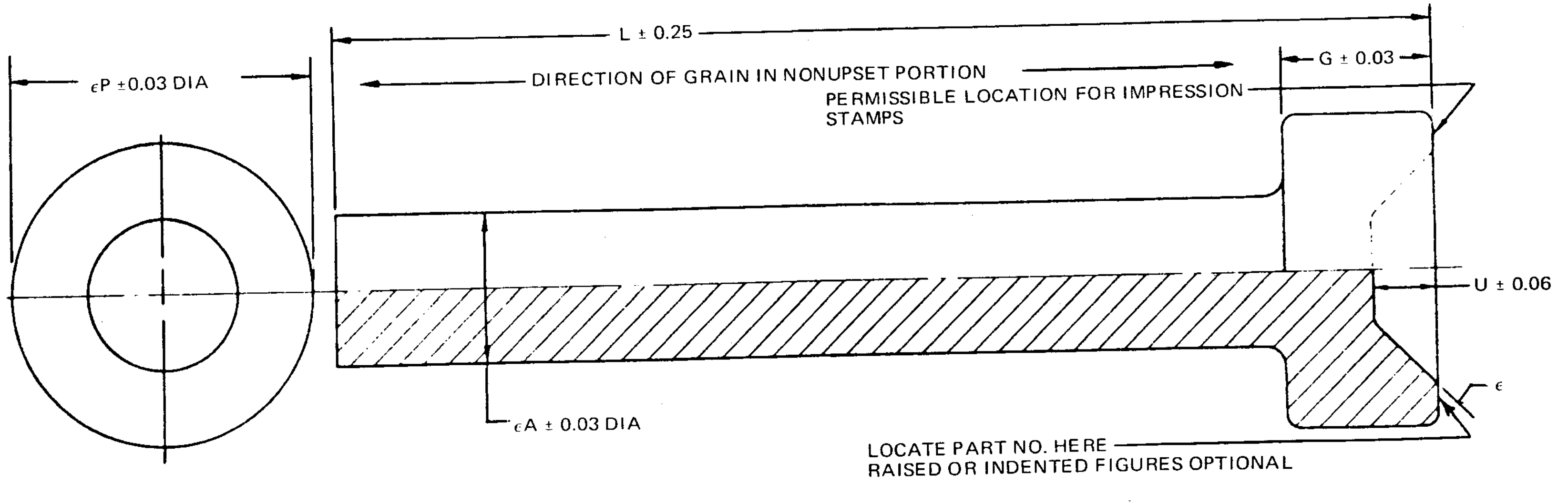
60.61 MATERIAL COMBINATIONS AND SURFACE PROTECTION Before proceeding with the actuator detail design sketch the designer should be familiar with the types of material used, and the material surface protection required.

The actuator barrels and pistons are normally fabricated from 4130, 4140, or 4340 steels. Heat treat ranges are normally 160,000 psi to 180,000 psi or 180,000-200,000 psi. Other heat treat ranges and steels such as stainless, or Hy-tuff can be used. Exact material and heat treat range selected will depend on strength required, maximum "Equivalent Round" (ER) dimension allowed, and surface treatment or material of adjacent part if there is relative motion between parts. Any surface of the barrel or piston that normally is in contact with hydraulic fluid does not require a protective coating to prevent corrosion. Any part of the barrel not in contact with hydraulic fluid is cadmium plated except for barrels made from some stainless steels and in the area of the gland static seal which is chrome plated. The cylinder gland threads are sealed with a silicone sealant (Reference DPS 2.50) to prevent corrosion. The piston and the piston rod outside diameters are chrome plated. The remaining surfaces of the piston and piston rod must have a protective coating of cadmium on exterior surfaces and zinc chromate primer or cadmium on interior surfaces exposed to atmosphere. The actuator gland, the name given the part fitting over the piston rod, is usually threaded into the barrel and used to retain the piston; is normally made from 6061 aluminum. The gland bore, the surface adjacent to the piston rod, is hard anodized. The remainder of the gland is also hard anodized but the coat is thinner than the coat on the bore. These materials and surface treatments are today's standards used in commercial aircraft actuators. Other materials and surface treatments have been used and for information pertaining to types and their applications or limitations, consult with design supervision, process or hydraulic test engineering. For a general list of process notes, see the Douglas drafting manual.

60.62 METHODS OF FABRICATION This heading is not to imply that the methods covered in the following paragraphs are all the methods or the only methods to be used in manufacturing the actuator.

The actuator barrel is normally machined from round bar stock. The barrel bosses used for connecting the hydraulic piping to the actuator are not machined as an integral part of the barrel. The barrel is first round machined. The bosses, also in rough machined state, are then welded to the barrel. The barrel is then finish machined and the ports machined in the bosses.

The piston is usually machined from an upset forging. The rod ID is then machined. The end of the piston rod is swaged to bring the material down to a diameter that will accommodate the required thread size. This type of piston rod fabrication is shown in Figures 60-9 through 60-12. Product Design and Value Engineering should be contacted when use of this method is questionable. Another type is a flash welded piston/rod assembly. This type may be more economical in some cases, depending on the design.



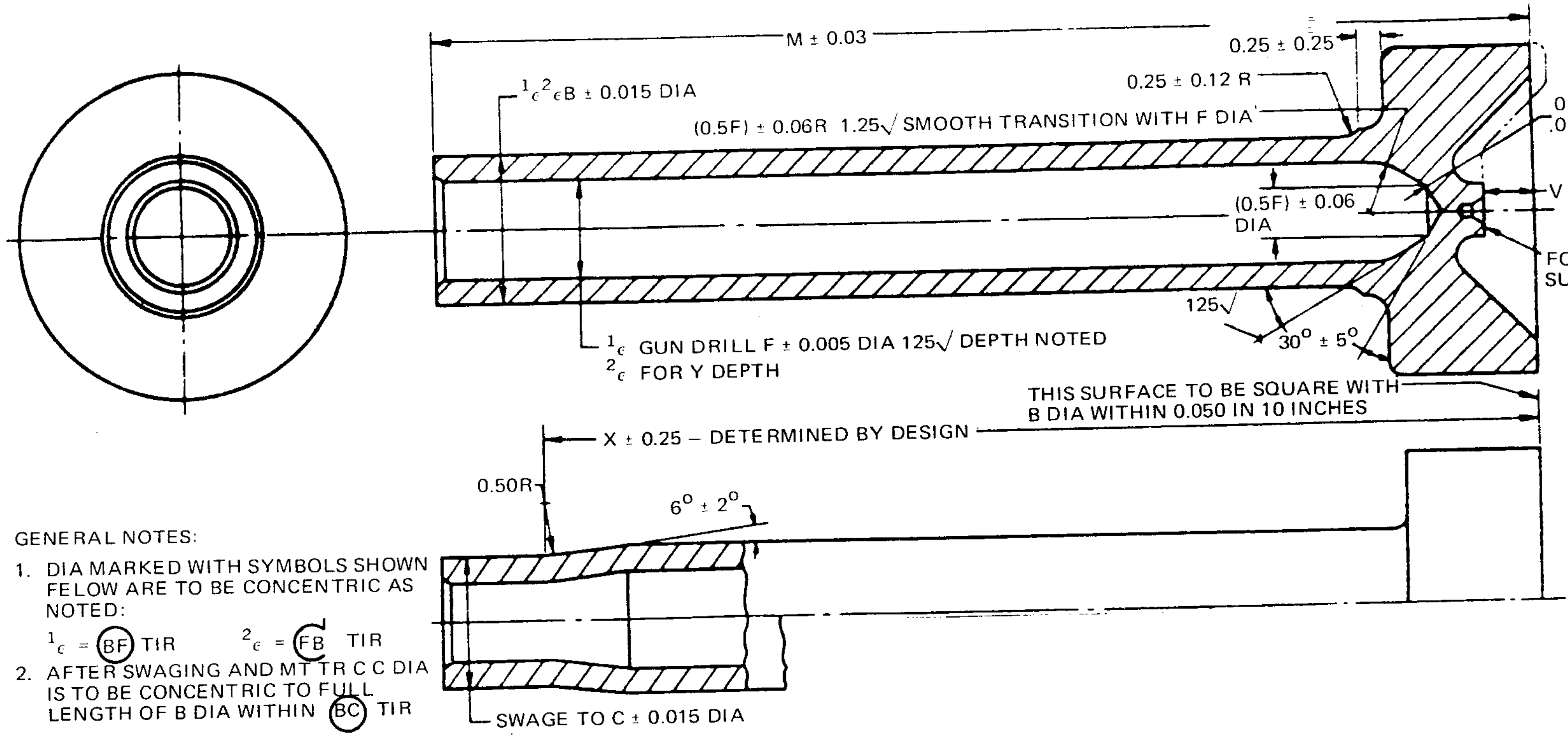
GENERAL NOTES:

1. MAY BE MADE FROM A DIA BAR.
2. OUTSIDE CORNER RADII 0 TO 0.125
3. FILLETS 0.06 ± 0.06 RADIUS
4. ALL DIA MARKED THUS ε MUST BE CONCENTRIC WITHIN 0.090 TIR
5. MAX HEAD VOLUME = $10\pi A^3/4$

PISTON FORGING

SEE PRODUCTION DESIGN FOR APPLICATIONS REQUIRING GREATER UPSET.

FIGURE 60-9. UPSET FORGE



GENERAL NOTES:

1. DIA MARKED WITH SYMBOLS SHOWN FELOW ARE TO BE CONCENTRIC AS NOTED:

   1_ϵ = BF TIR 2_ϵ = FB TIR

2. AFTER SWAGING AND MT TR C C DIA IS TO BE CONCENTRIC TO FULL LENGTH OF B DIA WITHIN BC TIR

$$Y = M - X + \frac{(B - C)\cot 6^\circ}{2}$$

BC = 2Y TAN 1/2° LIMIT TO 0.010 MIN AND 0.031 MAX

BF = LIMIT TO 0.006 MIN:

F DIA	BF	
1/4-1	0.0008	TIMES
1-1/32-2	0.0010	DEPTH
2-1/32-3 1/2	0.0012	OF F

DEPTH OF F

FB F MINUS NOMINAL THREAD SIZE	THREADS PER INCH	FB
1/32	>20	0.006
1/32	18 AND 20	0.004
>1/32	ALL	0.006

DEPTH OF Y

THREAD CLEARANCE REQUIREMENT:

(THREAD MAY BE SMALLER IF REQUIRED BY DESIGN.)

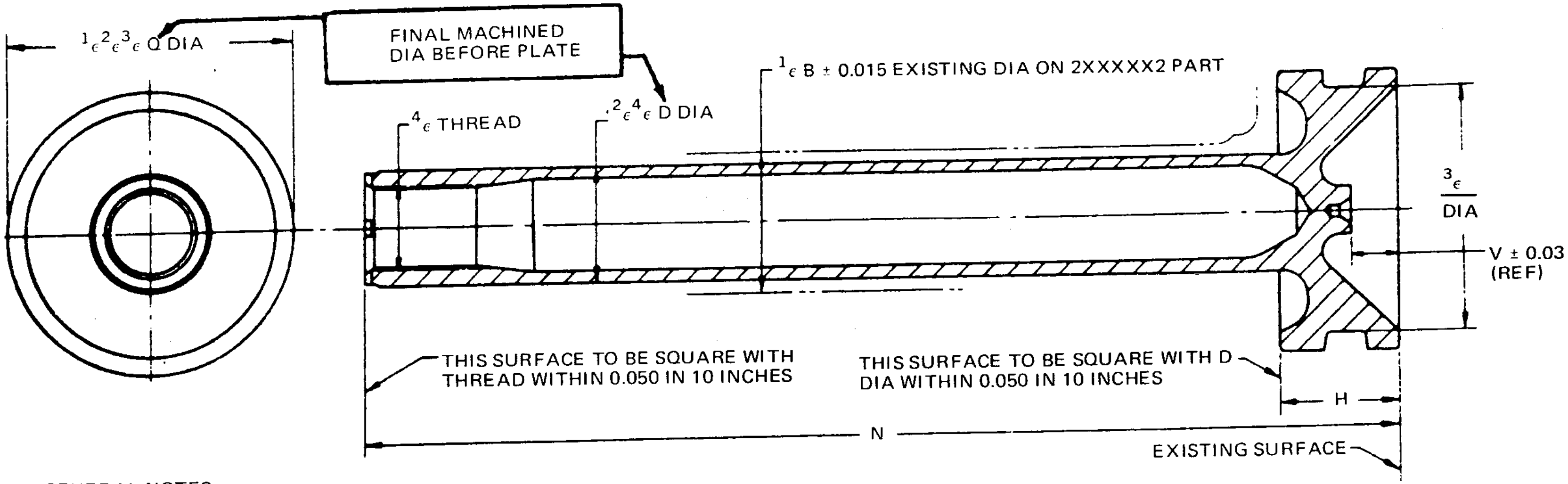
NOMINAL THREAD SIZE: (ON FINISHED PART):

= F-1/32 IN., WITH 18 OR MORE THREADS PER IN.

= F-1/16 IN., WITH LESS THAN 18 THREADS PER IN.

DO NOT SWAGE THE END WHEN LESS THAN 0.2 CU IN. WOULD BE SAVED.

FIGURE 60-10. ROUGH MACHINE OD. GUN DRILL ID CONCENTRIC TO OD. MACHINE OUTER SIDE OF HEAD. AND SWAGE ROD END.



GENERAL NOTES:

1. DIAMETERS MARKED WITH SYMBOLS SHOWN BELOW ARE TO BE CONCENTRIC AS NOTED:

$^{1}\epsilon$ = 0.003 TIR $^{2}\epsilon$ = 0.002 TIR
$^{3}\epsilon$ = 0.002 TIR $^{4}\epsilon$ = DT TIR

F ROD ID MINUS NOM THREAD SIZE	THREADS PER INCH	DT
0.03	>18	0.005
0.03	18	0.004
>0.03	ALL	0.005

FIGURE 60-11. FINAL MACHINE

DETERMINATION OF DIMENSIONS FROM FINAL DIMENSIONS:

Given:
- D = rod OD (nominal fraction)
- F = rod ID (nominal)
- S_{min} = tap drill dia (min.)
- T = thread size (nominal fraction)

Find:
- C = swaged OD (nominal)
- B = rough machined rod OD (nominal)
- A = rod OD after upset forging (nominal)

$$\boxed{C = D + .020 + BC}$$

Derivation: $C = D + \text{minus tol. on } C + \text{max TIR of } C \text{ to } D = D + .015 + .005 + BC$

$$\boxed{B = \frac{C - S_{min} + .025 + BC + FB}{1 + \Delta} + F + .020}$$

where Δ = increase is wall thickness plotted vs $\frac{B-C}{B}$ in Drafting Manual Section 23.3.
Δ must be estimated or determined by trial.

Derivation: $B = \frac{C + \text{plus tol. on } C - S_{min} + \text{max TIR of swaged ID to } T}{1 + \Delta}$ $+ F + \text{plus tol. on } F + \text{minus tol. on } B$

$$= \frac{C + .015 - S_{min} + .010 + BC + FB}{1 + \Delta} + F + .005 + .015$$

Limitations: $B \div F = 1.75$ max, $B \div C = 1.25$ max } See Production Design if these limitations must be exceeded.

$$\boxed{A \geqslant B + .125 \text{ Also } A \geqslant D + .250 \text{ (Specify A in 1/8 inch increments.)}}$$

Given: H, N. Q, V (nominal)
Find: G, M, L. P. U (nominal)

G = H + 3/8
M = N + 1/16
L = M + 9/16
P = Q + 1/4
U = V + 5/32

Consult Production Design on whether each individual application of this type piston is economical, and on whether there is a similar existing forging die which can be used.

FIGURE 60-12. NOTES

60.621 TOLERANCES AND FINISHES Control of tolerances used when dimensioning actuator details is very important. This includes concentricity, perpendicularity, parallelism, and surface roughness.

Examples of tolerances used for critical actuator dimensions are listed below.

Dimension Controlling		Tolerance
Piston Head OD, Rod OD, Barrel and Gland ID		±0.001
Piston Head Length		±0.002
Barrel Bore Length		±0.005
Gland Length (Inside Barrel)		±0.002
Concentricity -	Piston OD to Rod OD	0.002 TIR
	Gland ID to Thread PD	0.002 TIR
	Gland OD (at Static Packing) to Thread PD	0.002 TIR
	Barrel ID to Thread PD	0.002 TIR
	Rod OD to ID	0.010 TIR
Surface Finish on Barrel ID		16 max
Piston OD and ID Corner Radii		32 max
Rod OD and ID		32 max
Rod and Barrel Grind Relief		32 max
Gland ID and OD (at Static Packing)		32 max
Perpendicularity Piston Ends to Rod OD		0.002 TIR
Barrel Face and Bottom of Bore to OD		0.002 TIR
Gland Faces (that Piston and Barrel Bottom Against) to Thread PD		0.002 TIR
Parallelism between Gland ID and PD		0.001 TIR

Where: OD = Outside Diameter

ID = Inside Diameter

PD = Pitch Diameter

60.63 ENDS AND BEARINGS Actuator bearings at the structure attach points must be designed for the critical ultimate static load application and to meet bearing life requirements for dynamic loads encountered under normal operating conditions. Actuator bearings can be either journal bushings or the spherical self aligning bearing type. The journal bushing type permits more flexibility in designing the lug in which the bearing is installed. The journal bushing outside diameter for any given bore will be less than the outside diameter of a spherical self-aligning bearing of the same bore. The journal bushing also is less expensive. A universal joint should be considered where actuator motion is not in one plane or where misalignment is possible. This type of joint prevents the actuator from rolling about its axis which causes undesirable loading in the hydraulic pipes at the actuator ports. Where space does not permit a universal joint, a spherical self-aligning bearing can be used. Spherical self-aligning bearings are also used where actuator misalignment is possible.

When spherical self-aligning bearings are used, anti-rotation lugs should be machined on the bearing housing to prevent the actuator from rotating about its axis. Journal bushings are pressed into the housing with a slight interference. They can be reamed after installation if required. Spherical bearings are pressed into the housing and the outer race of the bearing roll or anvil staked over the housing. (Reference: DPS 1.33-2.)

Prior to the DC-9 bearing material was usually aluminum bronze, steel, or beryllium copper, depending on bearing stress required. Bearings of these materials require lubrication. To reduce airplane maintenance and servicing, Teflon-lined bearings have been installed in some actuators. This type of bearing does not require lubrication since it has a lower coefficient of friction (0.01 as compared to 0.1 used for metal-to-metal bearings). However, it has been found that the bearing stress of the Teflon liner will vary with the manufacturer. The dynamic bearing stress recommended by the manufacturer for Teflon-lined bearings is 18,000 psi and the ultimate static bearing stress 32,000 psi.

These values are somewhat optimistic, based on DAC experience. As a result of past experience, Teflon-lined bearings should be considered for use only in actuators or mechanisms where bearing failure will not adversely affect operation of component, system, or airplane.

It is recommended that the Teflon bearing manufacturers be contacted for exact bearing load limitations. If a Teflon-lined bearing must be used, the Teflon bearing should have the dimensions of a metal (aluminum, bronze, etc.) bearing that will take the desired loads and the bearing housing should have lubrication provisions incorporated. This is required until more experience is gained in the applications and limitations of Teflon-lined bearings. This provides for replacing the Teflon-lined bearing with a lubricated, metal bearing with minimum rework in the event the Teflon-lined bearing proves unsatisfactory.

60.631 <u>LUBRICATION</u> The preferred lube fitting is the pressed-in NAS516-1, based on fatigue experience with the MS15001 threaded-lube fittings. If a threaded fitting is necessary, special consideration must be given to assure a low stress level at the threaded region. Lube fittings must be located so as to be accessible for servicing.

60.632 <u>ROD END ADJUSTMENT</u> It should be a design objective to provide actuating cylinders that do not require rod end adjustment. Accomplishing this objective requires the combined effort of all the design sections involved. It also permits easier and faster installation and replacement. This objective has been accomplished on the DC-10 flight control actuators resulting in rod end configurations as shown in Figures 60-13 and 60-14.

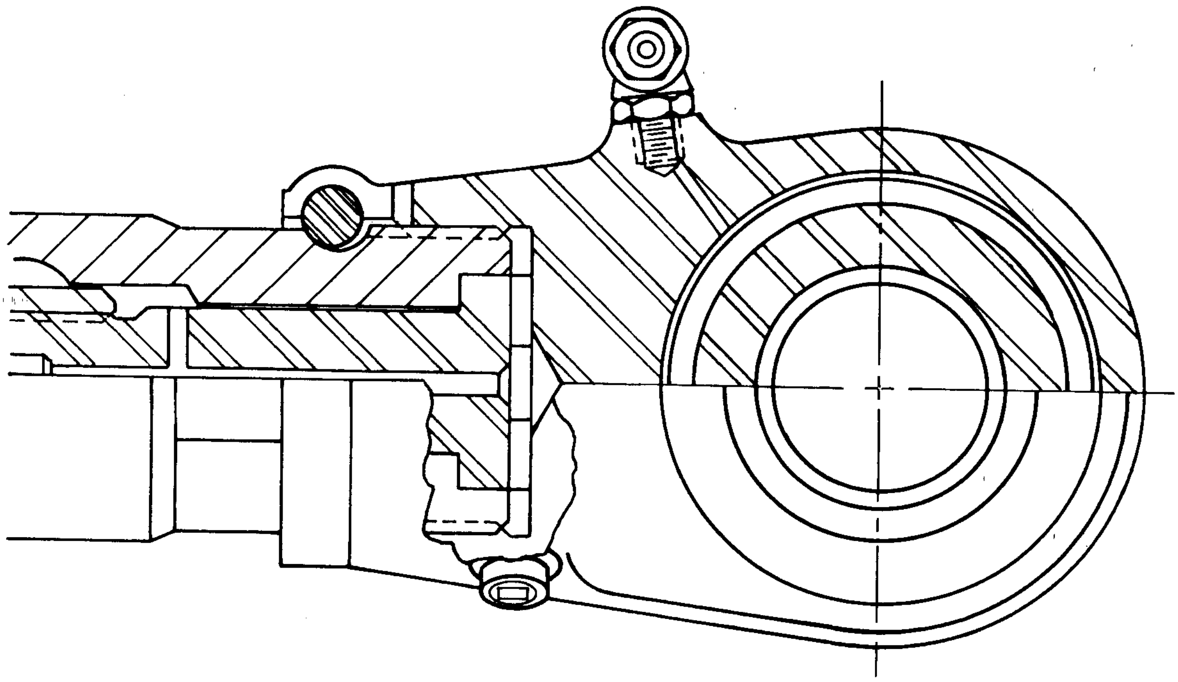


FIGURE 60-13. DC-10 OUTBOARD ELEVATOR ROD END EXTERNAL ROD THREAD

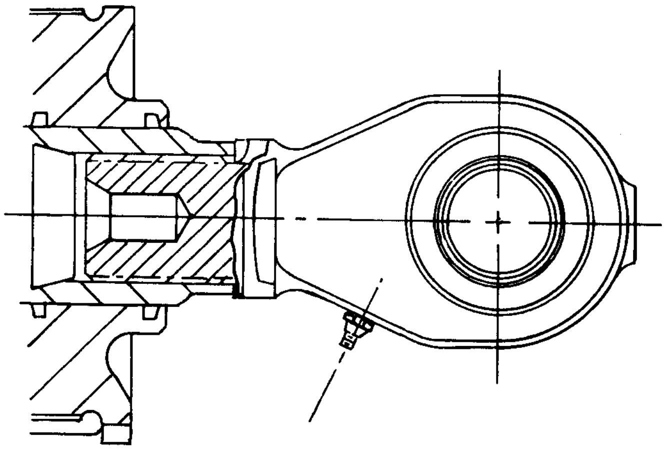


FIGURE 60-14. DC-10 SLAT ROD END INTERNAL ROD THREAD

Where adjustable rod ends are required, adequate thread engagement must be provided at the extremes of adjustment. For standard adjustable rod end locking methods refer to NAS513 and NAS559 in the Standards Manual.

60.64 GLANDS AND PACKINGS It is recommended that cylinders utilize packing glands in accordance with MIL-G-5514 unless the design requirements dictate otherwise. MIL-G-5514 dimensional data is shown in Section 100. Standard O-rings and back-up rings should also be used to minimize hardware and development costs. MS28775 is the standard 'O' ring for cylinders utilizing MIL-H-5606 fluid and NAS 1611 is the standard O-ring for Skydrol cylinders. MS28774 is the standard back-up ring and can be used with both fluids. Use MS27595 when back-up rings are installed in threaded-in members such as end glands.

There are cylinder operating conditions that standard O-rings cannot satisfy. Some of these conditions are:

1. High frequency, short-stroke cycling
2. Conditions contributing to spiral failures
3. Lo friction and long life.

Some of more commonly used packings to overcome these conditions are slipper seals, T-rings, and piston rings.

Slipper seals consist of a teflon cap and a standard O-ring. The teflon cap is the wear surface and permits the O-ring to become a static seal. Slipper seals can be installed in standard MIL-G-5514 "O"-ring glands. (1 back up). Redundant seals are used on DC-10 primary flight control actuators where the primary seal is a slipper seal and the secondary seal is an O-ring. The Douglas standard slipper part number is S-3891430 for internal gland, or S-3891431 for external gland.

The T-ring is an elastomeric ring having a T-shaped cross section. It is made to fit standard MIL-G-5514 O-ring glands. The T-ring carries its own anti-extrusion devices in the form of special plastic rings (normally scarf cut) similar to MS28774 but comparatively narrower radially to permit its accommodation on either side of the elastomeric member. Because of the interlocking of the elastomer with these back-up rings, it is resistant to spiral-type failures. T-rings are used in the following DC-10 cylinder applications:

1. Main gear retract cylinder piston and piston rod
2. Slat cylinder piston
3. Landing gear shock strut cylinders

The Douglas split piston ring (S3929694) with maraging steel expander (S4929713) is used in piston head applications where low friction and long life are required. This seal has bidirectional sealing capabilities and is installed in narrow grooves. The DC-10 primary flight control actuators and surface dampers utilize piston rings.

60.65 WRENCHING METHODS Wrench flats are required on piston rods to facilitate assembly and removal of rod ends. The wrench flats should accommodate standard wrenches. A typical wrench flat design is shown in Section 60.673.

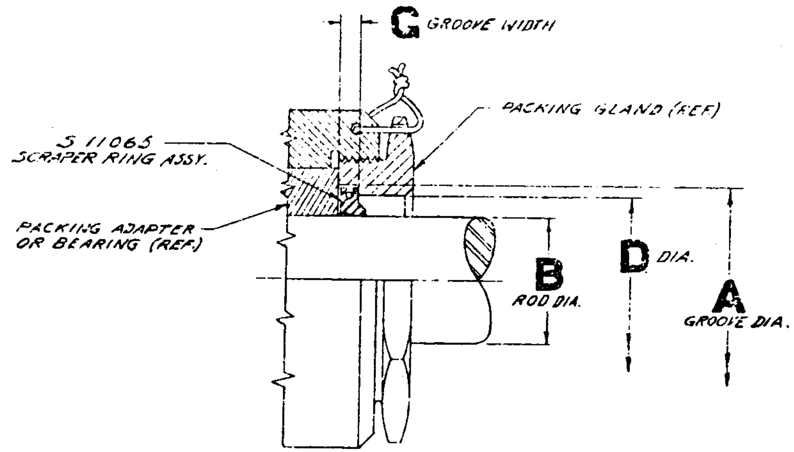
Spanner wrench slots are permissible to accomplish threaded end glands or their jam nuts. A guide for spanner wrench slot dimensions can be found in MIL-C-5503.

60.66 SCRAPERS AND WIPERS Scrapers are normally installed on all piston rods and tail rods. Teflon scrapers per W. S. Shamban & Co. drawing S-11065 or equivalent are recommended and are installed as shown in Figure 60-15. For actuators installed in a poor environment such as the landing gear, MS28776M2 (metal) scrapers are used with S4774753 washers and MS16625 retaining rings as shown in Figures 60-16, and 60-17.

Wipers are not normally used in commercial cylinders. For military requirements see MIL-C-5503.

DASH NO	B ROD	A GROOVE	G WIDTH	D
	+.000 -.002	+.010 -.000	+.010 -.000	+.010 -.000
3/8	.373	.636	.104	.523
7/16	.435	.697		.585
1	.498	.760		[illegible]
2	.560	.823		[illegible]
3	[illegible]	.885		.717
4	.685	.948		.839
5	.748	1.010		.902
6	.810	1.052		.954
7	.873	1.114		1.017
8	.935	1.177		1.079
9	.998	1.239		1.141
10	1.060	1.302		1.204
11	1.123	1.364		1.267
12	1.185	1.427		1.329
13	1.248	1.489		1.391
14	1.310	1.614		1.485
15	1.373	1.677		1.547
16	1.435	1.739		1.610
17	1.498	1.802		1.673
18	1.623	1.927		1.798
19	1.748	2.052		1.923
20	1.873	2.177		2.048
21	1.998	2.302		2.205
22	2.123	2.427		2.330
23	2.248	2.552		2.455
24	2.373	2.677		2.580
25	2.498	2.802		2.705
26	2.623	2.989	.119	2.861
27	2.748	3.114		2.986
28	2.873	3.239		3.111
29	2.997	3.364		3.236
30	3.122	3.489		3.361
31	3.247	3.614		3.486
32	3.372	3.739		3.611
33	3.497	3.864		3.736
34	3.622	3.989		3.861
35	3.747	4.114		3.986

DASH NO.	B ROD	A GROOVE	G WIDTH	D
	+.000 -.002	+.010 -.000	+.010 -.000	+.010 -.000
35	[illegible]	4.239	.119	4.111
37	3.997	4.427	.135	4.267
38	4.122	4.552		4.392
39	[illegible]	4.677		4.517
40	4.372	4.802		4.642
	+.000 -.003			
41	4.497	4.927		4.767
42	4.522	5.052		4.892
43	4.747	5.177		5.017
44	4.872	5.302		5.142
45	4.997	5.427		5.267
46	5.122	5.552		5.392
47	5.247	5.677		5.517
48	5.372	5.802		5.642
49	5.497	5.927		5.767
50	5.622	6.104	.151	5.923
51	5.747	6.234		6.049
52	5.872	6.364		6.173
53	5.997	6.489		6.299
54	6.247	6.739		6.549
55	6.497	6.989		6.799
56	6.747	7.239		7.049
57	6.997	7.489		7.299
58	7.247	7.739		7.549
59	7.497	7.989		7.799
60	7.747	8.239		8.049
61	7.997	8.489		8.299
62	8.497	8.989		8.799
63	8.997	9.489		9.299
64	9.497	9.989		9.799
65	9.997	10.489		10.299
66	10.497	10.989		10.799
67	10.997	11.489		11.299
68	11.497	11.989	.166	11.799
69	11.997	12.489		12.299
70	12.497	12.989		12.799
71	12.997	13.489		13.299



NOTES:

1. DASH NOS. 1 THRU 71 OF THIS DWG. CORRESPOND TO DASH NOS OF MS 28776.
2. THIS INSTALLATION COMPLIES WITH ARMY-NAVY AERONAUTICAL DESIGN STD. AND 10075.
3. SCRAPER RING ASSY. MAY BE INSTALLED IN ONE PIECE GLAND AS WELL AS IN TWO PART GLAND SHOWN ABOVE.
4. SCRAPER RINGS ARE AVAILABLE IN VIRGIN TFE, TFE COMPOUNDS OR NYLON PER MIL-M-20693 SHAMBAN SPEC. ZZ-53.
5. TO ORDER SPECIFY AS FOLLOWS:- EXAMPLE: S-11065-25-1
   PART NO.
   DASH NO. (2½ DIA. ROD)
   MATERIAL CODE

S-11065 | K

MATERIAL PER W.S.S. SPEC. ZZ-5:	CODE
VIRGIN TFE	1
BRONZE REINFORCED TFE - 16966	4
TURCON-WEAR RESISTANT TFE-16547	5
NYLON PER MIL-M 20693	Y

REMOVE ALL BURRS AND BREAK ALL SHARP EDGES TO — MAX.

UNLESS OTHERWISE NOTED LIMITS ON DIMENSIONS XX DECIMAL = — XXX DECIMAL = — FRACTION = ± 1/32 ANGLES = ± 1/2°

NEXT ASSY. USED ON | QTY. REQ'D.

ITEM NO. | PART NO. | NO. REQ'D | SIZE AND DESCRIPTION

BILL OF MATERIAL

THIS PRINT IS THE PROPERTY OF W. S. SHAMBAN & COMPANY AND MUST NOT BE REPRODUCED WITHOUT THE WRITTEN PERMISSION OF SAID COMPANY

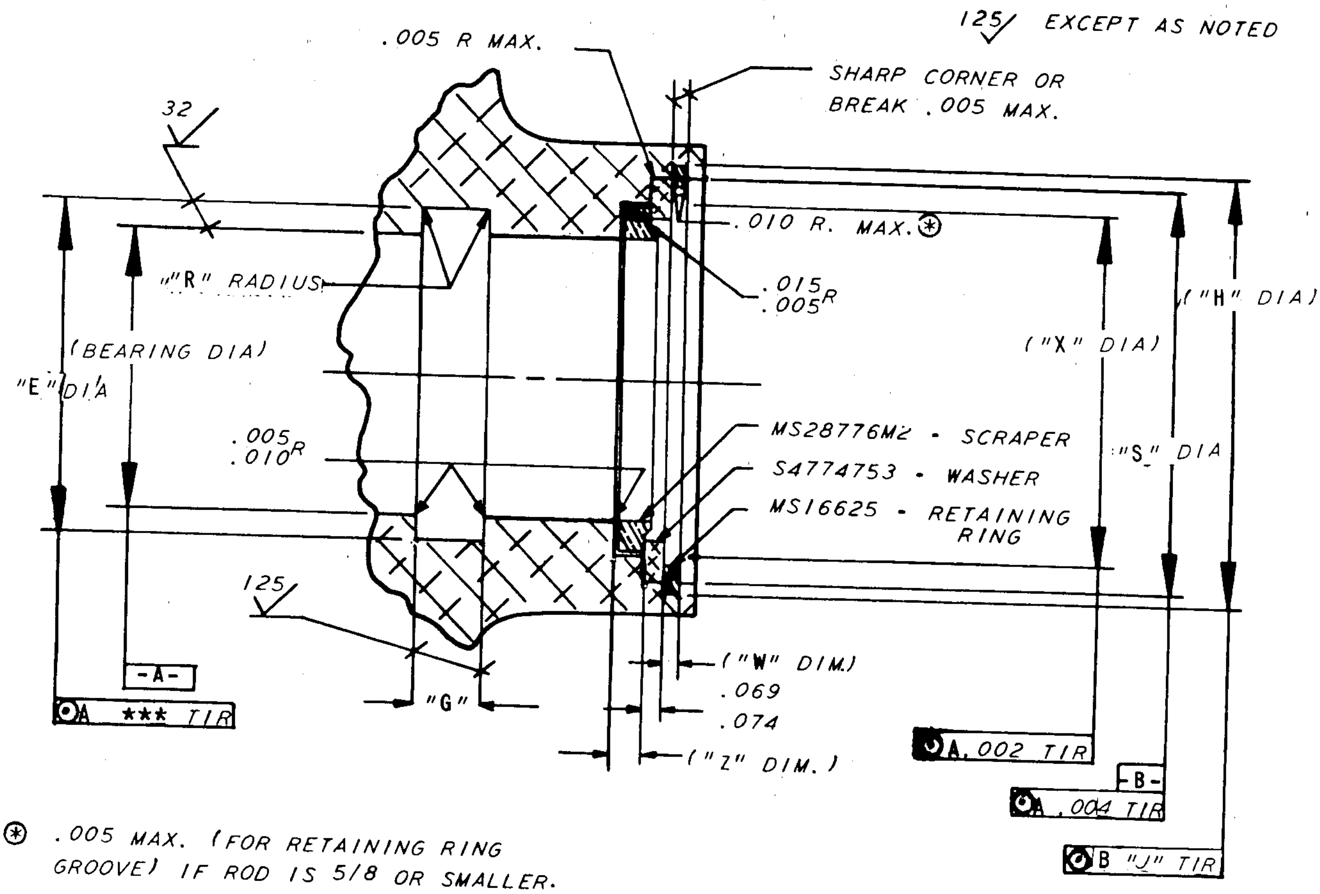
SCALE: NONE | MATERIAL: NOTED SEE NOTE 4 | CHANGE: K

SCRAPER RING INSTALLATION

W. S. SHAMBAN & CO. 11617 WEST JEFFERSON BLVD. CULVER CITY, CALIF.

DWG. NO. S-11065

FIGURE 60-15.

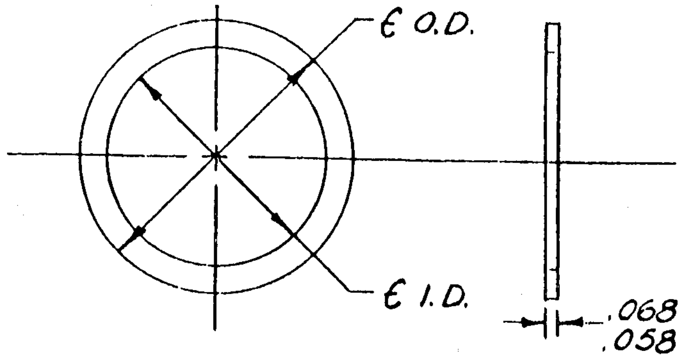


(*) .005 MAX. (FOR RETAINING RING GROOVE) IF ROD IS 5/8 OR SMALLER.

FIGURE 60-16.

SCRAPER

S4774753 WASHER



MATERIAL: 2024-T4

DIA. MARKED THUS ℄ TO BE CONCENTRIC WITHIN .002 FULL INDICATOR

DRAWING NO.	DASH NO.	OD +.000 -.010	ID +.010 -.000	DRAWING NO.	DASH NO.	OD +.000 -.010	ID +.010 -.000
S4774753	-1	.863	.657	S4774753	-19	2.305	1.928
	-2	.898	.720		-20	2.493	2.053
	-3	.997	.782		-21	2.618	2.178
	-4	1.018	.844		-22	2.806	2.303
	-5	1.183	.907		-23	2.993	2.428
	-6	1.245	.959		-24	3.055	2.553
	-7	1.307	1.022		-25	3.243	2.678
	-8	1.370	1.084		-26	3.339	2.834
	-9	1.433	1.146		-27	3.462	2.959
	-10	1.451	1.209		-28	3.618	3.084
	-11	1.556	1.272		-29	3.743	3.209
	-12	1.682	1.334		-30	3.931	3.334
	-13	1.744	1.396		-31	4.118	3.459
	-14	1.806	1.490		-32	4.243	3.584
	-15	1.869	1.552		-33	4.323	3.709
	-16	1.932	1.615		-34	4.493	3.834
	-17	1.994	1.678		-35	4.618	3.959
S4774753	-18	2.181	1.803	S4774753	-36	4.743	4.084

FIGURE 60-17.

60.67 RETENTION METHODS Piston rods shall be positively locked to both the piston head and the rod end to prevent loosening under load and vibration. Jam nuts and lockwire are not considered positive locking means. Lockwire is used to prevent rotation of end caps, locknuts and adjustment nuts. An example is shown in Figure 60-18.

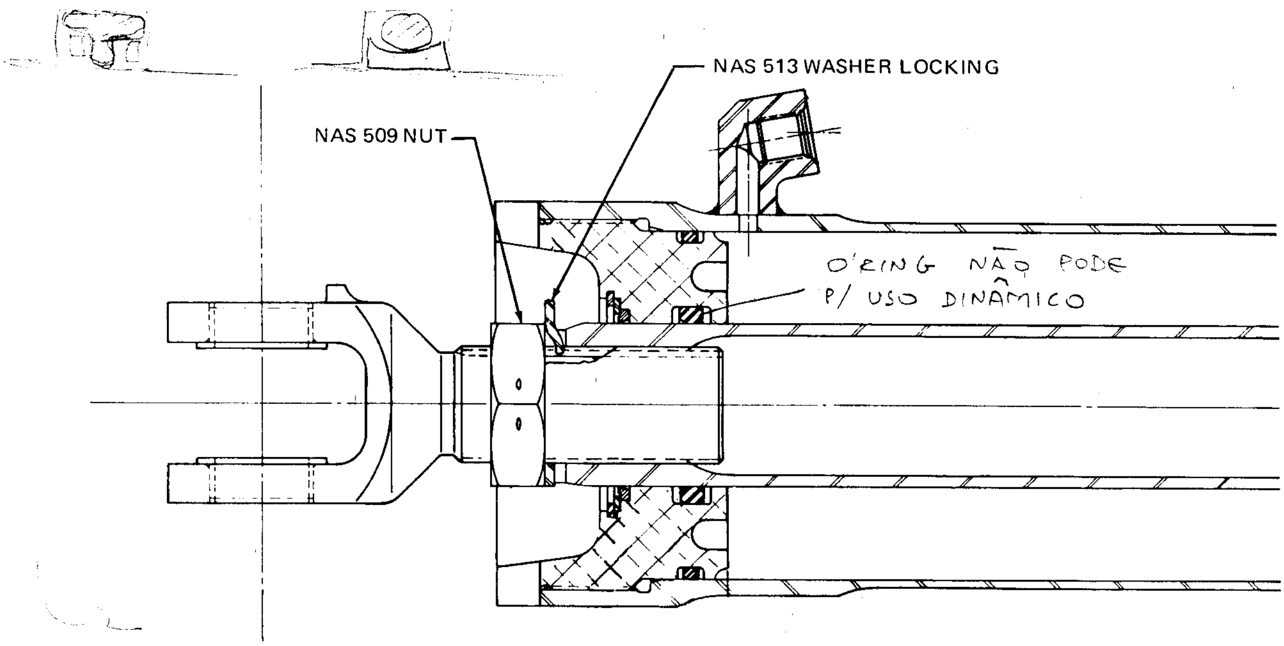


FIGURE 60-18. NOSE GEAR RETRACT CYLINDER

60.68 ACTUATOR DASHPOTS Actuator dashpots or snubbers are used to decrease the piston velocity and thereby reduce the impact of the piston as it bottoms in the actuator. The desired dashpot action is achieved by decreasing the flow of fluid from the actuator. This is accomplished by forcing the fluid through a variable orifice during the last part of the actuator stroke. The fluid through an orifice dissipates energy and produces a decelerating force proportional to the fluid pressure produced. In some servoed systems, the control valve may function as a variable orifice, thereby eliminating the requirement for a dashpot in the actuator. There are several ways the variable orifice may be designed into the actuator. Figures 60-19a and 60-19b show variable orifice type dashpots. Figure 60-23c is a fixed orifice type dashpot. Figure 60-19c dashpot functions in two directions, using a different fixed orifice in each direction. The last part of the actuator retract stroke is dashpotted through one orifice and the first part of the extend stroke is dashpotted through a different orifice.

The actuator dashpot should be calculated to determine dashpot flow requirements and pressures developed. If dashpot pressures developed exceed 3000 psi the actuator strength requirements may be affected. The actuator packings should not be subjected to pressures in excess of 3000 psi. The design of the dashpot, therefore, will be partially dictated by the pressure developed.

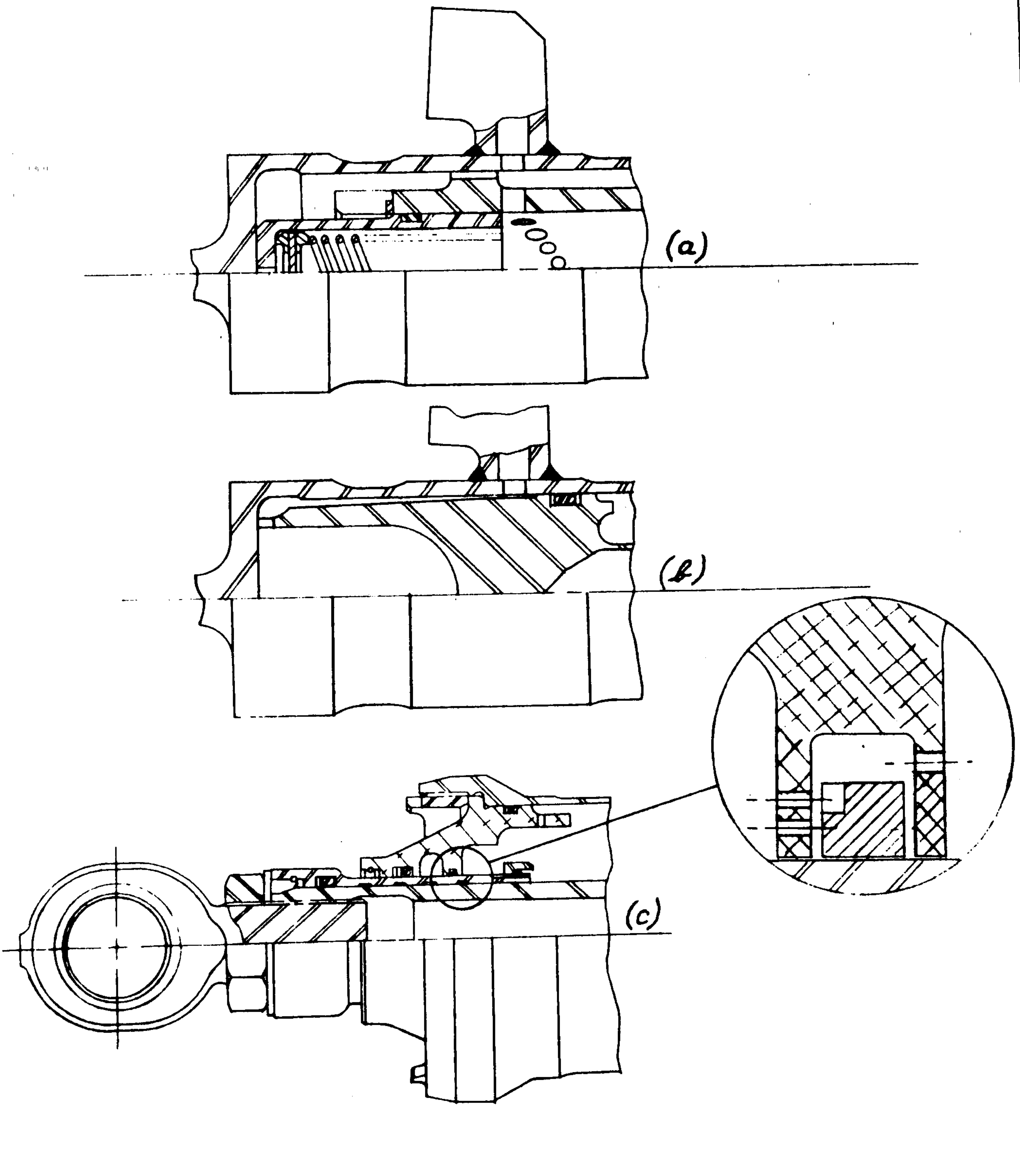
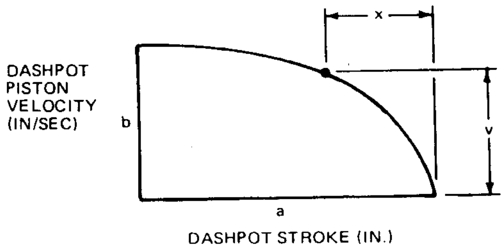


FIGURE 60-19. ACTUATOR DASHPOTS

The dashpot is calculated by assuming a dashpot stroke. The assumed dashpot stroke should then be divided in increments. From the load stroke curve of the actuator determine the average load for each increment of dashpot stroke. Next determine the average actuator piston velocity. This can be determined by dividing the actuator stroke by the actuator operating time required. The required dashpot velocity for each increment of dashpot stroke can be assumed (if assumed, the velocity should be decreased for each increment of stroke) or it may be calculated by assuming the dashpot velocity vs dashpot stroke curve is that of a parabola.



$$\frac{v^2}{b^2} = \frac{x}{a}$$

$$v = \left(\frac{b^2}{a} \cdot x\right)^{1/2} \qquad (33)$$

where: a = total dashpot stroke

b = average actuator piston velocity at start of dashpot stroke

x = dashpot stroke remaining from mid-point of each increment of stroke

By substituting for x (the amount of dashpot stroke remaining from the mid-point of each increment of dashpot stroke) the dashpot velocity at that point can be calculated.

Determine the dashpot area (A_{DP}).

Knowing the dashpot area and velocity the average dashpot flow (Q) can be calculated for each increment of dashpot stroke.

$$A\ (\text{GPM}) = 0.26\ A_{DP}\ (\text{in.}^2) \times V_{DP}\ (\text{in./sec}) \qquad (34)$$

Now calculate the dashpot pressure (ΔP_{DP}) required to develop the load (P_{DP}) (from the actuator load stroke curve) for each increment of stroke.

$$\Delta P_{DP}\ (\text{psi}) = \frac{P_{DP}\ (\text{lb})}{A_{DP}\ (\text{in.}^2)} \tag{35}$$

Since orifice diameters are a function Q and ΔP the orifice diameter (d) required for each increment of dashpot stroke can now be calculated.

$$d = \left[\frac{0.0346 \times Q_{DP}}{0.6 \times P_{DP}^{1/2}}\right]^{1/2}$$

For Skydrol and orifice coefficient of discharge = 0.6

Simplifies to:

$$d = \left[0.0576\ \frac{Q_{DP}}{P_{DP}^{1/2}}\right]^{1/2} \tag{36}$$

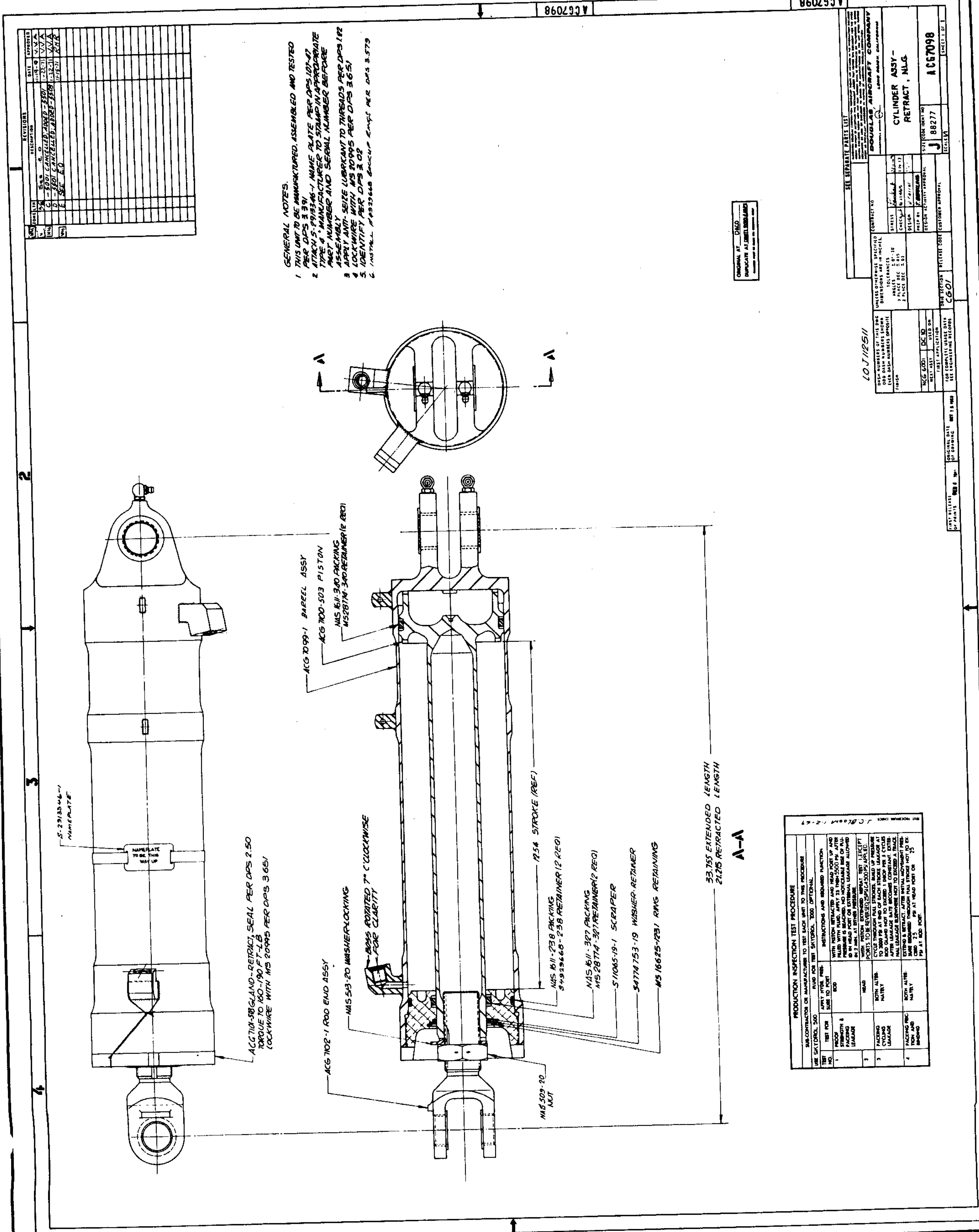
These diameters may be used as the equivalent diameters where orifices of other shapes are used.

This is a greatly simplified description of dashpot calculating. There are many other factors that must be considered when calculating dashpot pressures and flows. Some other factors to consider are:

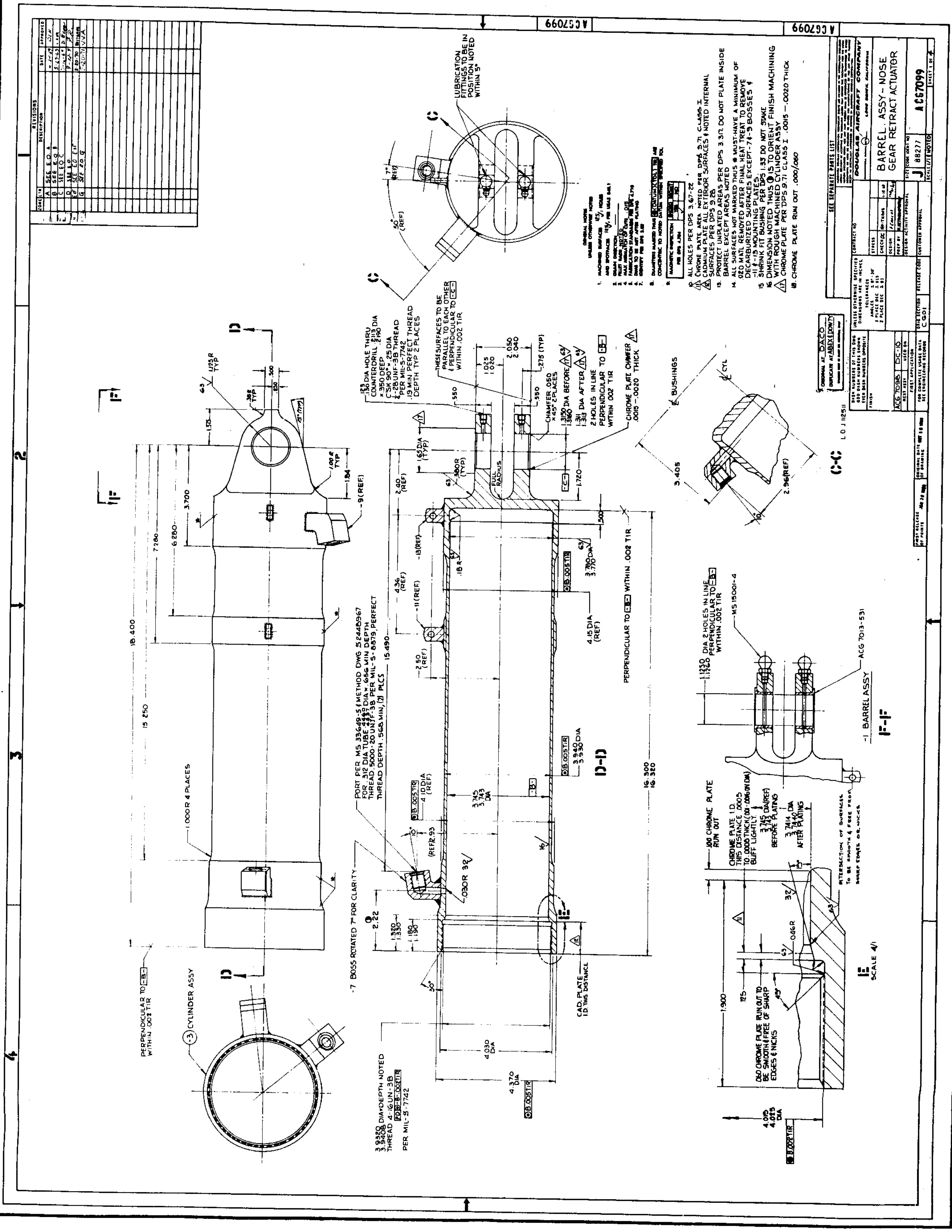
1. Effects of hydraulic system pressure on actuator. This is in addition to external loads? Is it opposing external load? How does it affect dashpot?

2. Effects of pressure drop in hydraulic system (Pressure side and return side ΔP) and effects on dashpot.

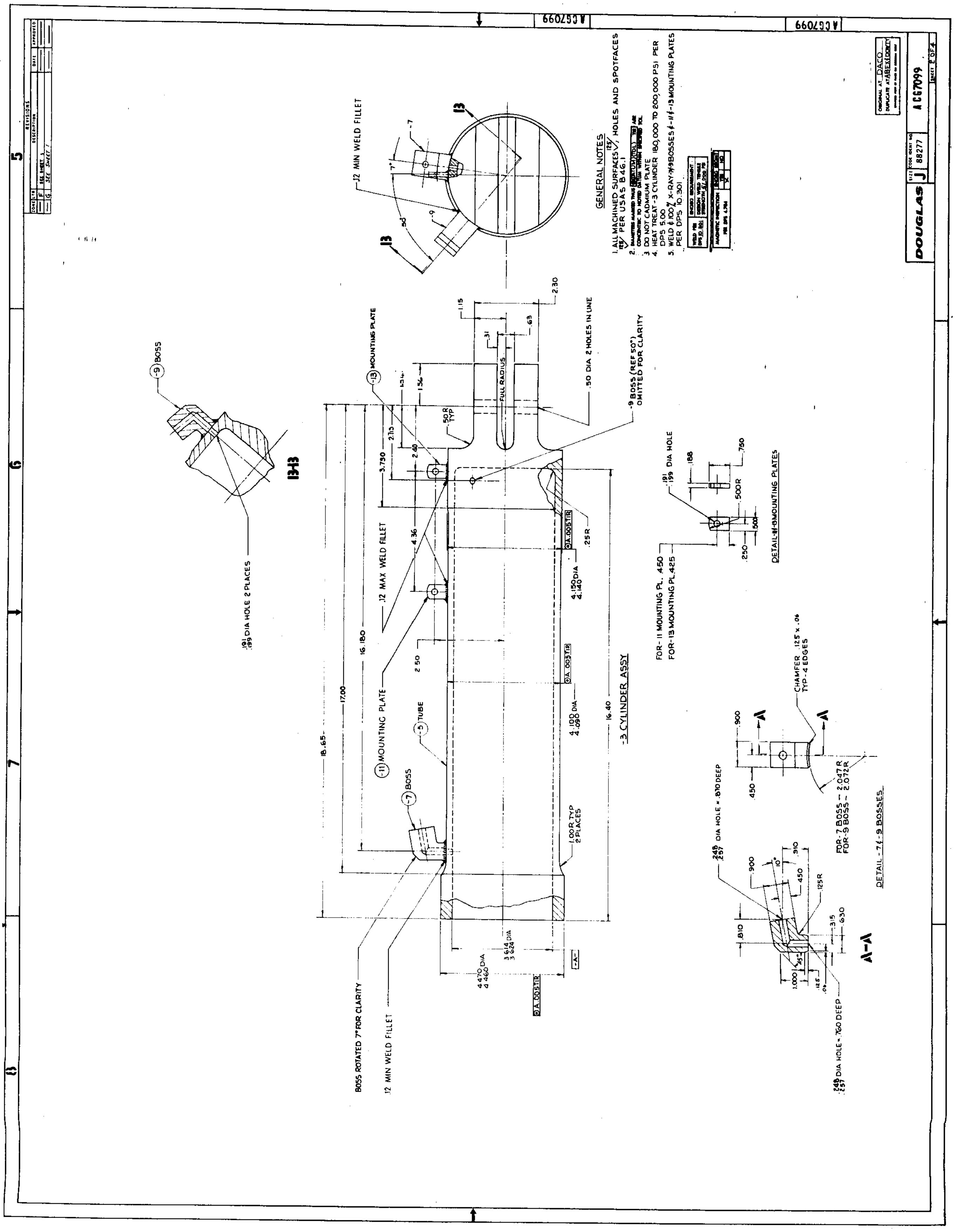
60.69 SAMPLE DRAWINGS



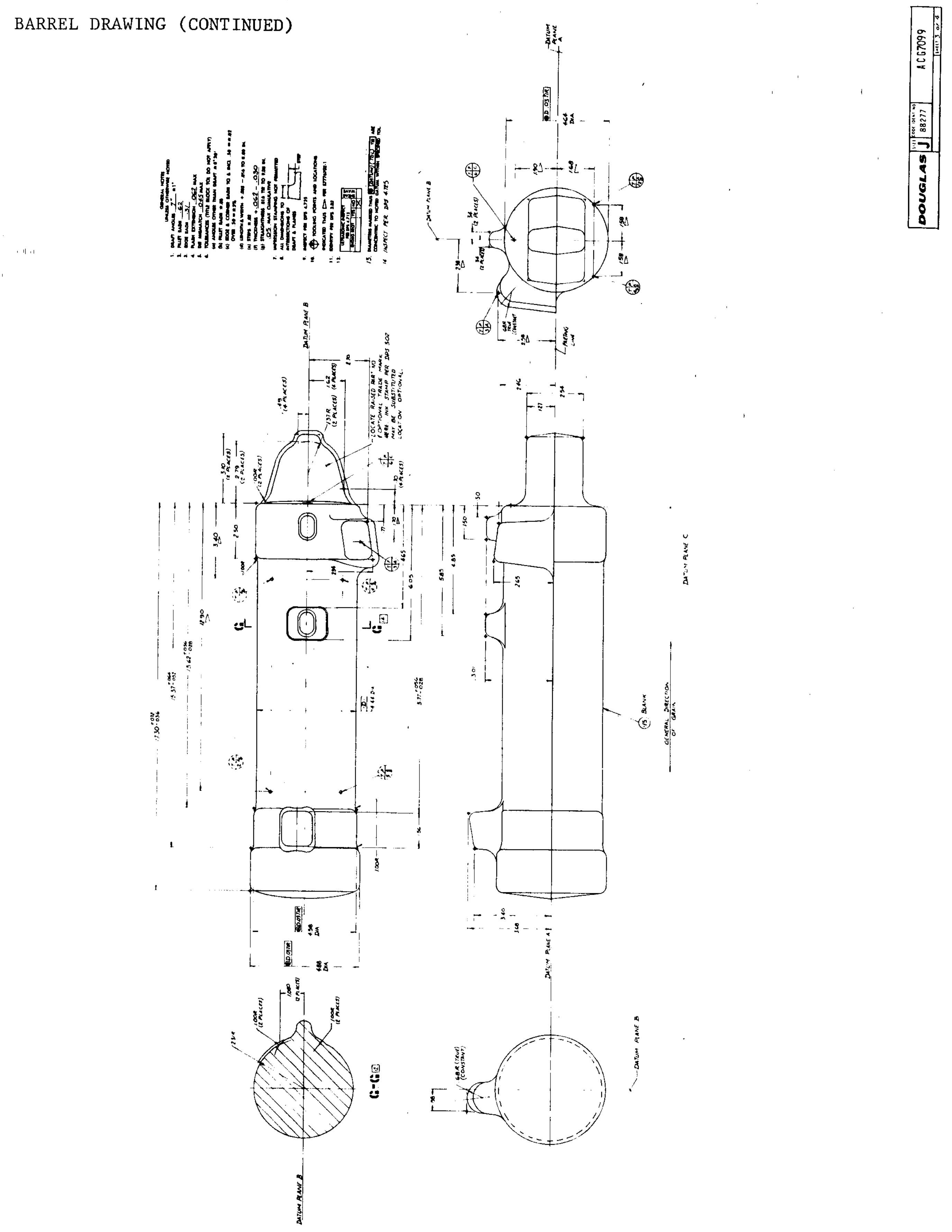
60.691 BARREL DRAWING



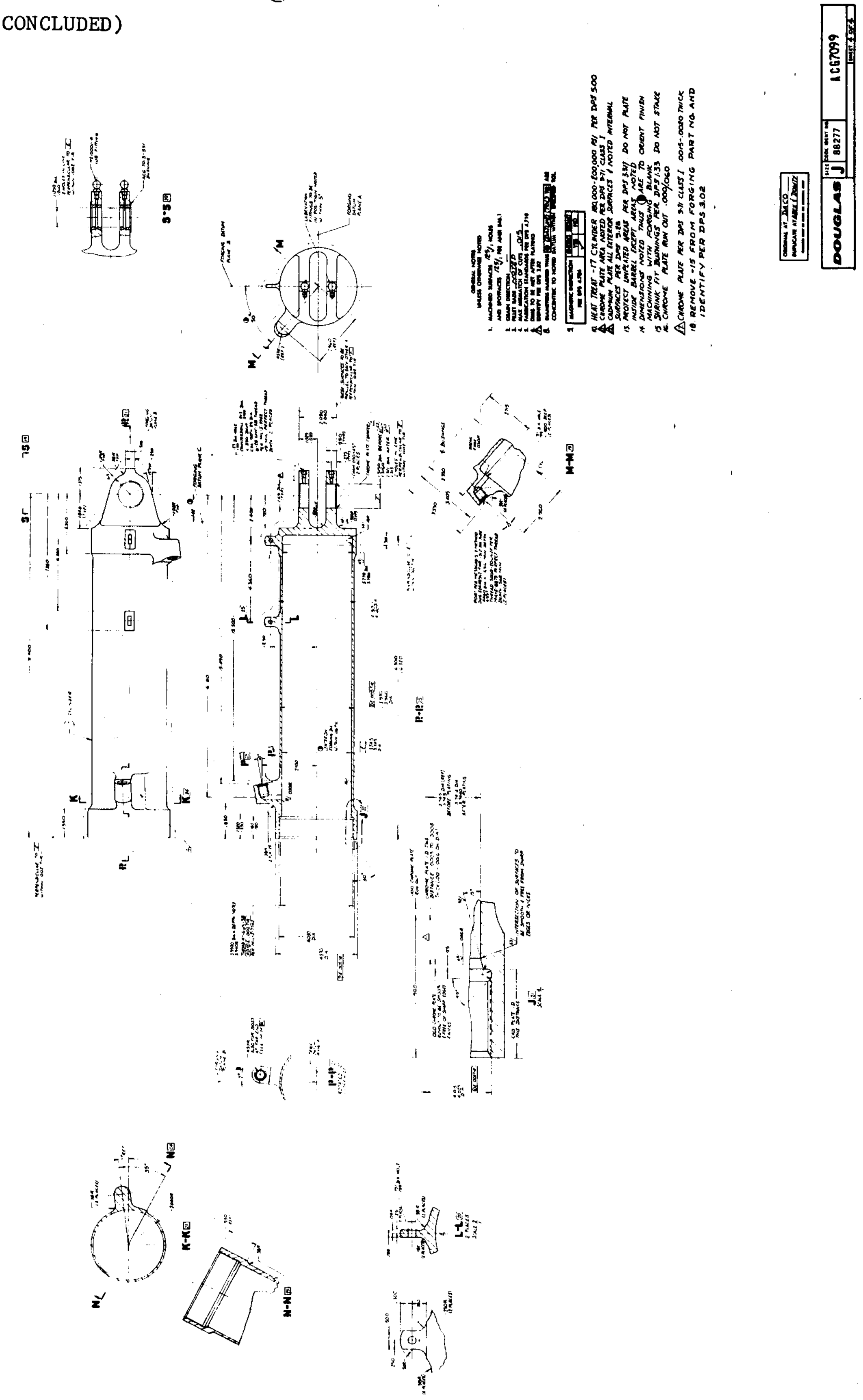
BARREL DRAWING (CONTINUED)



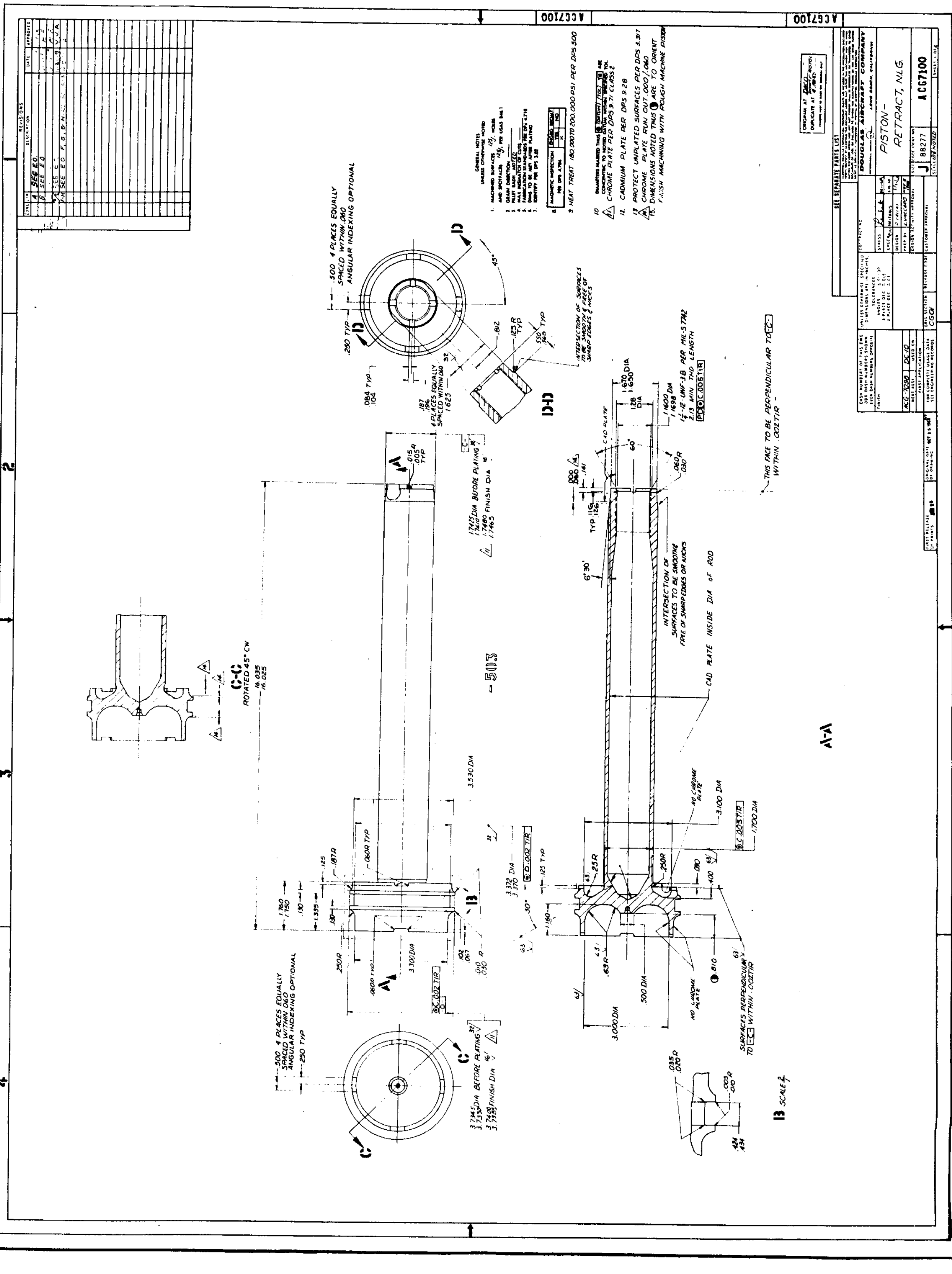
BARREL DRAWING (CONTINUED)



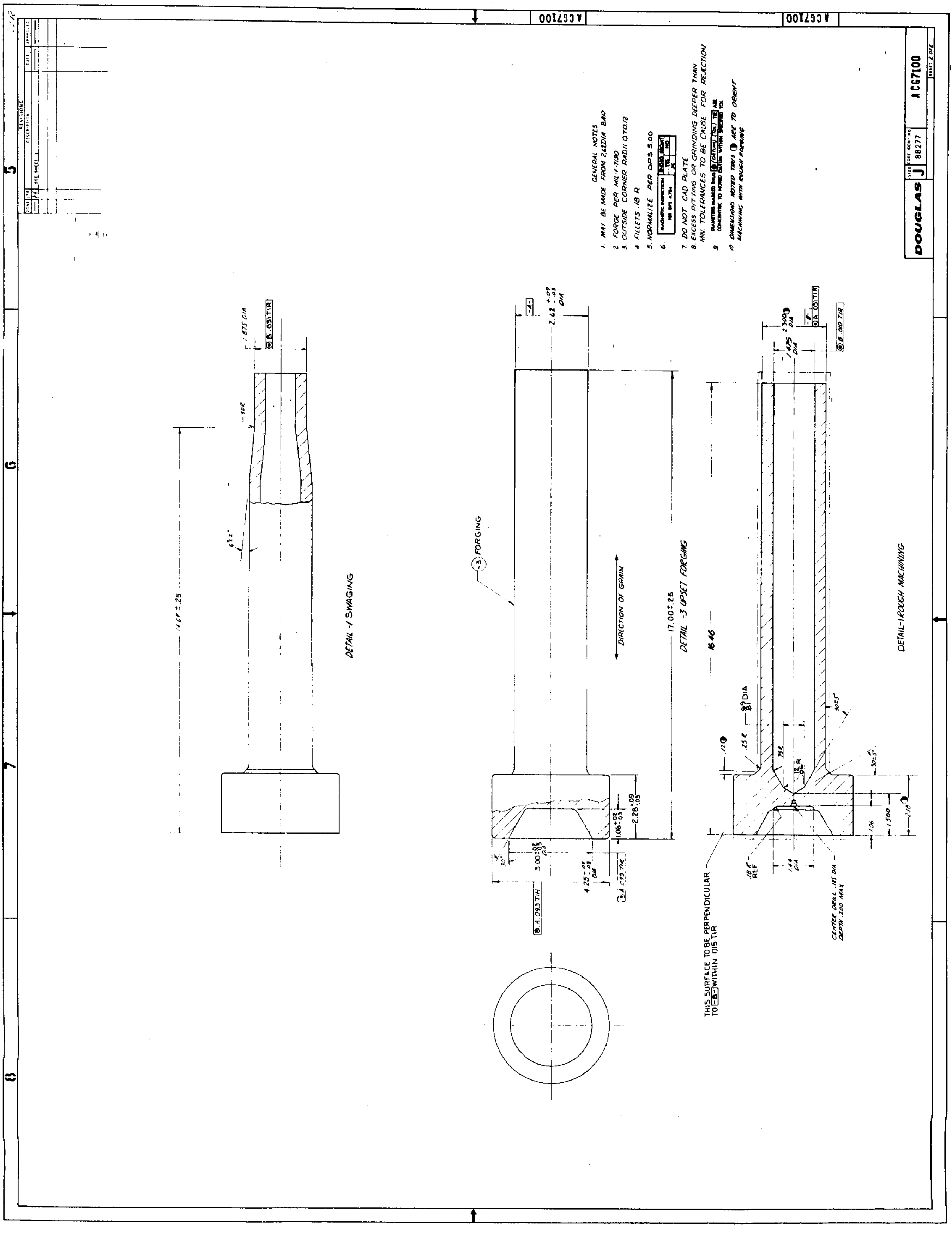
BARREL DRAWING (CONCLUDED)



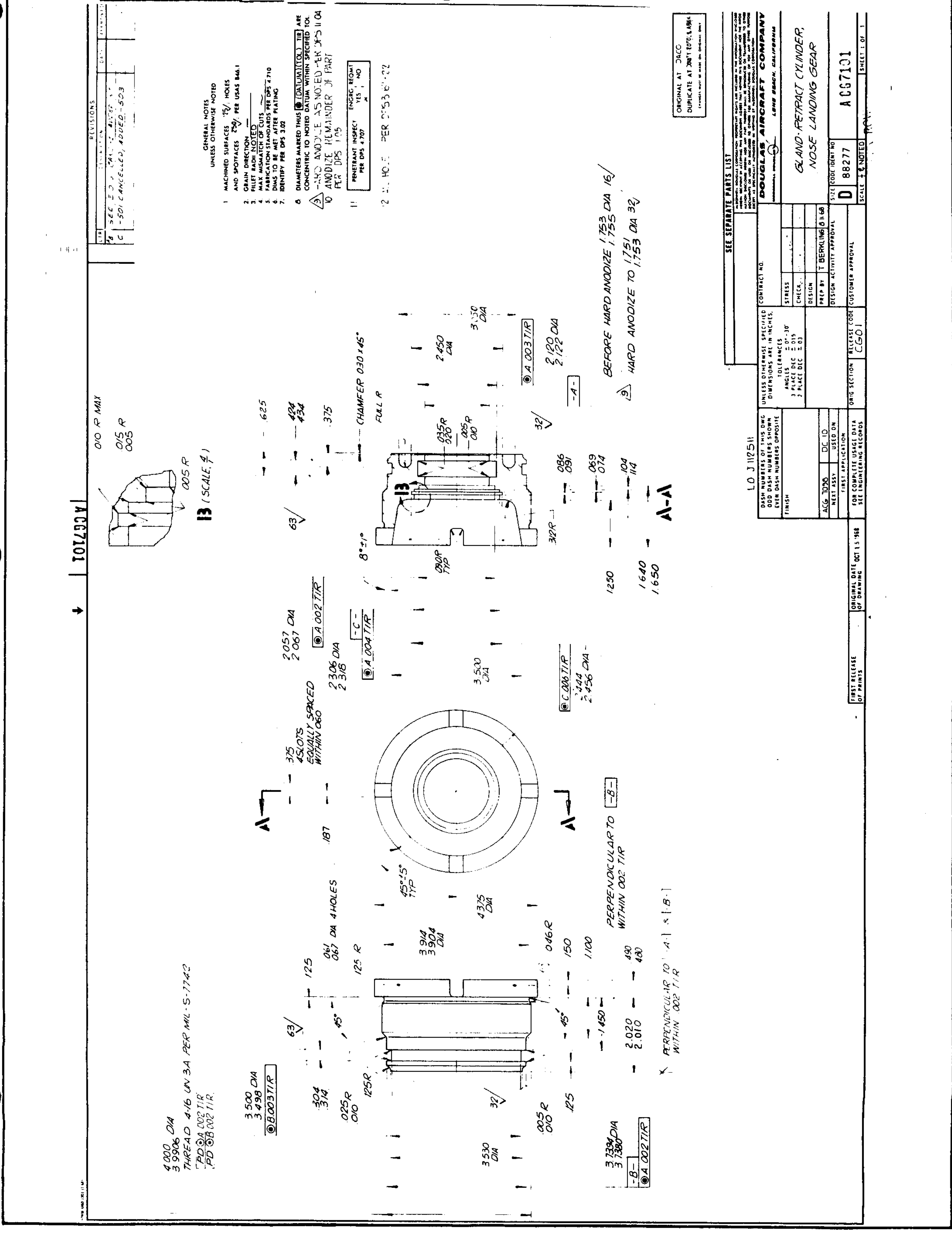
60.692 PISTON DRAWING



PISTON DRAWING (CONCLUDED)



60.693 END GLAND DRAWING



60.7 QUALIFICATION

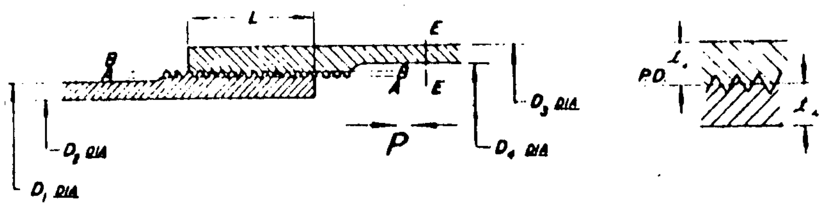
The qualification testing of each actuating cylinder must be considered in order to give confidence of satisfactory operation when the actuating cylinder is subjected to the aircraft environment for the service life intended. For information relative to qualification testing, refer to Section 90 and consult with the Hydro-Mechanical Test Group. A successful qualification test program depends to a large extent on the cylinder designer's ability to define all of the design requirements and to recognize any special requirements early in the design phase. Refer to MIL-C-5503 for military hydraulic cylinder performance and test requirements.

60.8 REFERENCE DATA

In addition to the data shown herein, the data contained in Report No. MDC A2781, HYDRAULIC ACTUATOR STRESS MANUAL, are recommended for reference.

60.81 STRESS ANALYSIS OF THREADED JOINTS

ILLUSTRATION:



UNIT TENSION OR COMPRESSION STRESS

FIND: INNER AND OUTER MEMBER STRESS (f_t OR f_c)

WHEN: f_t = UNIT TENSILE STRESS (PSI)

f_c = UNIT COMPRESSIVE STRESS (PSI)

P = DESIGN AXIAL LOAD (LBS)

D_1 = OD TUBE (INCHES)

D_2 = ID TUBE (INCHES)

D_3 = OD OUTER MEMBER (INCHES)

D_4 = ID OUTER MEMBER (INCHES)

SOLUTION:

INNER MEMBER

$$f_t \text{ OR } f_c = \frac{P}{\frac{\pi}{4}(D_1^2 - D_2^2)}$$

OUTER MEMBER

$$f_t \text{ OR } f_c = \frac{P}{\frac{\pi}{4}(D_3^2 - D_4^2)}$$

NOTE: $83\% = \dfrac{1 \times 100}{1.2 \text{ FITTING FACTOR}}$

F_t ALLOWABLE = 83% ULTIMATE TENSILE STRESS OR 50% IF CASTING

F_c ALLOWABLE = 83% ULTIMATE COMPRESSIVE STRESS OR 50% IF CASTING

RADIAL UNIT TENSION OR COMPRESSION STRESS (HOOP STRESS)

FIND: INNER AND OUTER MEMBER STRESS (f_t OR f_c)

WHEN: f_c = RADIAL UNIT COMPRESSIVE STRESS (PSI)

f_t = RADIAL UNIT TENSILE STRESS (PSI)

P = DESIGN AXIAL LOAD (LBS)

L = LENGTH THREADS ENGAGED (INCHES)

t_i = THICKNESS ID TO PD INNER MEMBER (INCHES)

t_o = THICKNESS OD TO PD OUTER MEMBER (INCHES)

SOLUTION:

INNER MEMBER

60° THREAD

$$f_c = .0919 \frac{P}{Lt_i}$$

45° THREAD

$$f_c = .066 \frac{P}{Lt_i}$$

OUTER MEMBER

60° THREAD

$$f_t = .0919 \frac{P}{Lt_o}$$

45° THREAD

$$f_t = .066 \frac{P}{Lt_o}$$

NOTE: $83\% = \frac{1 \times 100}{1.2 \text{ FITTING FACTOR}}$

F_t ALLOWABLE = 83% ULTIMATE TENSILE STRESS OR 50% IF CASTING

F_c ALLOWABLE = 83% ULTIMATE COMPRESSIVE STRESS OR 50% IF CASTING

60° THREAD AMERICAN NATIONAL STANDARD

45° THREAD DOUGLAS SPECIAL

UNIT SHEAR STRESS

Δ FIND: INNER AND OUTER MEMBER STRESS (f_s)

WHEN:

60° THREAD:

$n = .577$ (A CONSTANT)
$m = .0919$ (A CONSTANT)
$q = .650$ (A CONSTANT)
$R = .758$ (A CONSTANT)
$S = .866$ (A CONSTANT)
$U = .50$ (A CONSTANT)
D_F = THREAD PITCH DIAMETER
D_M = THREAD PITCH DIAMETER

45° THREAD:

$n = .414$ (A CONSTANT)
$m = .066$ (A CONSTANT)
$q = .848$ (A CONSTANT)
$R = .848$ (A CONSTANT)
$S = 1.207$ (A CONSTANT)
$U = .60$ (A CONSTANT)
D_F = FEMALE THREAD DIA
D_M = MALE THREAD DIA

f_s = UNIT SHEAR STRESS (PSI)
p = PITCH = 1 DIVIDED BY NO. OF THREADS PER INCH
E_i = MODULUS OF ELASTICITY - INNER MEMBER
E_o = MODULUS OF ELASTICITY - OUTER MEMBER
ρ = HYDRAULIC DESIGN PRESSURE (PSI)
f_c = UNIT COMPRESSIVE STRESS (PSI)
f_t = UNIT TENSILE STRESS (PSI)
t_i = THICKNESS ID TO PD INNER MEMBER (INCHES)
t_o = THICKNESS OD TO PD OUTER MEMBER (INCHES)
L = LENGTH OF THREADS ENGAGED (INCHES)
PD = PITCH DIAMETER
P = DESIGN AXIAL LOAD (LBS)

Δ REVISED 8-15-54

Δ

UNIT SHEAR STRESS (CONT)

SOLUTION:

a. INNER MEMBER (MALE THREAD – SECTION "A–A")

$$f_s \text{ (MAX)} = \sqrt{f_s^{\,2} + \left(\frac{f_c}{2}\right)^2}$$

WHEN: $f_s = \frac{P}{\text{AREA A–A}}$

(Shear)

$f_c = \frac{mP}{Lt_i}$ OR $\frac{nf_s}{1}$ (WHICHEVER IS GREATER)

(Hoop Compression or Radial Compression)

This term to be deleted if no hyd pressure on thd

$$\text{MIN SHEAR AREA "A–A"} = \left[\frac{qp}{1} - \frac{1}{2}\left(\frac{m\,P\,PD}{Lt_iE_i}\right) - \frac{1}{2}\left(\frac{m\,P\,PD}{Lt_oE_o}\right) - \overbrace{\frac{1}{4}\left(\frac{\rho\,(PD)^2}{E_ot_o}\right)} - \frac{\text{PD TOL (MALE)}}{2} - \frac{\text{MINOR DIA TOL (FEMALE)}}{2}\right] \frac{\pi L \text{ (MAX MINOR DIA (FEMALE)}}{Sp}$$

APPROX AREA "A–A" = $U \pi D_F{}^{*} L$ (ROUGH ASSUMPTION ONLY)

b. OUTER MEMBER (FEMALE THREAD – SECTION "B–B")

$$f_s \text{ (MAX)} = \sqrt{f_s^{\,2} + \left(\frac{f_t + f_c' + f_t'}{2}\right)^2}$$

WHEN: $f_s = \frac{P}{\text{AREA B–B}}$ $f_c' = nf_s$

(Shear) *(Radial Tension)*

$f_t = \frac{m\,P}{Lt_o}$ $f_t' = \frac{\rho\,PD}{2t_o}$

(Hoop Tension) *(Hoop Tension, Pressure on Thd)*

This term to be deleted if no hyd pressure on thd

$$\text{MIN SHEAR AREA "B–B"} = \left[\frac{Rp}{1} - \frac{1}{2}\left(\frac{m\,P\,PD}{Lt_iE_i}\right) - \frac{1}{2}\left(\frac{m\,P\,PD}{Lt_oE_o}\right) - \overbrace{\frac{1}{4}\left(\frac{\rho\,(PD)^2}{E_ot_o}\right)} - \frac{\text{PD TOL (FEMALE)}}{2} - \frac{\text{MAJOR DIA TOL (MALE)}}{2}\right] \frac{\pi L \text{ (MIN MAJOR DIA (MALE)}}{Sp}$$

APPROX AREA "B–B" = $U \pi D_M{}^{*} L$ (ROUGH ASSUMPTION ONLY)

NOTE: F_S ALLOWABLE = 83% ULTIMATE SHEAR STRESS OR 50% IF CASTING.

* FOR THE 60° THREAD, D_M MAY BE SUBSTITUTED FOR D_F AS EACH HAS THE SAME VALUE. FOR THE 45° THREAD, D_M AND D_F CANNOT BE SUBSTITUTED ONE FOR THE OTHER BECAUSE GREATER ACCURACY IS NECESSARY IN DETERMINING THE DIMENSION IN QUESTION.

Δ REVISED 8-15-54

60.82 AXIAL LOAD FATIGUE BEHAVIOR The S-N curves shown in Figures 60-20, 60-21, 60-22, and 60-23 are presented to serve as a guide in the design of parts for fatigue strength. The actual test points have been faired and the lines represent probable behavior of the materials. Scatter has not been evaluated and therefore these diagrams should not be used for safe design values. K_t is the theoretical stress concentration factor calculated for the specimens as shown in Figure 60-24. It should be noted that no advantage is realized in using high heat treat steels if long fatigue life is required and stress raisers are present.

FATIGUE LIFE RESULTING FROM MAXIMUM STRESS EQUAL TO TWO-THIRDS ULTIMATE TENSILE STRESS FROM NACA TN 3866

MATERIAL	MEAN NOMINAL STRESS KSI	FATIGUE LIFE, N. CYCLES, FOR $K_T = 1.0$	$K_T = 2.0$	$K_T = 4.0$
2024-T3 Aluminum Alloy	0	9,000	200	22
	20	-----	3,000	200
7075-T6 Aluminum Alloy	0	3,500	200	22
	20	-----	1,400	130
Normalized SAE 4130 Steel	0	3,900	610	130
	20	-----	3,200	420
Hardened SAE 4130 Steel	0	80,000	300	80
	50	105,000	1,900	500

"Maximum Stress" is the maximum tensile stress applied in a particular endurance test.

"Minimum Stress" is the minimum stress applied, considering tension to be positive, and compression negative.

"Mean Nominal Stress" is that stress half way between the "Maximum Stress" and the "Minimum Stress."

Thus a specimen subjected to 50 ksi maximum stress and 0 ksi maximum stress and 0 ksi mean nominal stress would be cycled through a range between 50 ksi tension and 50 ksi compression, which would be more severe than for a specimen under 50 ksi maximum stress and 20 ksi mean nominal stress. (A range of 50 ksi tension to 10 ksi compression.)

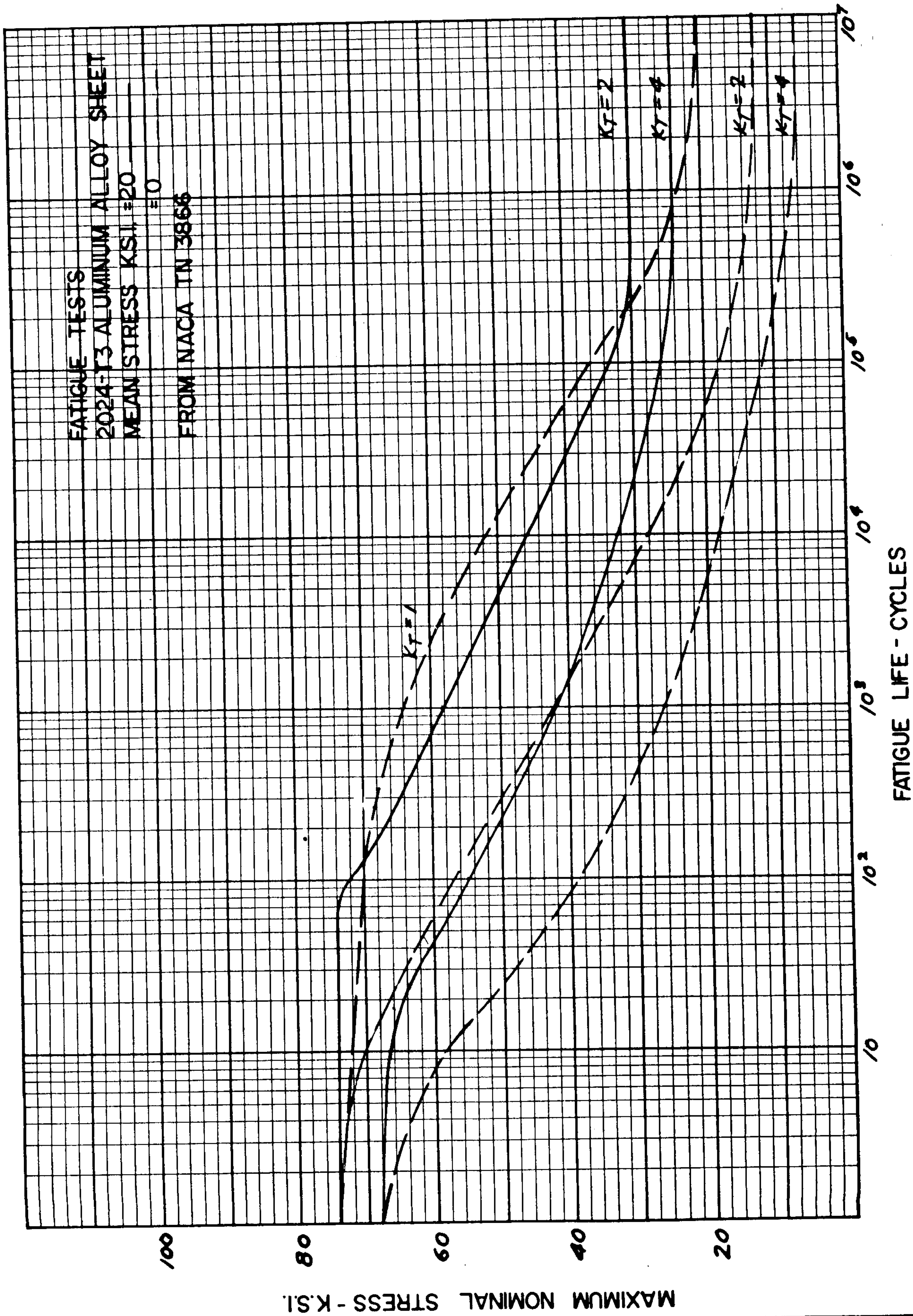

FATIGUE TESTS
2024-T3 ALUMINUM ALLOY SHEET
MEAN STRESS KSI. =20
=0
FROM NACA TN 3866
$K_T=1$
$K_T=2$
$K_T=4$
$K_T=2$
$K_T=4$
10
10^2
10^3
10^4
10^5
10^6
10^7
FATIGUE LIFE - CYCLES
100
80
60
40
20
MAXIMUM NOMINAL STRESS - K.S.I.

FIGURE 60-20.

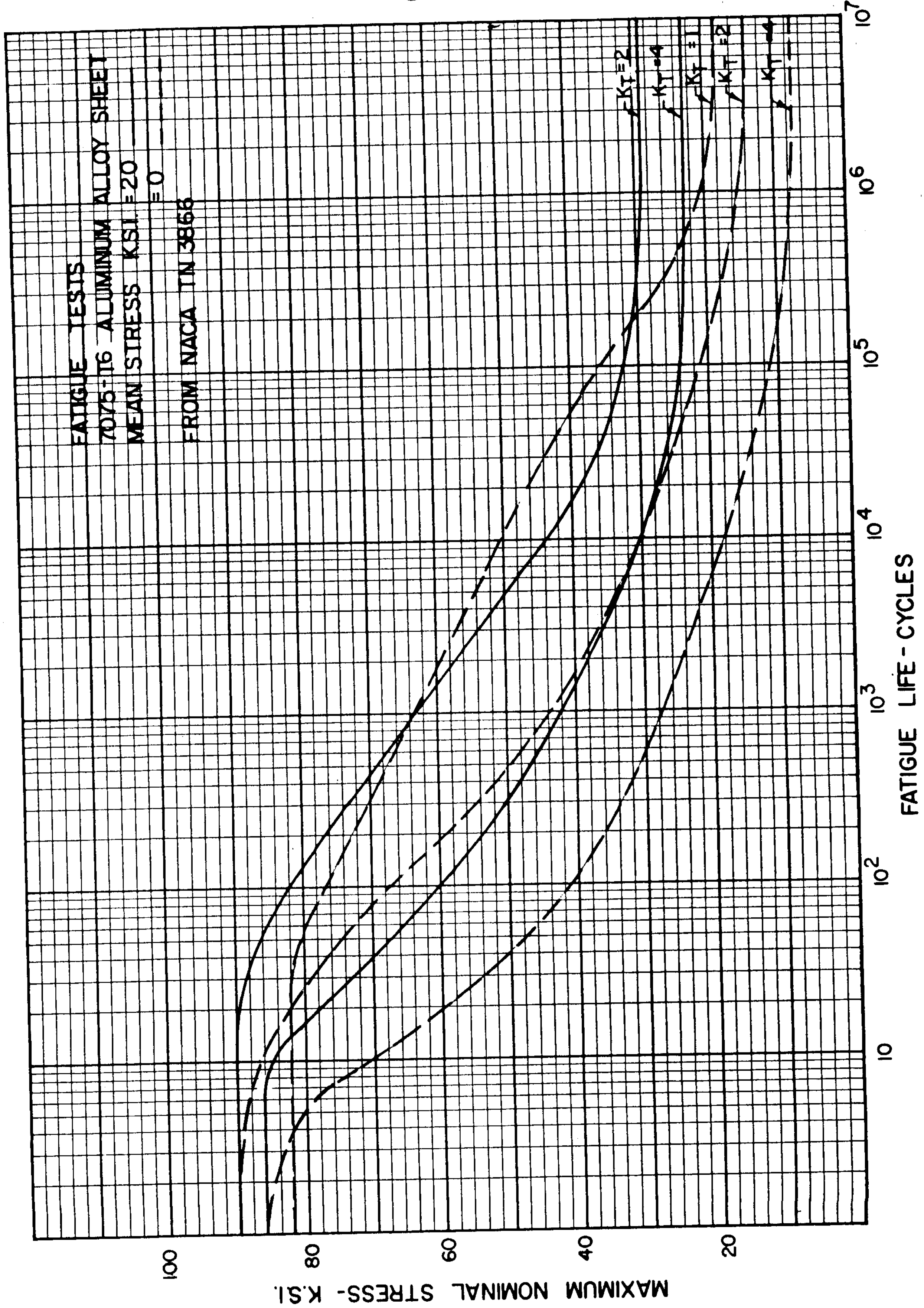

FATIGUE TESTS
7075-T6 ALUMINUM ALLOY SHEET
MEAN STRESS KSI =20
=0
FROM NACA TN 3866
K_T=2
K_T=4
K_T=1
K_T=2
K_T=4
10
10^2
10^3
10^4
10^5
10^6
10^7
FATIGUE LIFE - CYCLES
100
80
60
40
20
MAXIMUM NOMINAL STRESS- K.S.I.

FIGURE 60-21.

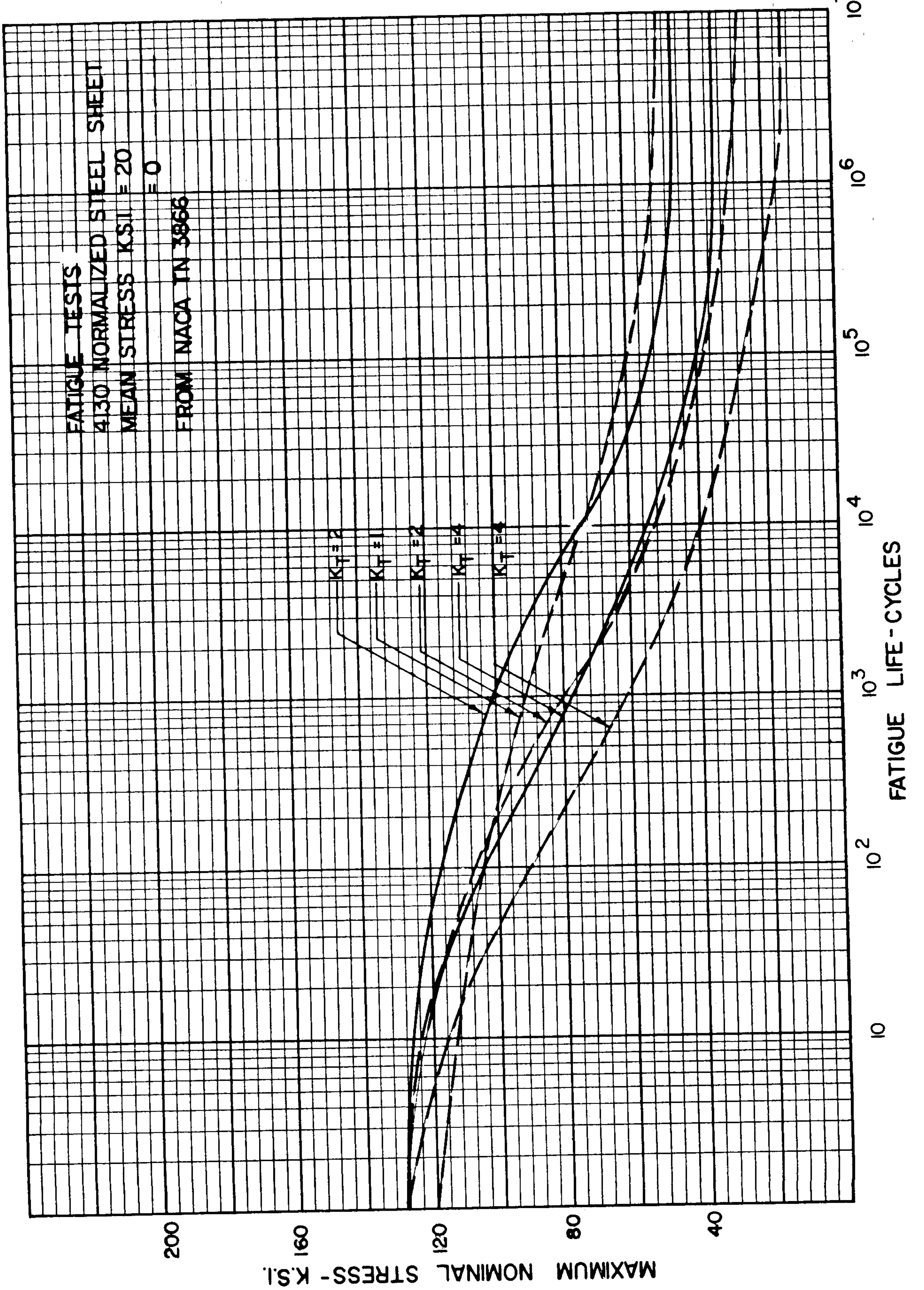

FATIGUE TESTS
4130 NORMALIZED STEEL SHEET
MEAN STRESS KSI = 20
= 0
FROM NACA TN 3866
$K_T = 2$
$K_T = 1$
$K_T = 2$
$K_T = 4$
$K_T = 4$
FATIGUE LIFE - CYCLES
10
10^2
10^3
10^4
10^5
10^6
10^7
MAXIMUM NOMINAL STRESS - K.S.I.
40
80
120
160
200

FIGURE 60-22.

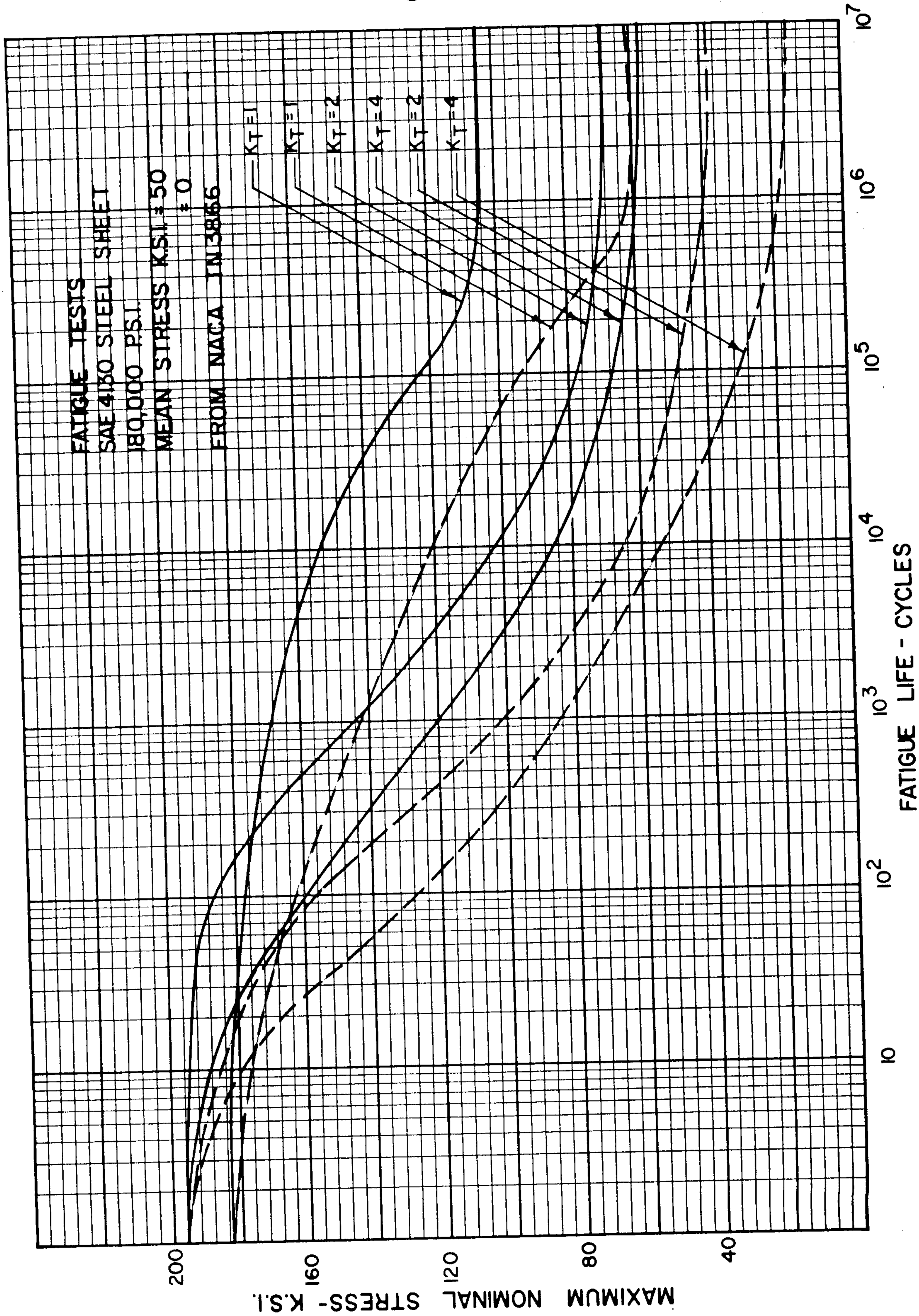
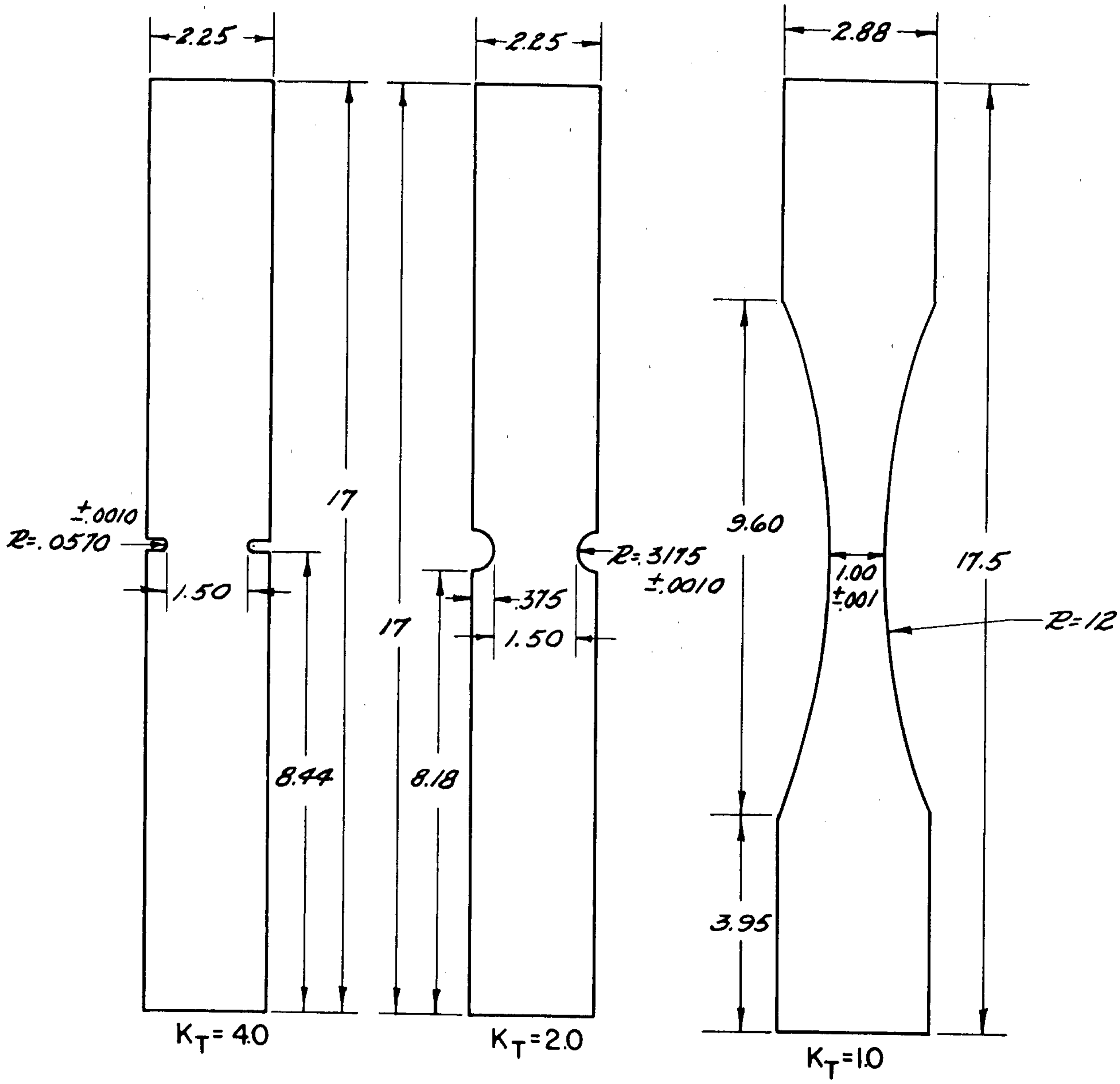

FATIGUE TESTS
SAE 4130 STEEL SHEET
180,000 PSI.
MEAN STRESS K.S.I. = 50
= 0
FROM NACA TN 3866
K_T=1
K_T=1
K_T=2
K_T=4
K_T=2
K_T=4
10
10^2
10^3
10^4
10^5
10^6
10^7
FATIGUE LIFE - CYCLES
200
160
120
80
40
MAXIMUM NOMINAL STRESS- K.S.I.

FIGURE 60-23.

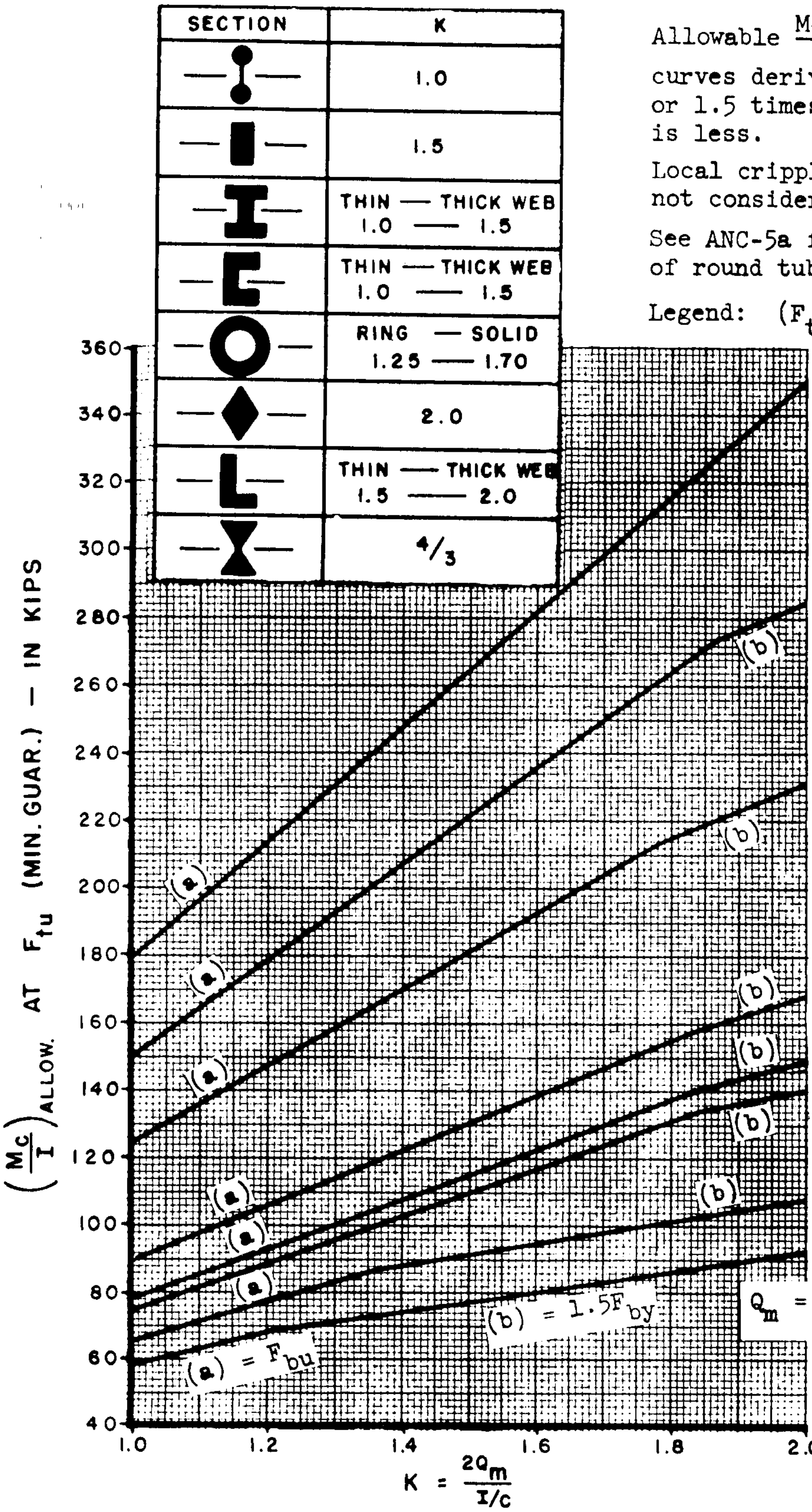


CONFIGURATION OF TEST SPECIMENS
ALUMINUM -.090 THICK
STEEL -.075 THICK

FIGURE 60-24.

60.83 MODULUS OF RUPTURE

MODULUS OF RUPTURE OF STEEL AND DURAL SHAPES



Allowable $\frac{Mc}{I}$ vs. K curve is envelope of curves derived from ultimate allowable or 1.5 times yield allowable, whichever is less.

Local crippling may be critical but is not considered in this chart.

See ANC-5a for modulus of rupture value of round tubing.

Legend: (F_{ty}) F_{tu} - Minimum guaranteed value in Kips

Alloy Steel (165) 180

Alloy Steel (135) 150
4330XX Steel Cast. (135) 150
X4130 Steel Forging (135) 150

Alloy Steel (100) 125

Alloy Steel (70) 90

75ST Bar or Extrusion (70) 78
75ST Forging (65) 75

14ST Forging (50) 65

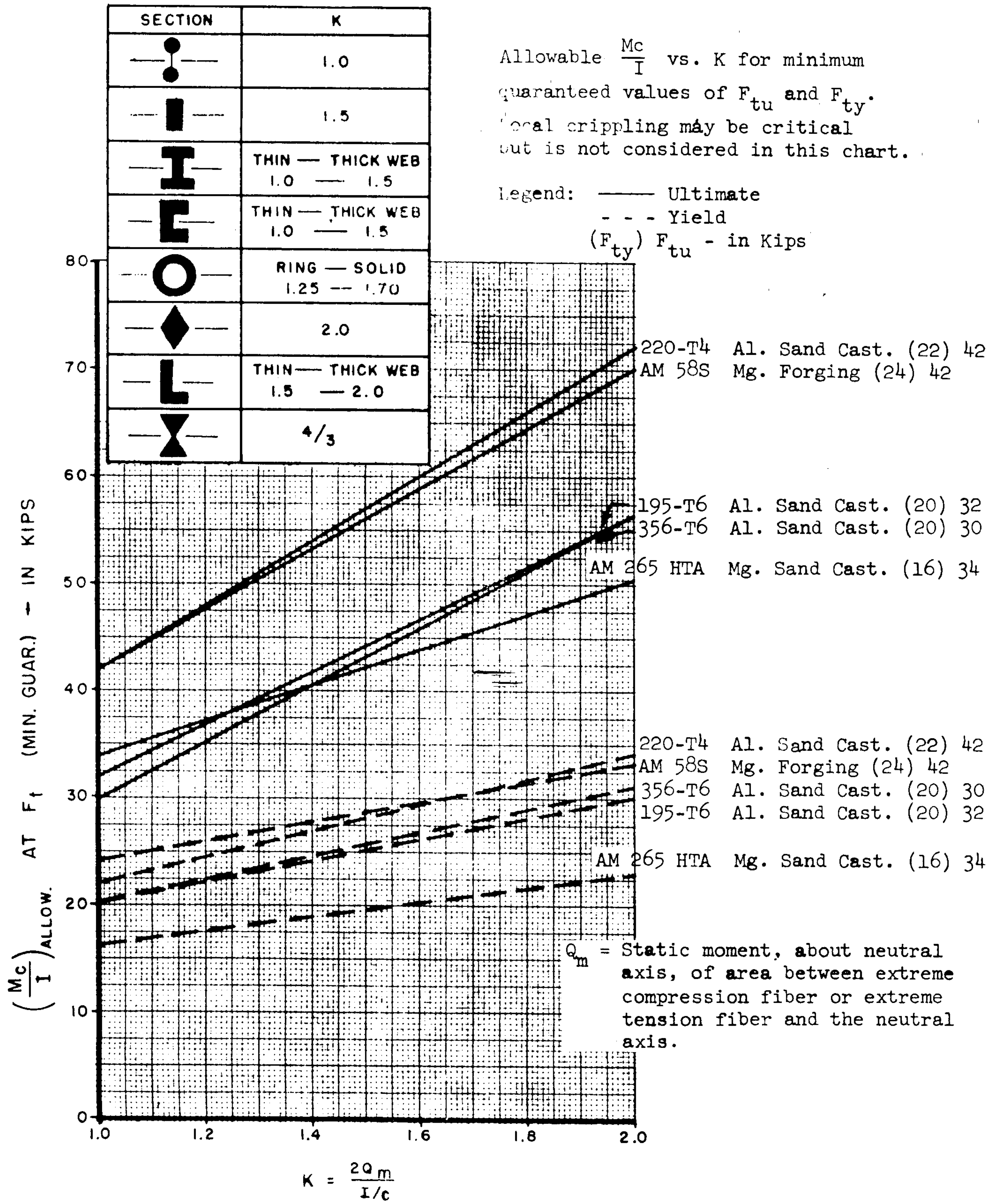
24ST Extrusion (42) 58

Q_m = Static moment, about neutral axis, of area between extreme compression fiber or extreme tension fiber and the neutral axis.

MODULUS OF RUPTURE OF DURAL AND MAGNESIUM SHAPES

Allowable $\frac{Mc}{I}$ vs. K for minimum guaranteed values of F_{tu} and F_{ty}. Local crippling may be critical but is not considered in this chart.

Legend: ——— Ultimate
- - - Yield
(F_{ty}) F_{tu} - in Kips



Q_m = Static moment, about neutral axis, of area between extreme compression fiber or extreme tension fiber and the neutral axis.

60.84 SPHERICAL BEARING DATA FROM MAC RPT 339 The following is an excerpt from MCAIR Report 339 and can be used to quickly determine the acceptability of spherical bearing dimensions.

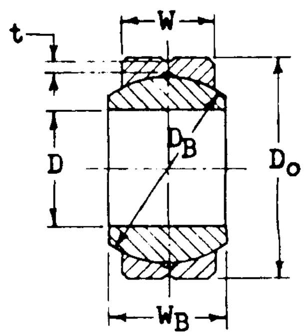
60.841 SPHERICAL BEARING CONFIGURATION Bearings of this type are available in a great number of sizes, shapes, materials, surface treatments and design features. Extensive MAC testing has shown some of them to be satisfactory and others to be very unsatisfactory. The following discussion lists the parameters found to be most important to the bearing configuration and application, and defines what is satisfactory in regard to each.

1. Material - The ball is almost invariably of extremely hard steel, chrome plated. The outer race is available in bronze, carbon steel or stainless steel. Bronze outer race bearings have proven to be definitely superior for all normal applications of a highly loaded rotating joint. The stainless steel outer race bearing has an advantage in the less frequent application in which the joint must withstand very high loads without joint rotation and much lower loads while rotating. (Some landing gear joints, for example.)

2. Dimensional Proportions - The basic design of the spherical bearing is such that the ball is inherently weakest at the ends of the bolt hole and the outer race is inherently weakest at the centerline or half width of the bearing. From the sketch below, it is evident that a poor choice of dimensions D, D_B and W_B would result in a near knife-edge between the bore and the spherical surface of the ball. This situation, existent in many catalogued bearings, results in early fatigue failures of the ball due to bolt bending effects and in damage to the inner faces of the clevis due to the high clamp-up bearing stresses. It is likewise apparent that too small a difference between dimensions D_O and D_B, especially if the bearing has lubrication grooves, would make the outer race thin and weak. This is a very prevalent condition among the "standard" bearings presently available and it invariably results in the outer race spreading and eventually splitting when subjected to repeated rotations under load. This thin outer race situation is serious enough to merit the establishment of definite criteria for acceptable outer race proportions.

The spreading and finally splitting of the outer race is the result of plastic extrusion of the metal under the compressive pressures and the lateral tension stresses which result from the lateral components of the bearing pressures between the spherical surfaces. From the geometry of the loading it can be shown that these lateral tension stresses, for a given bearing stress, are proportional to

$$\frac{W^2}{D_B t}$$ Defined by the Sketch



It is not feasible to attempt to actually compute this stress and analytically arrive at a life expectancy but by evaluating test data in terms of these criteria it is possible to establish the dividing line between satisfactory and unsatisfactory outer race proportions.

$$\frac{W^2}{D_B t} < 3.6 \qquad \text{Satisfactory}$$

$$\frac{W^2}{D_B t} > 3.6 \qquad \text{Unsatisfactory}$$

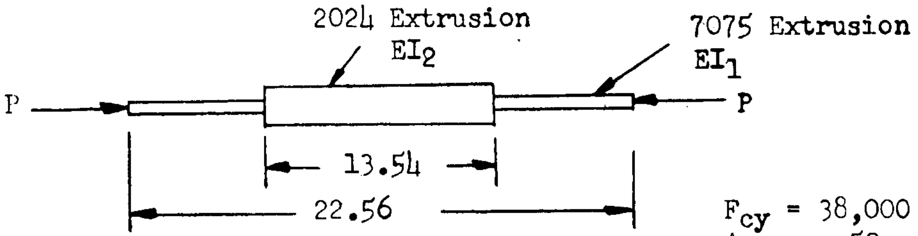
60.85 COLUMN ANALYSIS FROM MAC RPT 339

1. If the stress in either portion of the column exceeds the proportional limit, the tangent modulus, E_T, should be used for E. The two portions of the column should also be checked for crippling. Tangent modulus curves for some materials are available in MIL-HBK-5. For other materials use the following equation:

$$\frac{E_T}{E} = \frac{1}{1 + \frac{3}{7} n \left(\frac{F}{F_{.7}}\right)^{n-1}}$$

Values of n and $F_{.7}$ are given for a variety of materials in MAC Report 6763, Bulletin 4.1.

Example:



$F_{cy} = 70{,}000$
$A_1 = .30$
$I_1 = .10$

$F_{cy} = 38{,}000$
$A_2 = .50$
$I_2 = .50$

Assume $P_{cr} = 18{,}220$ lbs.

$$f_1 = \frac{18220}{.30} = 60{,}750 \text{ psi} \qquad f_2 = \frac{18220}{.5} = 36{,}440 \text{ psi}$$

$$E_{T_1} = .590\ (10.5 \times 10^6) \qquad E_{T_2} = .19\ (10.5 \times 10^6)$$

$$\frac{EI_1}{EI_2} = \frac{(.59)\ (10.5 \times 10^6)\ .10}{(.190)(10.5 \times 10^6)\ \ .50} = .622 \qquad a/L = \frac{13.54}{22.56} = .60$$

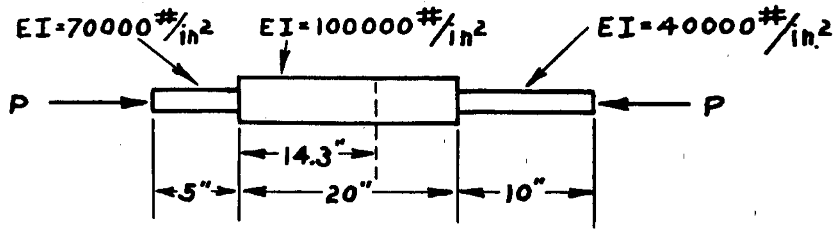
$$\frac{P_{cr}}{P_E} = .940 \quad P_E = \frac{\pi^2\ (.19)(10.5)(10^6)(.5)}{(22.56)^2} = 19{,}390$$

$$P_{cr} = (19{,}380)(.940) = 18{,}220 \text{ lbs.}$$

In general, the first try will not check as it did above; therefore more tries must be made until agreement is reached.

2. If the column is not symmetrical, the curves on page 16.42 can still be utilized to find the critical load. A guess must be made of the location of the point of maximum deflection of the column. Based on this assumption, a critical load can be calculated for each end of the column by assuming symmetry about the point of maximum deflection. In general the two critical loads thus computed will not agree. More tries must then be made at guessing the point of maximum deflection until the two critical loads computed for each end are brought into agreement.

Example:



Assume 14.3" dimension as shown to the point of maximum deflection.

Left Side

$$\frac{EI_1}{EI_2} = .7 \qquad \frac{a}{l} = \frac{28.6}{38.6} = .741 \qquad \frac{P_{cr}}{P_E} = .977$$

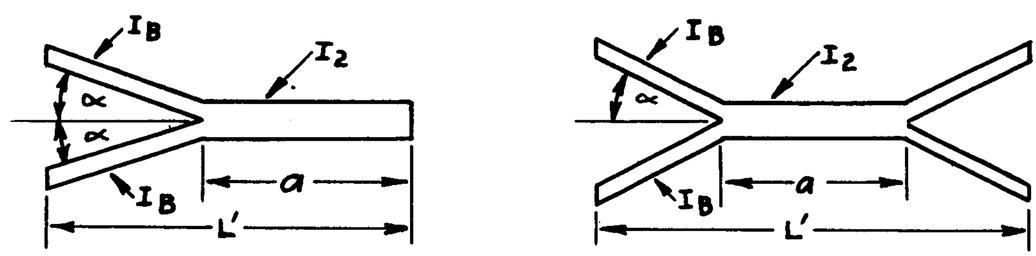
$$P_{cr} = \frac{(.977)(9.86)100000}{(38.6)^2} = 647 \text{ lbs.}$$

Right Side

$$\frac{EI_1}{EI_2} = .4 \qquad \frac{a}{l} = \frac{11.4}{31.4} = .361 \qquad \frac{P_{cr}}{P_E} = .647$$

$$P_{cr} = \frac{(.647)(9.86)(100000)}{(31.4)^2} = 647 \text{ lbs.}$$

3. In some cases (such as landing gear drag braces, etc.) one step of the column is actually composed of a truss as shown below.

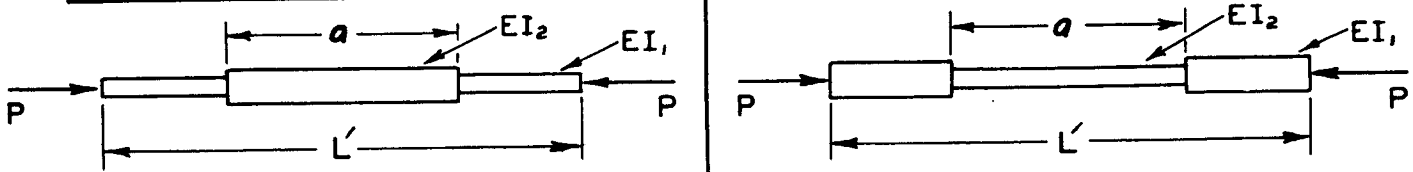


To calculate the critical load (for buckling out of the truss plane) the effective moment of inertia that must be used in the stepped column curves becomes:

$$I_1 = 2I_B \cos^3\alpha$$

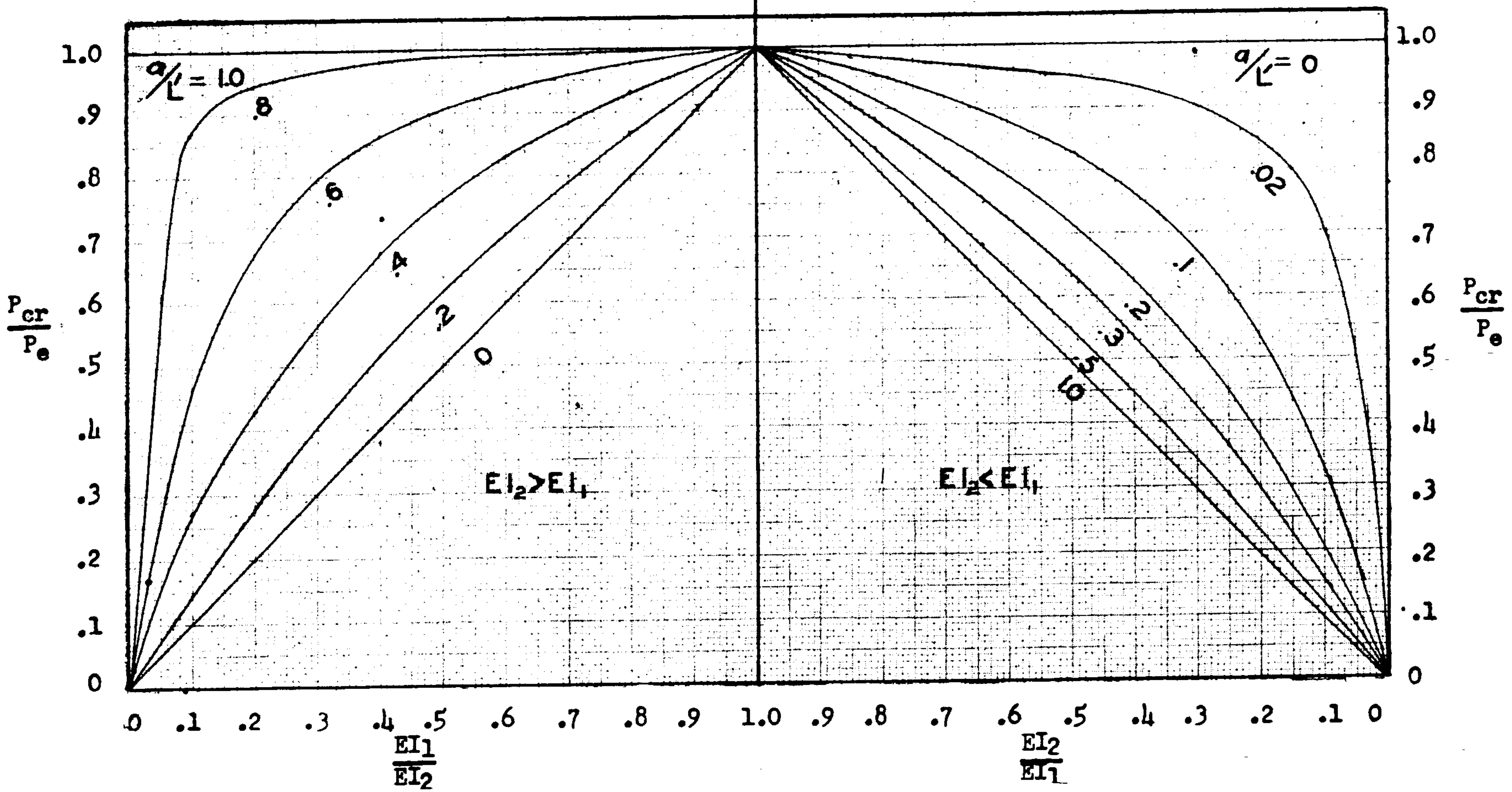
It should be noted that the effective length of "truss" step is the projected length on the plane of symmetry.

CRITICAL LOADS FOR SYMMETRICAL STEPPED COLUMNS

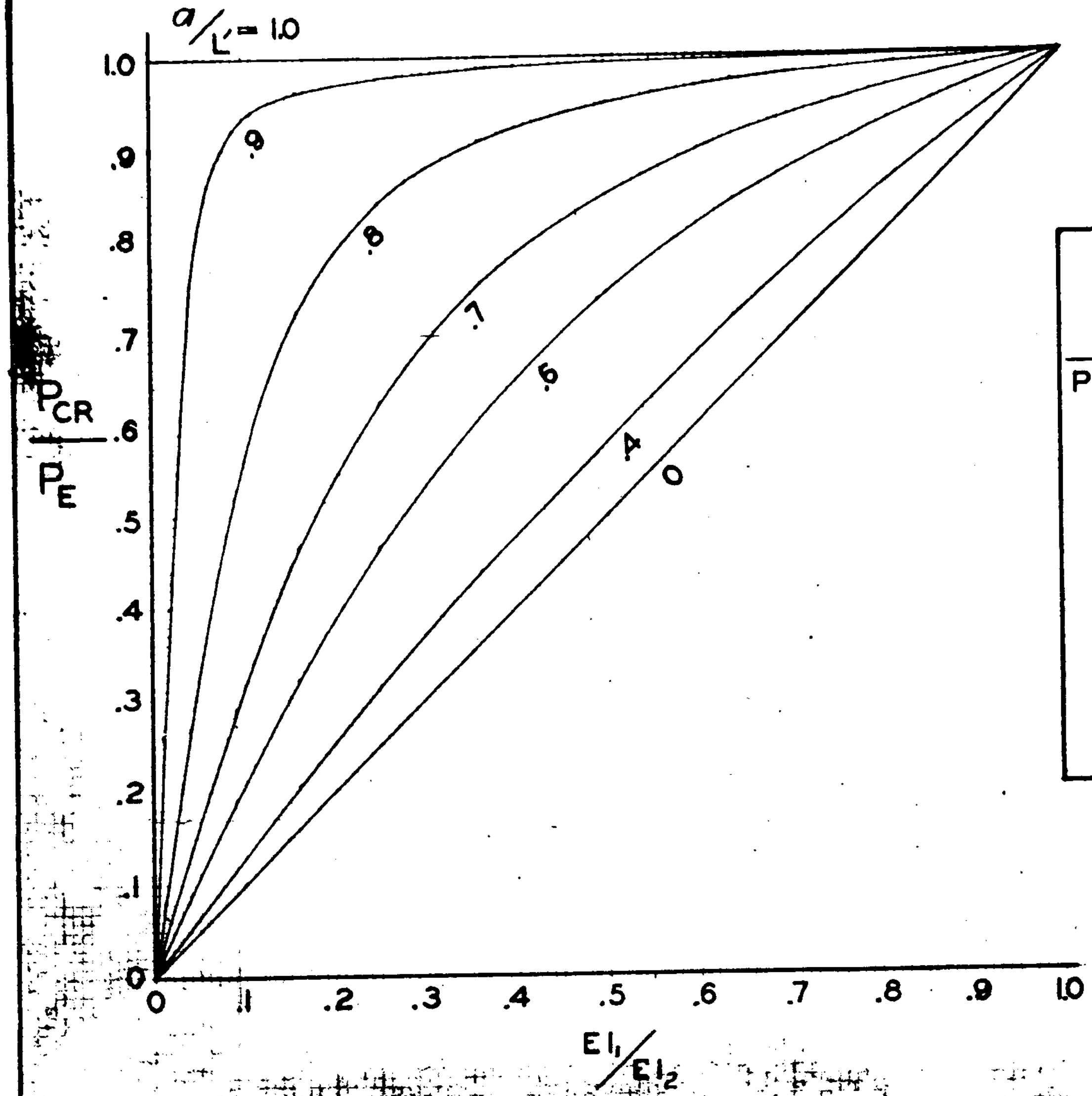


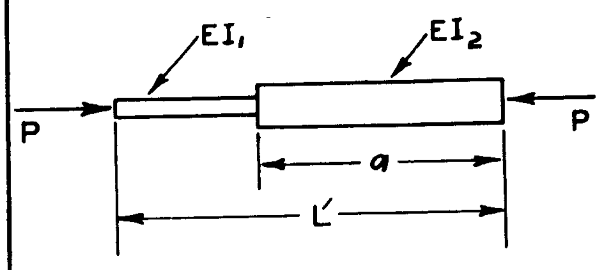
$$P_E = \frac{\pi^2 EI_2}{(L')^2} \quad (EI_2 > EI_1)$$

$$P_E = \frac{\pi^2 EI_1}{(L')^2} \quad (EI_1 > EI_2)$$



CRITICAL LOADS FOR UNSYMMETRICAL STEPPED COLUMNS





$$P_E = \frac{\pi^2 EI_2}{(L')^2}$$

60.86 CYLINDER DESIGN PROBLEMS

a. Fatigue Loading of Internal Stops Internal hydraulic stops in cylinders should be sufficiently strong or snubbing should be provided to prevent fatigue-loading failures. Do not depend on rigging to prevent fatigue loading. However, the alert engineer will emphasize rigging controls to stick-stop limits to avoid control-cylinder bottoming.

b. Bolt Interference Incorrect bolt length or reverse bolt installation in flight-control system or adjacent assemblies should not cause system interference.

c. Bolts Use of special bolts invites substitution in service unless they are properly designed.

d. Filling and Bleeding Provide positive filling and bleeding of actuators that are depended upon to provide surface snubbing.

e. Aluminum Cylinder Barrels Use of aluminum cylinder barrels is not recommended for commercial applications; however, if aluminum barrels are used, the use of hard anodize drastically reduces the fatigue life and should be accounted for in the design.

f. Trapped Areas Pressure traps (i.e., unvented space between two seals) must be eliminated.

g. Locks With Insufficient Margin to Withstand Pressure Surges
An example of this problem is the surge in a landing-gear actuator with internal locks when the solenoid valve returns to neutral as the generator drops off the line on engine shutdown. This also can occur to autopilot servo cylinders subject to unlocking with return surges. Balanced area design is the surest solution.

h. Control Actuator Shaft Seal Leaks For protection from dynamic seal failures in critical control actuators, incorporate dual shaft seals with return vent between seals.

i. Actuator Piston Head and Nut Backing Off from Piston Shaft
Do not safety-wire piston head retaining nuts to a piston head that can rotate on the piston shaft. Use of acceptable locking devices such as NAS 559 keys is recommended.

j. Cylinder Stops Be sure that cylinder bottoming loads are transmitted only into the designed stop. Insufficient bushing chamfer can cause interference and result in cracked end caps.

k. Shot-Peening Shot-peening can be a crutch for poor machining. It is effective 0.001 to 0.002 inch deep. Thus it may not completely eliminate but rather hide the effects of poor machining.

60.86 CYLINDER DESIGN PROBLEMS (CONT)

l. Internal Drilled Passages Feather edges around internal drilled passages create stress concentration areas that can eventually crack and fail.

m. Line Attachment On actuating cylinders, provide clipping tie points so that piping can be held in place.

n. Dashpots Hydraulic cylinders without dashpots that have high extend/retract rates or are adjacent to the cabin result in impact damage and/or objectionable noise. Hydraulic cylinders in this category should incorporate dashpots.

o. Cylinder End Cap Design Avoid internally threaded end caps utilizing AND 10050-type boss-sealing design. Squareness of threads to boss surface and torquing problems cause static seal leakage.

p. Return Pressure Forces Design systems to accept extending forces produced by return pressure on the differential cylinder area regardless of upstream valving.

q. End Caps Threaded detail parts (i.e., end caps and glands) are preferred over bolted joints in pressure vessels.

r. Inserts in Aluminum Where threaded fasteners are used in aluminum, inserts must be provided.

s. Cylinder Internally Threaded End Caps End caps that are internally threaded into a cylinder barrel and locked with a jam nut are subject to barrel stretch under pressure. This causes leakage or jam nut loosening. Sufficient barrel material thickness is required in the cap area to prevent barrel stretch.

t. Cylinder Piston and Rod Bearing Area To minimize binding and seal wear from cylinder side loading, provide a minimum of one diameter overlap for piston and rod bearing area when fully extended.

u. Piston-Head-to-Rod Radius Too sharp a piston-head-to-rod radius can cause fatigue failure from stresses during pressure surges or cylinder bottoming. Control of finish to 32/ is recommended. Shot-peening to cover poor machining is sometimes desirable.

v. Preload for Fatigue Resistance Preloading end caps, piston head retaining nuts, etc., to load values higher than normally applied from pressure will give greater fatigue resistance. Excessive preload can create stress-corrosion problems.

60.86 CYLINDER DESIGN PROBLEMS (CONT)

w. Teflon Seals Avoid using Teflon piston seals in a system that must operate with air, as blowby can occur.

x. Actuator Sizing Actuator sizes may or may not be sized by limit load criteria. If specified rates are required, larger cylinders may be required to accomplish the work in the desired time. System delta pressure reduces the available (pressure X flow) horsepower and must be accounted for in the design.

y. Unbalanced Cylinders Design for boost effect of unbalanced cylinders. Don't forget effect is compounded when more than one cylinder per surface is used.

z. Lines Hydraulic lines should be designed and developed so as to preclude their being connected to incorrect ports. Use different sizes or locate the ports 90 degrees to each other where possible.

aa. Flight-Control Jamming Flight-control system assemblies should have adequate clearance, guards, or other protection to prevent jamming by a foreign object.

60.87 SERVOACTUATOR DESIGN PROBLEMS

a. Clearances Carefully review detail hardware that is actuated by autopilot engagement or disengagement. It should have ample clearance in all modes and tolerance accumulations.

b. Dashpot Primary flight-control actuators did not incorporate snubbing in design of autopilot engage lockout mechanism because aircraft response was unaffected in flight. It was later determined that audio levels from structural response was unacceptable for passenger comfort on a commercial aircraft.

c. Yaw Damping Yaw damping (SAS) commands can cancel autopilot commands during the initial part of the align maneuver on crosswind landings.

d. Overtorquing of Bellcrank and Control-Rod Bolts Overtorquing of control-system bolts should not result in increased friction. Do not use bolts in control linkage less than 1/4 inch in diameter.

e. Heat and Cold Tests of Power Cylinders Temperature tests should incorporate a check for thermal bind-up of main or servo spools and transducer failures. The transducer can be sensitive to thermal shock.

f. Moisture Trap The design of power-control cylinder valving should prevent moisture from being trapped at the ends of the main control-valve spool.

60.87 SERVOACTUATOR DESIGN PROBLEMS (CONT)

g. Power-Control Actuator Piston Hammer The piston may bottom against internal stops because of autopilot (stability augmentation) input added to the extremes of manual input and rigging and manufacturing tolerances. Erratic stability augmentation input during aircraft taxiing can cause excessive loads from hammering against stops. The unit should be designed and tested for this condition.

h. Dual Power-Control Cylinders Common walls between power-control systems should be avoided.

i. Dual Hydraulic Flight Controls Avoid areas of possible hydraulic interflow between systems.

j. Autopilot Override Provisions Autopilot override devices should be backed up by breakaway linkage or be jam-proof and not be sensitive to system pressure changes.

k. Servovalve Oscillation Design the autopilot electrical system to avoid vibration signals that can cause servovalve oscillation and auxiliary ram seal wear. Electromagnetic interference susceptibility of power-control actuators is important and should be defined in the Specification Control Drawing.

l. Leakage from Seals Subjected to High-Frequency, Small-Amplitude Oscillations Cylinders and valves that are subjected to oscillations from electrohydraulic (transfer valve) control should have seals that will give adequate component life. Auxiliary ram seal failures have been caused by oscillating electrical inputs. Fix history was standard seals, to slipper seals, to cap seals. Consider unit design that allows leakage to go to system return.

m. System Interflow Interflow leakage across tandem valve spools should be considered when evaluating the effects of high-temperature oil, low viscosity, clean oil, and true ΔP.

n. Ground Checkout of Tandem-Powered Actuators Preflight tests are required to be sure both systems are powering.

o. Power-Control Actuator Tandem Valving Tandem cylinder valving should be acceptance-tested for excessive pressure buildup resulting from poor valve phasing or leakage by sleeve static seals. (Shrunk-in sleeves can be used to avoid static seals.) Pressure surges resulting from poor phasing, particularly in unbalanced cylinders, can result in actuator failures. Tests to develop representative surges should be based on realistic system setup and rates. To avoid setting change or breakage, the tandem valving should be adequately connected.

70. VALVES

71. SLIDE VALVE DESIGN

71.01 GENERAL Slide type valves should be used for flow direction and metering control when low operating forces are required and some internal leakage can be tolerated. The valves offer a simple device to accurately phase and/or sequence multi-path hydraulic flows.

71.1 DESIGN GUIDES

71.11 PRIMARY CONTROL VALVES Valves whose reliability is critical for aircraft operation, such as in primary flight controls, require special design consideration. These valves are subject to repeated operation and pressure cycling during aircraft flight. For valves with this type of duty cycle and integrity requirement it is recommended that:

1. The valve sleeve be shrunk fit into the valve housing, eliminating the sleeve-to-housing packing seals.

2. The sleeve should be mechanically caged to provide longitudinal end restraint to prevent sleeve separation in case of a sleeve failure.

3. The housing material should provide compatible coefficients of expansion between the shrunk fit sleeve and the housing.

4. The valve should be designed with a positive flow force tending to shut off the unit in case of an input linkage failure.

5. The valve slide should never bottom - provide stops in the linkage system.

6. For high flow gain modulated valves, the slide should be flow force compensated. At least, adequate slide and sleeve proportions shall be provided in the unit to add flow force balance if required.

7. The valve inlet shall be filtered with a screen, adequate to protect the lap assembly from being jammed by foreign material.

8. Valve slide end chambers must be interconnected with adequate size passages to prevent slide unbalance under all valve operating conditions.

9. It is recommended that sleeve port metering edges be aligned to within 0.000030 inch.

10. The finish of metering surfaces of both spool and sleeve should be 8 microinch (arithmetic average) or better.

71.12 SECONDARY CONTROL VALVES Valves whose reliability is NOT critical for aircraft operation, such as flaps, slats, gear retraction, etc., may be designed to somewhat less stringent criteria. These valves are operated only a few times during flight and are normally not subjected to frequent pressure reversals. For valves with this type of duty cycle, the following is recommended:

1. O-ring packings are used to accomplish the sleeve-to-housing sealing. The O-rings must have backup rings for all high pressure applications and where practicable the O-rings shall have two backups, regardless of pressure.

2. It is desirable to mechanically cage the sleeve to provide longitudinal end restraint in case of a sleeve failure.

3. The housing material may be aluminum-alloy permanent-mold casting or of material with superior strength allowables.

4. Valve slide should never bottom - provide stops in the linkage system or adequate overtravel.

5. Valve slide end chambers must be interconnected with adequate size passages to prevent slide unbalance under all valve operating conditions.

71.13 VALVE SIZING The valve size is a function of the system flow requirement and the pressure drop the system can tolerate. The system flow requirement is dictated by the actuator displacement, the actuator actuating time and the number of actuators controlled by the valve.

$$\text{Valve Flow} = Q(\text{GPM}) = \frac{\text{Actuator Displ (In.}^3\text{) X No. of Actuators}}{\text{Critical Actuating Time (Sec.)}} \text{ X} .26$$

The critical flow for each valve housing port (pressure, extend, retract and return) should be determined using the above method. The valve housing ports should be sized by choosing the NOM. tube OD having a recommended flow for the port being sized.

The flow passage through the housing from each port to the sleeve should have a diameter equal to the "C" diameter specified by MS 33649 for the port (Boss) size (NOM. tube OD) selected.

The next step is to determine the equivalent orifice diameter required for the fluid chamber. Knowing the system requirements and the valve environment the designer must estimate the allowable pressure drop ($\Delta\rho$) across the passages at each sleeve chamber. Using the estimated pressure drop and critical flow determine the equivalent orifice diameter required for each sleeve chamber. The equivalent orifice diameter (d_e) may be found by using the pressure drop through orifice curves found in the hydraulics manual or by using the following formula.

$$d = \left(\frac{K}{C_d} \frac{Q}{\sqrt{\Delta P}} \right)^{1/2}$$

where: d = orifice dia (inches)
k = fluid coefficient = 0.176 $\sqrt{\rho}$ where ρ is the fluid density in lb/cu in.
= 0.0334 for Skydrol LD
= 0.0346 for Skydrol 500
= 0.0311 for MIL-H-5606
Q = fluid flow rate (GPM)
C_d = orifice discharge coefficient (Usually 0.65)
ΔP = pressure drop (PSI)

Once the equivalent orifice diameter (d_e) is determined, proportion the valve fluid passages as shown:

The sleeve chamber flow area (A_{s1}) as shown in Figure 70-1 is equal to X times Y. The spool chamber flow area (A_{sp}) as shown in Figure 70-1 is equal to $\pi/4$ (D_1^2 - D_2^2). Both chamber flow areas, A_{s1} and A_{sp}, should ideally equal four times the critical equivalent orifice area (A_e).

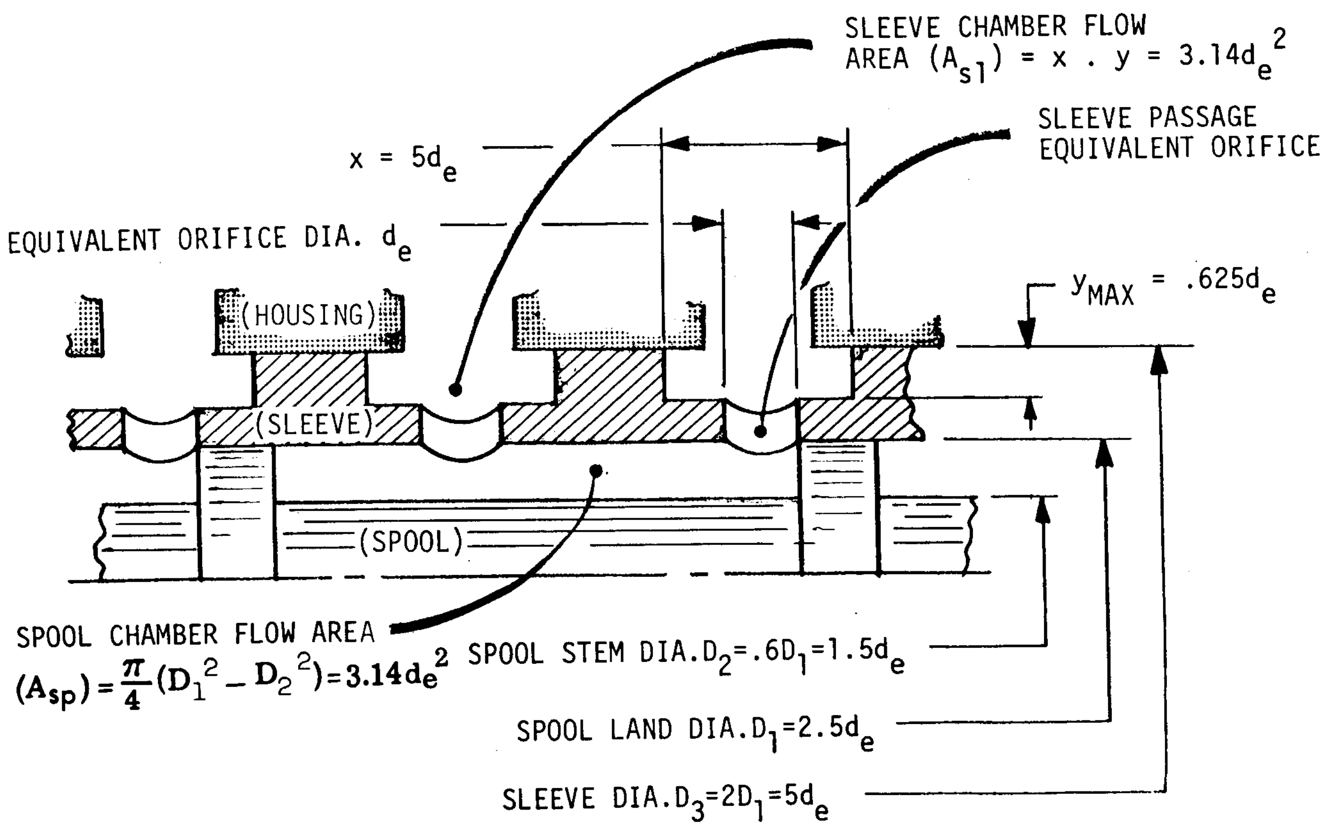


Figure 70-1. Sleeve and Spool Chamber Flow Area Calculations

Passages and chambers having flow areas of the magnitude just discussed will prevent the valve from becoming flow saturated. (Flow saturation is the condition where valve flow becomes restricted before the spool land fully uncovers the fluid passages through the sleeve.)

Where design requirements permit, the valve proportions specified in Figure 70-1 are to be used. However, the requirements (space, stress, weight, etc.) generally will necessitate deviating from the ideal proportions of Figure 70-1.

The actual number and size of fluid passages through the sleeve at each sleeve chamber will depend on the spool stroke and sleeve stress level desired. The number of fluid passages at each chamber should be a minimum of three and preferably as many as possible, equally spaced radially. These equally spaced radial passages will help balance the spool, thereby keeping it centered and keeping friction forces and leakage to a minimum.

It is also desirable to shape the fluid passages through the sleeve at the spool control lands to provide gradual opening and closing (metering) of the passages. This will prevent the occurrence of pressure surges and "water hammer" normally associated with rapid opening and closing of valves. Another benefit of metering is the reduction in the rate of travel of the actuator as it nears its selected position.

The sleeve must be stiff to minimize deflection and prevent binding of the spool as the pressure gradient across the sleeve changes when the spool is displaced. Locate the edges of the spool control lands under or as close as possible to the stiff (thick) section of sleeve. Reinforce the sleeve chamber corners by using generous fillet radii or incline the chamber side walls.

After the valve has been sized and the flow passage and chamber dimensions established, the valve total pressure drop can be calculated, using the following formula for pressure drop (ΔP) through orifices.

$$P = KQ^2$$

Where $K = \dfrac{1}{(kA_{EFF})^2}$

and k = 23.9 for Skydrol 500 and $C_d = 0.65$
24.8 for Skydrol LD and $C_d = 0.65$
26.6 for MIL-H-5606 and $C_d = 0.65$

$$A_{EFF} = \left[\frac{1}{\sum \frac{1}{A_i^2}}\right]^{1/2} = \text{(Effective area of orifices in series)}$$

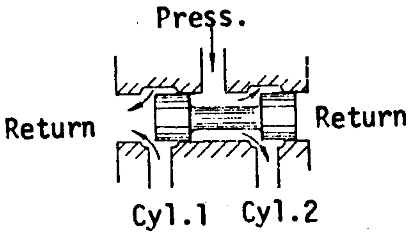
A_i^2 is effective flow area of each flow passage (for flow passage i = 1 through passage i = n) in square inches.

Q = Flow through valve in GPM.

The pressure drop should be calculated for each half the valve loop (pressure to actuator and actuator to return) at various flows and a curve plotted.

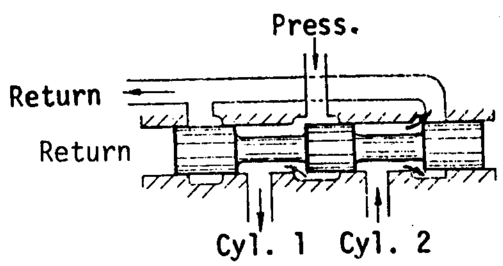
71.14 VALVE CONFIGURATIONS The four-way control valve is the most common and preferred version of directional control valve. The term four-way refers to the number of fluid connections to the valve, pressure, return, and two cylinder ports. The schematics below show a four-way valve having two, three and four spool lands.

71.141 TWO LAND SPOOL The two land unbalanced spool is undesirable for use in a modulated or metered system. This land configuration tends to be unstable because of the thrust load produced to open the valve when the jet exhausts directly into a large chamber. The valve stability or instability will depend on the stiffness of actuating means plus the flow forces. The fluid acquires momentum from the spool as it leaves the sleeve bore, resulting in a lateral force being exerted against the spool. This force tends to open the valve more widely, increasing flow and causing instability. The thrust force is equal to $\frac{q^2 \ell}{A}$ where:

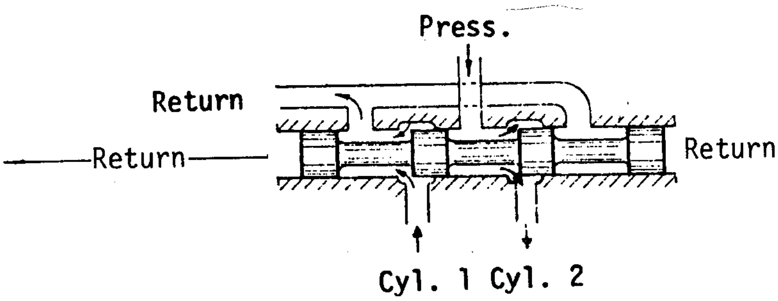


q = flow rate (in.^{3}/sec)
ℓ = fluid density (lb sec^2/in.^{4})
A = spool area (in.^{2})

71.142 THREE AND FOUR LAND SPOOLS



THREE LAND SPOOL



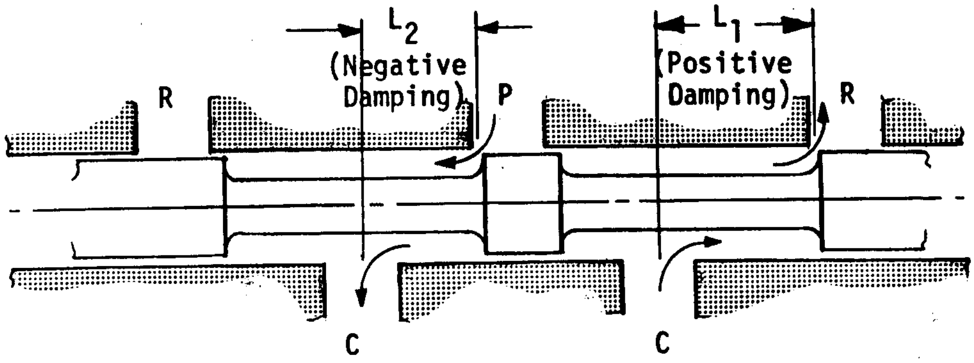
FOUR LAND SPOOL

The balanced three and four land spool configurations will not exhibit the instability described in the preceding paragraph. The sealing land at each end of the spool forces the fluid to exit radially through the sleeve and the momentum of the exhausting fluid no longer exerts a lateral force against the spool.

The four land spool configuration has an advantage over the three land spool in that it is easier to flow force compensate.

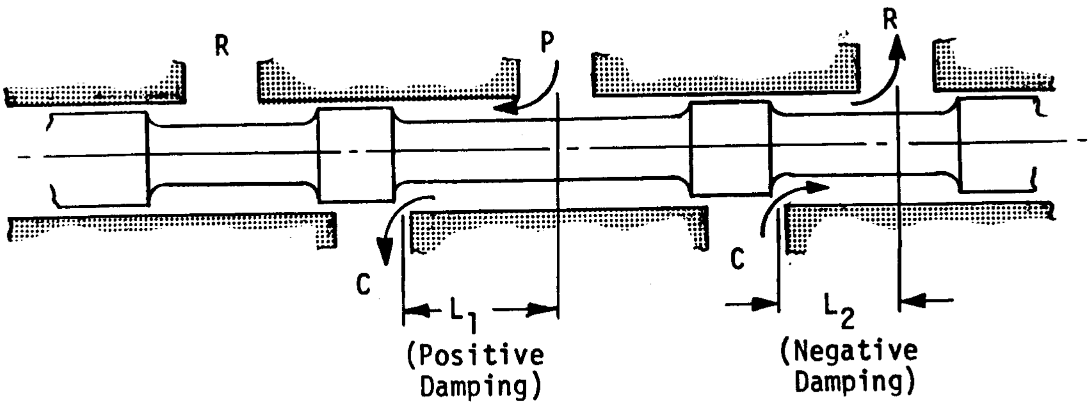
Instability may still occur in three and four land spool valves. The reason, simply stated, is that as the flow through the valve increases, the fluid between the spool lands must be accelerated in a horizontal direction and therefore exerts a force against the spool land. The force is equal to L dq/dt, where L is the horizontal distance between the centers of the incoming and outgoing flows. With fluid flowing outward through the metering orifice, as shown in Figure 70-2, L_1 is positive, the damping is positive and the valve is dynamically stable. With fluid flowing inward through the metering orifice, Figure 70-2, L_2 is negative, the damping negative and the valve is dynamically unstable.

The effective damping length for the whole valve is the algebraic sum of the L's of the separate orifices. Therefore, to have a dynamically stable valve, the positive damping length L_1 must be greater than the negative damping length L_2.



Four-Way, Three Land Valve

Note: For dynamically stable valve, L_1 must be $>$ L_2.



Four-Way, Four Land Valve

FIGURE 70-2. EFFECTIVE DAMPING LENGTH

71.15 <u>VALVE OPERATING FORCES</u> The force that must be overcome to operate the valve is normally due to crank friction, spool force and the valve spring load when applicable.

71.151 <u>CRANK FRICTION</u> Crank friction is a result of packing friction and the friction due to the crank being hydraulically loaded against the crank bushing. To minimize friction, the operating crank should be located in a low-pressure return chamber and the external O-ring dynamic seal should be of minimum size consistent with crank rigidity and strength. Crank friction can be reduced further by supporting the crank shaft in ball bearings installed in the crank bushing. These bearings must be capable of withstanding the thrust load of the hydraulically loaded crank. Packing friction can be decreased by using a slipper ring (Reference Seal and Gland Design Section) between the O-ring packing and crank shaft. Further friction reduction can be a attained if the "slipper" ring incorporates "mini grooves" (a series of radial grooves in the "slipper" ring surface in contact with the crank shaft). Crank packing friction will vary with temperature and pressure, therefore the critical parameters should be chosen for the design.

Figure 70-3 shows typical low friction designs along with crank breakout friction test data:

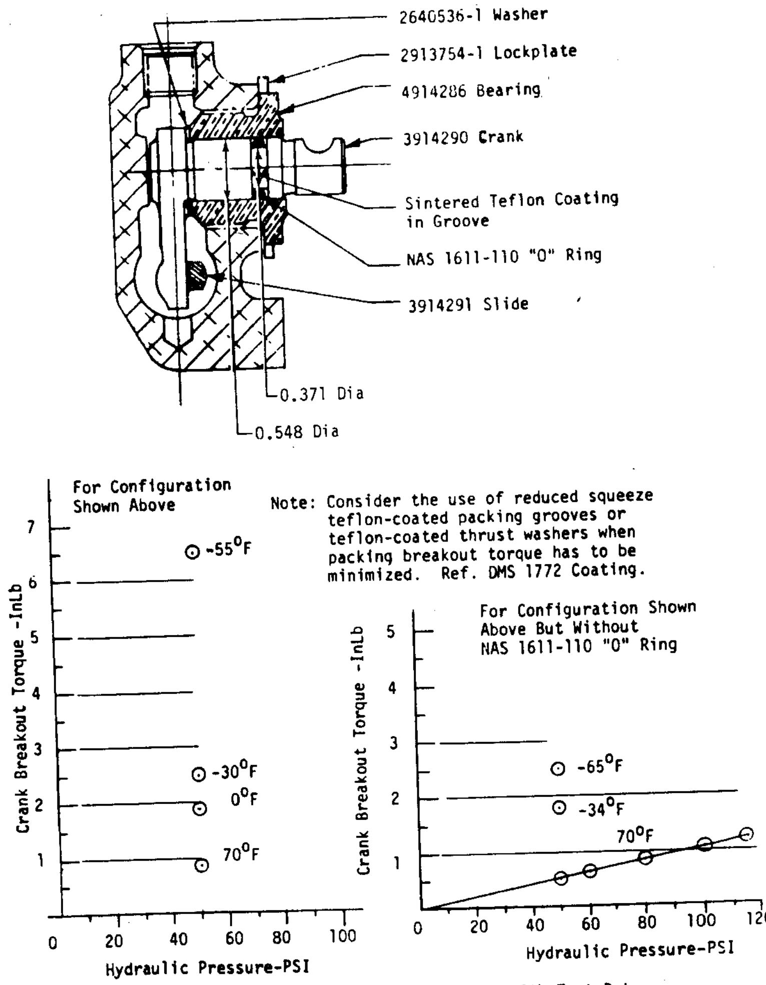
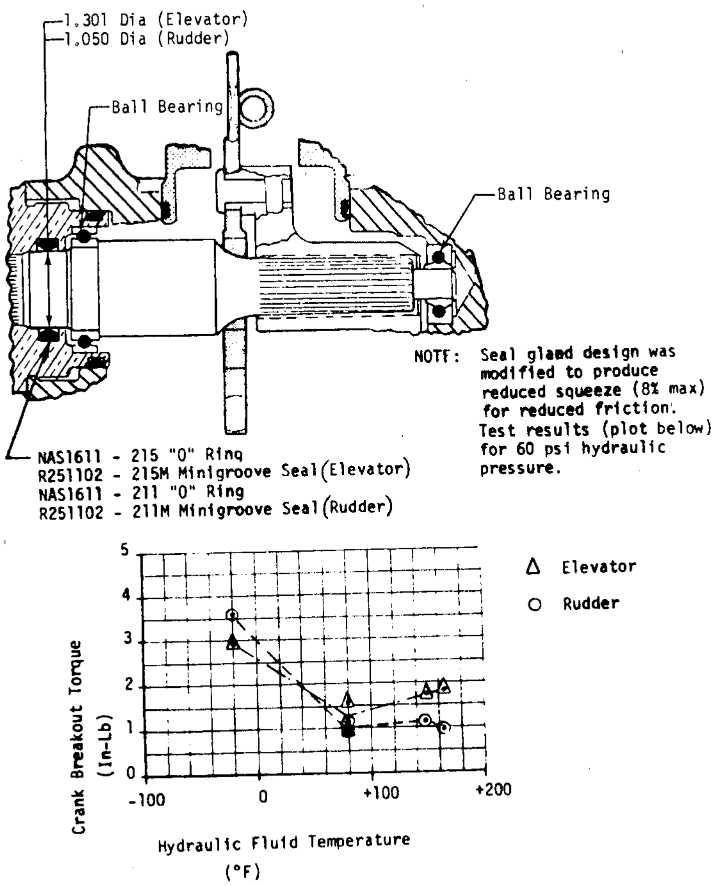


Figure 70-3. Typical Low Friction Design with Test Data

FIGURE 70-3 (CONT'D)

Torque Friction characteristics
DC-10 Elevator & Rudder Power Servo Actuators



71.152 SPOOL FRICTION Spool force, simply stated, is the sum of the static and dynamic forces acting on the valve spool. This section covers only the static forces. Dynamic forces (flow forces, etc.) are covered in later paragraphs. Static forces acting on the spool are caused by hydraulic and/or mechanical forces that eccentrically load the spool, resulting in friction between the spool and sleeve. This static friction, or stiction can be minimized by:

1. Keeping the ratio of spool lapped length to spool land OD to a minimum.

2. Incorporating balancing grooves (radial grooves approximately 0.030 wide by 0.015 deep) along spool land OD.

3. Equally spacing the fluid passages around the sleeve. (Provide the maximum number of fluid passages allowed in keeping with desirable sleeve stress levels.)

The crank should contact the valve spool as close as possible to the spool center. Eccentric crank-to-spool contact will side load the spool and increase the operating force. Crank design must incorporate features that prevent hydraulic pressure and external force from driving the crank against the spool in a manner that would result in undesirable spool side loading.

71.153 SPOOL DYNAMIC FORCES In the detail design of the control valve a complete static balance of hydraulic forces parallel and normal to the slide is desirable. Flow forces and frictional forces only degrade the response characteristic of the system.

With the slide displaced and fluid flowing through the valve the slide becomes unbalanced due to the Bernoulli effect. See Figure 70-4. Flow causes a reduction in pressure at the slide controlling edge where the flow rate is extremely high and in effect wants to resist the opening of the metering orifice.

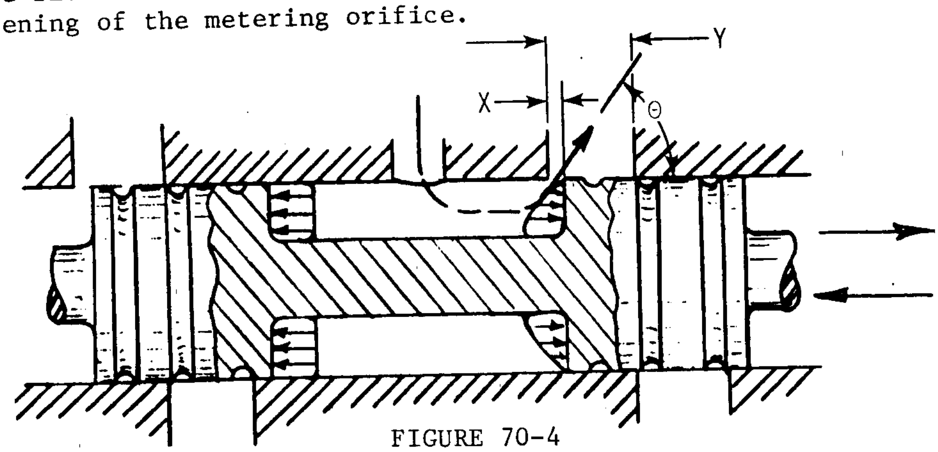


FIGURE 70-4

For manual control valves or valves controlled through a modulating piston this effect is ignored if the flow requirements are in the range of 10 gpm or less. For valves where the driving force is limited or higher flows are required, contouring of the slide passageway reduces this force.

Force Equation:

$$F = 2\ C_q\ W\ \Delta P \sqrt{X^2 + C_r^2}\ \cos\theta$$

where:

F	=	flow force along slide axis
C_q	=	coeff of discharge
W	=	metering port width
X	=	metering port axial length (uncovered)
Y	=	metering port axial length
ΔP	=	pressure drop through metering port
$\cos\Theta$	=	angle between metering port fluid stream and slide axis
C_r	=	slide/sleeve radial clearance

C_r influences cos Θ and increases overall flow force, however for $X/C_r > 40$, X may be neglected. This is true for most aircraft control valves, including those being considered here, therefore F simplifies to:

$$F = 2\ C_q\ WX\Delta P \cos\Theta$$

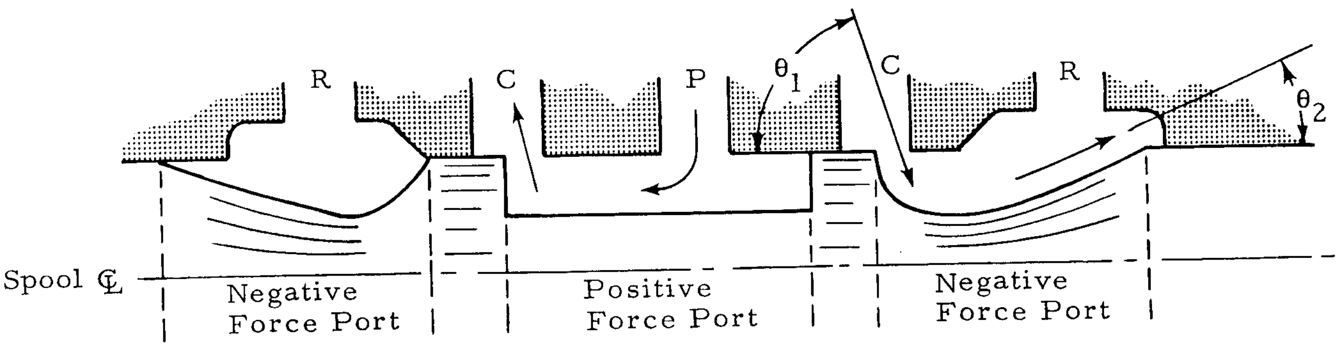
For small values of X compared to other flow path dimensions, $\Theta = 69^{\circ}$. (DC-10 Power Servo Actuator control valve test data indicate that Θ = constant from X = 0 to approximately X = 1/3 Y maximum.)

$$F/X = 2\ C_q\ W\Delta P \cos\Theta = \text{const.}$$

For quick estimating, peak force may be assumed to equal 1 lb per gpm per 3000 psi system with the metering ports at 2/3 to 3/4 of full open. (As an example, for a 30 gpm, 2 system valve, peak flow force may be assumed to be: 1 lb/gpm/sys x 2 sys x 2/3 x 30 gpm = 40 lb.)

71.154 <u>SPOOL DYNAMIC FORCE COMPENSATION</u> Slide valve flow force compensation is achieved by utilizing the dual flow paths occurring in all four-way control valves. Each path is controlled through the slide and sleeve assembly to produce cancelling force effects on the valve slide.

The normal square-edged spool land, when used on both inlet and outlet sleeve ports, will produce a positive force on the spool in a direction to shut off flow. The equation for calculating the positive flow force is shown in the preceding section "spool dynamic forces". By tailoring the spool metering lands in the second flow path a negative force tending to open the valve can be obtained. The positive flow forces are then balancing the negative flow forces. Design data for calculating the negative force port is shown below:



$$F = QU\rho\ (\cos\theta_1 - \cos\theta_2)$$

where: Q = total rate of flow in in.^{3}/sec

$U = \sqrt{\dfrac{(2\Delta P)}{\rho}}$ velocity of jet at vena contracta

ρ = density of fluid in $\dfrac{\text{lb sec}^2}{\text{in.}^4}$

71.16 LAPPED LEAKAGE Lapped leakage across each spool land is determined for the critical spool positions by using the Nomagraph shown in Figure 70-5. The spool overlap (L) used in determining the lapped leakage is not constant for entire spool circumference. The overlap for the width of the fluid passages will be less than the overlap between the fluid passages. The overlap for the width of the fluid passage will depend on the shape of the fluid passage. If the fluid passage is rectangular, the overlap will be the actual spool overlap. If the fluid passage is circular, the overlap is not constant for the width of the passage and some average overlap must be calculated. (For the case of the circular passage, the average overlap would be equal to the actual overlap plus 0.108 times the passage diameter.) A minimum leakage should be determined, assuming a concentric spool and minimum diametral clearance between spool and sleeve. A maximum leakage should be determined, assuming an eccentric spool and maximum diametral clearance between spool and sleeve.

The lapped leakage for each land is the sum of the leakage determined in each of the two following steps.

1. Lapped Leakage at Fluid Passages

   a. Determine lapped leakage from Figure 70-5, using average overlap for width of fluid passage.

   b. Multiply leakage obtained in 1a above by the percent of spool circumference having overlap.

2. Lapped Leakage Between Fluid Passages

   a. Determine lapped leakage from Figure 70-5, using overlap between fluid passages.

   b. Multiply leakage obtained in 2a above by the percent of spool circumference having overlap.

The above procedure should be repeated for each land for each spool position. The total lapped leakage for each spool position can now be determined. If the lapped leakage is greater than desired, the lapped leakage can be decreased by decreasing the diametral clearance, sleeve fluid passage diameter, or spool diameter or by increasing the overlap. Some control valves have a diametral clearance of 90×10^{-6} (0.00009). The actual diametral clearance and overlap will be obtained on assembly with the sleeve by lapping and trimming the spool lands to meet the leakage, spool operating force, and pressure or flow gain requirements specified in the acceptance test procedures.

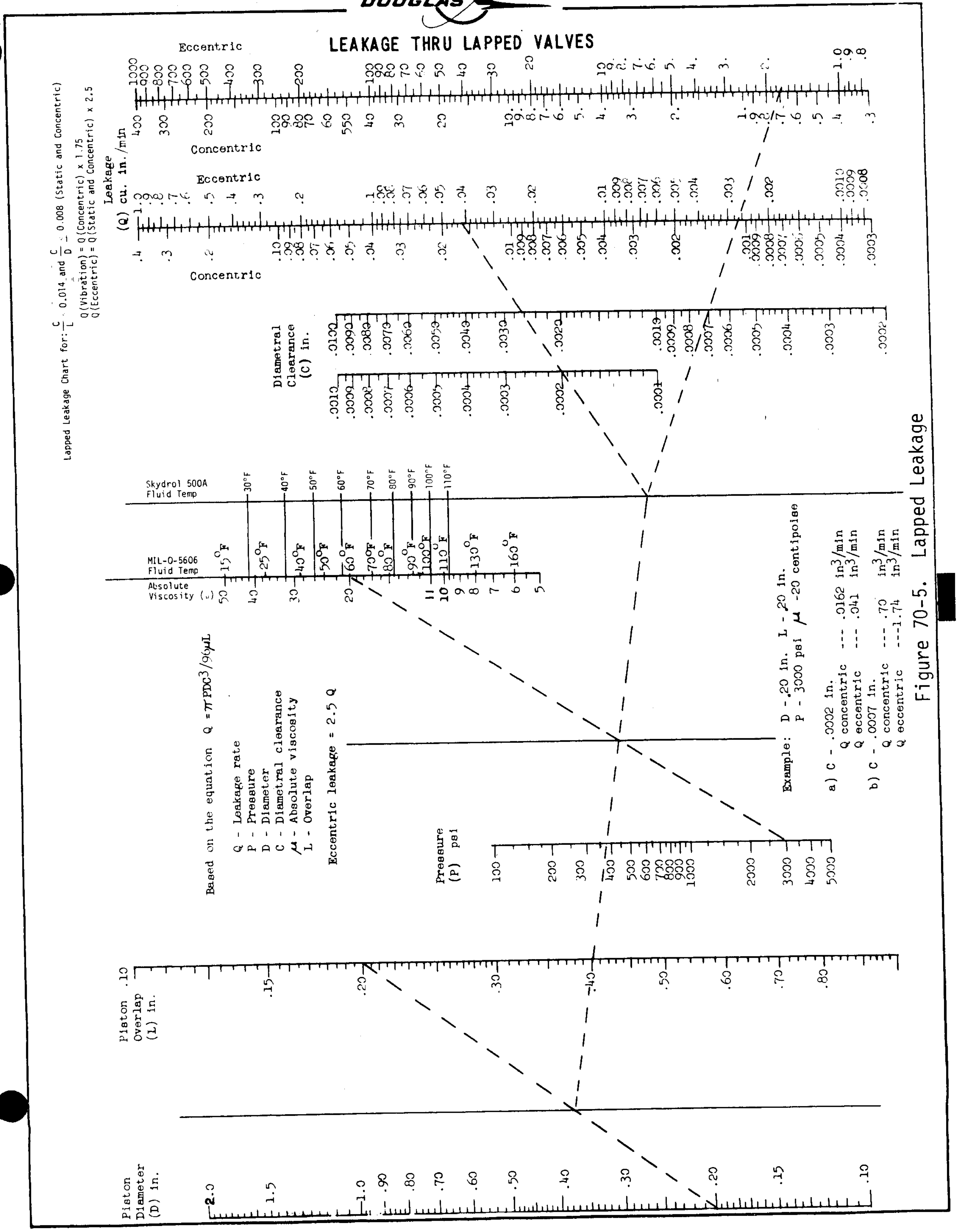


Figure 70-5. Lapped Leakage

71.17 SPOOL LAND LAP CONFIGURATION The spool lands at the control ports can be constructed to provide overlap, underlap or zero lap, as shown in Figure 70-6.

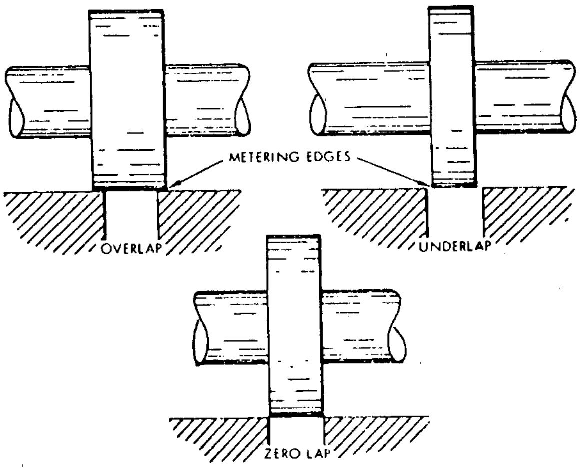
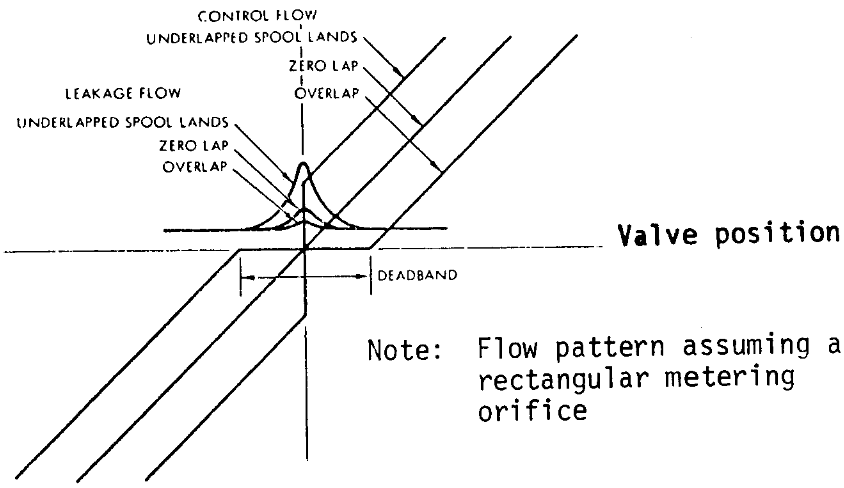


Figure 70-6. Lap Configurations

Zero lap is virtually impossible to manufacture and most valves are designed with a small amount of overlap. This results in a dead band equal to the amount of overlap and a loss of sensitivity near the null position. Null is the spool position where the valve control port pressures are equal (pressure differential across the actuator piston is zero) and the control flow is zero. However, overlap keeps the leakage flow to return to a minimum. Figure 70-7 shows the effect of degree of lap on controlled flow and leakage flow.



Note: Flow pattern assuming a rectangular metering orifice

Figure 70-7. Typical Flow Pattern

Figure 70-8 is a typical pressure versus stroke (pressure gain) curve for a four-way valve having equal overlaps to pressure and to return at null. When the overlap to pressure equals the overlap to return, the control port pressure (P_1 and P_2) is one-half the supply pressure.

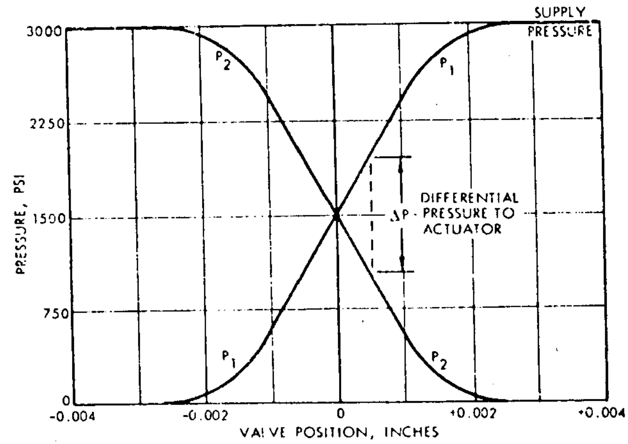


Figure 70-8. Pressure Gain Plot

As the spool is displaced from null, the control port pressure (P_1 and P_2) will change directly as the ratio of pressure overlap to total overlap changes.

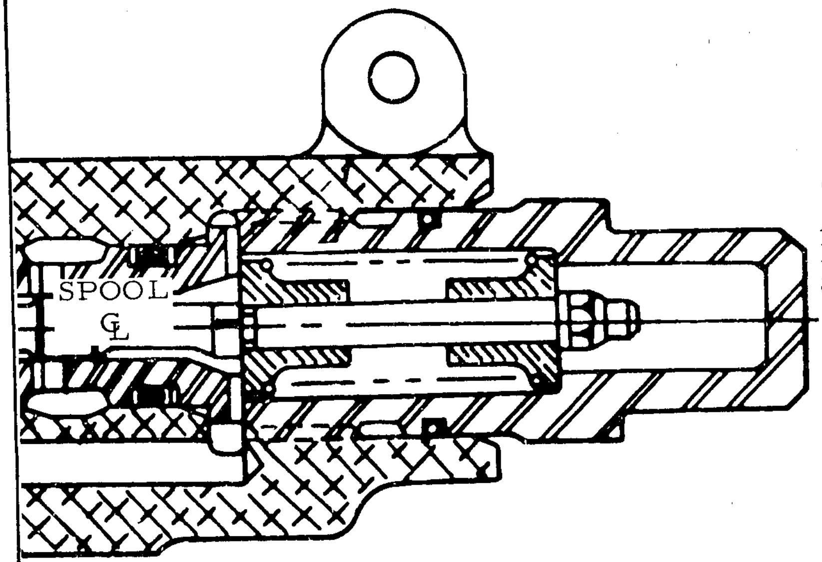
$$\text{Control Pressure} = \text{Supply Pressure}\left(1 - \frac{\text{Pressure Overlap}}{\text{Total Overlap}}\right)$$

As the spool is displaced, in the overlapped region near null, the pressure will increase at one control port and decrease at the other. The pressure differential between the two control ports is the pressure available at the actuator for load. The pressure gain curve traced with the spool moving in one direction will not coincide with the pressure gain curve traced with the spool moving in the opposite direction. The difference between the two curves is known as hysteresis. Hysteresis is the result of spool rotation within the sleeve and the relative position of spool lands to sleeve bore (concentric or eccentric).

71.18 DETAIL CONCEPTS

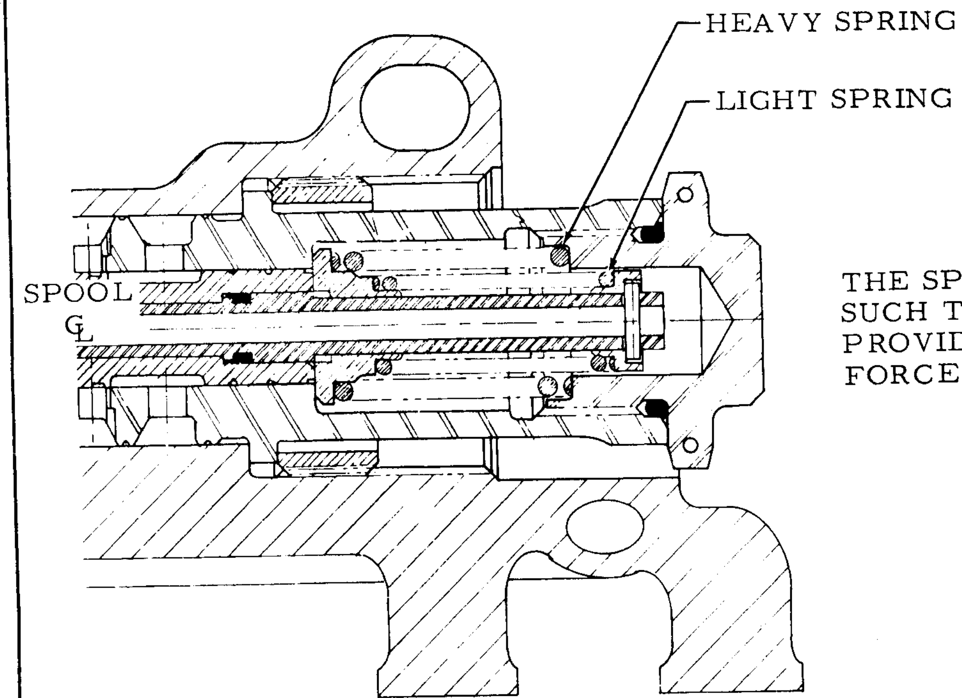
71.181 VALVE SPOOL CENTERING

1. SINGLE SPRING (SMALL NEUTRAL SLOP)



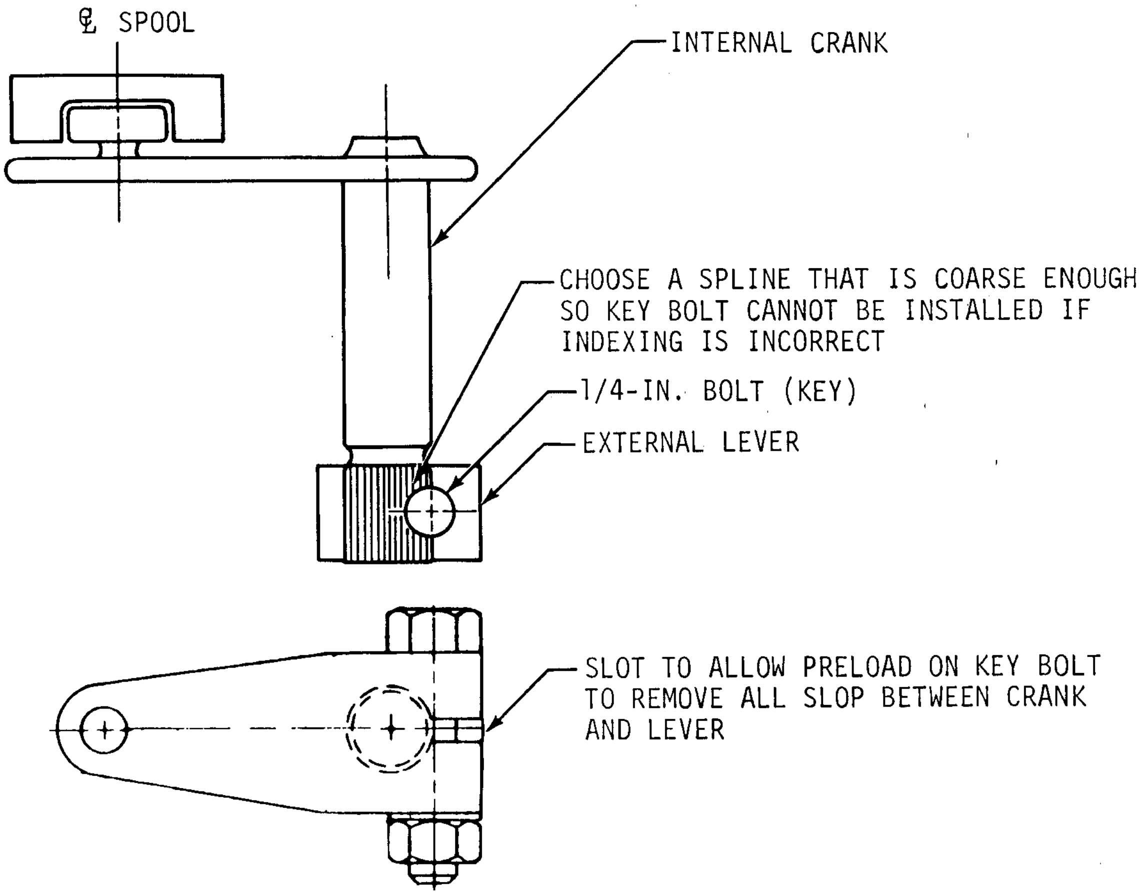
THIS DESIGN MUST BE ADJUSTED AND WILL ALWAYS HAVE A SMALL CENTERING SLOP BAND DUE TO PRODUCTION TOLERANCES.

2. DOUBLE SPRING (NO NEUTRAL SLOP)



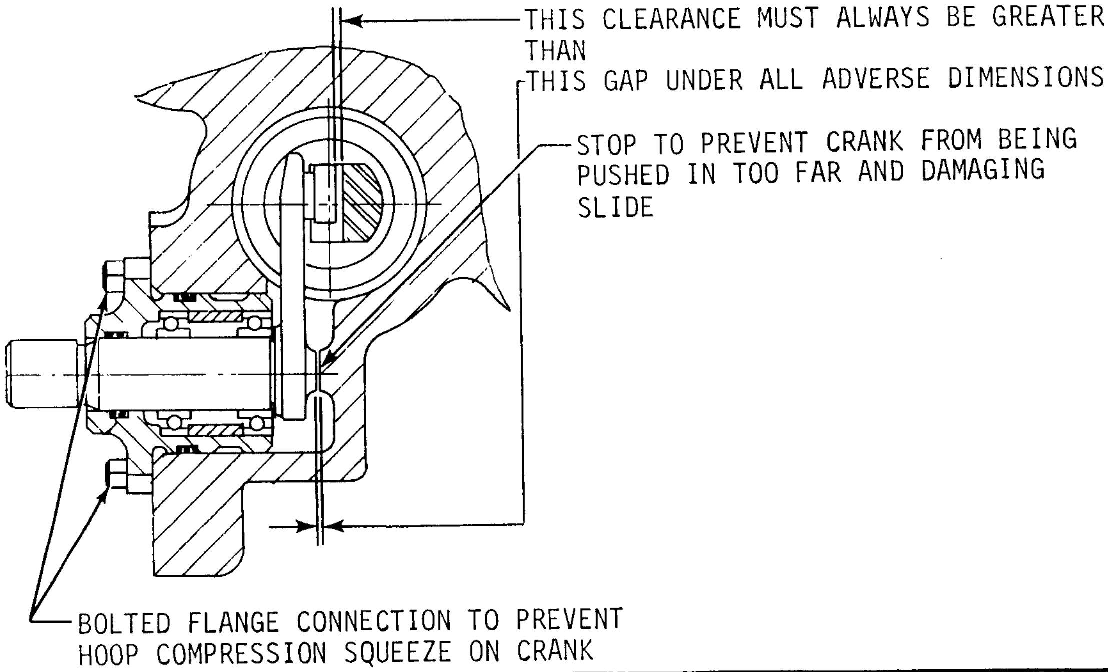
THE SPRINGS ARE SIZED SUCH THAT THEIR SUM WILL PROVIDE THE SAME CENTERING FORCE IN BOTH DIRECTIONS.

71.182 EXTERNAL LEVER ATTACHMENT

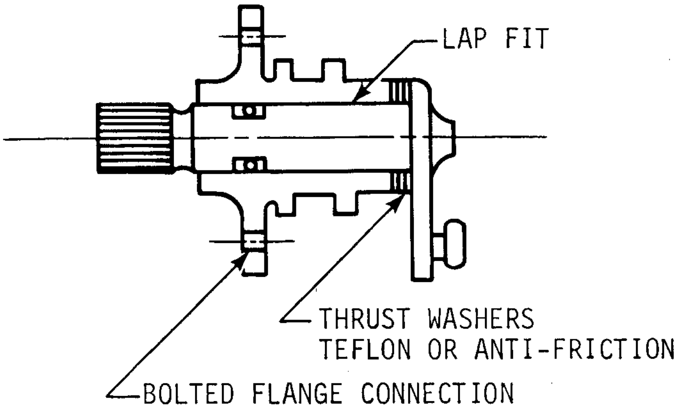


71.183 CRANK BUSHING CONFIGURATIONS

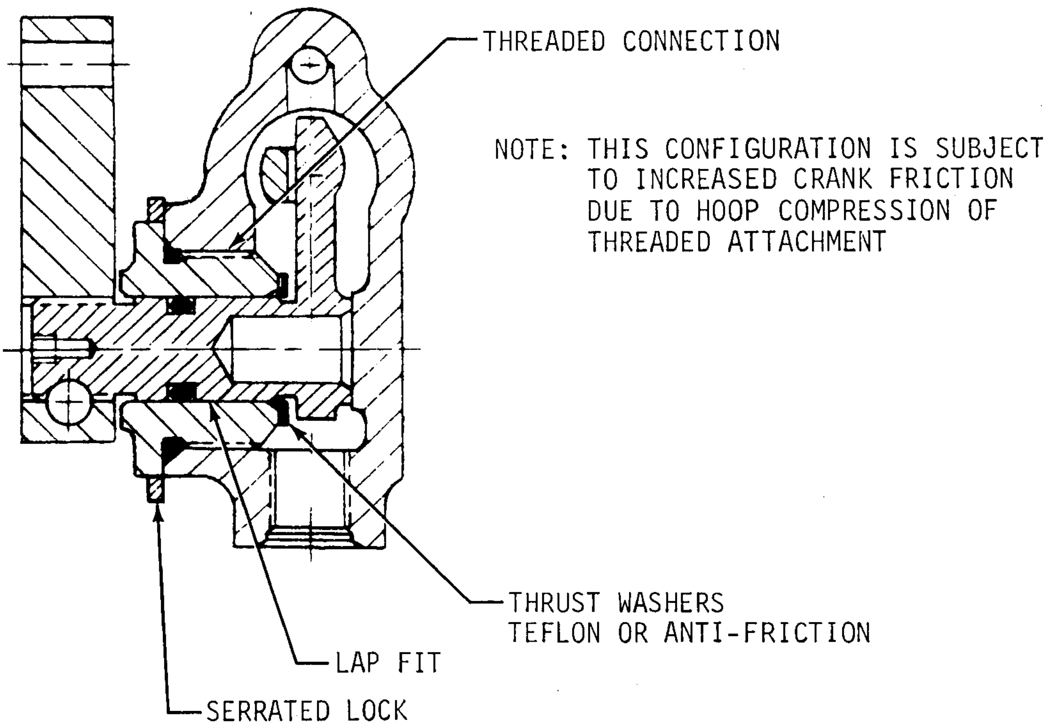
1. Anti-Friction Bearing-Bolted Flange



2. Lap Bearing - Bolted Flange



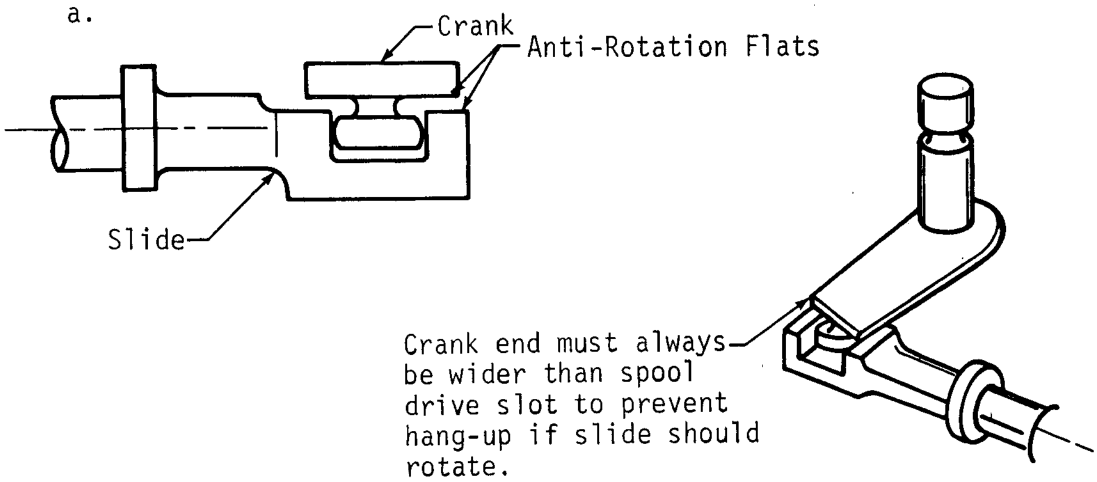
3. Lap Bearing - Threaded Connection



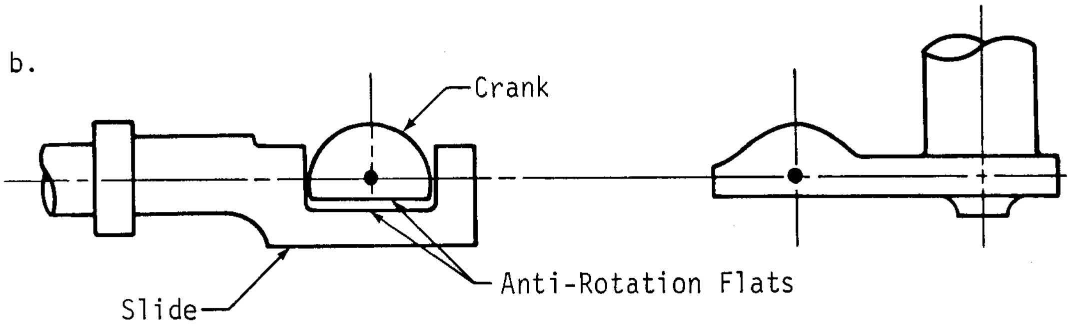
71.184 SLIDE DRIVE CONFIGURATIONS

1. Ball in a slot

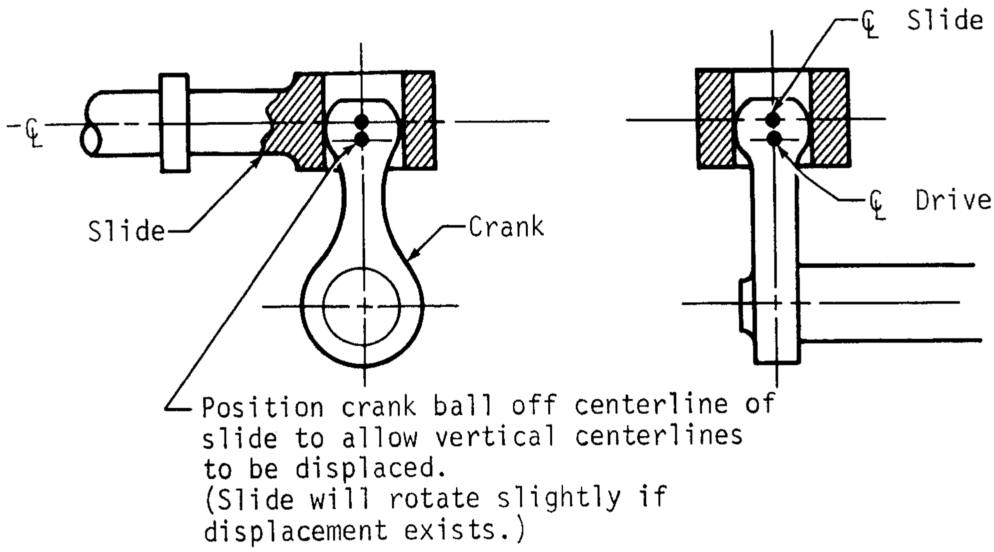
a.



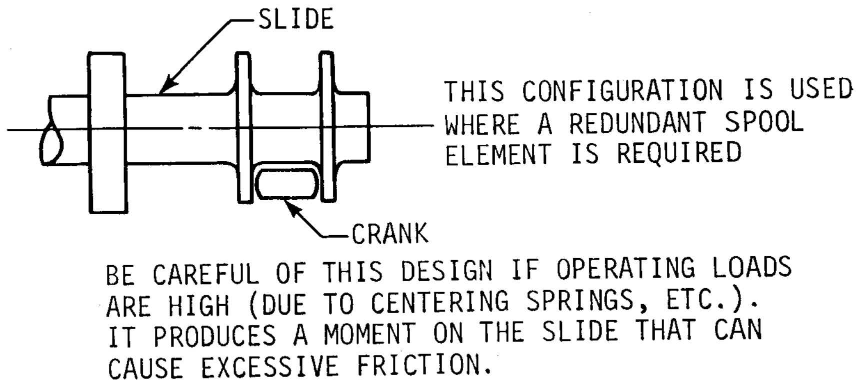
b.



2. Ball in a hole



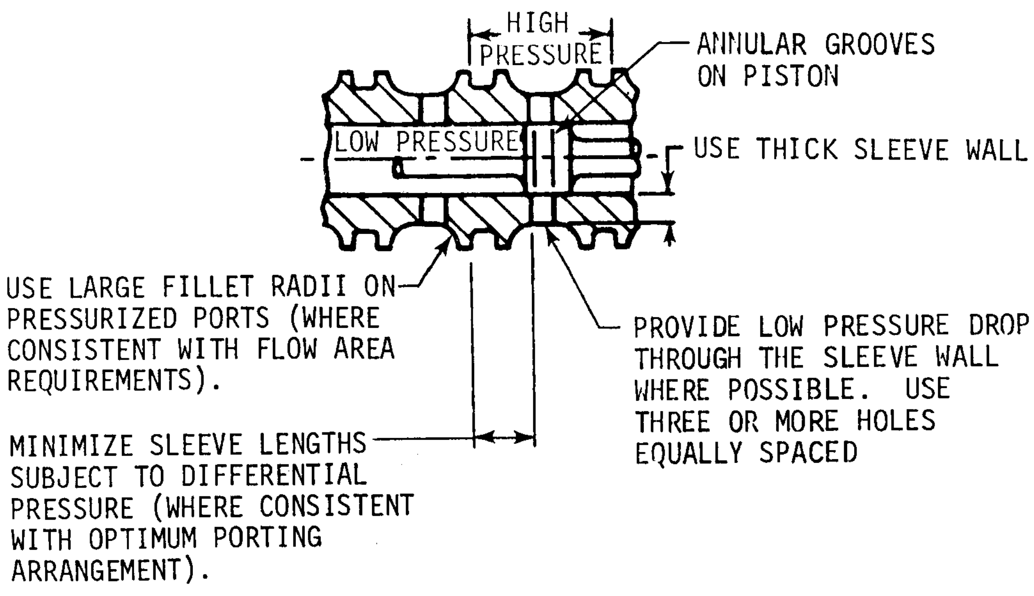
3. Ball in a flange



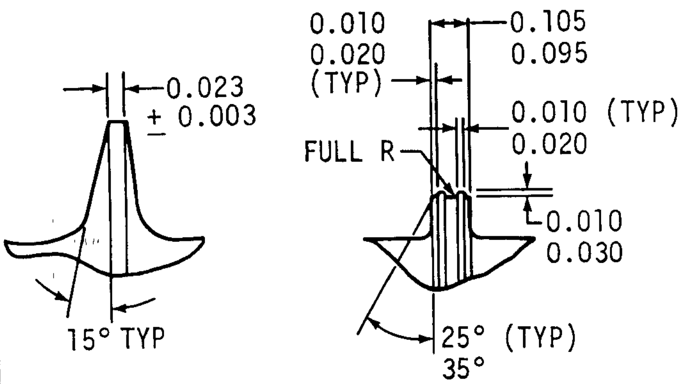
71.185 SLEEVE SQUEEZE-DOWN When pressure on the outside of the sleeve exceeds the pressure on the inside, the sleeve ID contracts, reducing the small (approximately 0.0002) clearance between slide and sleeve. When not properly controlled, this squeeze-down can cause excessive operating force or galling.

This pressure differential occurs with system pressure outside the sleeve adjacent to lower pressure inside. Differential pressure across metering slots in the sleeve wall causes a similar condition.

Means for minimizing squeeze-down are noted below.



71.186 SHRINK FIT SLEEVE CONFIGURATION



Provide a $\overset{32}{\surd}$ finish on sleeve lands and a $\overset{64}{\surd}$ finish on housing bore.

Provide an interference fit between the sleeve housing bore and sleeve land. Shrink fit should be adequate to accommodate expansion of housing due to the most adverse pressure conditions and changes due to differences in coefficients of thermal expansion between the housing and sleeve.

During the manufacturing process prior to lap-fitting the slide to the sleeve, the shrunk fit sleeve-to-housing must be thermal cycled and proof pressure cycled to distortion relieve the assembly. This process will prevent stiction of the lap assembly during service.

71.19 MATERIALS

The materials to be used in the design of control valve components are important in order to avoid certain problems such as:

1. Erosion in the slide and sleeve
2. Wear
3. Siezure or binding of slide and sleeve due to differences in materials and temperature variations.

Through the years certain combinations of materials have been found to give a satisfactory performance and they are to be used in any new design. Special emphasis is placed in any of the moving parts of the control valve as to the materials used. The following is the recommended selection of materials to use.

Part	Usage	Material	Heat Treat
Slide	A	Nitralloy 135M per MIL-S-6709 Consummable vacuum melt (CVM) preferred.	Core Strength = 135 to 155 KSI Nitride 0.-02 - 0.003 Case Depth. Case Hardness = Rockwell 15N - 72 Min.
		52100 Steel Bar (CVM) per MIL-S-7420	Rockwell C-58 to 64(1) per DPS 5.00-1 including Cold Treatment Cycle.
Sleeve	A	52100 Steel Bar (CVM) per MIL-S-7420	
Valve Body	B	2024-T4 Bar or A356-T6 Permanent Mold Casting	
	C	17-4PH Casting (AMS 5343)	Cond. H1000

A = Valves operating in hydraulic fluid. In some special cases where slide extends through a seal into atmosphere, material and/or plating must provide corrosion resistance.

B = Valve with "O" rings in the sleeve

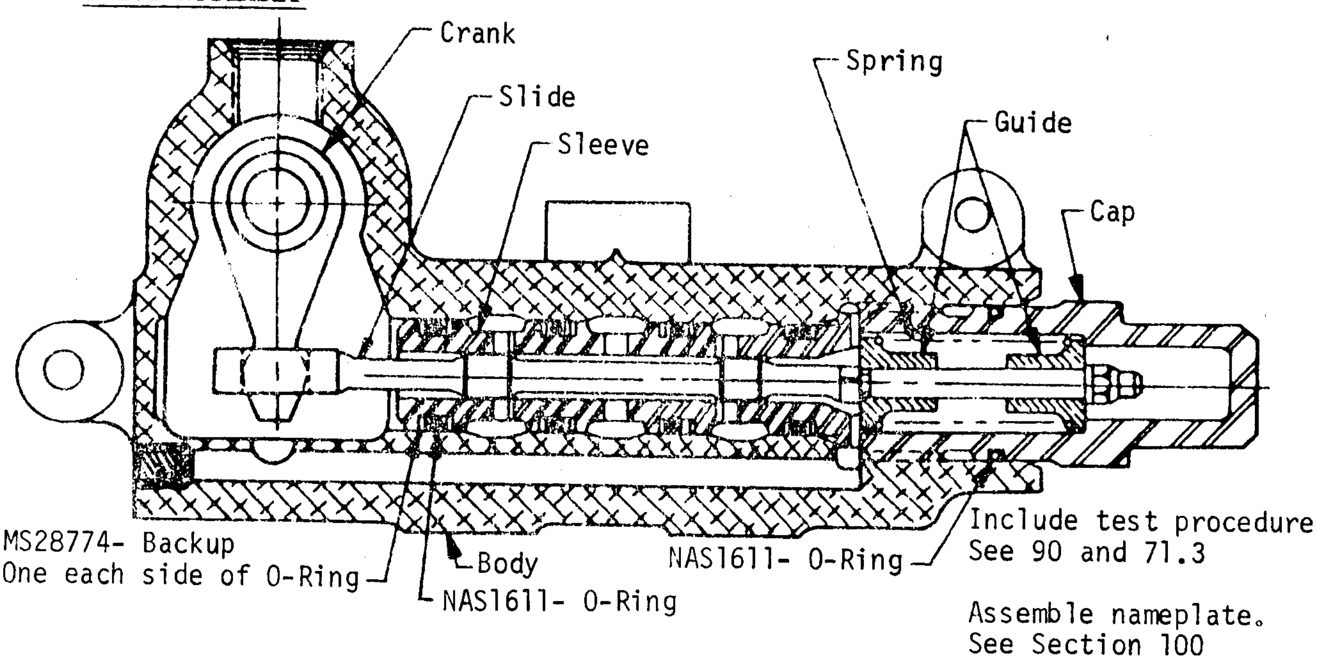
C = Valve with no "O" rings in the sleeve

Notes:

(1) The heat treatment of 52100 slides and sleeves shall be Rc 58-60 where ductility may be required over hardness (wear) considerations, RC 62-64 if wear resistance is desired, but no shock or high stresses are anticipated, or Rc 60-62 for compromise conditions.

71.2 PRODUCTION DRAWINGS

71.21 VALVE ASSEMBLY

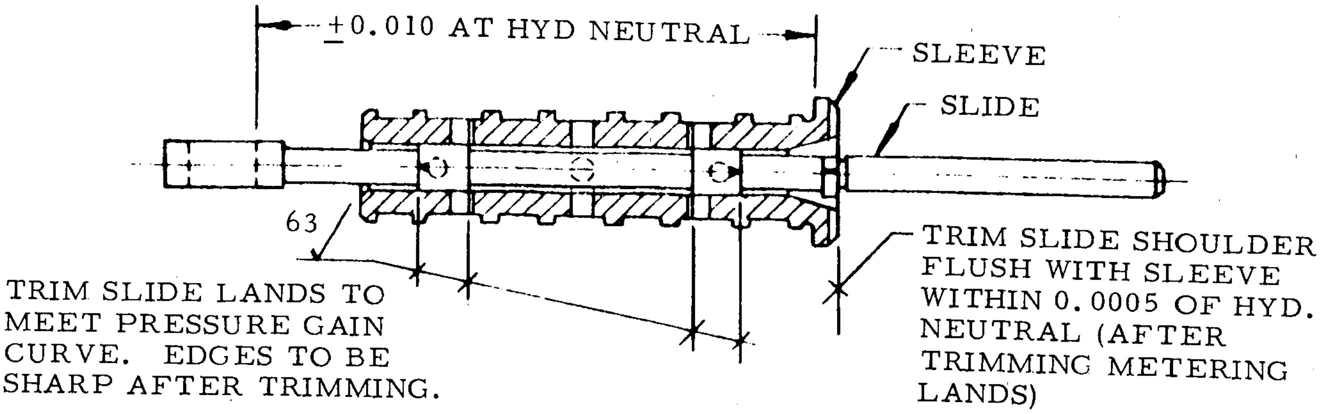


GENERAL NOTES (Typical) UNLESS OTHERWISE NOTED

1. This unit to be manufactured, assembled and tested per DPS 3.391 (for Skydrol units) This unit to be assembled and tested per DPS 3.334 (for mineral oil units)
2. √ indicates surface roughness per MIL-STD-10
3. Apply anti-seize lubricant to threads per DPS 1.22
4. Assembly Shop practice per DPS 2.70 (for standard bolts, standard nuts or cotter pins)
5. Lockwire with MS20995 per DPS 3.651
6. Identify per DPS 3.02
7. Procurement Sources and Quality Control of NAS1611 and NAS1612 O-rings per 7912037 (Use only for drawings with integral material lists. On separate material lists, code each NAS1611 and NAS1612 callout with the following note: Procurement Sources and Quality Control per 7912037)

SLIDE VALVE

71.211 SLEEVE AND SLIDE LAP ASSY



LAP SLEEVE BORE AND SLIDE PER DPS 3.05 TO MEET OPERATING FORCE AND LEAKAGE REQUIREMENTS AS NOTED. (APPROXIMATELY 0.____ TO 0.____ DIAMETRAL CLEARANCE REF.) KEEP AND INSTALL TOGETHER AS LAP ASSY'S. INDIVIDUAL PARTS NOT INTERCHANGEABLE. CAUTION: DO NOT REMOVE MORE THAN 0.0005 FROM SLIDE DIA TO REMOVE TAPER.

0.2111 TYPICAL PRESSURE GAIN PLOT (UNDER LAP CONFIGURATION)

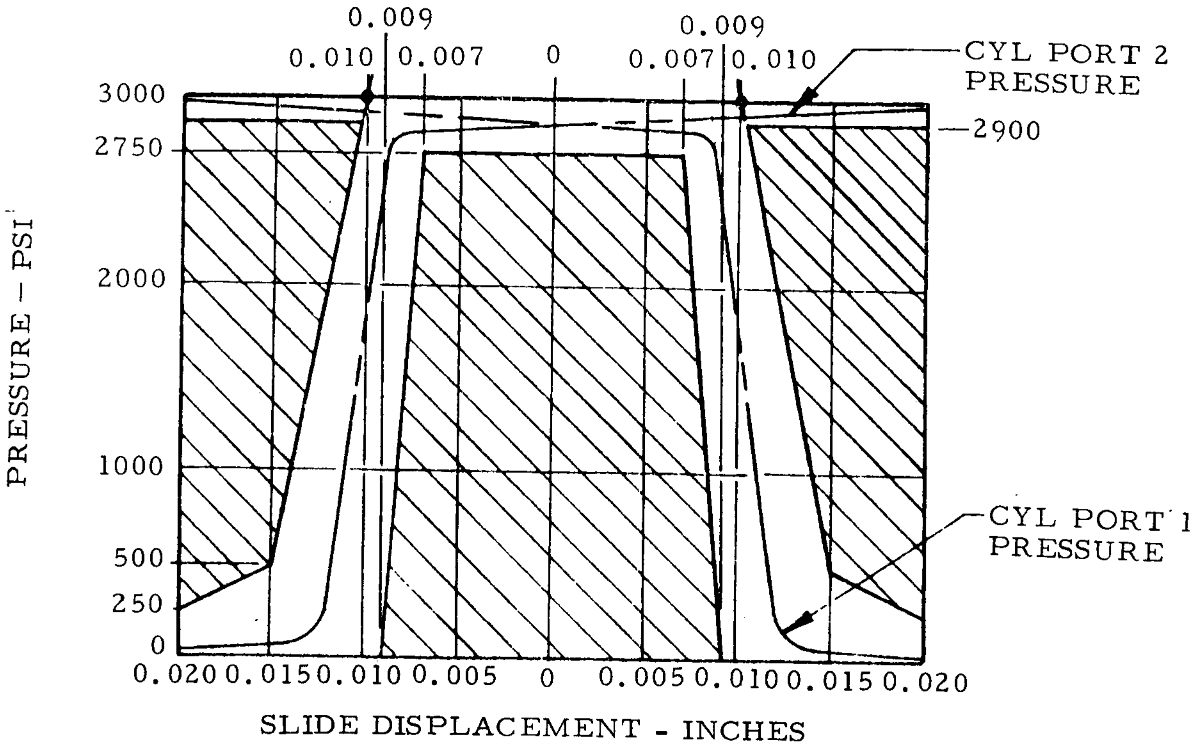
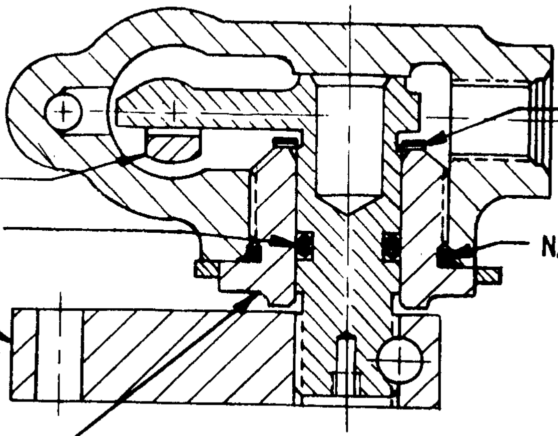
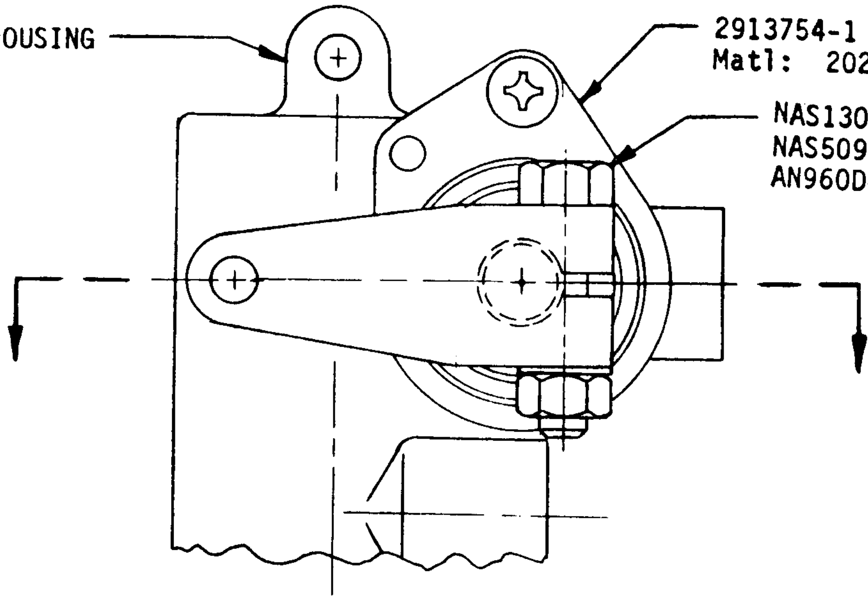


FIGURE 70-9. PRESSURE GAIN PLOT

71.212 CRANK JOINT Dimensions and tolerances are to be such that when the crank is pushed in, the crank must bottom in the housing before the lever can bottom on the bearing. When the crank is pulled out, the crank must bottom against the thrust washers without contacting the slide.

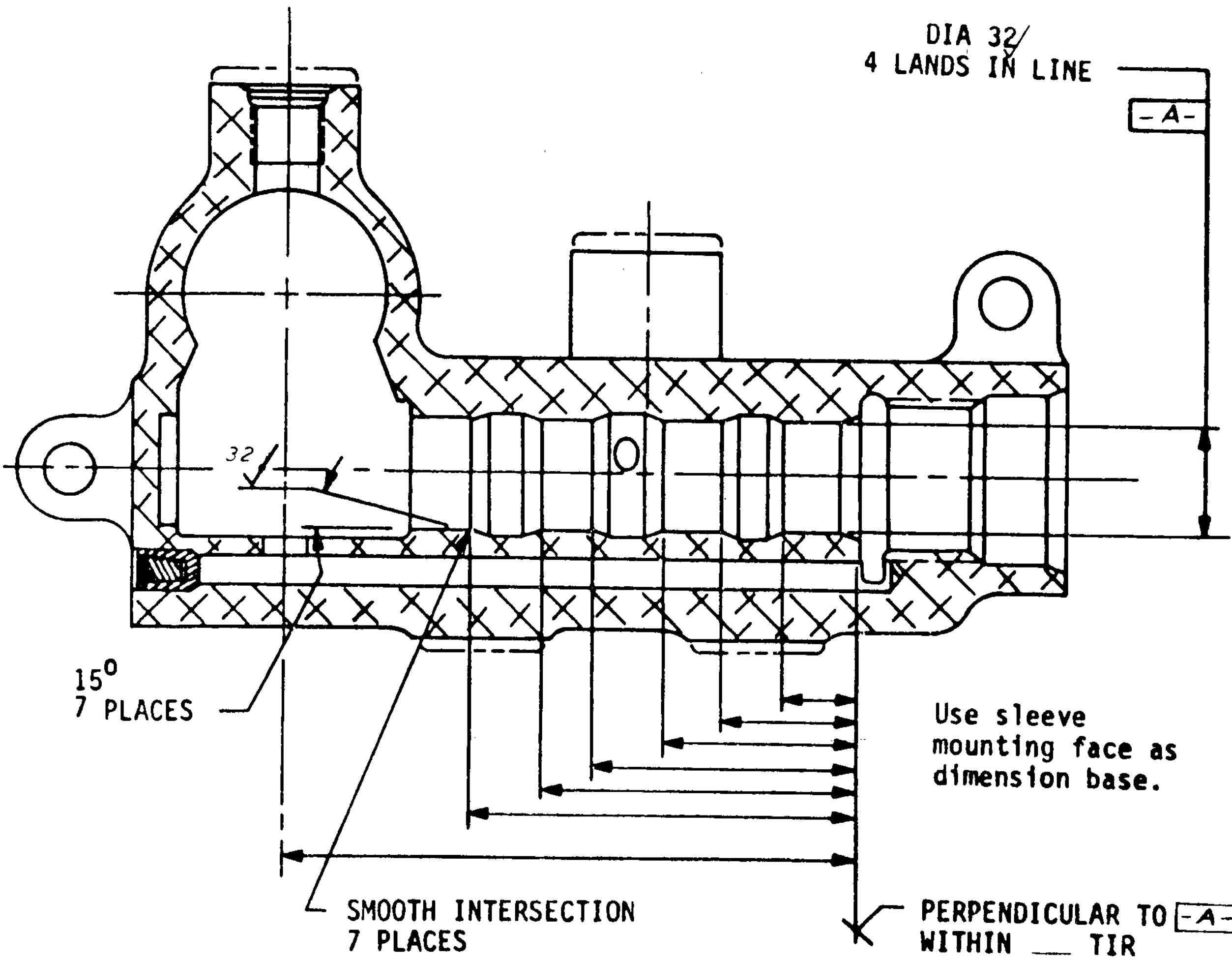




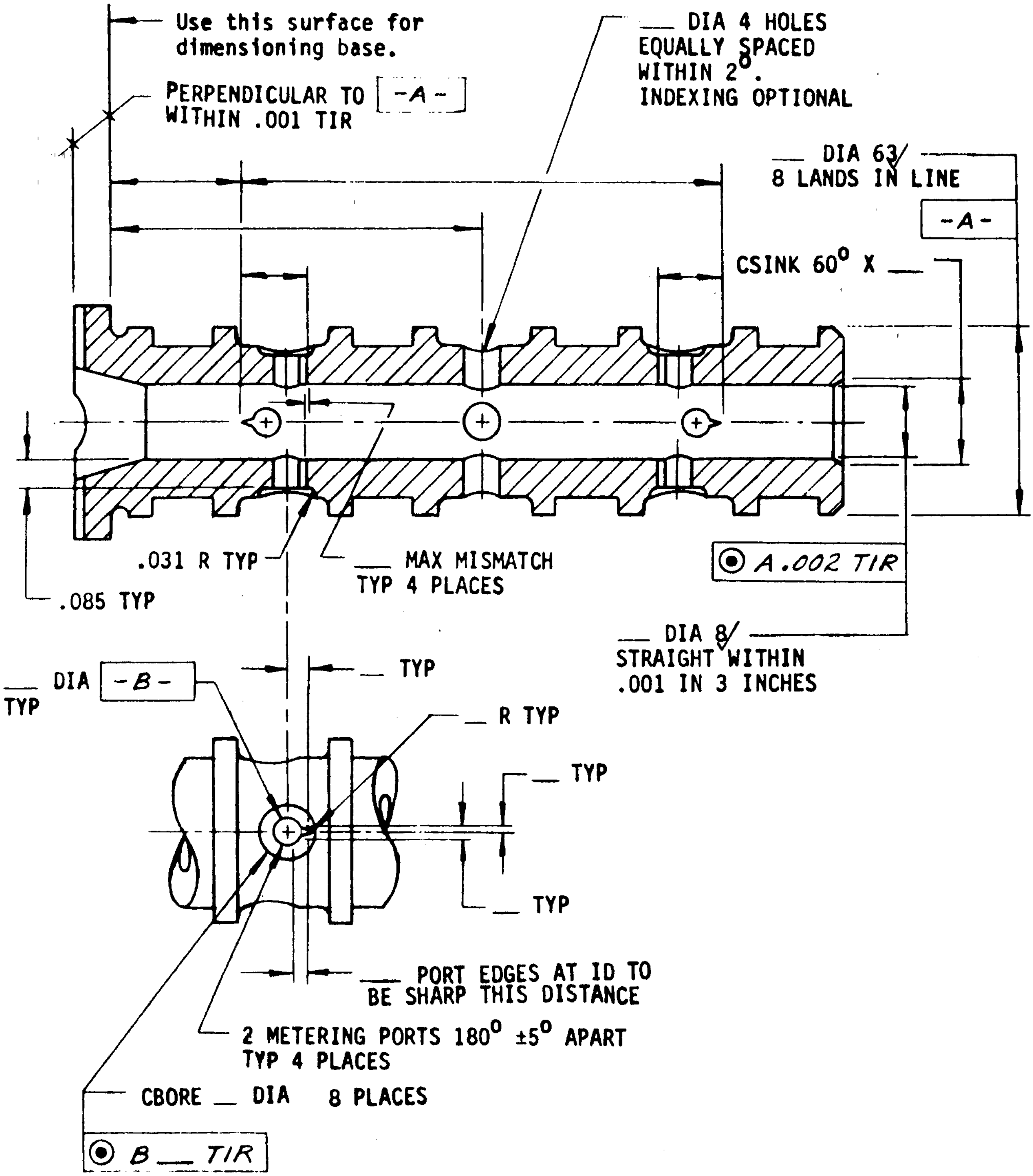
Commonly used part numbers are noted, but designs are not limited to the configuration shown.

71.22 HOUSING For dimensions and notes of aluminum alloy castings and machining, see drawings 5914298 and 5916461 and Section 33 of the Drafting Manual.

Drawing 3641747 is an example steel housing for critical flight control valves.



71.23 SLEEVE A typical sleeve is shown, with some typical dimensions. The actual sleeve configuration, metering slot style, dimensions, and tolerances depend on design.



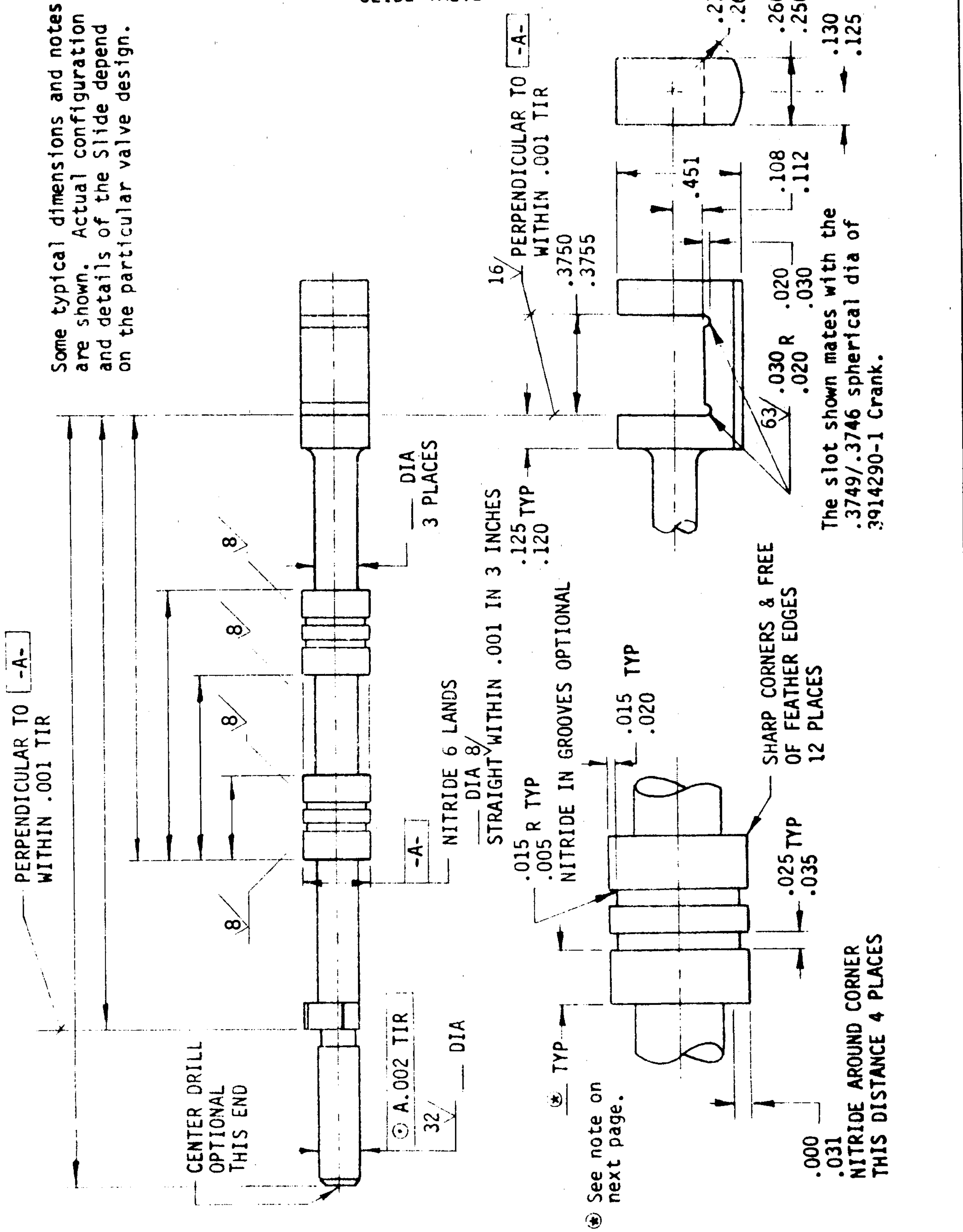
MATERIAL: 52100 Steel Bar

GENERAL NOTES (Typical)
UNLESS OTHERWISE NOTED

1. Machined surfaces 125√ ; holes and spotfaces 125√ ; per MIL-STD-10
2. Fabrication standards per DPS 4.710
3. Identify per DPS 3.02
4. Heat treat (see Paragraph 71.19, Note (1))
5. Remove 0.05 min from each surface after heat treat or protect from decarburization per DPS 5.00
6. Magnetic inspection . . . (block)
7. (Concentric Note)
8. Protect per DPS 3.317. Do not plate
9. Not to be separately furnished as spare. See next assy.

71.24 SLIDE

SLIDE VALVE



MATERIAL: Nitriding Steel Bar per MIL-S-6709

GENERAL NOTES (Typical)
UNLESS OTHERWISE NOTED

1. Machined surfaces $\sqrt{125}$; holes and spotfaces $\sqrt{125}$; per MIL-STD-10
2. Fillet radii 0.005 to 0.015
3. Max mismatch of cuts 0.010
4. Fabrication standards per DPS 4.710
5. Identify per DPS 3.02
6. Remove 0.06 min from each stock surface before nitriding
7. Heat treat and nitride noted surfaces only per DPS 5.00 with controlled white layer and special metallurgical control.
   Core strength 135,000-155,000 psi.
   Case depth 0.002-0.006. Case hardness Rockwell 15N-92 min
8. Magnetic inspection . . . (block)
9. (Concentric Note)
10. Protect per DPS 3.317. Do not plate
11. Not to be separately furnished as spare.
    See next assy

The distance from each slide metering corner to the first pressure-equalizing groove usually should be at least 0.06 inch, after this metering land is trimmed on the next assembly. In most cases this final distance should be reasonably consistent for each corner. Thorough dimensioning control must be used in the Slide and Sleeve detail drawings.

71.3 TESTING Each valve assembly must be tested by the manufacturer (or by Douglas) prior to being accepted by Douglas, to prove that the valve meets operating requirements. The test procedures and required results are specified in the Acceptance Test Procedures. The Acceptance Test Procedures should include at a minimum tests to cover the following items:

1. Slide Friction
2. Proof Pressure Test/External Leakage
3. Crank Packing Leakage (Cycling)
4. Internal Leakage (Neutral and Hardover)
5. Operating Force/Lever Travel
6. Pressure or Flow Gain (as applicable)

Testing may be required at various stages of valve assembly. Test for slide friction and flow or pressure gain are accomplished on the lap assembly (consisting of spool, sleeve and sometimes the housing). It may be desirable to specify a waiting period of several minutes between application of pressure and measurement of spool friction. This will allow silting of the lap assembly. Silting is a build-up (damming) of small particles between the spool land and the sleeve at the upstream end of the land. The small particles are those not removed by the system filters. The test procedure must clearly describe the valve configuration if other than complete assembly. Valve operation and operating force should be checked with operating pressure applied to the valve and the cylinder ports looped (interconnected). This will simulate actual valve operating conditions in the airplane and provide a check on valve stability.

Allowable leakage, minimum and maximum, should be specified in the test procedure.

Figure 70-10 is a typical valve assembly acceptance test procedure.

Note: All acceptance test procedures are to be reviewed and signed by the group test engineer.

MANUFACTURER TO TEST EACH UNIT TO THIS PROCEDURE USING SKYDROL 500B OR 7000 FLUID AT 90° TO 110°F. ALLOWABLE LEAKAGE RATES ARE FOR NEW-CONDITION 500B, APPROX 12.6 CENTIPOISES (12 CENTISTOKES) AT ZERO PRESS. AND NOTED TEMP, AND MUST BE CORRECTED: MULTIPLY RATE BY THE RATIO OF THE ABOVE VISCOSITY TO THE ACTUAL VISCOSITY OF THE TEST FLUID. TEST WITH FLUID CHAMBERS FREE OF AIR.

DATA SHOWING CONFORMANCE TO THE FOLLOWING TESTS SHALL BE SUBMITTED BY THE MANUFACTURER TO ENGINEERING HYDRO/MECH SECTION, DEPT C1-250. THIS DATA SHALL BE OBTAINED FOR EACH UNIT BY MANUFACTURER AND SHALL BE SUBMITTED PRIOR TO OR TOGETHER WITH DELIVERY OF THE PART.

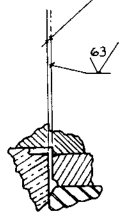
TEST	TEST FOR	METHOD	REMARKS
1	VALVE SLIDE FRICTION	PLUG CYLINDER PORTS C,D,3 AND 4 AND APPLY 3000 PSI AT PRESSURE PORTS A AND 1 OF LAP ASSEMBLY ACG7115. OBSERVE AXIAL FORCE REQUIRED TO MOVE THE SPOOL .125 MIN OFF NEUTRAL IN BOTH DIRECTIONS AFTER A 10 MINUTE WAIT.	THE FORCE SHALL NOT EXCEED 3.0 LBS

PRODUCTION INSPECTION TEST PROCEDURE

TEST NO.	TEST FOR	TEST PRESS	PORTS PRESSURIZED	PORTS PLUGGED	PORTS OPEN	INSTRUCTIONS AND/OR VALVE CRANK POSITION	REQUIREMENTS
1	PROOF STRENGTH AND EXTERNAL LEAKAGE	1500	2 AND B	3 AND 4 C AND D	1 AND A	HOLD CRANK LEVER IN EACH EXTREME POSITION FOR 2 MINUTES	NO EXTERNAL LEAKAGE ALLOWED. NO PERMANENT DEFORMATION ALLOWED.
2		6000	1 AND A		2 AND B	HOLD CRANK LEVER IN EACH EXTREME POSITION FOR 2 MINUTES. SLOWLY MOVE CRANK LEVER FROM FREE POSITION TO EACH EXTREME POSITION.	NO EXTERNAL LEAKAGE ALLOWED. NO PERMANENT DEFORMATION ALLOWED. CRANK TRAVEL SHALL BE SMOOTH WITH NO INDICATION OF BINDING.
3	CYCLING EXTERNAL LEAKAGE	60	2	3 AND 4	1	CYCLE CRANK LEVER FROM FREE POSITION TO EACH EXTREME POSITION AND BACK TO FREE POSITION 25 TIMES.	NO EXTERNAL LEAKAGE ALLOWED EXCEPT A TRACE AT CRANK SHAFT.
4	INTER-SYSTEM LEAKAGE	60	B	C AND D 3 AND 4	A 1 AND 2	CRANK LEVER TO BE IN FREE POSITION	LEAKAGE OUT PORT 2 TO BE 0.02 CC/MIN MINIMUM TO 0.15 CC/MIN MAXIMUM
5 A	INTERNAL LEAKAGE	3000	A AND 1	NOTED	B AND 2	ATTACH PRESSURE GAUGES AND/OR PRESSURE TRANSDUCERS TO PORTS C,D,3 AND 4. ROTATE THE CRANK LEVER IN A COUNTER-CLOCKWISE DIRECTION TO A POSITION WHERE THE PRESSURE AT PORT 4 IS 1400 PSI GREATER THAN THE PRESSURE AT PORT 3 AND THE PRESSURE AT PORT D IS 1400 PSI GREATER THAN THE PRESSURE AT PORT C.	FLOW OUT OF PORTS B AND 2 TO BE BETWEEN 1.5 CC/MIN AND 72 CC/MIN AT EACH PORT
5 B				3 AND 4 C AND D		CRANK LEVER TO BE IN HYD NEUTRAL POSITION.	FLOW OUT OF PORTS B AND 2 TO BE BETWEEN 3 CC/MIN AND 133 CC/MIN AT EACH PORT.
6	OPERATING FORCE AND LEVER TRAVEL	70 ± 10	B AND 2	3 AND 4		ROTATE VALVE CRANK CW & CCW, EACH AFTER A 10 MIN WAIT, TO TEST BREAK-OUT AND RUNNING FORCE FROM HYDR NEUTRAL TO 6° EACH SIDE OF NEUTRAL.	MAXIMUM FORCE SHALL NOT EXCEED 5.4 INCH POUNDS
		3000	A AND 1	C AND D		ROTATE THE VALVE CRANK LEVER TO EACH EXTERNAL STOP POSITION	THE MINIMUM TRAVEL SHALL BE 10° EACH SIDE OF NEUTRAL
						RELEASE THE VALVE CRANK LEVER AFTER DETERMINING TRAVEL	VALVE CRANK LEVER MUST RETURN TO NEUTRAL AND SHALL NOT MOVE WHEN 2.7 INCH POUNDS ARE APPLIED IN EACH DIRECTION

FIGURE 70-10. TYPICAL ACCEPTANCE TEST PROCEDURE

7	PRESSURE GAIN AND HYDRAULIC NEUTRAL POSITION	3000±50	A AND 1	NOTED	B AND 2	ATTACH PRESSURE GAUGES AND/OR PRESSURE TRANSDUCERS TO PORTS C, D, 3 AND 4. ROTATE THE VALVE CRANK EACH DIRECTION FROM NEUTRAL, FOR A DISTANCE OF 0.040 MEASURED AT POINT Y	RECORD CYLINDER PORT PRESSURES VERSUS THE LINEAR TRAVEL OF POINT "Y". THE PRESSURE GAIN CURVE OF CYLINDER PORTS C AND D TO PRESSURE PORT A AND CYLINDER PORTS 3 AND 4 TO PRESSURE PORT 1, SHALL BE WITHIN THE SATISFACTORY REGION SHOWN **BELOW**.
8	DETENT NEUTRAL POSITION					AFTER TRIMMING SLIDE TO MEET PRESSURE GAIN CURVE REQUIREMENTS OF FIGURE 1. ATTACH PRESSURE GAUGES AND/OR PRESSURE TRANSDUCERS TO PORTS C, D, 3 AND 4. TRIM END OF SPOOL AS SHOWN IN VIEW "C" TO OBTAIN "NEW" FREE POSITION. PRESSURE AT PORTS 3 AND C TO BE 3000 PSI AND PRESSURE AT PORTS 4 AND D TO BE 700 ± 300 PSI.	
9	CYLINDER PORT LEAKAGE		D AND 4		1, 2 & 3 A, B & C	AFTER COMPLETION OF TEST #8, CRANK LEVER TO BE IN DETENT NEUTRAL POSITION.	FLOW OUT OF PORTS B AND 2 TO BE BETWEEN 6 CC/MIN AND 180 CC/MIN AT EACH PORT.
10	FLOW	3000	A AND 1	NONE	B & 2	INTERCONNECT PORT C TO D AND 3 TO 4 WITH GAUGES AT EACH PORT AND A HAND VALVE BETWEEN PORTS C AND D, AND BETWEEN 3 AND 4.	FLOW OUT OF PORTS B AND 2 FOR FULL VALVE CRANK TRAVEL IN BOTH DIRECTIONS TO BE 12 GPM MINIMUM.



C 11

SCALE 4/1

Continuation of Acceptance Test Procedure and Pressure Gain Curve

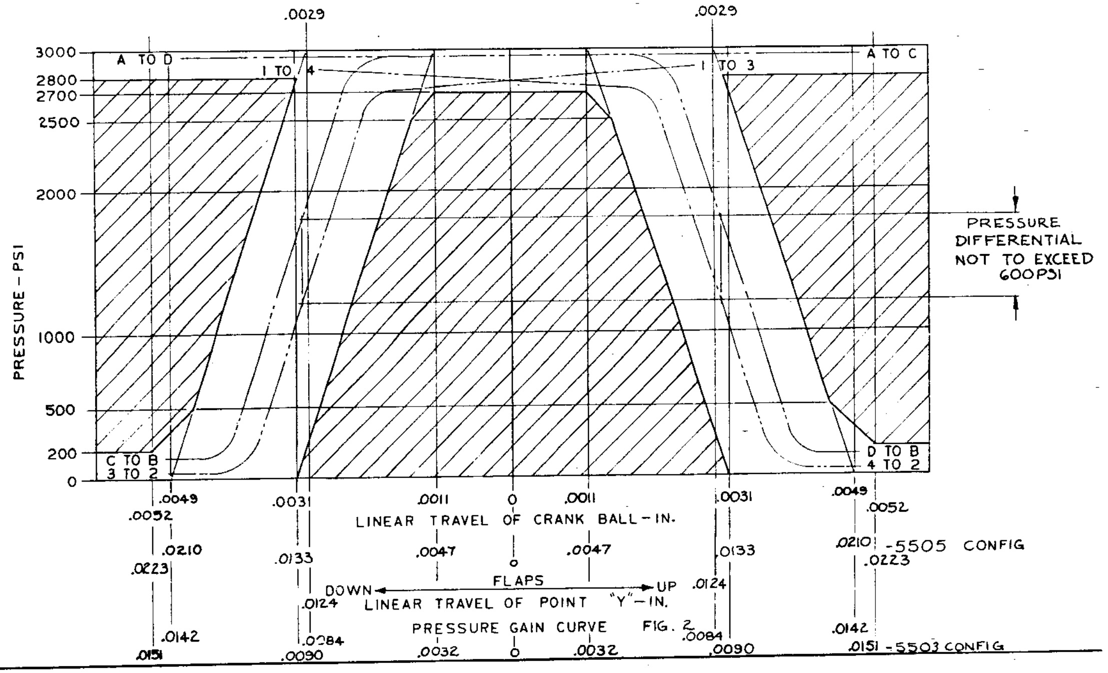


FIGURE 70-10. TYPICAL ACCEPTANCE TEST PROCEDURE (CONT)

72. ELECTROHYDRAULIC VALVES

72.1 RECOMMENDED TERMINOLOGY The recommendations contained in SAE specification: Aircraft Recommended Practices (ARP 490B) are confined to the input and output characteristics of electrohydraulic flow-control servovalves. The information presented should be useful in standardizing the terminology, the specification of physical and performance parameters, and the test procedures used in conjunction with these components.

Servovalve, Electrohydraulic Flow-Control - An electrical input, flow-control valve, which is capable of continuous control.

Hydraulic Amplifier - A fluid valving device which acts as a power amplifier, such as a sliding spool, or a nozzle flapper, or a jet pipe with receivers.

Stage - A hydraulic amplifier used in a serovalve. Servovalves may be single-stage, two-stage, three-stage, etc.

Output Stage - The final stage of hydraulic amplification used in a servovalve.

Port - A fluid connection to the servovalve, e.g., supply port, return port, control port.

Two-Way Valve - An orifice flow-control component with supply and one control port arranged so that action is in one direction only, from supply to control port.

Three-Way Valve - A multiorifice flow-control component with supply, return and one control port arranged so that valve action in one direction opens supply to control port and reversed valve action opens the control port to return.

Four-Way Valve - A multiorifice flow-control component with supply, return, and two control ports arranged so that the valve action in one direction opens supply to control port No. 1 and opens control port No. 2 to return. Reversed valve action opens supply to control port No. 2 and opens control port No. 1 to return.

Electrical Characteristics

Torque Motor - The electromechanical transducer commonly used in the input stages of servovalves.

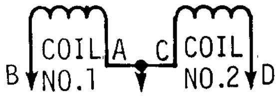
Input Current - The current to the valve, expressed in ma, which commands control flow.

Rated Current - The specified input current of either polarity to produce rated flow, expressed to ma. Rated current must be specified for a particular coil connection (differential, series, or parallel), and does not include null bias current. When coil leads A and C are positive with respect to B and D the following will apply:

72.1 (Continued)

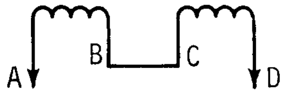
Differential: The net difference between the currents in the two coils in a three or four lead differential coil connection.

4-WIRE



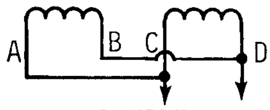
3-WIRE

Series: With the coils in a series, rated flow will be obtained with an input current of one half the rated differential current.



2-WIRE

Parallel: With the coils in parallel, rated flow will be obtained with an input current equal to the differential connection, but will utilize the same power as series connected coils.



2-WIRE

Quiescent Current - A d-c current that is present in each valve coil when using a differential coil connection, the polarity of the current in the coils being in opposition such that no electrical control power exists.

Electrical Quiescent Power - The power dissipation required for differential operation when the current through each coil is equal and opposite in polarity.

Electrical Control Power - The power dissipation required for control of the valve. Control power is a maximum with full input signal, and is zero with zero-input signal. It is independent of the coil connection (series, parallel, or differential) for any conventional two-coil operation. For differential operation, the control power is the power consumed in excess of the electrical quiescent power. This power increase is a result of the differential current change.

Total Electrical Power - The sum of the instantaneous control power and the quiescent power, expressed in mw.

Coil Impedance - The complex ratio of coil voltage to coil current. It is important to note that the coil impedance may vary with signal frequency, amplitude, and other operating conditions due to back emf generated by the moving armature.

Coil Resistance - The d-c resistance of each torque motor coil, expressed in ohms.

Polarity - The relationship between the direction of control flow and the direction of input current.

Dither - A low amplitude, relatively high frequency periodic electrical signal, sometimes superimposed on the servovalve input to improve system resolution. Dither is expressed by the dither frequency (cps) and the peak-to-peak dither current amplitude (ma).

Static Performance Characteristics

Control Flow - The flow through the valve control ports, expressed in cis or gpm. Control flow is referred to as No-Load Flow when there is zero load-pressure drop. Control flow is referred to as Loaded Flow when there is load-pressure drop. Conventional test equipment normally measures no-load flow.

Rated Flow - The specified control flow corresponding to rated current and specified load pressure drop. Rated flow is normally specified as the no-load flow.

Flow Curve - The graphical representation of control flow versus input current. This is usually a continuous plot of a complete cycle between plus and minus rated current values.

Normal Flow Curve - The locus of the midpoints of the complete cycle flow curve, which is the zero hysteresis flow curve. Usually valve hysteresis is sufficiently low, such that one side of the flow curve can be used for the normal flow curve.

Flow Gain - The slope of the control flow versus input current curve in any specific operating region expressed in cis/ma or gpm/ma. Three operating regions are usually significant with flow-control servovalves: (1) the null region, (2) the region of normal flow control, and (3) the region where flow saturation effects may occur. Where this term is used without qualification, it is assumed to mean normal flow gain.

Normal Flow Gain - The slope of a straight line drawn from the zero flow point of the normal flow curve, throughout the range of rated current of one polarity, and drawn to minimize deviations of the normal flow curve from the straight line. Flow gain may vary with the polarity of the input, with the magnitude of load differential pressure and with changes in operating conditions.

72.1 (Continued)

Rated Flow Gain - The ratio of rated flow to rated current, expressed in cis/ma or gpm/ma.

Flow Saturation Region - The region where flow gain decreases with increasing input current.

Flow Limit - The condition wherein control flow no longer increases with increasing input current. Flow limitation may be deliberately introduced within the servovalve.

Symmetry - The degree of equality between the normal flow gain of one polarity and that of the reversed polarity. Symmetry is measured as the difference in normal flow gain of each polarity, expressed as percent of the greater.

Linearity - The degree to which the normal flow curve conforms to the normal flow gain line with other operational variables held constant. Linearity is measured as the maximum deviation of the normal flow curve from the normal flow gain line, expressed as percent of rated current.

Hysteresis - The difference in the valve input currents required to produce the same valve output during a single cycle of valve input current when cycled at a rate below that at which dynamic effects are important. Hysteresis is normally specified as the maximum difference occurring in the flow curve throughout plus or minus rated current, and is expressed as percent of rated current.

Threshold - The increment of input current required to produce a change in valve output, expressed as percent of rated current. Threshold is normally specified as the current increment required to revert from a condition of increasing output to a condition of decreasing output.

Internal Leakage - The total internal valve flow from pressure to return with zero control flow (usually measured with control ports blocked), expressed in cis or gpm. Leakage flow will vary with input current, generally being a maximum at the valve null (null leakage).

Load-Pressure Drop - The differential pressure between the control ports, expressed in psi. In conventional three-way servovalves, load-pressure drop may be expressed as an equation, wherein it is equated to the supply pressure, less return pressure, and less the pressure drop across the single active control orifice. (Ps-Pr-Po = Pl).

72.1 (Continued)

Valve Pressure Drop - The sum of the differential pressures across the control orifices of the output stage, expressed in psi. Valve-pressure drop will equal the supply pressure minus the return pressure minus the load pressure drop.

Pressure Gain - The rate of change of load pressure drop with input current at zero control flow (control ports blocked), expressed in psi/ma. Pressure gain is usually specified as the average slope of the curve of load pressure drop versus current between $\pm 40\%$ of maximum load-pressure drop.

Null Region - The region about null wherein effects of lap in the output stage predominate.

Null - The condition where the valve supplies zero control flow at zero load-pressure drop.

Null Pressure - The pressure existing at both control ports at null, expressed in psi.

Null Bias - The input current required to bring the valve to null, excluding the effects of valve hysteresis, expressed as percent of rated current.

Null Shift - A change in null bias, expressed as percent of rated current. Null shift may occur with changes in supply pressure, temperature and other operating conditions.

Lap - In a sliding spool valve, the relative axial position relationship between the fixed and movable flow-metering edges with the spool at null. For a servovalve, lap is measured as the total separation at zero flow of straight line extensions of the nearly straight portions of the normal flow curve, drawn separately for each polarity, expressed as percent of rated current.

Zero Lap - The lap condition where there is no separation of the straight line extensions of the normal flow curve.

Overlap - The lap condition which results in a decreased slope of the normal flow curve in the null region.

Underlap - The lap condition which results in an increased slope of the normal flow curve in the null region.

Dynamic Performance Characteristics

Frequency Response - The complex ratio of flow-control flow to input current as the current is varied sinusoidally over a range of frequencies. Frequency response is normally measured with constant input current amplitude and zero load pressure drop, expressed as amplitude ratio, and phase angle. Valve frequency response may vary with the input-current amplitude, temperature, supply pressure, and other operating conditions.

Amplitude Ratio - The ratio of the control-flow amplitude to the input-current amplitude at a particular frequency divided by the same ratio at the same input-current amplitude at a specified low frequency (usually 5 or 10 cps). Amplitude ratio may be expressed in decibels where db = $20 \log_{10}$ AR.

Phase Lag - The instantaneous time separation between the input current and the corresponding control-flow variation, measured at a specified frequency and expressed in degrees (time separation in seconds x frequency in cps x 360 degrees per cycle).

72.2 <u>A METHOD FOR DETERMINING THE OPTIMUM SERVOVALVE FOR EACH HYDRAULIC SERVOSYSTEM</u> The problem of properly matching a servovalve and actuator to its load is very important. If the servovalve is too small, the system may become velocity limited during its duty cycle and have poor dynamic response. When the system operates under this condition, a large portion of the pressure drop occurs across the servovalve instead of across the actuator connected to the load, and little useful work is done. The power expended in the servovalve heats the oil excessively, which is undesirable and inefficient. If the actuator is too small for its load, the system may be force limited during its duty cycle. Somewhere between these extremes exists the optimum relationship between the servovalve, the actuator, and the load.

It can be shown mathematically that for any valve configuration operating at any supply pressure the maximum power the valve can deliver to the load occurs when the load differential pressure equals two-thirds of the supply pressure. Neglecting losses, the remaining one-third of the system pressure drop occurs across the servovalve.

The efficiency of the hydraulic system may be improved by using a somewhat larger servovalve than indicated by the maximum power to the load theory. The improved system efficiency is achieved by increasing servovalve size, weight, and, probably, cost. Also there will be an increase in all the system factors that contribute to system drift because the valve will not ever be working at its full capacity. Most systems have a combination of load requirements, such as several force conditions superimposed upon a velocity condition. Fortunately, these may be analyzed by the superposition principle. If the servovalve can supply sufficient power to the load for the most adverse combination of load requirements, then the lesser system requirements will also be satisfied and the design will be satisfactory. The following procedure is offered to help engineers determine the optimum servovalve for each hydraulic servosystem.

Three load conditions are analyzed: the first at the point of maximum load force, the second at maximum velocity, and the third at maximum power (maximum product of force times velocity). Three corresponding points are plotted on a nomograph (Figure 70-11), which indicates the correct servovalve for each of these load conditions. The optimum valve for the system will be the largest one indicated.

To proceed, plot the total system force (dynamic plus other forces) and corresponding velocity as a function of time for the complete load duty cycle.

Most load forces and velocities are easily determined. However, when the load is driven at high frequencies and/or the load has high inertia, the dynamic forces become appreciable. Therefore, these forces need to be determined and plotted. Assuming a pure inertia load driven sinusoidally:

72.2 A METHOD FOR DETERMINING THE OPTIMUM SERVOVALVE FOR EACH HYDRAULIC SERVOSYSTEM (CONT)

The dynamic force at any time is given by

$$F_D = M\omega^2 \text{ a sin } \omega t$$

where

F_D = dynamic force, pounds

M = mass, lb sec^2/in.

a = amplitude, inches (±the maximum load displacement from mid-position)

ω = angular velocity, radians/sec ($2\pi f$)
(f = frequency, cps)

a sin ωt = displacement, inches

This function reaches a maximum where $\omega t = \pi/2$ (±90° from mid-position).

The dynamic velocity is given by

$$V_D = a\omega\cos\omega t$$

where

V_D = dynamic velocity, in./sec

This function reaches a maximum where $\omega t = \pi$ (0° or 180°, at mid-position.)

In order to use the nomograph, the supply pressure (P_S), the load differential pressure (P_L), and the load flow (Q) must be known for each of the three load conditions outlined above.

Referring to the force plot, determine the maximum load force. Assume a supply pressure and assume further that the differential pressure across the load actuator is equal to supply pressure; then calculate the minimum actuator area from the following equation:

$$A_{min.} = \frac{F_{max.}}{P_S}$$

where

$A_{min.}$ = min. actuator area, in.^{2}

$F_{max.}$ = max. load force, pounds

P_S = supply pressure, psi

In order that the valve may properly control the load under maximum force conditions, assume an actuator area 20 percent larger than the minimum area calculated above.

72.2 A METHOD FOR DETERMINING THE OPTIMUM SERVOVALVE FOR EACH HYDRAULIC SERVOSYSTEM (CONT)

Calculate P_L for the load condition of maximum force:

$$P_L = \frac{F_{max.}}{A}$$

where

P_L = load differential pressure, psi

Referring to the velocity plot, determine the velocity which corresponds to the load condition of maximum force and calculate Q:

$$Q = 0.26\ VA$$

where

Q = flow, GPM

V = velocity, in./sec

With the assumed P_S and the calculated P_L and Q for the load condition of maximum force, plot this point on the nomograph.

Referring to the velocity plot, determine the point of maximum velocity and its corresponding force. Calculate P_L and Q for the load condition of maximum velocity. With the assumed P_S, plot this point on the nomograph.

Referring to the plots of force and velocity, determine the maximum product of these load conditions (point of maximum power). Calculate P_L and Q for this load condition, and plot this point on the nomograph. The optimum valve for the system will be the largest one indicated.

If the servovalve indicated does not yield a reasonable system efficiency, especially at the condition of maximum power to the load, assume a larger valve and/or supply pressure and repeat the procedure outlined above.

If the system consists of a pure inertia load only, it is not necessary to plot the dynamic load force and velocity. Also the correct actuator area can be calculated to match this unique load.

It can be shown that the maximum power required to drive a pure inertia load sinusoidally occurs when $\omega t = \pi/4$ ($\pm 45^o$ from mid-position). Using the equation for F_D, dynamic force, and substituting, we have

$$F_D = \frac{Ma\omega^2}{1.4}$$

Using the equation for V_D, dynamic velocity, and substituting, we have

$$V_D = \frac{a\omega}{1.4}$$

72.2 A METHOD FOR DETERMINING THE OPTIMUM SERVOVALVE FOR EACH HYDRAULIC SERVOSYSTEM (CONT)

Next choose a suitable P_L/P_S ratio (abscissa of the nomograph). If it is desired to determine the smallest servovalve that would drive this load, P_L/P_S should be 0.67. The final point that determines the size of the required valve will lie on this vertical line on the nomograph.

Assume a system pressure and calculate the actuator area, A, from the following equation:

$$A = \frac{F_D}{P_L} = \frac{F_D}{0.67\ P_S}$$

Obtain the value of Q by substituting A in the following equation:

$$Q = 0.26\ V_D A$$

Using the vertical scales on the left of the nomograph, and the values of P_S and Q, determine the point on the selected P_L/P_S ratio line that indicates the optimum valve size.

Neglecting system losses, the maximum dynamic power required to drive a pure inertia load sinusoidally is given by

$$HP_{max.} = \frac{Ma^2\omega^3}{13,200}$$

where

$HP_{max.}$ = horsepower

This point occurs where $\omega t = \pi/4$ ($\pm 45^{\circ}$ from mid-position).

Also the power to the load in a more general sense is given by

$$HP_L = \frac{QP_L}{1714}$$

The total system hydraulic power is given by

$$HP_S = \frac{QP_S}{1714}$$

To analyze a system with a hydraulic motor, plot the motor torque requirements in inch-pounds and the motor shaft speed in radians per second as a function of time, and substitute the following equations for those shown in the preceding text.

$$(F_D) \qquad T_D = I\omega^2\theta \sin \omega t$$

where

T_D = dynamic torque, inch-pounds

72.2 A METHOD FOR DETERMINING THE OPTIMUM SERVOVALVE FOR EACH HYDRAULIC SERVOSYSTEM (CONT)

θ = amplitude, radians ($\pm$ maximum motor rotation from mid-position)

$\theta \sin \omega t$ = angular displacement, radians

$$(V_D) \qquad W_D = \theta\omega\cos\omega t$$

where

W_D = dynamic motor velocity, radians/sec

$$(A_{min.}) \qquad D_{min.} = \frac{T_{max.}}{P_S}$$

where

$D_{min.}$ = motor displacement, in.^{3}/radian

$$(P_L) \qquad P_L = \frac{T_{mxa.}}{D}$$

$$(Q) \qquad Q = 0.26\ DW$$

where

W = motor velocity, radians/sec

To analyze a system with a hydraulic motor driving a pure inertia load, substitute the following equations for those shown above:

$$(F_D) \qquad T_D = \frac{I\omega^2\theta}{1.4}$$

$$(V_D) \qquad W_D = \frac{\omega\theta}{1.4}$$

$$(A) \qquad D = \frac{T_D}{P_L} = \frac{T_D}{0.67\ P_S}$$

$$(Q) \qquad Q = 0.26\ W_D D$$

$$(HP_{max.}) \qquad HP_{max.} = \frac{I\omega^3\theta^2}{13,200}$$

HP_L and HP_S equations in the text may be applied without modification to all rotary systems.

In order to keep system losses at a reasonably low value, the hydraulic lines should be as short as possible and the oil velocity should be less than 20 ft/sec at all times.

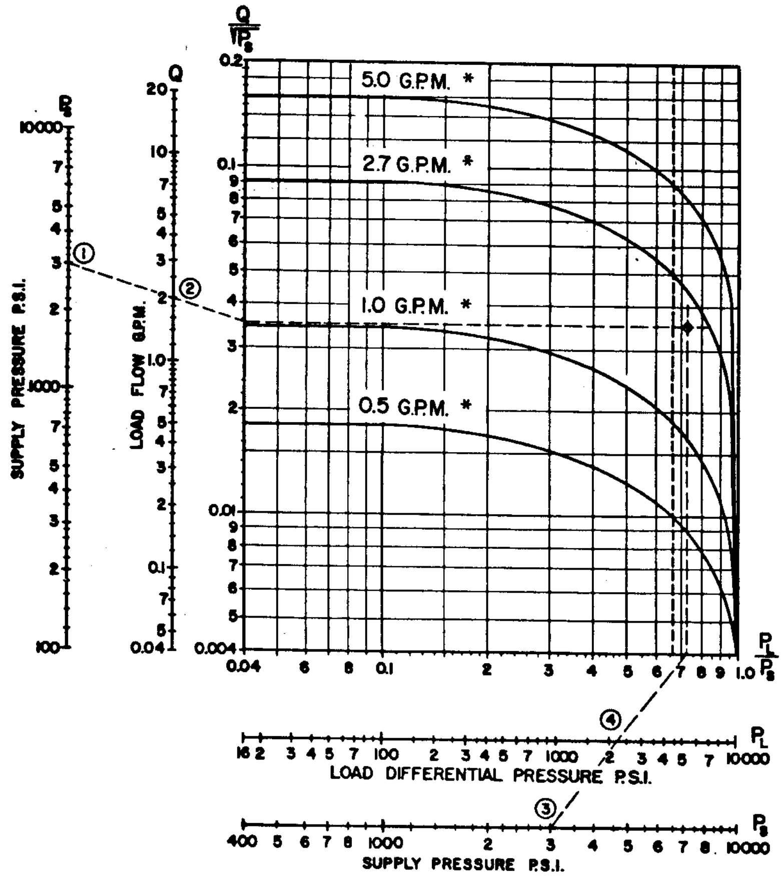
The nomograph shown on Figure 70-11 not only can be used to determine the optimum servovalve size for a particular system, but it has other uses. For example, with a valve size given, any combination of Q and P_L for a certain P_S can be determined directly. Also, by

72.2 A METHOD FOR DETERMINING THE OPTIMUM SERVOVALVE FOR EACH HYDRAULIC SERVOSYSTEM (CONT)

trial and error, with a valve size given, the P_S required to fulfill a known load requirement can be determined.

The nomograph relates the variables in the following equation and four sizes of servovalves.

$$\frac{Q}{\sqrt{P_s}} = K\sqrt{1 - \frac{P_L}{P_s}}$$



*Rated flow at 1000 psi drop across the valve.

✱ The four sizes of servovalves shown have standard port configurations

The following load conditions and supply pressure are plotted as an example on the servovalve nomograph:

$Q = 2$ GPM

$P_s = 3000$ psi

$P_L = 2200$ psi

This condition was plotted as follows: Referring to the vertical scales on the left of the nomograph, draw a line from the supply pressure (1) through the load flow (2). Draw a horizontal line as shown. Referring to the horizontal scales, draw a line from the supply pressure (3) through the load differential pressure (4). Draw a vertical line as shown. The optimum valve for this load condition and supply pressure is the 2.7 GPM size.

FIGURE 70-11. OPTIMUM SERVOVALVE SIZING NOMOGRAPH

73. VALVE DESIGN PROBLEMS

73.1 POPPET VALVES

a. Valve Seats Do not use cast surface for a valve seat. Do not use casting surface for metering; use a sleeve.

b. Steel Poppets, Spools Against Aluminum Seats Insufficient seat area and high valve seating forces can cause peening of aluminum seat. Valve moving parts may stick as a result. In general, avoid using aluminum seats; install steel seat insert.

c. Flow-Past Soft Seat Soft seals or poppets that depend on assembly compression to prevent secondary leak paths around the material can leak from distortion or compression under operating pressures. A fix is to incorporate static "O" rings to protect secondary leak paths.

d. Return Check Valve Install check valves in the return line from subsystems to prevent back pressures caused by high return flows from acting on cylinder locks, differential cylinder areas, and return cavities. The miniature check valve should be used because it has a faster response time than the standard AN-type. Balanced areas for lock devices are recommended.

e. Pressure Check Valves Install check valves in the pressure line of subsystems where air loads can cause a flow reversal if system pressures are reduced by combined demand. Consider relief-type check valve if overloading can occur at high speeds. Also, orificing may be used to reduce system demand if excess operation speed exists.

f. Materials Which Can Cold-Flow Such materials as Teflon and nylon, used in components for poppets or poppet seats, should be sufficiently supported to prevent excessive material creep. Component malfunction can occur from change in travel or leakage characteristics. Tests to determine cold-flow characteristics should be realistic, particularly for the time the material remains under load.

g. Soft-Seat Valves All soft-seat valve designs should have the seal material supported in a manner to prevent washout and cold-flow of the material.

h. Spring Retention Provide for retention of springs in poppet valves to prevent them from being washed out by high flows, such as rapid discharge from an accumulator.

73.2 SOLENOID VALVES

a. Electrically Operated Valves Consider the valve positioning and the effect on the system if both electrical inputs have electrical power applied concurrently as from a short, etc. Relays can ensure predictable operation in this abnormal situation. Some motor-operated shutoff valve installations can fail closed and/or open without system indication. A means of indication is needed to determine that the valve has traveled to the selected position.

b. Direct Operating Solenoid Valves Valves of this type must have adequate return spring force to overcome silting action. The compromise to be considered is the solenoid weight.

c. Split-Coil (Holding) Solenoids Such solenoids should be avoided in valve design. Unreliable operation, caused by starting coil switch settings and malfunctions, overheats the solenoids. This overrides any desirable features of split-coil design such as less weight.

d. Pilot-Operated Solenoid Valves These valves usually require a stated minimum system pressure, which may not be available, particularly under assisting load conditions. Design provisions may be incorporated within the pilot section to prevent a valve shift when system pressure falls momentarily.

e. Continuous-Duty Solenoid Valves Qualification tests should be conducted to determine if a continuous-duty solenoid will withstand a given temperature rise without causing valve failure. Test at the actual environmental temperature the valve will encounter. Valve failures from leaking seals and sticking pilot valves have been caused by overheating.

f. Valves Dependent on Low Operating Forces for Movement
Low-operating-force valves, such as pilot section of solenoid valves, should be designed with poppets. Spool valves are subject to sticking from contamination (silting, etc.). Poppet valves are self-cleaning with flow. Failures of four-way valves have been noted with spool-type pilot-operating section.

73.3 SERVOVALVES

a. Return Port Testing Unrealistic and undesirable backflow can occur during return port pressure testing. This can damage delicate valves such as servo transfer valves. Servovalve damage from back pressure to flapper support causes out-of-null condition.

b. Servovalve Screens Install inlet screens on servovalves to protect small openings and passages.

73.4 MISCELLANEOUS VALVES

a. Component Part Concentricity Part concentricity should not depend on thread concentricity.

b. Return Side of Valve For equipment with return or discharge ports, the return section shall be designed to a proof pressure at least equal to the nominal pressure of the high-pressure side.

c. Flow-In Valves Flow paths in valves can be affected by indexing of radial holes in spools. Springs can affect flow paths. Poppets with flutes can rotate with flow, which may be desirable in some designs.

d. Self-Locking Nut Avoid use of self-locking nut to hold spring-loaded adjustment. Use lockwire and lead seal after adjustment.

e. Internal Seal Leakage Consider the possibility of leakage past internal seals into component cavities. This can result in overloading of end caps or malfunction of the component from greater than normal pressures on moving parts. Static seal leakage from hydraulic motor to gear case can cause binding of gears. Fix is to vent gear case.

f. Valve Damper A valve that is not driven by a direct mechanical input (driven through springs is not direct) must have a damping chamber working on the valve spool.

g. Location of Air-Bleeding Passages Locate stagnant cavities of oil in a way that they are self-bleeding or provide an escape passage high enough to be able to bleed with normal unit functions. Orientation refers to actual aircraft installation.

h. Hydraulic Pressure Regulators Transient back pressures at the valve return port can affect regulated pressure. Fix is to install fast-acting miniature check valve in the return line to valve.

i. Emergency Dump Valves Hydraulic emergency overboard dump valves should have sufficient flow capacity to avoid back-pressure buildup. In some systems, this buildup can divert oil to the reservoir through selector valves and check valves. Directing oil to the reservoir can result in reservoir failures.

j. Balanced In-Line Relief Valves Differential area vent seal wear and leakage can occur from plunger motion during normal system pressure fluctuations. Provide long-life seals or a valve design incorporating little or no plunger motion during normal system pressure fluctuation. Fix by installing cap seals.

73.4 MISCELLANEOUS VALVES (CONT)

k. Valve Interflow Valves that can be in an open center condition when passing through neutral (press open to return or dump) can be detrimental to system operation. Valve stopping, sticking, or otherwise failing in neutral can result in system dumping, low system pressures, or high system return pressures. Effects can be (1) loss of oil or (2) pressure on nearby subsystems.

l. Inspection All hydraulic components shall be inspected after machining using penetrant or magnetic-particle inspection methods.

m. Flow Regulators Aluminum assemblies such as flow regulators, which may operate constantly, can fail from operational wear and metal erosion from fluid flow. Make parts from steel.

n. "O" Ring Entry Chamfer The recommended "O" ring entry chamfer in the hydraulic manual is to be used on all designs in order to preclude installation problems.

80. LANDING GEAR DESIGN

NOTE: Landing gear design data, which were included as Section 80 of the previous Hydraulics and Landing Gear Manual, are now available in a separate volume entitled Landing Gear Manual.

90. TESTING

90.1 HYDROMECHANICAL TEST GROUP FUNCTIONS

The functions of the Hydromechanical Test Group include the following:

a. Definition of test programs to meet specification requirements or to prove design objectives.

b. Design test installations, manage test parts procurement, and supervise performance of tests.

c. Prepare reports for submittal to certifying agency.

d. Review, or help formulate, qualification test requirements in Douglas specifications for vendor-designed equipment, and production test procedures on Douglas and vendor drawings.

e. For vendor-designed and tested equipment: approval of test plan, witnessing of tests if warranted, and approval of test reports. Also if required, resubmittal of test reports to certifying agency.

f. Perform research and development tests, which may be oriented toward solving an immediate design problem or to advance the state of the art (IRAD, Independent Research and Development or CRAD, Contract Research and Development).

g. Recommend procurement of certain test equipment or facilities needed to carry out hydromechanical group missions.

90.2 TYPES OF TESTS

Tests fall into the following catagories:

90.21 QUALIFICATION TESTS These tests are sometimes called pre-production tests and are required by a certifying agency - The FAA or a military service. Qualification tests or pre-production tests may be accomplished in accordance with FAA, military or company specifications, and have the primary objective of assuring that the product will function as intended and have a satisfactory service life. FAA requirements are defined in Federal Air Regulations, Section 25. That portion pertaining to hydraulic testing is quite short, but says a lot in a few words. Military specifications, on the other hand, cover almost every kind of component in great detail. Section 120 of this manual lists most of these specifications.

90.22 PRODUCTION TESTS Production tests are also called acceptance tests or quality conformance tests and are required to ensure that production equipment, when delivered, possesses the required characteristics of quality, strength, weight, and performance under normal and proof test conditions.

90.23 DEVELOPMENT TESTS The objective of development tests is to obtain information for use in design. This category includes occasional research and development projects which may not have immediate applications but are intended to expand knowledge and capabilities for future designs.

90.24 TROUBLESHOOTING TESTS Troubleshooting tests are intended to determine the cause and/or cure for a system or component malfunction.

90.3 APPLICATIONS REQUIRING TESTS

90.31 <u>COMPANY-DESIGNED HYDRAULIC COMPONENTS</u> Qualification and production tests are usually required on company-designed hydraulic components such as cylinders, reservoirs, control valves, and mechanisms. These units are usually tested in the company laboratories. Where applicable, adjacent connecting fittings and mechanisms are also tested with the hydraulic component, although these are often designed by controls or structures engineers.

90.32 <u>COMPLETE SYSTEMS</u> Complete systems or subsystems of a new model are tested in order to determine if the design or concept meets specific requirements established by contract or specification. These tests are performed on the first production airplane, the controls test stand (Iron Bird), or in the laboratory, whichever is most appropriate. Each subsequent aircraft hydraulic system is subjected to production type testing to verify conformance to functional requirements.

90.33 <u>NEW COMPONENTS OR MATERIALS</u> Tests are usually required on new components and materials, as in the search for better seals (also seal configurations), or investigation of new hydraulic fluids. Long-wearing, nongalling combinations of metals or plating for hydraulic cylinder and valve components has been another area of investigation, along with fluid piping and tube fitting evaluation.

90.34 <u>VENDOR-DESIGNED ITEMS</u> Vendor-designed items are usually qualification tested and acceptance tested by the vendor, or subcontracted by him to a commercial laboratory. Tests by Douglas are usually limited to function verification of a new design, troubleshooting, etc.

90.4 TEST PROCEDURES

Refer to Control Procedure DAC 8.038 in conjunction with the following:

90.41 QUALIFICATION TEST PROCEDURES Qualification test procedures for company-designed components and complete hydraulic systems are usually specified by a Task Assignment Drawing (TAD). Qualification test procedures for vendor-designed items are prepared by the vendor in accordance with the test requirements of the Specification Control Drawing (SCD). The vendor bids the test program as part of his price quotation. If the vendor's product or a similar product has previous flight experience or has been previously qualified for a different contractor, Douglas may accept full or partial qualification based on this similarity together with previous qualification. This requires a detail comparison of the new design with the previously qualified one, including tolerance checking, materials, stresses and clearances.

90.42 ACCEPTANCE TEST PROCEDURES Acceptance test or production test procedures for company-designed components are specified on the production drawings of the component. Acceptance test procedures for vendor-designed items are handled in the same manner as qualification test procedures on vendor-designed items (Reference Section 90.41). A booklet-type drawing is prepared by the hydromechanical group and utilized to specify (on-aircraft) function test procedures for complete systems. As an example of content and format of this type document, refer to TXG 7002, (on-aircraft) Functional Test Procedure Hydraulic Systems, which is used on the DC-10.

90.43 DEVELOPMENT TEST PROCEDURES or troubleshooting test procedures are usually specified on a Task Assignment Drawing (TAD) or test/work request document (T/WR). The T/WR is utilized when the development test or troubleshooting test needs to be done on a flight test airplane.

90.44 TEST REQUEST DOCUMENTATION Complete test work request (Form 250-2069) and submit to hydromechanical supervisor for approval. TAD Z bar drawing numbers can be obtained from the numbers clerk in the Engineering Department drawing vault.

90.5 PRODUCTION TESTS

a. Installations The tests are specified in the functional test procedure drawing. For an example of this type of drawing, Refer to TXG 7002.

b. Subinstallations Occasionally it may be necessary to test installed equipment before final testing of the complete installation. Consult design and test engineers for the need of such tests. Specify the tests on the functional test procedure or installation drawing.

c. Bench Assemblies Tests of hydraulic equipment such as panels, units-plus-mechanism, etc., are specified on the applicable assembly or installation drawing.

d. Unit Assemblies Tests of hydraulic units are specified on the assembly or specification control drawing. (Where Douglas receiving inspection tests are required, they are called out on the drawing.)

e. Tube Assemblies DPS 3.80 and 3.801 calls out the proof pressure test method, and the tabulated piping assembly drawing list the pressure value.

f. Hose Assemblies Proof pressure and bulge tests are called out in DPS 3.572-2. Where necessary these tests are also specified in the Douglas Standard or Specification Control Drawing.

g. Castings For castings which are for pressure vessels, rough and machined casting drawings note this fact. Where necessary a proof pressure test is specified on the machined casting drawing. Consult design and test group engineers.

90.51 PRODUCTION TEST REQUIREMENTS Production tests shall be designed to verify:

a. Proof strength of all fluid chambers

b. Packing leakage (static and/or dynamic) and lapped-fit leakage is within design limits. All pressure responsive seals are to be tested for leakage at low pressure (e.g., 25 psi) as well as at proof pressure.

c. That all fluid passages are open, (important for intersecting drilled holes, cored castings, etc.)

d. That packing and mechanism friction is within design limits, and binding induced by pressure deformation of parts is not excessive.

e. That the unit functions as intended. This may include checks of flow rates at given pressures, rate of motion, handle loads at given pressures, direction, and rate of flow for specific applied voltages, etc.

90.52 TEST BLOCK On Douglas assembly drawings, the tests are normally tabulated in a test block. For simple actuators, a decal (DAC-24) is available. For more complex assemblies, the decal may be modified or augmented, or an original procedure devised. Usually, a precedent can be found on a previous drawing of a similar unit. Be sure all modes of operation, including emergency operating conditions, standby relief valves, integral check valves, etc., are tested. Particularly complex assemblies may contain functions which cannot be verified on the final assembly, so must be tested on a subassembly. In this case, separate test procedures on different areas of the drawing may be used. The test block shall be submitted to the test group engineer for approval.

90.521 HEADING A typical heading for Skydrol-unit test blocks is as follows:

PRODUCTION INSPECTION TEST PROCEDURE

MANUFACTURER TO TEST EACH UNIT TO THIS PROCEDURE USING DMS 2014 FLUID (CLASS IV PREFERRED) AT 90° F TO 110°F. ALLOWABLE LEAKAGE RATES ARE FOR NEW-CONDITION CLASS IV FLUID, APPROXIMATELY 11.2 CENTIPOISES (11.5 CENTISTOKES) AT ZERO PRESSURE AND NOTED TEMPERATURE, AND MUST BE CORRECTED: MULTIPLY RATE BY THE RATIO OF THE ABOVE VISCOSITY TO THE ACTUAL VISCOSITY OF THE TEST FLUID. TEST WITH FLUID CHAMBERS FREE OF AIR.

The note for correcting allowable leakage refers only to laminar flow (including lapped-leakage.) The note must be modified if some test steps involve faster flow (such as orifice drops).

After use, the viscosity of all types of hydraulic fluid decreases because of shearing, requiring periodic measurement of test stand fluid viscosity.

90.522 AUXILIARY INFORMATION All ports, levers, electrical pins, etc., must be identified on the assembly drawing so the test procedure can be followed without misinterpretation.

A schematic diagram of the assembly and/or of the hydraulic or electrical test circuit should be included if necessary to clarify the procedure.

A test aid, such as a dummy plug to replace a cartridge-type valve, may be detailed on the drawing. (The tooling department will then make this special tool available to the appropriate testing department.)

90.53 TEST FLUID (REF) Phosphate ester fluids per DMS 2014 are specified on commercial designs. Some inspection labs may use a DMS 2014 fluid other than the one used to determine flow rate limits. Typical properties for new fluids at 100°F (Ref. Section 10) and relative flow rates are:

	Skydrol 7000	Skydrol 500 B	Skydrol LD	Skydrol 500 B-4
Centistokes Kinematic Viscosity	15.5	12.0	11.2	11.5
Specific Gravity	1.072	1.048	0.988	1.057
Centipoises Absolute Viscosity	16.6	12.6	11.1	12.2
Relative Viscous Flow Rate	0.67	0.88	1.0	0.92
Relative Orifice Flow Rate	0.96	0.971	1.0	0.967

HYDRAULIC TESTS

90.54 PRODUCTION TEST PROOF PRESSURES FOR TRANSPORT AIRCRAFT When installed in the airplane, each portion of the hydraulic system should be tested to its critical relief pressure or to 125 percent of its system operating pressure, remembering each preceding test was to a higher pressure in accordance with Section 23.4 of this manual.

Judgment should be exercised in using the higher critical value when several identical units are subjected to different critical relief pressures.

When no critical relief pressure governs, the test pressures should be obtained from the Test Group Engineer.

When a part (as manually operated brakes) is not connected into a pressure system, the pressure of the system operating the part is taken as the maximum pressure possible under any condition except temperature expansion, while the critical relief pressure is taken as the maximum pressure possible under any condition. The system's operating pressure must also be calculated on a basis using 2/3 of the ultimate design load when that figure is available, and should be used unless either of the maximum pressures is higher.

When the critical relief valve is included in the test set-up, it must be tested to its relief pressure.

90.55 TABLE OF TEST PRESSURES To be applied to each production assembly and installation. Use the higher of the alternative pressures (or negative pressure where noted).

.551 HYDRAULIC INSTALLATION

PORTION OF SYSTEM	TEST PRESSURE
HYDRAULIC INSTALLATION EXCEPT RETURN OR SUCTION SYSTEMS	100% OF CRITICAL RELIEF PRESSURE OR 125% OF SYSTEM OPERATING PRESSURE
INSTALLATION SUBJECT TO ACCELERATED LOADS	100% OF MAXIMUM PRESSURE THAT ACCELERATED LOAD WILL DEVELOP
HYDRAULIC INSTALLATION – RETURN SYSTEM	40% OF SYSTEM OPERATING PRESSURE OR 150% OF ACTUAL PRESSURE
HYDRAULIC INSTALLATION – SUCTION SYSTEM EXCLUDING RESERVOIR	50 PSI OR 200% OF RESERVOIR OPERATING PRESSURE
HYDRAULIC SUCTION SYSTEM INCLUDING RESERVOIR	100% OF RESERVOIR RELIEF PRESSURE, 125% OF RESERVOIR OPERATING PRESSURE OR 50 PSI

.552 HYDRAULIC SUBINSTALLATION AND BENCH ASSEMBLIES

PORTION OF SYSTEM	TEST PRESSURE
HYDRAULIC SUB–INSTALLATIONS AND BENCH ASSEMBLIES EXCEPT RETURN OR SUCTION SYSTEMS	110% OF CRITICAL RELIEF PRESSURE OR 135% OF SYSTEM OPERATING PRESSURE
HYDRAULIC SUB–INSTALLATIONS AND BENCH ASSEMBLIES – RETURN SYSTEMS	45% OF SYSTEM OPERATING PRESSURE OR 165% OF ACTUAL PRESSURE
HYDRAULIC SUB–INSTALLATIONS AND BENCH ASSEMBLIES – SUCTION SYSTEMS EXCLUDING RESERVOIR	60 PSI OR 210% OF RESERVOIR OPERATING PRESSURE

HYDRAULIC TESTS

90.553 HYDRAULIC UNIT ASSEMBLIES

PORTION OF SYSTEM	TEST PRESSURE (PROOF)
HYDRAULIC UNIT ASSEMBLIES EXCEPT THOSE CONTAINING GAS	112.5% OF THERMAL RELIEF PRESSURE OR 125% SYSTEM RELIEF PRESSURE OR 150% OF SYSTEM OPERATING PRESSURE
*HYDRAULIC UNIT ASSEMBLIES CONTAINING GAS	200% OF SYSTEM OPERATING PRESSURE
HYDRAULIC UNIT ASSEMBLIES – RETURN CHAMBERS	50% OF SYSTEM OPERATING PRESSURE OR 187% OF ACTUAL PRESSURE
HYDRAULIC UNITS IN SUCTION LINE EXCLUDING RESERVOIR	70 PSI OR 225% OF RESERVOIR OPERATING PRESSURE
RESERVOIR	5 PSI ON UNPRESSURIZED RESERVOIRS
	50 PSI OR 200% OF OPERATING PRESSURE IN PRESSURIZED RESERVOIRS, WHICHEVER IS HIGHER

*FOR PNEUMATIC SYSTEMS CONSULT YOUR SUPERVISOR.

90.554 LINES, FITTINGS, AND HOSE

For hose assemblies, listed value does not apply in many cases. Use value in DPS 3.572-2 or hose specification. Tube fittings are not production tested at proof pressure except in special cases. (Strength is assured by qualification tests, etc.)

PORTION OF SYSTEM	TEST PRESSURE (PROOF)
LINES, FITTING, AND HOSE EXCEPT IN RETURN AND SUCTION SYSTEMS	250% OF SYSTEM OPERATING PRESSURE OR 167% OF CRITICAL RELIEF PRESSURE
LINES, FITTINGS, AND HOSE – RETURN SYSTEM	*80% OF SYSTEM OPERATING PRESSURE OR 300% OF ACTUAL PRESSURE
LINES, FITTINGS, AND HOSE – SUCTION SYSTEM	*75 PSI INTERNAL PRESSURE OR 300% OF RESERVOIR OPERATING PRESSURE **

* CHECK YIELD STRENGTH OF LINES TO INSURE NO SET AT THIS PRESSURE, AND CHANGE MATERIAL OR SIZE IF NECESSARY.

** CHECK SUCTION HOSE FOR LINER COLLAPSE WITH NEGATIVE PRESSURE

90.6 QUALIFICATION TESTS

90.61 TEST SPECIMENS Where hydraulic system components and related fittings and mechanisms are concerned, production units are used whenever possible. This is because of the importance of having a test specimen which fully represents the aircraft part. On occasion, certain types of test specimens are manufactured in the laboratory shops to gain a time advantage. However, if the production unit involves forged or cast parts, these may not be adequately simulated by parts machined from bar stock. The endurance test may be repeated for additional assurance. Each case is individually considered. Obviously, if a life test can be performed before quantity production is started, deficiencies in design can be corrected without reworking or scrapping production parts or declaring them to have a limited life. This is of such importance that usually one of the first two or three units made of a new design is assigned to the qualification test.

90.611 PREPARATION OF TEST SPECIMENS The typical hydraulic cylinder is subject to manufacturing tolerances on wall thickness, fits and clearances, heat-treat strengths, surface finishes, and many others. Any or all of these tolerances may affect the endurance life of the assembly. It is obviously impractical to adjust all of the tolerances in an assembly to the adverse extreme; however, it is usually practical to adjust the diametral clearances between sliding parts, the packing squeeze, and sometimes the surface roughness. A diametral clearance which is adverse for O-ring packing life (i.e., maximum) is least likely to cause binding at extreme temperatures. To cover all situations, two test specimens, or one with some interchangeable parts may be required. However, the minimum clearance situation with regard to thermal binding is almost always covered by mathematical analysis, because of extreme difficulties in obtaining adverse clearances and, most importantly, the effects of eccentricities.

Adverse clearances are obtained by honing of bores, lapping of external round surfaces, plating, and if practical, selective assembly. Sometimes a special part is fabricated to a particular dimension, with virtually no tolerance.

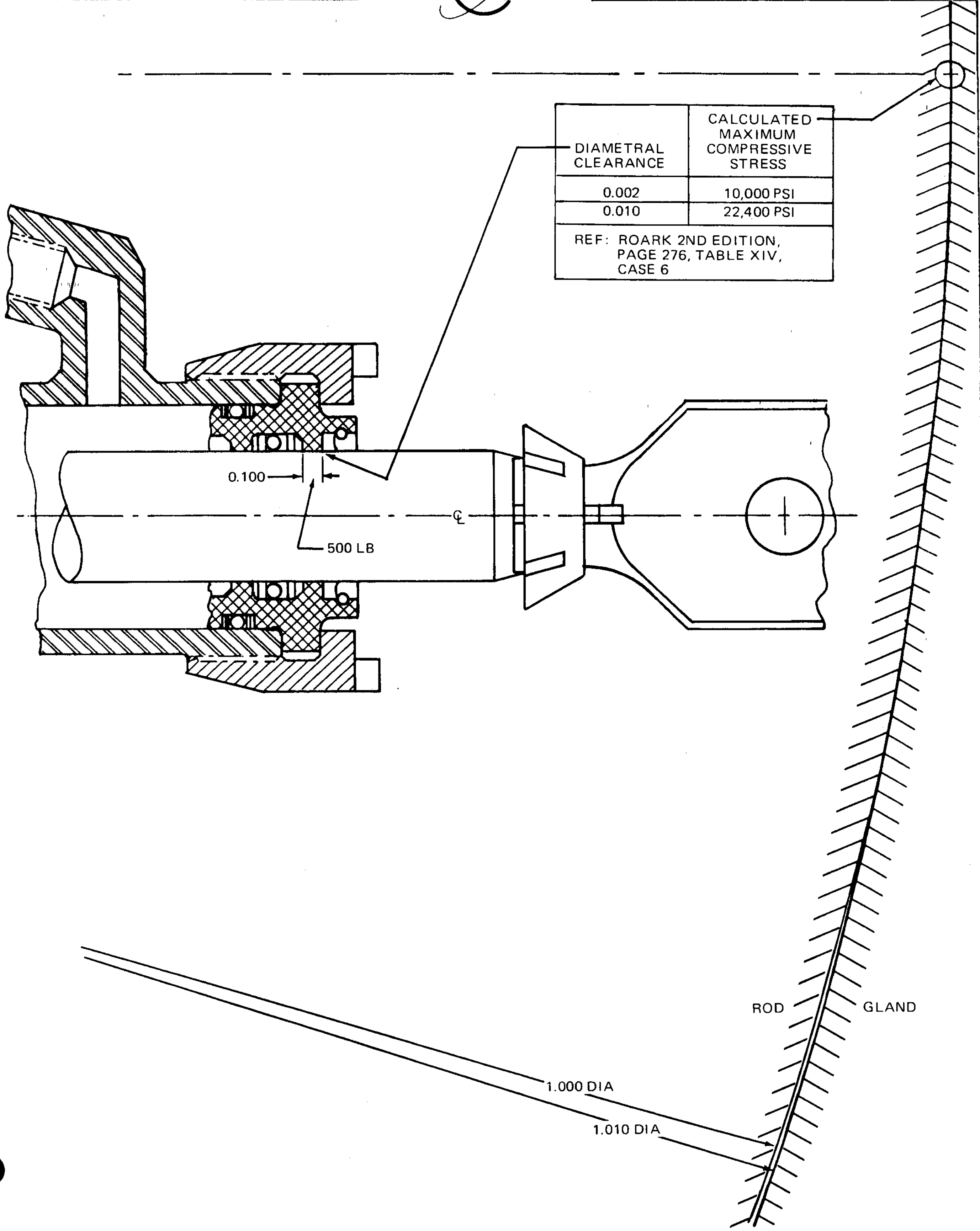
The importance of production tolerances cannot be over-emphasized. Figure 90-1, for example, shows a magnified cross section of the contact area of a 1-inch-diameter piston rod. Assuming the rod slides through a bearing 0.10-inch-long, and the bearing sideload is 500 pounds, the calculated maximum compressive stress is:

a. At 0.002 diametral clearance, 10,000 psi

b. At 0.010 diametral clearance, 22,400 psi

which is a ratio of 2.24:1.

The actual sideload in such an actuator is not constant as the clearance changes. It is composed of at least two components: one due to the torque resulting from rod-end friction, and one due to axial compression. The sideload due to axial compression varies with the angle between piston and barrel permitted by the diametral clearances, and is aggravated by secondary bending.



DIAMETRAL CLEARANCE	CALCULATED MAXIMUM COMPRESSIVE STRESS
0.002	10,000 PSI
0.010	22,400 PSI
REF: ROARK 2ND EDITION, PAGE 276, TABLE XIV, CASE 6	

FIGURE 90-1. EFFECT OF DIAMETRAL CLEARANCES ON BEARING LOADS

Some other areas sensitive to production tolerances are:

a. Packing Glands The extrusion gap (diametral clearance) can have a profound effect on packing life.

b. Packing Friction The O-ring squeeze and the surface roughness can account for wide variations. This could be very important for such items as spring-returned pistons, where the spring may be too weak for the maximum friction case.

c. Damping Chambers Some units, such as pressure reducers and relief valves, use damping chambers to prevent unstable operation (chatter). As little as 0.001 change in diametral clearance of a plunger to a bore may destroy the stability of a unit.

There are many other such areas. It is hydromechanical group practice to establish the adverse tolerance condition where practicable, if it is significant to the test.

To condition the rubber seals, the complete test assembly is sometimes submerged in hydraulic fluid and "hot soaked" at an appropriate temperature, such as 160°F, for a week.

Other preparation of the test specimen prior to testing may include addition of strain gage instrumentation, providing extra test pressure ports, etc. A complete precision inspection, in which all dimensions are recorded for later comparisons, is usually performed.

90.62 QUALIFICATION TEST PROGRAM Typical tests performed on hydraulic units include the following, although not all are applicable to all types of units:

a. Proof Pressure
b. External Leakage
c. Internal Leakage
d. Extreme Temperature Binding and Leakage
e. Operating Force
f. Sliding Friction
g. Pressure Drop
h. Impulse Life
i. Endurance or Operating Life
j. Burst Pressure
k. Crack, Flow and Reseat Tests (Relief Valves).

All but items h, i and j of these are fairly straightforward, self-explanatory, and nondestructive. As a group, they are referred to as minor first-article tests. The burst test is saved until last, as the unit may be deformed or destroyed in that test.

The Endurance or Life Test is the most significant of these tests, and the planning and operation of such tests accounts for the major part of the test group workload as far as qualification testing is concerned.

Planning the endurance life test can be quite involved. In the case of military aircraft, the number of load cycles may be quite well defined, but even then we may deviate from the specifications for a variety of reasons. In general, both for military and commercial applications, the goal is to prove that the test article will last the life of the airplane. This may be 4000 flight hours or less for military planes, and 40,000 flights or more for commercial planes.

For any major hydraulic cylinder installation and some of the minor ones, the structural mechanics section prepares a flight-load profile, such as the one shown in Figure 90-2.

The life test program is based on such a load profile, multiplied by the number of flights which constitute the "airplane life," and by a scatter factor which is intended to account for the variables which may affect the life of the single test specimen. These variables include manufacturing tolerances, surface finishes, fillet radii, the metal alloy, its heat treatment, and its cleanliness, and variations in grain direction, plating, etc. For the DC-9 we used a scatter factor of 2. For the DC-10, Structural Mechanics wants this increased to 3.

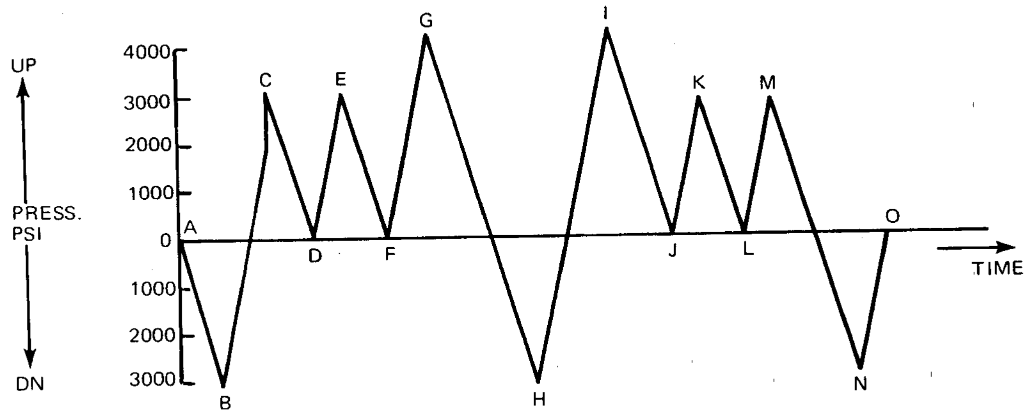
When the test program consists of several different load conditions for each flight, the easiest way to apply the loads is to apply all of the load cycles for the first load condition, followed by all the load cycles of the second condition, etc. Unfortunately, when a failure occurs, it is impossible to tell what the completed portion of the test is equivalent to in terms of flights or flight hours. Sometimes, therefore, the life test is programmed to apply the loads in the same sequence as the flight-load profile. In other cases, a compromise solution is used. The test is divided into equal phases, perhaps ten, so that 10 percent of the load "A" cycles are performed, followed by 10 percent of the load "B" cycles, etc., until 10 percent of the complete test program has been performed. Then the load "A" cycles are repeated for another 10 percent, and this sequence continued until 100 percent of all scheduled loads have been applied.

This method has proved satisfactory. Failures can be evaluated in terms of equivalent flight time, yet test installations are kept relatively simple.

In some cases, it has been preferable to apply complete flight sequences of loads, if this can be done without unduly complicating the test installation.

It has long been our policy to perform cycling tests at as high a frequency as possible to minimize test time and expense. Obviously, inadvertent loading due to inertia of moving parts, unforeseen flow effects, etc., must be guarded against, so these do not affect the test results.

It has also been Douglas policy to test hydraulic components utilizing the same mounting conditions as on the airplane including actual attaching hardware and fittings. Flexible piping is also qualification tested along with the hydraulic component to which it attaches.



A → B	AT POINT "A," AIRPLANE IS PARKED (ZERO HYDRAULIC PRESSURE). POINT "B" REPRESENTS DOWN-SIDE PRESSURE AFTER STARTING ENGINES.
B → C	GEAR IS RETRACTED AFTER TAKEOFF. CONSERVATIVELY ASSUME AIRPLANE SPEED DURING RETRACTION AS 200 KEAS AND THE AIRPLANE LOAD FACTOR = x 1.5G.
C → D	MOVE LANDING GEAR HANDLE TO UPLATCH CHECK POSITION TO DEPRESSURIZE SYSTEM TO CHECK THAT GEAR IS LOCKED IN "UP" POSITION.
D → E	MOVE LANDING GEAR HANDLE TO UP POSITION AFTER UPLATCH CHECK OPERATION.
E → F	IN FLIGHT, THE LANDING GEAR SYSTEM CAN DEPRESSURIZE IF THE AIRPLANE 1500-PSI HYDRAULIC PRESSURE SYSTEM IS SELECTED.
F → G → H	INCREASE AIRPLANE HYDRAULIC PRESSURE TO 3000 PSI AND EXTEND GEAR FOR AIRPLANE SLOW-DOWN. CONSIDER A MAXIMUM EXTENSION SPEED OF 300 DIAS (M = 0.7) AND AN AIRPLANE LOAD FACTOR = +2.0G DURING EXTENSION.
H → I	RETRACT GEAR AFTER SLOW-DOWN. CONSIDER A MAXIMUM RETRACTION SPEED OF 250 KEAS (M = 0.7) AND AN AIRPLANE LOAD FACTOR = +2.0G.
I → J → K → L	PERFORM UPLATCH CHECK PROCEDURE AND REDUCE THE AIRPLANE HYDRAULIC PRESSURE TO 1500 PSI.
L → M	INCREASE THE AIRPLANE HYDRAULIC PRESSURE TO 3000 PSI PRIOR TO LANDING.
M → N	EXTEND LANDING GEAR. CONSIDER AIRPLANE SPEED DURING EXTENSION AS 200 KEAS AND AN AIRPLANE LOAD FACTOR = +1.5G.
N → O	SHUT OFF ENGINES SUBSEQUENT TO LANDING.

FIGURE 90-2. TYPICAL FLIGHT OPERATION

90.63 LOADING METHODS In endurance cycling tests of hydraulic actuating cylinders, it is usually desirable to closely duplicate the load versus stroke cycle experienced on the airplane. If there ia a peak load, for example, which occurs at 30 percent of the stroke, and in the test, this load is applied at 70 percent of the stroke, the attaching brackets or mechanisms may be loaded in a direction which would induce a failure, whereas in the airplane, the loads are taken in a different path and no failure would occur. Conversely, a potential weak point in the system may pass the test because the critical load wasn't applied at the proper time.

Figure 90-3 shows some typical load-cycle fixtures, which have been used in our lab. Many types of load-cycles can be approximated by variations in the geometry of these fixtures. Sometimes, however, geometry alone won't suffice, and it is necessary to resort to sequence valving, variable orifices, and other devices to vary the pressure in the load cylinders. Servo-valves with programmed signal generators are a last resort, if no simple mechanical method of load control is adequate.

Cycling controls are also kept as simple as possible. Sometimes a simple selector valve driven by a geared-down motor is sufficient. More often, however, an array of limit switches, solenoid valves, drum-type timers, electronic timers, and digital counters are used. Figure 90-4 is a fairly typical schematic diagram of a test cycling installation. Notice the multiple disc rotary sequence timer at the lower left corner. Timers of this type with up to 20 switching functions per cycle were used in the DC-9 test program.

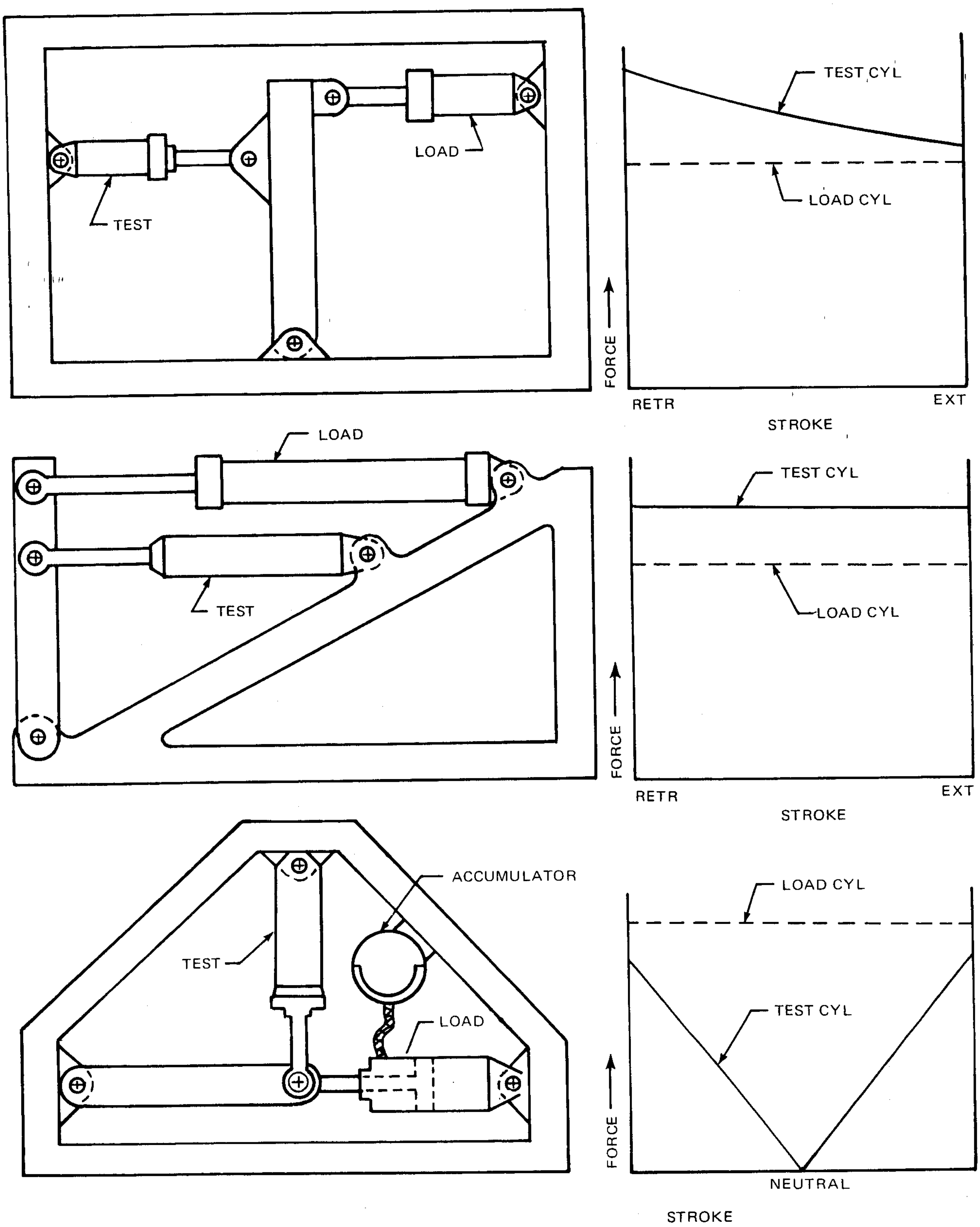


FIGURE 90-3. TYPICAL LOAD-STROKE JIGS

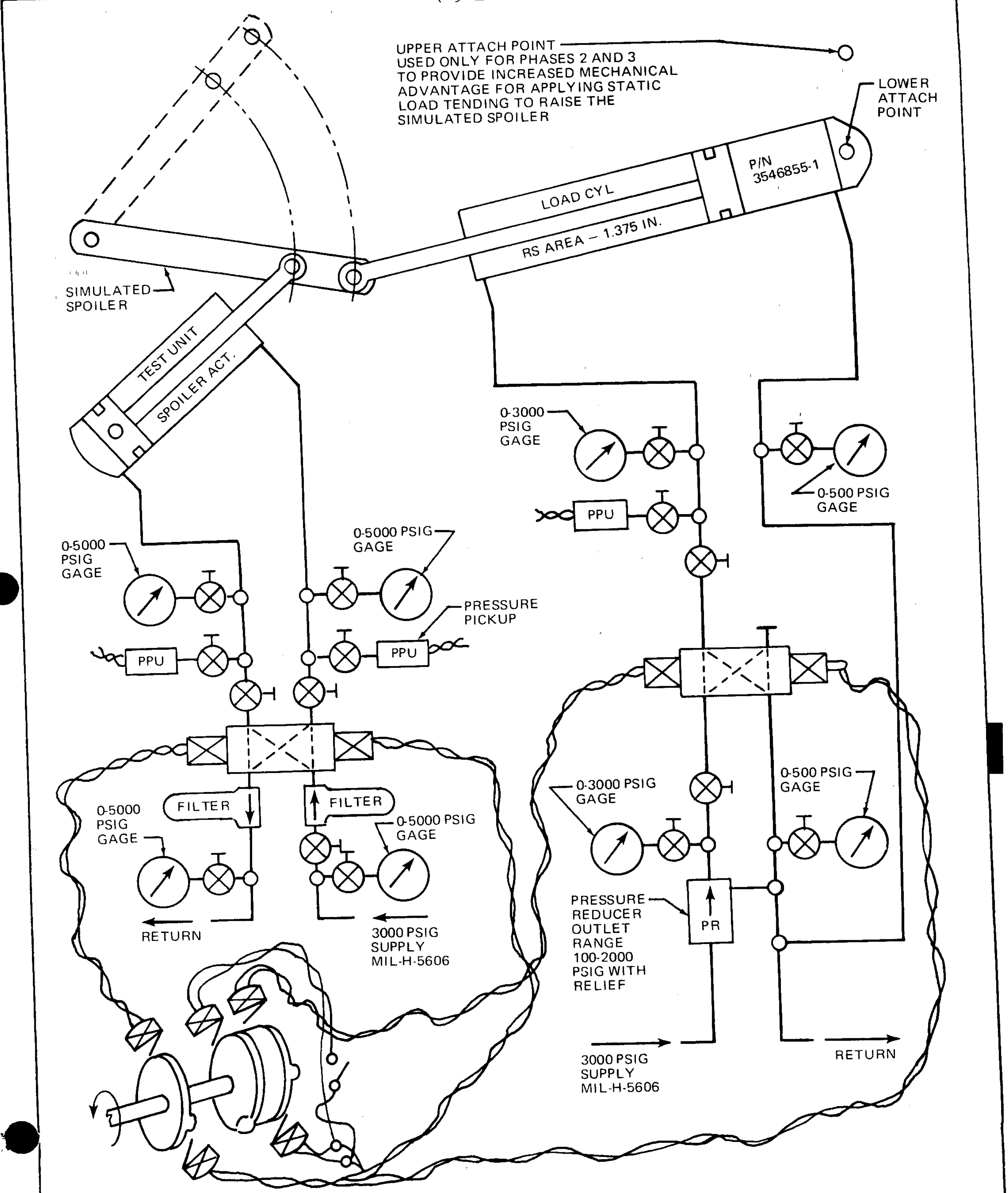


FIGURE 90-4. ENDURANCE TEST SETUP

90.7 EVALUATION OF FAILURES

When a test component fails, it is necessary to find out why. The answer usually falls into one of four categories:

a. <u>The Test Was Too Severe</u> A re-evaluation of test loads, sometimes augmented by flight test data obtained after the test conditions were originally established, sometimes results in reducing the test.

b. <u>Fault Test Installation</u> Test fixtures which are too rigid or too flexible may induce loads in the airplane parts which are not encountered in-service.

c. <u>Faulty Manufacture of the Specimen</u> Process engineering specialists are employed to find such problems as faulty alloys, improper heat treatment, voids or inclusions, hydrogen embrittlement from plating operations, etc.

d. When all other possibilities have been eliminated, it is necessary to face facts and admit that the design is deficient. This brings up all kinds of problems: How will it be changed? What will be the effectivity? Can existing parts be reworked? Should parts be delivered on a limited-life basis and a later field change made? How does the life relate to the warranted life? These are not basically test-group problems, but we often get involved quite heavily anyway. Sometimes our test fixtures are used to prove out the soundness of a proposed retrofit scheme.

90.8 TEST REPORTS

While the object of a test program is to establish that an article is satisfactory in its function, strength, endurance life, etc., the permanent evidence of the test is a test report. Unless a complete report is prepared and indexed, and the original stored in the vault along with other permanent documents, the report and the test data might be thrown out during the next cleanup campaign. The loss of such data, or the necessity to repeat the test program, has sometimes been quite costly.

Under military contracts, the reports must be submitted to the customer for approval. Also, the customer's representative must witness, or at least be invited to witness, the tests in progress.

On commercial airplanes, the FAA represents the customer. FAA engineers may choose to witness certain tests, and may request submittal of specific reports. They don't have the manpower to monitor the tests and read the reports in the detail that the military services do. However, all the information must be available, because any component may come under suspicion and the test report be requested by FAA.

After an airplane is originally certified by either the FAA or military service, however, the reports have many additional functions. When new versions of the airplane are built, the reports may be reviewed to see if the existing hydraulic units are suitable for a changed flight-load profile. If the military services buy a version of a commercial airplane, they review the test reports to help decide whether the system has been adequately tested for military use, even if not strictly in accordance with military specifications. Hydraulic test reports are often useful where a similar item is being designed for a new airplane, to avoid repeating old mistakes. Sometimes these reports have been useful in crash investigations and patent litigation.

A test report, to be of maximum value, must be complete and accurate. In particular, it must tell: (1) What the test specimen was, with the part number and change letter of every detail of the assembly, and include the details of any modification or reworks. All critical dimension must be recorded before and after the test, and perhaps periodically during the test. (2) The test load spectrum applied, and the source of these loads, that is, exactly what they represent in terms of airplane service. (3) The geometry of any test mechanism, and a schematic diagram of the hydraulic circuitry. (4) Evidence of the load cycle(s) applied (oscillograph traces, oscilloscope photos, etc.). (5) Details of any failures, redesigns or modifications of the test specimen, or changes in the installation or procedure during the course of the test. (6) Photos in sufficient detail to support the descriptions and diagrams, showing the installation, the specimen, details or any failures, etc. (7) A cross-sectional drawing of the specimen, showing all materials, heat-treat strengths, platings, etc. (8) A thorough discussion of any possible considerations that could be pertinent or of interest.

100. STANDARDS

100. STANDARDS

100.1 GENERAL This section includes standard designs that are associated with almost every major component of aircraft hydraulic systems. Seals, glands, bosses, threads, nameplates, and lee plugs are included to the extent that they are utilized at Douglas.

101. SEALS

101.1 DESIGN CONSIDERATIONS Some of the important features or characteristics which must be considered in hydraulic seal design are listed below:

Leakage
Friction
Operating Temperature Range
Durability
Fluid Compatibility
Bidirectional Sealing Capability
Instability
Cost
Surface Roughness
Shelf Life
Multiple-Usage Capability (Simplification of Stocking)
Compactness of Design
Rigidity or Volumetric Backlash as related to System Response
Compatibility with Metals
Gland Tolerance Requirements
Contaminant Generating Tendencies
Gland Design Simplicity

101.2 O-RINGS The selection of O-rings is dependent upon the type of fluid in the system in which the O-ring is used and upon the location of the O-ring. See the tabulation shown below:

		O-Ring Part No.	
Fluid	O-Ring Material	MIL-P-5514 Gland Location	Boss-Seal Location
Phosphate-Ester	Ethylene-Propylene Rubber (EPR)	NAS1611(1)(3)	NAS1612(2)(3)
MIL-H-5606	Buna-N	MS28775(1)	MS28778

Notes:

1. The NAS1611 dash number is the same as the MS28775 dash number for equivalent sizes.

2. NAS1612 dash numbers correspond to the tubing OD in sixteenths of an inch.

3. Controlling note for NAS1611 and NAS1612 O-rings:

   a. On assembly drawings with separate material lists, code each NAS1611 and NAS1612 callout with the following note:

      "Procurement Sources and Quality Control per 7912037."

   b. On other assembly drawings, include the following note, as applicable:

      "Procurement Sources and Quality Control of NAS1611 and NAS1612 O-Rings per 7912037."

   c. Specification drawings and bid documents are to carry a note stating, "Skydrol-Resistant O-Ring Seals are to be called for on drawings by NAS1611 and NAS1612 numbers. Procurement sources and quality control of NAS1611 and NAS1612 O-rings are to be per Douglas Specification 7912037."

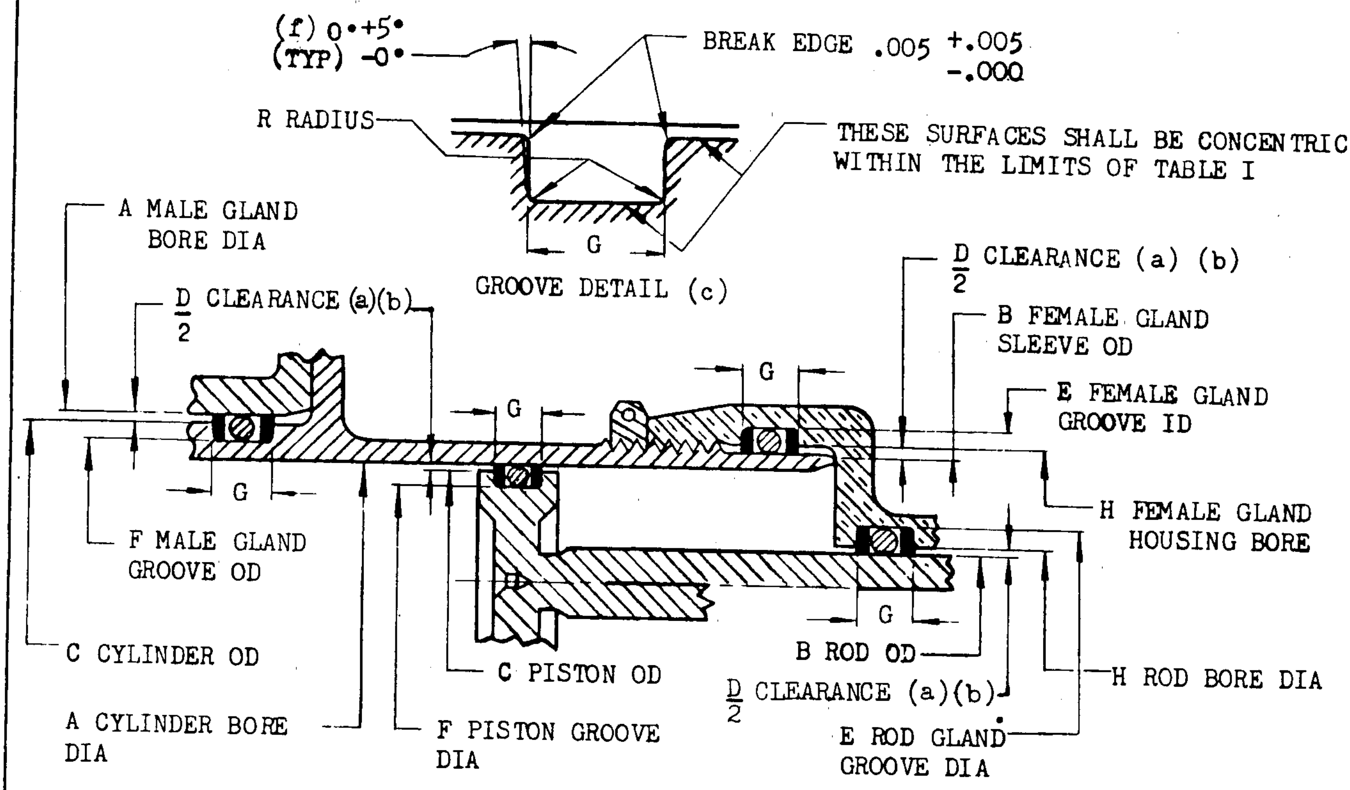
101.3 BACK-UP RINGS MS27595 and MS28774 back-up rings are available for use in MIL-G-5514 glands. MS27595 is an uncut back-up ring and is available in all sizes from -004 through -460. MS28774 is a scarf-cut back-up ring and is available in sizes -004 through -139, -210 through -230, and -325 through -437. (The dash numbers of the MS27595 and MS28774 back-up rings correspond to the dash numbers of the NAS1611 and MS28775 O-rings.)

The MS28774 back-up rings are the simplest to install; however, they are subject to being pinched and damaged during the assembly process, especially when the mating components containing the seal are screwed together on assembly (as opposed to being pushed together on assembly).

The MS27595 back-up rings usually require special tooling to ensure proper assembly into its gland in piston-type sealing applications, but it is capable of overcoming the installation problem. Therefore, it should be a design objective to use the MS27595 uncut back-ups when the mating components containing the seal are screwed together on assembly.

It also shall be a design objective, when back-up rings are required, to install a back-up ring on both sides of the O-ring.)

101.4 O-RING GLAND DESIGN The dimensions of MIL-G-5514F should be followed and are shown below and in Table I.



(SEE TABLE I FOR DIMENSIONS.)

(a) DIAMETRICAL CLEARANCE IS THE TOTAL DIFFERENCE BETWEEN THE BORE ID AND THE MEMBER CONTAINED THEREIN.

(b) TOTAL INDICATOR READING, BETWEEN GROOVE AND ADJACENT BEARING SURFACE. SEE GROOVE DETAIL.

(c) CAUTION SHOULD BE OBSERVED TO INSURE THAT THE RADIUS USED AT THE BOTTOM OF THE GLAND DOES NOT RESULT IN NOTCH SENSITIVITY OF THE GLAND DESIGN OR CREATE AN INSTALLATION PROBLEM.

(d) FOR THE GROOVE ANGLE, BETTER PERFORMANCE IS OBTAINED AT THE 0 DEGREE ANGLE.

(e) EITHER THE GROOVE DIAMETER DIMENSION OR THE OPPOSING SEALING SURFACE DIMENSION MAY BE HELD WITHIN CLOSER LIMITS THAN THOSE SPECIFIED TO GAIN ADDITIONAL MACHINING TOLERANCE ON ITS OPPOSING DIMENSION, PROVIDED THE ACCUMULATED TOLERANCE OF THE TWO DIMENSIONS DOES NOT EXCEED THAT SPECIFIED.

EXAMPLE: FOR AN MS28775-221 O-RING
"A" DIAMETER MAY BE HELD TO 1.678/1.679
IN LIEU OF 1.678/1.680 TO GAIN AN "F" DIAMETER
DIMENSION OF 1.435/1.432 IN LIEU OF 1.435/1.433.

NOTE: CAUTION SHOULD BE USED IN APPLYING THE −001 THROUGH −005 SIZES. WHILE BEING INSTALLED IN AN EXTERNAL GROOVE, THEY MIGHT BE STRETCHED BEYOND THE ELASTIC LIMIT, WITH PROBABLE FAILURES OR INCIPIENT FAILURES RESULTING. MOREOVER, THERE IS NO STANDARD BACKUP RING FOR −001 THROUGH −003; THEREFORE, FOR PRESSURES IN EXCESS OF 1500 PSI THE DIAMETRAL CLEARANCE (EXTRUSION GAP) MUST BE REDUCED.

IT IS RECOMMENDED THAT WHEREVER POSSIBLE, O-RINGS WITH A LARGER CROSS SECTIONAL DIAMETER ("W" DIMENSION) BE USED IN PREFERENCE TO −020 THROUGH −028 AND −131 THROUGH −149, SO AS TO PROVIDE A MORE ADEQUATE SEAL. THEREFORE, THESE SIZES ARE NOT PREFERRED.

101.4 (Continued)

O-RING GLAND DIMENSIONS

AN6227 DASH NO.	AN6230 DASH NO	MS28775 DASH NO.	SEAL INSTALLATION DIMENSIONS – EXTERNAL – "C" PISTON OR CYLINDER OD	SEAL INSTALLATION DIMENSIONS – EXTERNAL – "A" CYLINDER BORE OR MALE GLAND CYLINDER BORE ID	SEAL INSTALLATION DIMENSIONS – EXTERNAL – "F" GROOVE OD	SEAL INSTALLATION DIMENSIONS – INTERNAL – "B" ROD OR GLAND SLEEVE OD	SEAL INSTALLATION DIMENSIONS – INTERNAL – "H" ROD BORE OR FEMALE GLAND HOUSING BORE ID	SEAL INSTALLATION DIMENSIONS – INTERNAL – "E" GROOVE ID	DIAMETRAL CLEARANCE (a) "D" MAX – EXT	DIAMETRAL CLEARANCE (a) "D" MAX – INT	SQUEEZE – MINIMUM – ACTUAL	SQUEEZE – MINIMUM – PERCENT (REF)	SQUEEZE – MAXIMUM – ACTUAL	SQUEEZE – MAXIMUM – PERCENT (REF)	GROOVE WIDTH G – NO BACKUP RING	GROOVE WIDTH G – ONE BACKUP RING	GROOVE WIDTH G – TWO BACKUP RINGS	GROOVE CORNER RADIUS	ECCENTRICITY (c)	O-RING CROSS SECTION	O-RING INSIDE DIAMETER
		001	0.093 0.092	0.095 0.096	0.033 0.032	0.033 0.032	0.035 0.036	0.095 0.096	0.004	0.004	0.005	13.5	0.012	27.9	0.063 0.073			0.015 0.005	0.002	0.040 ± 0.003	0.029 ± 0.004
		002	0.126 0.125	0.128 0.129	0.048 0.047	0.048 0.047	0.050 0.051	0.128 0.129			0.006	12.8	0.013	24.5	0.073 0.083					0.050 ± 0.003	0.042
		003	0.157 0.156	0.159 0.160	0.063 0.062	0.063 0.062	0.065 0.066	0.159 0.160			0.008	14.0	0.015	23.8	0.083 0.093					0.060 ± 0.003	0.056
		004	0.188 0.187	0.190 0.191	0.076 0.075	0.076 0.075	0.078 0.079	0.190 0.191			0.009	13.4	0.016	21.9	0.094 0.104	0.149 0.159	0.207 0.217			0.070 ± 0.003	0.070
		005	0.219 0.218	0.221 0.222	0.108 0.107	0.108 0.107	0.110 0.111	0.221 0.222			0.0095	14.2	0.0165	22.6							0.101 ± 0.004
1		006	0.233 0.232	0.235 0.236	0.123 0.122	0.123 0.122	0.125 0.126	0.235 0.236			0.010	13.4	0.017	23.3							0.114 ± 0.005
2		007	0.264 0.263	0.266 0.267	0.154 0.153	0.154 0.153	0.156 0.157	0.266 0.267													0.145
3		008	0.295 0.294	0.297 0.298	0.185 0.184	0.185 0.184	0.187 0.188	0.297 0.298													0.176
4		009	0.327 0.326	0.329 0.330	0.217 0.216	0.217 0.216	0.219 0.220	0.329 0.330													0.208
5		010	0.358 0.357	0.360 0.361	0.248 0.247	0.248 0.247	0.250 0.251	0.360 0.361													0.239
6		011	0.420 0.419	0.422 0.423	0.310 0.309	0.310 0.309	0.312 0.313	0.422 0.423													0.301
7		012	0.483 0.482	0.485 0.486	0.373 0.372	0.373 0.372	0.375 0.376	0.485 0.486	0.004	0.004	0.010	13.4	0.017	23.3	0.094 0.104	0.149 0.159	0.207 0.217	0.015 0.005	0.002	0.070 ± 0.003	0.364 ± 0.005
		013	0.548 0.547	0.550 0.552	0.438 0.436	0.435 0.433	0.437 0.438	0.547 0.549	0.005	0.005	0.009	13.4	0.017	23.3	0.094 0.104	0.149 0.159	0.207 0.217	0.015 0.005	0.002	0.070 ± 0.003	0.426 ± 0.005
		014	0.611 0.610	0.613 0.615	0.501 0.499	0.498 0.496	0.500 0.501	0.610 0.612													0.489
		015	0.673 0.672	0.675 0.677	0.563 0.561	0.560 0.558	0.562 0.563	0.672 0.674													0.551
		016	0.736 0.735	0.738 0.740	0.626 0.624	0.623 0.621	0.625 0.626	0.735 0.737													0.614
		017	0.798 0.797	0.800 0.802	0.688 0.686	0.685 0.683	0.687 0.688	0.797 0.799													0.676
		018	0.861 0.860	0.863 0.865	0.751 0.749	0.748 0.746	0.750 0.751	0.860 0.862													0.739 ± 0.005
		019	0.923 0.922	0.925 0.927	0.813 0.811	0.810 0.808	0.812 0.813	0.922 0.924													0.801 ± 0.006
		020	0.989 0.988	0.991 0.993	0.879 0.877	0.873 0.871	0.875 0.876	0.985 0.987													0.864
		021	1.051 1.050	1.053 1.055	0.941 0.939	0.935 0.933	0.937 0.938	1.047 1.049													0.926
		022	1.114 1.113	1.116 1.118	1.004 1.002	0.998 0.996	1.000 1.001	1.110 1.112													0.989
		023	1.176 1.175	1.178 1.180	1.066 1.064	1.060 1.058	1.062 1.063	1.172 1.174													1.051
		024	1.239 1.238	1.241 1.243	1.129 1.127	1.123 1.121	1.125 1.126	1.235 1.237													1.114
		025	1.301 1.300	1.303 1.305	1.191 1.189	1.185 1.183	1.187 1.188	1.297 1.299													1.176
		026	1.364 1.363	1.366 1.368	1.254 1.252	1.248 1.246	1.250 1.251	1.360 1.362													1.239
		027	1.426 1.425	1.428 1.430	1.316 1.314	1.310 1.308	1.312 1.313	1.422 1.424													1.301
		028	1.489 1.488	1.491 1.493	1.379 1.377	1.373 1.371	1.375 1.376	1.485 1.487	0.005	0.005	0.009	13.4	0.017	23.3	0.094 0.104	0.149 0.159	0.207 0.217	0.015 0.005	0.002	0.070 ± 0.003	1.364 ± 0.006
8		110	0.548 0.547	0.550 0.552	0.372 0.370	0.373 0.371	0.375 0.376	0.551 0.553	0.005	0.005	0.009	9.0	0.017	16.0	0.141 0.151	0.183 0.193	0.245 0.255	0.015 0.005	0.002	0.103 ± 0.003	0.362 ± 0.005
9		111	0.611 0.610	0.613 0.615	0.435 0.433	0.435 0.433	0.437 0.438	0.613 0.615													0.424
10		112	0.673 0.672	0.675 0.677	0.497 0.495	0.498 0.496	0.500 0.501	0.676 0.678													0.487
11		113	0.736 0.735	0.738 0.740	0.560 0.558	0.560 0.558	0.562 0.563	0.738 0.740													0.549
12		114	0.798 0.797	0.800 0.802	0.622 0.620	0.623 0.621	0.625 0.626	0.801 0.803													0.612
13		115	0.861 0.860	0.863 0.865	0.685 0.683	0.685 0.683	0.687 0.688	0.863 0.865													0.674
14		116	0.923 0.922	0.925 0.927	0.747 0.745	0.748 0.746	0.750 0.751	0.926 0.928	0.005	0.005	0.009	9.0	0.017	16.0	0.141 0.151	0.183 0.193	0.245 0.255	0.015 0.005	0.002	0.103 ± 0.003	0.737 ± 0.005

STATIC APPLICATIONS ONLY

101.4 (Continued)

O-RING GLAND DIMENSIONS

AN6227 DASH NO.	AN6230 DASH NO	MS28775 DASH NO.	SEAL INSTALLATION DIMENSIONS – EXTERNAL: "C" PISTON OR CYLINDER OD	EXTERNAL: "A" CYLINDER BORE OR MALE GLAND CYLINDER BORE ID	EXTERNAL: "F" GROOVE OD	INTERNAL: "B" ROD OR GLAND SLEEVE OD	INTERNAL: "H" ROD BORE OR FEMALE GLAND HOUSING BORE ID	INTERNAL: "E" GROOVE ID	DIAMETRAL CLEARANCE (a) "D" MAX EXT	DIAMETRAL CLEARANCE (a) "D" MAX INT	SQUEEZE MINIMUM ACTUAL	SQUEEZE MINIMUM PERCENT (REF)	SQUEEZE MAXIMUM ACTUAL	SQUEEZE MAXIMUM PERCENT (REF)	GROOVE WIDTH G: NO BACKUP RING	GROOVE WIDTH G: ONE BACKUP RING	GROOVE WIDTH G: TWO BACKUP RINGS	GROOVE CORNER RADIUS	ECCENTRICITY (c)	O-RING CROSS SECTION	O-RING INSIDE DIAMETER
		117	0.989 0.988	0.991 0.993	0.813 0.811	0.810 0.808	0.812 0.813	0.988 0.990	0.005	0.005	0.009	9.0	0.017	16.0	0.141 0.151	0.183 0.193	0.245 0.255	0.015 0.005	0.002	0.103 ± 0.003	0.799 ± 0.006
		118	1.051 1.050	1.053 1.055	0.875 0.873	0.873 0.871	0.875 0.876	1.051 1.053													0.862
		119	1.114 1.113	1.116 1.118	0.938 0.936	0.935 0.933	0.937 0.938	1.113 1.115													0.924
		120	1.176 1.175	1.178 1.180	1.000 0.998	0.998 0.996	1.000 1.001	1.176 1.178													0.987
		121	1.239 1.238	1.241 1.243	1.063 1.061	1.060 1.058	1.062 1.063	1.238 1.240													1.049
		122	1.301 1.300	1.303 1.305	1.125 1.123	1.123 1.121	1.125 1.126	1.301 1.303													1.112
		123	1.364 1.363	1.366 1.368	1.188 1.186	1.185 1.183	1.187 1.188	1.363 1.365													1.174
		124	1.426 1.425	1.428 1.430	1.250 1.248	1.248 1.246	1.250 1.251	1.426 1.428													1.237
		125	1.489 1.488	1.491 1.493	1.313 1.311	1.310 1.308	1.312 1.313	1.488 1.490													1.299
		126	1.551 1.550	1.553 1.555	1.375 1.373	1.373 1.371	1.375 1.376	1.551 1.553		0.005											1.362
		127	1.614 1.613	1.616 1.618	1.438 1.436	1.435 1.433	1.437 1.439	1.613 1.615		0.006											1.424
		128	1.676 1.675	1.678 1.680	1.500 1.498	1.498 1.496	1.500 1.502	1.676 1.678													1.487 ± 0.006
		129	1.739 1.738	1.741 1.743	1.563 1.561	1.560 1.558	1.562 1.564	1.738 1.740	0.005												1.549 ± 0.010
		130	1.802 1.801	1.805 1.807	1.627 1.625	1.623 1.621	1.625 1.627	1.801 1.803	0.006												1.612
		131	1.864 1.863	1.867 1.869	1.689 1.687	1.685 1.683	1.687 1.689	1.863 1.865													1.674
		132	1.927 1.926	1.930 1.932	1.752 1.750	1.748 1.746	1.750 1.752	1.926 1.928		0.006											1.737
		133	1.989 1.988	1.992 1.994	1.814 1.812	1.810 1.808	1.813 1.815	1.988 1.990		0.007											1.799
		134	2.052 2.051	2.055 2.057	1.877 1.875	1.873 1.871	1.876 1.878	2.051 2.053													1.862
		135	2.115 2.114	2.118 2.120	1.940 1.938	1.936 1.934	1.939 1.941	2.114 2.116													1.925
		136	2.177 2.176	2.180 2.182	2.002 2.000	1.998 1.996	2.001 2.003	2.176 2.178													1.987
		137	2.240 2.239	2.243 2.245	2.065 2.063	2.061 2.059	2.064 2.066	2.239 2.241													2.050
		138	2.302 2.301	2.305 2.307	2.127 2.125	2.123 2.121	2.126 2.128	2.301 2.303													2.112
		139	2.365 2.364	2.368 2.370	2.190 2.188	2.186 2.184	2.189 2.191	2.364 2.366													2.175
		140	2.427 2.426	2.430 2.432	2.252 2.250	2.248 2.246	2.251 2.253	2.426 2.428	0.006												2.237
		141	2.490 2.488	2.493 2.495	2.315 2.313	2.311 2.309	2.314 2.316	2.489 2.491	0.007												2.300
		142	2.552 2.550	2.555 2.557	2.377 2.375	2.373 2.371	2.376 2.378	2.551 2.553													2.362
		143	2.615 2.613	2.618 2.620	2.440 2.438	2.436 2.434	2.439 2.441	2.614 2.616													2.425
		144	2.677 2.675	2.680 2.682	2.502 2.500	2.498 2.496	2.501 2.503	2.676 2.678													2.487
		145	2.740 2.738	2.743 2.745	2.565 2.563	2.561 2.559	2.564 2.566	2.739 2.741													2.550
		146	2.802 2.800	2.805 2.807	2.627 2.625	2.623 2.621	2.626 2.628	2.801 2.803													2.612 ± 0.010
		147	2.865 2.863	2.868 2.870	2.690 2.688	2.686 2.684	2.689 2.691	2.864 2.866													2.675 ± 0.015
		148	2.927 2.925	2.930 2.932	2.752 2.750	2.748 2.746	2.751 2.753	2.926 2.928													2.737
		149	2.990 2.888	2.993 2.995	2.815 2.813	2.811 2.809	2.814 2.816	2.989 2.991	0.007	0.007	0.009	9.0	0.017	16.0	0.141 0.151	0.183 0.193	0.245 0.255	0.015 0.005	0.002	0.103 ± 0.003	2.800 ± 0.015

STATIC APPLICATIONS ONLY

101.4 (Continued)

O-RING GLAND DIMENSIONS

AN6227 DASH NO.	AN6230 DASH NO.	MS28775 DASH NO.	SEAL INSTALLATION DIMENSIONS – EXTERNAL: "C" PISTON OR CYLINDER OD	EXTERNAL: "A" CYLINDER BORE OR MALE GLAND CYLINDER BORE ID	EXTERNAL: "F" GROOVE OD	INTERNAL: "B" ROD OR GLAND SLEEVE OD	INTERNAL: "H" ROD BORE OR FEMALE GLAND HOUSING BORE ID	INTERNAL: "E" GROOVE ID	DIAMETRAL CLEARANCE (a) "D" MAX EXT	DIAMETRAL CLEARANCE (a) "D" MAX INT	SQUEEZE MINIMUM ACTUAL	SQUEEZE MINIMUM PERCENT (REF)	SQUEEZE MAXIMUM ACTUAL	SQUEEZE MAXIMUM PERCENT (REF)	GROOVE WIDTH G NO BACKUP RING	GROOVE WIDTH G ONE BACKUP RING	GROOVE WIDTH G TWO BACKUP RINGS	GROOVE CORNER RADIUS	ECCENTRICITY (c)	O-RING CROSS SECTION	O-RING INSIDE DIAMETER
15		210	0.989 0.988	0.991 0.993	0.748 0.746	0.748 0.746	0.750 0.751	0.991 0.993	0.005	0.005	0.0115	8.5	0.0215	15.0	0.188 0.198	0.235 0.245	0.304 0.314	0.025 0.010	0.003	0.139 ± 0.004	0.734 ± 0.006
16		211	1.051 1.050	1.053 1.055	0.810 0.808	0.810 0.808	0.812 0.813	1.053 1.055													0.796
17		212	1.114 1.113	1.116 1.118	0.873 0.871	0.873 0.871	0.875 0.876	1.116 1.118													0.859
18		213	1.176 1.175	1.178 1.180	0.935 0.933	0.935 0.933	0.937 0.938	1.178 1.180													0.921
19		214	1.239 1.238	1.241 1.243	0.998 0.996	0.998 0.996	1.000 1.001	1.241 1.243													0.984
20		215	1.301 1.300	1.303 1.305	1.060 1.058	1.060 1.058	1.062 1.063	1.303 1.305													1.046
21		216	1.364 1.363	1.366 1.368	1.123 1.121	1.123 1.121	1.125 1.126	1.366 1.368													1.109
22		217	1.426 1.425	1.428 1.430	1.185 1.183	1.185 1.183	1.187 1.188	1.428 1.430													1.171
23		218	1.489 1.488	1.491 1.493	1.248 1.246	1.248 1.246	1.250 1.251	1.491 1.493													1.234
24		219	1.551 1.550	1.553 1.555	1.310 1.308	1.310 1.308	1.312 1.313	1.553 1.555													1.296
25		220	1.614 1.613	1.616 1.618	1.373 1.371	1.373 1.371	1.375 1.376	1.616 1.618													1.359
26		221	1.676 1.675	1.678 1.680	1.435 1.433	1.435 1.433	1.437 1.438	1.678 1.680													1.421
27		222	1.739 1.738	1.741 1.743	1.498 1.496	1.498 1.496	1.500 1.501	1.741 1.743	0.005	0.005											1.484 ± 0.006
	1	223	1.864 1.863	1.867 1.869	1.624 1.622	1.623 1.621	1.625 1.627	1.866 1.868	0.006	0.006											1.609 ± 0.010
	2	224	1.989 1.988	1.992 1.994	1.749 1.747	1.748 1.746	1.750 1.752	1.991 1.993		0.006											1.734
	3	225	2.115 2.114	2.118 2.120	1.875 1.873	1.873 1.871	1.876 1.878	2.116 2.118		0.007											1.859
	4	226	2.240 2.239	2.243 2.245	2.000 1.998	1.998 1.996	2.001 2.003	2.241 2.243													1.984
	5	227	2.365 2.364	2.368 2.370	2.125 2.123	2.123 2.121	2.126 2.128	2.366 2.368	0.006												2.109
	6	228	2.490 2.488	2.493 2.495	2.250 2.248	2.248 2.246	2.251 2.253	2.491 2.493	0.007												2.234
	7	229	2.615 2.613	2.618 2.620	2.375 2.373	2.373 2.371	2.376 2.378	2.616 2.618													2.359
	8	230	2.740 2.738	2.743 2.745	2.500 2.498	2.498 2.496	2.501 2.503	2.741 2.743													2.484
	9	231	2.865 2.863	2.868 2.870	2.625 2.623	2.623 2.621	2.626 2.628	2.866 2.868													2.609 ± 0.010
	10	232	2.990 2.988	2.993 2.995	2.750 2.748	2.748 2.746	2.751 2.753	2.991 2.993													2.734 ± 0.015
	11	233	3.115 3.113	3.118 3.120	2.875 2.873	2.873 2.871	2.876 2.878	3.116 3.118													2.859
	12	234	3.240 3.238	3.243 3.245	3.000 2.998	2.997 2.995	3.000 3.002	3.240 3.242													2.984
	13	235	3.365 3.363	3.368 3.370	3.125 3.123	3.122 3.120	3.125 3.127	3.365 3.367													3.109
	14	236	3.490 3.488	3.493 3.495	3.250 3.248	3.247 3.245	3.250 3.252	3.490 3.492													3.234
	15	237	3.615 3.613	3.618 3.620	3.375 3.373	3.372 3.370	3.375 3.377	3.615 3.617													3.359
	16	238	3.740 3.738	3.743 3.745	3.500 3.498	3.497 3.495	3.500 3.502	3.740 3.742													3.484
	17	239	3.865 3.863	3.868 3.870	3.625 3.623	3.622 3.620	3.625 3.627	3.865 3.867													3.609
	18	240	3.990 3.988	3.993 3.995	3.750 3.748	3.747 3.745	3.750 3.752	3.990 3.992													3.734
	19	241	4.115 4.113	4.118 4.120	3.875 3.873	3.872 3.870	3.875 3.877	4.115 4.117													3.859
	20	242	4.240 4.238	4.243 4.245	4.000 3.998	3.997 3.995	4.000 4.002	4.240 4.242													3.984
	21	243	4.365 4.363	4.368 4.370	4.125 4.123	4.122 4.120	4.125 4.127	4.365 4.367	0.007	0.007	0.0115	8.5	0.0215	15.0	0.188 0.196	0.235 0.245	0.304 0.314	0.025 0.010	0.003	0.139 ± 0.004	4.109 ± 0.015

STATIC APPLICATIONS ONLY

101.4 (Continued)

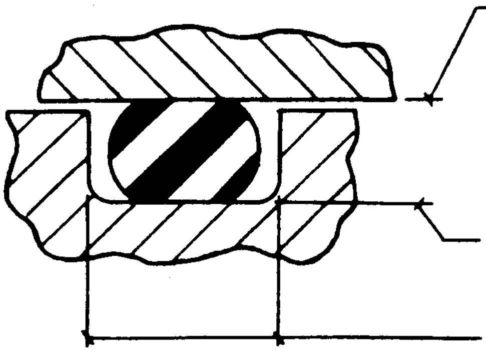
	AN6227 DASH NO.	AN6230 DASH NO.	MS28775 DASH NO.	O-RING GLAND DIMENSIONS — SEAL INSTALLATION DIMENSIONS — EXTERNAL — "C" PISTON OR CYLINDER OD	EXTERNAL — "A" CYLINDER BORE OR MALE GLAND CYLINDER BORE ID	EXTERNAL — "F" GROOVE OD	INTERNAL — "B" ROD OR GLAND SLEEVE OD	INTERNAL — "H" ROD BORE OR FEMALE GLAND HOUSING BORE ID	INTERNAL — "E" GROOVE ID	DIAMETRAL CLEARANCE (a) "D" MAX — EXT	DIAMETRAL CLEARANCE (a) "D" MAX — INT	SQUEEZE — MINIMUM — ACTUAL	SQUEEZE — MINIMUM — PERCENT (REF)	SQUEEZE — MAXIMUM — ACTUAL	SQUEEZE — MAXIMUM — PERCENT (REF)	GROOVE WIDTH G — NO BACKUP RING	GROOVE WIDTH G — ONE BACKUP RING	GROOVE WIDTH G — TWO BACKUP RINGS	GROOVE CORNER RADIUS	ECCENTRICITY (c)	O-RING CROSS SECTION	O-RING INSIDE DIAMETER
STATIC APPLICATIONS ONLY		22	244	4.489 4.487	4.493 4.495	4.250 4.248	4.247 4.245	4.250 4.252	4.490 4.492	0.008	0.007	0.0115	8.5	0.0215	15.0	0.188 0.198	0.235 0.245	0.304 0.314	0.025 0.010	0.003	0.139 ± 0.004	4.234 ± 0.015
STATIC APPLICATIONS ONLY		23	245	4.614 4.612	4.618 4.620	4.375 4.373	4.372 4.370	4.375 4.377	4.615 4.617		0.007											4.359
STATIC APPLICATIONS ONLY		24	246	4.739 4.737	4.743 4.745	4.500 4.498	4.497 4.495	4.501 4.503	4.740 4.742		0.008											4.484
STATIC APPLICATIONS ONLY		25	247	4.864 4.862	4.868 4.870	4.625 4.623	4.622 4.620	4.626 4.628	4.865 4.867	0.008	0.008	0.0115	8.5	0.0215	15.0	0.188 0.198	0.235 0.245	0.304 0.314	0.025 0.010	0.003	0.139 ± 0.004	4.609 ± 0.015
	28		325	1.864 1.863	1.867 1.869	1.495 1.493	1.498 1.496	1.500 1.502	1.870 1.872	0.006	0.006	0.017	8.3	[illegible]	13.5	0.281 0.291	0.334 0.344	0.424 0.434	0.035 0.020	0.004	0.210 ± 0.005	1.475 ± 0.010
	29		326	1.989 1.988	1.992 1.994	1.620 1.618	1.623 1.621	1.625 1.627	1.995 1.997													1.600
	30		327	2.115 2.114	2.118 2.120	1.746 1.744	1.748 1.746	1.750 1.752	2.120 2.122		0.006											1.725
	31		328	2.240 2.239	2.243 2.245	1.871 1.869	1.873 1.871	1.876 1.878	2.245 2.247		0.007											1.850
	32		329	2.365 2.364	2.368 2.370	1.996 1.994	1.998 1.996	2.001 2.003	2.370 2.372	0.006												1.975
	33		330	2.490 2.488	2.493 2.495	2.121 2.119	2.123 2.121	2.126 2.128	2.495 2.497	0.007												2.100
	34		331	2.615 2.613	2.618 2.620	2.246 2.244	2.248 2.246	2.251 2.253	2.620 2.622													2.225
	35		332	2.740 2.738	2.743 2.745	2.371 2.369	2.373 2.371	2.376 2.378	2.745 2.747													2.350
	36		333	2.865 2.863	2.868 2.870	2.496 2.494	2.498 2.496	2.501 2.503	2.870 2.872													2.475
	37		334	2.990 2.988	2.993 2.995	2.621 2.619	2.623 2.621	2.626 2.628	2.995 2.997													2.600 ± 0.010
	38		335	3.115 3.113	3.118 3.120	2.746 2.744	2.748 2.746	2.751 2.753	3.120 3.122													2.725 ± 0.015
	39		336	3.240 3.238	3.243 3.245	2.871 2.869	2.873 2.871	2.876 2.878	3.245 3.247			0.017	8.3									2.850
	40		337	3.365 3.363	3.368 3.370	2.996 2.994	2.997 2.995	3.000 3.002	3.369 3.371			0.017	8.1									2.975
	41		338	3.490 3.488	3.493 3.495	3.121 3.119	3.122 3.120	3.125 3.127	3.494 3.496													3.100
	42		339	3.615 3.613	3.618 3.620	3.246 3.244	3.247 3.245	3.250 3.252	3.619 3.621													3.225
	43		340	3.740 3.738	3.743 3.745	3.371 3.369	3.372 3.370	3.375 3.377	3.744 3.746													3.350
	44		341	3.865 3.863	3.868 3.870	3.496 3.494	3.497 3.495	3.500 3.502	3.869 3.871													3.475
	45		342	3.990 3.988	3.993 3.995	3.621 3.619	3.622 3.620	3.625 3.627	3.994 3.996													3.600
	46		343	4.115 4.113	4.118 4.120	3.746 3.744	3.747 3.745	3.750 3.752	4.119 4.121													3.725
	47		344	4.240 4.238	4.243 4.245	3.871 3.869	3.872 3.870	3.875 3.877	4.244 4.246													3.850
	48		345	4.365 4.363	4.368 4.370	3.996 3.994	3.997 3.995	4.000 4.002	4.369 4.371	0.007												3.975
	49		346	4.489 4.487	4.493 4.495	4.121 4.119	4.122 4.120	4.125 4.127	4.494 4.496	0.008												4.100
	50		347	4.614 4.612	4.618 4.620	4.246 4.244	4.247 4.245	4.250 4.252	4.619 4.621													4.225
	51		348	4.739 4.737	4.743 4.745	4.371 4.369	4.372 4.370	4.375 4.377	4.744 4.746													4.350
	52		349	4.864 4.862	4.868 4.870	4.496 4.494	4.497 4.495	4.500 4.502	4.869 4.871	0.008	0.007	0.017	8.1	0.029	13.5	0.281 0.291	0.334 0.344	0.424 0.434		0.004	0.210 ± 0.005	4.475
	88		425	4.970 4.968	4.974 4.977	4.497 4.494	4.497 4.494	4.501 4.503	4.974 4.977	0.009	0.009	0.0275	10.2	0.0425	15.1	0.375 0.385	0.475 0.485	0.579 0.589		(e)	0.275 ± 0.006	4.475
	53		426	5.095 5.093	5.099 5.102	4.622 4.619	4.622 4.619	4.626 4.628	5.099 5.102													4.600
	54		427	5.220 5.218	5.224 5.227	4.747 4.744	4.747 4.744	4.751 4.753	5.224 5.227													4.725
	55		428	5.345 5.343	5.349 5.352	4.872 4.869	4.872 4.869	4.876 4.878	5.349 5.352													4.850
	56		429	5.470 5.468	5.474 5.477	4.997 4.994	4.997 4.994	5.001 5.003	5.474 5.477	0.009	0.009	0.0275	10.2	0.0425	15.1	0.375 0.385	0.475 0.485	0.579 0.589	0.035 0.020	(e)	0.275 ± 0.006	4.975 ± 0.015

101.4 (Continued)

O-RING GLAND DIMENSIONS

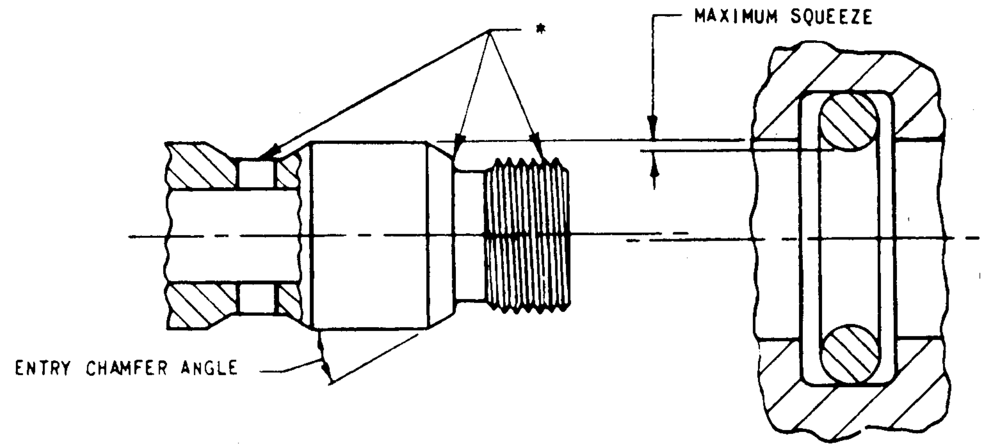
AN6227 DASH NO.	AN6230 DASH NO.	MS28775 DASH NO.	SEAL INSTALLATION DIMENSIONS – EXTERNAL: "C" PISTON OR CYLINDER OD	EXTERNAL: "A" CYLINDER BORE OR MALE GLAND CYLINDER BORE ID	EXTERNAL: "F" GROOVE OD	INTERNAL: "B" ROD OR GLAND SLEEVE OD	INTERNAL: "H" ROD BORE OR FEMALE GLAND HOUSING BORE ID	INTERNAL: "E" GROOVE ID	DIAMETRAL CLEARANCE (a) "D" MAX: EXT	DIAMETRAL CLEARANCE (a) "D" MAX: INT	SQUEEZE MINIMUM: ACTUAL	SQUEEZE MINIMUM: PERCENT (REF)	SQUEEZE MAXIMUM: ACTUAL	SQUEEZE MAXIMUM: PERCENT (REF)	GROOVE WIDTH G: NO BACKUP RING	GROOVE WIDTH G: ONE BACKUP RING	GROOVE WIDTH G: TWO BACKUP RINGS	GROOVE CORNER RADIUS	ECCENTRICITY (c)	O-RING CROSS SECTION	O-RING INSIDE DIAMETER
57		430	5.595 5.593	5.599 5.602	5.122 5.119	5.122 5.119	5.126 5.128	5.599 5.602	0.009	0.009	0.0275	10.2	0.0425	15.1	0.375 0.385	0.475 0.485	0.579 0.589	0.035 0.020	(e)	0.275 ± 0.006	5.100 ± 0.023
58		431	5.720 5.718	5.724 5.727	5.247 5.244	5.247 5.244	5.251 5.253	5.724 5.727													5.225
59		432	5.845 5.843	5.849 5.852	5.372 5.369	5.372 5.369	5.376 5.378	5.849 5.852													5.350
60		433	5.970 5.968	5.974 5.977	5.497 5.494	5.497 5.494	5.501 5.503	5.974 5.977													5.475
61		434	6.095 6.093	6.099 6.102	5.622 5.619	5.622 5.619	5.626 5.628	6.099 6.102													5.600
62		435	6.220 6.218	6.224 6.227	5.747 5.744	5.747 5.744	5.751 5.753	6.224 6.227													5.725
63		436	6.345 6.343	6.349 6.352	5.872 5.869	5.872 5.869	5.876 5.878	6.349 6.352													5.850
64		437	6.470 6.468	6.474 6.477	5.997 5.994	5.997 5.994	5.501 5.503	6.474 6.477													5.975
65		438	6.720 6.718	6.724 6.727	6.247 6.244	6.247 6.244	6.251 6.253	6.724 6.727		0.009											6.225
66		439	6.970 6.968	6.974 6.977	6.497 6.494	6.497 6.494	6.501 6.504	6.974 6.977		0.010											6.475
67		440	7.220 7.218	7.224 7.227	6.747 6.744	6.747 6.744	6.751 6.754	7.224 7.227													6.725
68		441	7.470 7.468	7.474 7.477	6.997 6.994	6.997 6.994	7.001 7.004	7.474 6.477													6.975 ± 0.023
69		442	7.720 7.718	7.724 7.727	7.247 7.244	7.247 7.244	7.251 7.254	7.724 7.727													7.225 ± 0.030
70		443	7.970 7.968	7.974 7.977	7.497 7.494	7.497 7.494	7.501 7.504	7.974 7.977													7.475
71		444	8.220 8.218	8.224 8.227	7.747 7.744	7.747 7.744	7.751 7.754	8.224 8.227													7.725
72		445	8.470 8.468	8.474 8.477	7.997 7.994	7.997 7.994	8.001 8.004	8.474 8.477	0.009												7.975
73		446	8.970 8.967	8.974 8.977	8.497 8.494	8.497 8.494	8.501 8.504	8.974 8.977	0.010		0.0275	10.2									8.475
74		447	9.470 9.467	9.474 9.478	8.997 8.994	8.997 8.994	9.001 9.004	9.474 9.478	0.011		0.027	9.9									8.975
75		448	9.970 9.967	9.974 9.978	9.497 9.494	9.497 9.494	9.501 9.504	9.974 9.978													9.475
76		449	10.470 10.467	10.474 10.478	9.997 9.994	9.997 9.994	10.001 10.004	10.474 10.478													9.975
77		450	10.970 10.967	10.974 10.978	10.497 10.494	10.497 10.494	10.501 10.504	10.974 10.978													10.475
78		451	11.470 11.467	11.474 11.478	10.997 10.994	10.997 10.994	11.001 11.004	11.474 11.478													10.975
79		452	11.970 11.967	11.974 11.978	11.497 11.494	11.497 11.494	11.501 11.504	11.974 11.978													11.475
80		453	12.470 12.467	12.474 12.478	11.997 11.994	11.997 11.994	12.001 12.004	12.474 12.478													11.975
81		454	12.970 12.967	12.974 12.978	12.497 12.494	12.497 12.494	12.501 12.504	12.974 12.978													12.475
82		455	13.470 13.467	13.474 13.478	12.997 12.994	12.997 12.994	13.001 13.004	13.474 13.478													12.975
83		456	13.970 13.967	13.974 13.978	13.497 13.494	13.497 13.494	13.501 13.504	13.974 13.978													13.475
84		457	14.470 14.467	14.474 14.478	13.997 13.994	13.997 13.994	14.001 14.004	14.474 14.478													13.975
85		458	14.970 14.967	14.974 14.978	14.497 14.494	14.497 14.494	14.501 14.504	14.974 14.978													14.475
86		459	15.470 15.467	15.474 15.478	14.997 14.994	14.997 14.994	15.001 15.004	15.474 15.478													14.975
87		460	15.970 15.967	15.974 15.978	15.497 15.494	15.497 15.494	15.501 15.504	15.974 15.978	0.011	0.010	0.027	9.9	0.0425	15.1	0.375 0.385	0.475 0.485	0.579 0.589	0.035 0.020	(e)	0.275 ± 0.006	15.475 ± 0.030

101.41 <u>O-RING PACKING GLAND SURFACE ROUGHNESS</u> The following surface roughness designations are to be called out on drawings for O-ring packing glands. The static seal requirements refer to both radial-seal and face-seal O-rings.

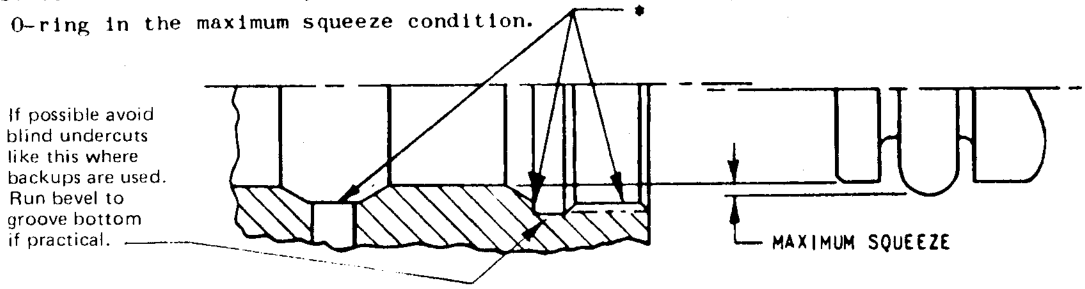


101.5 PACKING ENTRY

101.51 O-RING ENTRY PROVISIONS Units should be designed so that sharp corners including threads and serrations clear all backed and unbacked O-rings during assembly and disassembly. Entry chamfers are used as "shoehorns" to avoid cutting O-rings. The nominal entry chamfer angle should be 15 degrees if space allows, and should not exceed 30 degrees, but plus and minus tolerances on the 30-degree value are permissible.



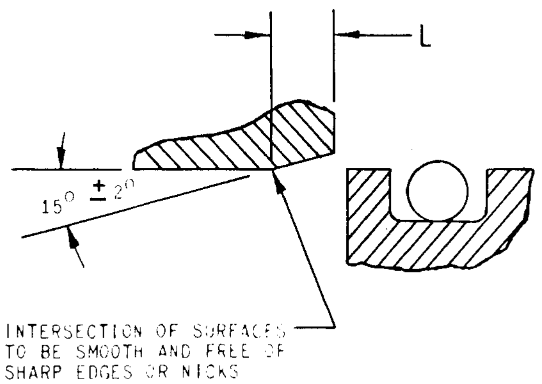
* These corners should clear, at adverse tolerances including eccentricities, the O-ring in the maximum squeeze condition.



$$\text{MAXIMUM SQUEEZE} = W - \frac{\text{MIN. GLAND OD} - \text{MAX. GLAND ID} - E}{2}$$

WHERE: W IS MAXIMUM CROSS SECTION DIAMETER TAKEN FROM THE O-RING DRAWING. (CHANGE IN O-RING CROSS SECTION DUE TO INSTALLATION IN THE GLAND IS NEGLECTED). E IS MAXIMUM FULL INDICATOR READING ECCENTRICITY BETWEEN OD AND ID OF GLAND.

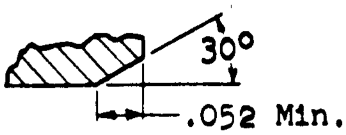
101.52 PREFERRED ENTRY BEVEL Where space permits, the following 15-degree entry chamfer is preferred to 30 degrees.



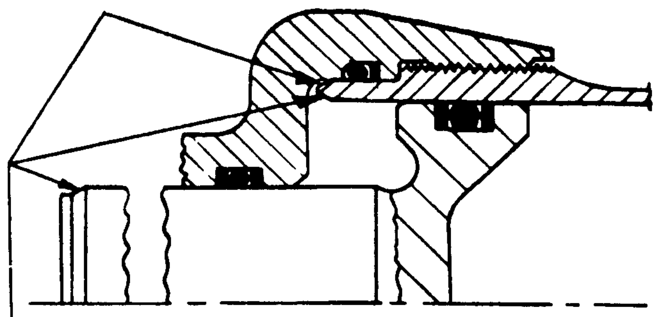
NOM. O-RING SECT.	MS28775 AND NAS1611 DASH NO.	PACKING RING SIZE AN6227	L ±.010
1/16	006 TO 028	–1 TO –7	.093
3/32	110 TO 149	–8 TO –14	.093
1/8	210 TO 247	–15 TO –27	.105
3/16	325 TO 349	–28 TO –52	.135
1/4	425 TO 460	–53 TO –88	.187
		AN6230	
1/8		–1 TO –25	.105

101.53 TEFLON BACK-UP RING ENTRY Unless chamfers are large enough, teflon back-up rings may be damaged when installed. The back-up that enters the chamfer before the O-ring requires no more entry chamfer than the O-ring, but the back-up that follows the O-ring may jam or be cut unless at least 1/32-inch radial entry is provided. However cylinder barrel inside end chamfers and piston rod chamfers need no more chamfer than specified in the Hydraulics Manual, since special guides may be used where necessary.

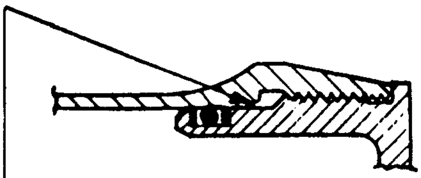
REQUIREMENT: All chamfers where a teflon back-up enters after the O-ring must be 30 degrees (tolerance ±2 degrees or less) times at least 0.052-inch minimum tolerance axial length; except for barrel inside end chamfers and piston rod chamfers. (Chamfer angles less than 30 degrees are also suitable, providing the minimum radial chamfer is 1/32 inch.) See examples below.



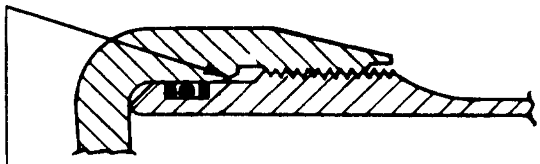
Required Chamfer



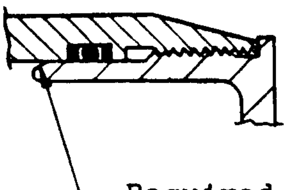
Chamfers noted in section 60 are sufficient.



Required Chamfer



Required Chamfer



Required Chamfer

101.6 SPECIAL DYNAMIC SEALS

101.61 SLIPPER SEAL

.611 GENERAL INFORMATION The S3891431 (internal) and S3891430 (external) teflon slipper rings provide dynamic seals for axial and rotary motion and for frequent short-stroke oscillation, when used in conjunction with NAS1611 or MS28775 O-ring packings. The combination of slipper ring and packing is most commonly known as a slipper seal. Slipper seals are used in the identical gland as an O-ring packing with one back-up ring. Radial squeeze of the packing provides the sealing force of the slipper ring against the mating member.

Slipper seals have two advantages over conventional O-ring packings. The first of these is lower break-out friction, particularly after a long stand-by. Running friction of slipper seals can be higher or lower than that of O-ring packings depending on the surface finish of the mating member, and the radial squeeze, which is a function of the tolerance stack-up of the gland and mating member diameters.

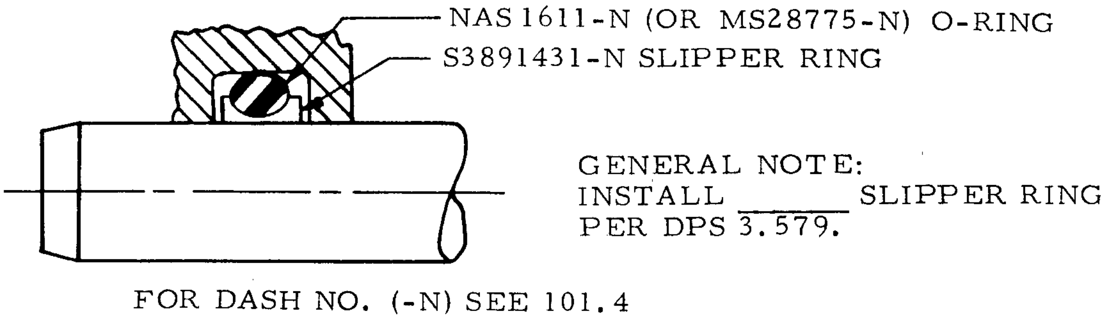
The second advantage of slipper seals is that the O-ring becomes a static seal, while the teflon slipper ring takes the abuse in dynamic applications where O-ring packings alone have shown rapid deterioration. Examples of such applications are: (1) In short-stroke, high-frequency reciprocating motion; (2) Under conditions which tend to produce spiral failure such as excessively long actuator stroke or excessively large ratio of packing ID to its cross-sectional diameter; (3) In high-pressure rotary devices such as swivel fittings.

Slipper seals are to be used in unidirectional pressure applications only; they should never be used where a reversal of pressure across the seal can occur. The exception to this rule is in rotary or oscillatory devices where there is no axial motion.

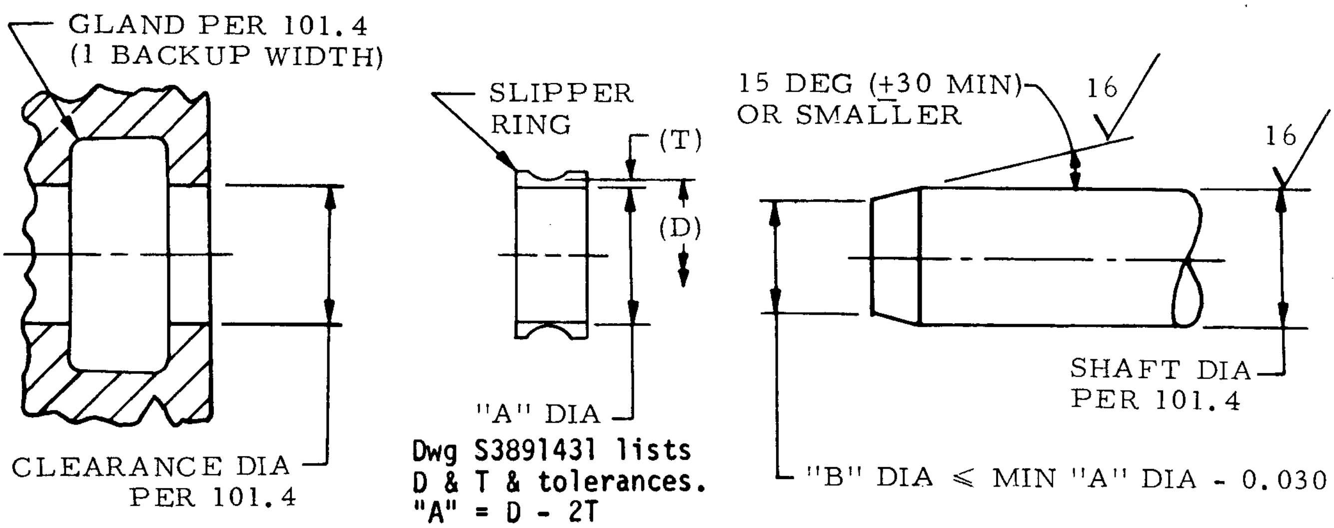
Slipper seals generally have a slightly higher leakage rate than O-ring packings because teflon is naturally less tolerant of sealing surface imperfections. Leakage, like running friction, is a function of radial squeeze and surface finish.

101.612 SLIPPER SEAL FOR INTERNAL GLAND

a. Assembly



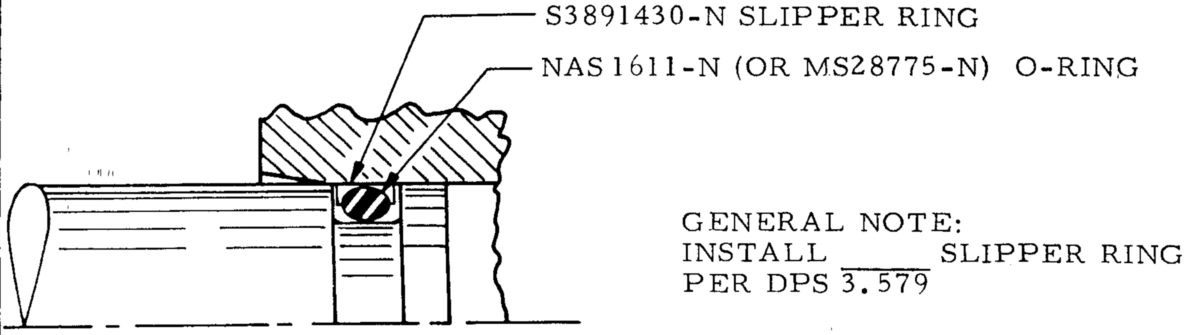
b. Shaft and Gland



SLIPPER SEAL DESIGN

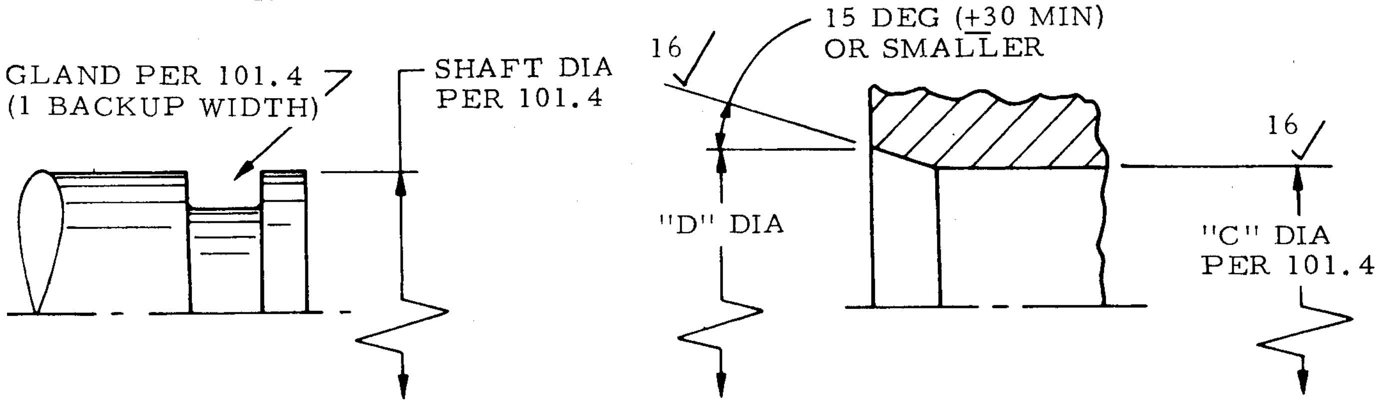
101.613 SLIPPER SEAL FOR EXTERNAL GLAND

a. Assembly



FOR DASH NO. (-N) SEE 101.4

b. Housing and Gland



SLIPPER DASH NO. (REF S3891430)	"D" DIAMETER
-004 to -028	"C" dia. + .036 to .046
-110 to -149	"C" dia. + .036 to .046
-210 to -247	"C" dia. + .046 to .056
-325 to -349	"C" dia. + .060 to .070
-425 to -460	"C" dia. + .088 to .100

Use housing entry chamfer shown above whenever possible to avoid special installation processing per DPS 3.579. If for design reasons a different chamfer is required, change the general note to:

INSTALL ____________ SLIPPER RING & FREEZE IN PLACE PER DPS 3.579 BEFORE INSTALLING (shaft)

101.62 TEFLON PISTON RING DESIGN In order to reduce piston head seal friction to acceptable levels in applications where a slight amount of leakage can be tolerated, the teflon piston ring has been successfully utilized. In addition to extremely good friction characteristics throughout a differential pressure range from zero to 3000 psi, the teflon piston ring has the following qualities:

a. Extremely rugged and long wearing.

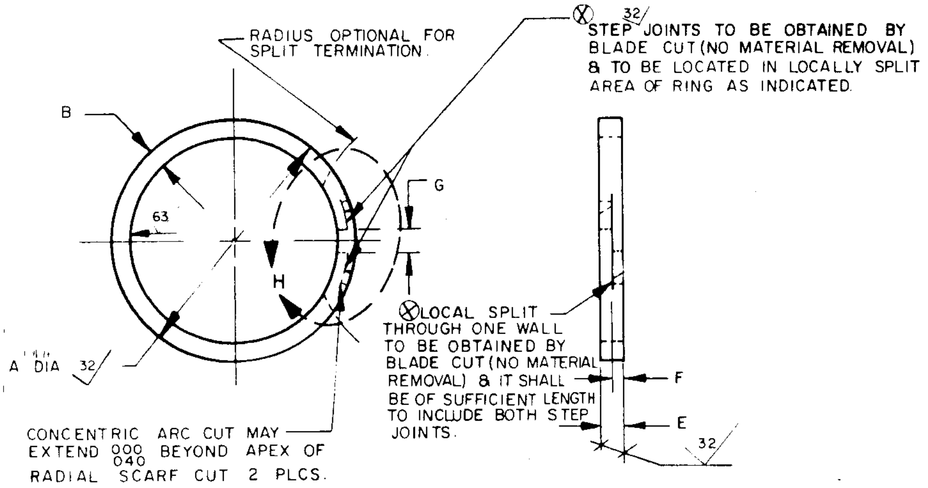
b. Very compact in design, permitting use of pistons with minimal thickness.

c. Superior to O-rings in regard to backlash caused by seal shift with pressure reversal. This feature is important in servomechanism stability and freedom from flight control surface flutter.

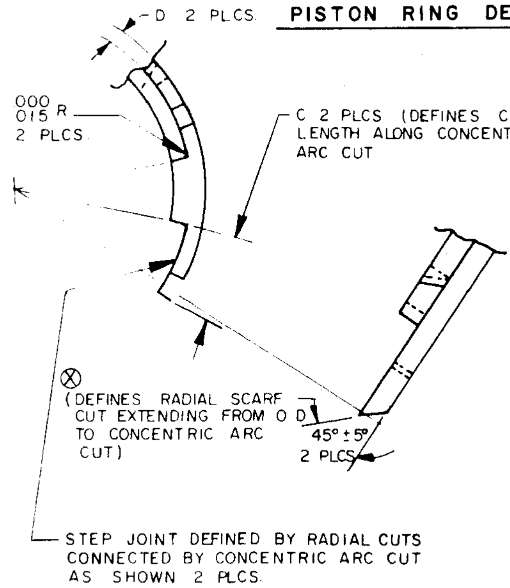
d. Bidirectional sealing capability.

.621 THE SPLIT PISTON RING The split piston ring is used on the DC-10 and is to be used for future design. Information from the standards manual is reproduced below as a convenient reference.

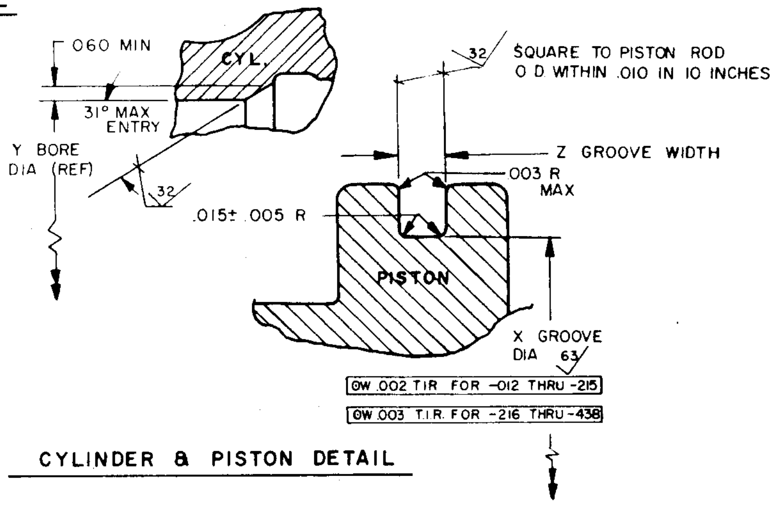
101.621 (Continued)



PISTON RING DETAIL



VIEW H
ONE END SHOWN FOR CLARITY



CYLINDER & PISTON DETAIL

MATERIAL:
TEFLON PER MIL-R-8791

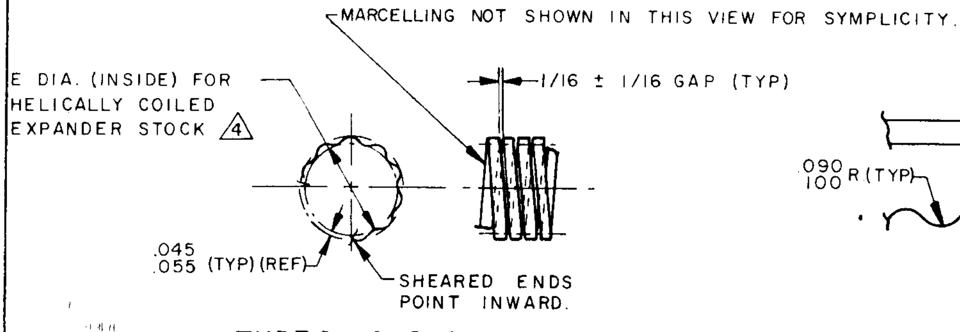
NOTES:

1. ▲ THIS SERIES OF PISTON RINGS IS DESIGNED TO FIT CYLINDER BORES SPECIFIED BY DAC HYDRAULIC DESIGN MANUAL.
2. ⊗ ALL EDGES PRODUCED BY BLADE CUT TO BE SHARP WITH NO CRUSHING & MINIMAL DISTORTION OF MATERIAL.
3. ★ DESIGNATES ENGINEERING REFERENCE INFORMATION ONLY.
4. PISTON RING DASH NUMBERS CORRESPOND TO DASH NUMBERS OF ARP568 UNIFORM DASH NUMBERING SYSTEM FOR O-RINGS.
5. PISTON RING DASH NUMBERS CORRESPOND TO THOSE OF S492971 3 EXPANDER WHICH ARE DESIGNATED FOR USE THEREWITH.

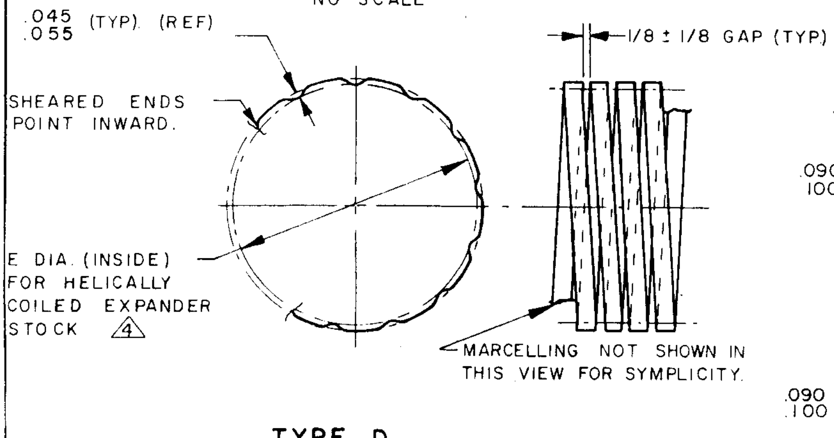
101.621 (Continued)

DASH NO.	PISTON RING DIMENSIONS A DIA	B ±.001	C ±.015	D ±.010	E ±.001	F ±.010	G ±.015	★ CYLINDER & PISTON GROOVE DIMENSIONS X DIA (GROOVE) USE WITH S4929713 EXPANDER	▲Y DIA (CYL. BORE)	Z ±.001
012	.484	.093	.187	.046	.121	.061	.156		.485 +.001 −.000	.126
110	.549								.550	
111	.612								.613	
112	.674	.093	.187	.046			.156		.675	
113	.737 ±.001	.125	.250	.062			.218		.738	
114	.799							.491	.800	
115	.862							.554	.863	
116	.924							.616	.925	
210	.990							.862	.991	
211	1.052							.744	1.053	
212	1.114							.807	1.116	
213	1.176							.869	1.178	
214	1.239							.932	1.241	
215	1.301	.125	.250	.062	.121	.061	.218	.994	1.303	.126
216	1.364	.187	.312	.093	.182	.091	.281	.933	1.366 +.002 −.000	.189
217	1.426							.995	1.428	
218	1.489							1.058	1.491	
219	1.551							1.120	1.553	
220	1.614							1.183	1.616	
221	1.676							1.245	1.678	
222	1.739							1.308	1.741	
325	1.865							1.434	1.867	
326	1.990							1.559	1.992	
327	2.116							1.685	2.118	
328	2.241							1.810	2.243	
329	2.366							1.935	2.368	
330	2.491							2.060	2.493	
331	2.616 ±.002							2.185	2.618	
332	2.741							2.310	2.743	
333	2.866							2.435	2.868	
334	2.991							2.560	2.993	
335	3.116							2.685 ±.002	3.118	
336	3.241							2.810	3.243	
337	3.366							2.935	3.368	
338	3.491							3.060	3.493	
339	3.616							3.185	3.618	
340	3.741							3.310	3.743	
341	3.866							3.435	3.868	
342	3.991							3.560	3.993	
343	4.116							3.685	4.118	
344	4.240							3.810	4.243	
345	4.365							3.935	4.368	
346	4.490							4.060	4.493	
347	4.615							4.185	4.618	
348	4.740							4.310	4.743	
349	4.865							4.435	4.868 +.003 −.000	
425	4.971							4.541	4.974	
426	5.096							4.666	5.099	
427	5.221							4.791	5.224	
428	5.346							4.916	5.349	
429	5.471 ±.005							5.041	5.474	
430	5.596							5.166	5.599	
431	5.721							5.291	5.724	
432	5.846							5.416	5.849	
433	5.971							5.541	5.974	
434	6.096							5.666	6.099	
435	6.221							5.791	6.224	
436	6.346							5.916	6.349	
437	6.471							6.041	5.474	
438	6.721	.187	.312	.093	.182	.091	.281	6.291	5.724	.189

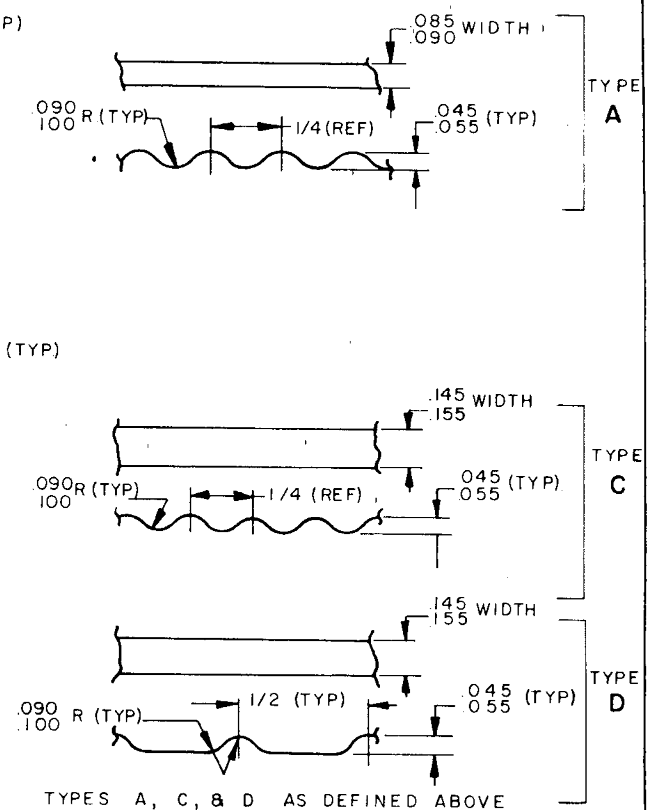
101.621 (Continued)



TYPES A & C
NO SCALE



TYPE D



TYPES A, C, & D AS DEFINED ABOVE DESCRIBE FORMED STRIP PRIOR TO BEING TRANSFORMED INTO HELICAL COILS.

FOR USE WITH S3929694 PISTON RING

<table>
<tr><th>DASH NO.</th><th>FREE LENGTH</th><th></th><th>TYPE</th><th>E</th></tr>
<tr><td>114</td><td>1 3/8</td><td rowspan="22">+1/32 -1/4</td><td rowspan="9">A</td><td rowspan="4">5/8 ± 1/16</td></tr>
<tr><td>115</td><td>1 9/16</td></tr>
<tr><td>116</td><td>1 3/4</td></tr>
<tr><td>210</td><td>1 15/16</td></tr>
<tr><td>211</td><td>2 1/8</td><td rowspan="5">3/4 ± 1/16</td></tr>
<tr><td>212</td><td>2 5/16</td></tr>
<tr><td>213</td><td>2 1/2</td></tr>
<tr><td>214</td><td>2 11/16</td></tr>
<tr><td>215</td><td>2 7/8</td></tr>
<tr><td>216</td><td>2 11/16</td><td rowspan="13">C</td><td rowspan="8">1 ± 1/16</td></tr>
<tr><td>217</td><td>2 7/8</td></tr>
<tr><td>218</td><td>3 1/16</td></tr>
<tr><td>219</td><td>3 3/16</td></tr>
<tr><td>220</td><td>3 3/8</td></tr>
<tr><td>221</td><td>3 9/16</td></tr>
<tr><td>222</td><td>3 3/4</td></tr>
<tr><td>325</td><td>4 3/16</td></tr>
<tr><td>326</td><td>4 9/16</td><td rowspan="5">1 1/2 ± 1/16</td></tr>
<tr><td>327</td><td>4 15/16</td></tr>
<tr><td>328</td><td>5 1/4</td></tr>
<tr><td>329</td><td>5 5/8</td></tr>
<tr><td>330</td><td>6</td></tr>
<tr><td>331</td><td>6 9/16</td><td rowspan="6">+1/32 -1/2</td><td rowspan="6">D</td><td rowspan="6">2 1/4 ± 1/8</td></tr>
<tr><td>332</td><td>6 15/16</td></tr>
<tr><td>333</td><td>7 5/16</td></tr>
<tr><td>334</td><td>7 11/16</td></tr>
<tr><td>335</td><td>8 1/16</td></tr>
<tr><td>336</td><td>8 7/16</td></tr>
</table>

<table>
<tr><th>DASH NO.</th><th>FREE LENGTH</th><th></th><th>TYPE</th><th>E</th></tr>
<tr><td>337</td><td>8 13/16</td><td rowspan="28">+ 1/32 - 1/2</td><td rowspan="28">D</td><td rowspan="2">2 1/4 ± 1/8</td></tr>
<tr><td>338</td><td>9 1/8</td></tr>
<tr><td>339</td><td>9 1/2</td><td rowspan="10">3 1/4 ± 1/8</td></tr>
<tr><td>340</td><td>9 13/16</td></tr>
<tr><td>341</td><td>10 1/4</td></tr>
<tr><td>342</td><td>10 5/8</td></tr>
<tr><td>343</td><td>11</td></tr>
<tr><td>344</td><td>11 3/8</td></tr>
<tr><td>345</td><td>11 3/4</td></tr>
<tr><td>346</td><td>12 1/8</td></tr>
<tr><td>347</td><td>12 1/2</td></tr>
<tr><td>348</td><td>12 7/8</td></tr>
<tr><td>349</td><td>13 1/4</td><td rowspan="16">4 1/2 ± 1/4</td></tr>
<tr><td>425</td><td>13 5/8</td></tr>
<tr><td>426</td><td>14</td></tr>
<tr><td>427</td><td>14 5/16</td></tr>
<tr><td>428</td><td>14 11/16</td></tr>
<tr><td>429</td><td>15 1/16</td></tr>
<tr><td>430</td><td>15 7/16</td></tr>
<tr><td>431</td><td>15 7/8</td></tr>
<tr><td>432</td><td>16 1/4</td></tr>
<tr><td>433</td><td>16 5/8</td></tr>
<tr><td>434</td><td>17</td></tr>
<tr><td>435</td><td>17 3/8</td></tr>
<tr><td>436</td><td>17 3/4</td></tr>
<tr><td>437</td><td>18 1/8</td></tr>
<tr><td>438</td><td>18 7/8</td></tr>
</table>

MATERIAL:

.003 THICK 12-9-2 CRES PER DMS 1893

NOTES:

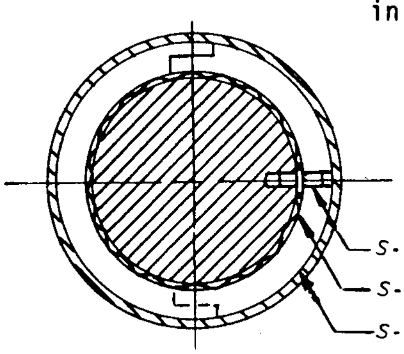
1. DASH NUMBERS OF THIS EXPANDER CORRESPOND TO THOSE DASH NUMBERS OF THE NOTED PISTON RING THEY ARE USED WITH.
2. HT TR HELICALLY COILED MARCEL STOCK TO STA 950 PER D.P.S. 500.
3. FINISH PER D.P.S. 9.07.
4. EXPANDERS SHALL BE SHEARED FROM HELICALLY COILED MARCEL STOCK AS NOTED. COIL DIRECTION IS OPTIONAL.
5. IDENTIFY PER D.P.S. 3.02.
6. BREAK ALL SHARP EDGES & CORNERS.
7. EXPANDER SPRING FREE LENGTH IS DEFINED AS THE UNCOILED LENGTH OF THE EXPANDER SPRING WITH MINIMAL STRAIGHTENING FORCES IMPOSED UPON IT.

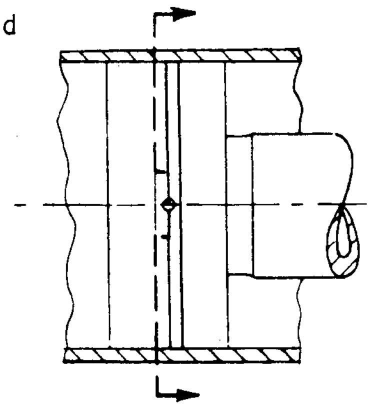
101.622 DUAL-STEP PISTON RING DESIGN

a. Assembly Materials and plating refer to cylinders for Navy aircraft noted in obsolete design section 41.9.

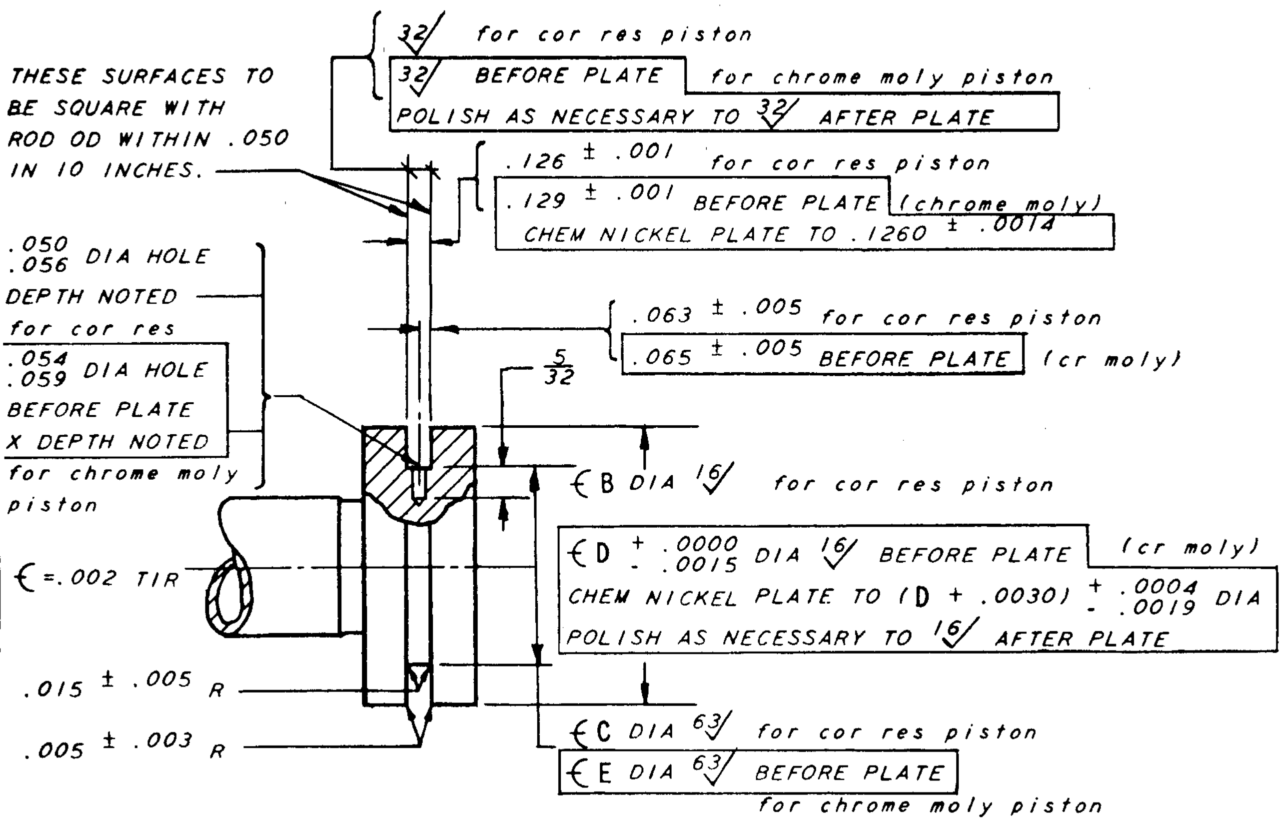
GENERAL NOTE:

FREE LENGTH OF S-4445080-A EXPANDER TO BE IN ACCORDANCE WITH DRAWING S-4445080





b. Piston Head The corrosion-resistant steel piston head is not plated. The cr moly steel piston head is chemical nickel plated 0.0015 ± 0.0002 thick.



c. Barrel ID Provide entry chamfers on the ID at least as much as in Section 41.14. Where possible axial chamfer length should be 0.060 minimum tolerance times 30 degrees.

ALUMINUM BARREL	CORROSION RESISTANT STEEL BARREL
HARD COATED .0020 ± .0002 THICK USE WITH COR RES PISTON	USE WITH CHROME MOLY PISTON
F DIA 16 HARD COAT TO G DIA 32 CRITICAL WEAR AREA	H DIA 16

101.622 (Continued)

NOMINAL HEAD DIA	A	B +.000 −.002	C ±.004	D +.0000 −.0015	E ±.004	F	G	H
1/2	8	.497	.255	.4935	.252	.5012	.499	.499
9/16	9	.559	.317	.5555	.314	.5637	.5615	.5615
5/8	10	.6215	.380	.6180	.377	.6262	.624	.624
11/16	11	.684	.442	.6805	.439	.6887	.6865	.6865
3/4	12	.7465	.441	.7430	.438	.7512	.749	.749
13/16	13	.809	.503	.8055	.500	.8137	.8115	.8115
7/8	14	.8715	.566	.8680	.563	.8762	.874	.874
15/16	15	.934	.628	.9305	.625	.9387	.9365	.9365
1	16	.997	.692	.9935	.689	1.0022	1.000	1.000
1 1/16	17	1.059	.754	1.0555	.751	1.0642	1.062	1.062
1 1/8	18	1.122	.817	1.1185	.814	1.1272	1.125	1.125
1 3/16	19	1.184	.879	1.1805	.876	1.1892	1.187	1.187
1 1/4	20	1.247	.942	1.2435	.939	1.2522	1.250	1.250
1 5/16	21	1.309	1.004	1.3055	1.001	1.3142	1.312	1.312
1 3/8	22	1.372	1.067	1.3685	1.064	1.3772	1.375	1.375
1 7/16	23	1.434	1.005	1.4305	1.002	1.4392	1.437	1.437
1 1/2	24	1.497	1.068	1.4935	1.065	1.5022	1.500	1.500
1 9/16	25	1.559	1.130	1.5555	1.127	1.5642	1.562	1.562
1 5/8	26	1.622	1.193	1.6185	1.190	1.6272	1.625	1.625
1 11/16	27	1.684	1.255	1.6805	1.252	1.6892	1.687	1.687
1 3/4	28	1.747	1.318	1.7435	1.315	1.7522	1.750	1.750
1 13/16	29	1.809	1.380	1.8055	1.377	1.8142	1.812	1.812
1 7/8	30	1.872	1.443	1.8685	1.440	1.8772	1.875	1.875
1 15/16	31	1.934	1.505	1.9305	1.502	1.9392	1.937	1.937
2	32	1.997	1.568	1.9935	1.565	2.0022	2.000	2.000
2 1/16	33	2.059	1.630	2.0555	1.627	2.0642	2.062	2.062
2 1/8	34	2.122	1.693	2.1185	1.690	2.1272	2.125	2.125
2 3/16	35	2.184	1.755	2.1805	1.752	2.1892	2.187	2.187
2 1/4	36	2.247	1.818	2.2435	1.815	2.2522	2.250	2.250
2 5/16	37	2.309	1.880	2.3055	1.877	2.3142	2.312	2.312
2 3/8	38	2.372	1.943	2.3685	1.940	2.3772	2.375	2.375
2 7/16	39	2.434	2.005	2.4305	2.002	2.4392	2.437	2.437
2 1/2	40	2.497	2.068	2.4935	2.065	2.5022	2.500	2.500
2 9/16	41	2.559	2.130	2.5555	2.127	2.5647	2.5625	2.562
2 5/8	42	2.622	2.193	2.6185	2.190	2.6277	2.6255	2.625
2 11/16	43	2.684	2.255	2.6805	2.252	2.6897	2.6875	2.687
2 3/4	44	2.747	2.318	2.7435	2.315	2.7527	2.7505	2.750
2 13/16	45	2.809	2.380	2.8055	2.377	2.8147	2.8125	2.812
2 7/8	46	2.872	2.443	2.8685	2.440	2.8777	2.8755	2.875
2 15/16	47	2.934	2.505	2.9305	2.502	2.9397	2.9375	2.937
3	48	2.997	2.568	2.9935	2.565	3.0027	3.0005	3.000
3 1/16	49			3.0555	2.627			3.062
3 1/8	50			3.1185	2.690			3.125
3 3/16	51			3.1805	2.752			3.187
3 1/4	52			3.2435	2.815			3.250
3 5/16	53			3.3055	2.878			3.312
3 3/8	54			3.3685	2.941			3.375
3 7/16	55			3.4305	3.003			3.437
3 1/2	56			3.4935	3.066			3.500
3 9/16	57			3.5555	3.128			3.562
3 5/8	58			3.6185	3.191			3.625
3 11/16	59			3.6805	3.253			3.687
3 3/4	60			3.7435	3.316			3.750
3 13/16	61			3.8055	3.378			3.812
3 7/8	62			3.8685	3.441			3.875
3 15/16	63			3.9305	3.503			3.937
4	64			3.9935	3.566			4.000

F: +.0016 /−.0000 (1/2 through 2 1/2); +.0021/−.0000 (2 9/16 through 3)

G: +.002 / −.000 (1/2 through 2 1/2); +.0025/−.0000 (2 9/16 through 3)

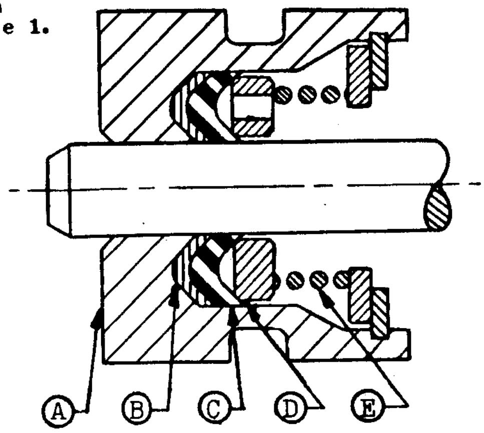
H: +.002/−.000 (1/2 through 3 1/4); +.003/−.000 (3 5/16 through 4)

101.63 CHEVRON PACKING DESIGN The only advantage of chevron packings is the low friction they sometimes produce at very low pressure, as rod seal glands. (Low friction piston head seals would be S-3929694 teflon piston rings as described in Section 101.621, except that chevron head packings would be used in airless-type reservoirs.) Chevron packing designs are complex, expensive, bulky and nonstandard and should not be used unless O-ring friction cannot be tolerated.

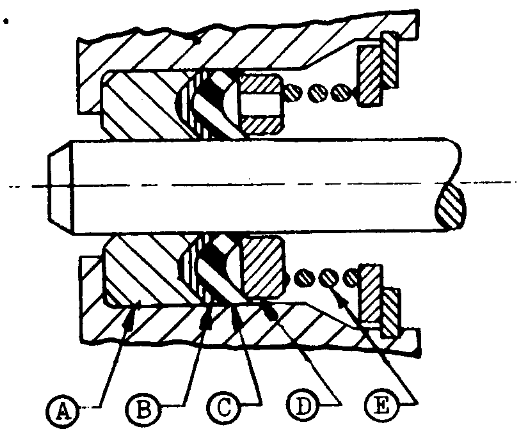
Tests indicate that the friction of chevron packings is not much less than that of low-squeeze O-rings except at pressures near zero, where little force is required in the direction that the chevron lip points, and where in the opposite direction break-out friction after pressurization (as in pressure switches) for some installations may be as low as half that of one O-ring. (Hydraulic Section Test Report 76 discusses low friction packings. Hydraulic Section Test Report 101 discusses chevron packings used on a power boost valve.)

It should be considered that, where a cylinder is returned by external forces faster than the rate of fluid inflow, creating a vacuum, the check valve nature of the chevron packing may allow air to be drawn in. There is some evidence that there is no way to prevent such leakage, particularly at low temperature.

Design Example 1.



Design Example 2.



(A) Female Adapter

1. Configuration to be determined by design.

2. Maximum diameter clearance should be as small as practicable, while allowing for eccentricities. It should normally not exceed 0.008 inch.

3. Packing seat dimensions should be per AN6228.

4. Example drawings: S-2434521 and 3437253.

101.63 CHEVRON PACKING DESIGN (Continued)

(B) Teflon Back-up Chevron

1. Use dash number of S-4388671.

2. Normally use where pressure exceeds 1500 psi, or at 500 psi and higher where female adapter diametric clearance exceeds 0.005 inch.

(C) Chevron Packing

1. Use dash number of S-2389536.

(D) Male Adapter

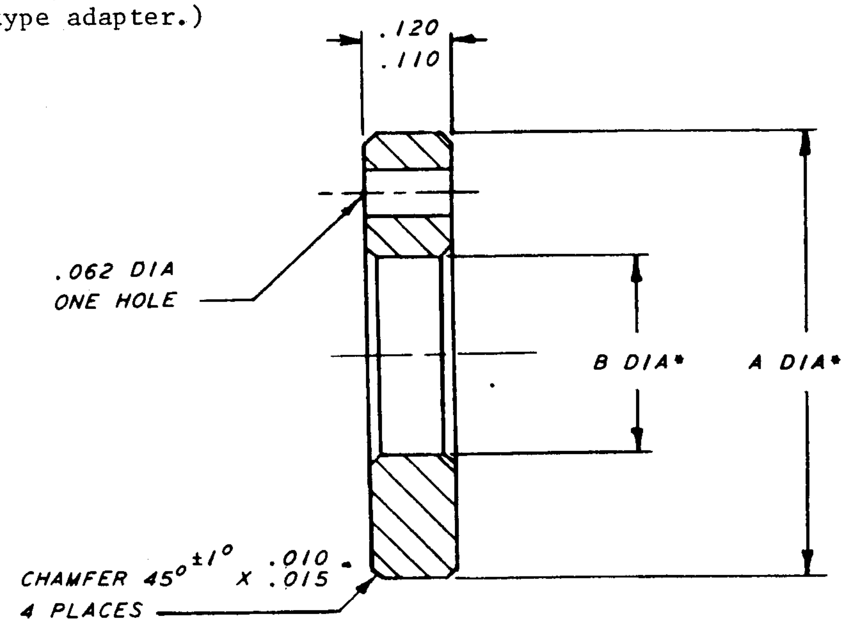
1. See below.

(E) Retaining Spring

1. Not a standard part.

2. Optimum spring force has not been established. The S-5432840 pressure switch uses approximately 4 pounds for 1/4 inch ID chevron; the 5444665 reservoir uses approximately 150 pounds for 7-inch ID chevron.

101.631 <u>BALANCED MALE ADAPTER FOR CHEVRON PACKING</u> In the event that a new design is made incorporating a chevron rod packing, a Douglas standard drawing should be set up as noted below for a male adapter for chevrons of 1-1/4 ID and smaller. (Hydraulic Section Test Report 101 discusses the features of this type adapter.)

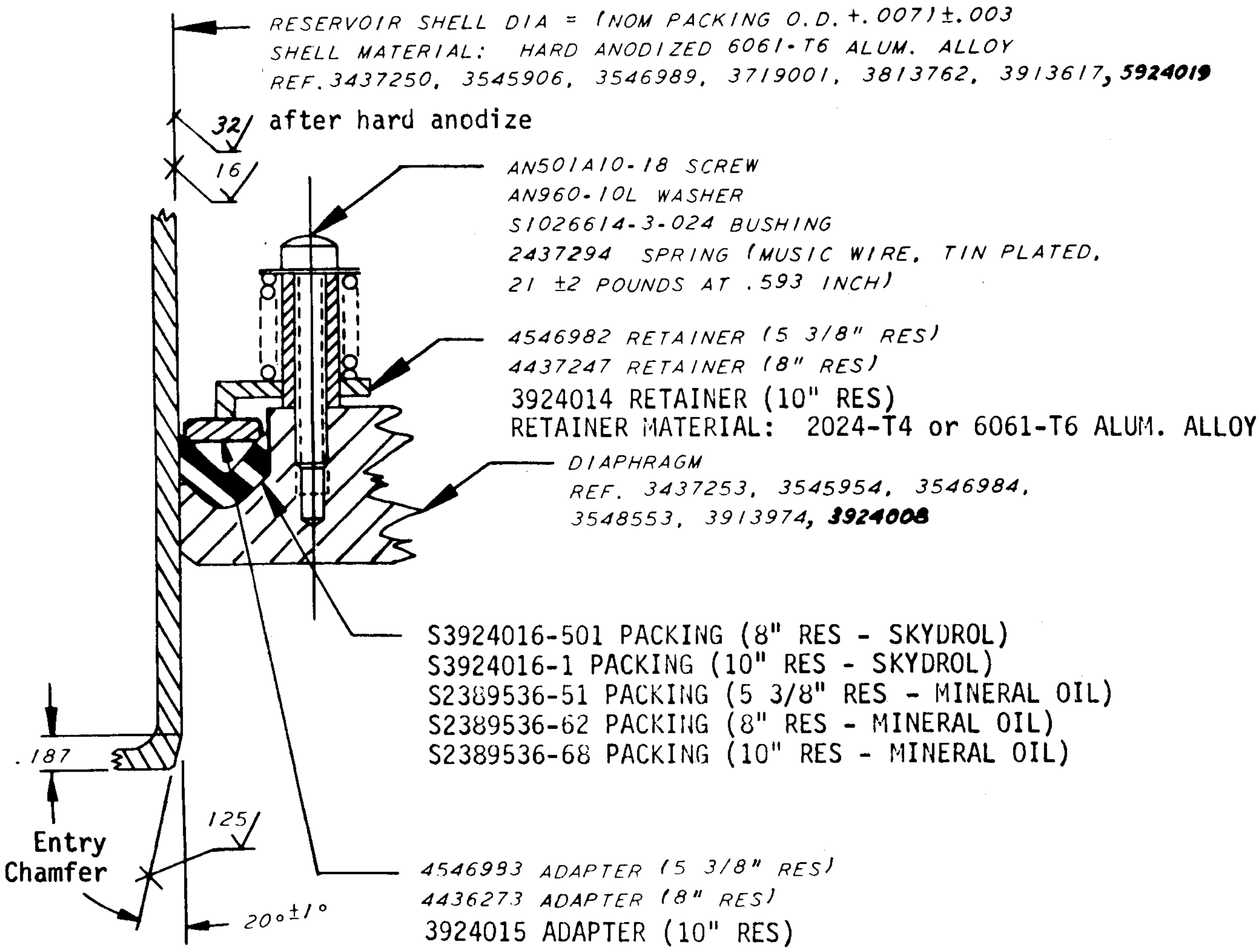


***A & B diameters to be same size as OD & ID of AN6229.**
***A & B diamerers to be concentric within .002 total indicator reading.**

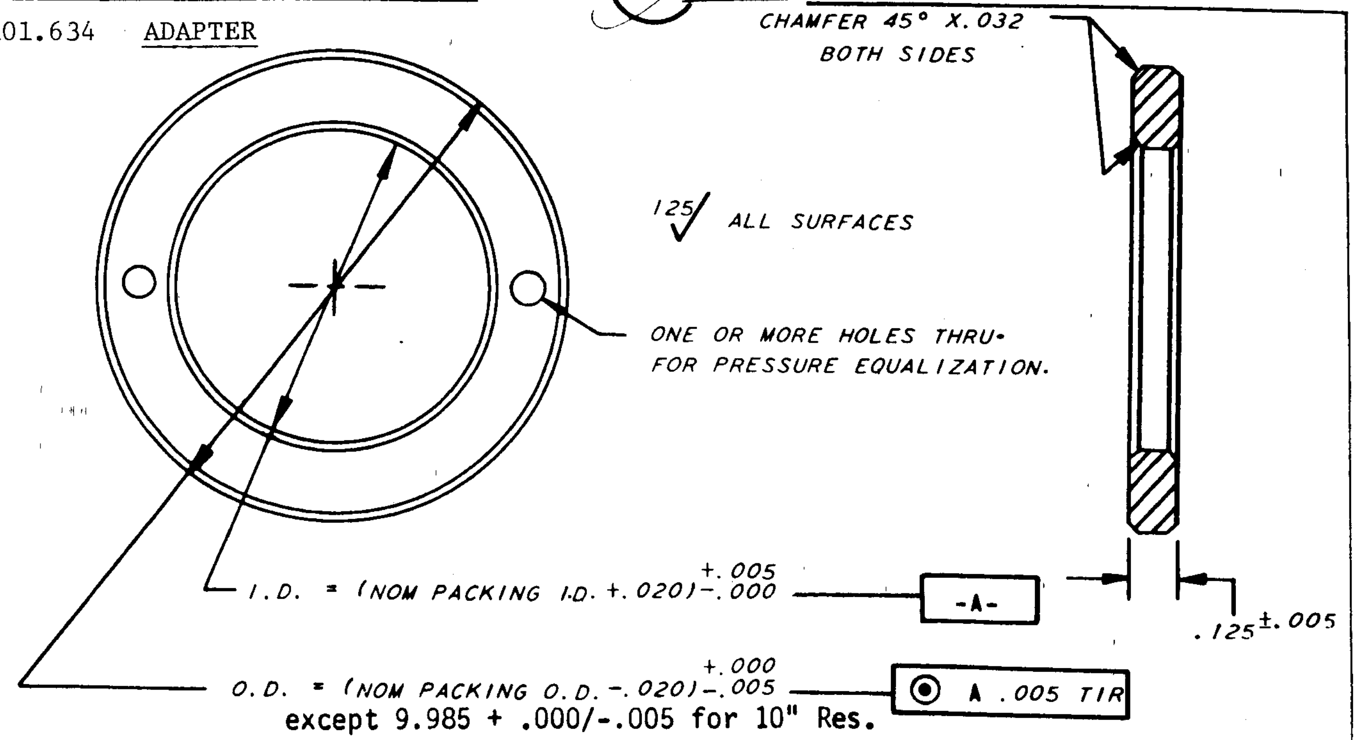
101.632 RESERVOIR PACKING DESIGN The current application of the "V"-ring packing is in airless reservoirs as the diaphragm seal between pressurized fluid and atmosphere. Although "V"-ring packings are available in a range of sizes from 0.5 to 16 inches OD, Douglas has used either a 5-3/8-, an 8- or a 10-inch reservoir bore diameter and the corresponding dash number packing. For sizes and dash numbers, refer to S3924016 for packings for use with Skydrol hydraulic fluid, or S2389536 for mineral oil applications.

Where practical reservoirs should be designed with a 5-3/8-, 8- or 10-inch bore in order to utilize existing adapters and retainers. See below for existing part numbers. Should a different diameter be required see below for gland and adapter design criteria. The new retainer should be designed to contact the mean diameter of the adapter, and maintain the existing spring installed height. As a rule of thumb provide one spring per inch of reservoir diameter. The same screw, washer, bushing and spring are used in all reservoir applications.

101.633 ASSEMBLY Ref 5444665, 5542535, 5542536, 5547510, 5553217, 5718470, 5813598, 5913704, 5924035



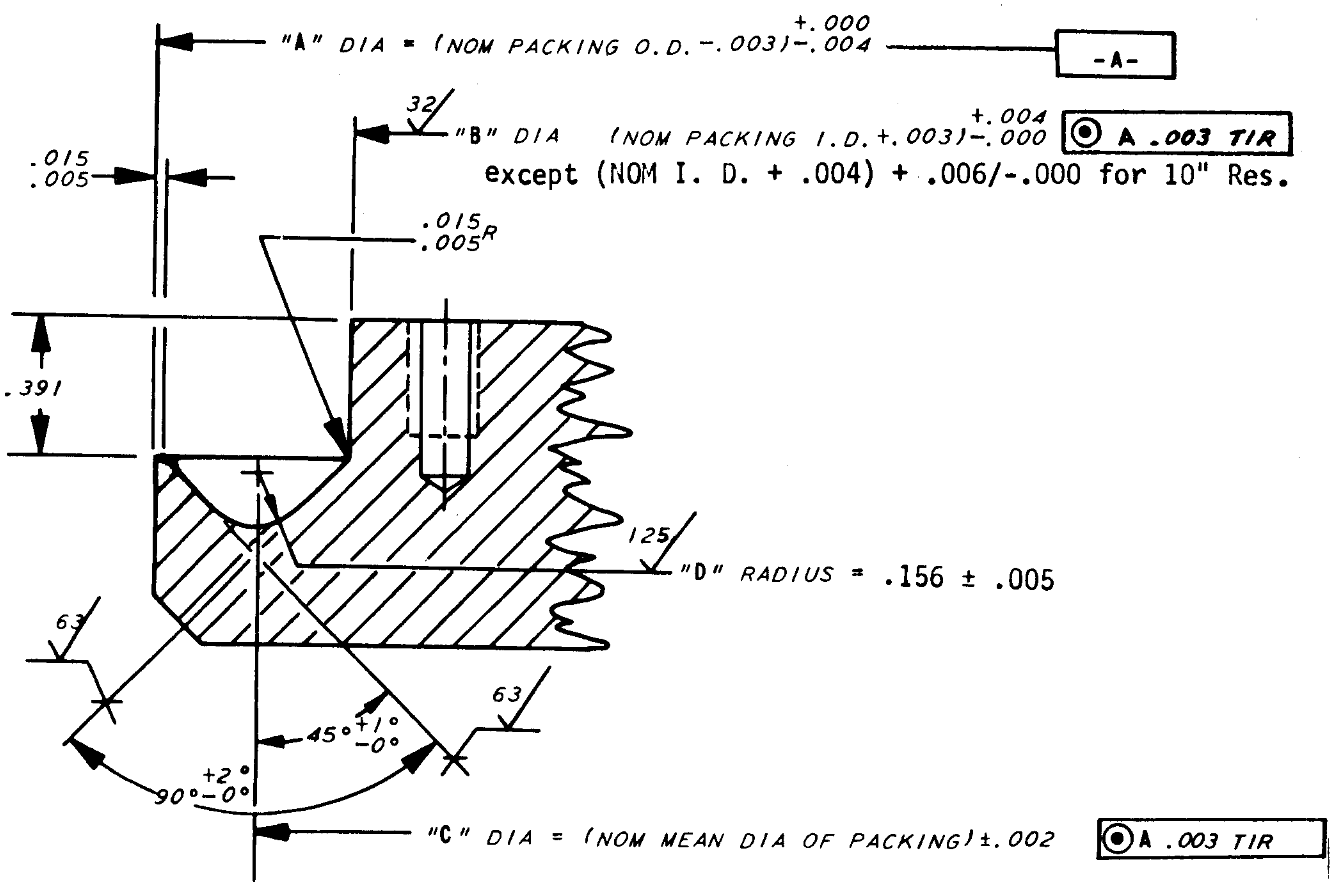
101.634 ADAPTER



except 9.985 + .000/-.005 for 10" Res.

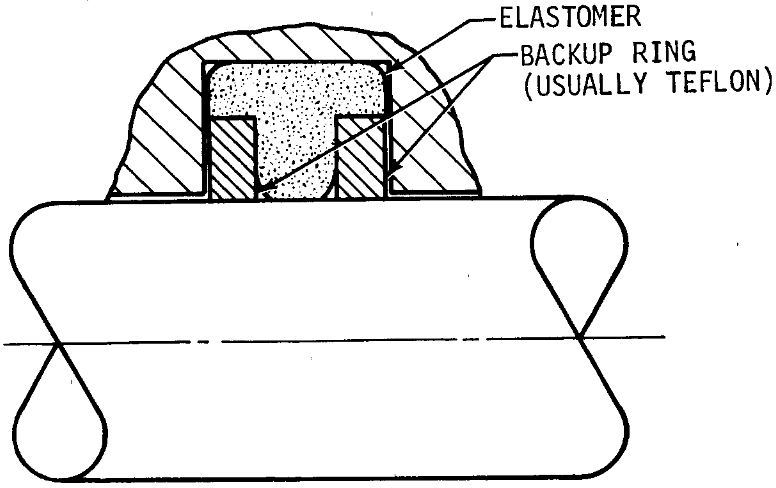
MATERIAL: Anodized 2024-T4 or -T3 ALUM. ALLOY

101.635 GLAND IN DIAPHRAGM (for nominal packing OD of 4-3/4 or larger)



MATERIAL: ALUM. ALLOY BAR OR CASTING AS APPLICABLE.

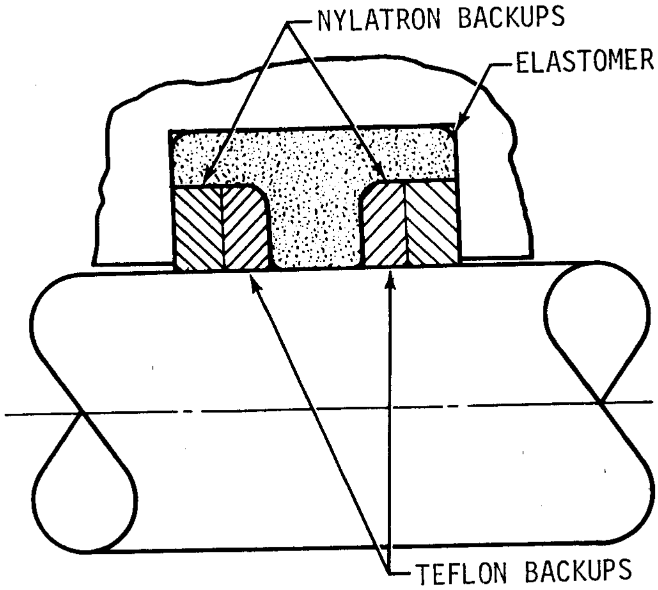
101.64 T-RING SEALS A special type of seal known as a T-ring has been on the market since the early 1950's. It was originally manufactured by the Greene-Tweed Company, but did not become popular in aircraft usage until about the middle 1960's. Patents on the basic concept have since expired and a similar version is now being produced in competition by the Rockwell Mechanical Packing Co., Inc., a subsidiary of the Parker Seal Co.



T-RING SEAL (ROD-TYPE)

The T-ring design features an elastomeric ring having a T-shaped cross section. It is made in three axial widths to fit MIL-G-5514 gland widths. Likewise, radial widths correspond to the O-ring cross sections. The T-ring carries its own anti-extrusion devices in the form of special plastic rings (normally scarf-cut) similar to MS28774 but comparatively narrower radially to permit its accommodation on either side of the elastomeric member. Because of the interlocking of the elastomer with these backup rings, it is immune to spiral-type failure.

101.64 (Continued)



T-RING SEAL WITH STAGED BACKUPS

In some instances dual or staged back-up rings are used to position a ring of a harder, more extrusion-resistant material (such as some suitable form of nylon) on the outside next to the gap(s) being sealed. The rings in contact with the elastomer are usually made from teflon. Such a combination is often used in shock strut seal applications.

In comparison with O-rings (with back-ups), T-rings can generally be characterized as having somewhat lower friction at very low pressures but higher friction at elevated operating pressures.

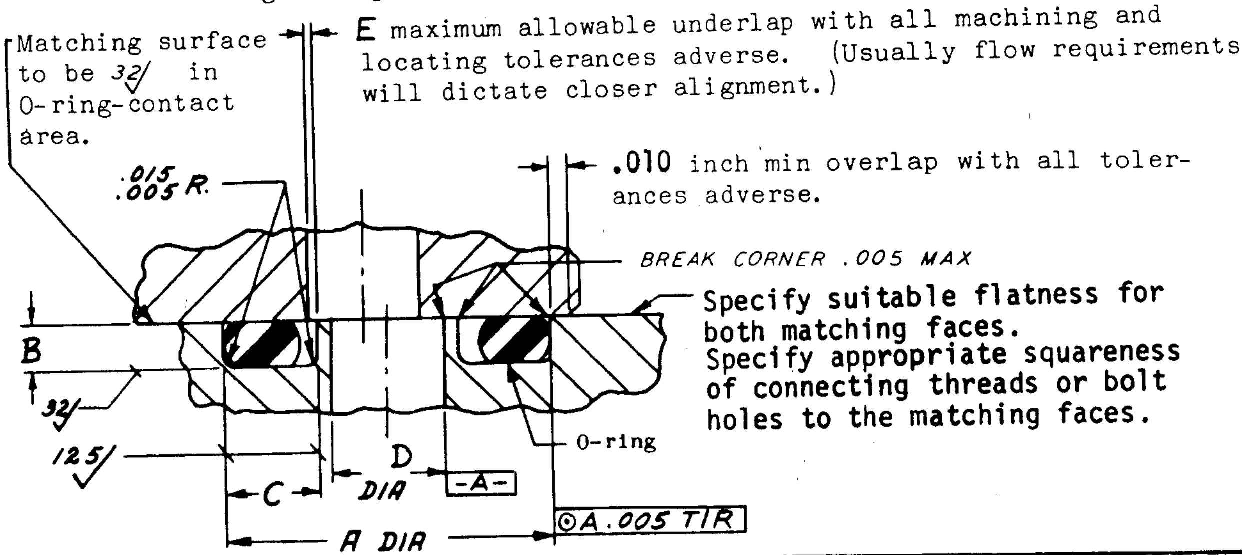
T-rings are now being used in the A-4 canopy actuator, the DC-9 main gear shock strut lower bearing seal, and in the following DC-10 applications:

Main gear lower bearing (dynamic)
Nose gear lower bearing (dynamic and static)
Centerline gear lower bearing (dynamic)
Main gear dual chamber floating piston
Main gear retract cylinder piston and piston rod
Slat cylinder piston

101.7 FACE SEALED JOINT

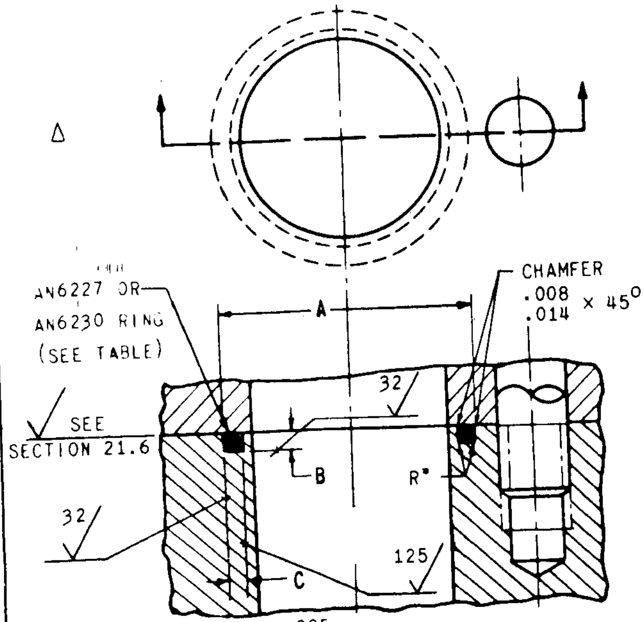
101.71 BACKED O-RING Wherever space permits, S3929237 teflon backup rings should be used with face seal O-rings. See S3929237 for gland dimensions. Maximum extrusion gap due to maximum operating pressure and out-of-flatness of mating faces must not exceed 0.004 inch.

101.72 UNBACKED O-RING The following gland dimensions have been used for small-size unbacked O-rings. (Minimum O-ring cross-section squeeze in these sizes is 0.010 inch. Maximum volumetric occupation is approximately 90 percent.) Also see 101.73 for a larger range of sizes and preload information.



NAS1611 or MS28775 O-ring	Equiv. AN6227 O-ring	A ±.005	B	C	D max drill	E max	Tube size with fitting ID near D
-008	AN6227-3	.328	.052 .057	.110±.003	.042 .047	.007	
-009	AN6227-4	.359		.110±.003	.071 .077		1/8
-010	AN6227-5	.391		.105±.005	.109 .116		3/16
-011	AN6227-6	.448		.105	.164 .172		1/4
-012	AN6227-7	.505	.052 .057	.105	.219 .227	.007	5/16
-111	AN6227-9	.625	.085 .090	.130	.288 .297	.010	3/8
-112	AN6227-10	.687	.085 .090	.130	.346 .355	.010	
-113	AN6227-11	.750	.085 .090	.130±.005	.411 .420	.010	1/2

101.73 FACE SEALED JOINT (ADDITIONAL INFORMATION)



* R = .005 / .010 THRU A = .938

R = .025 FOR LARGER SIZES

GIVEN: RING DIAMETER & PRESSURE

FIND: BOLT SIZE & TORQUE

SOLUTION:

1. DETERMINE TEST PRESSURE (PRELOAD BOLTS TO TEST PRESSURE).
2. DETERMINE AXIAL PRELOAD AT 1000 PSI FOR GIVEN RING DIAMETER FROM TABLE.
3. MULTIPLY THIS BY $\left(\frac{\text{TEST PRESSURE}}{1000}\right)$ = REQUIRED AXIAL PRELOAD DUE TO TEST PRESSURE.
4. Δ TOTAL AXIAL PRELOAD = AXIAL LOAD AT ZERO PSI PLUS AXIAL PRELOAD DUE TO PRESSURE.
5. ASSUME NUMBER OF BOLTS.
6. REQUIRED AXIAL PRELOAD PER BOLT = $\left(\frac{\text{REQUIRED TOTAL AXIAL PRELOAD}}{\text{NUMBER OF BOLTS}}\right)$ APPLIES ONLY TO BOLTS EQUALLY SPACED AROUND FACE SEAL; OTHERWISE ANALYZE LOAD DISTRIBUTION.
7. DETERMINE BOLT SIZE & TORQUE FROM SECTION 33.4 (SOLID BOLTS).
8. PRELOAD BOLTS TO TORQUE SHOWN IN SECTION 33.4. SPECIFY TORQUE ON DRAWING.

REF: AT SM SEE DEV-1004 AND DEV-1574.

RING NUMBER	NOMINAL RING OD	NOMINAL RING ID	A	B +.005 -.000	C ±.005	AXIAL PRELOAD REQD TO SEAL JOINT AT 1000 PSI (LBS)	AXIAL LOAD TO SEAL AT ZERO PSI (LBS)
AN6227-1	1/4	1/8	.249 ± .002	.055		49	32
-2	9/32	5/32	.281 ± .002			63	36
-3	5/16	3/16	.312 ± .003			77	40
-4	11/32	7/32	.343 ± .003			93	42
-5	3/8	1/4	.375 ± .004			110	48
-6	7/16	5/16	.437 ± .004			150	56
-7	1/2	3/8	.500 ± .004	.055		200	65
-8	9/16	3/8	.563 + .000 -.010	.085	.130	250	77
-9	5/8	7/16	.625			310	86
-10	11/16	1/2	.688			375	93
-11	3/4	9/16	.750			450	101
-12	13/16	5/8	.813			525	110
-13	7/8	11/16	.875			600	119
-14	15/16	3/4	.938	.085	.130	700	127
-16	1 1/16	13/16	1.063	.120	.165	790	155
-17	1 1/8	7/8	1.125			1000	164
-18	1 3/16	15/16	1.188			1110	174
-19	1 1/4	1	1.250			1230	183
-20	1 5/16	1 1/16	1.313			1360	192
-21	1 3/8	1 1/8	1.375			1490	201
-22	1 7/16	1 3/16	1.438			1625	210
-23	1 1/2	1 1/4	1.500			1770	219
-24	1 9/16	1 5/16	1.563			1920	228
-25	1 5/8	1 3/8	1.625			2075	235
-26	1 11/16	1 7/16	1.688			2240	245
-27	1 3/4	1 1/2	1.750			2410	255
AN6230-1	1 7/8	1 5/8	1.875			2765	118
-2	2	1 3/4	2.000			3140	126
-3	2 1/8	1 7/8	2.125			3550	134
-4	2 1/4	2	2.250			3980	142
-5	2 3/8	2 1/8	2.375			4430	150
-6	2 1/2	2 1/4	2.500			4910	158
-7	2 5/8	2 3/8	2.625			5415	166
-8	2 3/4	2 1/2	2.750 +.000 -.010	.120	.165	5940	174

101.8 SEAL FRICTION O-ring friction is quite often an important consideration in hydraulic design. Although considerable friction data is available, seal friction is so highly variable that very accurate prediction is almost impossible, especially at higher pressures. Some of the variables that affect seal friction and are responsible for friction scatter are as follows:

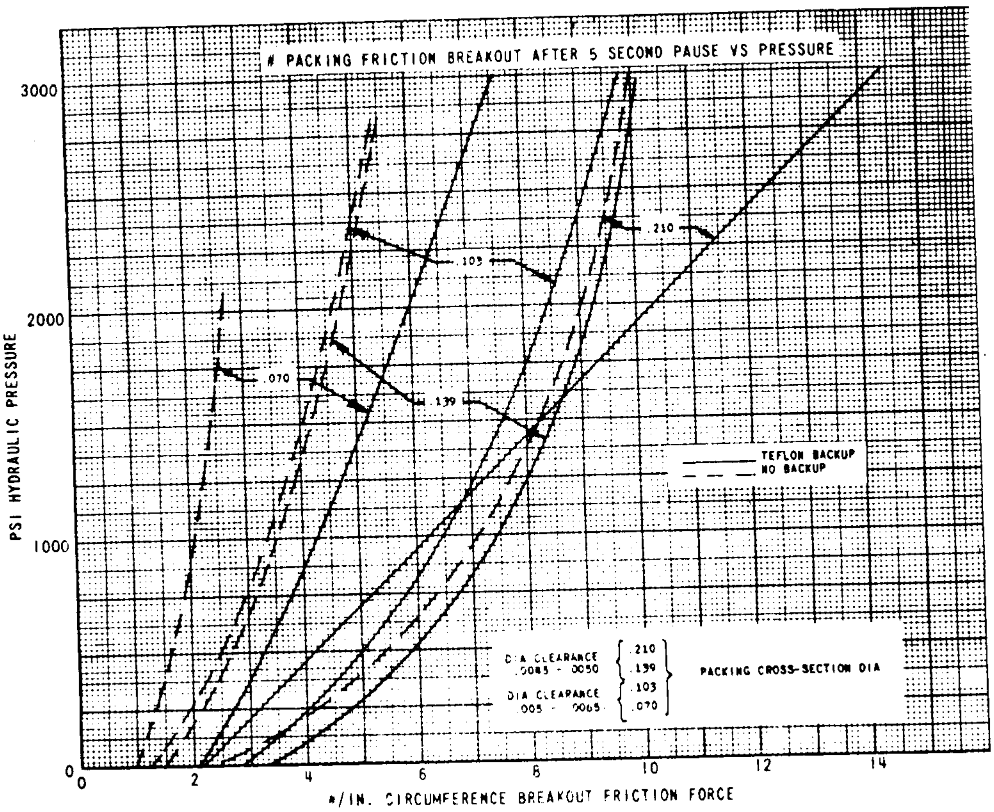
- Installed squeeze
- Operating temperature
- Standby time between actuations
- Surface finish
- Actuation velocity
- Direction of actuation relative to direction of pressure on seal
- Presence of ice crystals on sliding surfaces exposed to atmosphere (low temperature operation)
- Use of backups
- Fluid film (or absence of it) on sliding surfaces exposed to atmosphere.
- Contamination
- Extruded or feather edges on back-ups which can wedge into the sealing clearance
- Seal compound
- Prior cycling (break-in effect).

101.81 O-RING FRICTION DATA The following curves can be used for rapid, approximate solution of O-ring (NAS1611 or MS28775) break-out friction with or without teflon back-up rings.

Curves show breakout friction characteristics after 5-second pause as related to pressure for maximum Douglas standard squeeze and nominal RMS 12 surface roughness, for each of four O-ring cross-sectional sizes.

For delay periods less than 5 seconds such as continuous small amplitude corrective control motions in flight, packing friction will be somewhat less.

101.81 (Continued)



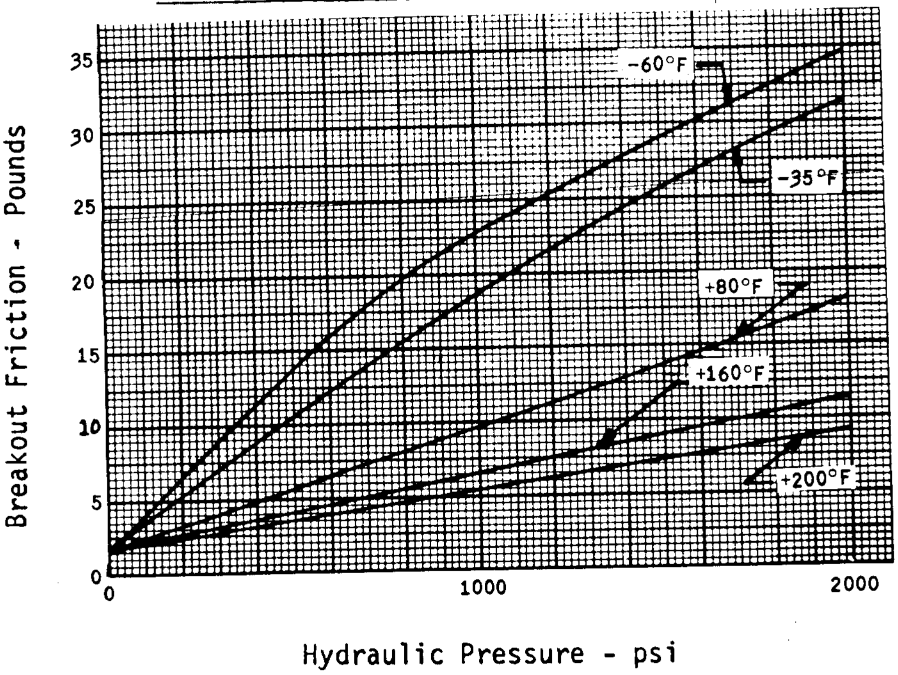
For longer delay periods the packing has a tendency to squeeze the fluid out from between the sealing surfaces resulting in a considerable increase in breakout friction. For an 8-hour standby the breakout friction may increase by at least 200 percent.

Little good data is available on low temperature friction. However, it is known that once a packing has been "broken out" low temperature running friction is not appreciably greater than at room temperature.

The test group engineer shall be consulted for additional reference to test data for those instances where packing friction has a significant effect on component function or design efficiency.

101.82 TEFLON PISTON RING FRICTION EXAMPLES Data is for the Douglas standard teflon piston rings described in Section 101.62 of the Hydraulics Manual. The data is presented for comparison purposes only, and does not include test scatter.

101.821 FRICTION CHANGE WITH TEMPERATURE (One Seal Assembly)

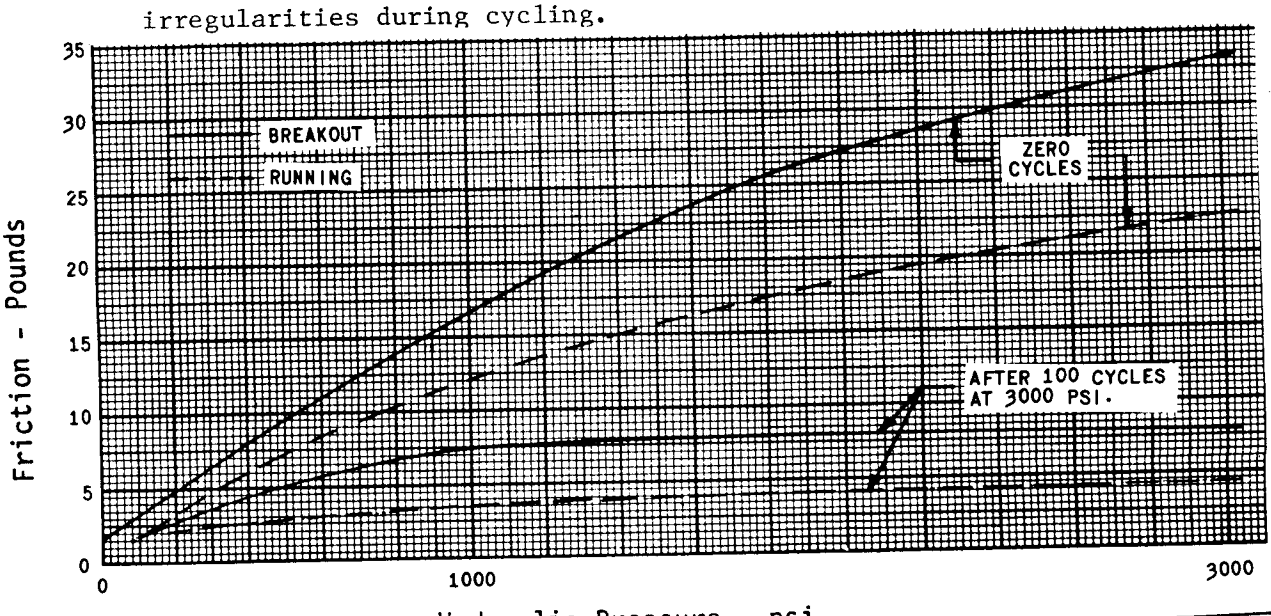


2 Inch Cylinder Bore

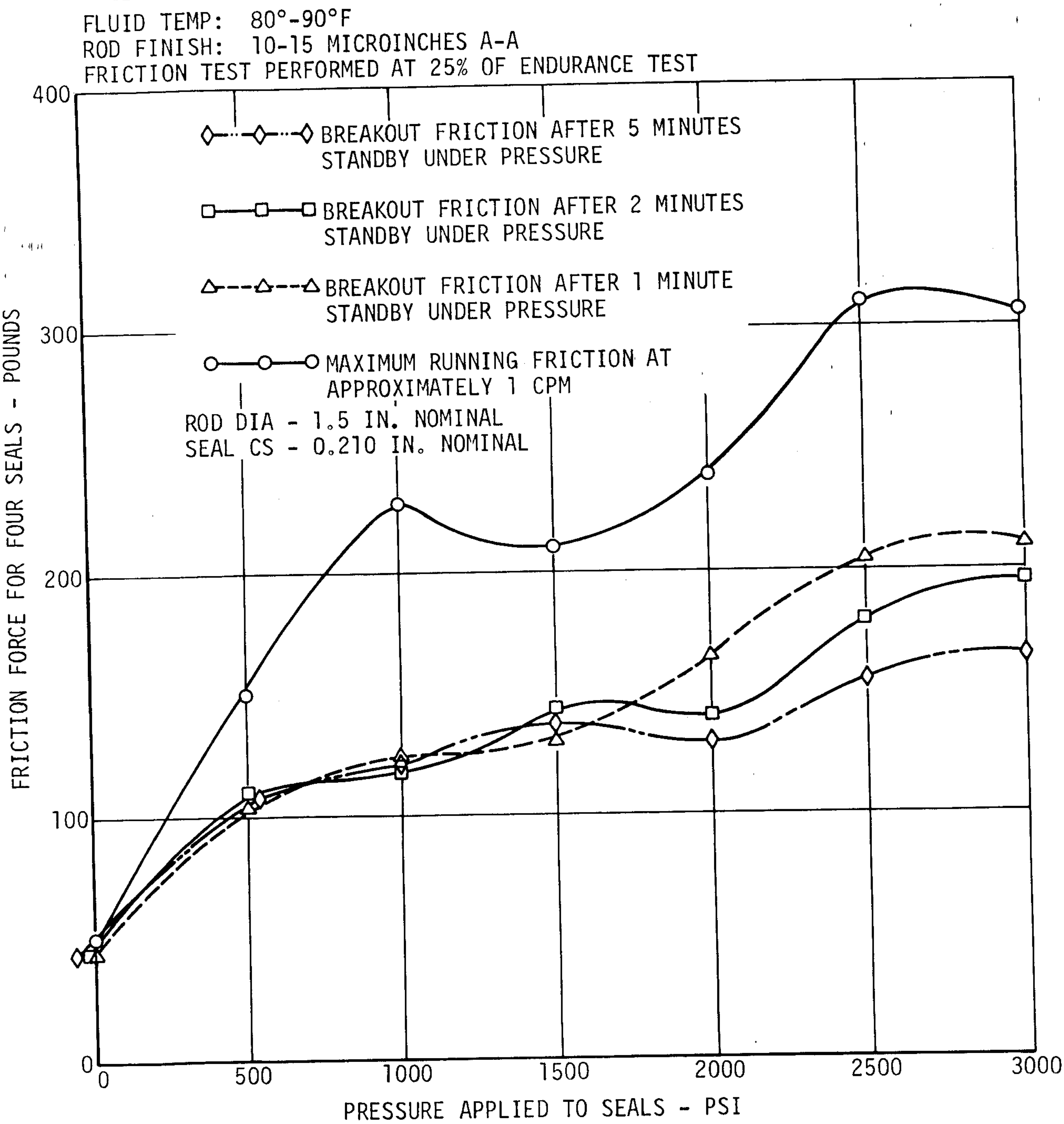
Fluid: MIL-H-5606 Mineral Oil

101.822 FRICTION CHANGE WITH CYCLING (One Seal Assembly) The tests were conducted with 1-5/8 inch diameter teflon piston rings (S-4444978-26). The barrel was steel with 15 microinches roughness. The fluid was MIL-H-5606 mineral oil at 70°F.

The breakout was not much affected by the standby time. The change in friction is assumed due to the teflon impregnating surface irregularities during cycling.



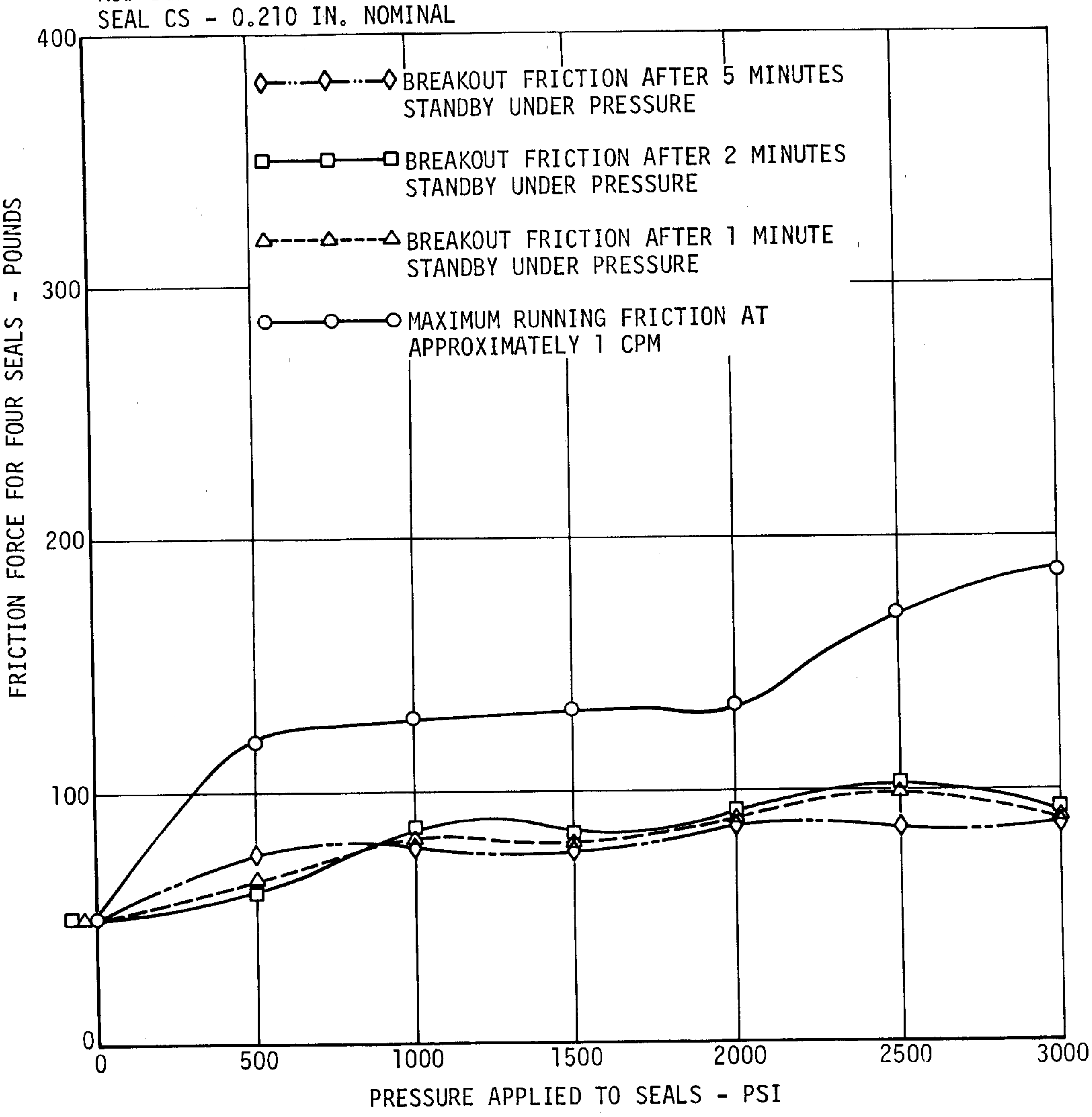
101.83 FRICTION FORCE VERSUS SEAL PRESSURE - TWO TEFLON AND TWO TURCON SLIPPERS (FROM HTR NO. 276)



101.83 FRICTION FORCE VERSUS SEAL PRESSURE - TWO TEFLON AND TWO TURCON SLIPPERS (FROM HTR NO. 276)

101.83 (Continued)

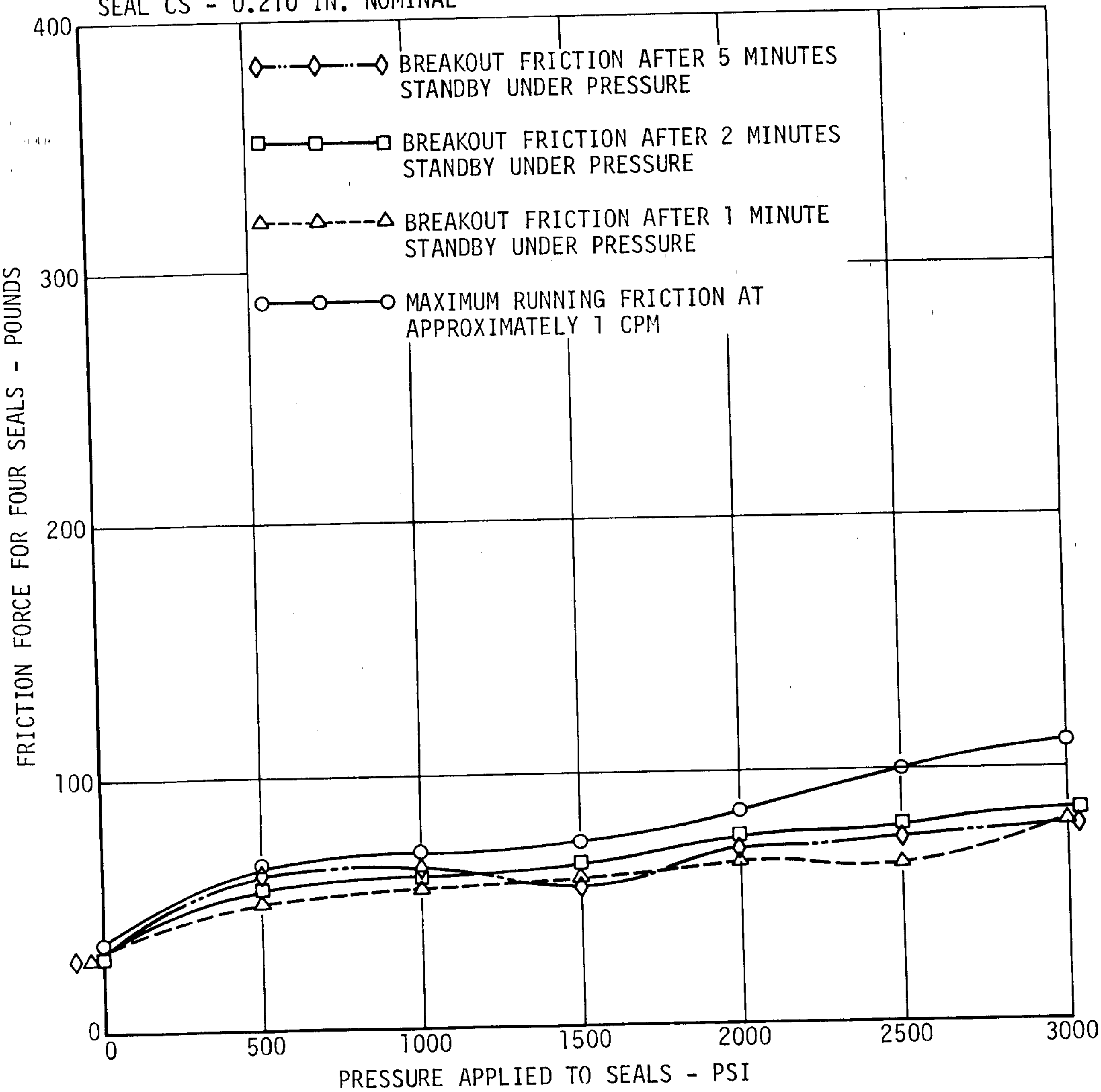
FLUID TEMP: 160°F
ROD FINISH: 10-15 MICROINCHES AA
FRICTION TEST PERFORMED AT 25% OF ENDURANCE TEST
ROD DIA - 1.5 IN. NOMINAL
SEAL CS - 0.210 IN. NOMINAL



101.83 FRICTION FORCE VERSUS SEAL PRESSURE - TWO TEFLON AND TWO TURCON SLIPPERS (CONT)

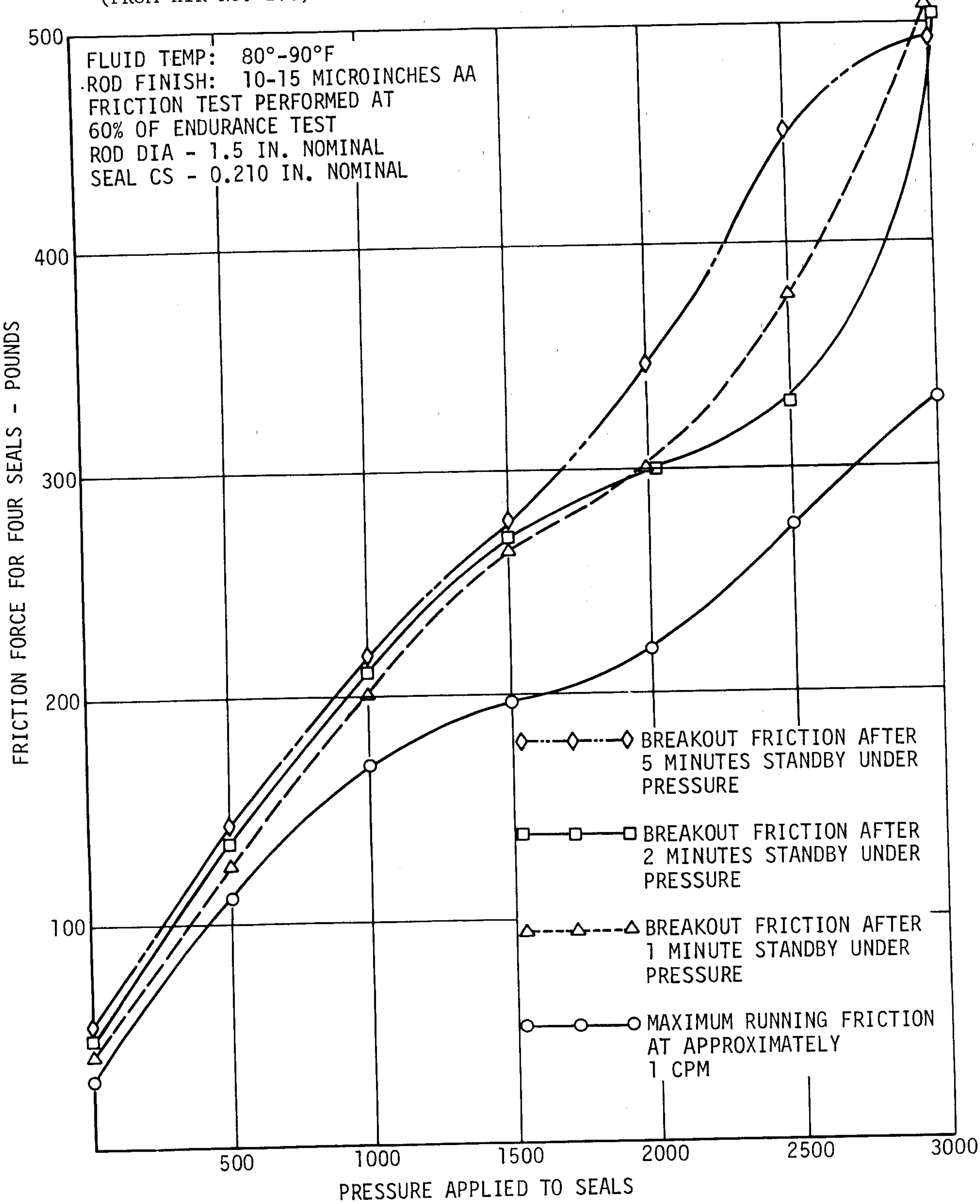
101.83 (Continued)

FLUID TEMP: 225°F
ROD FINISH: 10-15 MICROINCHES A-A
FRICTION TEST PERFORMED AT 25% OF ENDURANCE TEST
ROD DIA - 1.5 IN. NOMINAL
SEAL CS - 0.210 IN. NOMINAL



101.83 FRICTION FORCE VERSUS SEAL PRESSURE - TWO TEFLON AND TWO TURCON SLIPPERS (CONT)

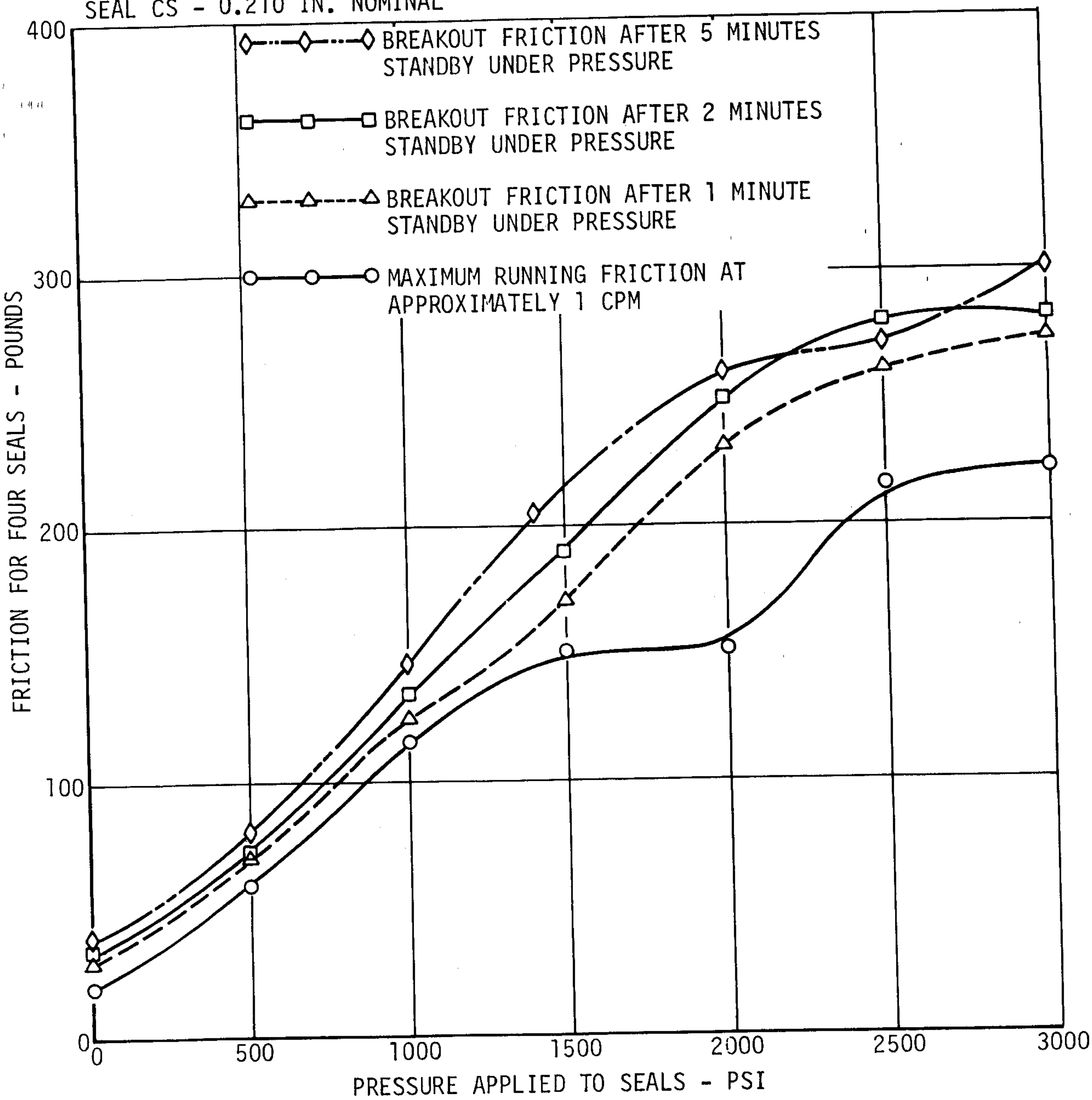
101.84 FRICTION FORCE VERSUS SEAL PRESSURE - FOUR GREEN TWEED T-SEALS
(FROM HTR NO. 276)



101.84 FRICTION FORCE VERSUS SEAL PRESSURE - FOUR GREEN TWEED T-SEALS (FROM HTR NO. 276)

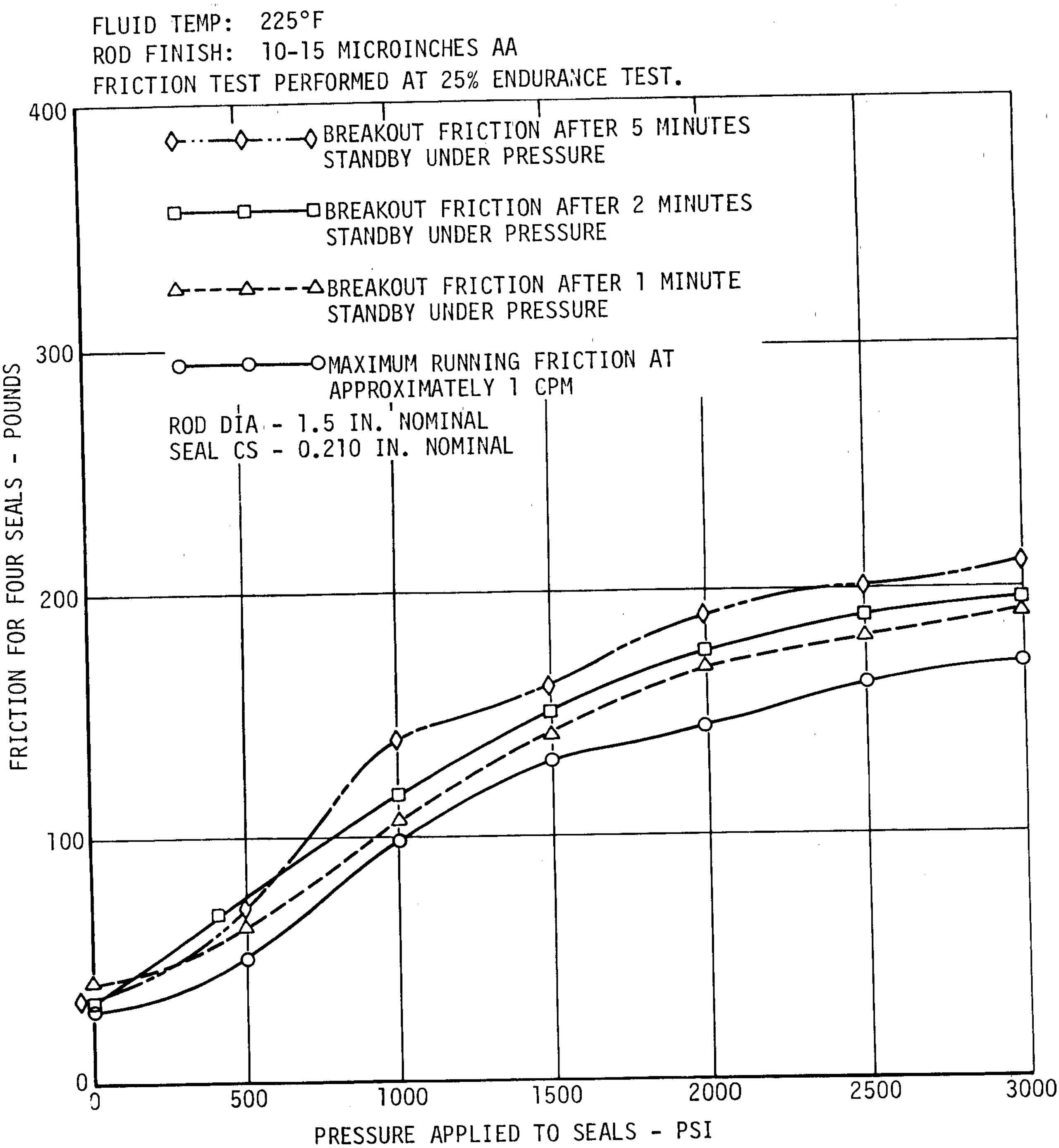
101.84 (Continued)

FLUID TEMP: 160°F
ROD FINISH: 10-15 MICROINCHES AA
FRICTION TEST PERFORMED AT 25% ENDURANCE TEST
ROD DIA - 1.5 IN. NOMINAL
SEAL CS - 0.210 IN. NOMINAL



101.84 FRICTION FORCE VERSUS SEAL PRESSURE - FOUR GREEN TWEED T-SEALS (CONT)

101.84 (Continued)



101.84 FRICTION FORCE VERSUS SEAL PRESSURE - FOUR GREEN - TWEED T-SEALS (CONT)

101.9 SEAL LEAKAGE O-ring leakage in dynamic applications can be characterized as generally good, but certainly not perfect. The amount that O-rings leak during actuation is influenced by the following:

- Installed squeeze
- Stroke length
- Pressure relative to direction of motion through the seal
- Surface finish
- Temperature
- Stroke velocity

Douglas specification 7912042, which sets forth the qualification requirements for EPR O-ring seals, permits 8cc per 1000 cycles per seal. The cycling conditions are as follows:

- 50,000 stroke cycles (6 in. stroke)
- 225°F fluid temperature
- 3000 psi seal pressure (momentarily reduced to zero at midpoint of each stroke)
- 24 cpm
- 0.004 diametral clearance.

The standard Douglas production inspection test procedure for actuating cylinders specify the following:

PRODUCTION INSPECTION TEST PROCEDURE

SUB-CONTRACTOR OR MANUFACTURER TO TEST EACH UNIT TO THIS PROCEDURE

USE ____ FLUID FOR TEST

TEST NO.	TEST FOR	APPLY HYDR. PRESSURE TO PORT	INSTRUCTIONS AND REQUIRED FUNCTION
1	PROOF STRENGTH & PACKING LEAKAGE	ROD	WITH PISTON RETRACTED AND HEAD PORT UP AND FILLED WITH FLUID, APPLY 25 THEN 4500 PSI. AFTER PRESSURE IS REACHED, NO NOTICEABLE RISE OF FLUID IN HEAD PORT OR EXTERNAL LEAKAGE ALLOWED IN 2 MIN. AT EITHER PRESSURE.
2		HEAD	WITH PISTON EXTENDED REPEAT TEST 1, EXCEPT PORTS TO BE REVERSED.
3	PACKING CYCLING LEAKAGE	BOTH ALTERNATELY	CYCLE THROUGH FULL STROKES, BUILD UP PRESSURE TO 3000 PSI AT END OF EACH STROKE. LEAKAGE AT ROD GLAND NOT TO EXCEED 1 DROP PER 5 CYCLES AFTER LEAKAGE RATE BECOMES CONSTANT. EXTERNAL LEAKAGE ELSEWHERE NOT TO EXCEED A TRACE.
4	PACKING FRICTION AND BINDING	BOTH ALTERNATELY	EXTEND & RETRACT. AFTER INITIAL MOVEMENT PRESSURE REQUIRED THROUGH FULL STROKE NOT TO EXCEED ____ PSI AT HEAD PORT OR ____ PSI AT ROD PORT.

TEST PROCEDURE CHECK ____

102. THREADS

102.1 STANDARD HYDRAULIC THREAD SERIES

SIZE	THREADS PER INCH	SYMBOL
• 10	32	NF-3
• 1/4	28	UNF-3
• 5/16	£ 24	UNF-3
• 3/8	£ 24	UNF-3
• 7/16	£ 20	UNF-3
• 1/2	£ 20	UNF-3
• 9/16	£ 18	UNF-3
• 5/8	18	UNF-3
11/16	24	NEF-3
• 3/4	£ 16	UNF-3
3/4	20	UNEF-3
13/16	20	UNEF-3
7/8	20	UNEF-3
• 7/8	£ 14	UNF-3
15/16	20	UNEF-3
• 15/16	18	NS-3
1	20	UNEF-3
• 1	14	NF-3
• 1 1/16	18	NEF-3
• 1 1/16	14	NS-3
1 1/16	£ 12	UN-3
• 1 1/8	18	NEF-3
• 1 3/16	18	NEF-3
• 1 1/4	18	NEF-3
• 1 5/16	£ 18	NEF-3
1 5/16	12	UN-3
• 1 3/8	18	NEF-3
• 1 7/16	18	NEF-3
• 1 1/2	18	NEF-3
1 9/16	18	NEF-3

SIZE	THREADS PER INCH	SYMBOL
• 1 9/16	16	UNS-3
1 5/8	18	NEF-3
• 1 5/8	16	UNS-3
1 5/8	£ 12	N-3
1 11/16	18	NEF-3
• 1 11/16	16	UNS-3
1 3/4	18	NS-3
• 1 3/4	16	UN-3
1 13/16	18	NS-3
• 1 13/16	16	N-3
1 7/8	18	NS-3
• 1 7/8	16	N-3
1 7/8	£ 12	N-3
1 15/16	18	NS-3
1 15/16	16	N-3
2	18	NS-3
• 2	16	UN-3
• 2 1/16	16	N-3
2 1/8	16	N-3
2 3/16	16	N-3
2 1/4	16	UN-3
2 1/4	£ 12	UN-3
• 2 5/16	16	N-3
2 3/8	16	N-3
2 7/16	16	N-3
• 2 1/2	16	UN-3
2 1/2	£ 12	UN-3
2 5/8	16	N-3
• 2 3/4	16	UN-3

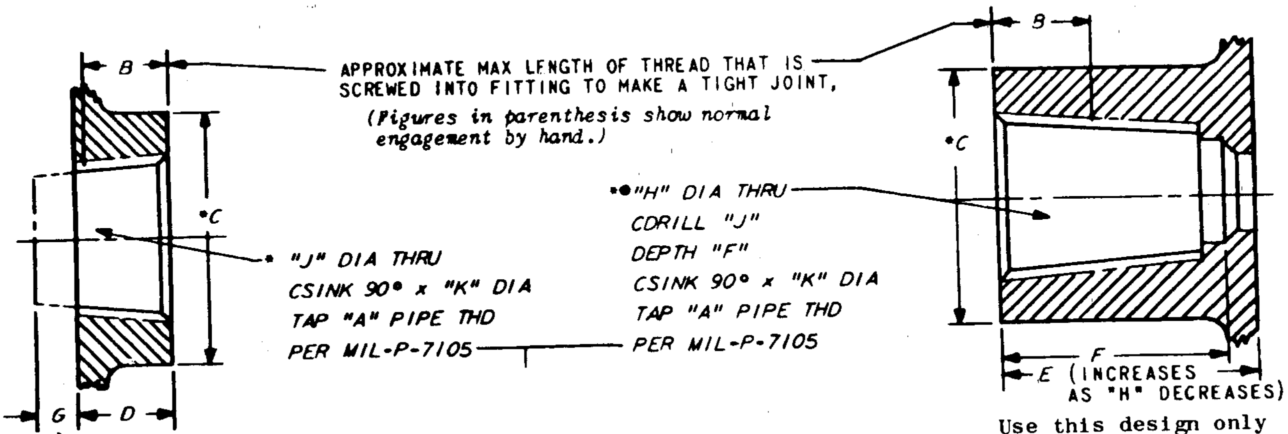
SIZE	THREADS PER INCH	SYMBOL
2 7/8	16	N-3
3	16	UN-3
• 3 1/8	16	N-3
3 1/4	16	UN-3
• 3 3/8	16	N-3
3 1/2	16	UN-3
3 5/8	16	N-3
• 3 3/4	16	UN-3
3 7/8	16	N-3
4	16	UN-3
• 4 1/8	12	UNS-3
• 4 1/4	12	UN-3
• 4 3/8	12	UNS-3
• 4 1/2	12	UN-3
• 4 5/8	12	UNS-3
• 4 3/4	12	UN-3
• 4 7/8	12	UNS-3
• 5	12	UN-3
• 5 1/4	12	UN-3
• 5 1/2	12	UN-3
• 5 3/4	12	UN-3
• 6	12	UN-3
• 6 1/4	8	UNS-3
• 6 1/2	8	UNS-3
• 6 3/4	8	UNS-3
• 7	8	UNS-3
• 7 1/4	8	UNS-3
• 7 1/2	8	UNS-3
• 7 3/4	8	UNS-3
• 8	8	UNS-3

NOTES:

1. Threads marked thus * are approved for general use. Others may be used if an approved thread is not available in required size, or in special design cases.
2. Threads marked thus £ should be used for tube fittings or bosses where there is any possibility of a tube fitting being used.
3. Threads bearing an S in the symbol are not listed in specification MIL-S-7742.
4. Threads lacking an S in the symbol are listed in specification MIL-S-7742.
5. See Drafting Manual, Section 69., for drafting conventions.

102.2 INTERNAL PIPE THREADS

△ NOTE: The following data is from AND10053, which is inactive for new design for military aircraft, except in oxygen systems, after 18 January 1955.



Clearance for standard tap.

Use this design only when port opening is required.

DIMENSIONS IN INCHES – STANDARD TOLERANCES WHERE APPLICABLE

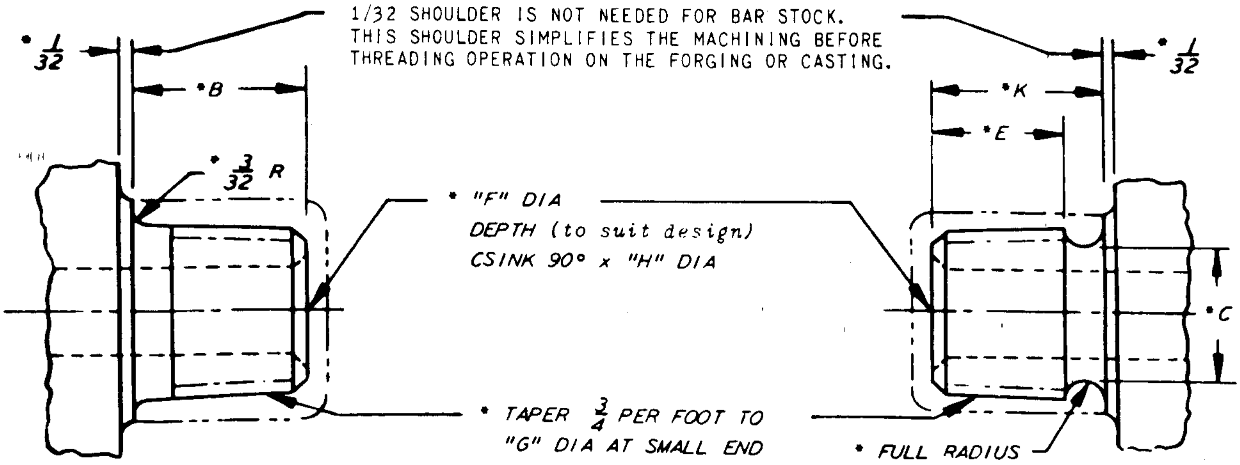
A NOM PIPE SIZE	NO. OF THREADS PER INCH	B ••	C MIN FORG AND BAR	C MIN CAST	D MIN	E MIN	E# MIN	F MIN	F# MIN	G MIN	J	K ±1/64 DIA	AVAILABLE TOOLS SM	ES	LB
1/16	27	(5/32) 17/64	1/2	19/32	19/64	43/64	27/64	39/64	23/64	15/64	.226 .234	5/16	X		X
1/8	27	(3/16) 9/32	9/16	11/16	5/16	11/16	7/16	5/8	3/8	1/4	.316 .327	13/32	X	X	X
1/4	18	(7/32) 13/32	23/32	29/32	7/16	7/8	17/32	13/16	15/32	9/32	.413 .421	9/16	X	X	X
3/8	18	(1/4) 13/32	7/8	1 1/32	15/32	29/32	19/32	27/32	17/32	5/16	.546 .556	11/16	X	X	X
1/2	14	(5/16) 17/32	1 3/32	1 1/4	19/32	1 1/8	3/4	1 1/16	11/16	3/8	.671 .686	7/8	X	X	X
3/4	14	(11/32) 17/32	1 11/32	1 1/2	5/8	1 1/8	13/16	1 1/16	3/4	13/32	.875 .896	1 1/16	X	X	X
1	11 1/2	(13/32) 21/32	1 5/8	1 13/16	3/4	1 3/8	29/32	1 5/16	27/32	15/32	1.109 1.126	1 5/16	X	X	X
1 1/4	11 1/2	(7/16) 11/16	2	2 1/4	25/32	1 3/8	29/32	1 5/16	27/32	15/32	1.453 1.470	1 43/64	X	X	X
1 1/2	11 1/2	(7/16) 11/16	2 9/32	2 9/16	13/16	1 13/32	15/16	1 11/32	7/8	15/32	1.687 1.709	1 29/32	X	X	X
2	11 1/2	(15/32) 23/32	2 27/32	3 5/32	13/16	1 13/32	31/32	1 11/32	29/32	15/32	2.171 2.182	2 3/8	X		
2 1/2	8	(11/16) 1 1/16	3 13/32	3 25/32	1 3/16	1 15/16	1 3/8	1 7/8	1 5/16	17/32	2.579 2.595	2 29/32			
3	8	(3/4) 1 5/32	4 1/8	4 19/32	1 9/32	2	1 7/16	1 15/16	1 3/8	17/32	3.203 3.216	3 17/32			

NOTES:

1. Internal pipe threads should be indicated as shown in the illustration above.
* 2. Notes and dimensions marked * are to appear on production drawings.
3. Dimensions "E" and "F" will permit full length use of standard taps.
4. Dimensions "E#" and "F#" allow for grinding off taps maximum amount.
   a. Designs should not require taps to be ground off unless absolutely necessary.
   b. Maximum amount taps can be ground off is that which will leave 3 1/2 threads extending through the standard ring gage.
● 5. Diameter "H" should be drilled to suit design, at least one size smaller than tap drill "J".
6. If a tapped boss is (or becomes through welding, etc.) not removable from a large or expensive complicated part, "C" and "D" for next larger pipe size should be used.
7. Dimension "C" should be increased 1/8 on non-ferrous parts and cast iron when subjected to a working pressure of 100 psi or over.
** 8. External pipe threads may exceed the approximate B max dimension due to overtightening to overcome leakage. In cases of possible interference, D min should exceed B max by 3/32.

102.3 EXTERNAL PIPE THREADS

△ NOTE: The following data is from AND10052, which is inactive for new design for military aircraft, except in oxygen systems after 18 January 1955.



RESTRICTED DESIGN
See design supervision

PREFERRED DESIGN
Taken from AND10052

DIMENSIONS IN INCHES - STANDARD TOLERANCES WHERE APPLICABLE												
A NOMINAL PIPE SIZE	NO. OF THREADS PER IN.	B MIN DIM	C	E	F MAX DIM	G	H	J	K MIN DIM	AVAILABLE TOOLS SM	ES	LB
1/16	27	1/2	.260	11/32	.125	.296 .303	.188	.031	25/64	X		X
1/8	27	1/2	.385	11/32	.203	.388 .396	.250	.031	25/64	X	X	X
1/4	18	11/16	.478	1/2	.296	.516 .525	.344	.047	19/32	X	X	X
3/8	18	11/16	.635	1/2	.421	.650 .660	.469	.047	39/64	X	X	X
1/2	14	7/8	.760	5/8	.546	.809 .820	.625	.062	25/32	X	X	X
3/4	14	7/8	.978	21/32	.734	1.018 1.030	.812	.062	51/64	X	X	X
1	11 1/2	1 1/16	1.228	13/16	.953	1.276 1.289	1.031	.078	63/64	X	X	X
1 1/4	11 1/2	1 3/32	1.572	13/16	1.265	1.620 1.633	1.344	.078	1	X	X	X
1 1/2	11 1/2	1 1/8	1.822	27/32	1.500	1.859 1.871	1.594	.078	1 1/32	X	X	X
2	11 1/2	1 1/8	2.291	7/8	1.937	2.332 2.344	2.031	.078	1 1/16	X		
2 1/2	8	1 5/8	2.728	1 5/16	2.312	2.812 2.828	2.406	.109	1 9/16			
3	8	1 11/16	3.353	1 3/8	2.906	3.433 3.449	3.000	.109	1 5/8			

NOTES:
1. Notes and dimensions marked * are to appear on production drawings.
2. Design approval may be obtained to increase or decrease 3/32 R providing "B" dimension is changed accordingly.

103. BOSS DESIGN

103.1 BOSS CALLOUT FOR HYDRAULIC UNITS Boss design standard MS33649 replaces AND10050, but the MS33649 thread and tap drill depths are unnecessarily deep for hydraulic parts in the commonly used sizes, causing a possible space (and weight) penalty averaging about 1/8 inch. (The deeper threads accommodate power plant installations of "universal" positionable fittings assembled by special methods.)

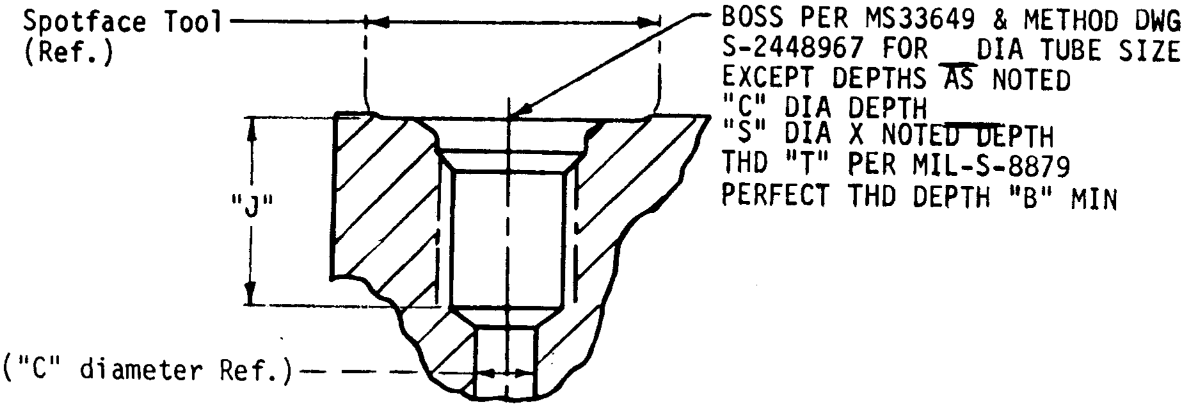
a. The following boss call-out must be used on hydraulic-unit production drawings, until a more suitable government boss standard becomes available.

b. S-2448967 controls spotface dimensions, and limits eccentricity of seal surfaces.

c. "C" is the through diameter, and may be more or less than the dimension listed in MS33649. Use of this "C" diameter is not mandatory. Cavity design must prevent flow blocking or internal interference caused by missassembly, per Section 20 requirements.

d. Depths "B" and "J" provide clearance for "straight" non-positionable fittings (MS33514 and MS33656 ends) at adverse tolerances, and are satisfactory for universal fittings in hydraulic units based on experience. The listed "B" and "J" generally follow AND10050 approximately, not MS33649. If required by design, "B" and "J" may be increased; minimum "J" should exceed minimum "B" by more than 2 threads, to provide runout.

e. The MIL-S-8879 threads of MS33649 replace the MIL-S-7742 threads of AND10050, to accommodate possible future rolled-thread fittings. The only significant difference in these two internal thread forms is 0.004 to 0.009 inch increase in the tap drill diameter for MIL-S-8879.



103.1 (Continued)

Tube OD Size	Fitg Dash No.	"B"	"J" ±.03	"S" (tap drill)	"T"	Ref. Data Spotface Dia ±.015	Ref. Data Spotface Rad ±.015
1/8	2	.380	.53	.2719/.2799	.3125-24 UNJF-3B	.828	.062
3/16	3	.425	.55	.3344/.3418	.3750-24 UNJF-3B	.906	.062
1/4	4	.456	.62	.3888/.3970	.4375-20 UNJF-3B	.984	.062
5/16	5	.456	.62	.4513/.4591	.5000-20 UNJF-3B	1.062	.062
3/8	6	.470	.63	.5084/.5166	.5625-18 UNJF-3B	1.125	.062
1/2	8	.562	.75	.6892/.6977	.7500-16 UNJF-3B	1.406	.093
5/8	10	.625	.81	.8055/.8152	.8750-14 UNJF-3B	1.562	.093
3/4	12	.688	.94	.9814/.9914	1.0625-12 UNJ-3B	1.843	.093
1	16	.688	.97	1.2314/1.2414	1.3125-12 UNJ-3B	2.130	.093
1 1/4	20	.688	1.01	1.5439/1.5539	1.6250-12 UNJ-3B	2.490	.093
1 1/2	24	.750	1.15	1.7939/1.8039	1.8750-12 UNJ-3B	2.780	.093
1 3/4	28	.812	1.27	2.1689/2.1789	2.2500-12 UNJ-3B	3.230	.093
2	32	.907	1.40	2.4189/2.4289	2.5000-12 UNJ-3B	3.700	.093

103.2 REFERENCE INFORMATION The following table compares minimum depths "B" and "J" of MS33649 and AND10050 with those listed in Section 103.1. Also the minimum thread runout from "B" to "J" and the minimum thread clearance for MS33514 flareless and MS33656 flare fitting ends are listed for Section 103.1 dimensions, as is the minimum amount depth "J" exceeds the length of MS33656.

Fitg Dash No.	MS33649 Boss Depth Min "B"	MS33649 Boss Depth Min "J"	AND10050 Boss Depth Min "B"	AND10050 Boss Depth Min "J"	Section 103.1 Min "B"	Section 103.1 Min "J"	Min Runout Min J less Min B Inch	Min Runout Min J less Min B Thds	Min Thread clearance Flare-less Fitg	Min Thread clearance Flare Fitg	Min J less Flared Fitg Length
2	.482	.577	.375	.500	.380	.50	.120	2.8	.001	.107	.037
3	.538	.583	.406	.500	.425	.52	.095	2.3	.000	.119	.026
4	.568	.656	.438	.594	.456	.59	.134	2.7	.000	.097	.025
5	.568	.656	.438	.594	.456	.59	.134	2.7	.001	.097	.025
6	.598	.709	.469	.594	.470	.60	.130	2.3	.000	.112	.029
8	.714	.834	.562	.718	.562	.72	.158	2.5	.000	.161	.048
10	.802	.930	.625	.781	.625	.78	.155	2.2	.000	.137	.007
12	.877	1.064	.688	.906	.688	.91	.222	2.7	.003	.146	.031
16	.877	1.064	.688	.938	.688	.94	.252	3.0	.003	.099	.014
20	.877	1.116	.688	.984	.688	.98	.292	3.5	.004	.103	.007
24	.877	1.127	.750	1.125	.750	1.12	.370	4.4	.066	.052	.022
28	.877	1.243	.812	1.250	.812	1.24	.428	5.1	--	.062	.017
32	.907	1.368	.938	1.375	.907	1.37	.463	5.6	.223	.042	.022

103.3 SPECIFICATION DRAWINGS To avoid communication problems, generally call for bosses per MS33649. But where space is critical, specify "AND10050 except threads per MIL-S-8879". Although some thread interference is possible at tolerance extremes, experience indicates AND10050 thread depths are satisfactory for hydraulic units.

103.4 RAISED BOSS OD The following outside diameters for raised bosses shall be specified on applicable drawings.

TUBE SIZE	BOSS OD ± .015					
	PRESSURE PORTS (3000 PSI)			RETURN PORTS (1000 PSI)		
	356-T6 or 6061-T6	2024 BAR	STEEL 125,000 H.T.	356-T6 6061-T6	2024 BAR	STEEL 125,000 H.T.
-2	.563	←			→	.563
-3	.625	←			→	.625
-4	.688	←			→	.688
-5	.750	←			→	.750
-6	.875	.813	←		→	.813
-8	1.188	1.062	←		→	1.062
-10	1.375	1.250	1.188	←	→	1.188
-12	1.688	1.500	1.438	←	→	1.438
-16	2.125	1.938	1.688	←	→	1.688
-20	2.813	2.500	2.000	←	→	2.000
-24	3.125	2.813	2.250	←	→	2.250

On expensive aluminum alloy parts, raised boss OD's shall be increased to the next larger size, to permit salvage rework, subject to design supervision approval.

104. NAMEPLATES AND INSTRUCTION PLATES

104.1 GENERAL Conform with Drafting Manual Section 73.

104.11 NAMEPLATES The DPS 1.07 Type 2 and Type 4 adhesive attachments are Skydrol resistant. Type 4 includes a strap, and is preferred over Type 2 which does not, for units with external cylindrical surfaces. For Type 2 attachments, also use attachment screws when practicable, since the adhesive bond can fail under adverse conditions.

GENERAL NOTES

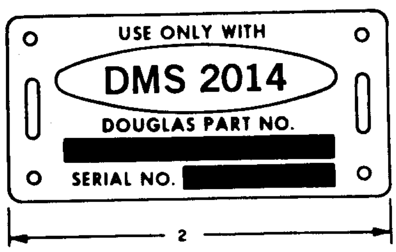
UNLESS OTHERWISE SPECIFIED

1. FABRICATE PER DPS 9.60 (DAC) OR DMS 1674 CLASS 2 (VENDOR)
2. LETTERS AND FIGURES TO BE DULL ALUMINUM ON 6606 GLOSS VIOLET BACKGROUND
3. ENGINEERING TO FURNISH PHOTO NEGATIVE, NO ART WORK TO BE DONE BY VENDOR
4. IDENTIFY PER DPS 3.02

0.020 THICK X 1 X 2	AL SHEET LABEL	DMS 1674 CLASS 2
STOCK SIZE	MATERIAL DESCRIPTION	MATERIAL SPECIFICATION

MATERIALS

104.12 CYLINDER NAMEPLATES



STANDARD
TYPE
S-2931817

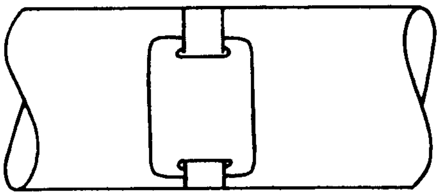
104.12 CYLINDER NAMEPLATES (Continued)

Generally, actuators should use S-2931817 plates, but special cylinders and/or special instructions must be adequately described by special plates.

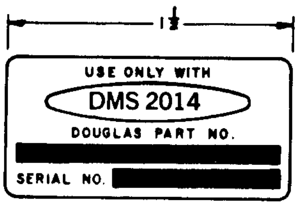
For attachment screws, consider use

SAMPLE INSTALLATION:

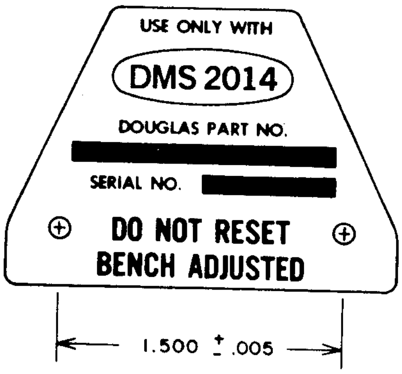
S-2931817 NAMEPLATE
ATTACH PER DPS 1.07-47 TYPE 4
MANUFACTURER TO STAMP IN
APPROPRIATE PART NO. AND
SERIAL NO. BEFORE ASSEMBLY.



104.13 VALVE NAMEPLATES The nameplate may be used without attachment screws, but use of a strap or screws in addition to the adhesive is preferred. For attachment with screws, consider use of S-2931817 or the smaller S-2778291, reworked to a special part by addition of screw holes. A special nameplate may be used when required by the housing shape, or to describe special instructions.



STANDARD TYPE
S-2778291



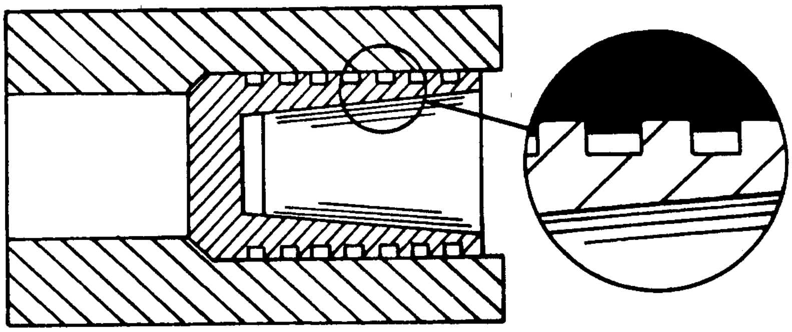
SPECIAL TYPE
2917326

ON ASSEMBLY DRAWING: ATTACH PER DPS 1.07-47 TYPE 2.
MANUFACTURER TO STAMP IN APPROPRIATE
PART NUMBER AND SERIAL NUMBER BEFORE
ASSEMBLY

105.0 LEE PLUGS

Lee plugs are a proprietary item made by The Lee Co., 2 Pettipaug Road, Westbrook, Conn. The Vendor Files Group has catalogs filed under "Lee Plugs". They are useful in sealing drilled passageways in hydraulic and pneumatic units.

A Lee plug consists of a cylindrical plug with a tapered reamed hole part way through its center, and a tapered pin. The plug is slipped into a counterbored hole, and the pin is then driven into the plug causing the plug to expand and effect a seal.



Lee plugs are made of aluminum alloy or corrosion resistant steel for use in aluminum or steel housings respectively.

They are made in a long and a short series, in standard diameters, and in oversize diameters for rework or salvage. Consult the Lee Co. catalog for information on oversize plugs.

The Lee plug part number consists of six digits, the significance of which is explained below.

PART NUMBER CODE:

X X X Y Z Z
- THE Z DIGITS SPECIFY MATERIAL.
- THE Y DIGIT SPECIFIES LENGTH (LONG OR SHORT).
- THE X DIGITS SPECIFY THE DIAMETER

EXAMPLE:

1 5 6 1 0 1
- 2024T4 ALUMINUM.
- LONG.
- 156 IN. NOMINAL O.D. OF LEE PLUG

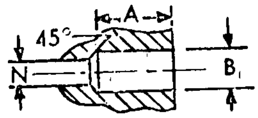
Y = 1 FOR LONG
Y = 0 FOR SHORT
ZZ = 01 FOR 2024T4 ALUMINUM
ZZ = 02 FOR 416 ST. STEEL
ZZ = 03 FOR 303 ST. STEEL (THIS MATERIAL IS NOT TO BE USED.)

105.1 <u>FITS AND PROOF PRESSURES</u> Three classes of fits are provided for each Lee plug. Proof pressures for each class of fit of each plug are given in the table below.

	ALUMINUM				STAINLESS STEEL			
		POUNDS PER SQ IN. PROOF PRESSURE X1000				POUNDS PER SQ IN. PROOF PRESSURE X1000		
	LEE PLUG NUMBER	CLASS 1	CLASS 2	CLASS 3	LEE PLUG NUMBER	CLASS 1	CLASS 2	CLASS 3
LONG	156101	10	13	16	156102	18	24	28
	187101	10	13	15	187102	18	24	28
	218101	9	12	15	218102	16	22	26
	250101	9	12	14	250102	16	22	25
	281101	9	12	14	281102	18	20	24
	343101	8	11	14	343102	18	20	22
	406101	8	10	13	406102	16	18	21
	468101	7	10	12	468102	16	18	20
	531101	5	8	12	531102	14	16	18
	656101	4	7	10	656102	10	14	16
SHORT	093001	10	13	16	093002	14	16	20
	125001	8	10	13	125002	14	16	20
	156001	8	9	12	156002	8	9	10
	187001	8	9	10	187002	12	14	16
	218001	8	9	10	218002	10	12	14
	250001	8	9	10	250002	12	14	16
	281001	8	9	10	281002	12	14	16
	343001	6	8	10	343002	10	12	14
	406001	6	8	10	406002	10	12	14
	468001	6	8	10	468002	10	12	14
	531001	6	8	10	531002	10	12	14

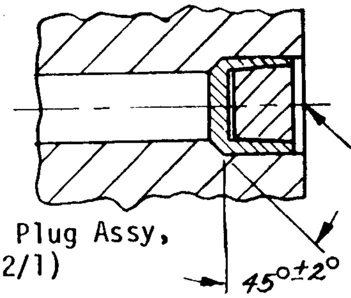
These "proof pressures" are advertized values for installation in housings of the same material as the plug. Data is unavailable on performance under adverse conditions of material, manufacturing, installation and environment. The designer should use conservatism, and the "proof pressure" should never be less than the design burst pressure.

105.2 HOUSING DIMENSIONS Cavity dimensions for the Lee plug are tabulated below:



LEE PLUG SIZE	MAX. DRILL HOLE "N"	MIN. HOLE DIA. "B"	MAXIMUM HOLE DIA. "B"			DEPTH OF REAM MINIMUM "A"	
			CLASS 1 MAX.	CLASS 2 MAX.	CLASS 3 MAX.	LONG	SHORT
093YZZ	.062	.0937	.0962	.0952	.0942	–	.097
125YZZ	.093	.1250	.1275	.1265	.1255	–	.125
156YZZ	.125	.1562	.1587	.1577	.1567	.250	.130
187YZZ	.156	.1875	.1900	.1890	.1880	.253	.150
218YZZ	.187	.2187	.2212	.2202	.2192	.312	.180
250YZZ	.210*	.2500	.2525	.2515	.2505	.352	.212
281YZZ	.240*	.2812	.2837	.2827	.2817	.375	.250
343YZZ	.280*	.3437	.3462	.3452	.3442	.437	.295
406YZZ	.340*	.4062	.4087	.4077	.4067	.500	.325
468YZZ	.400*	.4687	.4712	.4702	.4692	.562	.357
531YZZ	.460*	.5312	.5337	.5327	.5317	.625	.486**
656YZZ	.580*	.6562	.6587	.6577	.6567	.677	–

* SMALLER HOLE THAN IN CATALOG, FOR IMPROVED BEARING
** DEEPER REAMED HOLE, TO KEEP PLUG BELOW SURFACE



(250001 Plug Assy, Scale: 2/1)

TYPICAL MACHINING-ASSEMBLY DRAWING CALLOUT:

__ DIA __ DEEP (Max size of dia. not to exceed "N")
CDRILL (Min "B")/(Max "B") 125√("A" + .015) ± .015 DEEP
XXXYZZ PLUG ASSY.
INSERT PLUG AGAINST CDRILL BOTTOM
PRESS IN TAPERED PIN FLUSH WITH PLUG WITHIN ± .005

105.3 DESIGN CRITERIA

.331 The proof pressure of the Lee Plug must not be less than the burst pressure of the housing cavity. Generally specify a Class 3 fit for high pressure applications. Use of Class 3 facilitates possible salvage operations. A Class 1 fit may be used for low pressure applications in relatively inexpensive housings.

.332 The edge distance (from the plug centerline) must not be less than the nominal plug diameter.

.333 The cres steel Lee Plugs are not to be used in steel housings heat treated above 125,000-145,000 psi unless approved by design supervision. Consider the use of special plugs copper brazed or electron beam welded.

.334 Surface treatments such as passivating or anodizing should be applied before plug installation. The bore receiving the Lee Plug must not be plated.

110. SERVO SYSTEM DESIGN (COMPONENT RELATED)

CONTENTS

110.1 INTRODUCTION

This section provides a basic knowledge of general types of flight control systems and the hydraulic components used in these systems. The hydraulic flight control component designer needs to understand the overall functional requirements of the flight control system in order to optimize his design. At the preliminary design stage, system parameters must be ball-parked to get the aircraft design started. As the design develops, these parameters should not be allowed to be cast in concrete. The hydraulic component designer should determine what system/interface requirements are adding undue complexity, performance, reliability and cost to his design. These requirements should be reviewed with the design office and interfacing groups for possible compromise or elimination.

110.2 FLIGHT CONTROL SYSTEMS

Schematics and brief descriptions of various types of flight control systems follow. There are as many configurations of flight control systems as there have been variations in aircraft configurations. Each system has unique features or combinations of features. However, these systems have certain basic classifications. A typical version for each type will be described. Let it be understood, another, or several other versions could equally represent many of these systems. Wherever possible, the nomenclature used to describe these systems will be as defined in Addendum 1, Society of Automotive Engineers, Inc. (SAE) proposed AIR 1181 "Terminology of Redundant Flight Control Systems."

110.21 CLASSIFICATION OF FLIGHT CONTROL SYSTEMS

.211 MANUAL In a pure manual system, the pilot's force is supplied directly through mechanical load paths to an aerodynamic control surface, such as the controls of a light airplane or to a "flying tab" on larger airplanes which by use of the air loads give an aerodynamic boost to the pilot's effort. See Figure 110-1. The manual system has proven to be a reliable, safe means of aircraft control starting with the Wright Brothers' first flight at Kitty Hawk in 1903.

.212 HYDRAULIC BOOST WITH MANUAL REVERSION With progressive increases in aircraft size and speed, power boost control enabled the pilot to utilize the full maneuver capability of the aircraft. See Figure 110-2. With hydraulic power on, the pilot's force is applied directly through mechanical load paths to the aerodynamic control surface and to the boost actuator's servo valve. With the loss of hydraulic pressure, due to either a hydraulic system failure or a pilot operated shut-off valve, the boost actuator is in by-pass. An internal centering spring returns the actuator to neutral maintaining a fixed point for mechanical control of the surface.

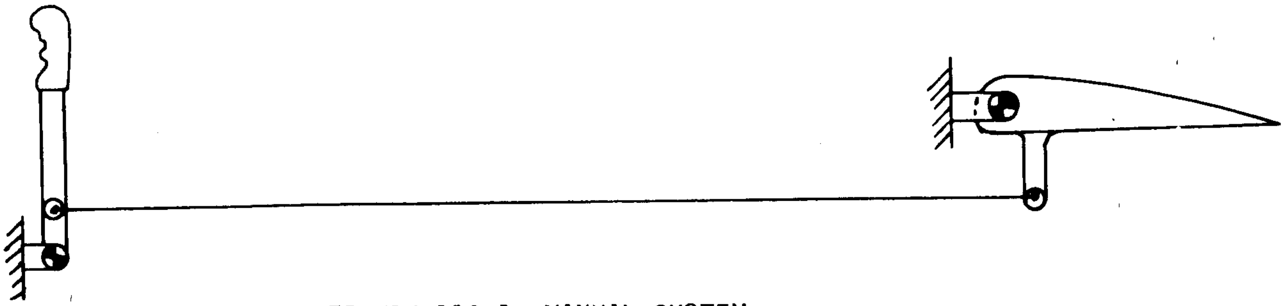


FIGURE 110-1 MANUAL SYSTEM

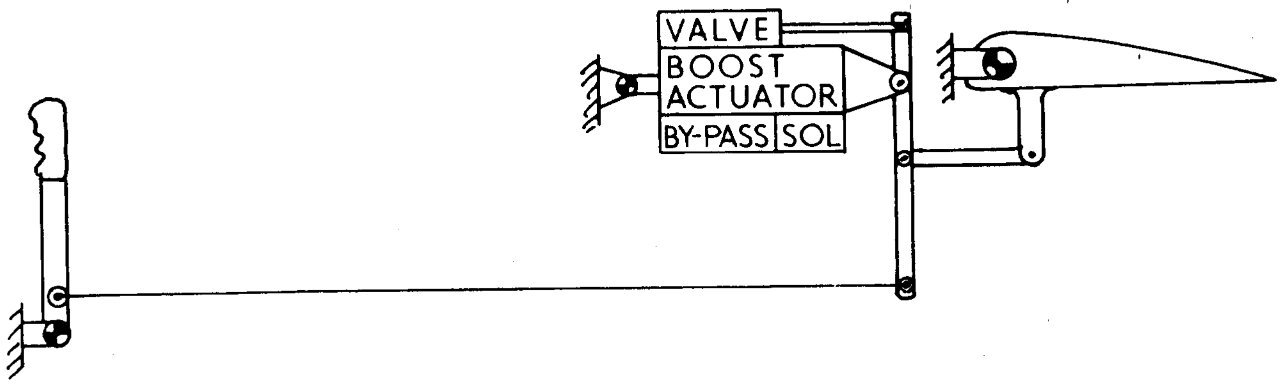


FIGURE 110-2 HYDRAULIC BOOST WITH MANUAL REVERSION

110.213 HYDRAULIC POWERED WITH MANUAL REVERSION As powered flight controls proved successful, it became possible to further increase aircraft size and speed by utilizing hydraulic power alone for surface actuation, except for emergency conditions. In a hydraulically powered system with manual reversion (Figure 110-3), the normal operation applies the pilot's force directly through mechanical load paths to a hydraulic servo valve which provides control of hydraulic power actuation of the surface. Artificial feel is provided to give the pilot proper handling qualities characteristics for control of the aircraft. The pilot has the capability of manual reversion where the pilot's force is applied directly to an aerodynamic control surface as described above. This type of actuation is used on the DC-8 and DC-9.

.214 FULLY POWERED Current aircraft designs have increased in size, speed, and maneuverability to where manual reversion is no longer desirable due to excessive pilot effort. In a fully powered flight control system (Figure 110-4) there are no manual reversion provisions. The pilot's force is applied directly through mechanical load paths to a hydraulic servo valve which provides control of hydraulic actuation of the surface. Artificial feel is provided to give the pilot the proper handling qualities characteristics for control of the aircraft.

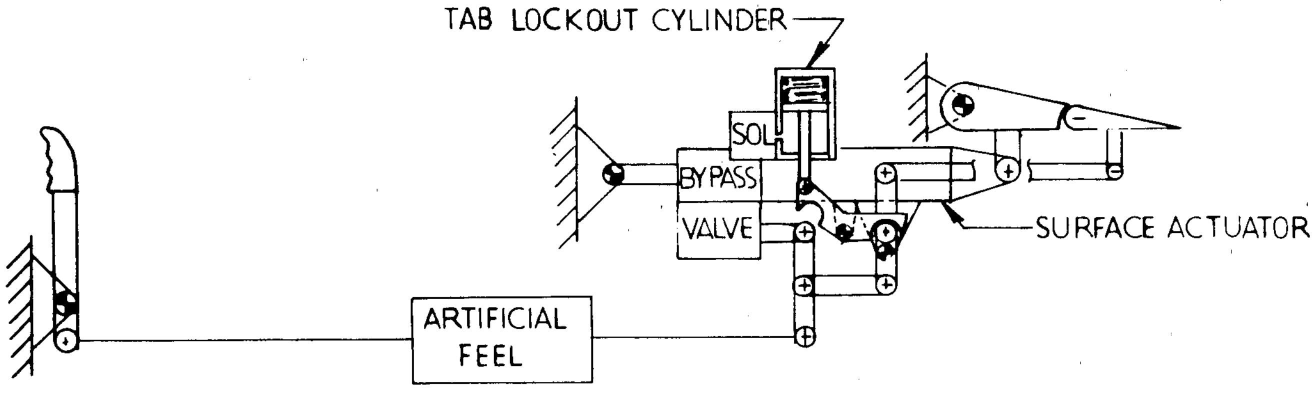


FIGURE 110-3 HYDRAULIC POWERED WITH MANUAL REVERSION

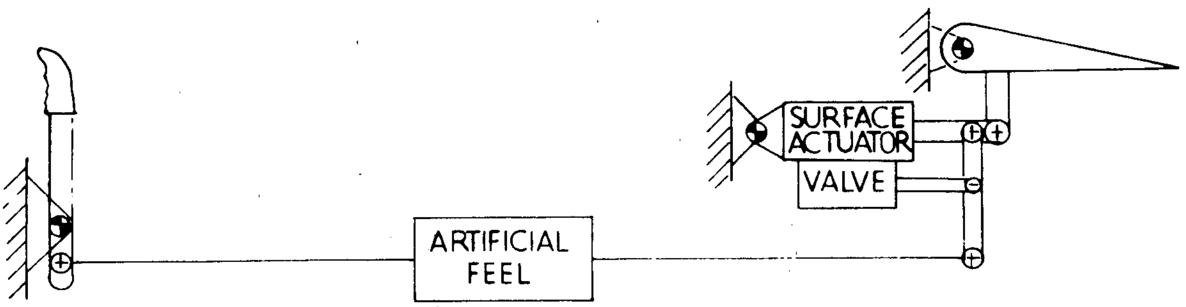


FIGURE 110-4 FULLY POWERED

Since the pilot no longer has any direct means to apply his force directly to an aerodynamic control surface, the FAA has updated FAR Part 25. Failures and combination of failures must be evaluated with the knowledge that the safety of the aircraft depends on the reliability of these fully powered systems. This type of system is used in most of the latest generation of transport and military aircraft such as the DC-10, B747, L-1011, F14 and F15, etc.

110.22 TYPES OF FULLY POWERED FLIGHT CONTROL SYSTEMS The fully powered aircraft system has many variations in control system configuration utilizing dual inputs of "manual" and "electrical." Manual input now refers to the direct pilot mechanical input to a hydraulic servo valve. [Note: There may be some confusion in terms. Previously, the word "manual" meant direct pilot control to an aerodynamic surface. Now, the term "manual" in a fully powered system means the direct pilot mechanical input control of a servo valve.]

There are many diverse powered flight control system configurations which will be encountered in current design. A few basic types will be described in order to present an overview of system classifications. It is outside the scope of this section to give the in-depth knowledge of these systems required for successful interface coordination. Also, as each system develops, the particular needs of the aircraft make each system unique.

110.221 <u>PILOT BOOST SERVOS</u> Pilot Boost Servos amplify the pilot's effort to overcome the mechanical system's friction, override forces, and surface actuator valve forces. The one shown in Figure 110-5 controls the surface actuator as a function of pilot input. With the boost servo inoperative, the boost servo centers, returning the surface to neutral or faired. This type is the most simple to mechanize and is suited for systems with multiple control paths and surfaces, such as the DC-10 spoiler system. The pilot boost servo can be designed to become a fixed link as shown in Figure 110-6. With the loss of hydraulic supply pressure due to operation of an electrical shut-off valve or hydraulic system failure, the boost servo acts as a fixed link, the pilot maintains control of the surface actuator by exerting higher forces. Since this would be a failure mode of operation, the increased pilot forces may be acceptable.

.222 <u>STABILITY AUGMENTATION SYSTEM (SAS)</u> Stability Augmentation allows the replacement of large surfaces required to provide inherent stability throughout the entire flight envelope with smaller surfaces and an associated control system to

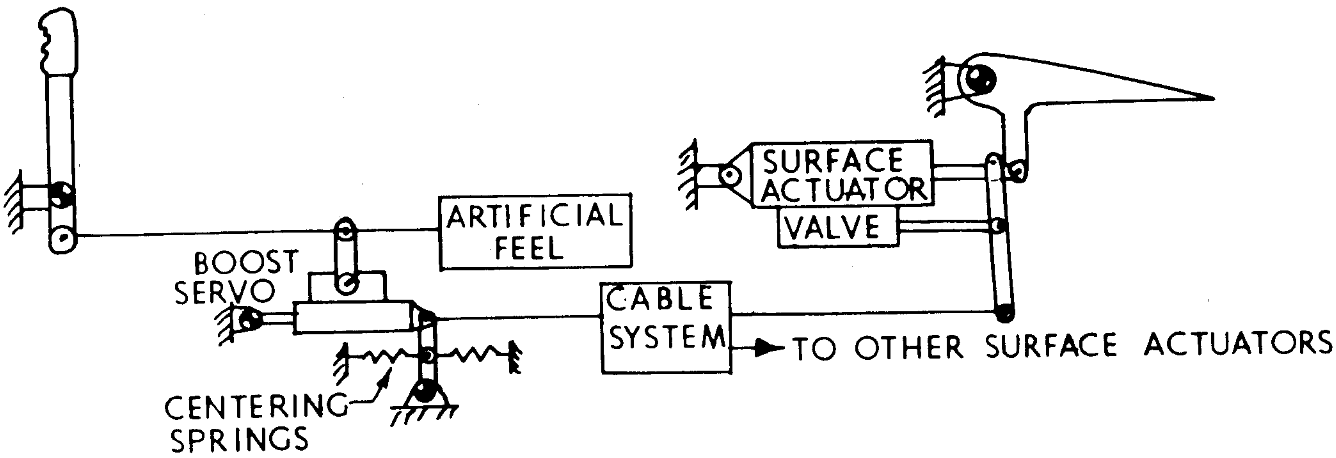


FIGURE 110-5 HYDRAULIC BOOST SERVO

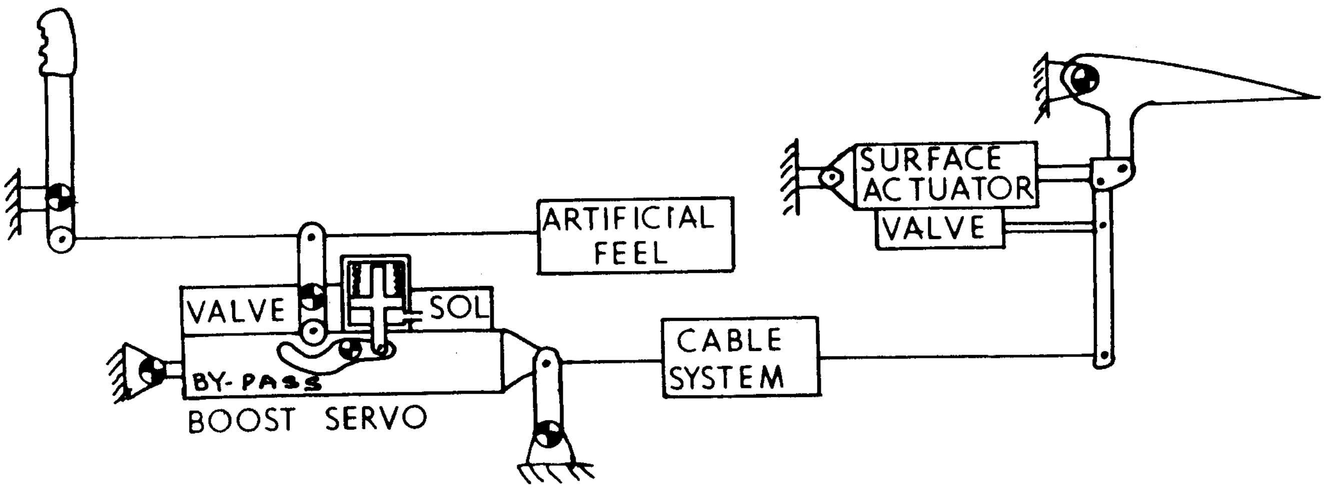
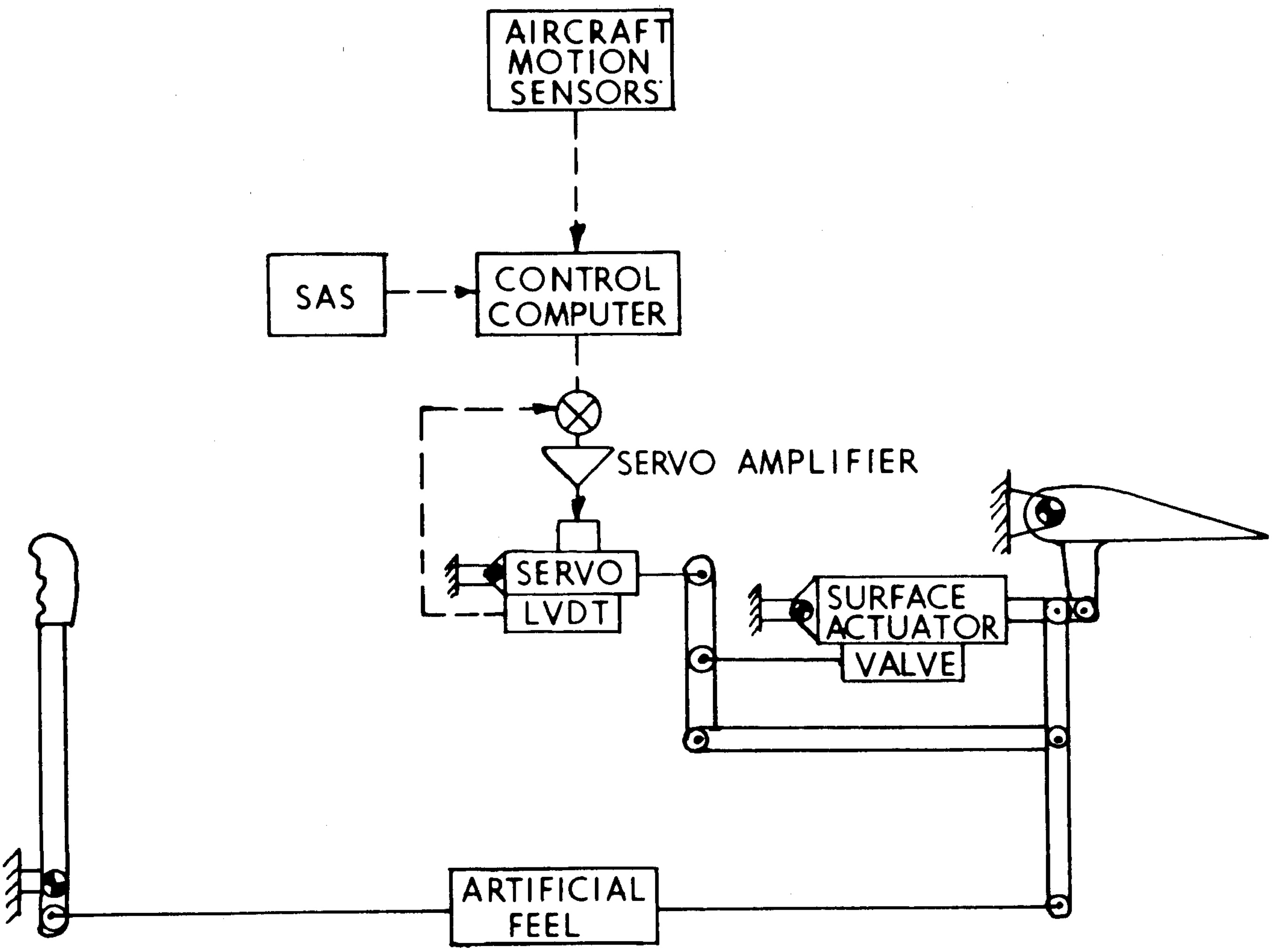


FIGURE 110-6 HYDRAULIC BOOST SERVO (WITH MANUAL REVERSION)

provide artificial stability, either for a small portion or most of the flight envelope, depending on design. For instance, SAS yaw-dampers allow the use of a smaller vertical tail surface. With the yaw dampers inoperative, some aircraft are restricted to slower flight at lower altitudes than their normal flight envelope, while other aircraft use SAS to improve passenger comfort and pilot handling qualities but have no flight restrictions with SAS inoperative.

SAS operates in series with the normal aircraft control system (Figure 110-7). Electrical command signals introduced into the SAS servo do not move the pilot's controls. Aircraft motion sensors feedback aircraft short period dynamics (such as dutch roll) to the SAS control computer. The SAS computer controls the surface actuator to position the corresponding aircraft control surface to develop stabilizing aerodynamic moments.



• MANUAL
• SERIES (SAS SUMMED WITH MANUAL)

FIGURE 110-7 STABILITY AUGMENTATION SYSTEM (SAS)

110.223 COMMAND AUGMENTATION SYSTEM (CAS) Command Augmentation is specifically designed to provide enhanced responses to pilot commands (Figure 110-8). For example, CAS can provide response quickening for large aircraft, by causing the control surface to first overshoot and then to washout to the pilot commanded surface position. Utilization of CAS in combination with altitude control will provide proper control coupling for STOL or V-STOL aircraft. CAS frequently involves proper integration of power and altitude control. A Stability and Command Augmentation System (SCAS) combines and optimizes SAS and CAS functions. CAS operates in series with the normal aircraft control system. Electrical command signals introduced by CAS do not move the pilot's controls. The electrical signal generated by movement of the pilot's control is fed to the Command Model. The model shapes the signal to produce the desired aircraft response. The signal is then summed with the measured aircraft dynamics to generate the surface command.

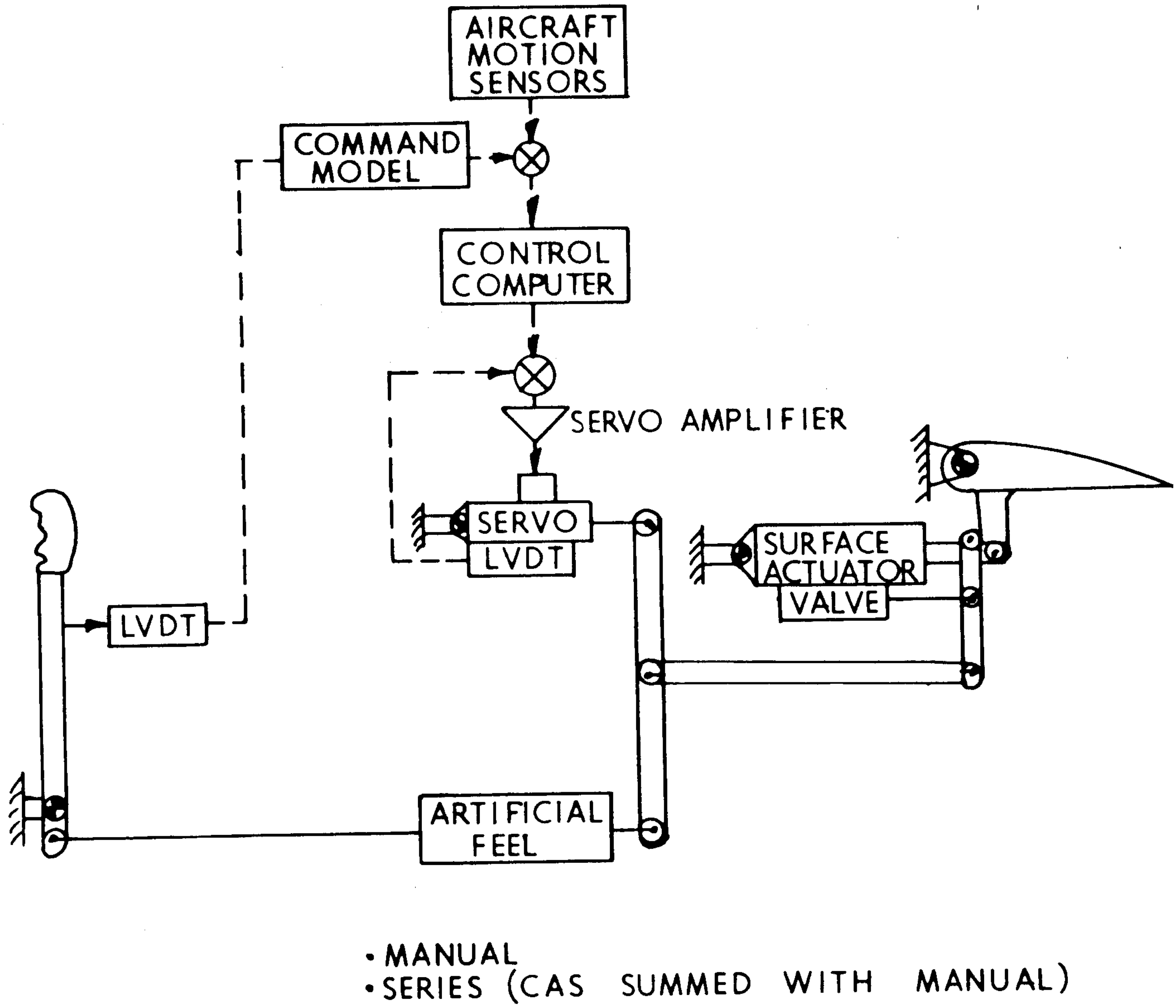
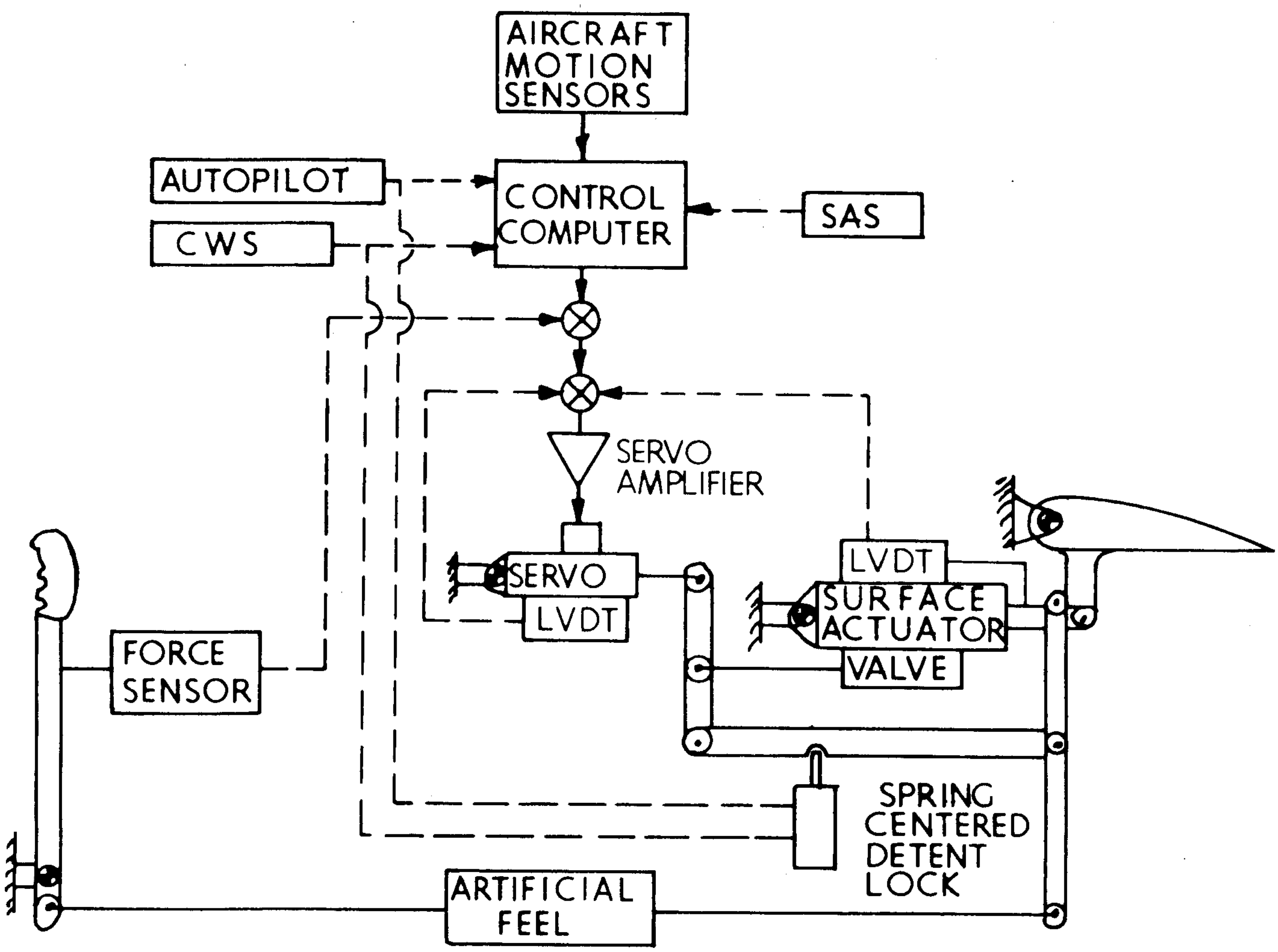


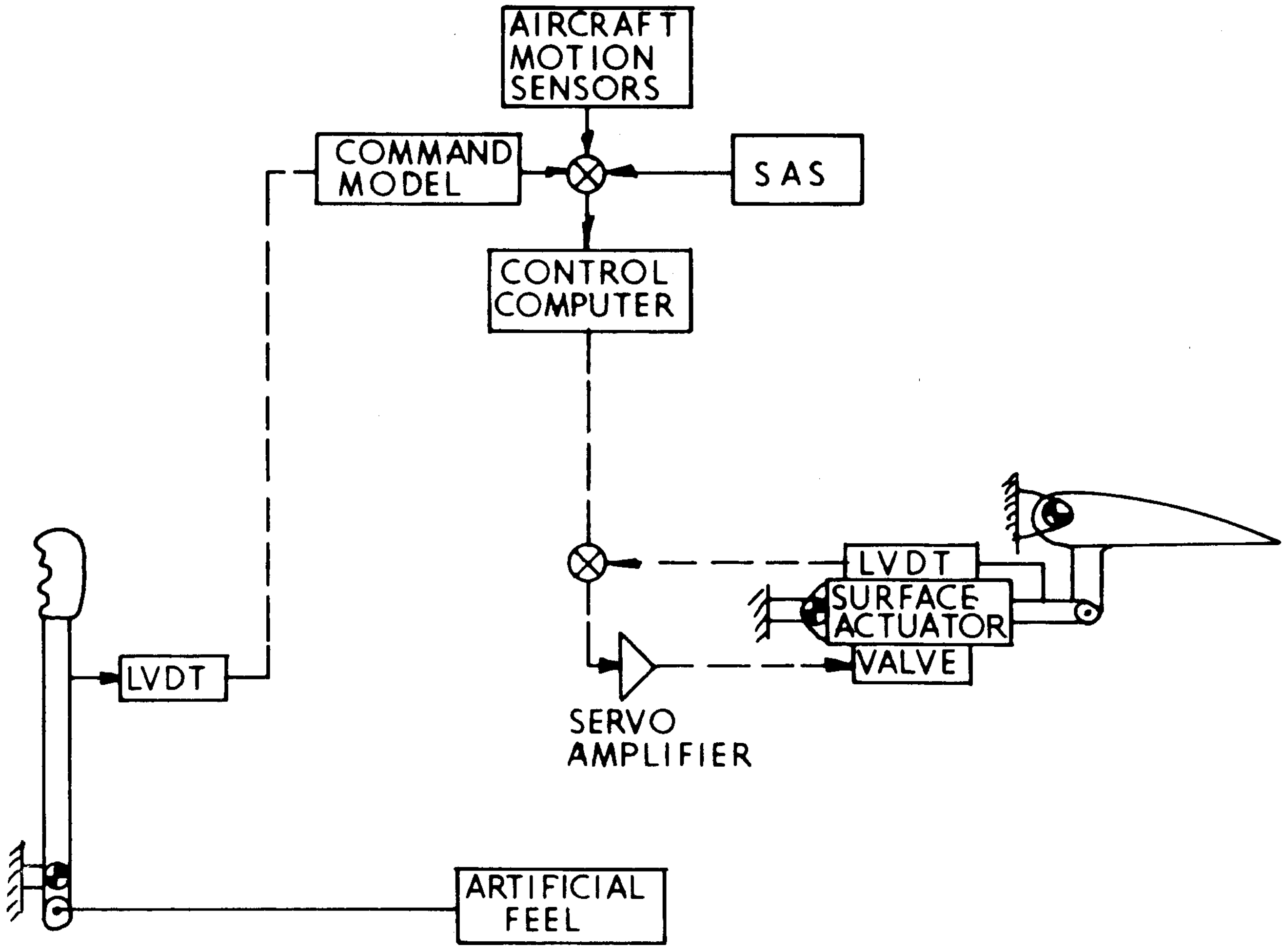
FIGURE 110-8 COMMAND AUGMENTATION SYSTEM (CAS)

110.224 INTEGRATED SYSTEMS The primary and automatic flight control systems are integrated into a complete airplane system (Figure 110-9). Pilot control of the aircraft is provided through the mechanical control system or through Control Wheel Steering (CWS). Automatic control may be selected for operation in cruise, landing, and rollout modes. The pilot can instantaneously disconnect the autopilot through a disconnect switch, or may override the autopilot through CWS or by exerting a force sufficient to break out the autopilot lockout detent. The inherent reliability requirements for each of these systems is dependent on the mode of operation and aircraft safety. For example, the DC-10 single channel cruise autopilot authority is mechanically limited and is not required to be fail-operative. Whereas, during Autoland, the autopilot has full authority and the system has fail-operational requirements.



(PRIMARY AND AUTOMATIC CONTROL SYSTEMS)
FIGURE 110-9 INTEGRATED SYSTEM

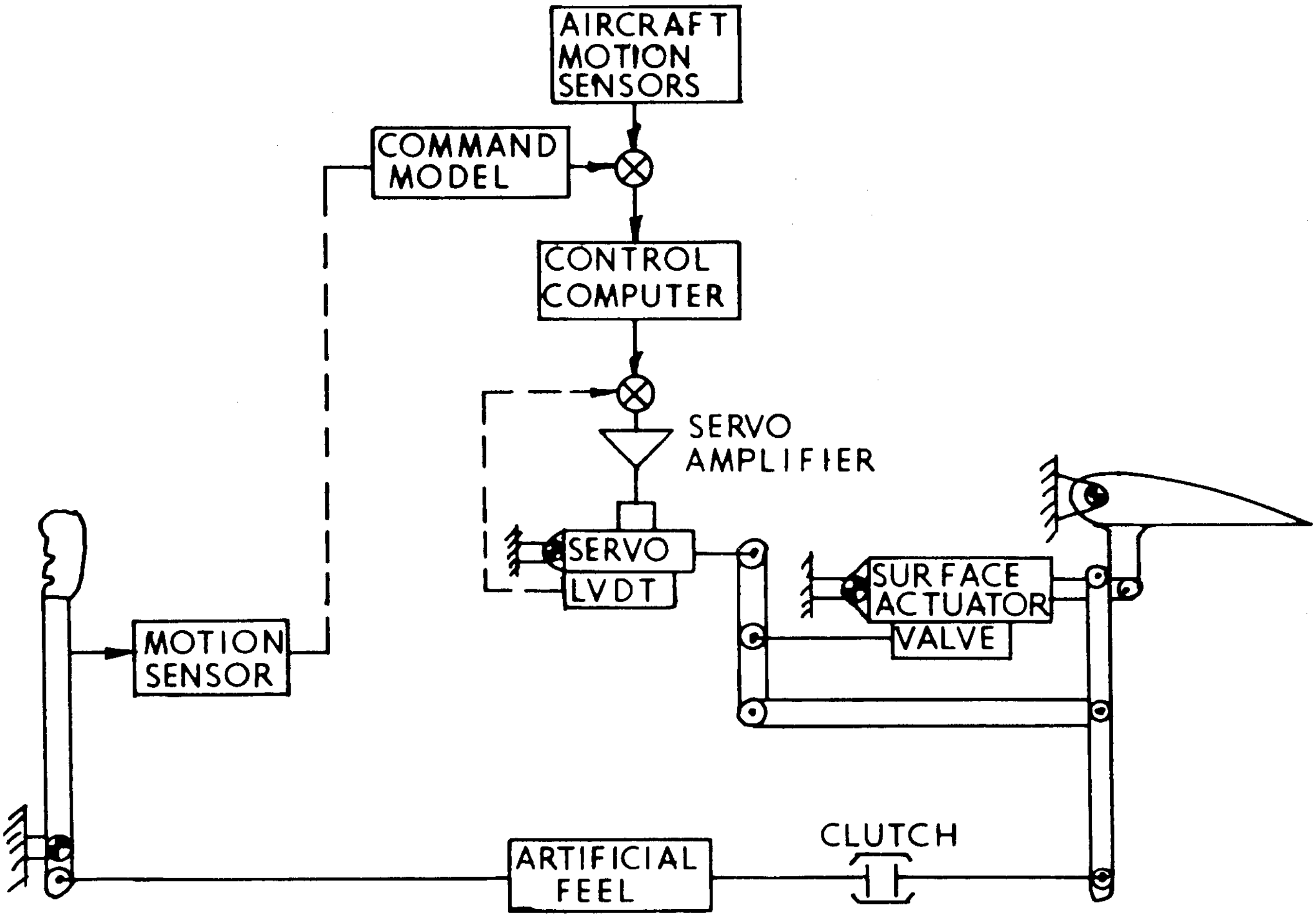
110.225 FLY-BY-WIRE (FBW) The wire in Fly-By-Wire means an electrical wire as shown in Figure 110-10. There is no mechanical connection between the pilot and the servo valve that controls the surface actuator. If the reliability of these systems were equal to that of mechanical cables and linkages, then the electrical system could replace the mechanical on a one-to-one basis. This is not true. The electrical system is not as reliable as the mechanical system and it is necessary to replace one mechanical system with multiple electrical systems. Studies to date indicate that for large military or commercial transports, the system should have two fail-operative capability. This means it will accept any two failures, including like failure with little or no degradation in system performance. In current mechanizations, the number of control channels varies from 4 to 5 depending on: (1) the type of logic to be used (medium select, majority vote, etc.), (2) failure effects removed or allowed to remain, and (3) number of control surface segments. The interface can be either electro-mechanical or electro-hydraulic depending on system mechanization and response requirements. FBW systems utilize either Central Hydraulic System power or localized Power By Wire (PBW).



•SERIES SAS/CAS
FIGURE 110-10 FLY-BY-WIRE

FBW control systems were developed for the control of space vehicles such as Gemini and Apollo. As aircraft flight control systems demand more and more artificial stability and more automatic control functions, the closer the systems come to FBW dependence. Therefore, it is hoped that research and development programs, such as the USAF modified F-4 and NASA's modified F-8, will yield substantial FBW technological gains.

110.226 PSEUDO FLY-BY-WIRE Pseudo Fly-By-Wire primary flight control system (Figure 110-11) is a FBW system with the capability to engage a mechanical "get home" backup control system. The degree of redundancy in the Pseudo FBW system can be less than for a true FBW system, since the backup system has the high degree of reliability inherent in a mechanical control system. The Concorde utilizes this type of system. (It is claimed that the Concorde's natural stability is satisfactory but autostabilization improves passenger comfort and results in easier piloting.) Therefore, the primary system has only two channels with a mechanical backup system. The mechanical system has acceptable but degraded performance. See Figure 110-11.



PRIMARY CONTROL — FLY-BY-WIRE
SECONDARY CONTROL — MANUAL

FIGURE 110-11 PSEUDO FLY-BY-WIRE

110.23 TYPES OF HYDRAULIC POWER SUPPLY FOR FULLY POWERED SYSTEMS

.231 CENTRAL HYDRAULIC SYSTEM The DC-10 hydraulic block diagram shown in Figure 110-12 is typical of central hydraulic systems. Prime power is generated at the engine accessory pad by mechanical energy conversion to hydraulic power by the engine-driven pumps. This hydraulic power is transmitted to the surface control actuator by means of hydraulic tubing. The closed circuit system operates at 3000 psi. Most commercial aircraft utilize flame resistant fluid such as Skydrol 500. MIL-H-5606 "red oil" has been used foremost in

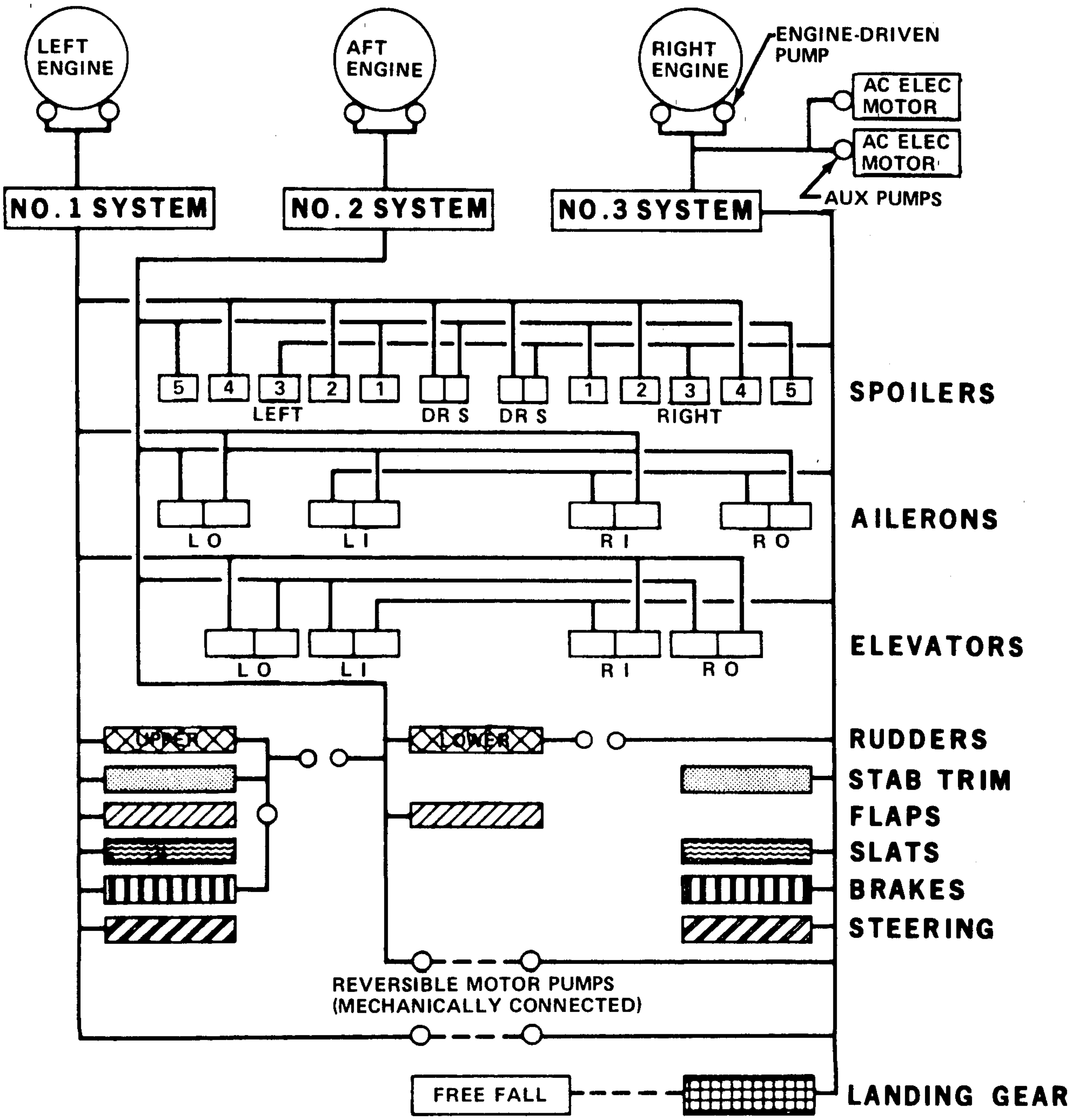


FIGURE 110-12 DC-10 HYDRAULIC SYSTEM

military application. This has been superseded by MIL-H-83282 which is compatible with MIL-H-5606 but is more flame resistant and operates at higher temperatures than MIL-H-5606. Central hydraulic systems for advanced military and commercial aircraft will probably operate at elevated pressure and temperatures such as 4000 psi and 500°F.

The central hydraulic system is capable of load sharing. Maximum roll, pitch, and yaw control rates are usually not required at the same time as maximum loads. Neither are maximum surface control rates required during maximum utility control such as landing gear retraction. Therefore, the total aircraft hydraulic horsepower requirement can be scaled down to meet realistic maximum combined power demands. The control designer obtains from aerodynamics the combinations of control rates which are required during various conditions, normal and abnormal (such as normal flare and abnormal one-engine-out wave-off). The control designer converts these rates into flow and power requirements. The optimum distribution to the surface actuators is obtained through coordination with the hydraulic system design engineer and aerodynamic engineering. Figure 110-13 shows a typical mission profile versus hydraulic system power requirements. Note flight control requirement effects on overall system power requirements.

The hydraulic flight control component's interface with the hydraulic power supply system presents more difficult problems in damping valve cut-off pressure surges as developing flight control systems demand larger surface throws and faster response systems. Some means of localized surge suppression without affecting actuator positionability or response may be required as higher flow and quicker response are demanded by advanced systems.

110.232 <u>POWER-BY-WIRE (PBW)</u> In a PBW aircraft the prime power is generated at the engine accessory pad by mechanical energy conversion to electrical power by the engine-driven generators. This electrical power is distributed to PBW units located throughout the aircraft. The PBW units contain an electric motor to convert electrical power to hydraulic power at the flight control surface drive station.

[Note: This type of unit is also referred to as an Integrated Actuator Package (IAP). This terminology is confusing, since units such as the DC-10 which integrate autopilot and manual inputs, are also called integrated actuators.]

Power-By-Wire units have been used successfully since the 1950's for missile applications such as the Polaris, where the duty cycle was short. The first commercial application was on the British VC-10 transports. In 1969, the Air Force began the survivable flight control system research and feasibility program. USAF development efforts on both power-by-wire and fly-by-wire were combined for this program. The decentralization of hydraulic power and multiple electrical control system have been attractive candidate methods for reducing military aircraft flight control system vulnerability. First part of the program was a "get home" F-4 stabilator power-by-wire unit called a "simplex" since it contained a single hydraulic power source.

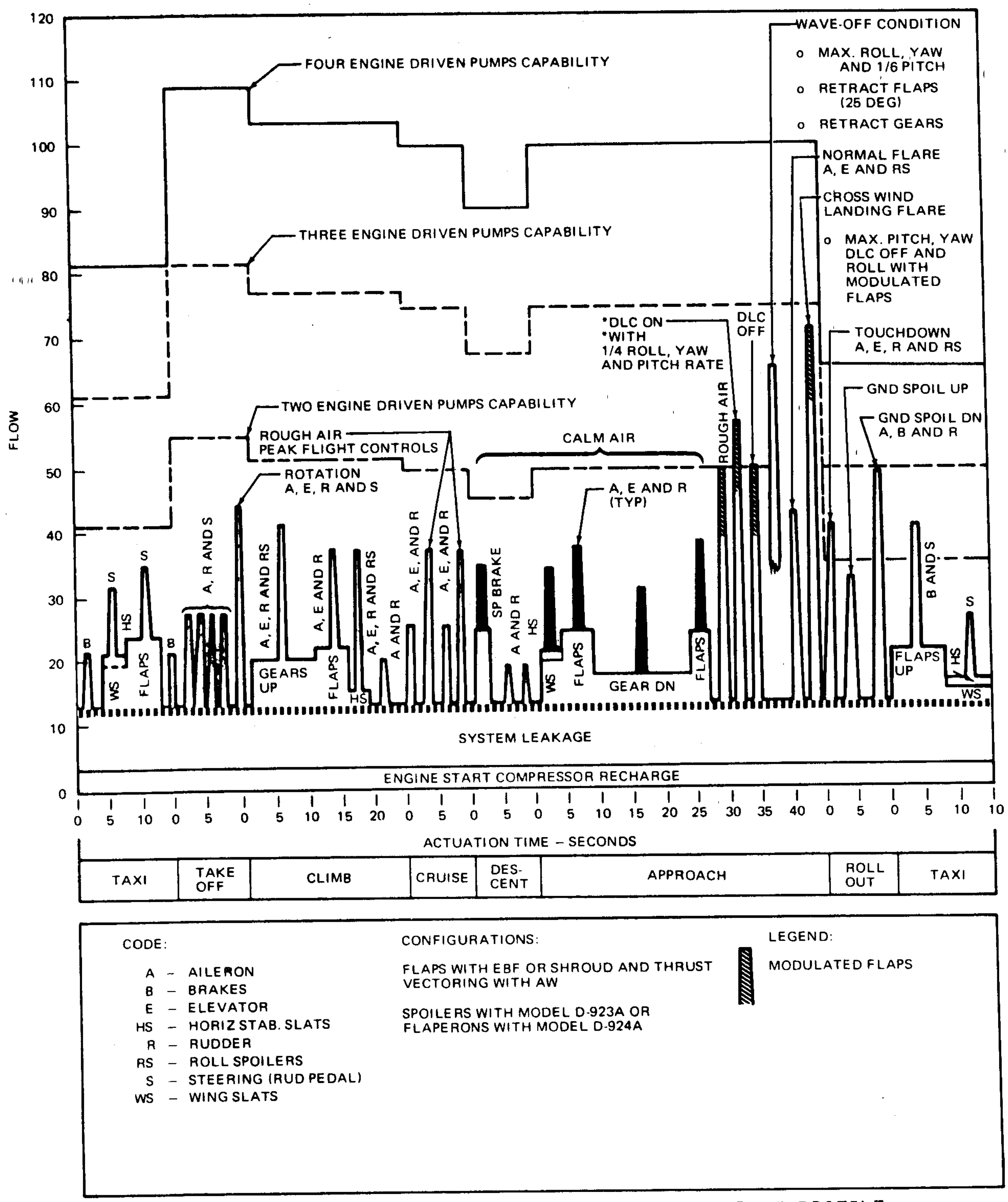


FIGURE 110-13 HYDRAULIC SYSTEM FLOW vs FLIGHT PROFILE

McDonnell Douglas, under USAF contracts, has successfully flight tested the "simplex." The second part of the F-4 survivability program will have a full performance 3-axis FBW-PBW flight control system with an emergency mechanical backup system. These PBW units are called "duplex" since they incorporate two hydraulic power sources. The last part of the program will be the removal of the mechanical backup system or pure FBW. See Figure 110-14.

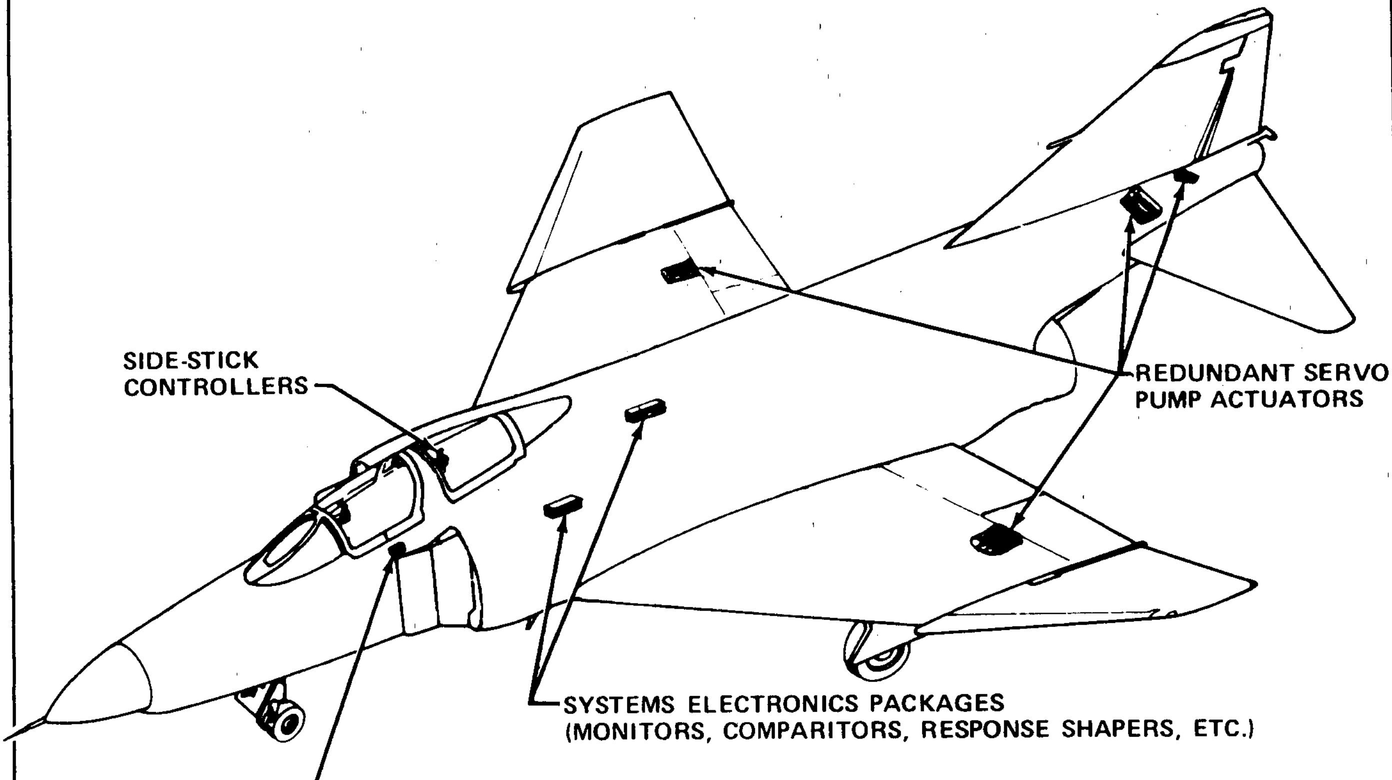


FIGURE 110-14 F-4C TWO FAIL OPERATIVE FLY-BY-WIRE POWER-BY-WIRE FLIGHT CONTROL SYSTEMS

Earlier versions of PBW for the VC-10 and the F-4 program operated at maximum pressures of 1500 to 2000 psi and used standard pumps. A servo valve controlled flow and pressure to either side of the main ram piston. Heat generation has been a problem in these units and low power requires large actuators. Later versions of the Boulton-Paul unit for the VC-10 and the Vickers units for the F-4 PBW program are servo-pump units utilizing higher pressures. Flow and load pressures are produced on demand by the servo control, thus reducing the heat generated. Also, the higher pressures may alleviate some of the weight penalty presently associated with PBW units.

Some of the advantages and disadvantages of PBW follow. The main military advantage seems to be survivability, particularly when used in conjunction with split surfaces. Thus when the aircraft is subjected to small arms fire the probability is that enough units will survive for completion of the mission. It is also argued that the units present a smaller target than the central hydraulic systems. From a commercial standpoint, PBW permits the apparent easier routing of electrical, rather than hydraulic power and there is some argument that troubleshooting is more easily accomplished since there is no interreaction with the other hydraulic power units except through the electrical power sources. Of course, the final survivability redundancy of the system depends on how many engines you have (the prime mover), how many independent electrical power sources, and how well electrical lines are separated and shielded. As mentioned previously, there is a weight penalty associated with the present level of PBW development. Since there can be no load sharing as is done in a central hydraulic system, each unit must provide its own power requirements. Therefore, the total horsepower requirement is greater than for a central hydraulic system.

110.24 FUTURE TRENDS

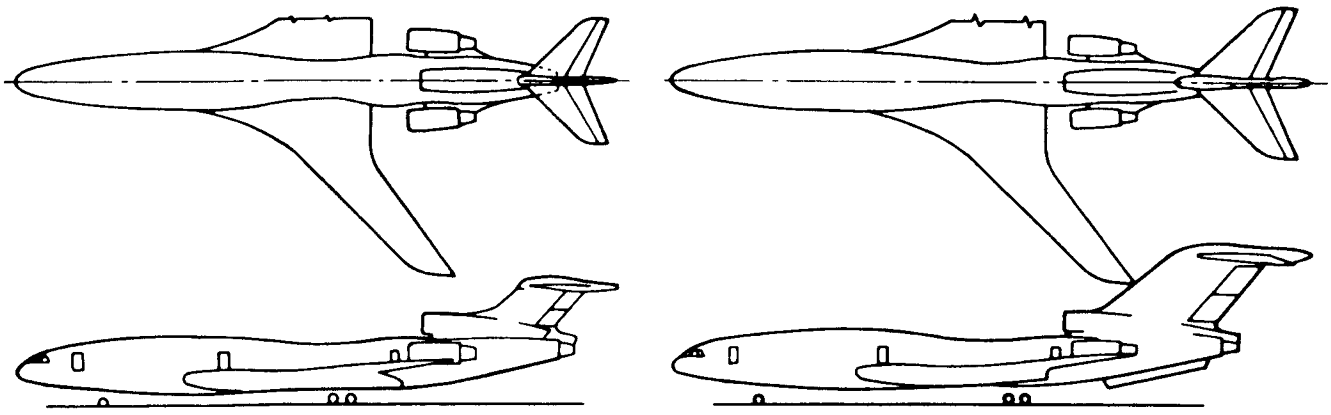
.241 ACTIVE CONTROL SYSTEMS The development of an air vehicle configuration from a given set of mission requirements has traditionally involved only three primary disciplines: aerodynamics, propulsion, and structures. Recent advances in control systems technology have provided new capabilities in this discipline which may have a significant impact on the vehicle configuration and on aircraft performance. Some of the areas of application presently being considered include:

a. Relaxed Aerodynamic Stability

Using conventional design practices, the vehicle configuration has been constrained in that it had to be statically stable. Utilizing full-time, fail-operative stability augmentation systems (SAS), this requirement may be relaxed with the subsequent benefits of reduced wing loading and trim drag, and possibly reduced tail volume with its advantages of reducing structural weight and wetted area drag. Some studies have shown a 25-percent saving in tail area with corresponding savings in drag and weight. See Figure 110-15.*

b. Maneuvering Load Control

This concept utilizes a sensing of vertical acceleration to shift the wing lift loads from the outboard wing to the inboard section during maneuvers and consequently reduces wing bending moment. The controls create this distribution by deflecting inboard and/



ADVANCED TECHNOLGOY

- FLIGHT CRITICAL STABILITY AUGMENTATION
- MANEUVER LOAD ALLEVIATION
- GUST LOAD ALLEVIATION
- OPTIMUM CRUISE C.G. AND REDUCED C.G. RANGE

CONVENTIONAL TECHNOLOGY

- STATICALLY STABLE
- FAIL-SAFE YAW DAMPER
- INHERENT ACCEPTABLE STALL

FIGURE 110-15 TAIL-SIZE SAVINGS

or outboard wing surfaces such as flaps or spoilers in response to maneuvering acceleration.

c. Fatigue Reduction (Gust-Load Alleviation)

Gust-load alleviation reduces fatigue damage due to gusts and is related to improving ride control. Sensors are located in strategic wing locations to sense the lower dynamic frequencies associated with structural fatigue. The controls deflect wing surfaces in response to acceleration and/or angle of attack. The system may simultaneously deflect the elevator to maintain airplane attitude.

d. Flutter Suppression

The higher dynamic frequencies of the wing structure (wing deflection and deflection rates) are sensed and the controls move wing surfaces (possibly auxiliary surfaces) to suppress flutter.

e. Ride Control

Reduction in accelerations and rotations by means of improved piloting performance in turbulence or critical maneuvers by means of a full-time, fail-operative Stability and Command Augmentation System (SCAS).

*Reprint from an article published in Astronautics and Aeronautics, "Active Controls Changing the Roles of Structural Design," August 1972, by Ray J. Hood, NASA Advanced Technology Transport Program Office.

A Control Configured Vehicle (CCV) is a vehicle which is designed to optimize performance and weight by applying some or all of the above technologies. NASA Advanced Technology Transport (ATT) studies show a predicted return on investment (ROI) and profit gains as shown in Figure 110-16. The evaluation of the predicted gains assumed a three-engine Mach -0.98 aircraft made of aluminum and just meeting FAR 36 noise regulations. The uncertainties of economic predictions must be considered when evaluating these results. Candidate vehicles are STOL, V-STOL, SST and high-performance attack aircraft.

These systems reduce drag and/or weight by replacing structure with active control technology. Since the very survivability of the vehicle depends on these systems, they must have as high a degree of reliability as the structure they replace. Hardware safety can be improved through improvement in failure rates and the use of redundant components (from hydraulic power units to "black box" components); however, the reliability of the software/hardware interface and the software systems pose new and complex problems. Digital-Controls technology may provide the necessary reliability through a combination of reliable hardware and software. Several approaches to implementing Digital-Controls are being developed which offer promise of providing greatly increased functional reliability, such as fault-tolerant computers and reconfigurable computer systems. Software will provide performance monitoring, data loading and self-testing to improve reliability as well as maintainability. Present NASA-sponsored programs are aimed at making these systems economically feasible for commercial applications by the early 1980's.

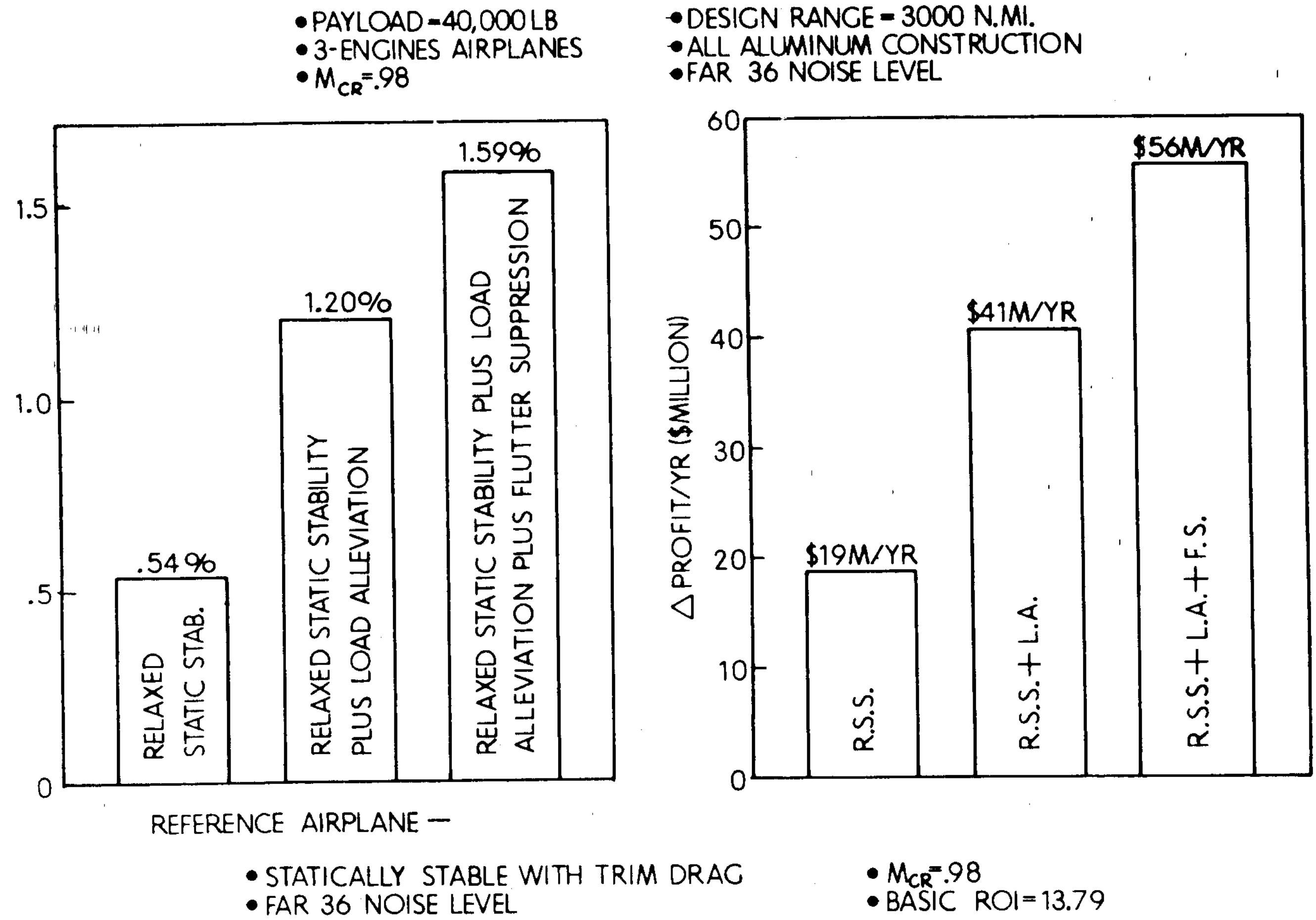


FIGURE 110-16 ECONOMIC BENEFITS OF ACTIVE CONTROLS

110.242 FLY-BY-WIRE Since survivability of the aircraft may depend on electrical control systems anyway, many advanced systems will be Fly-By-Wire or pseudo Fly-By-Wire. The control systems being developed for the space shuttle, NASA and Air Force-sponsored Fly-By-Wire programs, and private research programs such as our own IRAD Fly-By-Wire program, are providing many varied and promising design approaches to multiple fail-operative control systems. The development of fail-operational-hydraulic components will be required to perform voting or failure neutralization techniques.

Where pseudo Fly-By-Wire systems are used, servo valves with control valve forces less than present state of the art may be required. Pilot effort must be reduced to make a mechanical-hydraulic backup system feasible.

.243 SURVIVABILITY-VULNERABILITY Survivability developments are being made in the design of Central Hydraulic Systems such that failure effects are limited to hydraulic subsystem rather than the entire hydraulic system by automatic isolation. Reservoir Level Sensing or Flow Difference Sensing systems detect exterior hydraulic leaks and isolate the associated hydraulic subsystem. With this capability, only the functions serviced by the failed subsystem are inoperative. A schematic of a typical isolation manifold system is shown in Figure 110-17. The McDonnell Douglas F-15 uses this type of concept. Figure 110-18 shows a possible combination of Central Hydraulic Systems and Power-By-Wire Units. The electrically powered and controlled units provide a back up to the mechanically powered and controlled systems and vice versa.

Survivability will continue to receive in-depth research. For instance, how to establish realistic combat damage "hit" patterns which can be utilized for design evaluations. The relative vulnerability of Central Hydraulic Systems versus Power-By-Wire units needs to be determined.

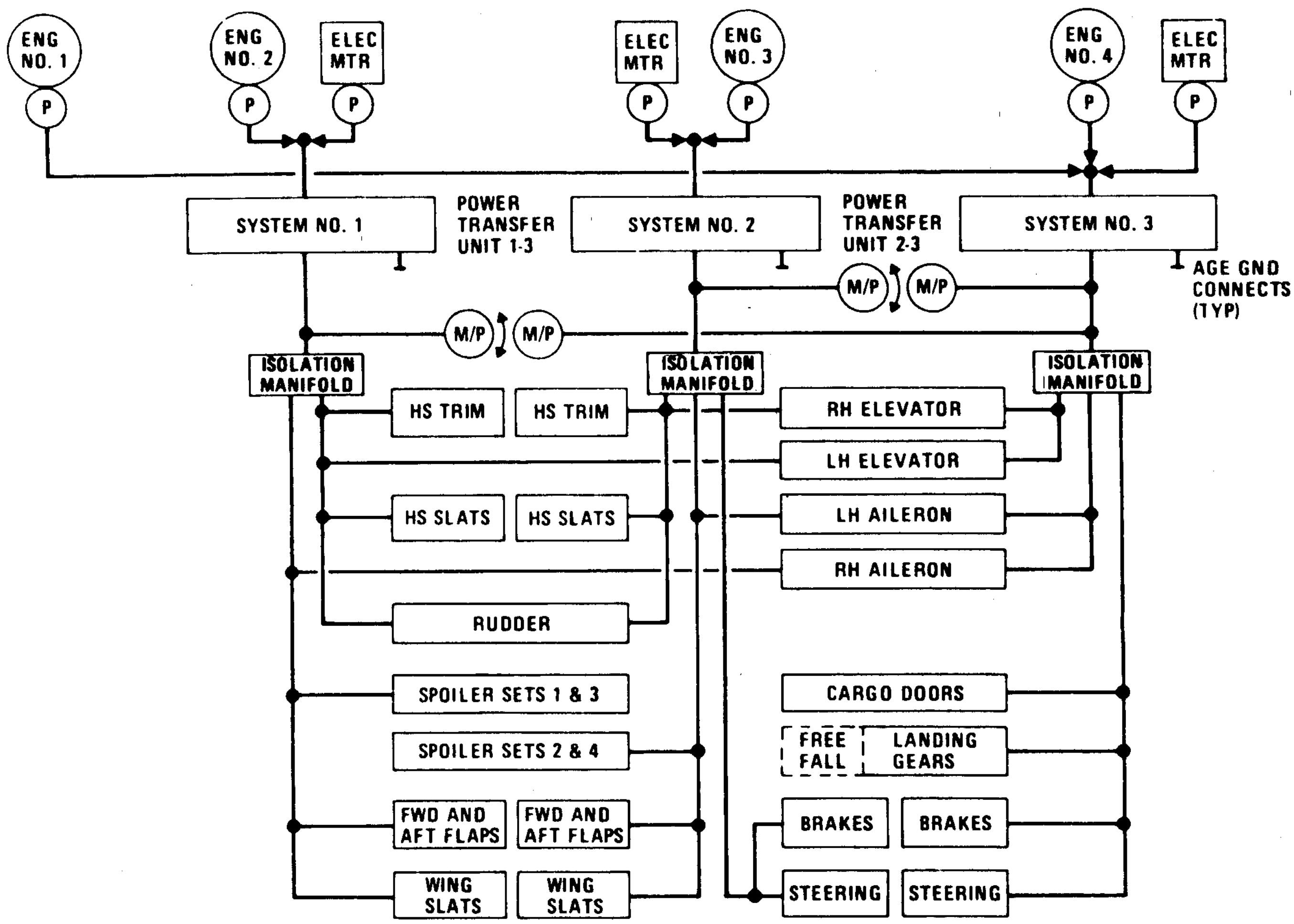


FIGURE 110-17 PROTOTYPE HYDRAULIC SYSTEM BLOCK DIAGRAM

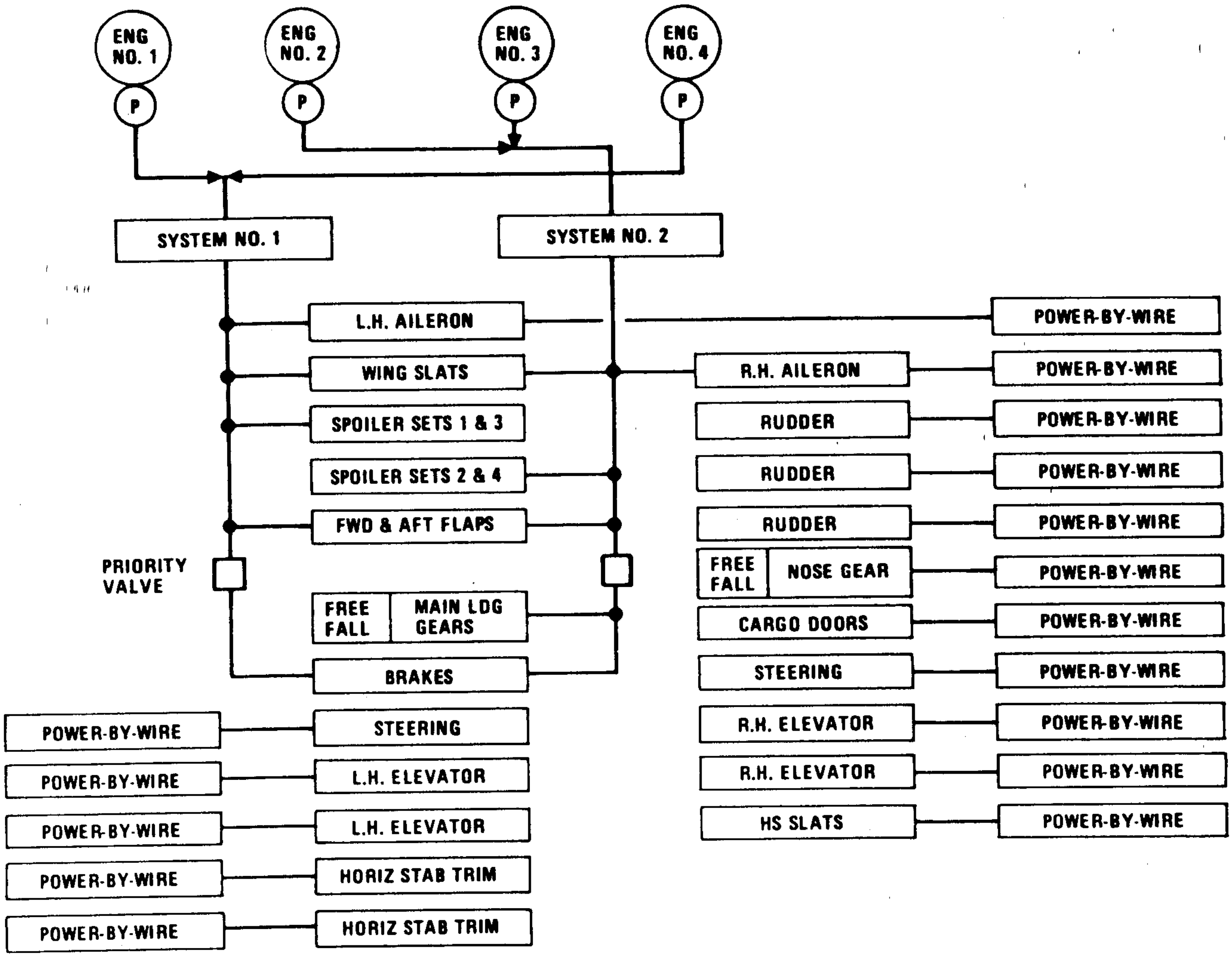


FIGURE 110-18 POWER-BY-WIRE HYDRAULIC SYSTEM BLOCK DIAGRAM

110.3 SUMMARY

These advanced inflight control systems are developing rapidly. The hydraulic flight control component designer must keep abreast of and participate in the development of these advance systems. He needs in-depth knowledge and understanding in order to optimize the hydraulic component interface requirements.

One of the best places to start learning about current technological advances is by studying some of the very useful data available in the Hydromechanical IRAD file. See 110.5, "Technical References."

110.4 TERMINOLOGY FOR FLIGHT CONTROL SYSTEMS

110.41 SCOPE This subsection, proposed by the SAE Committee A6, presents definitions of terminology used in conjunction with redundant flight control systems. No details of specific redundant system design approaches are given. Likewise, no recommendations are included for system performance and design requirements.

Redundancy may be provided within individual components of a flight control system, or it may be used for the system in total. The scope of this AIR is sufficiently broad to include both of these extremes.

110.42 ABBREVIATIONS

AFCS - automatic flight control system

CAS - control augmentation system

DFO - double fail operative

FBW - fly-by-wire

FCS - flight control system

FS - fail-safe

HO - hardover

MVL - mid-value logic

SAS - stability augmentation system

SFO - single fail operative

TVC - thrust vector control

110.43 TERMINOLOGY

Active - An adjective used to describe the nature of a device or the condition of a device after a failure. An active device will have at least two modes of failure: (1) open and (2) hardover.

Automatic Flight Control System - The portion of a flight control system which provides automatic augmentation and/or control of the stability, flying qualities, and flight path of an aircraft.

AFCS Pre-engage Synchronization - The process of integrating the AFCS output, prior to AFCS engagement, and summing the integrator signal with the AFCS servo command in a sense to drive the output to zero, thus minimizing AFCS engage transients.

Averaging System - The type of redundant system using two or more active channels wherein the individual channel outputs are summed to provide an average output. All channels are normally operative so performance degradation will occur after a failure. An example of an averaging system is the use of multiple control surfaces on an airplane, each individually actuated.

Channel - The term describing a single control path within a device or system that may contain many. A channel is an entity within itself and contains elements individual to that channel. Thus, it is implicit in this definition that a failure in one channel does not cause failure in another. A model may be used as a reference channel in a detection-correction system.

Control Augmentation System - A vehicle flight control system having both electrical and mechanical control of one or more axes wherein the electrical control system responds to the error between the commanded vehicle motion and the actual vehicle motion. The electrical control authority and gains are such that the electrical control predominates.

Control Authority - The amount of control surface deflection that can be produced by electrical signals relative to that which can be produced manually.

Detection-Correction System - The type of redundant system wherein a failure or out-of-operating tolerance condition is detected and corrective action is taken automatically. This may involve switching to a standby channel or if two or more channels are normally operating, correction may involve switching-out the failure. Inherent in this type of system is the existence of a finite time for detection-correction. With detection-correction systems it is possible to use a model of an active channel as a reference channel in order to extend the failure capability of the system.

Double Fail Operative - A condition or requirement wherein an active control device or system can sustain any two failures within the system and remain operative. It is implicit with DFO that the system be able to accept identical failures in two of its channels. Unless specifically stated it is understood that no nominal loss of performance occurs after one or two failures.

Dual Load Path - A type of passive paralleling wherein two separate load carrying paths exist from the control system input (if the input is mechanical) to the system output. Each load path is capable of carrying full load so that failure of any one member will not jeopardize system performance.

Electrical Logic - Logic, as for failure detection or correction, performed with electronic or electrical components.

Electrical Primary Flight Control System - A flight control system wherein the pilot's command signals are transmitted to the aircraft flight control actuators through electrical wires.

Equalization - The use of feedback to achieve close coincidence between the outputs of two or more channels in a redundant control system. Equalization may be necessary in order to reduce the transient that could occur with shutoff of a failed channel, or it may be necessary to minimize the adverse effects of normal tolerances.

Failure - A noun describing the state of having failed. In dealing with redundant flight control systems a failure occurs when a device within the system fails to function within prescribed limits without regard to the cause of the failure. Thus a failure may be:

1) Any loss of function of any element within the control system.

2) Loss of supply power to the system.

3) Erroneous hardover conditions or loss of control intelligence at the signal input.

4) Any out-of-tolerance condition that exceeds normal operating limits.

Fail Functional - A more limited case of fail operative wherein performance is significantly degraded following a failure.

Fail Neutral - A special case of fail-safe where the control device or system fails to a passive null or locked at null condition.

Fail Operative - A general term for describing a condition or system wherein operation continues after a failure. A more explicit description is given by SFO or DFO. In a true fail operative situation, a failure will cause no nominal loss of performance.

Fail-Passive - A condition or requirement wherein the failed device or system ceases to create any active output. In the purest sense a device that fails passive would simply remove its presence from the control system. However, a device is still considered fail-passive if it remains in the system but acts only as an additional load.

Fail-Safe - A condition or requirement wherein the control device or system ceases to function but the conditions or consequences resulting from the failure are not hazardous and do not preclude continued safe flight. The condition following failure may be completely passive, or it may involve driving to a predetermined nonactive condition.

Fly-By-Wire - A flight control system wherein one or more axes of vehicle control information is, at one point or another, completely electrical. A true FBW system does not have manual reversion, manual backup, or manual override.

Fully Powered Control - A vehicle flight control system that uses irreversible actuation such that no manual reversion capability exists.

Hardover - The type of failure wherein the output of the failed element is at its maximum value. In cases having a polarity of output, a hardover failure may be of either polarity.

Hydraulic Boost - The use of hydraulic power actuation to reduce the pilot effort needed for control of the vehicle.

Hydromechanical Logic - Logic, as for failure detection or correction, performed with only mechanical elements wherein the information is in the form of hydraulic pressures or flows.

Majority Voting System - A redundant system wherein the outputs of three or more active channels are summed and the summed output is fed back to each channel. When a failure occurs the feedback causes the unfailed channels to act to offset the failure, hence immediate shutoff of the failure is not necessary.

Manual Override - The condition of the pilot overriding the AFCS through a cable and/or linkage system and exerting control in excess of the AFCS authority or in opposition to the AFCS command.

Manual Reversion - The capability of reverting from hydraulic and/or electrical control to mechanical control wherein the pilot's control force is applied directly to an aerodynamic control surface.

Mid-Value Logic System - A redundant system having an odd number of active channels wherein the system output is the same as that of the center channel.

Model - A device used in a detection-correction redundant system to simulate the performance of a channel used for control. Typical models are electrical analogs or scaled-down electrohydraulic servoactuators.

Monitor - A device used in a redundant system for sensing the operation of a channel such that failures may be detected.

Open - The type of failure wherein the failed element disconnects the normal control path within a device. Such a failure either prevents the signal from passing or seriously distorts the signal that passes, through a channel.

Parallel Actuators - Two or more actuators arranged in parallel to drive a single load. Usually parallel actuators are physically separated, each with its own output rod, and are tied together by the load in a force or torque summing fashion. Thus parallel actuators usually provide a rip-stop design.

Parallel Servo - A servo located in a control system so that the servo output drives in parallel with the major input. Usually this arrangement is used with actuators which perform an alternate function to that of the pilot. The parallel servo output will drive both the pilot controls and the flight control system.

Passive - An adjective used to describe the nature of a device or the condition of a device after a failure (see fail-passive). A passive device will only fail open.

Passive Paralleling - The simplest and most common type of redundancy wherein two parallel functional devices are utilized such that if one fails the second is still available. This approach is limited to the more simple elements of the control system which can only fail-passive, such as springs and linkages. When failure of one element occurs, there is an acceptable change in performance or capability.

Pseudo Fly-By-Wire - A flight control system wherein one or more axes of vehicle control information is normally, at one point or another, electrical. Pseudo fly-by-wire systems have manual reversion or manual override.

Quadrex - An adjective meaning fourfold, as used for a quadruple redundant system.

Quadruple Redundant - A redundant control system having four channels so as to provide multiple failure capability such as SFO/FS, DFO or DFO/FS.

Reset - The process of returning a system to the original "no failure" condition after a failure has occurred and has been corrected. Some systems have the capability of providing this function automatically.

Reversion - In an active-standby system, the process of changing-over control from the active to the standby channel.

Rip-Stop - A mechanical design criteria that requires mechanical separation of hydraulic systems where more than one pressure source exists. With this criteria any one piece of material can be used only to contain the fluid from one pressure source, so if material fracture occurs it cannot progress and cause loss of two hydraulic systems. Thus, if two hydraulic supplies are to be used, they can never enter the same body anywhere in a system designed to provide rip-stop.

Series Servo - A servo located in a control system so that the servo output adds to that of a major input. This arrangement is commonly used with SAS actuators to superimpose control on primary commands. The series servo output will not cause motion at the major input.

Single Fail Operative - A condition or requirement wherein an active control device or system can sustain any single failure and remain operative. Unless specifically stated, it is understood that no nominal loss of performance occurs after the failure.

Stability Augmentation System - The portion of a flight control system that improves the handling characteristics by modifying the aerodynamic response of the vehicle. The SAS generally has limited authority. Because SAS signals are annoying to the pilot's "feel," they are normally introduced by a series servo that does not cause stick motion.

Standby - A term used to describe the normal status of a channel in a detection-correction redundant system that may be switched into control in the event of a failure of a normally operating channel.

Start-Up - The period of time between initial system energization and the point where nominal system control is attained.

Synchronization - The process of preadjusting the outputs of two or more channels so as to reduce deadzone if they operate together, or to reduce the switchover transient if they are operated separately.

Tandem Actuators - Two or more coaxial actuators that are mechanically constrained to move together. Usually a tandem actuator has two pistons on the same rod, carried in a single actuator cylinder housing. Separate cylinders (with a common piston rod) can be used to give partial rip-stop protection.

Thrust Vector Control - A type of vehicle attitude control wherein the direction of thrust of an engine or nozzle is caused to move with respect to the airframe.

Triple Redundant - A redundant control system having three channels so as to provide continued operation after a single failure or to provide SFO/FS operation. In a triple redundant detection-correction system, one channel may be a model.

Triple Tandem - A tandem actuator having three separate actuation sections.

Triplex - An adjective meaning threefold, as a triplex valve, a triplex actuator, etc.

Voter - A binary logic element or device that compares the signal condition in two or more channels and changes state when a predetermined signal mismatch occurs.

110.5 TECHNICAL REFERENCES

110.51 FLIGHT CONTROL SYSTEMS

"Integrated Servo Pump Actuator Aircraft Flight Control," K. Becker, N. Pedersen, Vickers Div., Final Presentation to AFFDL, WPAFB, Contract F-33615-70-C-1088, December 1970.

"Research and Development of A Servo Actuator Package for a Tactical Fighter Aircraft," D. H. Stalcup, P. S. Coppola, J. E. Rinde, General Electric Co., A Technical Report AFFDL-TR-71-19, January 1971.

"Research and Feasibility of An Integrated Servo Pump Actuator Package for Aircraft Flight Control," K. F. Becker, N. F. Pedersen, Vickers Div., Technical Report AFFDL-TR-71-19, January 1971.

"Fly-by-Wire Redundant Actuator," J. E. Emfinger, Sperry Rand Corp., Sperry Rand Engineering Review, Fluid Power and Control, 1969.

"Survivable Flight Control System Program - Simple Actuator Package," D. S. Hooker, R. L. Kisslinger, M. S. Smyth, MAC, MDC, Technical Report AFFDL-TR-70-135, November 1970 (PDL 110740).

"Hydraulic Power Controls (Brochure)," Dowty Rotol, Ltd., Boulton Paul, July 1969.

"The Dowty Group (Brochure)," Dowty Rotol, Ltd., September 1968.

"VC-10 Aircraft Overhaul Manual for Powered Flying Control Units - Types P163, P164 and P66; MK 1 and MK 3," November 1967.

"VC-10 Aircraft Overhaul Manual for Electronic Test Equipment," July 1964.

"VC-10 and Super VC-10 - A Review of the Flying and Control Characteristics," K. S. Lawson, Vickers-Armstrongs, Ltd., August 1960.

"Variable Delivery Closed Circuit Packaged Power Control Unit (Proposal)," Boulton Paul Acft., Ltd, November 1968.

"Evaluation Tests on Boulton Paul VC-10 Aileron Integrated Flight Control Actuator," A. F. Damp, Boeing Document No. D6-25527, March 1970.

"Selected Evaluation Tests - Boulton Paul VC-10 Integrated Rudder Actuator Package," R. L. Franzen, DAC, IRAD Report No. J5402 (Prereleased).

Cadillac Controls Integrated Products - Polaris Missile (Photographs).

"Survivable Controls Gain Emphasis," M. L. Yaffee, Aviation Week and Space Technology, February 2, 1970.

"Redundant Damper Servoactuators for the F-111 Airplane," W. J. Thayer, Moog Servo-controls, Ind.

"Electro-Hydraulic Power Units," S. H. Stone, Stone Hydraulic Industries, Inc.

"Unified Flight Control Actuator Package," R. C. Wild, J. E. Rinde, General Electric Company, July 1969.

"Survivability Becomes Key Goal in Fly-by-Wire Development," Space/Aeronautics, July 1970.

"Electronic Flight Control is Getting Set to Take Off," J. P. Sutherland, R. C. Hendrick, Honeywell, Inc., Aerospace Division, November 1970.

"Integrated Actuators Reduce Vulnerability of Aircraft Controls," V. R. Schmitt, W. G. Koch, Hydraulics and Pneumatics, August 1970.

"Polaris A2 Thrust Vector Control Hydraulic System," Lockheed Aircraft Corp.

"Multiples Electrohydraulic System for Fly-by-Wire Actuators, with Majority Voting and Pressure Logic," Dr. Ing. C. R. Himmler, British Hydromechanics Research Assoc., England, January 1971.

"Navy/LTV Aerospace F-8," Aviation Week and Space Technology, May 1972.

"Development of the Cessna Model 210 Hydraulic Power Pack," L. K. Evans, Society of Automotive Engineers, Inc., March 1972.

"Fly-by-Wire Advantages Explored," P. J. Klass, Aviation Week and Space Technology, July 1972.

"Fly-by-Wire Power Servo," Montek.

"Electro-Ram," Montek.

"747 Stabilizer Control Module," Montek.

"Flight Control System Design for Supersonic Transport," M. A. Ostgaard, Aeronautical Systems Div., Wright-Patterson Air Force Base, SAE #592B, October 1962 (C109097).

"Fly-by-Wire Primary Flight Control Systems for Advanced Fighter Aircraft," F. M. Krachmalnick, McDonnell, St. Louis, AFFDL-TR-69-58 F.B.W. Conf. - December 1968, (C119,368).

"Misc. Tradeoff Data," Cadillac Controls - Division of Ex-Cell-O Corp., January 1970.

"Engineering Data Hydraulic Power Package," Lear Siegler, Inc., May 1969.

"Piper Packaged Hydraulics," G. K. Gates and W. S. Vargo, Hydraulics and Pneumatics, March 1969.

"Vickers Packaged Hydraulics," D. Connett, C. E. Liles, Vickers Corp., 1969, (C33525).

"Self-Contained Electric-Hydraulic Accessory Power Unit," R. L. McManus, General Electric Company, (C54,612).

"Electro Mechanical Actuator Advantages," (C107,797).

"Integrated Pump Packages," Vickers (Capabilities Brochure) C4E-1, May 1972.

"The Application of Simplex, Servo-Pump, Integrated Actuator Packages for Large Aircraft Flight Control Systems," Vickers (Presentation to DAC), July 1972.

"Power-by-Wire and Fly-by-Wire Flight Control," Vickers (Marketing News) MN 2D-1, May 1972.

110.52 HIGH PRESSURE HYDRAULICS

"An Investigation of the Effects of Working Pressure of Aircraft Hydraulic Systems," R. H. Walsh, Royal Aircraft Establishment Tech. Report 69159, July 1969.

"Theoretical Study of Very High Pressure Fluid Power Systems," D. W. Deamer, S. J. Brighan, NAA, Inc., Columbus Div., Naval Air Systems Comm. NOw 65-0567-d, October 1966 (C-11,446).

Bound Volume of Papers presented to SAE, A-6 Symposium, October 1967:

"Pressure Considerations in Designing a Hydraulic Systems," J. W. Lebold, L. C. Garday, Lockheed-California Co.

"Pump Selection as a Function of System Pressure," F. L. Moncher, Vickers Div.

"Supply Pressure Considerations for Servoactuators," W. J. Thayer, Moog, Inc.

"Hydraulic Transmission Line Weight as a Function of System Operating Pressure," E. T. Raymond, Boeing Co.

"A Look at Fittings, Hoses and Seals for Pressures Above 3000 psi," S. C. Maggio, USAF, WPAFB.

"Dynamic Response of Very High Pressure Fluid Power Systems - Final Report," J. E. Stauffer, NAR, Columbus Div., NR 69-H-65, April 1969.

"Very High Pressure Fluid Power Systems - A Progress Report," J. E. Stauffer, NAR, Columbus Div., Presented to SAE, A-6, 1969.

"Dynamic Response Test of Very High Pressure Fluid Power Systems," J. N. Demarchi, NAR, Columbus Div., Tech Report No. NR 70h-533, December 1970.

"Application of Very High Pressure Hydraulic System to Aircraft - A Progress Briefing," J. N. Demarchi, NAR, Columbus Div., NR72H-111, 1972.

110.53 MISCELLANEOUS SUBJECTS

"Nonlinear Compensation of the Hydraulic Servo for Valve-Orifice Nonlinearity," J. F. Suerken, USAF, Air Force Institute WPAFB, (Thesis) June 1968 (AD 838883).

"Effect of Extreme Conditions on the Behavior of Lubricants and Fluids," D. R. Wilson, Midwest Research Inst., Technical Report AFML-TR-67-8, Part III, February 1969 (AD 851480).

"High Temperature Hydraulic Fluid Erosion Evaluation with the Control Valve Assembly having DC-10 Design Features," R. L. Green, Monsanto Company, R-AAP-1071-49, Testing from October 1970 to October 1971.**

"Identification and Control of Cavitation Damage in Aircraft Hydraulic Valves," Monsanto Company, Organic Division Report #3366, September 1966.**

"Helium Pressurization Systems for Liquid-Methane Fuel in Supersonic Transport," J. D. Eisenbery, Lewis Research Center, NASA TND-5519, October 1969.

"Some Factors Affecting the Stress-Corrosion Cracking of TI-6AL-4V Alloy in Methanol," W. B. Lisagor, Langley Research Center, NASA TND-5557, December 1969.

"Powered Flying Controls for the Harrier (Brochure)," Fairey Hydraulics, Ltd., August 1970.

"Surface Actuators for the Jaguar (Brochure)," Fairey Hydraulics, Ltd., January 1968.

"Measurement of Dynamic Flow of Hydraulic Fluid," J. R. Patten, Boeing Document No. DG-23975TN, October 1968.

"Determination of Wear Distributions," General Electric Company for AFSC. RADC-TR-69-225, July 1969 (AD 857520).

"Applications of Extreme Value Theory in the Reliability Analysis of Non-Electronic Components (M.S. Thesis)," C. B. Kubla, Air University, USAF Institute of Tech., 19 .

"New Approaches to Fluid Power Generation and Transmission for Mid-1970's Transport Aircraft," R. L. Franzen, DAC, IRAD Report No. DAC 67617. (Prereleased).

"Fluidics," G. W. Kibbee, DAC, 1967 (Printed material to supplement in-plant course).*

"Fluid Power Abstracts" Worldwide coverage of Hydraulics and Pneumatics Literature. The British Hydromechanics Research Association Cranfield, Bedford, England, (Separately bound monthly editions from January 1970 through December 1971).

The following subjects are contained in a loose-leaf volume. Each aircraft and system listed is described in various articles, etc., with principal emphasis on the hydraulic and control systems:

F-27 (All Pneumatic controls)

B747

L-1011

C-5A

Trident 3B

VC-10 (Includes totally integrated hydraulic packages)

Concorde

U.S. SST

A.C. (Nonflow) Hydraulics

Hydraulic Pumps

Pneumatics

Auxiliary and Emergency Power Supplies

Very High Pressure Hydraulics (See also "High Pressure Hydraulics" Listings)

Advanced Concepts

The following are related to "Flywheel" energy Storage:

"Solar Cell - Flywheel Energy Storage Power System," R. Gorman, Bellcom, Inc., September 1970 (PDL 109579, SL-1).

"Performance of Hybrid Flywheel Vehicles," C. A. Brown, United Aircraft, May 1971 (AIAA Paper No. 71-509).

"High Speed Energy Wheel Offers Trolleys Portable Electricity," Product Engineering, July 20, 1970.

"Flywheel Energy Storer Cuts Aircraft Weight," Machine Design, June 20, 1968.

"Convection Heat Exchange in A Closed Space Between Two Rotating Disks Under Turbulent Conditions Flow," V. M. Kapinos (Russian Translation), April 1965 (N68-33264).

"Application of Isotensoid Flywheels to Spacecraft Energy and Angular Momentum Storage," L. R. Adams, Astro Research Corp., February 1972, NASA CR-1971.

"Electric Motor-Engine Hybrids Get Nod from Engineers and Ecologist," Product Engineering, November 1971.

"Elements and Drives of Power Follow-Up Systems," Moscow Aviation Institute, 1961 (AD 646932), Collection of Servomechanism Articles: Theory of the Calculation of Combined Control Systems; Static and Dynamic Characteristics of Hydraulic Amplifier; Pumps and Valves, Hydraulic Motor Characteristics; Characteristic of Electro Mechanical Converters with A Magnetic Spring" Selecting the Ratio of Power Reducers for Follow-Up Drives; Control Characteristics of the Reversible Magnetic Amplifier Working on a Complex Load; Electron Tubes; Calculation of Bellows Strength with Large Pressure Drops.

"Hydraulic Elements in Control Systems," I. M. Drassov, (Translated Russian Text: Hydraulic Equipment, Hydraulic Pressure Amplifier, Hydraulic Pump, Hydraulic Fluid, Hydraulic Computer Component, Amplifier Stage, Time Relay, Automatic Control System) 1967 (AD 692632).

"Hydraulic Drives of Aircraft Volume I," T. M. Bashta, 1967 (AD700169) (730 pages in 2 Volumes devoted to theoretical fundamentals and designs of Hydraulic Units used on Civil Aviation Aircraft).

"Hydraulic Drives of Aircraft Volume II," (AD 700167) (see above).

*Additional 50+ "Fluidic" references are available from R. Franzen.

**Other articles on "Cavitation" contained in separate bibliography, available from R. Franzen.

120. REFERENCE DATA

CONTENTS

120.00 INTRODUCTION — SPECIFICATIONS The following list of specifications for hydraulic design and test is not all inclusive. Specifications listed can be used as starting points for locating or writing more specific requirements.

121.00 MILITARY SPECIFICATIONS

121.01 GENERAL SPECIFICATIONS

MIL-A 8860	Airplane Strength and Rigidity, General Specification
MIL-E-5272	Environmental Testing
MIL-F 9490	Flight Control Systems, Design, Installation and Test of, General Specifications
MIL-H-5440	Hydraulic Systems, Design, and Tests
MIL-P-5514	Packings; Installation and Gland Design, General Specifications
MIL-H-8775	Hydraulic Components, General Specifications
MIL-H-8891	Type III Hydraulic Systems, -65°F to +750°F
MIL-P-5518	Pneumatic Systems, Design, Installation and Tests
MIL-P-8564	Pneumatic Components, General Specifications
MIL-HDBK-5	Strength of Aircraft Metal Elements
MIL-S-8812	Steering System, General Requirements for
MIL-T-5522	Test Procedure, Hydraulic and Pneumatic Systems
SD-24 Vol. 1	General Specification for Design and Construction of Aircraft Weapons System (Naval Air Systems Command)
MIL-Std-17	Military Standard Mechanical Symbols.

121.02 PUMPS AND MOTORS

MIL-M-7997	Motor, Constant Displacement, Hydraulic
MIL-P-5515	Hand Pump, Hydraulic
MIL-P-5954	Pump and Electric Motor, Fixed Displacement
MIL-P-5994	Pump, Electric Driven, Fixed Displacement
MIL-P-7858	Pump, Power Driven, Fixed Displacement
MIL-P-19692	Pump, Variable Delivery

121.03 PIPING, FITTINGS, AND HOSE

MIL-C-25427 Coupling, Hydraulic Quick-Disconnect

MIL-F-5070 Fitting Ends; Hose

MIL-F-5506 Fitting, Method of Testing

MIL-F-5509 Fitting, Flared Tube

MIL-F-8789 Fitting End, High Pressure Hydraulic and Pneumatic

MIL-F-18280 Fitting, Flareless

MIL-H-5593 Hose, Aircraft, Low Pressure

MIL-H-8794 Hose, Rubber, Aircraft

MIL-H-8795 Hose Assemblies, Rubber, Aircraft

MIL-H-8788 Hose, Hydraulic and Pneumatic, High Pressure

MIL-P-5514 Pipe Threads, Taper, ANF, ANPT

MIL-S-5513 Swivel Joints, Hydraulic

MIL-S-7742 Screw Threads, Standard, Aeronautical

MIL-T-6845 Tubing, Steel, Corrosion Resistant (Type 304) Annealed, Aircraft Hydraulic, Seamless

MIL-T-7081 Tubing, Aluminum Alloy 6061 and 6062 Seamless, Round, Aircraft Hydraulic Quality.

MIL-T-8504 Tubing, Steel, Corrosion Resistant (1B-8) Annealed Aircraft Hydraulic System

MIL-T-8790 Hose Assembly, High Pressure Hydraulic and Pneumatic

MIL-T-8808 Tube, Corrosion Resistant, Type 321.

121.04 HYDRAULIC COMPONENTS

MIL-A-5498 Accumulators, Aircraft, 275°F

MIL-A-8897 Accumulators, Hydraulic, 3000 psi

MIL-C-5503 Cylinders, Aircraft, Hydraulic Actuating

MIL-F-5504 Filters, Hydraulic, Aircraft

MIL-F-5508 Fuses, Aircraft Automatic Quantity Measuring, Hydraulic

MIL-F-8815 Filter and Element, 15-micron Type II System

MIL-G-7734 Gages, Pressure, Aircraft

MIL-I-7084 Indicators, Pressure, Hydraulic 0-4000 psi

MIL-R-5520 Reservoirs, Hydraulic

MIL-S-5049 Scrapers, Hydraulic Piston Rod

121.05 HYDRAULIC VALVES

MIL-V-5519 Valves, Hydraulic Unloading, Aircraft

MIL-V-5523 Valves, Relief Pressure, Hydraulic

MIL-V-5574 Valves, Check, Type I Systems, Aircraft

MIL-V-5525 Valves, Power Brake, Aircraft

MIL-V-5527 Valves, Thermal Expansion Relief, Aircraft

MIL-V-5528 Valves, Hydraulic Controllable Check

MIL-V-5529 Valves, Hydraulic Directional Control

MIL-V-5530 Valves, Hydraulic Shuttle, Aircraft

MIL-V-7915 Valves, Hydraulic, Direction Control, Slide Selector

MIL-V-8566 Valve, Flow Regulator

MIL-V-8813 Relief Valve, Type II, Hydraulic System

MIL-V-19067 Controllable Check Valve, Type II Hydraulic System

MIL-V-19068 Shuttle Valve, Type II Hydraulic System

MIL-V-19069 Check Valve, Type II Hydraulic System

MIL-V-25517 Restrictor Valve, Hydraulic

MIL-V-25675 Check Valve, Miniature

121.06 WHEELS AND BRAKES

MIL-B-8584 Brake Systems, Wheel Design

MIL-C-5041 Casings, Tire and Tubeless Tires, Aircraft Pneumatic

MIL-C-5499 Cores, Aircraft High Pressure Air Valve

MIL-C-5500 Caps, High Pressure Air Valve

MIL-C-6026 Brake Control Unit, Pressure Generating, Manual

MIL-C-7726 Casings and Inner Tubes, Repair and Recapping of Used Aircraft

MIL-C-7979 Brake Cylinder, Master Power Assisted

MIL-T-5014 Tubes, Inner, Aircraft Pneumatic Tire

MIL-W-5013 Wheel and Brake Assemblies, Aircraft

121.07 SEALS AND GASKETS

MIL-G-5510 Gasket; Straight Thread Tube Fitting, Boss

MIL-P-5516 Packing and Gaskets; Hydraulic, Aircraft

MIL-P-25732 Packing, Preformed, 275°F, Hydraulic

MIL-R-5521 Rings, Aircraft Hydraulic Packing Back-up

MIL-R-8791 Teflon Ring, Hydraulic and Pneumatic Packing

121.08 PNEUMATIC COMPONENTS

MIL-C-7905 Cylinder, Non-Shatterable, Compressed Gas

MIL-C-8838 Cylinder, Pneumatic Actuating

MIL-L-4343 Lubricating Grease, Pneumatic

MIL-R-8572 Pressure Reducer, Pneumatic

MIL-R-8573 Reservoir, Air, Non-shatterable

MIL-V-6164 Filler Valve, Air, High Pressure

MIL-V-19593 Control Valve, Solenoid Operated Pneumatic

121.09 LANDING GEAR

MIL-S-8552 Strut, Landing Gear, Shock Absorber

MIL-T-6053 Tests- Shock Absorber Drop

121.10 HYDRAULIC FLUIDS

MIL-H-5606 Hydraulic Fluid, Petroleum Base, Aircraft

MIL-H-6083 Hydraulic Fluid, Petroleum Base, Preservative

MIL-H-83282 Hydraulic Fluid, Fire Resistant, Synthetic Hydrocarbon Base, Aircraft

VV-P-236 Petrolatum

121.11 MISCELLANEOUS

MIL-C-18805 Catapulting Landplanes

MIL-D-18717 Arresting Hook Installation

MIL-C-6641 Gear Box, Accessory Drive

MIL-G-7743	Pressure Gage
ANA 438	Bulletin: Age Control of Age Sensitive Elastomeric Items

122.00 FAA SPECIFICATIONS

FAR 25	Airworthiness Standards, Transport Category, Aircraft

123.00 INDUSTRY SPECIFICATIONS

123.01 GENERAL SPECIFICATIONS

AIR 737B	Hydraulic and Pneumatic Specifications and Standards, Aerospace
AIR 1082	Fluid System Component Specification Criteria
AIR 1083	Hydraulic System Survivability
ARP 24B	Hydraulic Pressure Drop Determination
ARP 243B	Hydraulic and Pneumatic Systems, Nomenclature for Aircraft
ARP 578	Detail Requirements, 4000 and 5000 psi Hydraulic Systems
ARP 819	Fluid System Characteristics Affecting Pump Operation

123.02 PUMPS AND MOTORS

AS 595A	Hydraulic Pump, Civil Aircraft, Variable Delivery
AS 696	Hydraulic Motor, Aircraft, Constant Displacement

123.03 PIPING, FITTING AND HOSE

ARP 683	Fluid Connections, Installation and Torque
AIR 1047A	Couplings, Quick Disconnect, Aerospace Fluid Systems

123.04 HYDRAULIC COMPONENTS

AIR 887	Filters, Liquid, Rating of
ARP 599	Filter, Test Downstream for Cleanliness
AS 714	Hydraulic Starters, Aircraft

123.05 HYDRAULIC VALVES

ARP 490B	Servovalves, Electrohydraulic Flow Control
ARP 699C	Electrohydraulic Flow Control Servovalves

123.06 WHEELS AND BRAKES

ARP 597A	Wheels and Brakes, Supplementary Criteria for Design Endurance, Civil Transport.
ARP 813	Wheels and Brakes, Maintainability for Aircraft
AS 227D	Wheels and Brakes, Minimum Requirements for Civil Aircraft

123.07 SEALS AND GASKETS

SAE J929	Piston Rings
AIR 1077	Metallic Seal Rings for High Temperature Reciprocating Hydraulic Service
ARP 820	O-Ring Packing, 3000 psi Hydraulic Service, Physical Performance Test for Type II

123.08 PNEUMATIC COMPONENTS

AIR 944	Starting Aircraft, Pneumatic Power for
AS 513	Air Valves, Qualification Test for Aircraft

123.09 HYDRAULIC FLUIDS

AIR 81	Hydraulic Fluid Characteristics
ARP 598A	Hydraulic Fluids, Procedure for Determination of Contamination by Particulate Count Method
ARP 785	Contamination, Particulate Determination in Hydraulic Fluid by Control Filter Gravimetric Procedures

123.10 MATERIALS

SAE J471	Sintered Metal Powder

124.00 DOUGLAS SPECIFICATIONS

124.01 GENERAL

DPS 3.391	Hydraulic, Skydrol, System for Aircraft
DPS 30.59	Manufacturing Requirements DC-9
DPS 30.101	Douglas Parts and Assemblies

124.02 PIPING, FITTING, AND HOSE

DPS 3.22	Fluid Lines, Aircraft, Identification
DPS 10.914	Brazing, Automatic, Tube Joints

DMS 1563 Tube, Corrosion Resistant, Type 321

DMS 1578 Tube, Corrosion Resistant, Type 304, Seamless

DMS 1864 Nylon Tubing

DMS 1897 Tube, Titanium

DMS 1900 Tube, Corrosion Resistant, Type 321

124.03 HYDRAULIC COMPONENTS

DPS 3.334 Preservation, Hydraulic Assembly

DPS 3.335 Hydraulic System Flushing

DPS 30.101 Douglas Hydraulic Units, Vendor Built

DPS 30.106 Hydraulic Units, Vendor Designed

DPS 30.110 Hydraulic Units, Cleanliness Requirements

124.04 SEALS AND GASKETS

DPS 2.50 Sealing Methods

DPS 3.576 Age Control - Synthetic Rubber

DPS 3.579 Molded O-Rings, Gaskets, Packing and Back-Up Rings

S4929668 Ring, Anti-Extrusion Packing Back-Up

124.05 LANDING GEAR

DPS 30.502.1 Landing Gear, Magnetic Inspection and Rework

DPS 30.503 Landing Gear, Manufacturing and Inspection Procedure

124.06 HYDRAULIC FLUIDS

DMS 2014 Fire Resistant Hydraulic Fluid

DPS 1.151-2 Fluid Requirements, Lubricant and Hydraulic

DPS 3.332 Fluid Testing and Preserving, Aircraft Systems

DPS 3.339 Fluid, Properties and Cleanliness Levels (Test Methods)

DPS 3.391 Skydrol Hydraulic Systems for Airplanes

124.07 MATERIALS

DPS 1.33 Bushing Installation

DPS 3.17	Lubricants
DPS 3.173	Anti-Friction Bearing
DPS 3.67	Fasteners
DPS 3.69	Forging, Aluminum
DPS 8.25	Staking Bearings

124.08 <u>MISCELLANEOUS</u>

DPS 2.401	Machine Finish, Roughness Control
DPS 3.53	Forging Identification
DPS 5.00	Heat Treatment
DPS 6.33	Heat Treatment - Titanium
DPS 11.01	Anodizing, Chromic Acid
DPS 11.04	Anodizing, Aluminum
DPS 11.07	Anodizing, Brush
WZZ7000	Electromagnetic Interference
WZZ7001	Bonding, Electrical
WZZ7002	Aircraft Wiring

125.00 DESIGN AIDS

125.01 FUNCTIONS OF DIAMETERS

D	D^2	$D^{1/2}$	D^3	$D^{1/3}$	D^4	$\pi D^2/4$	$\pi D^4/64$	BENDING MOMENT inch lbs. 125,000 PSI	BENDING MOMENT inch lbs. 160,000 PSI
1/32	.000976	.1768	.000031	.31488	.000001	.00077	.0000000465	.372	.476
1/16	.003906	.2500	.000244	.39685	.000015	.00307	.000000749	3.00	3.84
3/32	.008789	.3062	.000824	.45428	.000077	.00690	.00000379	10.10	12.93
1/8	.015625	.3536	.001953	.50000	.000244	.01227	.00001198	24.0	30.73
5/32	.024414	.3953	.003815	.53861	.000596	.01917	.00002926	46.8	59.90
3/16	.035156	.4330	.006592	.57236	.001236	.02761	.00006066	80.9	103.50
7/32	.047852	.4677	.010468	.60254	.002290	.03758	.0001125	128.5	164.5
1/4	.062500	.5000	.015625	.62996	.003906	.04909	.0001918	191.8	245.3
9/32	.079102	.5303	.022247	.65519	.006257	.06213	.000307	273.0	349.3
5/16	.097656	.5590	.030518	.67860	.009537	.07670	.000468	374.4	479.0
11/32	.118164	.5863	.040619	.70051	.013963	.09281	.0006875	500.	640.
3/8	.140625	.6124	.052734	.72113	.019776	.11045	.0009710	647.4	829.
13/32	.165039	.6374	.064047	.74062	.027238	.12962	.001335	822.0	1053
7/16	.191406	.6614	.083740	.75015	.036636	.15033	.001797	1027	1315
15/32	.219727	.6847	.102997	.776808	.048280	.17257	.00237	1264	1617
1/2	.250000	.7071	.125000	.79370	.062500	.19635	.003069	1535	1965
17/32	.282227	.7289	.149933	.80990	.079652	.22166	.00391	1840	2353
9/16	.316406	.7500	.177978	.825482	.100113	.24851	.004908	2196	2810
19/32	.354320	.7706	.210378	.84049	.125543	.27828	.005985	2520	3226
5/8	.390625	.7906	.244141	.85499	.152588	.30680	.007492	2996	3833
21/32	.430664	.8101	.282623	.86901	.185472	.33824	.0091	3465	4435
11/16	.472656	.8292	.324951	.88259	.223404	.37122	.01095	3980	5090
23/32	.516602	.8478	.371308	.89576	.266878	.40574	.0131	4560	5840
3/4	.562500	.8660	.421875	.90856	.316406	.44179	.01553	5176	6625
25/32	.610352	.8839	.476838	.92101	.37253	.47937	.0183	5850	7485
13/16	.660156	.9014	.536377	.933128	.435806	.51849	.0214	6580	8420
27/32	.711914	.9187	.600677	.944941	.506822	.55914	.0249	7375	9440
7/8	.765625	.9354	.669922	.95647	.586182	.60132	.02878	8223	10530
29/32	.821289	.9520	.744293	.96772	.674516	.64504	.0331	9135	11700
15/16	.878906	.9682	.823974	.97872	.772476	.69029	.0379	10100	12930
31/32	.938477	.9843	.909150	.98947	.880739	.73708	.04325	11150	14270

125.01 FUNCTIONS OF DIAMETERS (Continued)

D	D^2	$D^{1/2}$	D^3	$D^{1/3}$	D^4	$\pi D^2/4$	$\pi D^4/64$	BENDING MOMENT inch lbs. 125000 PSI	BENDING MOMENT inch lbs. 160000 PSI
1	1.00000	1.0000	1.00000	1.00000	1.00000	.78540	.04908	12260	15700
1- 1/32	1.06348	1.0155	1.09680	1.01031	1.13099	.83526	.0555	13460	17240
1- 1/16	1.12891	1.0308	1.1995	1.0204	1.2744	.88665	.0626	14740	18860
1- 3/32	1.19629	1.0458	1.3084	1.0303	1.4311	.93957	.07025	16060	20540
1- 1/8	1.26562	1.0606	1.4238	1.0400	1.6018	.99402	.0786	17460	22330
1- 5/32	1.33691	1.0753	1.5458	1.0496	1.7873	1.0500	.08775	18970	24300
1- 3/16	1.41016	1.0897	1.6746	1.0590	1.9886	1.10754	.0976	20520	26250
1- 7/32	1.48535	1.1040	1.8103	1.0682	2.2063	1.16659	.1083	22200	28400
1- 1/4	1.56250	1.1180	1.9531	1.0772	2.4414	1.22719	.1194	23900	30600
1- 9/32	1.64160	1.1319	2.1033	1.0861	2.6949	1.28931	.1324	25800	33000
1- 5/16	1.72266	1.1456	2.2610	1.0949	2.9676	1.35298	.1455	27700	35460
1-11/32	1.80566	1.1592	2.4264	1.1035	3.2604	1.41817	.1601	29800	38120
1- 3/8	1.89062	1.1726	2.5996	1.1081	3.5744	1.48489	.1755	31900	40800
1-13/32	1.97754	1.1845	2.7808	1.1195	3.9107	1.55316	.192	24170	43750
1- 7/16	2.06641	1.1990	2.9705	1.1286	4.2701	1.62296	.2095	36400	46600
1-15/32	2.15723	1.2119	3.1689	1.1367	4.6536	1.69429	.2287	39000	49900
1- 1/2	2.25000	1.2247	3.3750	1.1447	5.0625	1.76715	.2485	41400	53000
1-17/32	2.34473	1.2374	3.5904	1.1526	5.4977	1.84155	.270	44150	56500
1- 9/16	2.44141	1.2500	3.8147	1.1604	5.9605	1.91748	.2926	46760	59850
1-19/32	2.54004	1.2624	4.0482	1.1681	6.4516	1.99495	.317	49760	63700
1- 5/8	2.64063	1.2748	4.2910	1.1757	6.9729	2.07395	.343	52750	67500
1-21/32	2.74316	1.2870	4.5434	1.1832	7.5250	2.15448	.3693	55750	71350
1-11/16	2.84766	1.2990	4.8054	1.1906	8.1092	2.23655	.398	59000	75500
1-23/32	2.95410	1.31101	5.0774	1.1979	8.7267	2.32015	.428	62300	79750
1- 3/4	3.06250	1.3229	5.3594	1.2051	9.3789	2.40529	.460	65750	84200
1-25/32	3.1729	1.3346	5.6516	1.2122	10.0693	2.4920	.494	69400	88800
1-13/16	3.2852	1.3463	5.9544	1.2193	10.7923	2.5802	.530	73150	93650
1-27/32	3.3994	1.3579	6.2677	1.2262	11.5560	2.6699	.567	76800	98350
1- 7/8	3.5156	1.3693	6.5918	1.2331	12.3596	2.7612	.607	81000	103600
1-29/32	3.6338	1.3807	6.9269	1.2399	13.2044	2.8540	.648	85000	108750
1-15/16	3.7539	1.3919	7.2732	1.2467	14.0918	2.9483	.6915	89200	114100
1-31/32	3.8760	1.4031	7.6308	1.2533	15.0233	3.0442	.737	93500	119600
2	4.0000	1.4142	8.0000	1.2598	16.0000	3.1416	.7854	98000	125400

125.01 FUNCTIONS OF DIAMETERS (Continued)

D	D^2	$D^{1/2}$	D^3	$D^{1/3}$	D^4	$\pi D^2/4$
2-1/16	4.2539	1.4361	8.774	1.2729	18.096	3.341
2-1/8	4.5156	1.4577	9.596	1.2856	20.391	3.547
2-3/16	4.7852	1.4790	10.468	1.2981	22.898	3.758
2-1/4	5.0625	1.5000	11.391	1.3104	25.629	3.976
2-5/16	5.3477	1.5207	12.367	1.3224	28.598	4.200
2-3/8	5.6406	1.5411	13.397	1.3342	31.816	4.430
2-7/16	5.9414	1.5613	14.482	1.3458	35.300	4.666
2-1/2	6.2500	1.5811	15.625	1.3572	39.063	4.909
2-9/16	6.5664	1.6007	16.826	1.3684	43.118	5.157
2-5/8	6.8906	1.6202	18.088	1.3794	47.481	5.412
2-11/16	7.1985	1.6394	19.346	1.3903	51.818	5.654
2-3/4	7.5625	1.6593	20.797	1.4010	57.191	5.940
2-13/16	7.9102	1.6771	22.247	1.4116	62.571	6.213
2-7/8	8.2656	1.6956	23.764	1.4219	68.321	6.492
2-15/16	8.6289	1.7139	25.347	1.4322	74.458	6.777
3	9.0000	1.7321	27.000	1.4413	81.000	7.069
3-1/16	9.3789	1.7500	28.723	1.4522	87.964	7.366
3-1/8	9.7656	1.7678	30.518	1.4620	95.368	7.670
3-3/16	10.1602	1.7854	32.386	1.4717	103.229	7.980
3-1/4	10.5625	1.8028	34.328	1.4813	111.566	8.296
3-5/16	10.9727	1.8200	36.347	1.4907	120.399	8.618
3-3/8	11.3906	1.8371	38.443	1.5000	129.747	8.946
3-7/16	11.8164	1.8541	40.619	1.5092	139.628	9.281
3-1/2	12.2500	1.8708	42.875	1.5183	150.063	9.621
3-9/16	12.6914	1.8875	45.213	1.5273	161.072	9.968
3-5/8	13.1406	1.9039	47.635	1.5362	172.676	10.321
3-11/16	13.5977	1.9203	50.141	1.5449	184.896	10.680
3-3/4	14.0625	1.9365	52.734	1.5536	197.754	11.045
3-13/16	14.5352	1.9526	55.415	1.5622	211.271	11.416
3-7/8	15.0156	1.9685	58.156	1.5707	225.467	11.793
3-15/16	15.5039	1.9843	61.047	1.5791	240.371	12.177
4	16.0000	2.0000	64.000	1.5874	256.000	12.566
4-1/8	17.0156	2.0310	70.190	1.6001	289.53	13.364
4-1/4	18.0625	2.0616	76.766	1.6198	326.254	14.186
4-3/8	19.1406	2.0917	83.740	1.6355	366.364	15.033
4-1/2	20.2500	2.1213	91.125	1.6509	410.063	15.904
4-5/8	21.3906	2.1506	98.932	1.6661	457.56	16.800
4-3/4	22.5625	2.1795	107.172	1.6810	509.066	17.721
4-7/8	23.7656	2.2079	115.858	1.6956	564.805	18.666
5	25.0000	2.2361	125.000	1.7100	625.000	19.635
5-1/8	26.2656	2.2639	134.611	1.7241	689.883	20.629
5-1/4	27.5625	2.2913	144.703	1.7380	759.691	21.658
5-3/8	28.8906	2.3184	155.287	1.7517	834.669	22.691
5-1/2	30.2500	2.3452	166.375	1.7656	915.063	23.758
5-5/8	31.6406	2.3717	177.979	1.7785	1001.130	24.851
5-3/4	33.0625	2.3979	190.109	1.7915	1093.130	25.967
5-7/8	34.5156	2.4238	202.779	1.8044	1191.330	27.109
6	36.0000	2.4495	216.000	1.8171	1296.000	28.274

125.01 FUNCTIONS OF DIAMETERS (Continued)

D	D^2	$D^{1/2}$	D^3	$D^{1/3}$	D^4	$\pi D^2/4$
6- 1/8	37.516	2.4749	229.78	1.8297	1407.4	29.465
6- 1/4	39.063	2.5000	244.14	1.8420	1525.9	30.680
6- 3/8	40.640	2.5249	259.28	1.8542	1651.7	31.919
6- 1/2	42.250	2.5495	274.63	1.8663	1785.1	33.183
6- 5/8	43.891	2.5739	290.78	1.8781	1926.4	34.472
6- 3/4	45.563	2.5981	307.55	1.8899	2075.9	35.785
6- 7/8	47.266	2.6220	324.95	1.9015	2234.0	37.122
7	49.000	2.6458	343.00	1.9130	2401.0	38.485
7- 1/8	50.766	2.6693	361.71	1.9243	2577.1	39.871
7- 1/4	52.563	2.6926	381.08	1.9354	2762.8	41.383
7- 3/8	54.391	2.7157	401.13	1.9465	2958.3	42.718
7- 1/2	56.250	2.7386	421.87	1.9574	3164.1	44.179
7- 5/8	58.141	2.7613	443.32	1.9683	3380.3	45.664
7- 3/4	60.063	2.7839	465.48	1.9780	3607.5	47.173
7- 7/8	62.016	2.8062	488.37	1.9895	3845.9	48.707
8	64.000	2.8284	512.00	2.0000	4096.0	50.266
8- 1/8	66.016	2.8504	536.38	2.0104	4358.1	51.849
8- 1/4	68.063	2.8723	561.52	2.0206	4632.5	53.456
8- 3/8	70.141	2.8940	587.43	2.0308	4919.7	55.088
8- 1/2	72.250	2.9155	614.13	2.0408	5220.6	56.745
8- 5/8	74.391	2.9368	641.62	2.0508	5534.0	58.426
8- 3/4	76.563	2.9580	669.92	2.0606	5861.8	60.132
8- 7/8	78.766	2.9791	699.05	2.0704	6204.0	61.863
9	81.000	3.0000	729.00	2.0801	6561.0	63.617
9- 1/8	83.266	3.0208	759.80	2.0897	6933.2	65.397
9- 1/4	85.563	3.0414	791.45	2.0992	7320.9	67.201
9- 3/8	87.891	3.0619	823.97	2.1086	7724.8	69.029
9- 1/2	90.250	3.0822	857.38	2.1179	8145.1	70.882
9- 5/8	92.641	3.1024	891.67	2.1272	8582.3	72.760
9- 3/4	95.063	3.1225	926.86	2.1363	9036.9	74.662
9- 7/8	97.516	3.1425	962.97	2.1454	9509.3	76.589
10	100.00	3.1623	1000.	2.1544	10000.0	78.540
10- 1/8	102.52	3.1820	1038.0	2.1634	10509.5	80.516
10- 1/4	105.06	3.2016	1076.9	2.1722	11038.1	82.516
10- 3/8	107.64	3.2211	1116.77	2.1810	11586.5	84.541
10- 1/2	110.25	3.2404	1157.63	2.1898	12155.1	86.590
10- 5/8	112.89	3.2596	1199.5	2.1984	12744.3	88.664
10- 3/4	115.56	3.2787	1242.3	2.2070	13354.7	90.763
10- 7/8	118.27	3.2977	1286.1	2.2155	13986.8	92.886
11	121.00	3.3166	1331.0	2.2240	14641.0	95.033
11- 1/8	123.77	3.3354	1376.9	2.2324	15317.9	97.206
11- 1/4	126.56	3.3541	1423.8	2.2407	16018.1	99.402
11- 3/8	129.39	3.3727	1471.8	2.2490	16741.9	101.62
11- 1/2	132.25	3.3912	1520.9	2.2572	17490.1	103.87
11- 5/8	135.14	3.4096	1571.01	2.2653	18263.0	106.14
11- 3/4	138.06	3.4278	1622.2	2.2734	19061.3	108.43
11- 7/8	141.02	3.4460	1674.6	2.2815	19885.4	110.75
12	144.00	3.4641	1728.00	2.2894	20736.0	113.10

125.01 FUNCTIONS OF DIAMETERS (Continued)

DIA.	AREA	I	DIA.	AREA	I	DIA.	AREA	I
1.00	0.785	.0491	1.50	1.767	.2485	2.00	3.1416	.7854
1.01	0.801	.0510	1.51	1.791	.2552	2.01	3.173	.8012
1.02	0.817	.0531	1.52	1.815	.2620	2.02	3.205	.8172
1.03	0.833	.0552	1.53	1.839	.2689	2.03	3.237	.8335
1.04	0.849	.0574	1.54	1.863	.2760	2.04	3.269	.8501
1.05	0.866	.0596	1.55	1.887	.2833	2.05	3.301	.8669
1.06	0.882	.0619	1.56	1.911	.2907	2.06	3.333	.8839
1.07	0.899	.0643	1.57	1.936	.2982	2.07	3.365	.9012
1.08	0.916	.0667	1.58	1.961	.3059	2.08	3.398	.9188
1.09	0.933	.0692	1.59	1.986	.3137	2.09	3.431	.9366
1.10	0.950	.0718	1.60	2.011	.3217	2.10	3.464	.9547
1.11	0.968	.0745	1.61	2.036	.3298	2.11	3.497	.9729
1.12	0.985	.0772	1.62	2.061	.3380	2.12	3.530	.9915
1.13	1.003	.0800	1.63	2.087	.3465	2.13	3.563	1.0103
1.14	1.021	.0829	1.64	2.112	.3550	2.14	3.597	1.0295
1.15	1.039	.0860	1.65	2.138	.3638	2.15	3.631	1.0488
1.16	1.057	.0888	1.66	2.164	.3727	2.16	3.664	1.0685
1.17	1.075	.0919	1.67	2.190	.3818	2.17	3.698	1.0884
1.18	1.094	.0951	1.68	2.217	.3910	2.18	3.733	1.1086
1.19	1.112	.0984	1.69	2.243	.4004	2.19	3.767	1.1291
1.20	1.131	.1018	1.70	2.270	.4100	2.20	3.801	1.1499
1.21	1.150	.1052	1.71	2.297	.4197	2.21	3.836	1.1709
1.22	1.169	.1087	1.72	2.324	.4296	2.22	3.871	1.1923
1.23	1.188	.1123	1.73	2.351	.4397	2.23	3.906	1.2139
1.24	1.208	.1160	1.74	2.378	.4499	2.24	3.941	1.2358
1.25	1.227	.1198	1.75	2.405	.4603	2.25	3.976	1.2580
1.26	1.247	.1237	1.76	2.433	.4710	2.26	4.011	1.2806
1.27	1.267	.1277	1.77	2.461	.4818	2.27	4.047	1.3034
1.28	1.287	.1317	1.78	2.488	.4927	2.28	4.083	1.3265
1.29	1.307	.1359	1.79	2.516	.5039	2.29	4.119	1.3499
1.30	1.327	.1402	1.80	2.545	.5153	2.30	4.155	1.3737
1.31	1.348	.1445	1.81	2.573	.5268	2.31	4.191	1.3977
1.32	1.368	.1490	1.82	2.602	.5385	2.32	4.227	1.4234
1.33	1.389	.1535	1.83	2.630	.5505	2.33	4.264	1.4468
1.34	1.410	.1582	1.84	2.659	.5626	2.34	4.301	1.4718
1.35	1.431	.1630	1.85	2.688	.5749	2.35	4.337	1.4971
1.36	1.453	.1679	1.86	2.717	.5875	2.36	4.374	1.5227
1.37	1.474	.1729	1.87	2.746	.6002	2.37	4.412	1.5487
1.38	1.496	.1780	1.88	2.776	.6132	2.38	4.449	1.5750
1.39	1.517	.1832	1.89	2.806	.6263	2.39	4.486	1.6016
1.40	1.539	.1886	1.90	2.835	.6397	2.40	4.524	1.6286
1.41	1.561	.1940	1.91	2.865	.6532	2.41	4.562	1.6559
1.42	1.584	.1995	1.92	2.895	.6670	2.42	4.600	1.6836
1.43	1.606	.2052	1.93	2.926	.6810	2.43	4.638	1.7116
1.44	1.629	.2110	1.94	2.956	.6953	2.44	4.676	1.7399
1.45	1.651	.2170	1.95	2.986	.7097	2.45	4.714	1.7686
1.46	1.674	.2230	1.96	3.017	.7244	2.46	4.753	1.7977
1.47	1.697	.2292	1.97	3.048	.7393	2.47	4.792	1.8271
1.48	1.720	.2355	1.98	3.079	.7544	2.48	4.831	1.8526
1.49	1.744	.2419	1.99	3.110	.7698	2.49	4.870	1.8870

125.01 FUNCTIONS OF DIAMETERS (Continued)

DIA.	AREA	I	DIA.	AREA	I	DIA.	AREA	I
2.50	4.909	1.9175	3.00	7.069	3.9761	3.50	9.621	7.3662
2.51	4.948	1.9483	3.01	7.116	4.0293	3.51	9.676	7.4507
2.52	4.988	1.9796	3.02	7.163	4.0831	3.52	9.731	7.5360
2.53	5.027	2.0112	3.03	7.211	4.1375	3.53	9.787	7.6220
2.54	5.067	2.0431	3.04	7.258	4.1924	3.54	9.842	7.7087
2.55	5.107	2.0755	3.05	7.306	4.2478	3.55	9.898	7.7962
2.56	5.147	2.1083	3.06	7.354	4.3038	3.56	9.954	7.8845
2.57	5.187	2.1414	3.07	7.402	4.3604	3.57	10.01	7.9734
2.58	5.228	2.1749	3.08	7.451	4.4175	3.58	10.07	8.0631
2.59	5.269	2.2088	3.09	7.499	4.4751	3.59	10.12	8.1536
2.60	5.309	2.2432	3.10	7.548	4.5333	3.60	10.18	8.2448
2.61	5.350	2.2779	3.11	7.596	4.5921	3.61	10.24	8.3367
2.62	5.391	2.3130	3.12	7.645	4.6514	3.62	10.29	8.4296
2.63	5.433	2.3485	3.13	7.694	4.7113	3.63	10.35	8.5231
2.64	5.474	2.3844	3.14	7.744	4.7718	3.64	10.41	8.6174
2.65	5.515	2.4208	3.15	7.793	4.8330	3.65	10.46	8.7125
2.66	5.557	2.4575	3.16	7.843	4.8946	3.66	10.52	8.8084
2.67	5.599	2.4947	3.17	7.892	4.9568	3.67	10.58	8.9050
2.68	5.641	2.5322	3.18	7.942	5.0197	3.68	10.64	9.0025
2.69	5.683	2.5702	3.19	7.992	5.0832	3.69	10.69	9.1007
2.70	5.726	2.6087	3.20	8.042	5.1472	3.70	10.75	9.1998
2.71	5.768	2.6476	3.21	8.093	5.2119	3.71	10.81	9.2996
2.72	5.811	2.6868	3.22	8.143	5.2771	3.72	10.87	9.4003
2.73	5.853	2.7266	3.23	8.194	5.3430	3.73	10.93	9.5018
2.74	5.896	2.7668	3.24	8.245	5.4094	3.74	10.99	9.6041
2.75	5.940	2.8074	3.25	8.296	5.4765	3.75	11.04	9.7072
2.76	5.983	2.8484	3.26	8.347	5.5442	3.76	11.10	9.8112
2.77	6.026	2.8899	3.27	8.398	5.6126	3.77	11.16	9.9160
2.78	6.070	2.9319	3.28	8.450	5.6815	3.78	11.22	10.0216
2.79	6.114	2.9743	3.29	8.501	5.7511	3.79	11.28	10.1281
2.80	6.158	3.0172	3.30	8.553	5.8214	3.80	11.34	10.2350
2.81	6.202	3.0605	3.31	8.605	5.8923	3.81	11.40	10.3436
2.82	6.246	3.1043	3.32	8.657	5.9638	3.82	11.46	10.4526
2.83	6.290	3.1486	3.33	8.709	6.0359	3.83	11.52	10.5624
2.84	6.335	3.1933	3.34	8.762	6.1088	3.84	11.58	10.6486
2.85	6.379	3.2385	3.35	8.814	6.1823	3.85	11.64	10.7848
2.86	6.424	3.2842	3.36	8.867	6.2564	3.86	11.70	10.8970
2.87	6.469	3.3304	3.37	8.920	6.3312	3.87	11.76	11.0110
2.88	6.514	3.3771	3.38	8.973	6.4067	3.88	11.82	11.1250
2.89	6.560	3.4242	3.39	9.026	6.4829	3.89	11.88	11.2400
2.90	6.605	3.4719	3.40	9.079	6.5597	3.90	11.95	11.3560
2.91	6.651	3.5200	3.41	9.133	6.6372	3.91	12.01	11.4730
2.92	6.697	3.5686	3.42	9.186	6.7154	3.92	12.07	11.5910
2.93	6.743	3.6178	3.43	9.240	6.7943	3.93	12.13	11.7100
2.94	6.789	3.6674	3.44	9.294	6.8739	3.94	12.19	11.8290
2.95	6.835	3.7175	3.45	9.348	6.9542	3.95	12.25	11.9500
2.96	6.881	3.7682	3.46	9.402	7.0352	3.96	12.32	12.0690
2.97	6.928	3.8194	3.47	9.457	7.1168	3.97	12.38	12.1930
2.98	6.975	3.8711	3.48	9.511	7.1976	3.98	12.44	12.3170
2.99	7.022	3.9233	3.49	9.566	7.2824	3.99	12.50	12.4410

125.01 FUNCTIONS OF DIAMETERS (Continued)

DIA.	AREA	I	DIA.	AREA	I	DIA.	AREA	I
4.00	12.57	12.566	4.50	15.90	20.129	5.00	19.63	30.680
4.01	12.63	12.692	4.51	15.98	20.308	5.01	19.71	30.926
4.02	12.69	12.820	4.52	16.05	20.489	5.02	19.79	31.173
4.03	12.76	12.948	4.53	16.12	20.671	5.03	19.87	31.423
4.04	12.82	13.077	4.54	16.19	20.854	5.04	19.95	31.673
4.05	12.88	13.207	4.55	16.26	20.990	5.05	20.03	31.925
4.06	12.95	13.337	4.56	16.33	21.224	5.06	20.11	32.179
4.07	13.01	13.469	4.57	16.40	21.411	5.07	20.19	32.434
4.08	13.07	13.602	4.58	16.47	21.599	5.08	20.27	32.691
4.09	13.14	13.736	4.59	16.55	21.788	5.09	20.35	32.949
4.10	13.20	13.871	4.60	16.62	21.979	5.10	20.43	33.209
4.11	13.27	14.007	4.61	16.69	22.170	5.11	20.51	33.470
4.12	13.33	14.143	4.62	16.76	22.363	5.12	20.59	33.733
4.13	13.40	14.281	4.63	16.84	22.557	5.13	20.67	33.997
4.14	13.46	14.420	4.64	16.91	22.753	5.14	20.75	34.263
4.15	13.53	14.560	4.65	16.98	22.950	5.15	20.83	34.530
4.16	13.59	14.701	4.66	17.06	23.148	5.16	20.91	34.799
4.17	13.66	14.843	4.67	17.13	23.347	5.17	20.99	35.070
4.18	13.72	14.985	4.68	17.20	23.548	5.18	21.07	35.342
4.19	13.79	15.129	4.69	17.28	23.750	5.19	21.16	35.615
4.20	13.85	15.274	4.70	17.35	23.953	5.20	21.24	35.891
4.21	13.92	15.420	4.71	17.42	24.157	5.21	21.32	36.168
4.22	13.99	15.568	4.72	17.50	24.363	5.22	21.40	36.446
4.23	14.05	15.715	4.73	17.57	24.570	5.23	21.48	36.726
4.24	14.12	15.865	4.74	17.65	24.779	5.24	21.57	37.008
4.25	14.19	16.015	4.75	17.72	24.989	5.25	21.65	37.291
4.26	14.25	16.166	4.76	17.80	25.200	5.26	21.73	37.576
4.27	14.32	16.319	4.77	17.87	25.412	5.27	21.81	37.863
4.28	14.39	16.472	4.78	17.95	25.626	5.28	21.90	38.151
4.29	14.45	16.626	4.79	18.02	25.841	5.29	21.98	38.440
4.30	14.52	16.782	4.80	18.10	26.058	5.30	22.06	38.732
4.31	14.59	16.938	4.81	18.17	26.275	5.31	22.15	39.025
4.32	14.66	17.096	4.82	18.25	26.495	5.32	22.23	39.320
4.33	14.73	17.255	4.83	18.32	26.715	5.33	22.31	39.617
4.34	14.79	17.415	4.84	18.40	26.937	5.34	22.40	39.915
4.35	14.86	17.576	4.85	18.47	27.160	5.35	22.48	40.215
4.36	14.93	17.738	4.86	18.55	27.385	5.36	22.56	40.516
4.37	15.00	17.902	4.87	18.63	27.611	5.37	22.65	40.819
4.38	15.07	18.066	4.88	18.70	27.839	5.38	22.73	41.124
4.39	15.14	18.231	4.89	18.78	28.067	5.39	22.82	41.526
4.40	15.21	18.398	4.90	18.86	28.298	5.40	22.90	41.739
4.41	15.27	18.566	4.91	18.93	28.530	5.41	22.99	42.049
4.42	15.34	18.735	4.92	19.01	28.763	5.42	23.07	42.361
4.43	15.41	18.905	4.93	19.09	28.997	5.43	23.16	42.674
4.44	15.48	19.077	4.94	19.17	29.233	5.44	23.24	42.990
4.45	15.55	19.249	4.95	19.24	29.471	5.45	23.33	43.307
4.46	15.62	19.423	4.96	19.32	29.710	5.46	23.41	43.626
4.47	15.69	19.598	4.97	19.40	29.950	5.47	23.50	43.946
4.48	15.76	19.773	4.98	19.48	30.192	5.48	23.59	44.268
4.49	15.83	19.950	4.99	19.56	30.435	5.49	23.67	44.572

125.01 FUNCTIONS OF DIAMETERS (Concluded)

DIA.	AREA	I
5.50	23.76	44.918
5.55	24.19	46.574
5.60	24.63	48.275
5.65	25.07	50.022
5.70	25.52	51.817
5.75	25.97	53.659
5.80	26.42	55.550
5.85	26.88	57.490
5.90	27.34	59.481
5.95	27.81	61.523
6.00	28.27	63.617
6.10	29.22	67.965
6.20	30.19	72.533
6.30	31.17	77.327
6.40	32.17	82.355
6.50	33.18	87.624
6.60	34.21	93.142
6.70	35.26	98.917
6.80	36.32	104.956
6.90	37.39	111.266
7.00	38.48	117.859
7.10	39.59	124.739
7.20	40.72	131.917
7.30	41.85	139.399
7.40	43.01	147.196
7.50	44.18	155.316
7.60	45.36	163.766
7.70	46.57	172.557
7.80	47.78	181.697
7.90	49.02	191.197
8.00	50.27	201.062
8.10	51.53	211.305
8.20	52.81	221.935
8.30	54.11	232.960
8.40	55.42	244.392
8.50	56.75	256.239
8.60	58.09	268.512
8.70	59.45	281.220
8.80	60.82	294.375
8.90	62.21	307.985
9.00	63.62	322.062
9.10	65.04	336.616
9.20	66.48	351.659
9.30	67.93	367.199
9.40	69.40	383.249
9.50	70.88	399.820
9.60	72.38	416.922
9.70	73.90	434.567
9.80	75.43	452.766
9.90	76.98	471.531
10.00	78.50	490.874

125.02 CONVERSION FACTORS, ENGLISH - METRIC The following is a list of the more common English measurement units used in Hydromechanical Design, and the equivalent matching units used in the Metric system. Multiply the English unit by the factor to convert to Metric units. Divide the Metric unit by the factor to convert to English units.

Dimension	English	Factor	Metric
Velocity	Feet Per Second (Ft/Sec)	3.048	Meters Per Second (M/Sec)
	Miles Per Hour (Mph)	1.6093	Kilometers Per Hour (Km/Hr)
	Knots (Kn)	1.8532	Kilometers Per Hour (Km/Hr)
Liquid Flow	Gallons Per Minute (Gpm)	3.7854	Liters Per Minute (L/M)
	Cubic Inches Per Second ($In.^3/Sec$)	16.3870	Cubic Centimeters Per Second (CM^3/Sec) (Millimeters Per Sec = Ml/Sec)
Pressure	Pound Per Square Inch (psi)	0.07031	Kilogram Per Square Centimeter (Kg/Cm^2)
Moment and Torque	Inch-Pounds	1.1521	Centimeter-Kilograms (Cm-Kg)
	Foot-Pounds	1.3825	Meters-Kilograms (Nm-Kg)
Power	Horsepower (Hp)	0.7457	Kilowatts (Kw)
Length	Inches (In.)	25.4000	Millimeter (Mm)
Area	$Inches^2$ ($In.^2$)	6.4516	$Centimeters^2$ (Cm^2)
Volume	$Inches^3$ ($In.^3$)	16.3870	$Centimeters^3$ (Cm^3)
Liquid	Gallons (Gal)	3.7854	Liters (L)
Gas	Cubic Inches ($In.^3$)	0.01639	Liters (L)
Force	Pounds-Force (Lb)	0.4536	Kilogram-Force (Kgf)

Dimension	English	Factor	Metric
Weight	Pounds (Lb)	0.4536	Kilogram-Force (Kgf)
	Ounces (Oz)	453.592	Grams (gm)
Mass	Pounds-Mass (Lb)	0.4536	Kilogram-Mass (Kgm)
Density	Pounds Per Cubic Inch ($Lb/In.^3$)	27680	Kilograms Per Cubic Meter (Kg/M^3)
Heat	BTU	1054.8	Joules
	BTU	0.2928	Watt-Hours
Mechanical Energy	Foot-Pound (Ft/Lb)	0.13825	Meter-Kilogram (M-KG)

125.03 ADDITIONAL CONVERSION FACTORS

Item	Units	Conversions
Length	inch feet mile	1 micron = 0.000039 inches 1 nautical mile = 1.1515 statute miles 1 statute mile = 5280 feet (1 knot = 1 nautical mile per hour)
Mass	lb (mass)	$1 \text{ lb (mass)} = \dfrac{1 \text{ lb (of force)}}{32.2 \text{ ft/sec}^2}$ $= \dfrac{.031 \text{ lb (of force) sec}^2}{\text{ft}}$ = .031 slug
Acceleration due to gravity	ft/sec^2	g (standard) = 32.174 ft/sec^2
Energy or Work	ft/lb btu Kw/hr	1 Btu = 778 foot-pounds = 0.0000293 kilowatt-hour
Power	horsepower	$1 \text{ Hp} = \dfrac{550 \text{ ft/lb}}{\text{sec}}$ or = 746 watts
Stress	$lb/in.^2$	1 psi = 6895 $Newtons/meter^2$

Item	Units	Conversions
Pressure	$lb/in.^2$ or psi	1 psi = 0.068 atmospheres = 2.309 ft of water at 70°F = 2.045 inches of mercury at 70°F = 2.67 ft or MIL-H-5606 at 70°F
Volume	gallon cubic inches	1 gallon = 231 $in.^3$
Temperature	°F °C	°F = 32° + (1.8 x °C) °C = 1.8 (°F - 32)
Absolute Temperature	°R = °Rankine °K = °Kelvin	°R = °F + 459.69 °K = °C + 273.16
Mass Density	$\frac{lb/sec^2}{in.^4}$	1 $slug/ft^3$ = 4.83×10^{-5} $lb/sec^2/in.^4$ = 5.78×10^{-4} $slugs/in.^3$
Mass Unit Volume		
Kinematic viscosity	Stoke	The stoke is the CGS unit of kinematic viscosity in $\frac{cm^2}{sec}$ units. 1 $in.^2/sec$ = 6.452×10^2 CS (centistokes) 1 CS = 1.55×10^{-3} $in.^2/sec$ 1 stoke = 100 CS The Saybolt Second Universal, SSU, is a measure of kinematic viscosity from a particular viscometer, the Saybolt. Stokes = $.00226T \left(\frac{-1.95}{T}\right)$ when $32° < t\ (°F) < 100°$ Stokes = $.0022T \left(\frac{-1.35}{T}\right)$ when $t\ (°F) > 100°$ where T = SSU reading in seconds
Absolute (Dynamic) Viscosity	Poise	The poise is the CGS unit of absolute viscosity, its units are $\frac{\text{grams (of mass)}}{\text{(sec) (cm)}}$ or $\frac{\text{(dyne) (sec)}}{\text{(cm)}^2}$

Item	Units	Conversions
Absolute to Kinematic Viscosity	—	1 poise = 100 CP (centipoise) $1 \frac{(lb)\ (sec)}{(in.)^2} = 6.897 \times 10^6$ CP $1\ CP = 1.45 \times 10^{-7} \frac{(lb)\ (sec)}{(in.)^2}$ $1\ stoke = \frac{1\ poise}{\rho}$ where ρ = mass density of fluid in $\frac{grams\ (of\ mass)}{(cm)^3}$ units or (specific gravity)

125.04 USEFUL FORMULAS

125.041 TUBING PRESS DROP

1. Reynolds No. $= N_R = \frac{DV}{\nu} = \frac{\rho DV}{\mu}$

   where D = Tube I.D., inches
   V = fluid velocity, in./sec
   ν = kinematic viscosity, in.^{2}/sec
   ρ = mass density, lb (sec)^2/in.^{4}
   μ = dynamic viscosity, lb sec/in.^{4}

2. Friction Factor $= f = \frac{64}{N_R}$ (Laminar Flow)

   $$f = \frac{.316}{(N_R)^{.25}} \text{ (Turbulent Flow)}$$

3. Pressure Drop $= \Delta P = \frac{.0071\mu Q}{D^4} = \frac{.04575\nu(SG)QL}{D^4}$ = psi/ft

   (for laminar flow)

   where Q = flow rate, in.^{3}/sec
   μ = dynamic viscosity, poise
   D = Tube I.D., inches
   ν = kinematic viscosity, in.^{2}/sec
   SG = specific gravity
   L = tubing length, feet

where Q_c = flow rate, $in.^3/sec$ (concentric)
ΔP = pressure drop, psi
D = piston dia, inches
C = diametral clearance, inches
μ = absolute viscosity, $lb\text{-}sec/in.^2$
L = length, inches

$$Q_{eccentric} = 2.5\ Q_{concentric}$$

125.043 HORSEPOWER EQUATIONS

$$HP = \frac{QP}{1714} = \frac{TS}{63025} = T_\omega\ (2.644 \times 10^{-6})$$

where HP = horsepower
Q = flow rate, GPM
T = torque, in.-lb
S = speed, RPM
ω = angular rate, deg/sec

Note: For actual design, the results of the above equations must be modified by the efficiency of the unit or system being considered.

125.044 CYLINDRICAL PRESSURE VESSELS

Thin wall tubes, where t is less than 0.1 b

1. $$f_t = \frac{PR}{t}$$

where P = pressure, psi
t = wall thickness, inches
a = inside radius, inches
f_t = hoop stress, psi
R = mean radius, inches $= a + \frac{t}{2}$
b = outside radius, inches

2. $$\Delta a = \frac{R}{E}\left(f_t - \frac{\alpha PR}{2t}\right)$$

where α = Poisson's ratio
= 0.36 for aluminum alloy
= 0.25 for cast iron
= 0.26 for steel
= 0.355 for titanium
Δa = radial deflection, inches
E = modulus of elasticity, psi

Thick wall tubes, where t is greater than 0.1 b

3. Under internal pressure

$$f_t = P\left(\frac{b^2 + a^2}{b^2 - a^2}\right)$$

where f_t = max hoop stress at "a" radius

4. Pressure Drop $= \Delta P = \dfrac{.000908f(SG)Q^2}{d^5} = \dfrac{.00027\nu^{.25}(SG)Q^{1.75}L}{d^{4.75}}$

$= \text{psi/ft}$

(for turbulent flow)

where f = friction factor
Q = flow rate, $in.^3/sec$
D = Tube I.D., inches
ν = kinematic viscosity, $in.^2/sec$
SG = specific gravity
L = tubing length, feet

$$Q = AV = \frac{\pi D^2}{4} V$$

125.042 ORIFICE EQUATIONS

1. $Q = C_o A_o \sqrt{2g \dfrac{\Delta P}{\rho}}$,

where Q = flow rate, $in.^3/sec$
A_o = orifice area, $in.^2$
g = gravitational constant, $in./sec^2$
Δp = pressure drop, $lb/in.^2$ (psi)
ρ = density, $lb/in.^3$
C_o = orifice discharge coefficient (.5 → 1.0)

2. Orifices in Series

$$\frac{1}{(A_{eff})^2} = \frac{1}{(A_1)^2} + \frac{1}{(A_2)^2} + \ldots \frac{1}{(A_n)^2}$$

where A_1, etc = orifice areas
A_{eff} = effective orifice area

3. Orifices in Parallel

$$A_{eff} = A_1 + A_2 + \ldots A_n$$

4. Annular Orifice

$Q_c = \dfrac{\pi \Delta P\ DC^3}{96\mu,\ L}$ (laminar flow) (concentric)

4. $\Delta a = P \frac{a}{E}\left(\frac{b^2 + a^2}{b^2 - a^2} + \alpha\right)$ where Δa = radial deflection at "a" radius, inches

5. $\Delta b = P \frac{b}{E}\left(\frac{2a^2}{b^2 - a^2}\right)$ where Δb = radial deflection at "b" radius, inches

6. Under external pressure

$f_c = -P\left(\frac{2b^2}{b^2 - a^2}\right)$ where f_c = max stress at inner surface

7. $\Delta a = -P \frac{a}{E}\left(\frac{2b^2}{b^2 - a^2}\right)$

8. $\Delta b = -P \frac{b}{E}\left(\frac{a^2 + b^2}{b^2 - a^2} - \alpha\right)$

Pressure change in cylinders (tubing) due to differential thermal expansion of entrapped fluids and cylinder material

9. $$\Delta P = \frac{\Delta t\ (C_1 - C_2)}{C_3 + \frac{d^2}{4Ea\ (d+a)} + \frac{d}{Ea}}$$

where ΔP = pressure change, psi
Δt = temperature change, °F
C_1 = Coefficient of thermal expansion of fluid, in.^{3}/in.^{3}/°F
C_2 = Coefficient of thermal expansion of cylinder material, in./in./°F
C_3 = Coefficient of volumetric elasticity of fluid, in.^{3}/in.^{3}/psi
d = cylinder inside diameter, inches
a = cylinder wall thickness, inches
E = modulus of elasticity for cylinder material, psi

125.05 DECIMAL EQUIVALENTS

1/64	.015625	33/64	.515625
1/32	.03125	17/32	.53125
3/64	.046875	35/64	.546875
1/16	.0625	9/16	.5625
5/64	.078125	37/64	.578125
3/32	.09375	19/32	.59375
7/64	.109375	39/64	.609375
1/8	.125	5/8	.625
9/64	.140625	41/64	.640625
5/32	.15625	21/32	.65625
11/64	.171875	43/64	.671875
3/16	.1875	11/16	.6875
13/64	.203125	45/64	.703125
7/32	.21875	23/32	.71875
15/64	.234375	47/64	.734375
1/4	.25	3/4	.75
17/64	.265625	49/64	.765625
9/32	.28125	25/32	.78125
19/64	.296875	51/64	.796875
5/16	.3125	13/16	.8125
21/64	.328125	53/64	.828125
11/32	.34375	27/32	.84375
23/64	.359375	55/64	.859375
3/8	.375	7/8	.875
25/64	.390625	57/64	.890625
13/32	.40625	29/32	.90625
27/64	.421875	59/64	.921875
7/16	.4375	15/16	.9375
29/64	.453125	61/64	.953125
15/32	.46875	31/32	.96875
31/64	.484375	63/64	.984375
1/2	.5	1	1.

125.06 STRENGTH-TO-HARDNESS CONVERSION

1. Tensile strength and equivalent hardness for carbon and low alloy steels are listed in the following table.

2. Tensile strength versus hardness readings for aluminum alloys and other soft metals have little or no relationship; however, hardness readings generally indicate whether hardenable alloys have been heat treated.

Rockwell C-Scale Hardness Number (1)	Brinell Hardness Number 10mm Ball, 3000 KG Load			Rockwell Hardness Number (1)			Rockwell Superficial Hardness Number, Superficial Brale Penetrator			Shore Sclero-scope Hardness Number	Tensile Strength (Approx-imate) KSI (2)	Rockwell C-Scale Hardness Number (1)
	Standard Ball	Hultgren Ball	Tungsten Carbide Ball	A-Scale, 60 KG Load, Brale Penetrator	B-Scale, 100 KG Load, 1/16 In. Diam. Ball	D-Scale, 100 KG Load, Brale Penetrator	15-N Scale, 15 KG Load	30-N Scale, 30 KG Load	45-N Scale, 45 KG Load			
55	—	(546)	560	78.5	—	66.9	87.9	73.0	60.9	74	301	55
54	—	(534)	543	78.0	—	66.1	87.4	72.0	59.8	72	292	54
53	—	(519)	525	77.4	—	65.4	86.9	71.2	58.6	71	283	53
52	(500)	(508)	512	76.8	—	64.6	86.4	70.2	57.4	69	273	52
51	(487)	494	496	76.3	—	63.8	85.9	69.4	56.1	68	264	51
50	(475)	481	481	75.9	—	63.1	85.5	68.5	55.0	67	255	50
49	(464)	469	469	75.2	—	62.1	85.0	67.6	53.8	66	246	49
48	(451)	455	455	74.7	—	61.4	84.5	66.7	52.5	64	237	48
47	442	443	443	74.1	—	60.8	83.9	65.8	51.4	63	229	47
46	432	432	432	73.6	—	60.0	83.5	64.8	50.3	62	222	46
45	421	421	421	73.1	—	59.2	83.0	64.0	49.0	60	215	45
44	409	409	409	72.5	—	58.5	82.5	63.1	47.8	58	208	44
43	400	400	400	72.0	—	57.7	82.0	62.2	46.7	57	201	43
42	390	390	390	71.5	—	56.9	81.5	61.3	45.5	56	194	42
41	381	381	381	70.9	—	56.2	80.9	60.4	44.3	55	188	41
40	371	371	371	70.4	—	55.4	80.4	59.5	43.1	54	181	40
39	362	362	362	69.9	—	54.6	79.9	58.6	41.9	52	176	39
38	353	353	353	69.4	—	53.8	79.4	57.7	40.8	51	170	38
37	344	344	344	68.9	—	53.1	78.8	56.8	39.6	50	165	37
36	336	336	336	68.4	(109.0)	52.3	78.3	55.9	38.4	49	160	36
35	327	327	327	67.9	(108.5)	51.5	77.7	55.0	37.2	48	155	35
34	319	319	319	67.4	(108.0)	50.8	77.2	54.2	36.1	47	150	34
33	311	311	311	66.8	(107.5)	50.0	76.6	53.3	34.9	46	147	33
32	301	301	301	66.3	(107.0)	49.2	76.1	52.1	33.7	44	142	32
31	294	294	294	65.8	(106.0)	48.4	75.6	51.3	32.5	43	139	31
30	286	286	286	65.3	(105.5)	47.7	75.0	50.4	31.3	42	136	30
29	279	279	279	64.7	(104.5)	47.0	74.5	49.5	30.1	41	132	29
28	271	271	271	64.3	(104.0)	46.1	73.9	48.6	28.9	41	129	28
27	264	264	264	63.8	(103.0)	45.2	73.3	47.7	27.8	40	126	27
26	258	258	258	63.3	(102.5)	44.6	72.8	46.8	26.7	38	123	26
25	253	253	253	62.8	(101.5)	43.8	72.2	45.9	25.5	38	120	25
24	247	247	247	62.4	(101.0)	43.1	71.6	45.0	24.3	37	118	24
23	243	243	243	62.0	100.0	42.1	71.0	44.0	23.1	36	115	23
22	237	237	237	61.5	99.0	41.6	70.5	43.2	22.0	35	112	22
21	231	231	231	61.0	98.5	40.9	69.9	42.3	20.7	35	110	21
20	226	226	226	60.5	97.8	40.1	69.4	41.5	19.6	34	107	20
(18)	219	219	219	—	96.7	—	—	—	—	33	103	(18)
(16)	212	212	212	—	95.5	—	—	—	—	32	100	(16)
(14)	203	203	203	—	93.9	—	—	—	—	31	97	(14)
(12)	194	194	194	—	92.3	—	—	—	—	29	93	(12)
(10)	187	187	187	—	90.7	—	—	—	—	28	90	(10)
(8)	179	179	179	—	89.5	—	—	—	—	27	88	(8)
(6)	171	171	171	—	87.1	—	—	—	—	26	85	(6)
(4)	165	165	165	—	85.5	—	—	—	—	25	83	(4)
(2)	158	158	158	—	83.5	—	—	—	—	24	81	(2)
(0)	152	152	152	—	81.7	—	—	—	—	24	78	(0)

(1) Values in ( ) are beyond normal range and are given for information only.

(2) It is possible that steels of various compositions and processing histories will deviate in hardness-tensile strength relationship from the data presented in this table.

125.07 COMPUTER PROGRAMS - The computer programs available to analyze hydraulic and landing gear systems are listed in the following tables. For further information, the reader is directed to Report No. MDC J5680.

125.071 AIRCRAFT LANDING GEAR SYSTEM

Topic	Program No.	Title
Take-Off Landing and Taxi Dynamics	G3QA	Aircraft Landing and Take-Off Dynamics (pitch or pitch and yaw)
	G3QL	Aircraft Landing Analysis
	G3QR	Rotation - Take-Off Analysis
	G3QS	Space Shuttle "Crabbed" Landings
	G3QX	Inversion of G3QL
	G4XA	DC-9 Soft Gear Analysis
	H5CA	Catapult Yaw Response
	H5DA	Catapult Analysis With Nose Gear Tow and Articulating Nose Gear
	H9EA	Soft Riding Gear
	J3RK	3 Degrees of Freedom Transfer Functions
Statics	J6XD	Aircraft Static Ground Attitude
Steering and Manuevering	D1TA	Aircraft Turning Radius Program
Braking	B2EA	Brake Energy Program
	H2MA	Antiskid Simulation Checkout Program
Flight Test Data	H2UA	DC-10 Flight Test Data Tape Conversion
Flotation	ATKA	Soil Finite Element Representation
	C2XA C2XB	Master Ground Flotation Computer Program
	C2XG	Table Data Processing
	C2XM	Computerized Airport Data Bank
	H6AA	Portland Cement Association Program PDILB
	M6BA	Port of New York Authority Pavement Design Method
	M7C	Airport Pavement Design - PCA PDILB Program
	N5WA	Chevron N - Layered Flexible Pavement Analysis

125.072 LANDING GEAR UNIT

Topic	Program No.	Title
Dynamics	G3QP	Gear Extend/Retract Program (Nose or center gears)
	H6GA	Rough Terrain Landing Gear Taxi Analysis
	J3RA	Landing Impact, DC-10 Main Gear
	J3RB	DC-10 Main Gear Analysis, Including Bogie Trim Cylinder Dynamics
	J3RC	Double Metering Strut - DC-10
	J3RD	Triple Metering Strut - DC-10
	J3RF	Floating Metering Pin Analysis - DC-10
	J3RM	Matrix Generation for Landing Gear Stiffness
	J3RX	DC-10 Main Gear Landing Analysis (Debug Pack)
Steering System	F9LA	Nose Gear Steering Analysis
Vibration and Stability	F9CB	DC-10 Nose Gear Frequency Response Analysis
	M6VA	Landing Gear Shimmy Analysis
	M6XA	Modal Response Program
	N1TA	DC-9 Nose Gear Frequency Response Analysis
Landing Gear Components	A7GA	Clark Model of a Free Rolling Tire
	D4LA	Air Pressure Analysis, Shock Strut
	H8PA	Bogie Brake Torque Compensation Linkage Analysis

125.072 HYDRAULICS

Topic	Program No.	Title
Systems	D5RA	Hydraulic System Heat Transfer Analysis
	F9LB	Control Surface Hardover Response
	H7CA	Transient Hydraulic System Temperature Analysis
	J6GA	Hydraulic System Pressure Drop
	J6GB	Aircraft Hydraulic Piping Analysis
	M15A	Dynamic Performance Analysis, DC-10 Lateral Control Sys
	M6YA	Dynamic Characteristics, DC-10 Direct Lift Control Sys
Components	F7DA	DC-10 Surface Rate Analysis
	K9FA	Linear Stability Analysis, DC-10 Flight Control Actuators
	M6U	Stability Analysis, DC-10 Flight Control Actuators
	M6NA	Active Vibration Isolator
	M6ZA	Frequency Response Analysis, DC-10 Flight Control Actuators

125.073 Miscellaneous:

Topic	Program No.	Title
	A7JA	Mechanical Engineering Task Schedules
	L9AA	Task Schedule and Status Summary
	M4NA	Polynomial Fit - Least Squares