THE CLAMP CONNECTOR: AN ALTERNATIVE TO CONVENTIONAL FLANGES IN THE OFFSHORE AND INDUSTRIAL MARKETPLACES

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For more than thirty years the clamp connector has seen successful service in high pressure piping applications both offshore and in the industrial market-place. The connector incorporates a pressure energised sealing concept which is insensitive in service to bolting torque, and so lends itself to installations where external bending and tensile loads exist, and the use of conventional flanges would result in leakage occurring. The more widespread use of clamp connectors has been limited by both cost and the absence of a freely available design Standard. Increased competition amongst manufacturers has made these connectors available at acceptable prices, but published Standards covering pressure ratings and interchangeability are not yet available. In establishing such a standard there is clearly a need to include adequate reference to interchangability and external load graphs, similar to those under discussion within the American Petroleum Institute.

1.0 INTRODUCTION

The early Romans were known to have employed segmented wedge-type clamp devices in the construction of wooden water pipes. With some certainty we can say that such connectors would not have been subjected to the rigours of the ASME design criteria! However, any failure of these early connections would have resulted in merely spilled water, whereas failure of a fluid connector in a modern petrochemical plant or offshore oil platform is known to have catastrophic consequences both in terms of loss of human life and equipment damage.

The development of pipeline joint design, particularly with respect to flanges, has been essentially experimental based on various 'codes of design and construction'. However, the need for higher integrity connections and increased analysis and verification imposed by operators and Regulatory Authorities, has prompted many designers to look at the clamp connector as an alternative means of joining pipe.

This presentation attempts to provide a general background to clamp connector design and application to assist designers in determining the suitability or otherwise of specifying this type of joint. Without doubt the widespread application of the clamp connector has been hampered by the hitherto proprietary nature of the connector and the lack of any National and/or International Clamp Connector Standard. EEMUA has recognised this situation and is working towards the generation of an Industry Standard — as described later.

2.0 HISTORICAL DEVELOPMENT OF FLANGES

Present day flange configurations and design standards are the result of gradual evolution over the past 100 to 150 years (Ref: 1). Although integral flanges on cast iron pipe have probably been utilised for upwards of 500 years, it was John Russell who first patented in 1825 'Mechanical means of manufacturing butt-welded wrought iron pipe' and 'Screwed couplings made from pipe to join lengths of smaller pipe together'. The earliest types of loose or companion flanges, originating in the 1840's seem to have been made of wrought iron and apparently were either hammer or forged welded onto the pipe. The first reference to these flanges in the ASA Standard appears to be between 1927 and 1932.

Screwed flanges, although in use around 1850, did not become popular until after the development of the thread tapping machine by C. C. Walworth in 1855 and pipe threading machines in 1859.

The welding neck flanges may be considered the basic type of welding flange since it is the only welding type standardised for all pressure series. The idea was pioneered by John H. Zinc who commissioned Crane Company, to produce the fist welding neck flanges. This type of flange was developed primarily to avoid the necessity of refacing the gasket surface after attachment of a slip-on flange to the pipe. An additional advantage was the resultant stronger and more economical method of attachment, this being the only really acceptable method of joining pipe for severe service applications.

3.0 FLANGE TYPES

Six basic configurations of flanges are available, (Ref: 1):—

— Integral
— Welding neck
— Slip-on welding
— Lap-joint
— Screwed
— Blind

3.1 Integral Flange

Integral flanges are cast or forged integral with a fitting or valve body. Generally these flanges are the least highly stressed because of the support they receive from the thicker hub sections found in flanged valves and fittings.

3.2 Welding Neck Flange

Welding neck flanges, when butt welded to a pipe, constitute the closest practical approach to an integral flange of all the companion (ie mating with integral) flange types.

Their relative uniform stress condition provide the greatest factor of safety against leakage of all companion flange types and consequently are the preferred type for every severe service condition, whether this results from high pressure, sub-zero or elevated temperatures, or the handling of flammable, corrosive or costly fluids. Except for special applications (eg very low leakage requirements) welding neck flanges have become the industry standard worldwide.

3.3 Slip-on Flange

Slip-on flanges require to be double fillet welded to the pipe, the considerably shorter hub not providing the same resistance to bending as the welding neck flange. Slip-on flanges are more subject to fatigue failure particularly when located adjacent to reciprocating machinery such as pumps and compressors.

3.4 Lap Joint Flange

Lap joints are very similar to slip-on flanges from a stress analysis point of view. The ability to rotate the flange does aid assembly, although generally a lap joint stub is used which also requires welding to the pipe.

3.5 Screwed Flange

Screwed flanges are at least as highly stressed as slip-on flanges in the larger sizes, since they likewise cannot be given any real reinforcement by the pipe. The threads act as stress raisers and provide a path for leakage, a condition which is accentuated in high temperature service by differential expansion of the male thread in the flange.

3.6 Blind Flanges

Blind flanges when subjected to the ASME design rules for flat head closures, are required to be of a greater thickness than when designed in accordance with the ASA pipe line Standards. For high temperature applications welding neck flanges with caps are often used since less distortion occurs thereby minimising the possibility of leakage.

4.0 STANDARD FLANGES

The ASA Standard flange series is described by the designations given in Table (1).

Flange analysis considers two distinct load systems; gasket seating and internal pressure. The forces considered are:

— Hydrostatic end force
— Pressure force on the flange face
— The total gasket load required to maintain seal
— The bolt load

All except the bolt load are acting to separate the flange pair, with the gasket reaction load assumed to decrease as internal pressure is applied. The actual gasket load is affected by flange rotation, bolt stretch and the gasket's ability to resist and recover from compression. Bolt loads also change such that retightening must be carried out.

4.1 RTJ Flanges

RTJ flanges cause the total bolt force to be reacted at a seal ring inside the bolt circle. When fully made-up the flanges exhibit a 'stand-off' at the bolt circle which allows flange rotation to occur during assembly. Excessive bolt tension can cause permanent distortion of the flange ring.

TABLE 1

ASA FLANGE DESIGNATIONS

SERIES	SIZE RANGES
150 lb	1/2" – 24"
300 lb	1/2" – 24"
400 lb	4" – 24"
600 lb	1/2" – 24"
900 lb	3" – 24"
1500 lb	1/2" – 24"
2500 lb	1/2" – 12"

NOTE:-
1. 600lb flanges are used for 400lb conditions in sizes $3^{1/2}$" and below.
2. 1500lb flanges are used for 900lb conditions in sizes $2^{1/2}$" and below.

4.2 Flat Face Gasket Flanges

Flanges utilising flat faced gaskets cannot rotate under the action of applied bolt torques, but have the disadvantage that a large area of gasket needs to be compressed. Generally, these flanges are used for lower pressure applications.

5.0 LOADS OTHER THAN INTERNAL PRESSURE

For general piping applications it is necessary to consider the effect of other loadings in combination with internal pressure. These loadings include longitudinal forces, bending and torsional moments due to external loadings.

In the chemical and process industries a range of standard flanges are in widespread use:—

ANSI B16.5
Steel Pipe Flanges and Flanged Fittings

ANSI B16.1 Cast Iron Flanges and Flanged Fittings

ANSI B 16.24
Bronze Flanges and Flanged Fittings

MSSP SPP 44
Steel Pipeline Flanges

MSS SP 51
150lb Corrosion Resistant Cast Flanges and Flanged Fittings

ANSI B16.31
Non Ferrous Flanges

EEMUA 145/1987
CuNi Flanges for Off-shore Applications

The majority of piping flange design are based on ANSI Standard B16.5, which although giving allowable pressure/temperature ratings, offers no guidance as to permissible bending moments. Traditionally, these flanges have been used up to the allowable ratings without any check of their capability to carry additional loadings external loads.

5.1 Equivalent Pressure Methods

Flange strengths are not uniform when assessed by ASME Code analytical methods, with smaller and lower pressure classes having a greater reserve strength than larger sizes and higher pressure classes.

Pipeline practice has, therefore, been to keep joints to a minimum and to position flanges such that applied moments are reduced. The 'equivalent pressure formula' developed by Kellogg some forty years ago is still widely used (Ref: 2):—

$$Pe = \frac{16M}{\Pi G^3} + \frac{4F}{\Pi G^2}$$

The flange is checked for its ability to withstand a pressure of (Pe + Pd), where Pd is the design operating pressure. Pe is the sum of the external bending moment and axial force and is based on thin wall pressure vessel theory.

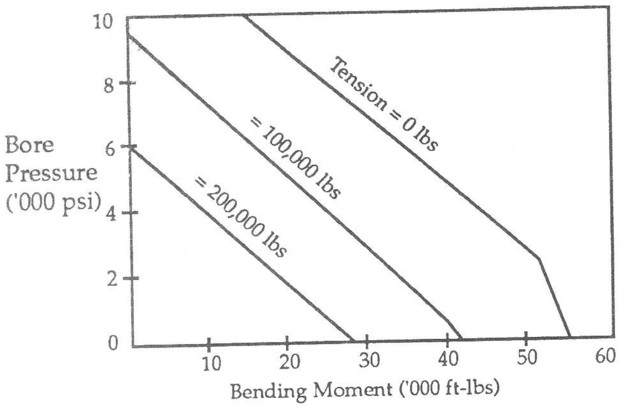
5.2 API 6A Flanges

A recent study on the 'Capability of API Flanges under Combination Loading' (Ref: 3), attempted to provide load ratings for all 69 API flanges for various combinations of external loads. A UKOOA Workgroup reviewed the draft report, and found it lacking in both adequate detailed analysis and practical testing, thereby rendering the tables of loading somewhat suspect. API also acknowledged that a number of flanges were overstressed at rated internal pressure and that the necessary bolt make-up stress of one-half the bolt yield stress was essential to maintain joint integrity.

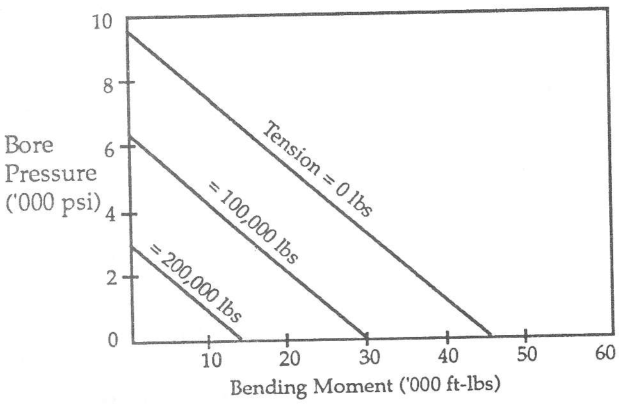
A typical loading curve for a 4-1/16in 10,000 psi 6BX flange is shown in Figure (1).

API $4^{1/16}$", 10,000psi FLANGE UNDER EXTERNAL LOAD

a) Bolt make-up Stress = 52,000 psi



b) Bolt make-up Stress = 40,000 psi



6.0 COMPARISON OF EUROPEAN DESIGN RULES

An informative comparison of 'Overseas Flange Design Rules versus ASME' (Ref: 4) compares flange strength based on DIN Rules to that calculated based on ASME methods, and arrives at the conclusion that the former are weaker — up to a factor of 0.75 for larger flanges. Furthermore, to overcome leakage during hydraulic testing re-tightening of the bolts often takes place causing permanent deformation of the flanges.

Dutch design rules require that actual bolt forces be considered when calculating the elastic rotation of the flange, with the maximum rotation limited to 1 degree. The author comments that designers very often prescribe higher bolting stresses than allowable in order to obtain 'leaktight' connections, these decisions more generally being based on experience and knowledge rather than with reference to any Rules.

High pressure API flanges, which are generally much smaller and lighter with smaller bolts, than equally rated standard flanges, were developed by the Standardisations Committee of AWHEM and were adopted by API in the second edition of Standard 6A (1963). API flange/gasket combinations are as follows:—

<u>6B Flanges (with stand-off)</u>

'R' gasket (standard: Octagonal, oval, groove).

'RX' gasket (Pressure energised).

For pressures up to 5,000 psi.

<u>6BX Gasket (no stand-off)</u>

'BX' gasket (pressure energised).

For pressures 5,000 psi and above.

<u>Clamp Connectors (no stand-off)</u>

'RX' gasket.

For all pressure ratings.

The high pressure 6BX flanges have raised faces and make-up face-to-face. Under zero pressure conditions, the bolt load is carried by the faces, thereby allowing the seal ring diameter and hence flange stresses to be reduced. Also, much higher allowable stresses have been used to reduce still further the flange and bolt dimensions.

The justification for these higher allowable stresses has been based on field experience, with no real consideration given to the effect of additional external loads.

8.0 THE CLAMP CONNECTOR

The clamp connector is a two piece four bolt connector developed in the early 1950's as a high pressure, high integrity means of joining pipework to pipework or pipework to equipment.

The design of the connector is such that many of the disadvantages of conventional flanges are eliminated with savings being possible in terms of weight, size envelope and installation time, Figure (2).

The connector shown in Figures (3) and (4) comprises four components:—
— Hubs
— Seal Ring
— Clamp
— Bolting

FIGURE 2

SIZE COMPARISON
CLAMP CONNECTOR Vs ANSI FLANGE

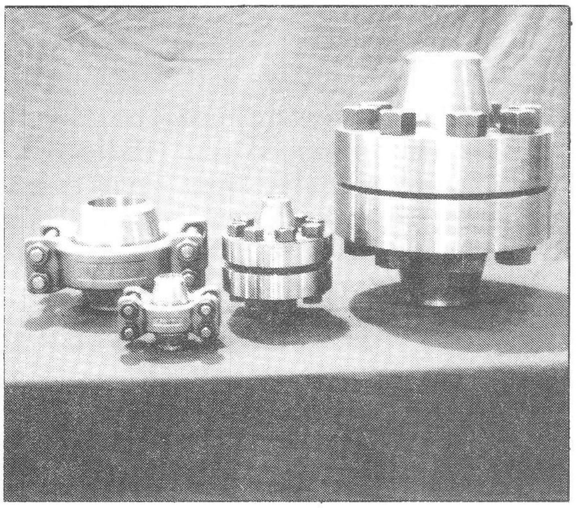
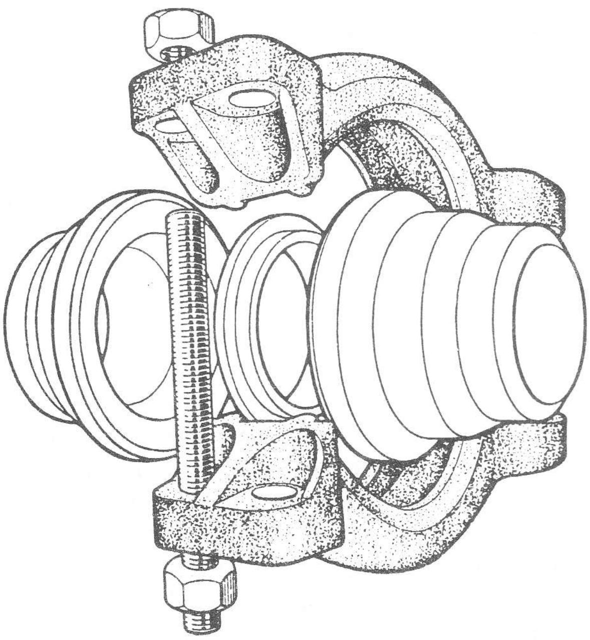


FIGURE 3

TYPICAL CLAMP CONNECTOR



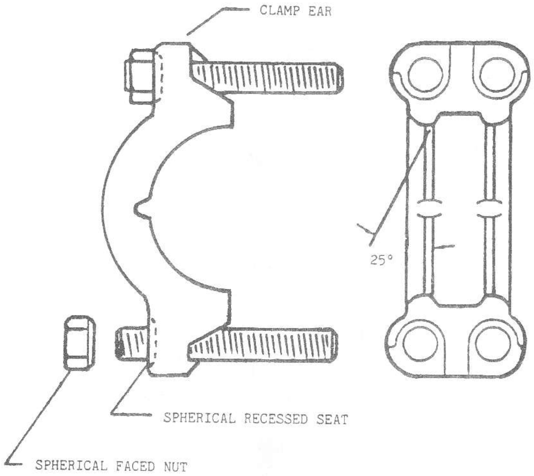
Hubs

Hubs would be supplied from the same generic material as other components of the connected system. For general industrial applications the hub material may only have the requirement of meeting a yield strength of 36,000 psi, although far more stringent conditions can be applied if necessary. Typically, for offshore applications A350 LF2 material modified to 52,000 psi yield is used when carbon steel is required. Normally, hubs would incorporate standard ANSI B16.5 butt welded ends, but socket welded, threaded or blind would also be available.

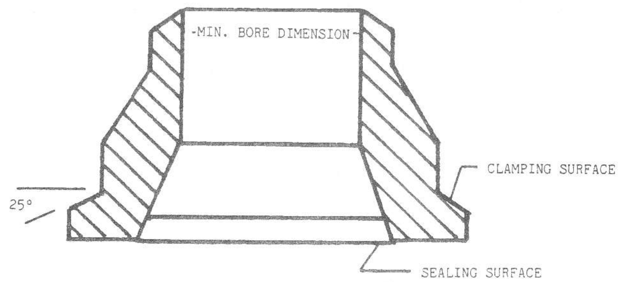
FIGURE 4

CLAMP CONNECTOR COMPONENT PARTS

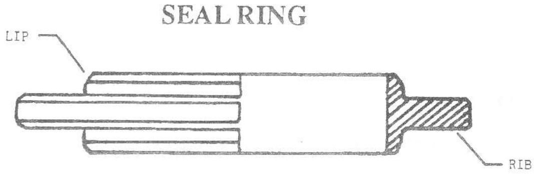
CLAMP



BUTT WELD HUB



SEAL RING



Seal Rings

During connector 'make-up' the seal ring surfaces are elastically deformed onto the hub sealing surface, Figure (5), due to the differential taper which exists. Internal pressure increases the effectiveness of the seal, deformation remaining elastic throughout thereby rendering the seal ring reusable.

Coating the seal ring with a baked-on lubricant such as PTFE prevents galling of the surfaces during make-up thereby enhancing seal re-usability. For high temperature applications molybdenum disulphide coating is normally specified in place of PTFE.

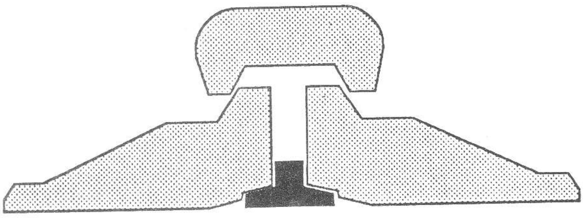
Clamps/Bolting

During assembly the hubs are drawn together under the twin wedge action of the clamp segments to provide the initial

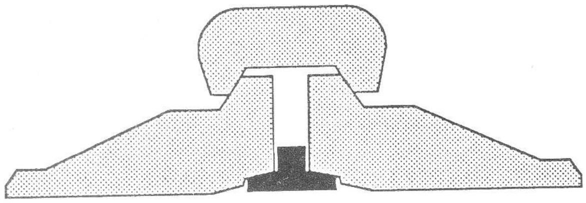
FIGURE 5

MAKE-UP OF SEAL RING

Prior to Assembly



Fully Assembled



self energised seal. Once the seal is energised the hub surfaces makes contact with the seal ring rib, with additional clamping loads being transmitted directly through the hubs with virtually 360 degree contact between clamp and hubs.

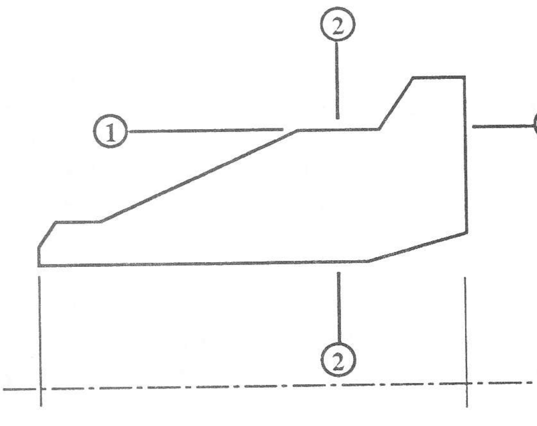
This represents a major variation to convention flanges in that the bolting is virtually isolated from the operation loads.

8.1 Hub Stresses

Hub diameter is maintained as small as possible consistent with the need for reinforcement local to the clamp bearing surface where the stresses are highest. Hub stress analysis is based upon classical theory similar to that for flanges, with stresses being determined at sections (1) and (2) as shown in Figure (6), where section (1) coincides with the base of the lip section.

FIGURE 6

LOCATIONS FOR HUB STRESS ANALYSIS

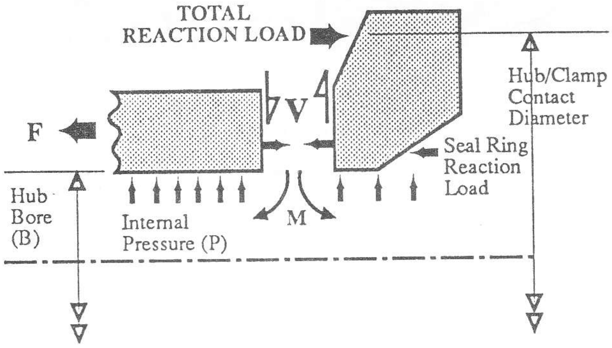


The free body diagramme given in Figure (7) forms the basis of the analysis where the stiffness of the flange and cylinder are evaluated and the rotation, displacement, shear force and the moment are determined. The shear and bending stresses across section (1), Figure (6), are also evaluated since this may be the limiting section for thick wall, high pressure hubs. The resulting stresses (hoop, axial and radial) are combined to give the stress intensity and are compared to the design code allowables — normally ASME VIII Division 2 criteria.

FIGURE 7

FREE BODY FORCE DIAGRAMME FOR HUB

$F = \Pi B^2 P$ + Axial Force + Equivalent Bending Force



8.2 Clamp Stresses

Clamps are designed to suit the full nominal size range and pressure rating of any given hub. This safety feature ensures that one clamp size can be used on any hub of the same nominal size, although on low pressure hubs this inevitably means that the clamp will be oversized in terms of strength. This feature has also prevented the clamp connector effectively penetrating the low pressure flange market from purely a cost viewpoint, even though it may be comparable in terms of weight and overall size. This drawback has become less apparent with the gradual introduction of high strength super stainless steels where the opportunity for greater savings of size and weight are possible. The moves which the industry is making with the advent of these materials has led manufacturers to develop complementary lighter weight connectors.

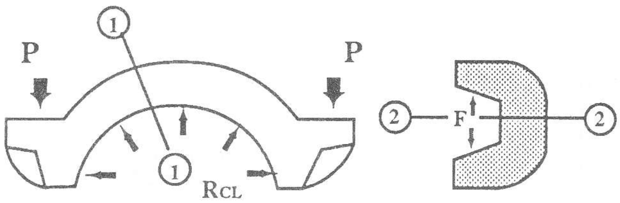
Clamp stresses are determined at sections (1) and (2) as shown in Figure (8).

The stresses at locations along section (2) are calculated as a result of the applied hub reaction load, F, generated when the clamp segments are in contact with hubs.

RcL, the radial component of F, is reacted by the bolts, P, giving rise to the induced

FIGURE 8

LOCATIONS FOR CLAMP STRESS ANALYSIS



hoop stresses across section (1). These stresses are combined with the axial stresses to obtain a total stress intensity.

Shear forces and torsional moments are not normally considered to react through the clamp, there being a maximum torsional moment beyond which rotational slippage will occur.

8.3 Design — General

Make-up stresses are determined in a similar manner to the operating stresses but with forces and moments being generated by the bolt load, based on the recommended torque value. In addition a seal ring (gasket) make-up load is also considered this being a constant for any given ring size; in the operating case the seal ring load is taken as zero.

Test condition stresses are based on the load resulting from the hydrotest pressure, with no external bending moments assumed present.

Stresses determined are categorised as primary membrane, primary membrane plus bending or secondary. Secondary stresses are applicable to the make-up condition and are assessed purely to prevent gross permanent deformation occuring under the action of the bolt torque, whilst the primary stresses apply to the operating conditions.

The primary membrane stresses represent the forces necessary to achieve equilibrium and balance the pressure forces (ie average stress across the wall thickness). The primary membrane plus bending stresses represent the maximum 'through thickness' stresses and are typically limited by the following criteria:—

$$Pm + Pb \leq 1.5 \times Sa$$

(Ref: ASME VIII, Div.2, App.4)

where Pm = Primary Membrane Stress
Pb = Primary Bending Stress
Sa = Basic Allowance Stress

Failure to adhere to this requirement could result in highly stessed local areas and/or local yielding. Whilst a limited amount of local yielding can be tolerated (generally during hydrostatic testing), this can only be allowed if the integrity of the components is not impaired causing the connector to leak.

9.0 INSTALLATION

Clamp connectors can be used in nearly all applications where flanges might be specified, and beyond the normal pressure rating of standard flanges, where otherwise fully welded construction might have been necessary.

TABLE 2

MAXIMUM MISALIGNMENT TOLERANCE FOR CLAMP CONNECTOR ASSEMBLY

CLAMP SIZE	MAXIMUM MISALIGNMENT	
	ANGULAR	DISPLACEMENT
2	2°	0.1in.
6	2°	0.1in.
10	2½°	0.15in.
16	2½°	0.2in.

Installation is straightforward with no angular bolt hole alignment requirement when fabricating hubs to pipework or equipment, thereby allowing cheaper off-site pre-fabrication of pipework and equipment. The connector can withstand a limited misalignment tolerance during assembly as indicated in Table (2).

Clamp assembly comprises the make-up of four bolts (irrespective of connector size) to the manufacturers' recommended torque values. Bolting torque requirements are considerably less than those required for flanges, typical values being given in Table (3).

Since the sealing ability of the connector is not affected by the bolting torque — beyond achieving the correct seating of the seal ring — over-torquing the bolts cannot compensate for a leaking joint. The seal

TABLE 3

TYPICAL CLAMP CONNECTOR BOLTING TORQUE REQUIREMENTS

CLAMP SIZE	BOLT TORQUE ft-lbf
1	17
1 1/2	35
2	55
2 1/2 / 3	60
4	90
6	200
8	300
10	600
12	900
14	1200
16	1300
18	1400
20	1700
24	1850

ring rib also acts as a stop to ensure that the seal lips are not overstressed leading to plastic deformation.

10.0 WEIGHT/COST SAVING

Offshore operators have, over the past few years become far more concious of the need to minimise platform topside weight in order to reduce installation cost. Recent significant reductions in oil revenues as a result in the fall in the price of oil, has emphasised the need for close attention to be paid to weight control.

Recent reviews on the subject (Ref: 5 and 6) estimate that one tonne of topside costs approximately 25,000 pds/stg 'all up', based on a simple division of some representative topside costs by their total dry weight. By adding the elements of stripping out, redesign and hook-up manpower, the Offshore Supplies office arrives at their cost estimate of up to 100,000 pds/stg per tonne.

The basic elements of structural and pipework costs should not be ignored as these can represent significant proportions of the topside dry weight. A saving of 125 tonnes on the piping components is a modest enough amount, representing 2% of a typical topside total of 6,500 tonnes for machinery, equipment and piping. However, it also implies a 175 tonnes saving in module structural weight and another 300 tonnes in jacket steel. Thus, the total weight savings would be worth well in excess of 1 million pds/stg.

Weight comparisons between clamp connectors and standard ANSI flanges shown in Figures (9) through (11).

FIGURE 9

WEIGHT COMPARISON
ANSI 900lb / PIPE SCH.80

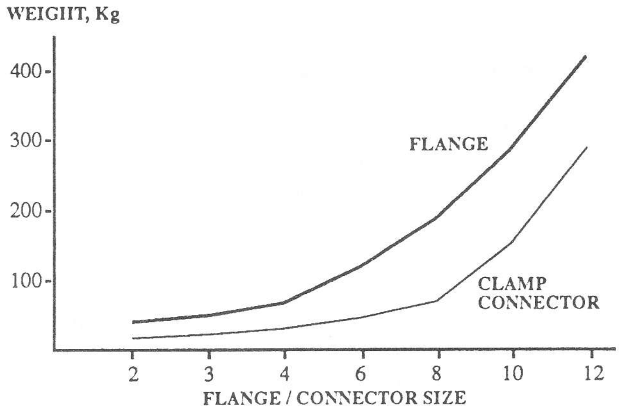


FIGURE 10

WEIGHT COMPARISON
ANSI 1500lb / PIPE SCH. 160

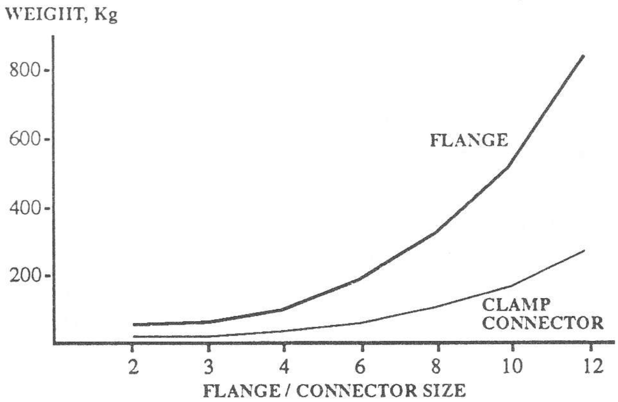
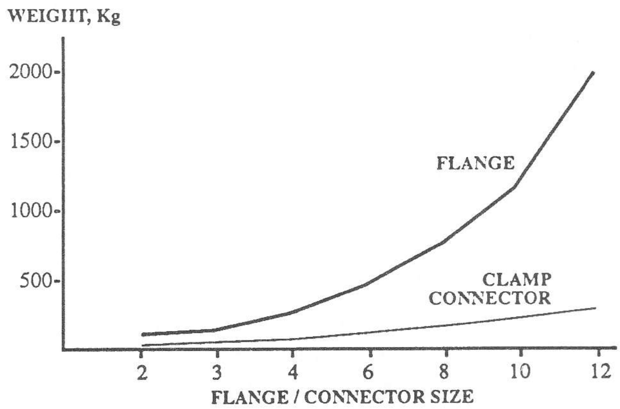


FIGURE 11

WEIGHT COMPARISON
ANSI 2500lb / PIPE SCH. 160



The potential cost savings to be realised by replacing flanges by clamp connectors has been analysed for the following typical installations:—

— Oil processing platform
— 10 slot gas wellhead platform
— Gas complex

Using flange requirement data provided by the major operators, the weight and resultant installed cost saving can be determined, as shown in Figures (12) to (14) for the oil processing platform.

A summary of typical overall installed cost savings given in Table (4).

FIGURE 12

OIL PROCESSING PLATFORM
NUMBER OF HIGH PRESSURE FLANGES

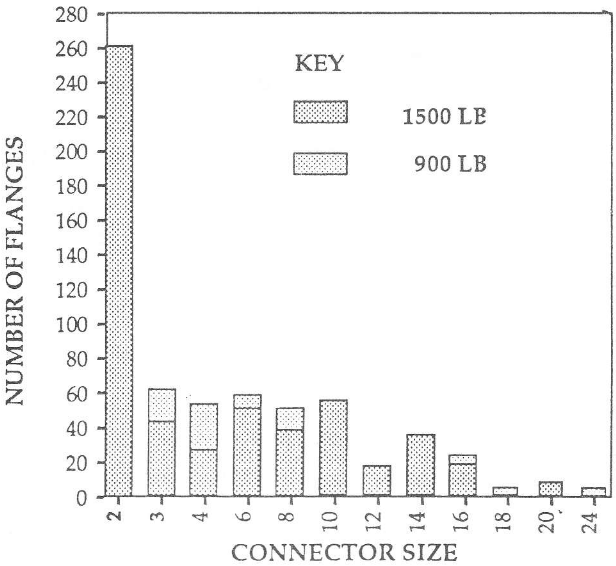


FIGURE 13

OIL PROCESSING PLATFORM
WEIGHT SAVING POSSIBLE BY USING
CLAMP CONNECTORS

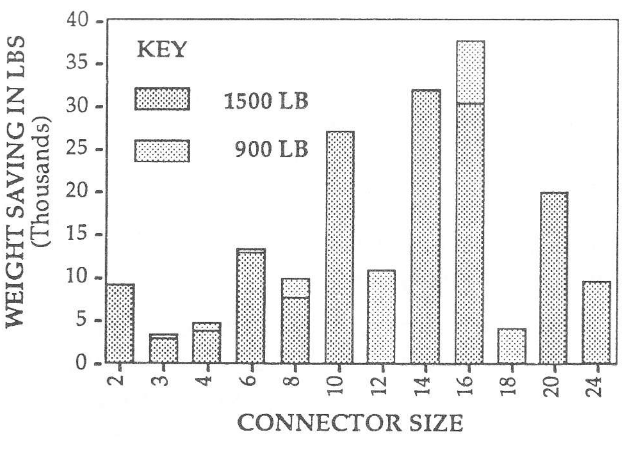


FIGURE 14

OIL PROCESSING PLATFORM
COST SAVING POSSIBLE BY USING
CONNECTORS

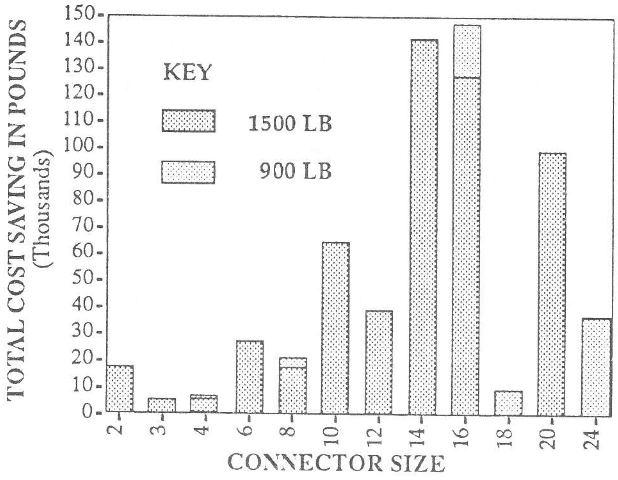


TABLE 4

SUMMARY OF INSTALLED COST SAVINGS (£)

COST SAVING	OIL PROCESSING PLATFORM	GAS WELLHEAD PLATFORM	GAS COMPLEX
Cost of Flanges	80,507	26,158	269,280
Weight Saved (Tonnes)	80.0	14.4	120.0
Total Value of Weight Saved	870,670	157,884	1,319,000

11.0 INTERCHANGABILITY

An 'unwritten' dimensional standard for clamp connectors has been in operation for a number of years and is based on the original design introduced some thirty years ago. The proprietary nature of the design has restricted the more widespread application of the connector, with potential users not always being able to satisfy themselves that components from different manufacturers will function when assembled in the same unit. The dimensional data to be given in the proposed Standard will cover the following basic interchangability requirements:

— Seal ring to hub
— Hubs to clamp
— Bolting to clamp

Thus, specifying that connector components shall be 'in accordance with the Standard' will ensure that the minimum design requirements are met, irrespective of component supplier.

12.0 LEAKAGE CONSIDERATIONS

Only a limited amount of published information is available comparing leakage performance between flanges and clamp connectors. One of the most extensive development programmes carried out, (Ref: 7), reported on the leak rates of a number of mechanical joint types having potential application for high temperature service in nuclear environments.

Leakage mesurement was achieved by measuring the mass of gas entering an evacuated enclosure in unit time; units of measurement being 'clusecs' where 1 clusec is approximately equal to 1 cc of gas per day at stp.

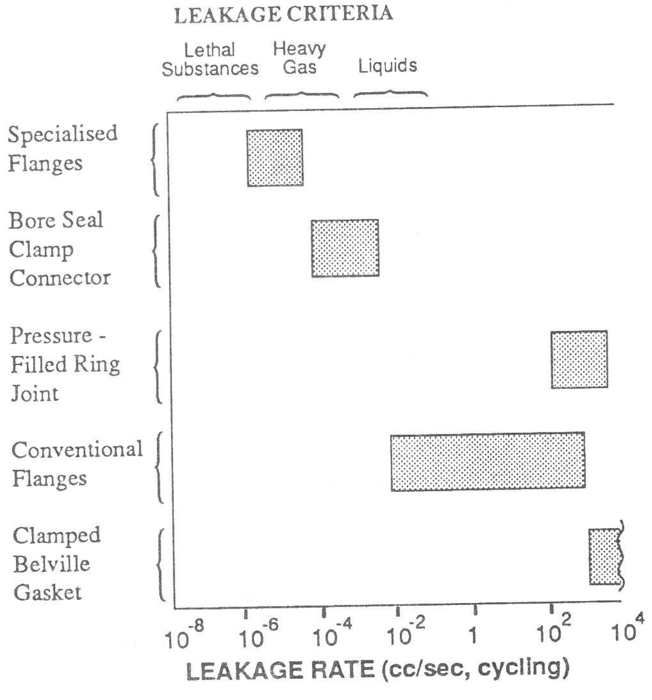
Establishing an 'acceptable' leakage rate (given that complete leak tightness in any joint is impossible to achieve, even though actual leak rates may be almost undetectable), must be a subjective decision. The following broad criteria are proposed:—

Liquids:	10^{-4} cc per sec
Heavy Gas:	10^{-5} cc per sec
Lethal Substances:	10^{-7} cc per sec

From the testing in Ref: (7) a general comparison has been determined to show the average performance of various jointing techniques when subjected to high temperature operation, Figure (15).

FIGURE 15

JOINT LEAKAGE PERFORMANCE FOR GAS AT 350 psi, 600°C SERVICE



13.0 REWORKING ON-SITE

The dimensions of clamp connectors, unlike flanges, allows for the facility of on-site remachining of the angled hub sealing surface and, if necessary the hub face without necessarily altering the pressure rating of the connector. A range of special purpose machines can be purchased or hired for this work which can take one of the following forms:—

— machining the angled sealing surface and hub face to accept the same size ring.

— machining the angled sealing surface of the hub to accept the next largest seal ring (and hence reducing the pressure rating of the connector).

— machining a conventional flange to accept the bore type seal.

This last alternative can provide an effective means of converting a conventional flange into a higher integrity joint without undertaking additional fabrication work.

14.0 API HUBBED END CONNECTIONS

API Specification 16A controls the use of two types of outlet hubs, Type 16B and 16BX. Table (5) gives the nominal sizes and pressure ratings.

TABLE 5

PRESSURE RATINGS AND SIZES OF API TYPE 16B AND 16BX HUBS

PRESSURE (psi)	TYPE 16B	TYPE 16BX
2000	7 1/16, 16 3/4, 21 1/4	
3000	11, 13 5/8, 16 3/4	
5000		2 1/16 thru 21 1/4
10,000		1 13/16 thru 21 1/4
15,000		1 13/16 thru 18 3/4
20,000		1 13/16 thru 11

Type 16B hubs are of the ring joint type and are designed for face-to-face make-up, using Type RX ring gaskets seating in special Type SR ring grooves.

Type 16BX hubs are also ring joint type and are designed for face-to-face make-up with Type BX gaskets.

Whilst hub dimensions have been detailed in API Specification 16A (1st Edition) clamps have been the subject of on-going Committee 16 evaluation. A new standard is proposed specific to ring joint design clamp connectors for Drill through Equipment.

15.0 API PROPOSED CLAMP STANDARD

API, through Committee 16, is proposing minimum design, material and dimensional requirements for clamps used in conjunction with API 16B and 16BX hubs. Proposed nominal clamp sizes and pressure ratings are given in Table (6).

API has adopted an existing hub design that is well known and widely used on Drill through equipment. This is to ensure that clamps meeting the new specification will also fit 'pre-API' hubs already in service. Vice versa, 'pre-API' clamps manufactured to the same number identification system will also fit current API 16A hubs.

The principle of one nominal clamp size to suit a hub with a range of bore sizes has been continued in the interests of safety and consistency. This inevitably means that a clamp may be subjected to relatively low stresses for several sizes of hub bores, but stresses approaching the design limit when used in conjunction with at least one of the hubs which it fits.

The tendency by API is to increase the extent of the specification coverage for clamp connectors, with API 6A Specification similarly now making provision for 'flange or other end connections'.

16.0 PROPOSED PIPELINE CLAMP CONNECTOR STANDARD

The proposed EEMUA Standard for clamp connectors will address, initially, connectors utilising the rib type bore seal suitable for pipeline and wellhead applications. This design of connector has been in widespread use for many years, and has recently undergone extensive independent strain gauge and destructive testing at the National Engineering Laboratory in Scotland, UK.

The results of this testing, as described in outline below, provide the means of verifying the theoretical analysis particularly with respect to the application of external loads.

16.1 Type Testing

Type testing undertaken consisted of;
(i) Independent Third Party approval of the design principles and calculation methods

TABLE 6

PROPOSED API PRESSURE RATING AND SIZES FOR RTJ TYPE CLAMP CONNECTORS

CLAMP Nº	HUB		API MATERIAL DES.
	NOMINAL BORE	PRESSURE (Kpsi)	
1	$1\frac{13}{16}$	10	
	$2\frac{1}{16}$	5	60K
2	$1\frac{13}{16}$	15	
	$2\frac{1}{16}$	10	75K
	$2\frac{9}{16}$	5	
3	$1\frac{13}{16}$	20	
	$2\frac{1}{16}$	15	
	$2\frac{1}{16}$	20	75K
	$2\frac{9}{16}$	15	
4	$2\frac{9}{16}$	10	
	$3\frac{1}{8}$	5	60K
5	$3\frac{1}{16}$	10	
	$4\frac{1}{16}$	5	60K
6	$2\frac{9}{16}$	20	
	$3\frac{1}{16}$	15	75K
	$4\frac{1}{16}$	10	
8	$3\frac{1}{16}$	20	
	$4\frac{1}{16}$	15	
	$7\frac{1}{16}$	5	75K
	9	5	
9	11	3	60K
10	$4\frac{1}{16}$	20	
	$7\frac{1}{16}$	10	
	9	10	75K
	11	5	
11	$13\frac{5}{8}$	3	60K
12	$16\frac{5}{8}$	2	60K
13	$13\frac{5}{8}$	5	60K
14	$16\frac{3}{4}$	3	60K
15	$7\frac{1}{16}$	20	
	11	15	75K
	$13\frac{5}{8}$	10	
18	$21\frac{1}{4}$	2	60K
19	$16\frac{3}{4}$	5	60K
22	$7\frac{1}{16}$	15	
	11	10	75K
25	$7\frac{1}{16}$	15	
	11	10	75K
26	$18\frac{3}{4}$	15	
	$21\frac{1}{4}$	10	75K
27	$18\frac{3}{4}$	10	
	$21\frac{1}{4}$	5	60K
28	11	20	
	$13\frac{5}{8}$	15	75K
	$16\frac{3}{4}$	10	

traditionally used, and (ii) conducting a series of tests to verify the analytical methods. Analysis was completed on connectors from 1in through 24in, supported by strain gauge analysis on 1.5in, 3in, 6in and 12in sizes.

Clamp connectors do not have a specific pressure rating as do conventional flanges; and so a more flexible approach has been adopted, whereby stress characteristics in the hub/clamp combination are used to determine the pressure rating for a given size and material, based on the selected design code. Following satisfactory correlation between the calculations and the test results it has been possible to generate pressure ratings for a range of connector sizes, including combined load cases.

16.2 General Test Requirements

Connector component parts were fully strain gauged including a control gauge for monitoring load conditions to ensure that no gross error existed, either in the strain gauge readings or the load application. Bolt stresses were not monitored on account of these stresses being due only to the make-up bolt torques, although elongation was recorded as an approximate measure of bolt force.

Strain gauges were connected to a data recorder capable of displaying both measured strains and the direction of the principal stresses. The principal stresses were then calculated using the following equations:—

$$S1 = \frac{E\,(e1 + Ve2)}{1 - V^2}$$

$$S2 = \frac{E\,(e2 + Ve1)}{1 - V^2}$$

$$E = 30{,}000 \text{ psi}, \quad V = 0.3$$

All test hubs were fitted with internal strain gauges insulated from the fluid medium and exiting the pipe via a high pressure gland seal.

Figure (16) is a flowchart of the Type Testing undertaken, with typical tests being shown in Figures (17) through (19).

FIGURE 16

FLOWCHART OF TYPE TESTING PROCEDURE

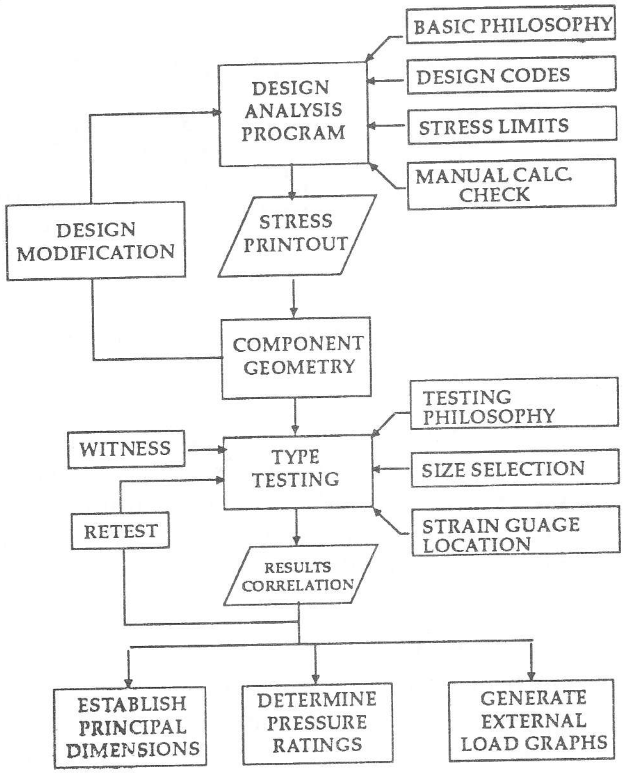


FIGURE 17

6" CLAMP CONNECTOR UNDERGOING COMBINED PRESSURE AND BENDING TEST

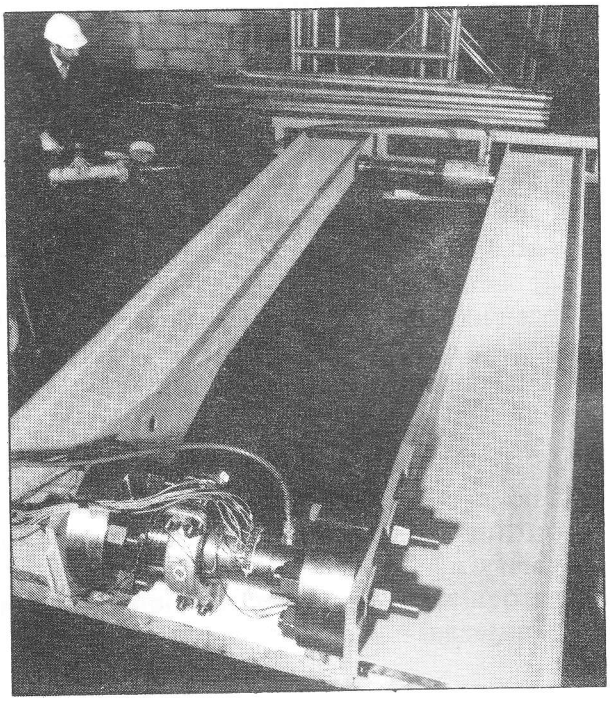


FIGURE 18

3" CLAMP CONNECTOR UNDERGOING COMBINED PRESSURE AND BENDING TEST

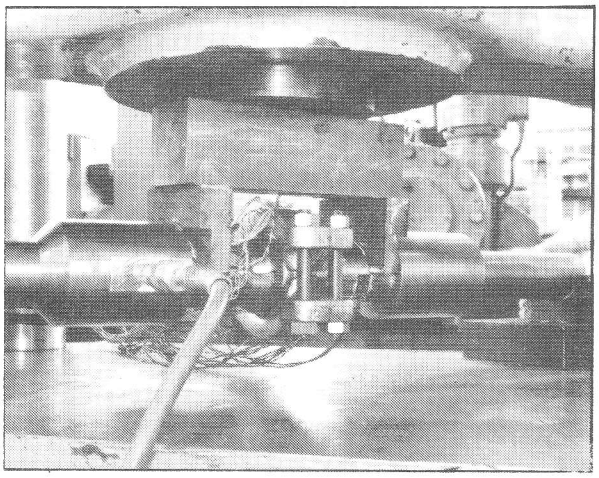
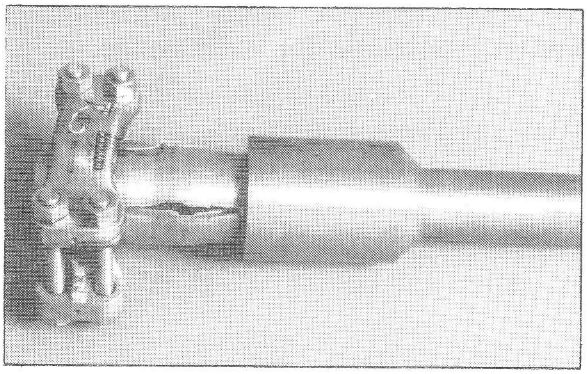


FIGURE 19

FAILURE OF 3" XXS PIPE AT 51,000 psi DESTRUCTION TEST



16.3 Selected Summary of Testing Undertaken

1.5 in Connector

— Pressure test to 10,000 psi
— Axial tension to 50,000 lbf
— Burst test to 50,000 psi (burst not achieved)

3 in Connector

— Axial tension to 170,000 lbf
— Pressure test to 10,000 psi
— Bending test to 20,300 ft lbf
— Combined bending plus pressure to 15,600 ft lbf and 8,000 psi
— Burst test to 51,000 psi (pipe failure achieved)

6 in Connector

— Combined axial tension plus pressure to 170,000 lbf and 10,000 psi
— Combined bending plus pressure to 105,000 ft lbf and 8,000 psi

12 in Connector

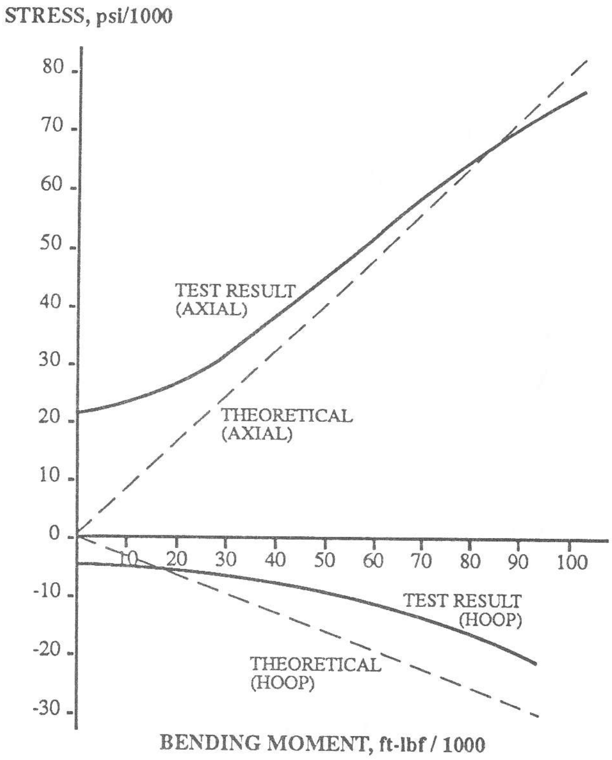
— Pressure test to 7,400 psi (onset of pipe yield)
— Bending test to 218,000 ft lbf
— Burst test to 11,000 psi (fixture failure)

Typical results for a 6in connector obtained from the Type Testing are shown in Figure (20). The graph shows good correlation between the theoretical analysis and the practical testing. An Independent Certifying Authority (Bureau Veritas) has approved the computer analysis for the determination of hub/clamp stresses for all sizes under all combined loading conditions.

This data will form an integral part of the proposed EEMUA Standard allowing, in particular, the generation of external load graphs similar to the format shown in Figure (1). Such graphs are not currently available in existing flange Specifications.

FIGURE 20

STRESS RESULTS FOR 6in HUB SUBJECTED TO EXTERNAL BENDING MOMENT



16.4 Contents of Standard

Scope and Design Basis

Clamp connectors covering the range 0.5in through 24in which incorporate a bore type seal ring. Material coverage would be low carbon steel and stainless steel as commonly in use today. Design basis would be to meet the requirements of ASME, Section VIII, Division 2 (Latest edition).

Dimensional Data

All hub, clamp and seal ring dimensions, tolerances and surface finish requirements necessary to ensure the correct operation and interchangability of components between manufacturers. Dimensional data not given will be established by the manufacturers when working in accordance with the stated design principles. Variations in these dimensions will not affect either interchangability or pressure/temperature ratings.

Design Verification

The manufacturer will be responsible for providing independent design verification based on Type Testing, in support of component dimensions not specified in the Standard.

Materials

Materials for hubs, clamps, seal rings and bolting will be specified with the following categories addressed:—

Hubs

ASTM A105 (Industrial)

ASTM A350-LF2 (Offshore/Industrial)

AISI 4130 (Offshore)

ASTM A182 F316

ASTM A182 F321

Clamps

AISI 4140 (Forged)

ASTM 487-4Q (Cast)

Seal Rings

AISI 4140

AISI 630

AISI 660

Bolting

ASTM A193 B7/A194 2H

ASTM A193 B8/A194 GR8

Process Methods

Acceptable process methods for components will be specified, including:—

— open and closed die forgings for hubs

— forgings for seal rings

— forgings and castings for clamps

Quality Control

Including: Inspection and NDE, certification, Data Book information, and marking.

Selection Criteria

Method by which a connector would be selected for a particular application covering:—

— hub/clamp material compatibility with pipe/vessel
— size and material
— seal ring and clamp selection
— hub/clamp selection to meet pressure/external load requirements

Pressure/Temperature Tables

Tables of minimum pressure ratings for connectors of given materials operating at various temperatures.

External Loading Graphs

A series of pressure versus external loading graphs providing a safe minimum operational envelope for each connector size.

Coating and Protection

Including PTFE colour coating for seal ring identification and dry lubricant properties. Corrosion protection requirements for hubs, clamps and bolting.

17.0 CONCLUSION

The clamp connector can provide an effective 'engineered' solution to higher pressure pipe jointing requirements. The pressure energised bore type seal can out-perform conventional flanges, although more care must be exercised during assembly since joint integrity depends upon seal and mating hub surface finish. Unlike conventional flanges seal integrity is insensitive to bolting make-up torque, thereby reducing the likelihood of leakage.

In its development from the established position in the Offshore Industry, where the connector can show significant weight and cost savings over flanges, into the more general Industrial market the connector is beginning to demonstrate its technical and overall installed economic benefits.

However, in order to gain widespread acceptance potential users will wish to have a reference Standard by which to guarantee interchangability of components between manufacturers. By also including in the Standard external loading graphs based on the results of independent strain gauge and destructive testing, a significant contribution to the pipeline designers' toolkit will be made for both offshore and Industrial piping design.

NOTATION

EEMUA — The Engineering Equipment and Materials Users Association.

ASME — American Society of Mechanical Engineers.

ANSI — American National Standards Institute.

AWHEM — Association of Wellhead Equipment Manufacturers.

API — American Petroleum Institute.

UKOOA — United Kingdom Offshore Operators Association.

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AUTHOR

Dr Dennis Briaris has a PhD in Fluid Dynamics and an Honours Degree in Mechanical Engineering. He has held a number of Senior Managerial positions in industry, and has also assisted with the preparation of Technical Standards both in the UK and the USA. In 1987 he formed Techlock Limited, a Company specialising in the design and manufacture of Clamp Connectors for both offshore and Industrial Applications. Techlok Limited is now a subsidiary of Furmanite PLC, where Dr Briaris holds the position of Director, Pipeline Equipment Operations.