CHAPTER

8

INTERFACE WITH STATIONARY EQUIPMENT

The main purpose of piping stress analysis is to ensure the structural integrity of the piping and to maintain the operability of the system. The latter function is mainly to ensure that the piping forces and moments applied to connecting equipment are not excessive. Excessive piping loads may hinder the proper functionality of the equipment. The function of maintaining system operability requires the investigation of the interface effects with connecting equipment. There are three main interface effects between piping and connecting equipment:

(a) *The loads imposed on piping from equipment.* This involves mainly the expansion of the equipment. The expansion of a vessel, for instance, can be large enough to have a considerable effect on piping stress. Occasionally, the vibration of equipment, although not detrimental to the equipment itself, can amplify through the piping to create a problem.
(b) *Flexibility of equipment.* The equipment is generally considered a rigid member in the analysis of piping. However, the flexibility, either from equipment supports or from the equipment itself, may significantly affect the results of the analysis.
(c) *Effect of piping loads on equipment.* The equipment is generally designed for loads based on the function of the equipment. It may or may not consider the potential piping loads that it might eventually have to take. It is important to ensure that the piping loads are acceptable to the equipment whether the equipment is designed for any piping loads or not.

This chapter discusses the interfaces of piping with stationary equipment, such as valves, heaters, and pressure vessels. The interfaces with rotating equipment will be discussed in the next chapter.

8.1 FLANGE LEAKAGE CONCERN

The possibility of flange leakage occurring well before the failure of the pipe or the flange is a major concern of piping engineers when the allowable piping expansion stress-range runs well over the yield strength of the piping material. Even with the structural integrity of the flange intact, the system is still not functional if the flange tightness is not maintained. Flange leakage is a very complex problem involving many factors. Inadequate pressure rating, poor gasket selection, insufficient bolt loading, temperature gradient, bolt stress relaxation, piping forces and moments, and so forth, can all cause leakage at a flange. In this chapter, we will limit our discussion to the effects of piping forces and moments.

The pipe loads applied to the flange have received considerable attention from piping engineers. Many new construction projects require that these piping loads be evaluated to assure the tightness as well as the structural integrity of the flanges. This problem has been broadly investigated. Blick [1], Koves [2], Tschiersch and Blach [3] have presented some theoretical treatments, and Markl and

George [4] have conducted a large number of experiments on the subject. From the extensive tests made on 4-in. Class 300 American Society of Mechanical Engineers (ASME) B16.5 [5] flanges, Markl and George have found that "even under unusually severe bending stresses, flange assemblies did not fail in the flange proper, or by fracture of the bolts, or by leakage across the joint face. Structural failure occurred almost invariably in pipe adjacent to the flange, and in rare instance, across an unusually weak attachment weld; leakage well in advance of failure was observed only in the case of threaded flanges." These test results pretty much confirm the expectation of the conventional flange selection process. In the conventional design process, the flange is selected based on the pressure rating. If the design pressure is smaller than the pressure rating of the flange, then the flange is expected to function just as well as the pipe of the same pressure rating.

The test results by Markl and George are so convincing that, for B16.5 flanges, as long as the design or service pressure is within the pressure rating of the flange, the flanges are considered capable of taking the same pipe load as that of the pipe-to-flange welds with appropriate stress intensification factors applied [6]. The stress intensification factor for welding neck flanges is 1.0. However, due to the large varieties of flanges involved in many different types of industries, the question remains as to whether the 4-in. Class 300 data is applicable to all flanges.

In this "design by analysis" era, engineers are generally not satisfied with "rule of thumb" approaches. A more definite evaluation approach is preferred. That is when the confusion starts. Several methods have been proposed and used, yet they give widely different results. To determine the merits of each method, the standard flange design procedure will be briefly discussed first.

8.1.1 Standard Flange Design Procedure

The standard flange design procedure was first developed in the 1930s and was adopted by the ASME Code for Unfired Pressure Vessel in 1934 [7, 8]. Through decades of practices and refinements, the current ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, Appendix 2 design rules, generally referred to as Appendix 2 rules, have been universally adopted for the design of flanges subject to internal pressure. Countless flanges have been designed by this simple and easy-to-use cookbook approach.

Using an integral type flange as an example, the flange is idealized into three ring sections as shown in Fig. 8.1. It consists of the flange ring, hub ring, and an effective length of pipe section. The stresses at each ring section are calculated by applying a circumferentially uniform bending moment at the face of the flange ring. This bending moment represents the total loading applied at the flange. Once

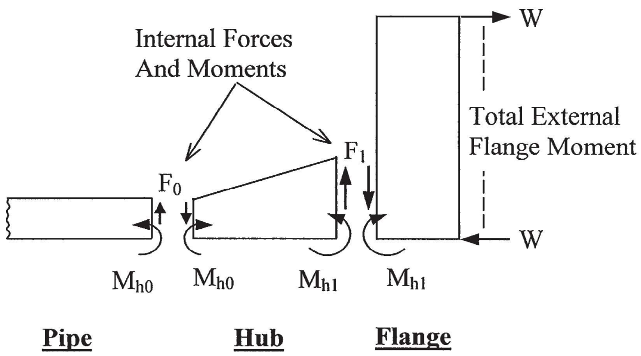

FIG. 8.1
IDEALIZED FLANGE ANALYTICAL MODEL

this moment is known, the stresses at the flange and at the hub are calculated with code formulas and charts. The main task of the designer is to determine the magnitude of this bending moment under seating and operating conditions. Appendix 2 rules do not cover the pipe force and moment.

Figure 8.2 shows the loadings on a flange subject to internal pressure. Their magnitudes are calculated as follows:

$$\begin{aligned}
H_D &= \frac{\pi B^2}{4}P, \quad \text{Pressure end force from pipe (operating condition)} \\
H_T &= \frac{\pi(G^2-B^2)}{4}P, \quad \text{Pressure force at flange face (operating condition)} \\
H_{G,2} &= b\pi Gy, \quad \text{Gasket force for seating (seating condition)} \\
H_{G,1} &= m(2b)\pi GP, \quad \text{Gasket force for sealing (operating condition)}
\end{aligned} \tag{8.1}$$

where b is the effective gasket seating width, m is the gasket factor, and y is the minimum required gasket seating stress. These b, m, and y values are given by the Appendix 2 rules. Due to the rotational deformation (cupping) of the flange, gasket stress is not uniform. In general, the outer edge receives a much higher stress than average, thus making the outer rim portion of the gasket the effective area. The effective gasket seating width is roughly equal to one-half of the gasket contact width, and the gasket load diameter, G, is located between the outer contact diameter and the mean contact diameter of the gasket.

The bolt load, W, is the summation of H_D, H_T, and H_G under each of the two conditions. However, to guard against overstressing the flange from the actual tightening of the bolts, the design bolt load is further adjusted by the total allowable bolt force. The total allowable bolt force is calculated by

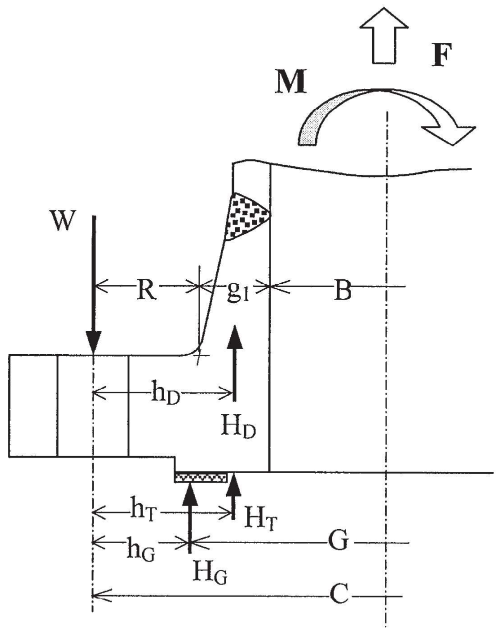

FIG. 8.2
LOADINGS AT FLANGE

multiplying the total bolt root areas by the bolt allowable stress. The design bolt load is taken as the average of the total allowable bolt force and the bolt force required by the above H_D, H_T, and H_G. This additional bolt force, over what is required for pressure and seating, balances with H_G, thus causing H_G to also increase by the same amount.

The total flange moment is calculated by using the bolt centerline as the pivot. The moment arms are taken as

$$\begin{aligned} h_D &= R + g_1/2 \\ h_G &= (C - G)/2 \\ h_T &= h_G + (G - B)/4 \end{aligned} \tag{8.2}$$

The total flange moment becomes

$$\begin{aligned} M_O &= H_D h_D + H_T h_T + H_{G,1} h_G && \text{Operating condition} \\ M_G &= (H_{G,2} + H_B) h_G && \text{Seating condition} \end{aligned} \tag{8.3}$$

H_B is the additional bolt force of the actual design bolt force used over the required bolt force based on H_D, H_T, and H_G. Once M_O and M_G are determined, flange stress and hub stress are calculated using the formulas and charts given. Three stresses — longitudinal hub stress, radial flange stress, and tangential flange stress — are calculated. The actual stress calculation is beyond the scope of the book. We will only use the procedure outlined to discuss the various methods proposed for evaluating the pipe force and moment.

8.1.2 Unofficial Position of B31.3

ASME B31 Mechanical Design Committee Report [9] issued for B31.3 [10] has stipulated that the moment, M_L, to "produce leakage" of a flanged joint with a gasket inside the bolt circle can be estimated by

$$M_L = \frac{C}{4}(S_b A_b - P A_p) \tag{8.4}$$

where A_p is the area of the circle to the outside of gasket contact, S_b is bolt stress, and A_b is total root area of flange bolts. This equation uses a rigid flange model having the pipe moment resisted by the bolt and gasket combination. By idealizing the bolt force and also the sealing force as distributed line loads located around the bolt circle as shown in Fig. 8.3(b), the residual sealing force per unit circumference, after subtracting the pressure force, is uniform and equal to $(S_b A_b - P A_p)/(\pi C)$. With a

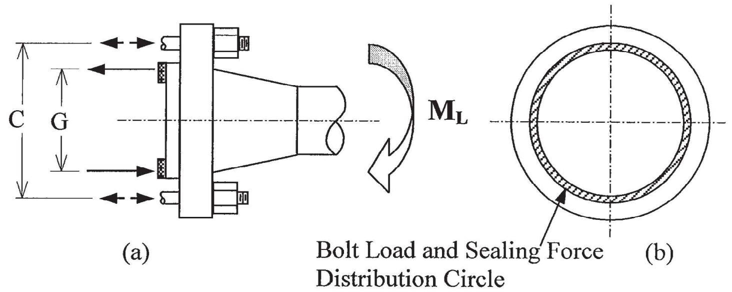

FIG. 8.3
GASKET AND BOLT FORCES DUE TO BENDING MOMENT

bending moment applied, the bolt force and thus the sealing force will be linearly redistributed across the diametrical direction. The maximum and minimum forces per unit circumference due to the moment occur at two extreme points and equal to $M_L/(\pi C^2/4)$. The moment will cause the sealing force at one end to increase and at the other end to decrease. The flange is assumed to leak when the sealing force, after subtracting pressure and moment forces, at any point of the circumference is zero. In other words, the flange will leak when $(S_b A_b - PA_p)/(\pi C) - M_L/(\pi C^2/4) = 0$.

This equation is not officially included in the code, but is being used by some engineers. However, because it is not a design formula, it is occasionally misapplied. In using this formula, there are a few things that need to be noted.

(1) Many engineers have mistaken the M_L as the allowable moment, which is not correct. Equation (8.4) is used to predict the moment to produce leakage. We do not normally design a flange to leak. Therefore, to use the formula, some type of margin has to be included. This can be done by applying a proper safety factor or to set aside some residual gasket loading for maintaining tightness. The latter is given by Appendix 2 rules [8] as $H_{G,1} = m(2b)\pi GP$, which is also given in Eq. (8.1). In other words, the total bolt force has to be subtracted by $H_{G,1}$ as well as PA_p to become the surplus sealing force before being converted to the allowable bending moment.

(2) Bolt stress, S_b, can be taken as the basic allowable stress of the bolt. However, it should be noted that the flange might not be designed for the full allowable stress of the bolt. Assuming the flange is designed for the full basic allowable stress of the bolt, then the use of $S_b = (S_c + S_h)/2$ may be considered as having a safety factor of 1.5. This is because the allowable stress is generally increased by 50% when dealing with piping loads. (See also item (B) in Section 8.1.3). Occasionally, engineers might even use the empirical stress [11] actually applied on the bolt while tightening. This stress can be as high as twice the allowable stress for 1-in. (25 mm) bolts as an example. This single application tightening stress shall not be considered as available for repetitive piping loads.

(3) Figure 8.3 shows the relationship of gasket force, bolt force, and the pipe moment. Theory and experiments have shown that with a given direction of moment, the load at each particular bolt can either increase or reduce [1], depending on the relative stiffness of the flange, gasket, and bolt. The bolt load in response to the pipe moment changes very little. The pipe moment is resisted mainly by the gasket. Therefore, it appears to be logical to use the gasket loading diameter, G, instead of the bolt circle diameter, C, in Eq. (8.4).

Equation (8.4) is simple and its intent clear, but is not very easy to apply. Judging from the above listed concerns, clear specifications are needed for using the equation.

8.1.3 Equivalent Pressure Method

The fact that we have a set of very reliable formulas and rules for designing flanges subject to internal pressure provides the incentive for evaluating the pipe force and moment based on these rules. To use the pressure design procedure, the first thing needed is to find the relationship between the internal pressure and the pipe force and moment. The M. W. Kellogg [12] has suggested an equivalent pressure approach. The approach assumes that the action of the moment and force is equivalent to the action of the pressure, which produces a gasket stress that is the same as the gasket stress produced by the force and the moment. Figure 8.4 shows this equivalence relation. By letting S_F represent the gasket stress due to force, S_M the maximum gasket stress due to moment, and S_P the gasket stress due to equivalent pressure, we have

$$S_F + S_M = S_P$$

or

$$\frac{F}{\pi Gb} + \frac{M}{\pi G^2 b/4} = \frac{\pi G^2 P_e/4}{\pi Gb}$$

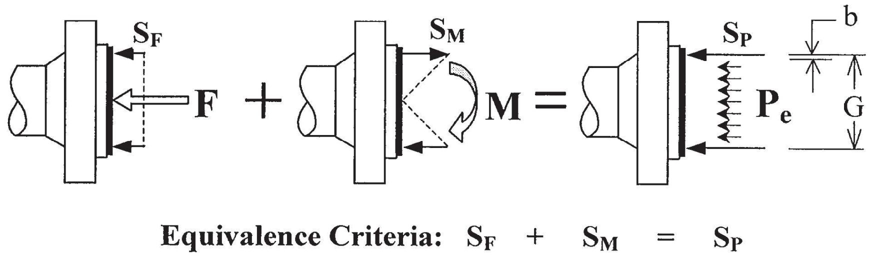

FIG. 8.4
EQUIVALENT PRESSURE DUE TO FORCE AND MOMENT

$$\text{i.e.,} \quad P_e = \frac{4F}{\pi G^2} + \frac{16M}{\pi G^3} \tag{8.5}$$

Once equivalent pressure P_e is determined, flange stress can be calculated using Appendix 2 formulas and rules. The above equivalent pressure formula is considered conservative because the maximum gasket stress due to moment occurs only at the small areas near two diametrically extreme points. The bending moment is the dominant loading.

In practice, equivalent pressure is combined with design pressure to become the total equivalent pressure. That is,

$$P = P_d + P_e \tag{8.6}$$

The equivalent pressure approach is very popular among piping engineers. There is little doubt about the usefulness of this approach, but its applications are still not uniform. The following are two major diversities:

(A) Rating table lookup. This approach has been adopted by a number of computer software packages. It uses the rating table as the sole evaluation tool. Once the total equivalent pressure is calculated, the rating table is checked for this total pressure. The pipe force and moment is considered acceptable to the flange if the total equivalent pressure is within the flange rating pressure. (See Table 1.1 for pressure ratings.) This method is simple and conservative, but has very limited practical use. It can be used for quick checking the acceptability of the piping load. However, because it is exceptionally conservative when used as a definite evaluation rule, it usually creates — rather than solves — problems.

In designing a piping system, the designer generally selects the flanges and valves based on the pressure rating. For a given design pressure, the flange selected will have a rating pressure equal to or greater than the design pressure. It is not unusual to select the flange having the rated pressure the same as or close to the design pressure. In this case, the flanges are still expected to perform satisfactorily under moderate piping forces and moments, in addition to the design pressure. Experiences do confirm this expectation. Had the piping system in this case been evaluated for leakage using equivalent pressure with the rating table lookup procedure, most of the flanges would have been disqualified even with minimal piping loads. The total equivalent pressure, including design pressure and pipe forces and moments, is generally greater than the rating pressure, which is almost entirely taken up by the design pressure in this case. The flanges would have to be replaced with ones of higher rating. This unusual requirement can generate a shocking impact to the plant involved. Some systems that have gone through this type of leakage evaluation have installed different classes of flanges at different locations on the same piping system. This creates not only waste and confusion, but also the quality assurance problem of installing the right flange at the right location.

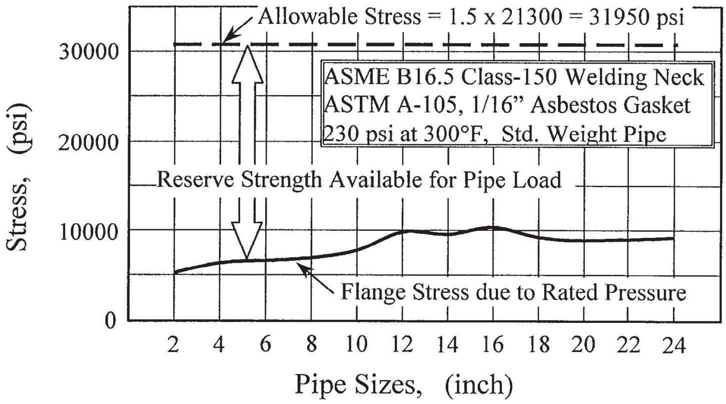

FIG. 8.5
ASME B16.5 CLASS 150 FLANGES

It is to be emphasized that the rating table lookup approach is a quick and conservative evaluation method. However, a flange connection is still most likely satisfactory even if it fails to pass the rating table look up evaluation.

(B) Flange stress calculation. In addition to providing sufficient strength against the rated pressure, the rating of the standard flanges [5, 13–15] also provides some reserve strength to cover other potential loads, including pipe forces and moments. The reserve strength, as judged by ASME Section-VIII Appendix 2 rules, is not uniform across sizes and classes. Smaller sizes and lower pressure classes generally have higher reserve strengths. Figures 8.5, 8.6, and 8.7 show these trends [16]. The exact amount of reserve strength can only be determined by calculating the flange stress.

The stresses given in Figs. 8.5, 8.6, and 8.7 are calculated according to ASME Section-VIII Appendix 2 procedures. Three stresses are calculated: longitudinal hub stress (S_H), radial flange stress (S_R), and tangential flange stress (S_T). The allowable value for these calculated stresses depends on the nature of the loading. For the pressure design, the allowable for S_H is $1.5S_h$, and the allowable for

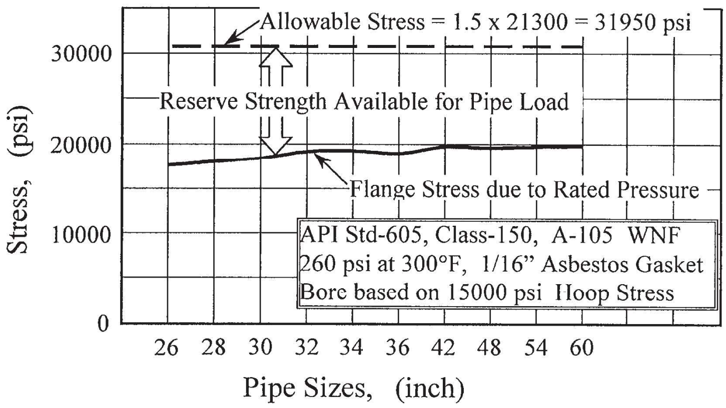

FIG. 8.6
API STD-605 CLASS 150 FLANGES

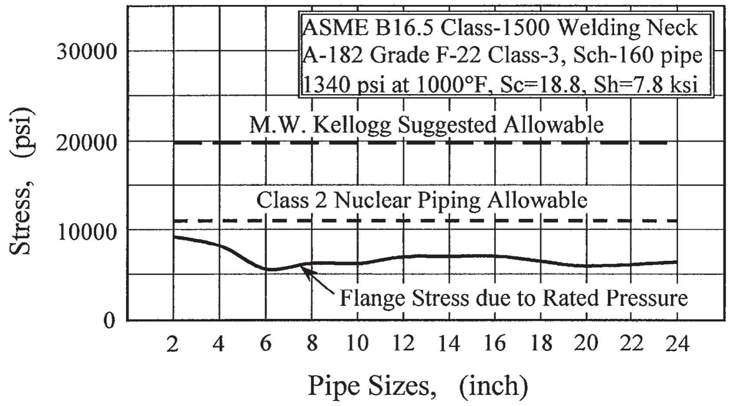

FIG. 8.7
ASME B16.5 CLASS 1500 FLANGES

each of S_R and S_T is $1.0S_h$. S_h is the allowable stress of the flange material at design temperature. When the piping load is included, the allowable stresses are increased to $1.5S_h$ for each of the three stresses. These same allowable stresses are applicable to the flanges subject to steady static loads as well as thermal expansion loads. These allowable values, however, are considered too small when dealing with thermal expansion load ranges at high operating temperatures. Based on experiences, M. W. Kellogg [12] suggested an allowable of $1.5(S_c + S_h)/2$ for bolt stress and for each of the three calculated flange stresses, for loads including thermal expansion. This is 1.5 times of the average of hot and cold allowable stresses. S_c is the allowable stress at ambient temperature.

8.1.4 Class 2 Nuclear Piping Rules

The Class 2 nuclear piping code [6] has comprehensive rules regarding the evaluation of flanged connections subject to piping loads. Based on this code, the flanged connections can be evaluated by any of the following methods.

1. Any flanged joint

This method is applicable to any type of flanged joint. It is based on the equivalent pressure approach described in Section 8.1.3. Either the rating table lookup method or stress calculation method can be used.

The rules are the same as in Section 8.1.3, except that both bending and torsional moments are evaluated, but separately. The longitudinal pressure membrane stress at the smaller end of the hub is added to the longitudinal hub stress calculated by the standard formula. The allowable stresses are all set to $1.5S_h$ for each of the three calculated flange stresses. As usual, the calculated three stresses are longitudinal hub stress, radial flange stress, and tangential flange stress. These allowable stresses are significantly different from the values suggested by Kellogg on flanges operating at high temperatures as shown in Fig. 8.7. This, however, should not create any difficulty for nuclear piping due to the moderate temperature environment in water reactor plants. When dynamic loads are included, the equivalent pressure is halved, thus doubling the allowable forces and moments.

2. Standard flanged joints at moderate pressures and temperatures

Flanged joints confirming ASME B16.5, MSS SP-44, API Std-605, or AWWA C207 Class E (275 psi), with design and service pressures below 100 psi, and design and service temperatures below 200°F, have an allowable piping moment of

$$M_A \le \frac{A_b S_b C}{4} \tag{8.7}$$

This formula is similar to Eq. (8.4) by ignoring the pressure term. By ignoring the pressure term, it may appear to be very un-conservative. However, the formula is based on two prerequisites that make it sufficiently conservative: (1) the flange has to be manufactured by an established standard; (2) the operating pressure is less than 100 psi, and the operating temperature does not exceed 200°F. For a flange made by one of the standards listed above, the minimum rating pressure is 235 psi for Class 150 or equal. This means that the 100-psi limiting pressure is less than one-half of the rated pressure. From Fig. 8.5, it is clear that flange stress due to one-half of the rated pressure for Class 150 ASME B16.5 flange is negligible compared with the allowable stress. By taking into account the inherent conservatism of the standard flange, it is obvious that Eq. (8.7) is still fairly conservative. The allowable moment is doubled when the dynamic loads are included.

One thing that the code does not mention is the special case with API Std.-605 Class 75 flanges. API-605 Class 75 has a rating pressure of only 140 psi, which is very close to the 100-psi limiting pressure. Therefore, there is a concern that Eq. (8.7) may not be sufficiently conservative for API-605 Class 75 flanges.

3. ASME B16.5 flanged joints with high strength bolting

ASME B16.5 was available before the standard flange design procedure was finalized. Without clear-cut procedures, the design tends to be more conservative. This makes B16.5 more conservative than other standards. This evaluation is applicable only to B16.5 flanges using bolting material having an allowable stress of not less than 20,000 psi at 100°F. For flanged joints in this category, the allowable bending or torsional moment (considered separately) is

$$M_A \le \frac{C}{4} S_{bm} A_b \frac{S_y}{S_{yn}} \tag{8.8}$$

where S_{bm} = 12,500 psi (86.2 MPa) is fixed for the reserve bolt stress available for pipe moment and S_{yn} = 36,000 psi (248.2 MPa) is fixed for the nominal yield strength of the flange material. S_y is the yield strength of the flange material at design temperature. The allowable moment is doubled when the dynamic loads are included.

8.2 SENSITIVE VALVES

There are many valves required in a plant to effectively manipulate and control the process flows. Some of these valves are sensitive to piping loads due to either their limited stroking forces or their weaker cross-sections. A safety relief valve, for instance, operates upon the balance of the pressure force and the set spring force. In a sense, the set spring force determines at what pressure the valve shall pop open to safeguard the system. The pipe force, if large enough, may strain the valve body so much as to create binding between the valve stem and its guide. This will create an obstruction or generate an excessive friction force, which is not included in the original balance equation of pressure force and spring force. As a result, the valve will open at a higher pressure than originally set. The same phenomenon is also applicable to some of the control valves.

Normally, valves are made as strong as the connecting pipe of the same size and thickness. However, due to control characteristics requirements and possibly economic reasons, safety relief valves and control valves are often one size smaller than the main connecting pipe as shown in Fig. 8.8. The inlet of a safety relief valve is also generally one size smaller than the outlet. This makes the inlet connection two sizes smaller than the main outlet piping.