EVALUATION OF FLANGED CONNECTIONS DUE TO PIPING LOAD

D. L. McKeehan, Division Manager
Piping Mechanical Division

L-C. Peng
Principal Staff Engineer

The M. W. Kellogg Company
Houston, Texas

ABSTRACT

The pipe loading applied to flanged connections has been a piping engineer's constant concern. This problem has been treated by various authors. The latest tendency in evaluation is leaning toward the so called equivalent pressure approach. The paper describes briefly the two popular approaches, namely rating table method and direct stress calculation method. With extensive stress calculations performed on the standard flanges, it is shown that the standard flanges not only can take the rating pressure allowed, but also have reserve strength available to resist significant pipe loads. Caution is also given to non-standard flanges designed by ASME B&PV Code rules. Without proper consideration of pipe load, a custom designed non-standard flange is prone to leak.

NOMENCLATURE

D - Outside diameter of pipe

t - Thickness of pipe

P - Internal pressure

Pe - Equivalent pressure due to pipe load

M - Bending moment from pipe

F - Axial force from pipe

G - Diameter at location of gasket load reaction

b - Effective gasket width

SH - Longitudinal stress in hub

SR - Radial stress in flange

ST - Tangential stress in flange

Sc - Basic allowable stress at ambient temperature (cold)

Sh - Basic allowable stress at operating temperature (hot)

INTRODUCTION

The pipe load applied to a flanged connection has been a constant concern to piping engineers. An excessive pipe load can cause leaking or even failure. This problem has been investigated by Markl and George, Blick, and others (1,2). From the extensive tests made on 4 inch class 300 ANSI flanges, Markl 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 the pipe adjacent to the flange, and in rare instances, across an unusually weak attachment weld. He stated that leakage well in advance of failure was observed only in the case of three [illegible] flanges. Although the test results were definite, they are no means telling [illegible] that the pipe load on a flanged connection should not be a concern. Because of the wide ranges of factors such as size, class, type, and material involved, a leak or failure can still occur at flange connections if the pipe load is excessive. Proper precautions should be taken to ensure the integrity of the connections.

The current tendency of evaluating the pipe load is leaning toward the equivalent pressure approach. This conservative approach is relatively simple. It is especially suitable for high volume daily production analyses. However, two considerably different methods have emerged from the application of the equivalent pressure. One method uses the equivalent pressure to select the suitable class of flange from the rating table. The other performs stress check on the flange to make sure it will take the equivalent pressure. The paper discusses in detail the approaches used in evaluating the pipe load. It presents the results of extensive stress analyses performed on standard flanges to give the readers a feeling on the magnitude of reserve strength available from these flanges. The paper cautions that proper pipe load should be included in designing a special flange for piping applications.

METHOD OF EVALUATION

A tight joint can be maintained when there is a sufficient gasket pressure to ensure at least a continuous line contact around the entire circumference at all conditions. This requires that the bolts be strong and stiff enough to hold the joint, the gasket be resilient enough to allow axial movement yet strong enough to prevent crushing or blowout, the flange be strong and stiff enough to prevent excessive cupping, thus making a continuous contact possible.

A flange is one of the few items in piping that requires the consideration of both stress and strain. It is also one of the few items that depends heavily on other components to make it functional. Designing a flange deals not only with the flange proper but also with bolt and gasket. The task is very complex. It involves setting up the gasket criteria, seating requirement, stress calculation procedure, allowable stresses, and so forth. Fortunately, the ASME Boiler and Pressure Vessel Code (3) has provided a complete and detailed rules for designing the flange against internal pressure. With the availability of this reliable procedure for internal pressure design, one will naturally attempt to use the same procedure for the pipe load evaluation. The first step toward this utilization is to convert the pipe load into equivalent pressure load.

Assuming the equivalent pressure is the pressure that will create the same amount of gasket stress as the pipe load does (4), we have

$$\frac{F}{\pi G b} + \frac{M}{\frac{\pi}{4} G^2 b} = \frac{\frac{\pi}{4} G^2 Pe}{\pi G b} \qquad (1)$$

After simplification, Equation (1) becomes

$$Pe = \frac{4F}{\pi G^2} + \frac{16M}{\pi G^3} \qquad (2)$$

Above equivalence is believed to be conservative because the maximum gasket stress produced by the pipe load exists only at the extreme edge of the gasket, whereas the stress generated by the pressure is uniform. Equation (2) can be considered as being in the same spirit as the area replacement approach used in the design of the reinforcement at branch connections. It is simple and reliable.

By converting the pipe load to equivalent pressure, the problem becomes the standard problem of flange under internal pressure which can be readily solved by ASME B&PV code procedure. The total equivalent design pressure becomes

$$P_{total} = P + Pe \qquad (3)$$

Equation (3) has been used by the petrochemical industry (4) for more than 30 years, and was formally adopted by the nuclear pipe Code (5) for evaluating class 2 piping flange connections. In fact Equation (2) has gained such a popularity that it is also used in converting torsional moment to equivalent pressure (5), although the nature of the stress generated from torsion is different from the one created by pressure.

APPLICATION OF EQUIVALENT PRESSURE

With the total equivalent pressure determined, the rest of the evaluation is straight forward. One can perform a stress calculation based on this pressure or simply select a flange that has the rating suitable for this pressure. Although both methods are acceptable the results can be quite different. This difference can be illustrated by the following example: Assuming there is a moderate pipe load of 8400 lbf-ft (11390 N-M) bending moment acting on a 8.625 inch (219mm) outside diameter, 0.322 inch (8.18mm) thick pipe operating at 300°F (149°C) temperature and 200 psi (1379 KPa) internal pressure. The bending moment will produce a bending stress of 6000 psi (41364 KPa) in the pipe. If a flanged connection is needed at this point, a design engineer will take the operating pressure and temperature and look up the rating table from standards such as ANSI B16.5 (6) to select a suitable class for the application. An ANSI B16.5 class 150 flange, made from A-105 material, would be selected in this case by disregarding the pipe load. However, based on Equation (2) this pipe load is equivalent to a pressure of 527 psi (3633 KPa). Adding this equivalent pressure to the internal pressure, the total pressure to be considered is 727 psi (5012 KPa). By using the rating table selection method a class 400 flange would have to be used, whereas a stress calculation will show the originally selected class 150 flange is satisfactory.

From the above example, it is clear that the rating table approach which ignores the reserve strength of a standard flange is overly conservative. In fact it probably will disprove most of the systems which are in existence because 6000 psi (41364 KPa) pipe stress is by no means excessive. The stress analysis method is much more realistic and appears to have become a preferred approach. However, it should be noted that for internal pressure loading the rating table should always be used for compliance with Code requirements (7).

In the stress calculation approach, the standard method (3) of calculation is to evaluate the following stresses:

(a) Longitudinal stress in hub, SH
(b) Radial stress in flange, SR
(c) Tangential stress in flange, ST

These stresses can be easily calculated using the Code Rules (3) based on the bending moment acting at the flange bolt circle circumference. However, the most important point for practical design is to establish a proper allowable stress. For example, the Class 2 Nuclear Piping Code (5) limits each of the above stresses to 1.5 times the basic allowable stress at operating temperature. It also requires that longitudinal pressure stress be included in the longitudinal hub stress. The basic allowable stress is the code tabulated stress mainly to be used in internal pressure design.

Most flange designs, other than for nuclear as above, is covered by ANSI/ASME Sec. VIII - Div. 1 (3) for vessels, and by ANSI/ASME B31 Piping Codes (7) for piping. Rules for allowable stresses are contained in each of the pertinent Codes. The piping Codes refer back to ref. (3) for stress calculation procedure. Allowable stresses for flanges (other than cast iron flanges) are generally as shown below.

S_H equal to or less than 1.5 S_f

S_R equal to or less than S_f

S_T equal to or less than S_f

$(S_H + S_R)/2$ equal to or less than S_f

$(S_H + S_T)/2$ equal to or less than S_f

Where S_f = basic allowable stress for flange material at design temperature, as defined by the applicable Code.

STANDARD FLANGES

Among the various standard flanges available in the industry, the ANSI B16.5 (6) and API 605 (8) are the most popular ones. The former covers up to 24-inch (610mm) with classes ranging from 150 to 2500, the latter covers 26-inch (660mm) through 60-inch (1524mm) with 75, 150, and 300 classes. In this section the reserve strength of these two categories of flanges will be investigated.

In selecting a standard flange for a specific application, the engineer will look-up the rating table from the standard to find the particular class of flange that offers a pressure rating higher than the design pressure of the system at the design temperature. The flange so selected will be adequate for the design pressure and at the same time provides some reserve strength to resist the pipe load. The magnitude of the reserve strength varies with the flange size, type, class, material and temperature. Therefore, no rule of thumb can be used to estimate the amount of reserve strength available. API STD 605 does clearly state that a bending moment capable of producing a longitudinal pipe stress equal to one half of the basic allowable stress is included in the pressure rating at rated temperature. However, the amount of pipe moment that can be absorbed at other temperatures is not defined.

To present a general idea of the reserve strength available in a standard flange, stress calculations on typical examples were performed and the results charted in Figures 1 through 5. Figure 1 shows the relation between the flange stress produced by the rated pressure load and the reserve strength available to absorb the pipe load for ANSI B16.5 class 150 flanges. In general, this class of flanges has a reserve strength much greater than the strength required to resist the rated pressure. The reserved strength can be converted to the bending stress at pipe as shown in Figure 2. This bending stress depends on the actual thickness of the connecting pipe; but in general is about 8000 psi (55000 KPa) for class 150 flanges. The reserve strength is considerably lower for the API STD 605 flanges as shown in Figure 3. It also decreases for the higher class flanges. Figure 4 shows the relation for ANSI B16.5 class 300 flanges. It is clear that the reserve strength of a class 300 slip-on flange decreases rapidly as the pipe size increases. For the weld neck flanges, because the flange bore is assumed to be the same as the inside diameter of the pipe, the reserve strength depends greatly on the thickness of the connecting pipe. Take a 24-inch (610mm) flange for instance, point A represents the stress with XS weight connecting pipe while point B represents the stress with scheduled 40 connecting pipe. Figure 5 shows the situation in high temperature and high pressure applications. In the high temperature range, the Class 2 Nuclear Piping allowable for example, predicts very little reserve strength available for the ANSI B16.5 class 1500 flanges. The Kellogg allowable on the other hand assumes considerable reserve for the pipe load. In the non-nuclear applications, it appears to be logical to limit the sustained loading to Class 2 Nuclear Piping allowable and to use the Kellogg allowable for the thermal expansion loading.

SPECIAL FLANGES

Flanges designed for special conditions have suffered the most leakage in actual applications. This is especially true in low pressure flanges that are specially designed for a cost savings over standard flanges. The difficulty experienced by the special flange is largely due to the omission of the pipe load in the design. Although ASME B&PV Code, Section VIII, Div, I, Appendix 2 (3) states, "Proper allowance shall be made if connections are subject to external loads other than external pressure," it is often disregarded due to lack of definite value imposed. A special flange should be designed with proper allowance provided for pipe load. This pipe load can be either calculated from the actual piping system or, to be in line with standard flanges, taken as the bending moment sufficient to produce a bending stress equal to one half of the basic allowable stress at the connecting pipe. After taking the above pipe load into consideration, one will find that the special flange, designed for low pressure application, is not much different in thickness than the standard flange available for higher pressure service. The advantage of using special low pressure flange is much less than what it appears to be.

CONCLUSION

The equivalent pressure approach has become a standard method in evaluating pipe load acting on flange connections. There are two acceptable ways of checking the pipe load using the equivalent pressure. They are rating table method and stress calculation method. The rating table method is simpler but is much more conservative by ignoring the reserve strength. It is so conservative that it would probably disprove most of the installations which are operating satisfactorily. The stress calculation method is a more realistic approach which evaluates the actual reserve strength available in a flange.

When a standard flange is rated for a certain pressure, it normally posses sufficient strength to resist the rated pressure load plus sustantial reserve strength to resist the pipe load. This reserve strength varies from flange to flange and is not known until a stress analysis is performed. It is also highly dependent on the thickness of the connecting pipe when a bore of the flange is specified to be the same as the inside diameter of the pipe.

In designing a special flange, it is necessary to provide some allowance for the pipe load. The allowance can be either based on the actual expected pipe load or based on the load that will produce a bending stress equivalent to one half of the basic allowable stress at the connecting pipe. In any case care should be exercised in the final piping system design to keep the pipe load from exceeding the allowance provided.

REFERENCES

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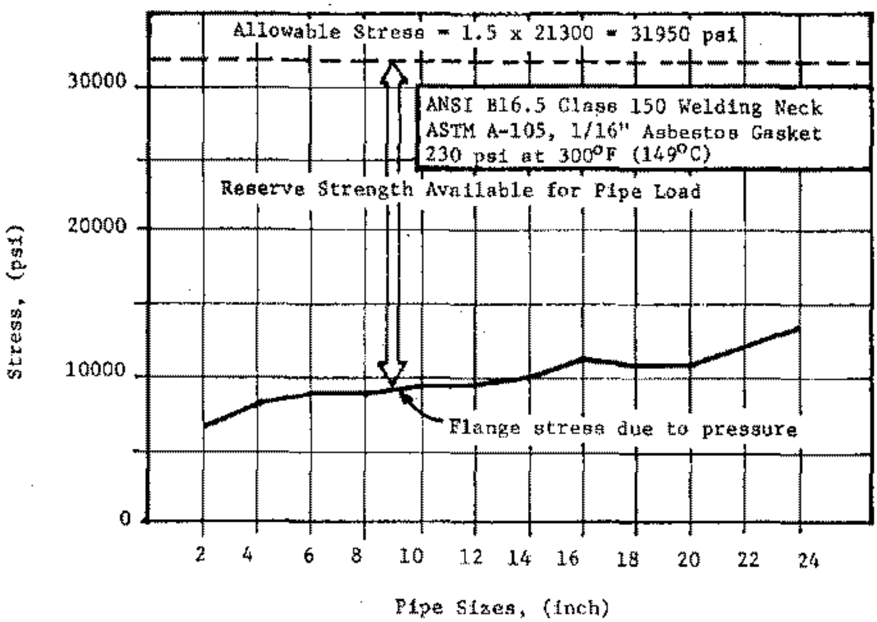


Figure 1. ANSI B16.5 Class-150 Flanges
(1 inch = 25.4mm, 1 psi = 6.894 KPa)

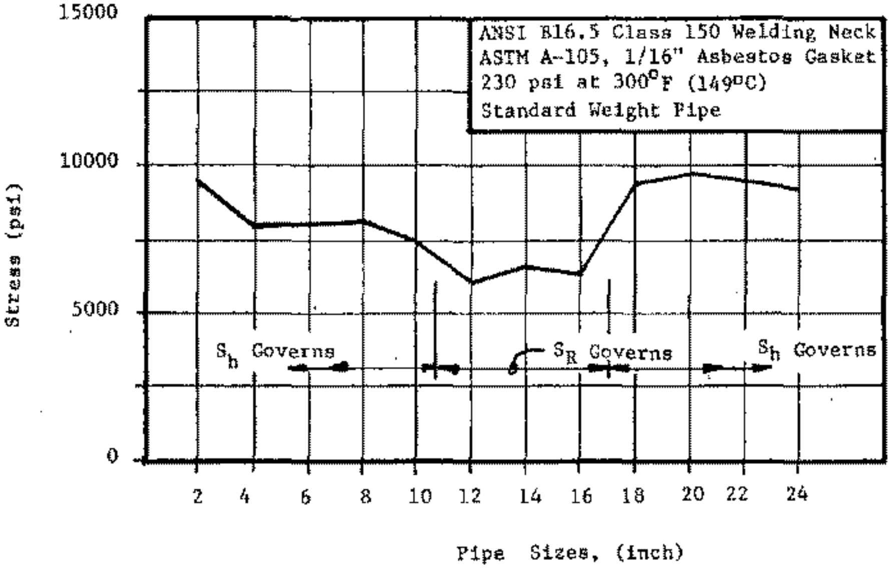
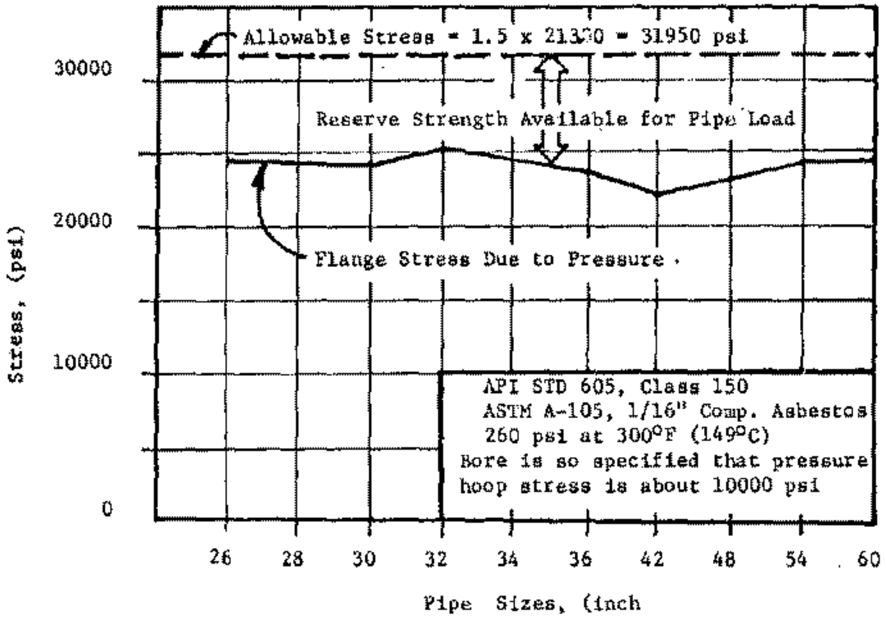
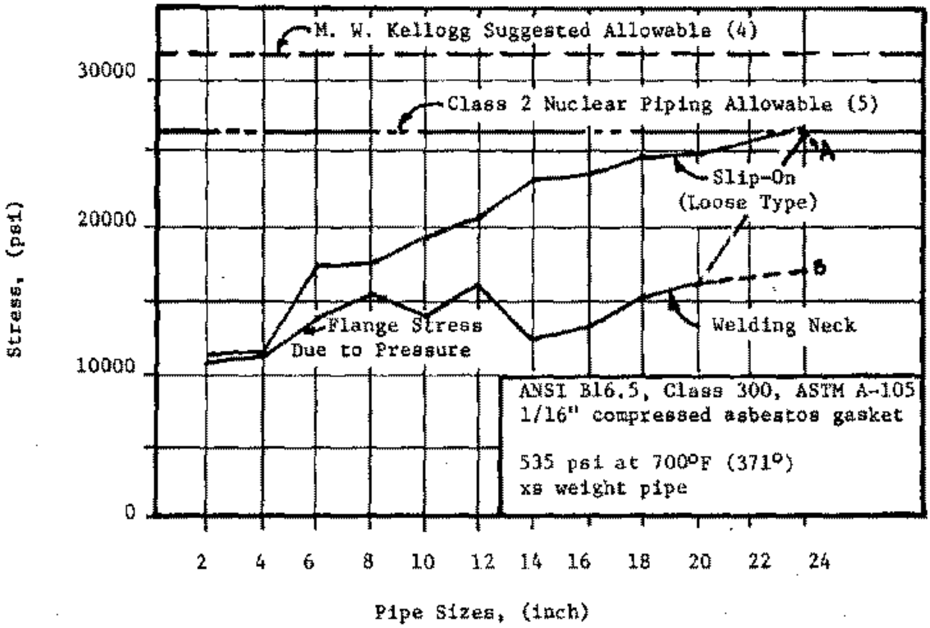


Figure 2. Allowable Pipe Load in Term of Bending Stress at Pipe
(1 inch = 25.4mm, 1 psi = 6.894KPa)



API STD 605, Class 150
ASTM A-105, 1/16" Comp. Asbestos
260 psi at 300°F (149°C)
Bore is so specified that pressure
hoop stress is about 10000 psi

Figure 3. API Std-605 Class-150 Flanges
(1 inch = 25.4mm, 1 psi = 6.894KPa)



ANSI B16.5, Class 300, ASTM A-105
1/16" compressed asbestos gasket

535 psi at 700°F (371°)
xs weight pipe

Figure 4. ANSI B16.5 Class-300 Flanges
(1 inch = 25.4mm, 1 psi = 6.894KPa)

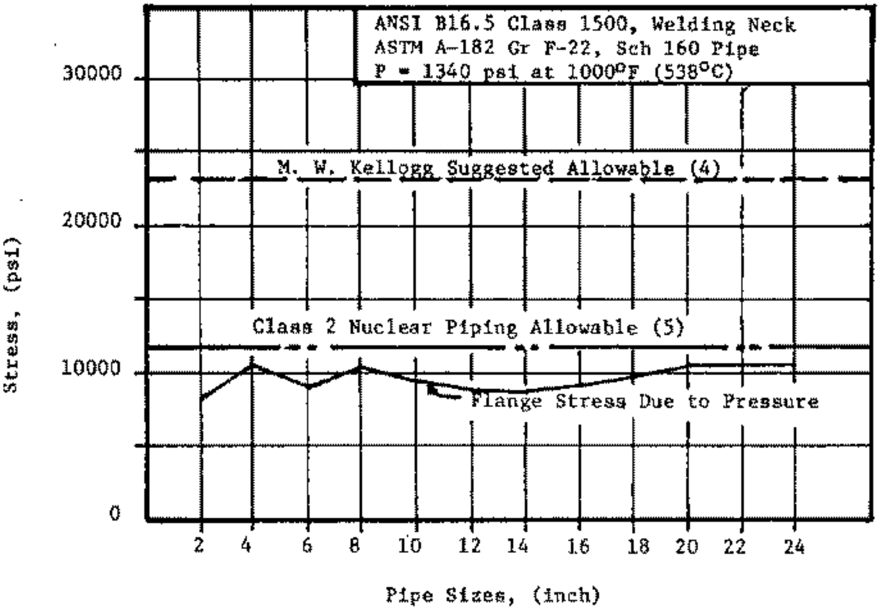


Figure 5. ANSI B16.5 Class-1500 Flanges
(1 inch = 25.4mm, 1 psi = 6.894 KPa)