RETHINKING THE ALLOWABLE PIPE LOAD ON ROTATING EQUIPMENT NOZZLES

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ABSTRACT

It is a consensual belief of piping engineers that the current allowable pipe loads on rotating equipment nozzles imposed by the equipment manufacturers are too low. A more realistic allowable should be established to better balance equipment costs against piping engineering and material costs.

This article reviews past practices of equipment nozzle loads and points out inadequacies and inconsistencies of the current standards. It is believed tha the extra manufacturing or engineering cost incurred in providing increased allowable nozzle loads for rotating equipment can be compensated by materials and engineering savings in the associated piping systems. This paper proposes a set of reasonable limiting allowables and suggests giving credit to equipment with higher allowable nozzle loads when a bid is being evaluated.

INTRODUCTION

A piping system has to be designed to satisfy the following rquirements:

(a) Functional Adequacy - The pipe shall be big enough to carry the amount of fluid required for the process. Its material shall be compatible with the fluid it carries. It is protected from excessive heat loss and from environmental damage such as corrosion, freezing and so forth.

(b) Structural Integrity - The pipe shall be thick enough to resit the internal pressure. It is properly supported for weight, wind, seismic, and other loadings. It should be flexible enough to absorb thermal expansion and contraction.

(c) System Operability - The piping shall not cause any excessive deformation to the connecting equipment thus hindering its proper operation. Flange leaking, valve sticking, rotating equipment vibration and overheating are some of the problems to be avoided.

While both insuring structural integrity and maintaining "system operability" are responsibilities of piping engineers, the task of maintaining the system operability is more difficult because it involves the strength of the connecting equipment which is beyond the control of piping engineers. What a piping engineer can do is to arrange the pipe in such a way that the pipe load applied at an equipment nozzle is less than the allowable load furnished by the equipment manufacturer. Unfortunately in actual practice this is not as easy as it sounds, since the allowable loads are generally very low. It is unusual to select a piping system meeting the allowable nozzle load without going through a considerable number of calculations. Flexible loops and special restraints are normally needed for hot piping systems to bring the pipe load within the acceptable limit.

It is a consensual feeling amont the pipe stress engineers that the current standard allowable nozzle loads should be higher. These low allowables have contributed to increased plant cost by requiring additional pipe loops, restraints, and engineering manhours. It is believed that the overall plant cost can be reduced by providing equipment with higher allowable nozzle load. This is believed to be particularly so in the case of rotating equipment.

It is the object of this paper to explore methods for setting an allowable pipe load on rotating equipment nozzles. The paper reviews past and current practices on nozzle allowable load, discusses the difficulties of meeting the current allowables, cites some special solutions to the problem, and establishes a set of rational allowable.

HISTORICAL BACKGROUND

Piping engineers have long realized that there is a limitation on the amount of pipe load which can be applied to the rotating equipment nozzle. Therefore, they have always refrained from putting too much load on the equipment. They also tried for forty years to rationalize and standardize the equipment allowable pipe loads. The difficulty faced by today's piping engineer is not any less than the difficulty faced by engineers forty years ago. The progress in standardizing the equipment allowable can be divided into classical and modern stages.

CLASSICAL ALLOWABLES

Prior to World War II, engineers didn't even bother to know how much pipe load a pump or a turbine could take. They resorted to common sense and judgement to design many successful plants. Although manufacturers insisted that absolutely no pipe load was allowed to be applied to their equipment, Rosshein and Markl (1) found that the average pipe reaction acting on the pumps were:

$$\begin{aligned}
&\text{Vertical thrust, Lb} = 3.25(D + 3)^3 \\
&\text{Lateral thrust, Lb} = 1.50\ (D + 3)^3 \qquad\qquad (1) \\
&\text{Bending or torsional moment, ft-lb} = 5\ (D + 3)^3
\end{aligned}$$

where D is the outside pipe diameter in inches. These loads are several times higher than the allowables of the modern poums 8-inch (203mm) or larger. These were the good old days when a pump could take a lot of load without even being designed for it.

Regardless of the successful operating experience represented by Equation (1) loading, in 1950 Wolosewick (2) suggested that the above loading was excessive. Based upon a large number of stress calculations on actual installations, he recommended a set of rules with considerably reduced allowables. His revised rules may have been based on the use of higher quality material available at the time thus resulting in thinner castings, and in higher speed equipment more sensitive to small deflections.

In this era, equipment was not designed directly to take any pipe loads. The loads applied were resisted by the inherient equipment strength designed by other factors such as pressure, architectural shape, and built-in conservatism. Equation (1) is no longer applicable, but it does reveal the magnitude of the loads piping engineers used to apply to equipment.

MODERN ALLOWABLES

In 1958, a new era began when National Electrical Manufacturers Association (NEMA) published the allowable pipe forces and moments on mechanical-drive steam turbines (3). This publication not only standardized the allowable loads but also divided the load into two categories; single nozzle load and combined machine load. The single nozzle is the direct measure of the nozzle and casing stress and deformation which might cause interference between moving and fixed parts. The combined machine load is the combination of all the loads acting on the machine through individual nozzles. This load is limited by the pedestal and baseplate strength and rigidity to assure shaft alignment. Because of their fundamental significance, the NEMA allowables are summarized in the following:

1. At each connection, the resultant forces and moment shall satisfy the following:

$$3F + M \leqq 500D \qquad (2)$$

where, F = Resultant force, lb.

M = Resultant moment, ft-lb

D = Pipe size of the connection (I.P.S.) in inches up to 8 inches in diameter. For sizes greater than this, use a value of D equal to (16 + I.P.S.)/3 inches.

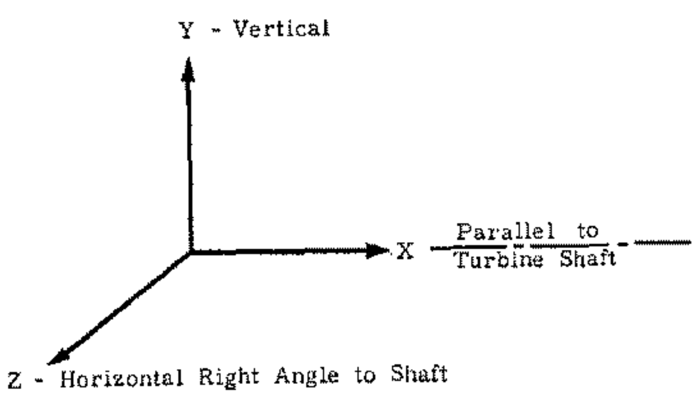

Figure 1. NEMA Allowable Directions

2. The combined resultant forces and moments of the inlet, extraction, and exhaust connections, resolved at the centerline of the exhaust connection, must not exceed the following conditions:

(a) These resultants shall satisfy:

$$2F_c + M_c \leqq 250\ D_c \qquad (3)$$

where, Fc = Combined resultant force, lbs

Mc = Combined resultant moment, ft-lbs

Dc = Equivalent diameter (in inches) of a circular opening equal to the total areas of the inlet, extraction, and exhaust openings up to a value of 9 inches in meter. For values beyond this, use a value of Dc equal to (18 + Equivalent Diameter)/3 inches.

(b) The components of these resultant shall not exceed:

$$
\begin{aligned}
Fx &= 50\ DC;\ Fy = 125\ Dc;\ Fz = 100\ Dc \\
Mx &= 250\ Dc;\ Fy = 125\ Dc;\ Mz = 125\ Dc
\end{aligned}
\qquad (4)
$$

where the directions are as shown in Figure 1.

As demonstrated in Figures 3 and 4, the NEMA allowable for a single nozzle is only about one eighth of the value given by Rossheim and Markl for a 12-inch (305 mm). Needless to say, this has created a lot of problems to the piping system. The allowable for the combined load is even more stringent.

Although the NEMA allowable is widely considered as very strict, it nevertheless is a standard with authority. It is also used conveniently as a reference value in other applications. For instance, Americal Petroleum Institute in its Standard 617 (4) requires that the centrifugal compressors shall be designed to withstand external forces and moments at least equal to 1.85 times the values calculated from NEMA SM-21 formulas.

PUMP ALLOWABLES

Standardized allowable nozzle load for pumps were not available until 1971, when API published in its Standard 610 (5). The original standard covered only the pumps with 4-inch discharge nozzle or smaller. The new 1981 6th edition has extended the scope to pumps having suction nozzles 12 inches and smaller.

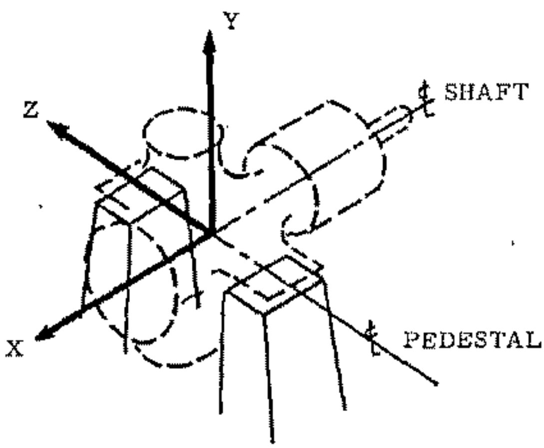

Figure 2. API Std 610 Allowable Axes

The allowable piping loads are summarized in Figures 3, 4, and 5. The lower values represent the allowable loads applied in the weak direction of the pumps. As will become clear later, these new revised 1981 allowables are approaching the rational values developed in this paper. The major improvement in the 1981 standards is the strengthening of the baseplates and pedestal

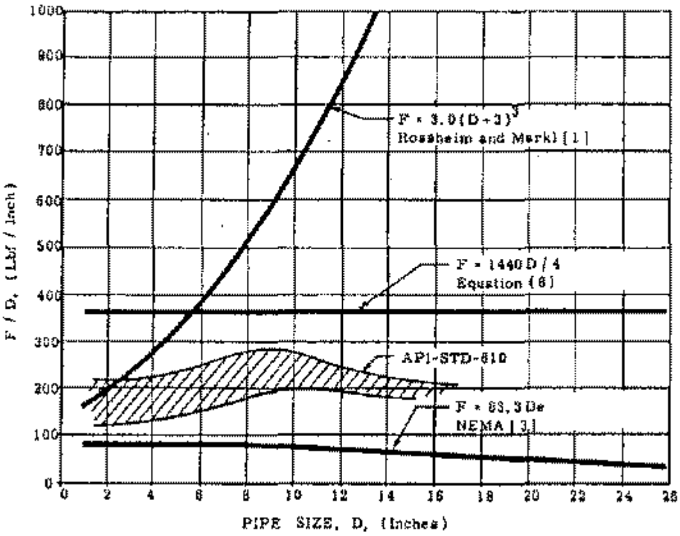

Figure 3. Single Nozzle Allowable Force (Resultant of 3-Directions)
Conversion Factors : 1 Inch = 25.4 mm
1 Lbf = 4.448 N

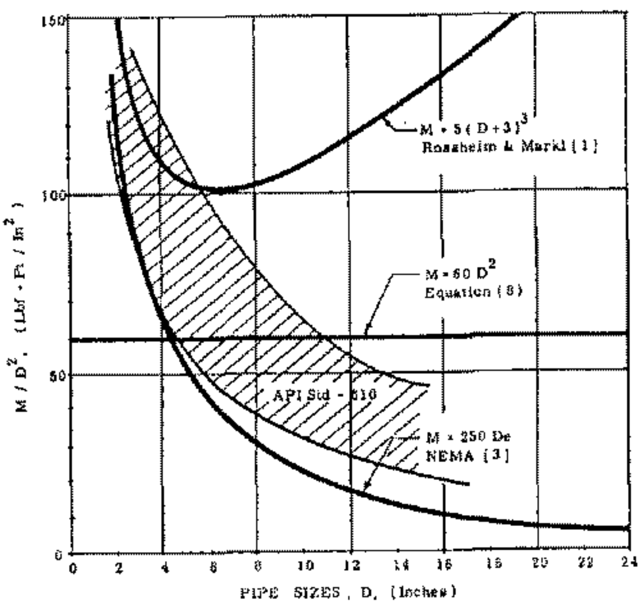

Figure 4. Single Nozzle Allowable Moment (Resultant of 3-Directions)
Conversion Factors : 1 Inch = 25.4 mm
1 Lbf = 4.448 N

supports. It also eliminates the original 1000 pounds minimum weight rule which had resulted in such an inconsistency that gave almost the same combined design moment for all sizes of the pumps covered.

However, it should be noted that a large number of pumps are not built by the API Standard. The so called AVS (9) pumps widely used in the chemical process industry, for instance, do not have a standard nozzle allowable. Most of the AVS pumps can take considerably less load than API pumps.

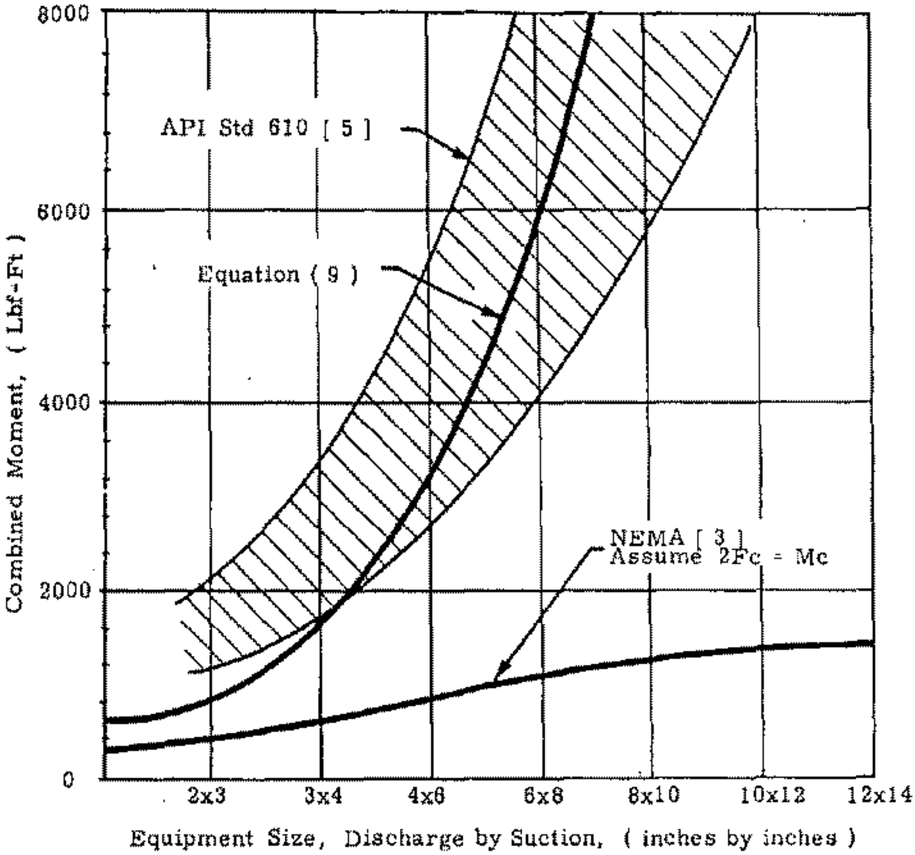

Figure 5. Combined Resultant Moments
Conversion Factors : 1 inch = 25.4 mm
1 Lbf = 4.448 N

From the charts presented, it appears that API STD 610 has a higher allowable than NEMA, but in reality, the STD 610 allowable is the more difficult one to meet. This is mainly due to the fact that:

a. Turbine piping is inherently more flexible than pump pipings due to the high temperature and pipe stress requirement.

b. Most turbines are independent drives therefore their piping can be more easily restrained. On the other hand, pumps are normally installed with 100% or 50% spares meaning two or three pumps are generally connected together with common piping. This creates twisting between pumps and also makes them more difficult to restrain.

c. Liquid can potentially create a more disturbing force than stream. Therefore, pump piping is more prone to vibration when excessive piping loops are installed; careful attention to loop supports and guides are a necessity.

DIFFICULTIES WITH CURRENT STANDARD ALLOWABLES

It is probably safe to say that the current standards for equipment nozzle allowable loads have caused considerable problems for piping engineers. They are too low, inconsistent and arbitrary.

(a) Too Low - This is exemplified in Figure 3. In this figure the Rossheim and Markl curve represents the allowables piping engineers used to apply. They are well above the NEMA recommendations. The realistic value can be expected to lie between these two curves.

(b) Inconsistency - The current standard values does not treat forces and moments in equal significance. In NEMA, for instance, although the moments and forces are combined into a single parameter, the fixed scale factor applied to forces is not consistent with the actual equipment geometry. The inconsistency also is apparent when Equations (2) and (3) are converted into the Metric system. The greatest inconsistency of the current standard allowables, however, is the variation with the pipe diameter. The standard allowable does increase with increasing pipe size, but is less than directly proportional to the pipe size. However, if based on strength of pipe or equipment, the allowable should vary proportionally to the square or cube of the pipe size as shown in Equation (1). The new 1981 API 610 has more or less corrected this inconsistency by requiring the consideration of both forces and moments in the combined evaluation.

(c) Arbitrary - Although some convenient values have to be used when a standard is being established, these values can make the whole standard appear to be arbitrary. Take NEMA allowable for instance, the allowables are proportional to the pipe size for pipes up to 8 inches; then the increasing rate abruptly drops to one third of the original rate for pipe sizes greater than 8 inches. Regardless of the reason behind this rate drop, the use of 8 inches as a change step is arbitrary. The arbitrary factor is even more apparent in the case of 1971 API-610 pump allowable which is still applicable to many plants currently under design. Owing to the minimum 1000 pound rule, the 1971 API-610 allowable of a 1 inch by 2 inches pump is the same as that of a 8 inch by 10 inch pump. This is due to the fact that except some small amount of high pressure pumps, most pumps of refinery and petrochemical services are light weight types. An 8 by 10 pump barely weights 1000 pounds, the minimum weight to be used in calculating the allowables. Although tests (6, 7) have shown that a 4 x 6 pump can resist the load allowed without exceeding the specified 0.010 inch displacement, it is valid to question if a 1 x 2 pump can take the same load. Also it will be logical to assume an 8 x 10 pump will be able to take more. Fortunately, this 1000 pound minimum weight rule has been eliminated in the 1981 API-610.

THE SOLUTIONS

The low equipment allowable nozzle loads have forced piping engineers to use excessive pipe loops coupled with complex restraint arrangements to meet the requirements. This not only increases capital expenditures but also increases potential operational problems. Vibration, cavitation, and loss of net positive suction head (NPSH) are some of the common operating problems resulting from excessive piping loops.

To overcome the above difficulties, engineers are occasionally forced to resort to unconventional approaches such as stiffening the pump base plates, putting the whole equipment set in suspension, making the whole equipment set free to slide, and so forth. However, even letting the equipment slide on a pad, considerable stiffening on the baseplate is still frequently required (8).

The fundamental solution to the problem is to increase the equipment allowable loads. In orther words, higher allowable nozzle loads are needed. Looking at current allowables, it is apparent that the pressure part of the casing of the equipment is generally sufficiently strong to be able to take the moment which will generate a pipe stress equal to approximately one third of the basic allowable stress. That is a large moment compared with the allowable moment. The limiting weak part in the equipment appears to be the baseplate and pedestal portion which can be very inexpensively stiffened.

RATIONALIZE THE ALLOWABLE

It has been argued that low equipment allowable nozzle loads have resulted in increasing the overall plant cost. Piping engineers have long realized that buying somewhat more expensive equipment with higher allowable nozzle loads may actually be cheaper, overall, when piping cost is included.

However, this philosophy is seldomly put into effect in buying the equipment. Because of this, the manufacturers don't have an incentive to devote attention to this problem. They frequently give allowable nozzle loads lower than the one their equipment can actually take to reduce the possibility of equipment problems.

Therefore, manufacturers must be given incentives to provide higher allowables as they are done in other areas. For instances, in buying pumps or turbines, a credit is generally given to the ones with higher efficiency or performance rate for the eventual saving in power or energy consumption. Although the energy saving is not immediately realizable at the time of purchasing, engineers are willing to capitalize the future saving into the present worth to compensate the vendors. Because of these incentives the vendors are constantly trying to improve their equipment efficiency without being pushed to do it. This process works real well in increasing equipment efficiency. There is no reason to believe that a similar process will not work for the equipment allowable load.

The savings from higher equipment allowable nozzle loads are not very tangible. It comes from saving of pipe and pipe support material and reduced engineering effort. When the stated allowable nozzle loading is lower than a practicable value, engineers will be forced to devote much time and effort to developing a solution for what may well be a ficticious problem.

In order to have a guideline of the magnitude of the allowable, it is beneficial to develop a set of rational allowables.

1. Allowable Loads

Because of the pipe stress, flange load, foundation capacity and other requirements, the allowable pipe loads cannot be arbitrarily increased without proper limit. An equipment allowable exceeding a certain limit will have no practical increased value at all.

In setting the practical allowable, it is necessary to avoid falling into the same difficulties the current standard allowables are facing. The allowables should consider the force and moment in equal significance. They should also vary more than directly proportional to the pipe diameter.

From Figure 6, for loads acting on the face of nozzle A or the designated resolving point B, the effect on the coupling displacement will be depending on the force and moment acting through A-B-C and A-B-D. In most cases the flexing deformation due to moment is much more significant than the shear deformation due to direct force. Therefore, the total effect of the load can be measured by a combined parameter $M + K_1LF$ where K_1 is the correlation constant and L the equipment dimension. Because the equipment dimension is proportional to pipe diameter, the above parameter can be revised to $M + K_2DF$. By setting $K_2D = 3$, this parameter would be identical to the NEMA parameter. But by using K_2D instead of the simple constant, the parameter can reflect the actual equal significance of forces and moments. K_2 can be set equal to 4 with consistent units in M, D, and F. This combined parameter can also be called equivalent moment. That is

$$M' = M + 4DF \tag{5}$$

The variation of the allowable with the diameter can be derived from the idea of limiting pipe stress. Based on past experiences and practices, the limiting pipe stress shall be smaller for larger pipes. In order to have a smooth variation, it appears to be logical to set the limiting pipe stress inversely proportional to the square root of the pipe diameter. That is

$$S = \frac{K_3}{\sqrt{D}} \tag{6}$$

K_3 can be roughly set to 18000 psi - $in^{1/2}$ (19,800 KPA - $m^{1/2}$). It should be noted that this is not the actual pipe allowable stress but rather a measure to set a reference maximum pipe load. Since the standard wall thickness varies roughly proportional to the square root of the pipe diameter, the section modulus of the pipe can be written as $Z = \pi r^2 t = K_4 D^{5/2}$. K_4 can be set as 0.08 $in^{1/2}$ (0.0128 $m^{1/2}$). The allowable equivalent moment $M' = SZ$ then becomes,

$$M + 4DF = K_3\ K_4\ D^2 \tag{7}$$

or, for individual nozzle

$$M + 4DF = KD^2 \tag{8}$$

where, M = Resultant moment, in-lb, (KN-M)

D = Outside diameter of pipe, in, (M)

F = Resultant force, lb, (KN)

K = Constant = 1440 lb/in

= 253 KN/M

Assuming M and 4DF have the same contribution, the relative comparison of this maximum load with the standard allowable is shown in Figure 3, 4, and 5.

Equation (8) represent the allowable for each individual nozzle. For combined allowable the same equation can also be used by replacing the outside diameter with an equivalent diameter. That is for combined load

$$Mc + 4\ Dc\ Fc = K\ Dc^2 \tag{9}$$

where, Mc = Combined resultant moment, in-lb, (KN-M)

Fc = Combined resultant force, lb, (KN)

Dc = $\sqrt{D_1{}^2 + D_2{}^2 + D_3{}^2 + \cdots}$, equivalent diameter, in, (M)

K = Constant, same as in Equation (8)

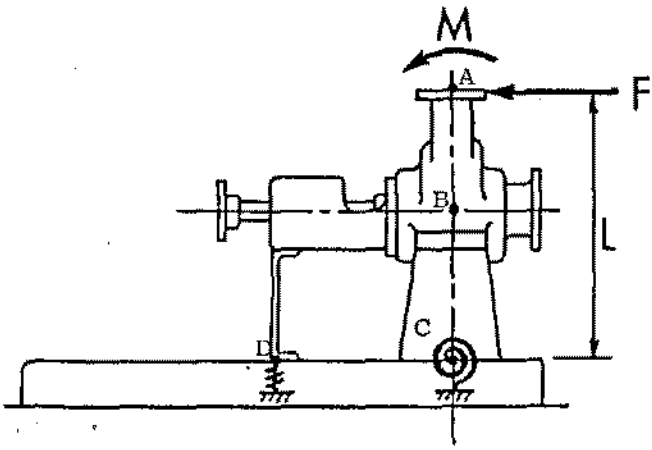

Figure 6, Equal Significance of Force and Moment

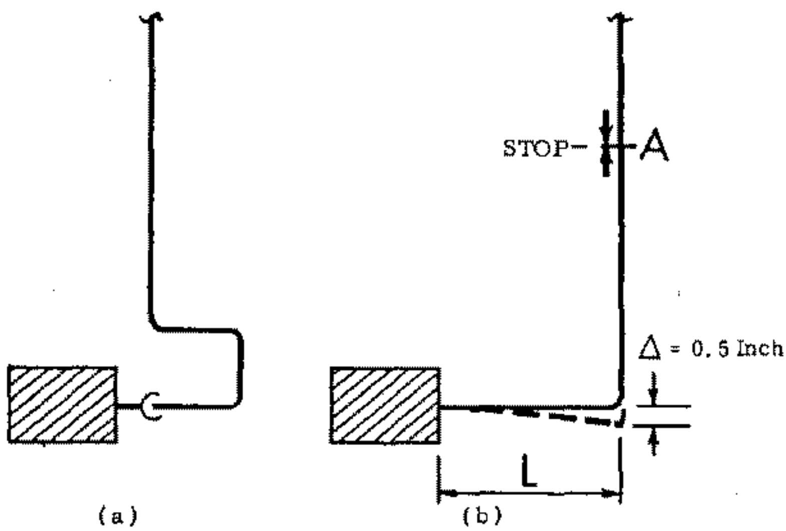

Figure 7. The Piping around an Equipment

Although each equipment has different stiffnesses at different directions, the variation of the allowables at different directions should not be greater than 50 percent of the maximum component. The difference between M and 4DF should not be more than 50 percent of the greater of the two either.

2. Cost Benefits

Equations (8) and (9) have laid out the allowable which piping engineers would like to have, now it is time to find out what is the cost benefit of the high allowable. Figure 7 shows a typical layout to be used in this investigation. The actual layout may look like the one shown in Figure 7-(a), but for investigation purpose the layout can be unfolded as in Figure 7-(b). The loop or the leg L is needed to absorb the expansion so the pipe will not create too much load on the equipment. The investigation is to find the pipe length required for different equipment allowables. The lower allowable will require longer length, and the larger the expansion the longer the L will also be needed. This L is tied to equipment allowable and the amount of expansion two variables. However, since the line can always be conveniently stopped at Point A located 20 or 30 feet away from the equipment, it really involves only the expansion of about 20 feet of pipe. Furthermore, the higher the temperature the longer the initial length is required just for the pipe stress purpose, the effect of temperature on the additional pipe length in reducing the pipe load has greatly reduced. Based on the above, a fixed delta of 0.5 inch (12.7mm) can be used for all the cases. This 0.5 inch movement is equivalent to the expansion of 20 feet of carbon steel pipe operating at about 400°F (204°C) and also is the amount of displacement that can be controlled relatively easily by restraints. The lenght required can be determined by using the guided cantilever approximation. From guided cantilever formulas,

$$F = \frac{12EI\Delta}{L^3}, \qquad M = \frac{FL}{2} = \frac{6EI\Delta}{L^2} \qquad (10)$$

where, E = Modulus of elasticity of pipe material, psi, (KPA)

I = Moment of Inertial of pipe, in^4, (M^4)

Δ = 0.5 in (0.0127 m)

Equation (10) can be used to estimate the capital value of the allowable in terms of the pipe length.

Of course, the saving from the higher allowable is not limited to the material saved. The supports, restraints, and space saved can easily exceed the saving from the pipe considering the fact that many restraints have to be framed up from the grade. A tight space can also cost engineering days to come up with an acceptable loop.

CONCLUSION

Piping systems connected to rotating equipment have occasionally caused operating problems. Large pipe forces have caused some pump seal to wear out permaturely. They have also caused some turbines to vibrate undesirably. On the other hand, excessive loops used to reduce the pipe force have caused some severe shaking in piping. These problems can be greatly reduced through a better coordination between piping engineers, equipment engineers and the equipment manufacturers. This can be achieved by the following practice.

1. Know the amount of pipe load the equipment can take:

It is not unusual for a piping engineer to discover toward the end of the design stage that an equipment can actually take much less pipe load than expected. To alleviate this situation whenever possible, equipment should be purchased that is built in accordance with standards providing allowable nozzle loads.

2. Encourage the vendors to provide the actual nozzle allowables particularly when it is higher than the standard values. It should be specified that high allowable nozzle loadings will be viewed as an advantage.

3. At equipment purchasing coordination meetings, the subject of allowable nozzle loads should be discussed and the design parameter agreed upon with the manufacturer.

ACKNOWLEDGEMENT

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