THE ART OF DESIGNING PIPING SUPPORT SYSTEMS

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ABSTRACT

The standard approach to pipe support design is to follow well known, accepted, practices; the art of designing support systems is to go beyond these common practices.

This paper uses specific examples to demonstrate that the use of some common practices can lead to real problems in some situations. Support type based on vertical thermal displacement, spring loads set to balance the weight at hot condition, anchors at expansion joint installation, springs sized to minimize the vertical load at equipments are among the specific items discussed. The potential of utilizing friction forces to replace the expensive snubbers is also presented.

INTRODUCTION

A basic design is normally created by following common rules formulated from the past experiences of the industry as a whole. The common rules are essential for the day to day design practice. However, the rules passed from generation to generation are only those which are broad enough and simple enough to warrant a space in the company standards or technical books. These rules are valid for most of the situations, but invalid for certain cases. The exceptions are often so inconspicuous that they can be overlooked even by experienced engineers. To deal with these is an art which requires exceptions to the rules.

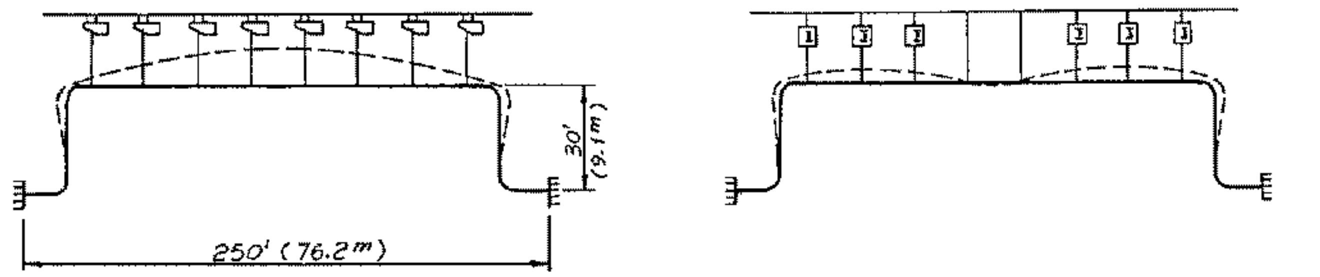
An art is undoubtly abstract. Therefore, instead of presenting principles, this article will use some specific examples to demonstrate the ideas. For instance, what can go wrong by (1) selecting the support types based on vertical thermal expansion displacement, (2) by making the equipment nozzle take no direct weight load, (3) by setting the spring to balance the weight at hot condition or (4) by installing anchor liberally at expansion joint installations.

With the advance of the computer technique, almost all the calculations today are done by computers. However, the old maxim "garbage in garbage out" is still true. In order to provide better data for the computer, a couple of practices have been evolved. First, all the support members and attachments are designed to be super stiff. Secondly, friction forces are greatly reduced by using teflon sliding plates or ball jointed struts. With these painstaking arrangements, analysts can now boast of the validity of their analyses. However, few have realized that the brute force approach has thrown away two of the major ingredients that have helped preserve the structural integrity of the design. These two ingredients are flexibility and friction, and they should be once again put to work to our advantage.

SUPPORT TYPES

The types of supports are normally selected based on the vertical thermal displacements expected at the support locations. Rigid supports are used at places where the expected thermal displacement is very small, variable springs are used for medium displacements, and constant effort supports are used when displacements are great. The practice is very logical, but problems arise occasionally. Oddly enough, the use of springs and constant supports create more problems than the use of rigid supports. Although it is true that a rigid support should not be used even at a place having a small expected thermal displacement, the misapplication of rigid supports will be detected as soon as an analysis is performed. On the other hand, the analysis on a spring or constant effort supported system cannot readily tell the mis-application of the springs.

Figure 1 shows one situation that might end up with a problem. A free thermal expansion analysis shows a vertical displacement of 11 inches (280 mm) at the middle of the span due to arching effect. The displacements at other support points are all greater than 3 inches (76 mm). By using the free thermal displacement as a guidance, constant effort supports will be used for the entire system as shown in Figure



(a) (b)

Fig. 1 Free Thermal Displacement

1 (a). Some snubbers might also be added if the system is to be designed for earthquake. Aside from the hardware cost, the arrangement appears to have no problems. The computer analysis shows perfect results, and the installation will have no problem either. The problem nevertheless occurs when the system is ready for operation and the travel stops are removed. It couldn't have come at a worse time, but that is the nature of most the problems.

This system may collapse if the actual pipe, insulation, and attachment weight is considerably heavier than the theoretical or assumed weight used in the design. The system may weigh as much as 15 percent more than the design capacity of the supports and this may make the field adjustments almost impossible. One may argue that the weight should have been estimated more conservatively, but the point is that the system designed is uanble to absorb the uncertainty due to manufacturing tolerance. The system can also be underweight making the field adjustment equally impossible. Even with a properly adjusted system, because of the friction associated with the linkages a lot of banging may be expected during the start-up and shut-down. The movement tends to be stuck for a while then an intermittent sudden release.

Figure 1 (b) shows a better design by placing rigid supports at the middle spans where the free thermal displacements are the greatest. This system is much more capable for absorbing the weigh variation disregarded by the computer. It also costs a lot less than the one shonw in 1 (a).

HOT BALANCE

In a high temperature system, in order to minimize the creep, the spring is set in such a way that the spring force and the system weight will balance out each other under the hot operating condition. It is important that the sustained stress be reduced to minimum at creep temperature. For a low temperature piping where little creep is expected, it is still a good idea to do the same so the unbalanced load is minimized under the operating condition. Hot balance is such a good practice that is considered as one of the basic principles of piping engineering.

By adopting the hot balance approach, the springs have to be locked in place during installation. The locks or stops are removed when the system is ready for operation. The problem is that in many cases severe twisting and jerking occur when the stops are removed. This is somewhat expected because the spring forces at cold condition are different from those required for balancing the weight. This calculated preloading is alright if everything is as ideal as calculated. As mentioned previously the pipe weight, insulation weight, clamp weight and so forth can vary considerably from the theorectical values. Therefore the actual loading applied to the system can be quite different from the one calculated.

The deviations of the weight and the analytical model are so difficult to predict that the hot balance approach intended to minimize th hot load is in fact applying unpredictable loads on both cold and hot conditions. Alignment problems have frequently occurred on large rotating equipment. The theoretical minimum hot load is actually only a paper promise. It hits the target some of the times and is off the target at other times. This kind of incertainty is simply too much of a risk to be taken on an expensive delicate machine which is often the heart of the entire plant. Therefore, a more reliable approach is needed. Contrary to the common practice, the reliable approach is the cold balance approach.

In the cold balance approach, the fit up of the piping to a compressor or turbine is normally done with springs unlocked. The construction engineer will then try to adjust the spring load to bring the pipe connection to the equipment nozzle with a minimum help of outside force. In this way, it is sure that the piping load at cold condition is almost zero, although some load is expected under the operating condition. However, this hot load caused by spring force variation is highly predictable. The cold balance has become more and more popular lately. Designers who fail to understand the situation will make the field adjustment very difficult and will also create unnecessary argument with the construction engineers. The system

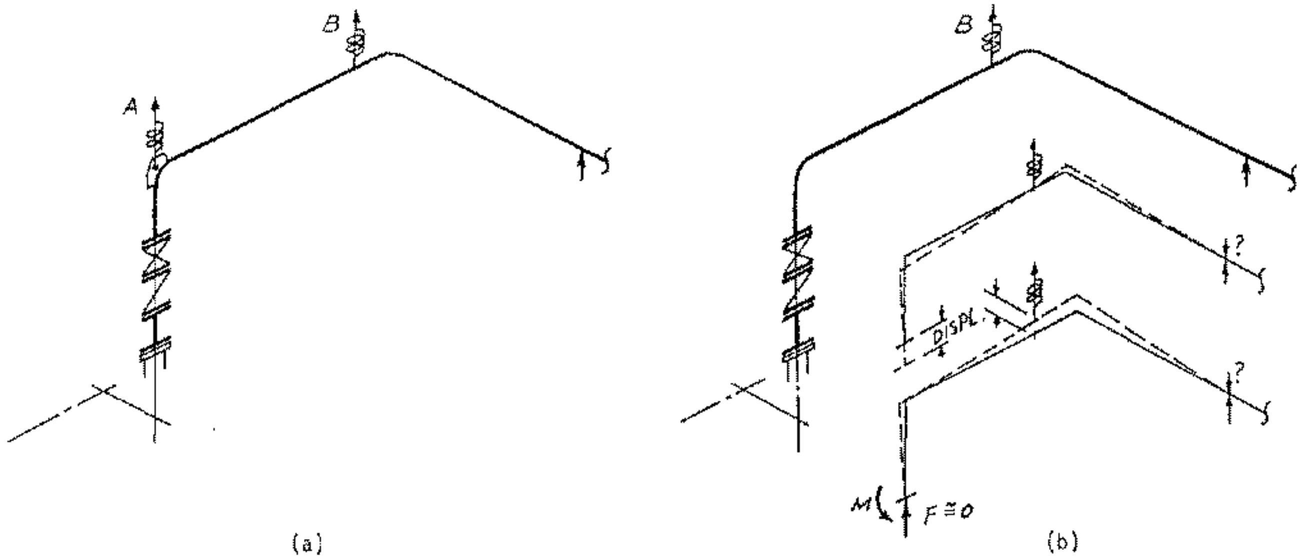


Fig. 2 Springs Sized By Releasing the Anchor

designed intentionally to balance the weight at cold condition will make the field adjustment much easier. With this understanding, the designer can simply instruct the field to set the spring at calculated hot load instead of the shifted cold load.

ZERO WEIGHT LOAD ON NOZZLE

It is a common practice to adequately support a piping system that no weight is imposed on rotating equipment. This can be done fairly easily by placing proper supports at proper locations. The only problem associated with this practice is the blind dependence on the computers.

Some piping stress computer programs used in the industry today have an automatic spring selection capability. They also have the option of releasing the vertical translational constraint at certain anchors during the spring selection process. This option will force the springs to carry all the weight leaving very little direct weight load on an equipment nozzle. This option is useful if it is applied correctly. For instance, many designers do not recognize that the scheme reduces only the direct weight load but not necessarily the weight moment. To have the scheme do the job right, springs have to be located at suitable locations. Otherwise, the springs selected by this anchor release option can make the system worse than the ones selected without the anchor release option.

Figure 2 shows a typical pump discharge piping. In 2 (a), since there is a spring directly over the valve assembly, the spring selection process with the anchor release option will force the springs to carry the entire weight leaving very little load to the pump nozzle. If the anchor release option is not used, then most likely spring A and pump nozzle will each carry about one half of the assembly weight. This of course happens only when the springs are selected by computer program

The situation will be different if the spring A is not available as shown in Figure 2 (b). In this case, if the spring is selected with the anchor release option, the spring B will be forced to pick up the entire weight including the whole valve assembly. This will leave very little vertical force on the nozzle but will create a huge bending moment on the nozzle. Unfortunately, this huge moment may escape the attention of the analyst in some cases. Some computer programs, in an attempt to speed up the process, give the weight load case results taken from the ones obtained with anchor released. By doing so, the high moment at the nozzle will not show up in the output report. The only clue to this problem is the significant vertical displacement shown at the anchor point. This vertical displacement is not very obvious and is often overlooked or ignored. A properly designed computer program will apply the spring force selected and the anchor fixed to recalculate the weight result. With this type of proper analysis, the high moment will appear at the nozzle together with an upward displacement at the spring location. With systems as shown in Figure 2 (b), it will be more favorable to select the spring with the anchor fixed. In this way the anchor will absorb some vertical weight force but not the huge bending moment.

EXPANSION JOINT ANCHOR

One of the most important requirements in designing bellow expansion joint system is to install sufficient anchors for resisting pressure end forces. Figure 3 (a) shows the potential pipe movement when no proper anchor is installed. Figure 3 (b) represents the system stabilized by the anchor. The anchor normally needs to be designed to absorb only the vectorial sum of the two end forces. However, if the system is expected to experience flow surges, the unequality of the two end forces at any time instant also needs to be considered. It is also possible that a valve is located at one side of the anchor as shown in Figure 3 (c). In this case the anchor has to be designed also for the condition when the valve is closed. If this valve shut-off condition is not designed for, the anchor can fail due to inadequate design, especially when the bend angle is small.

There are also cases when anchors should not be used. Figure 4 shows a tied expansion joint which is used to absorb the lateral differential expansion. By comparing with Figure 3 arrangement, it is tempting to put an anchor at the base support to resist the bellow end force. This anchor appears so natural that problems are often overlooked even by an experienced checker. The problem of the anchor can be explained from the start up sequence. When the pipe is heated up, both B and C ends expand into the bellow leaving slack at the tie-rods. As soon as the tie-rods get loose, the pressure end force pushing the turbine is not balanced. This pressure force normally is sufficient to push the turbine off alignment causing severe operational problems. In a correct installation, this anchor is not used. The pressure will push the base support outward ensuring a balancing force on the tie-rods to cancel the pressure end force acting on the turbine.

THREE HINGE SYSTEM

Symmetry and balance are normally considered two major principles in a good design. However, there are occasions when symmetry can also mean handicap. The Three hinge systme frequently used in solving plane expansion problem is one of the examples.

Figure 5 shows a three hinge system to be installed in large diameter piping connected between two major pieces of equipment. Figure 5 (a) is the perfect symmetric layout favored by many designers, including experienced ones. The only problem with this layout is that the three hinges are lined up in a perfect straight line. For the hinge 2 to be active it has to move when the system is heated up. However, this is almost impossible due to the perfect symmetry. For instance, if a line is drawn between hinges 1 and 3 to divide the space into two half spaces I and II, it is clear that any given point x in half space I there is a corresponding symmetric point xx in half space II. In other words, if the hinge 2 can move to x, it can certainly move to xx too. Since it cannot be at two different locations at the same time, the hinge will be simply stuck without moving anywhere. This is an example of pure symmetric case. In reality certain unsymmetrical effect will be built-in in the system to allow the hinge to move.

Figure 5 (b) shows the movement of hinge 2 which is located slightly off the symmetric line due to construction deviation. The hinge 2 in this case will move toward the half space II, but the magnitude of the movement can be unexpectedly high. For instance, with the dimension and the temperature shown, the calculated

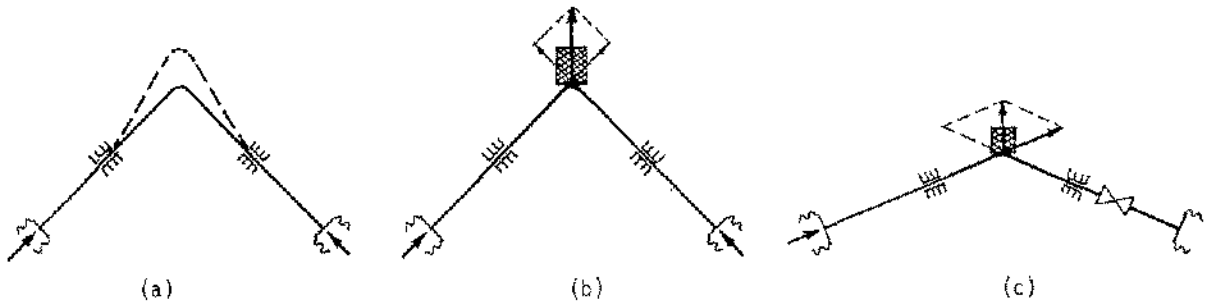


Fig. 3 Adequate Anchor Is Essential

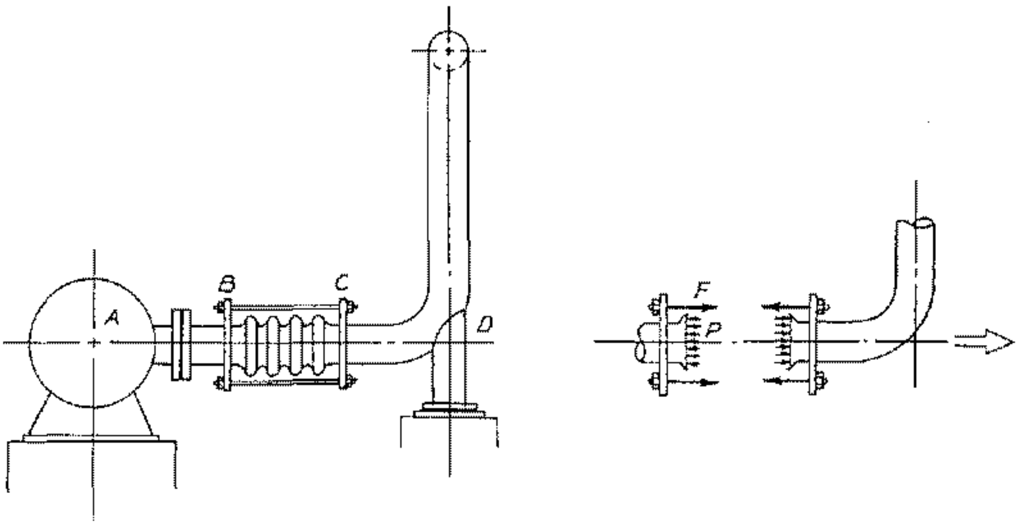


Fig. 4 No Anchor is Allowed

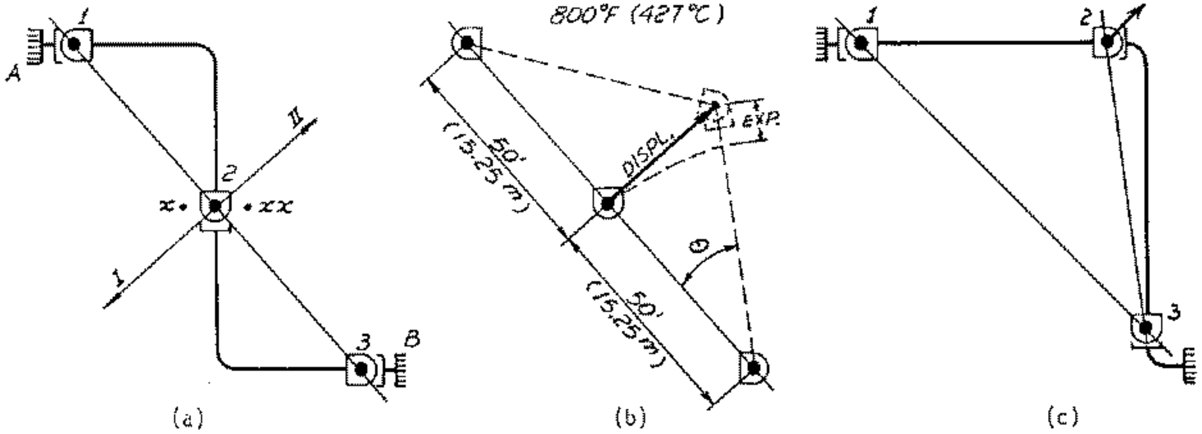


Fig. 5 Symmetry Can Also Mean Problems

movement of hinge 2 is 65 inches (1651 mm), and the angle of hinge rotation is 12 degrees. This movement is too much to be accommodated by support system. The friction on the support will also have very great effect on the equipment loading.

The system can be greatly improved by locating the hinge 2 away from the line connecting the end hinges 1 and 3 as shown in Figure 5 (c). With this alternative layout, the expected hinge 2 movement is reduced to about 5 inches (127 mm) with a hinge rotation of only about one degree. This order of magnitude is well within the normal support system capacity.

FRICTION RESTRAINT

For systems which need to be designed for shock loadings the common approach is to install snubbers, either hydraulic or mechanical, at points where rigid restraints are not permitted due to thermal expansion requirement. It works fine except there are also difficulties. These snubbers are not only expensive but also require constant maintenance. The snubber also has a built-in play that allows the restraint point to move a certain amount before being stopped. This slack makes the snubber a poor restraint for small amplitude, steady state vibrations. A frictional restraint may be more suitable for some cases.

Figure 6 shows a horizontal loop system whose vertical motion can be easily restrained with rigid supports. The horizontal motion, however, is somewhat complicated. Each leg of the loop needs an intermediate horizontal restraint to resist the earthquake load. However, because of the thermal expansion, a rigid horizontal restraint will create too much thermal expansion stress. In this case the straight forward approach is to install a snubber. The question here is if there is an alternative approach. The main reason the loop needs the horizontal restraints is because the unrestrained system will shake in the neighborhood of the peak response spectra. Once the horizontal restraints are installed, the natural frequency will shift upward to a more favorable spectra range.

Therefore, it is interesting to note that the horizontal restraint force required is only about 500 pounds (2224N). This magnitude of force is normally tolerable to an 8 inch (219 mm outside diameter) pipe. By using frictional sway braces adjusted at 500 pounds force, the braces will act as rigid stops during earthquake event, while putting limited restraint force against thermal expansiom movement. Table 1 shows the stresses for different support scheme used. The table is constructed by assuming that the snubbers impose no resistance to thermal expansion. In reality because of the tight seal requirement, the resistance imposed by a snubber can be significant.

Table 1, Pipe Stress Generated by Different Support Schemes

Restraint	Pipe Stress(psi), 1 psi=6.789 KPA	
	Thermal Expansion	Earthquake
Without Restraint	8950	8030
Rigid Restraints	38390	1380
Snubbers	8950	1380
Frictional brace	14400	1380

The above discussion demonstrates the use of frictional brace to stop a dynamic motion. The frictional restraints can also be used to absorb the dynamic motion. For energy absorption, the support point has to be allowed to move a small amount. The small movement coupled with a friction force can effectively absorb the vibration energy thus increases the damping of the system.

CONCLUSION

Piping support systems are generally designed by two major rules. The support locations are determined by the guidance of the maximum allowable spans, and the support types are selected based on the expected vertical thermal displacements. There are also rules and practices adopted to facilitate the design and to avoid common errors. However, as demonstrated in the above discussions, there are always exception to each of the rules. These exceptions, if not handled properly, can cause difficulties in installation and create problems during operation.

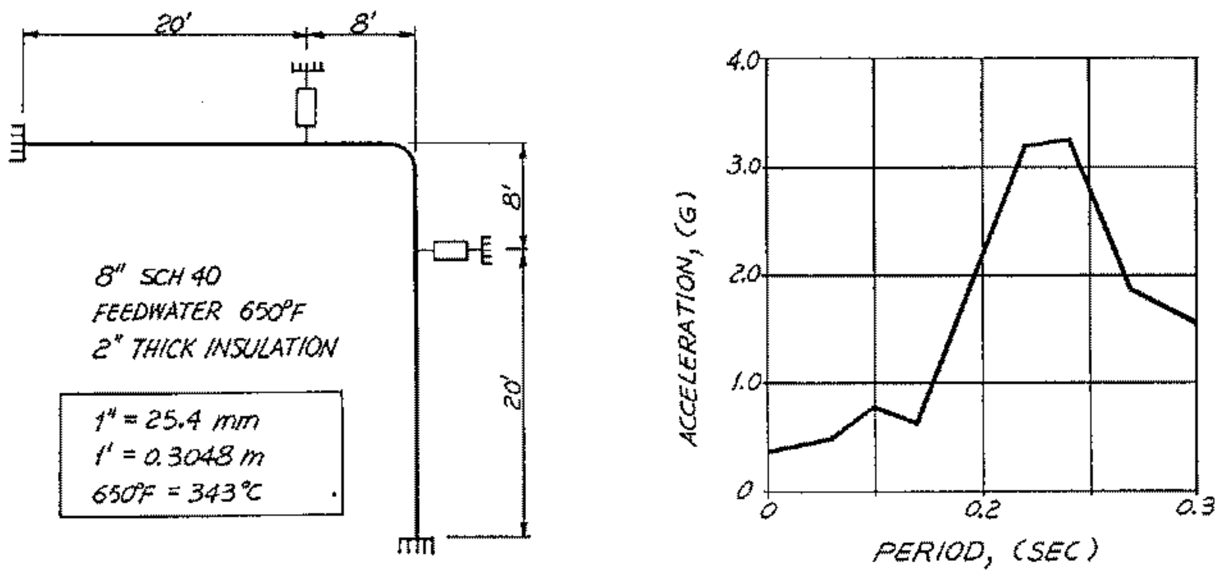


Fig. 6 Friction Restraints at Work