EQUIPMENT RELIABILITY IMPROVEMENT THROUGH REDUCED PIPE STRESS

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The load and stress imposed from a connecting piping system can greatly affect the reliability of an equipment. These loads, either from expansion of a pipe or from other sources, can cause shaft misalignment, as well as shell deformation, interfering with the internal moving parts. Therefore, it is important to design the piping system to impose as little stress as possible on the equipment. Ideally, it is preferred to have no piping stress imposed on an equipment, but that it is impossible. The practical practice is for the equipment manufacturer to specify a reasonable allowable piping load and for the piping designer to design the piping system to suit the allowables. The allowable piping loads given these days are generally determined solely by the equipment manufacturers without any participation from the piping engineering community. The values so determined are usually to low to be practical.

The low allowable pipe load given by the manufacturer results in a weaker machine for enduring the day to day operating environment. It also complicates the layout of the piping system in meeting the allowable. Unusual configurations and restraining systems are often used to make the calculated piping load satisfy the given allowable. However, all these efforts are very often just exercises of computer technology. The main reliability problem has not been solved. A better designed equipment with some common sense piping arrangement is the basis for improving reliability.

ALLOWABLE LOAD

Process equipment, especially the rotating equipment, generally have a very low allowable piping load. Piping engineers often think the manufacturers give low allowables to protect their own interests. This notion is not necessarily true, because many equipment indeed cannot take too much a load. The problem is that a weak link exists that is often overlooked in the design of an equipment. Figure 1 shows a typical pump installation which can be divided into three main parts: the pump body, the foundation, and the pedestal/base plate. Without the input or threat from the piping or equipment engineers, the routine

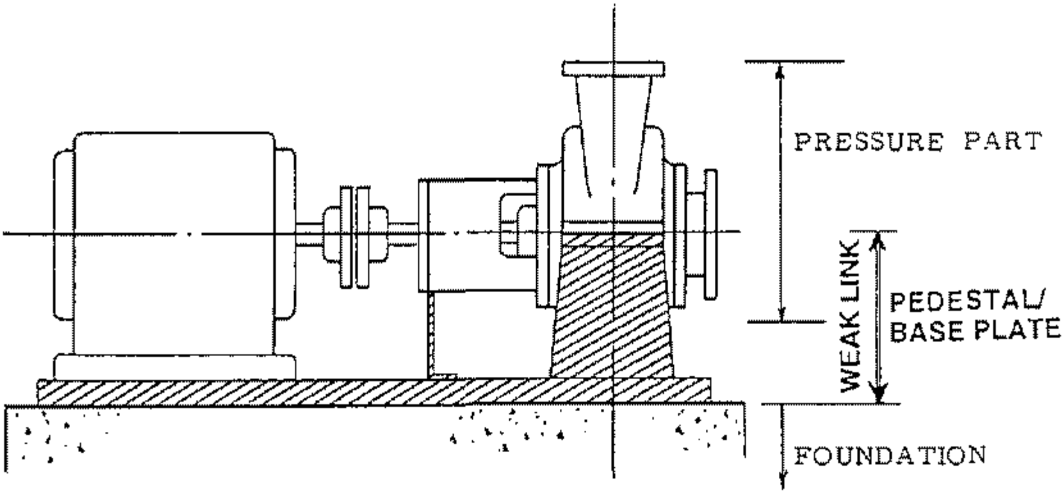


Figure 1, The Weak Link

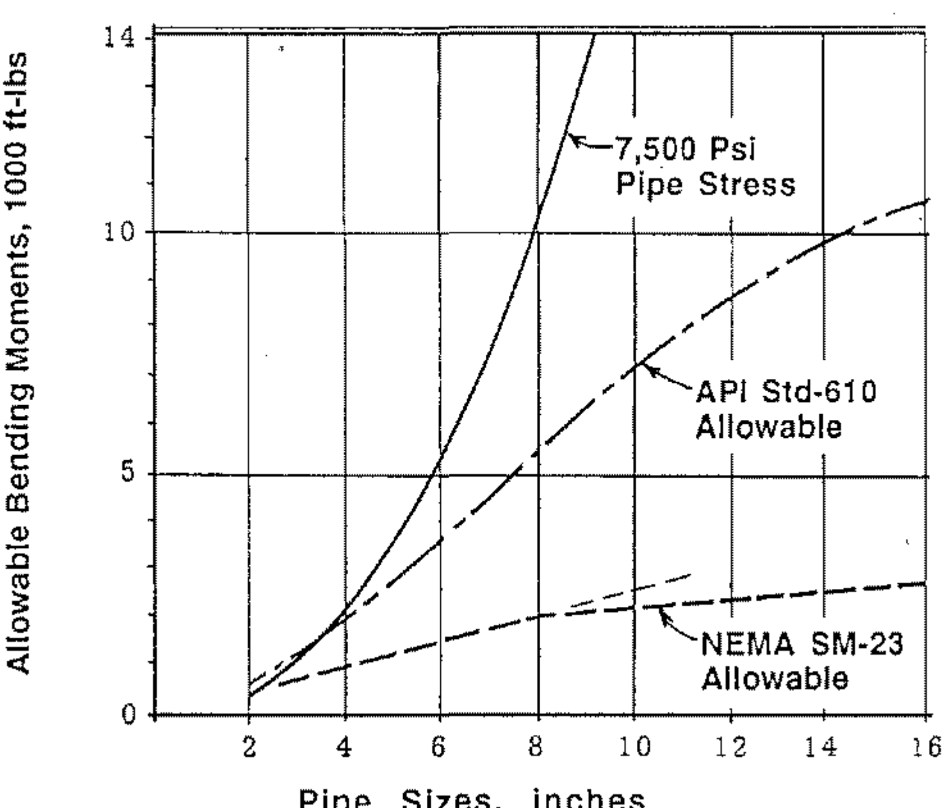


Figure 2, Allowable Piping Loads

design of the pump assembly can have different significance on different parts of the pump. The pump body is designed to be as strong, if not stronger, than the piping so that the body can resist the same internal design pressure as the piping. The foundation, normally designed with the combined pump/motor assembly weight, is also massive and stiff due to the limitations of the soil bearing capacity. However, the pedestal/baseplate is a different story. Without considering the taking of any piping load, the pedestal/baseplate is generally designed only by the pump weight. This design basis creates a very weak pedestal/baseplate which can take very little load from the piping, hence the famous story of the vendor who claimed his equipment cannot take any piping load. Nowadays, most vendors have more sense than to claim such a thing, but the allowable piping load is still not large enough to be desirable. The weak link, of course, is the pedestal/baseplate assembly.

By understanding the situation, the problem can actually be rectified very easily. Improvement has already been seen in pump applications. Pump application engineers who long realized the low allowable piping load problem customarily specified double (2X) or triple (3X) base plates to increase the allowable piping load by two or three times, respectively. Surprisingly, to most engineers, the cost of a 2X or 3X pump was only marginally more than that of a regular pump. Actually, it should not have been the least bit surprising, since all a vendor has to do to make it 2X or 3X is to provide a couple of braces or stiffners. Recognizing the popular demand for the 2X or 3X baseplate, the API formally adopted it to its pump standards. Since the sixth edition of the API Std-610 [1], the allowable has been increased to a level that makes the 2X and 3X specification no longer necessary. In other words, the strength of the whole pump assembly has become fairly uniform that no additional allowable can be squeezed out without adding a substantial cost. Unfortunately, at present this philosophy has not been shared by other manufacturers. For example, the 1956 NEMA [2] turbine allowable load is probably the most unreasonable of its kind. The API Std-617 [3] centrifugal compressor and the ASME/ANSI B73.1 [4] pump are not far behind. The API Std-617 uses 1.85 times the NEMA allowable, and the ANSI B73.1 vendors often use 1.30 times the NEMA values for the allowables. Figure 2 shows the comparison of the pipe strength, the allowable API Std-610 piping load, and the NEMA allowable piping load. The pipe strength curve is based on a 7500 psi bending stress. It should be noted that the allowable pipe stress against thermal expansion can be as much as three times higher than 7500 psi.

Looking at Figure 2, it is clear that the piping load that can be applied to an equipment is much smaller than the strength of the pipe itself. Therefore, in designing the piping connected to an equipment, the equipment allowable load is the controlling factor. For low allowable items, such as a large size steam turbine, an extensive expansion loop, and a restraining system is generally required. This is a fact and should be understood by all parties concerned.

Because of the elaborate design of the piping system attached to a sensitive equipment, engineers may sometimes get too trapped in the computer maze and overlook engineering fundamentals. Typical examples that can cause unreliable operation are discussed in the following.

EXCESSIVE FLEXIBILITY

Adequate piping flexibility at an equipment is required to reduce the piping load to the acceptable value. However, a good design should consider the realistic flexibility from the support structure and the proper use of the protective restraints. Without the properly located restraints, a piping system , no matter how flexible it is, has difficulty meeting the allowable load imposed by the equipment. Figure 3 shows a pump piping system which was designed without any restraints installed. This is a common mistake made by inexperienced engineers who think that a restraint can only increase the stiffness, thus increasing the load. It is true that a restraint will tend to decrease the flexibility of the system as a whole and will increase the maximum stress and force in the system. However, a properly designed restraint can shift the stress from the portion of piping near the equipment to a portion further away from the equipment.

Although extensive loops are used in the piping given in the figure, the piping load still may not meet the equipment allowable due to the lack of a restraining system. The excessive flexibility makes the system prone to vibration, because it is easily excited by small, disturbing fluid forces. In addition, the piping loops enhances the internal fluid disturbance by creating cavities and other flow discontinuities due to excessive pressure drops. A system similar to that shown in Figure 3 experienced very severe vibrations in one petrochemical plant. The operational engineer had to put a large cross beam to anchor all the loops in the field to suppress the vibration to a manageable level. This shows that the function of the original loops were lost by the anchoring system. The piping still experiences larger than normal vibrations due to flow disturbance caused by the loop

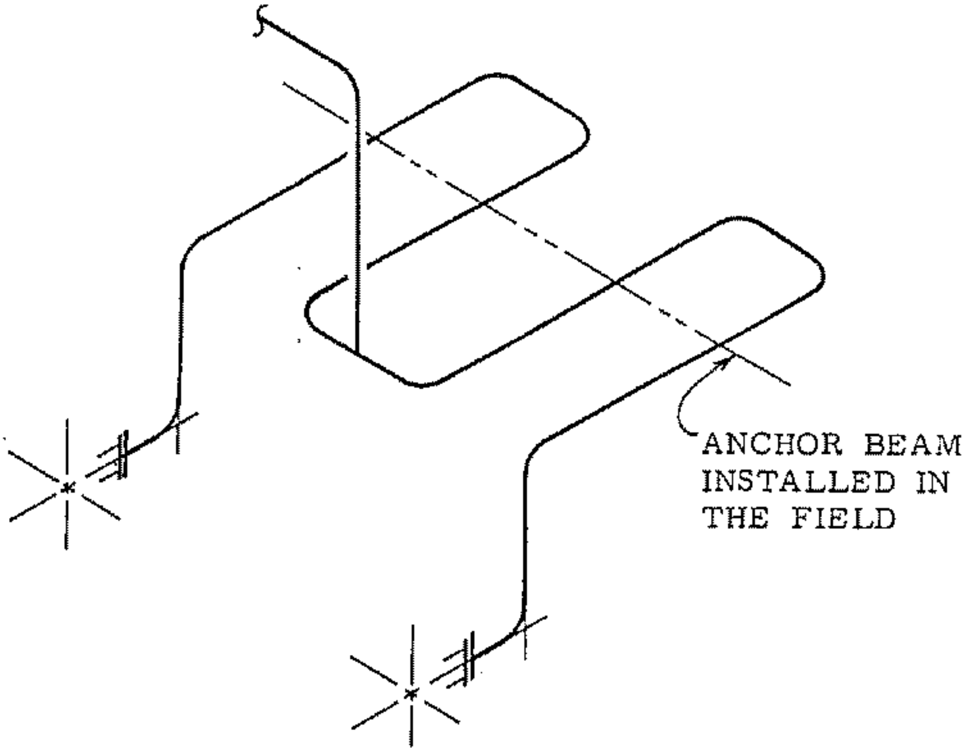


Figure 3, Too Much Flexibility

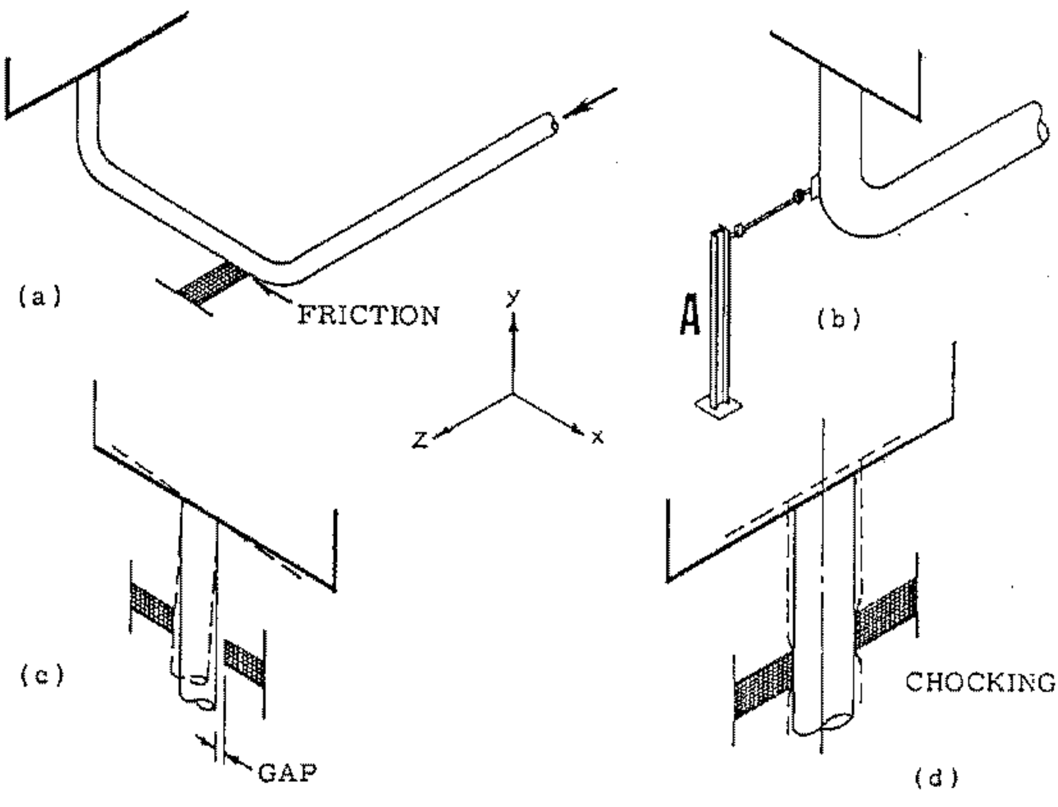


Figure 4, Problems with Theoretical Restraints which is structurally fixed, but hydraulically still open to many directional changes.

THEORETICAL RESTRAINTS

A properly designed piping system generally has some restraints to control the movements and to protect the sensitive equipment. However, there are also restraints which are placed in desperation by piping engineers trying to meet the allowable load of the equipment. These so-called computer restraints give a very good computer analysis result on paper, but are often very ineffective and sometimes even harmful. Figure 4 shows some typical situations which work on the computer, but do not work on a real piping system. These pitfalls are caused by the differences between the real system and the computer model. Some important discrepancies are described in the following.

FRICTION is important in the design of the restraint system near the equipment. Figure 4 (a) shows a typical stop placed against a long Z-direction line to protect the equipment. In the design calculations, if the friction is ignored, the calculated reaction at the equipment is often very small. However, in reality, the friction at the stop surface will prevent the pipe from expanding to the positive X-direction. This friction effect can cause a high X-direction reaction to the equipment. A calculation including the friction will predict this problem beforehand. A proper type of restraint, such as a low friction plate or a strut, would then be used.

An INEFFECTIVE SUPPORT MEMBER is another problem often encountered in the protective restraints. Figure 4 (b) shows a popular arrangement to protect the equipment. The engineers direct instinct is to always put the fix at the problem location. For instance, if the computer shows that the Z-direction reaction is too high, the natural fix is to place a Z-direction stop near the nozzle connection. This may be all right on the computer, but in reality it is very ineffective. For the support to be effective, the support member A has to be at least one order of magnitude higher than the stiffness of the pipe which is very stiff in this case due to the support's relatively short distance from the nozzle.

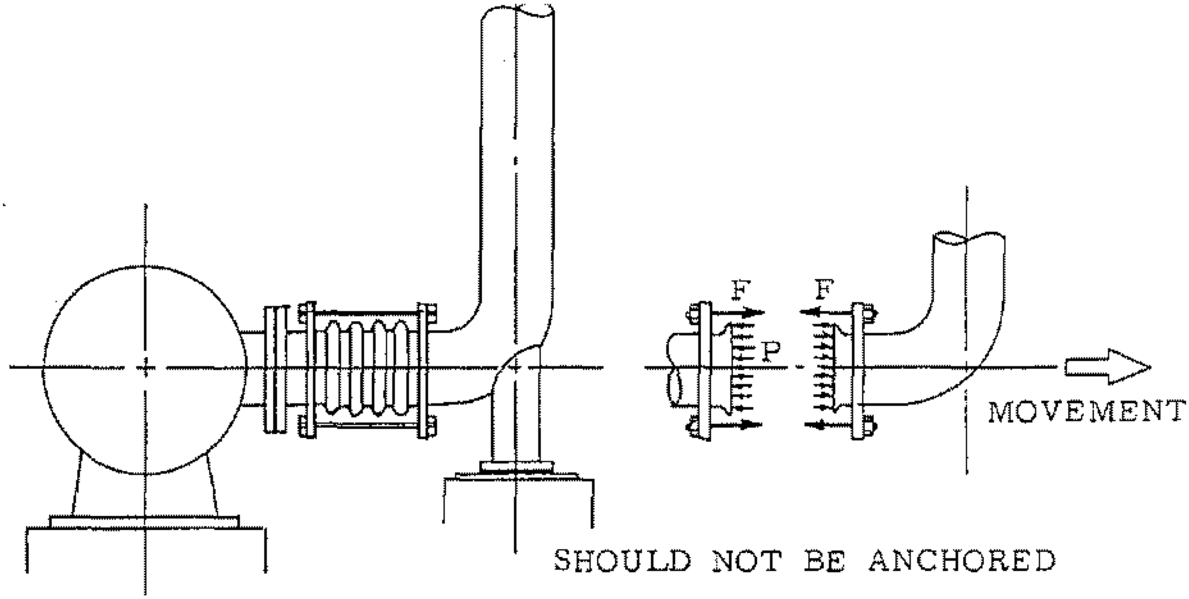
A GAP is generally required in the actual installation of a stop. Therefore, if a stop is placed too close to the nozzle connection, its effectiveness is questionable due to the inherent gap. As shown in Figure 4 (c), because of the gap, the pipe has to be bent or moved a distance equal to the gap before the stop becomes active. Due to the closeness of the stop to the equipment, this is almost the same as bending the equipment that much before the pipe reaches the stop. This is not acceptable, because the equipment generally can only tolerate a much smaller deformation than the construction gap of the stop.

CHOKING is another problem relating to the gap at the stop. Some engineers are aware of the consequences of the gap at the stop mentioned above and try to solve it by specifying that no gap be allowed at the stop. This gives the appearance of solving the problem, but another problem is actually waiting to occur. As shown in Figure 4 (d), when the gap is not provided, the pipe will be choked by the stop as soon as the pipe temperature starts to rise. We all know to pay attention to the longitudinal or axial expansion of the pipe, but we often forget that the pipe expands radially as well. When the temperature rises to a point when the radial expansion is completely choked by the support, the pipe can no longer slide along the stop surface. The axial expansion will then move upward, pushing the whole equipment up.

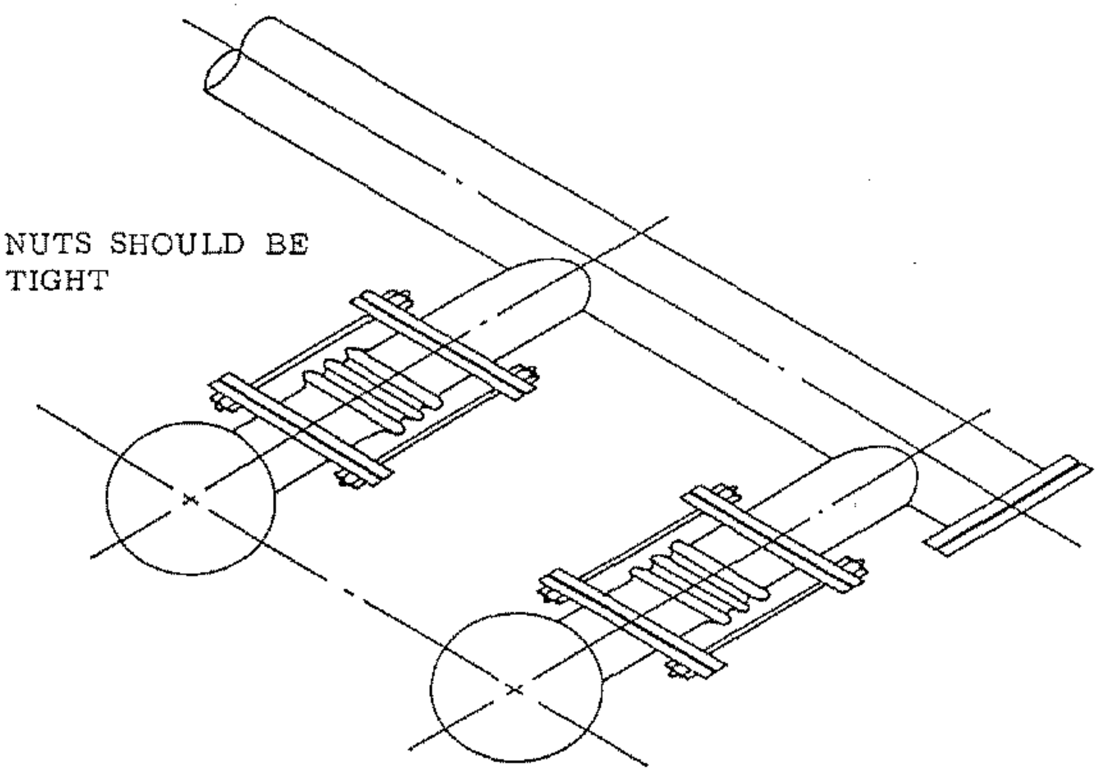
EXPANSION JOINT

An alternative solution to meet the allowable pipe loading to an equipment is the use of bellow expansion joints. Regardless of the constant objection from plant engineers, the bellow expansion joint is very popular in the exhaust system of a steam turbine drive which has an extremely low allowable pipe load for pipes 8" and above. The bellow joints are also often used for fitting the large multi-unit assemblies as shown in Figure 5 (b). Although a properly installed and maintained bellow expansion joint should have the same reliability as other components, such as flanges and valves. In real applications, it is often found to be very undesirable due to the difficulty in maintenance. For instance, when covered with insulation, the expansion joint looks just like a pile of blanketed scraps. Nobody knows exactly what is going on inside the mixed layers of covering. Due to blindness anxiety, many installers have resorted to an uninsulated arrangement. This not only creates an occupational safety concern, but it can also cause cracks due to thermal shock from the environment and/or weather changes.

One important factor often overlooked by engineers in the installation of a bellow expansion joint is the pressure thrust force inside the pipe. The bellow is flexible axially. Therefore, the bellow is not able to transmit or absorb the axial internal pressure end force. This pressure end force has to be resisted either by the anchor at the equipment or by the tie-rod straddling the bellow. With the exception of very low pressure applicators, such as the pipe connected to a



(a)



(b)

Figure 5, Tie-Rods on Expansion Joints

storage tank, most equipment are not strong enough to resist the pressure end force equal to the pressure times the bellow cross section area. The pressure thrust force has to be taken by the tie-rod. Somehow this idea is not obvious to many engineers, resulting in some operational problems. Figure 5 shows two actual problems. Figure 5 (a) shows one of many steam turbine exhaust pipings installed at a petrochemical plant. The expansion joint layout scheme appears to be sound, but the construction was not done properly. The actual installation had a sliding base elbow anchored with four bolts. This problem often escapes the eyes of even experienced engineers. When the base elbow is anchored, the tie-rod loses its function as soon as the pipe starts to expand. In this case, the pipe expands from the anchor toward the bellow joint, making the tie-rod loose and ineffective. The large pressure thrust force pushes the turbine, causing shaft misalignment and severe vibrations. Figure 5 (b) is a similar situation. The bellow expansion joints were used solely for fitting up the connections. The tie-rods were supposed to be locked. However, before the start-up operation, one engineer had loosened the tie-rod nuts, apparently thinking the tie-rods defeat the purpose of the expansion joint. The start-up was very shaky and had to be quickly halted. It took quite awhile before anyone discovered that the problem was caused by the loose tie-rods. When the nuts are loose, the pressure end force simply pushes the pump way out of alignment.

OTHER PRACTICAL CONSIDERATIONS

As discussed above, the reduction of pipe stress is not at all straight forward. Especially when dealing with the low allowable of some equipment, the technique becomes tricky and very often only works on paper. Other practical approaches may be explored to further improve overall reliability. One very important resource often ignored in this country is the experience found in operating plants. We often see a good, simple working layout changed to a complicated, shaky layout only because a computer liked it that way. Undoubtedly, computers are important tools, but they are only as good as the information we give them. Since there are so many things, like friction, anchor flexibility, etc., that cannot be given accurately, computer results need to be interpreted carefully. It is time to realize that if something works well in a plant day in and day out, it should be considered good, regardless of whether or not the computer predicted it to be good. The process of evolution is very important in designing a good, reliable plant.

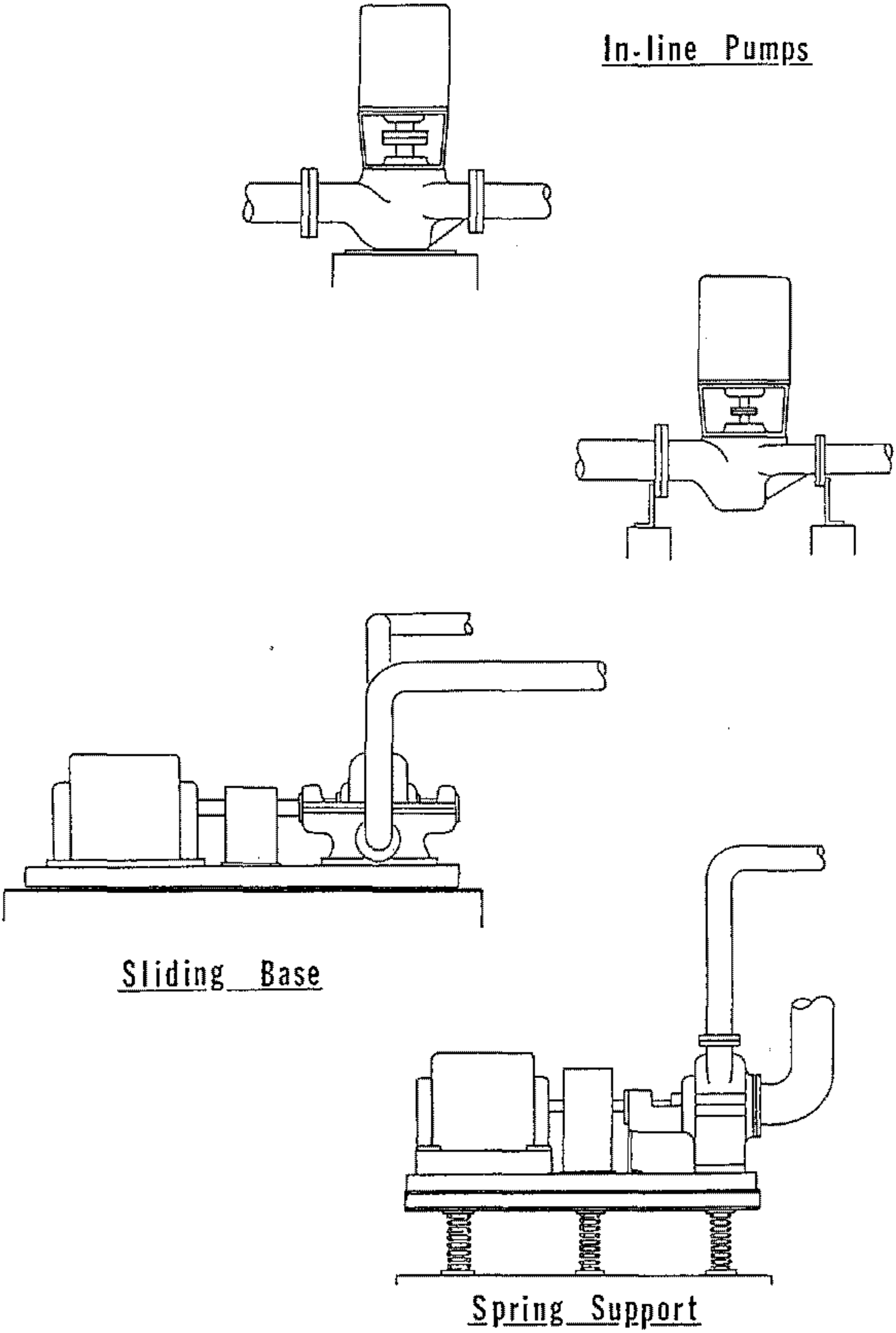


Figure 6, Alternative Machine Assemblies

Other ideas, such as the use of sliding supports, spring supports, and more compact in-line arrangements as shown in Figure 6, can also be seriously considered. It is understood that engineers do not feel too confident on the movable assembly, but it is important to distinguish the difference between the movement of the whole assembly and the movement of only the pump or turbine. When the whole assembly moves, the shaft alignment can still be maintained if the distortion of the equipment is not excessive. That is, if the piping load is still within the allowable. It should be noted, however, that these movable assemblies are just potential alternatives. One should not be oversold on the idea and blindly use it in a plant. To make the sliding base or the spring support scheme workable, an extra strong baseplate is required. Then again, if we have that strong of a baseplate in the first place, the allowable piping load would have increased substantially.

LITERATURE CITED

[1] API Standard 610, "Centrifugal Pumps for General Refinery Services," American Petroleum Institute, Washington, D.C.

[2] NEMA SM-23, "Steam Turbines for Mechanical Drive Service," National Electrical Manufacturers Association, Washington, D.C.

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[4] ASME B73.1M-1991, Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process," American Society of Mechanical Engineers, New York.