# API 610 BASEPLATE AND NOZZLE LOADING CRITERIA

**by**

**James E. Steiger**

**Lead Design Engineer**

**United Centrifugal Pumps**

**San Jose, California**

*James E. Steiger is a Lead Design Engineer with United Centrifugal Pumps of San Jose, California. Prior to joining United, he was involved with centrifugal pumps for the nuclear industry at Pacific Pumps. He has an M.S. degree in Mechanical Engineering from Brigham Young University, and is a registered professional engineer in the State of California.*

## ABSTRACT

The baseplate and nozzle loading criteria in the December 1985 draft version of API 610 7th Edition is substantially different from the criteria found in the 6th Edition. The optional heavy duty baseplate criterion has been eliminated, an ungrouted baseplate stiffness criterion has been added so that designs can be easily verified by a relatively inexpensive test in the manufacturer's shop, and Appendix C, Effect of External Nozzle Forces and Moments on Piping Design, has been retitled and rewritten. These changes and others have been made to clarify the intent of the document, and to make the design criteria both achievable for the manufacturer and enforceable by the purchaser.

Shortcomings found in sub-sections 2.4, 3.3.1, and Appendix C of the 6th Edition are discussed as is the corresponding modified text found in the proposed 7th Edition of API 610 Standard. Results are provided of grouted and ungrouted pump-baseplate stiffness tests. An evaluation chart and baseplate checklist are also included.

## INTRODUCTION

Ideally, a baseplate is a structure, generally fabricated from steel, to which shaft coupled pieces of rotating equipment are mounted and held in alignment for trouble free operation. Proper design, manufacturing, handling and installation are key ingredients necessary to provide reliable service. When one of the pieces of equipment is a pump that is subjected to external loadings via the attached piping, maintaining shaft alignment and minimizing equipment distress can become more complicated. Recognizing this, API Standard 610 puts more emphasis on external piping loads and baseplate design than any other rotating equipment standard. This is justified, due to the number of centrifugal pumps used in a refinery and the economic-reliability trade-off associated with the magnitude of allowable piping loads.

Pump manufacturers and rotating equipment engineers concerned with maintaining the shaft alignment of coupled equipment for operational reliability, prefer small piping loads. Several arguments can be made to defend this position. Field experience indicates that design or calculated piping loads can be significantly different from the loads actually imposed on the installed equipment. The pump-baseplate assembly represents a complex structure whose response to piping loads is difficult to predict with a high degree of certainty.

Contractor and piping designers prefer larger piping loads which result in simpler and significantly less expensive piping configurations. The argument is that the pump manufacturers and rotating equipment engineers are too conservative and the higher piping loads which have been proven by past experience do not usually lead to significant operability problems.

Before the 6th Edition of API 610 was published, there were no industry accepted standards for allowable piping loads acting on centrifugal pumps [1]. (The 5th Edition only addressed pumps with discharge nozzles four inches and smaller in diameter [2]). Each pump manufacturer, contractor and user had their own standards which frequently complicated plant design. The 6th Edition addressed this problem by establishing two sets of standards, one for the pump manufacturer (Table 2 of API 610, presented herein as Figure 1), and suggested values for the piping designer (Appendix C). Besides providing standardized allowable piping loads, the 6th Edition of API 610 defines two stiffness criteria for pump-baseplate and support assemblies (Standard-2.4.3 and Heavy Duty-2.4.6). These criteria were added to establish uniformity to the structural design of pump-baseplate assemblies. The 1985 Pressure Vessel Research Committee (PVRC) Pump-Piping Interaction Experience Survey indicates that API was right on target with the intent of their document [3]. The survey indicates that there is a significant pump-piping interaction problem and that it has an upper annual impact on the order of one-half billion dollars.

Although the intent of the 6th Edition was good, a careful review of the text found in Section 2.4 and Appendix C indicates that the specified design criteria are often vague, and, therefore, subject to interpretation. In most instances, there is not a cost effective method of proving that the design criteria are being met.

| | Nominal Size of Nozzle Flange (inches) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Force/Moment[a] | ≤2 | 3 | 4 | 6 | 8 | 10 | 12 | 14[b] | 16[b] |
| Each top nozzle | | | | | | | | | |
| $F_x$ | 160 | 240 | 320 | 560 | 850 | 1200 | 1500 | 1600 | 1900 |
| $F_y$ (compression) | 200 | 300 | 400 | 700 | 1100 | 1500 | 1800 | 2000 | 2300 |
| $F_y$ (tension) | 100 | 150 | 200 | 350 | 530 | 750 | 920 | 1000 | 1200 |
| $F_z$ | 130 | 200 | 260 | 460 | 700 | 1000 | 1200 | 1300 | 1500 |
| Each side nozzle | | | | | | | | | |
| $F_x$ | 160 | 240 | 320 | 560 | 850 | 1200 | 1500 | 1600 | 1900 |
| $F_y$ | 130 | 200 | 260 | 460 | 700 | 1000 | 1200 | 1300 | 1500 |
| $F_z$ | 200 | 300 | 400 | 700 | 1100 | 1500 | 1800 | 2000 | 2300 |
| Each end nozzle | | | | | | | | | |
| $F_x$ | 200 | 300 | 400 | 700 | 1100 | 1500 | 1800 | 2000 | 2300 |
| $F_y$ | 130 | 200 | 260 | 460 | 700 | 1000 | 1200 | 1300 | 1500 |
| $F_z$ | 160 | 240 | 320 | 560 | 850 | 1200 | 1500 | 1600 | 1900 |
| Each nozzle | | | | | | | | | |
| $M_x$ | 340 | 700 | 980 | 1700 | 2600 | 3700 | 4500 | 4700 | 5400 |
| $M_y$ | 260 | 530 | 740 | 1300 | 1900 | 2800 | 3400 | 3500 | 4000 |
| $M_z$ | 170 | 350 | 500 | 870 | 1300 | 1800 | 2200 | 2300 | 2700 |

NOTE: $F$ = force, in pounds; $M$ = moment, in foot-pounds; Subscript $x$ = horizontal (parallel to horizontal shafts); Subscript $y$ = vertical (parallel to vertical shafts); Subscript $z$ = horizontal (parallel to side nozzle centerlines). See Figure 1 for a diagram of the coordinate system. For vertical and in-line pumps that are turbine driven, use values for side nozzles; for vertical and in-line pumps that are motor driven, multiply values for side nozzles by 2.

[a] In summing moments about any point, the forces, $F$, multiplied by their respective moment arms are to be added to the moments, $M$, to give the total moment.

[b] These values are for guidance only and are subject to negotiation between the purchaser and the vendor for the specific application.

*Figure 1. Table 2—Nozzle Loadings [1].*

With such complicated structures, analytical methods are only as valid as the assumptions that are made concerning the behavior of the equipment. The standard baseplate stiffness criteria is difficult to achieve and frequently results in designs that cannot be economically justified except by the knowledgeable user. In most instances, the heavy duty baseplate stiffness criteria can only be achieved on paper by making appropriate assumptions. As a result of these problems, the intent of the 6th Edition concerning baseplate and nozzle loading criteria is not being achieved.

With so many purchasers and users buying equipment to the API 610 6th Edition specifications, pump manufacturers are forced into advertising compliance to remain in business. As a result of the previously mentioned shortcomings, there are wide variations in the structural design of pump-baseplate assemblies among pump manufacturers. For some manufacturers, the 6th Edition had a significant impact on the structural design of their pump cases and baseplates [4]. On the other hand, some of the pumping equipment being sold appears to be of the 5th Edition vintage, with no modifications whatsoever. In other instances, manufacturers have apparently made different interpretations as to the meaning of the stiffness criteria found in Section 2.4. The 7th Edition Task Force has made significant changes to the new 610 Standard (December 1985 Draft) in order to eliminate ambiguity of text and to make the stiffness requirements realistic and economically enforceable [5].

The intent herein is to make the reader more knowledgeable about API 610 baseplate and nozzle loading criteria, so that better pumping equipment will be selected. Shortcomings of the 6th Edition Standard have been critiqued and the new 7th Edition Standard is presented. Grouted and ungrouted stiffness test data for end suction overhung pumps, along with pertinent design parameters, are presented to provide a means by which pump-baseplate-pedestal support assemblies can be evaluated. A baseplate checklist is also included.

## SHORTCOMINGS OF THE 6TH EDITION STANDARD

### *Section 2.4—External Nozzle Forces and Moments*

As previously mentioned, Section 2.4 of the 6th Edition was written to provide standardized minimum nozzle loads for pumps used in general refinery service and to establish minimum system spring constants for pump-baseplate-pedestal support assemblies. The 0.005 in shaft displacement criterion adopted by the user controlled American Petroleum Institute was intended to eliminate the flimsy (shaft deflections on the order of 0.020 in to 0.050 in) 5th Edition assemblies that were sometimes being produced. Due to the publication of the 6th Edition of API 610, minimum allowable nozzle loadings for pump manufacturers have become standardized. Section 2.4 of the 6th Edition has also caused several manufacturers to upgrade their pump lines and provide heavy-duty pieces of equipment that have spring rates that fall in the 0.005 in to 0.015 in range. Some manufacturers, however, have found it unnecessary to change the structural design of their 5th Edition baseplate assemblies. Economic pressure, lack of knowledge among all parties ane design criteria that are vague, sometimes unrealistic and unenforceable, have prevented the intent of the API 610 standard and heavy-duty baseplate concept from being achieved. Section 2.4 is shown with ambiguous or unenforceable words or phrases italicized. An explanation of the ambiguity follows each Subsection of Section 2.4. It is hoped that a discussion of these shortcomings will make the proposed 7th edition more understandable and help the petroleum industry procure the type of pumping equipment they require.

*Subsection 2.4.1*

> 2.4.1 The requirements of 2.4 shall apply to pumps with suction nozzles 12 inches and smaller in size and with casings constructed of steel or alloy steel. Two effects of nozzle loading are considered: *development of stresses and strains in the case (see 2.4.2) and displacement of the shaft (see 2.4.3). The forces and moments given in Table 2* (Figure 1) are considered minimum loads and *should be adjusted where the vendor has experimental or test data permitting larger values.* The values shall submit *comparable nozzle loadings* for pump cases con.structed of other materials. [1]

The *development of stresses and strains in the case* is *italicized* because Section 2.4.2 does not provide any quantitive values of stress or strain for a designer to use as an acceptance criteria. Even if such values were provided, it would be necessary to also provide a standard method to evaluate casing stresses and strains. Due to the complexity, as well as the numerous variations in pump case geometry, plus the number of potential loading conditions that could be considered, it is not feasible to define standard analysis methods, let alone have them included in the API 610 document.

The phrase, *the forces and moments given in Table 2*, has been questioned because the literal meaning does not convey what was actually intended [1]. This is because the values shown in Table 2 of API 610 6th Edition (Figure 1) are unsigned and can be interpreted to mean one specific load set. Each of the loads shown in Table 2 (Figure 1) was intended to represent a range that could vary from plus to minus the values shown. For example, 160 lb, referring to Table 2 (Figure 1) for a two inch or smaller nozzle, was meant to represent any value between plus or minus 160 lb and not just plus 160 lb, as indicated. The difference between the literal and intended interpretation of Table 2 loads can affect a factor of two change in the required stiffness of the pump and its support assembly, which may explain some of the variations in construction features among manufacturers.

The literal interpretation *of the forces and moments given in Table 2*, i.e., a single load set, also simplifies the effort required to demonstrate an adequate design. Only one loading condition must be considered instead of 4096. The latter represents the number of possible load combinations if one considers the upper and lower bounds of the intended range for each component load.

As previously indicated, the pump manufacturer prefers small piping loads (one times Table 2 (Figure 1) values for horizontal pumps) to ensure reliable operation. This being the case, the pump manufacturer has little motive to analytically or experimentally prove that his equipment can withstand higher nozzle loadings as suggested by the third sentence of Subsection 2.4.1. In addition, Appendix C of the API 610 Standard suggests that piping designers can impose higher nozzle loadings than those guaranteed by the OEM. The knowledgeable user will compare and reconcile significant piping load differences between the pump vendor and the piping designer. In many instances, the pump manufacturer can accept nozzle loads outside the ranges given in Table 2 (Figure 1), if specific loading conditions can be specified.

What are *comparable nozzle loadings*? This is an undefined term. The text in the second sentence of Subsection 2.4.1 implies that comparable nozzle loads would produce the same relative stress, strain and no more than 0.005 in of shaft displacement. Based on this premise, comparable nozzle loadings can be obtained by multiplying Table 2 (Figure 1) values by the smallest of the following ratios: 1) allowable stress of other material/allowable stress of steel or 2) Young's modulus of other material/Young's modulus of steel. For example, a cast iron pump casing

(A-48 Class 30 with an ASME Boiler Pressure Vessel Code (BPVC) allowable tensile stress of 3000 psi) would be limited to nozzle loadings that are 17 percent of Table 2 (Figure 1) values [6].

*Subsection 2.4.2*

'2.4.2" The pump shall be capable of withstanding double *the forces and moments in Table 2* applied simultaneously to the pump through each nozzle, in addition to internal pressure, *without causing an internal rub or adversely affecting the operation of the pump or seal.* [1]

As previously explained, there are some interpretation problems with the phrase, *the forces and moments in Table 2* (Figure 1). It is interesting to notice that instead of stress and strain (reference subsection 2.4.1), subsection 2.4.2 defines the acceptance criteria for case nozzle loading in terms of functional requirements: *internal rub or adversely affecting the operation of the pump or seal.* Fortunately for the manufacturer, the allowable nozzle loads shown in Table 2 (Figure 1) are small and even when doubled for design purposes probably do not cause equipment distress in most API 610 pumps. It should be pointed out that although virtually all manufacturers accept this subsection for horizontal pumping equipment, they generally guarantee operation with Table 2 (Figure 1) loads. Appendix C and experience indicates that actual piping loads can differ significantly from the values shown in Table 2.

This subsection may describe what the user desires, but it is economically not enforceable and therefore never challenged. No manufacturer or user has been known to successively apply double Table 2 (Figure 1) loads simultaneously, plus internal pressure, to demonstrate an adequate design. It is not economically feasible. If pressured by the purchaser to prove compliance to this requirement, most manufacturers, for a price, would furnish simple strength of materials type calculations whose validity would depend on the nature, of the assumptions made in the analysis. For more money, a finite element analysis could be performed, eliminating some, but not all of the assumptions made concerning the behavior of the equipment and the imposed loading conditions. There is also a problem of semantics: *internal rub* is not a precise term since rotating and stationary parts of centrifugal pumps occassionally touch even when no piping loads are imposed. For this reason there are undoubtedly differences in opinion between manufacturers, contractors, inspectors, and users on what constitutes an *internal rub*. Likewise, how does one decide when nozzle loadings are *adversely affecting pump or seal operation*? Equipment failures are easy to identify; specific causes are much more difficult to come by especially in a refinery where there are so many variables to consider.

*Subsection 2.4.3*

2.4.3 *The pump, baseplate, and pedestal support assembly* shall be adequate to limit the shaft displacement measured at the coupling on the installed pump to a maximum of 0.005 inch in any direction when subjected to *the forces and moments in Table 2*. These forces and moments are to be simultaneously applied to the pump through each nozzle. (This shaft displacement is a measure of the stiffness of the assembly for design only and is not an allowable value for pump operation. Realignment at normal operating temperatures is recommended.) The pump will not always be subjected simultaneously to all the forces and moments in Table 2. *When loads in one or more directions are significantly less than those in Table 2, the purchaser may request and the vendor shall then advise of load increases in the other directions that will satisfy the above criteria* [1].

All API 610 pump manufacturers advertise that they meet the requirements of this subsection. This is necessary to promote business. Typical sales literature to this effect is illustrated in Figure 2 through Figure 6. A survey taken by a major engineering contractor confronted with a client specifying pump-baseplate assemblies five times stiffer than those required by this subsection indicates the written response time of nine API 610 pump manufacturers concerning the stiffness of their equipment. The results of this survey are shown in Appendix A-1 (Table 1). The consensus was that the stiffness requirements found in Section 2.4 are achievable. Only when the wide variations in structural design of pump assemblies among manufacturers are compared is there a hint that there are major differences in interpretation concerning the requirements of this subsection. Structural rigidity of the installed pump-baseplate assembly is very much a function of: 1) the size and thickness of the pump mounting pads and the tendons that attach them to the pump casing; 2) the size, number and tightness of the pump hold-down capscrews; 3) the thickness, size, height and shape of the support pedestals; 4) the size of the support pedestal to baseplate attachment welds; 5) the

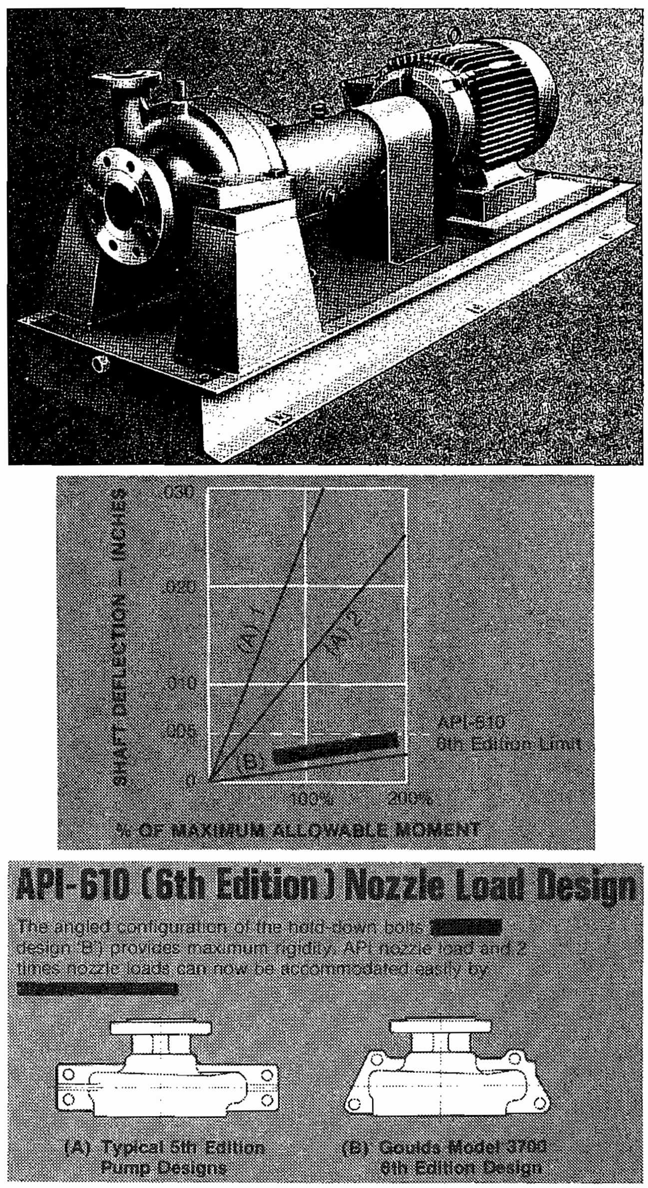
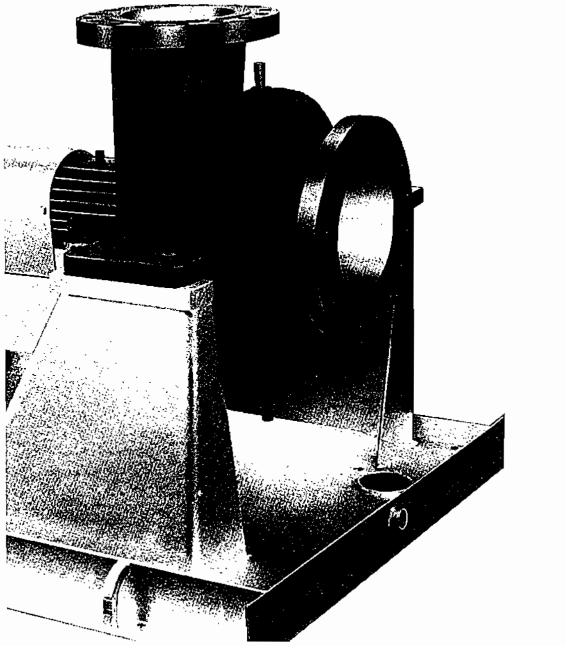



*Figure 2. Sales Literature, Exhibit "A."*

**API 610 6th Edition**
**Heavy Duty Process Pump**



**API 610 nozzles loadings/baseplates**

The [redacted] heavy duty casing and baseplate designs evolved from extensive nozzle load testing. The designs exceed API 610 6th Edition nozzle load criteria without exceeding .005 inch shaft deflection at the coupling. Standard grouted baseplates conform to API standard nozzle loadings. Optional heavy duty baseplates are available to accept API standard nozzle loads in the **ungrouted** condition and **double** the API standard loadings in the grouted condition. All baseplates conform to the dimensional standards established in appendix G of API 610 6th Edition.

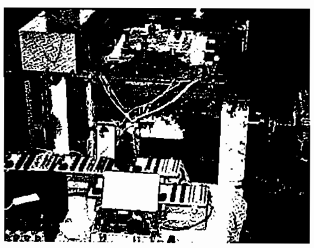


*Figure 3. Sales Literature, Exhibit "B."*

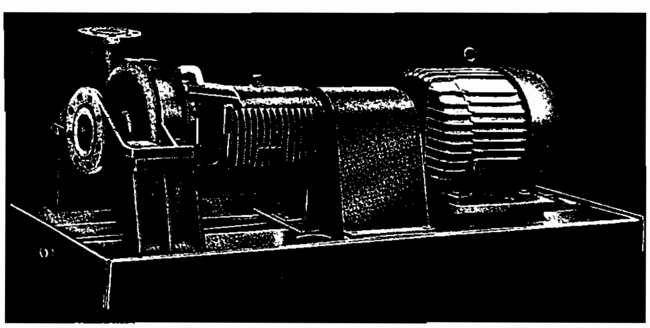


*Figure 4. Sales Literature, Exhibit "C." "The baseplate is of fabricated steel construction, computer designed for rigidity to limit shaft displacement at coupling in compliance with API Standard 610 for Forces and Moments."*

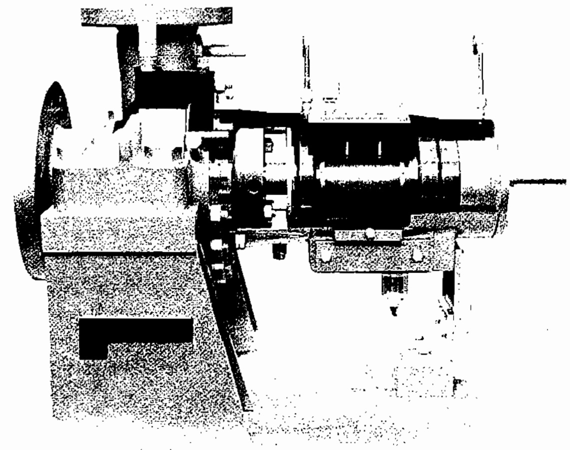


*Figure 5. Sales Literature, Exhibit "D." "The line meets or exceeds API 610 requirements and refinery specifications. Heavy duty foot design to meet API nozzle loading criteria. Design approach, verified by both factory testing and field testing, has resulted in a pump system (casing, support pedestals and baseplate) fully capable of meeting the simultaneous nozzle loads imposed by API Standard 610. Casing strain and pump shaft deflection have been reduced to substantially increase service life—under the most rigorous operating conditions."*

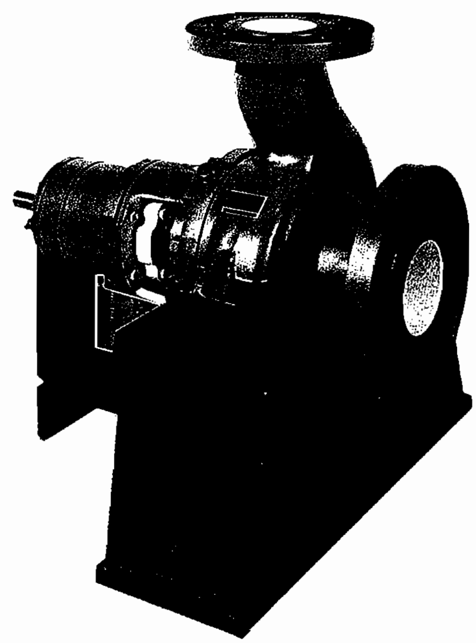


*Figure 6. Sales Literature, Exhibit "E." "For resinery, chemical and petrochemical services, hot water and cryogenics. Meets and exceeds requirements of API 610, sixth edition."*

size, number and location of the structural members that form the baseplate; and 6) the size, number and location of the grout and vent holes which can be contributing factors on whether or not there will be voids between the underside of the baseplate and the foundation. The pump-baseplate assembly can be no stiffer than its most flexible component. In some opinions, purchasers, contractors and users have been too lax in asking manufacturers to explain these differences in construction and their interpretations of this subsection.

The phrase, *The pump, baseplate, and pedestal support assembly* does not clearly define whether or not bearing housing supports or wobble plates can be used to achieve the 0.005 in shaft displacement criterion. A bearing housing support is a stay between the coupling end of a bearing housing and the baseplate and is capable of reacting to upward and downward vertical loads. A wobble plate is a brace bolted only to the bearing housing and bearing against the top of the baseplate making it capable of reacting only to downward vertical loads. These devices are commonly furnished on overhung process pumps and can significantly reduce the shaft displacement at the coupling if the pump casing is flexible, compared to the baseplate and support pedestals. The effectiveness of these devices is dependent upon the amount of load transmitted to them, which in turn is a function of the magnitude and direction of the applied loading, the flexibility of the casing, the foot to support pedestal interfaces, the bearing housing which is typically cast iron, and the support or wobble plate itself. In the case of the wobble plate, its effectiveness is also dependent on how it is positioned vertically. Since the performance of these devices are dependent on so many variables, most of which are difficult to predict analytically, it makes more sense to provide adequate casing stiffness and use bracket supports to help solve unforseen problems.

The phrase, *the forces and moments in Table 2* implies a single load set, as previously explained.

The words *significantly less* does not have a precise meaning and requires interpretation. As used in the context of Section 2.4.3, the significance of a load would depend not only the magnitude and the direction of the load, but also the directional stiffness characteristics of the pump and baseplate structure. The entire last sentence of Section 2.3 is misleading, since it implies that pump and baseplate structures have equal spring rates in all directions. From experimental test data presented later, it will be clear that this is not the case.

*Subsection 2.4.4*

2.4.4 The preceding paragraphs and the forces and moments in Table 2 are criteria for the pump design. The purchaser is referred to Appendix C for *guidance* in determining the allowable piping loads for a specific installation [1].

Appendicies of API Standards are provided for *guidance* only and are not considered as a binding portions of the specification.

*Subsection 2.4.5*

*2.4.5 The vendor shall submit criteria for pump flanges larger than 12 inches* [1].

*Subsection 2.4.6*

2.4.6 If specified by the purchaser, the vendor shall provide a *heavy-duty baseplate and support* intended to simplify piping layouts by allowing higher loads from piping. These baseplates shall be adequate to limit the shaft displacement measured at the coupling on the installed pump to a maximum of *0.005* inch in any direction when subjected to *forces and moments double those described in 2.4.3* [1].

The heavy-duty baseplate criterion is in conflict with the scope of the 610 Standard: "This standard covers the minimum requirements for centrifugal pumps for use in petroleum refinery service." The standard baseplate criterion defines the minimum requirements of the Standard. This point is made, because the criterion in Subsection 2.4.6 is generally not economically achievable for standard overhung process type pumps, assuming that the nozzle loadings in Table 2 represent a range and that rigid bearing housing supports are not used to limit shaft displacement. The test in Subsection 2.4.6 implies that the stiffness of the pump-baseplate assembly can be doubled by furnishing a heavy-duty baseplate with a standard pump. This is not true, assuming that the word *support* refers to the pedestal support assembly referenced in Subsection 2.4.3. Analytical and experimental test data indicate that there is significant flexibility in standard pump casings and in the bolted pump to support pedestal interfaces. As long as this is true, a heavy-duty baseplate will not significantly increase the stiffness os the pump-baseplate assembly. The pump-baseplate assembly can be no stiffer than its most flexible component.

The heavy-duty criterion also introduces potential operability problems for pumps in hot service. In most instances, the support pedestals of process type pumps absorb the thermal growth of the pump casing transverse to the pump shaft. Even with the standard baseplate criterion (Subsection 2.4.3), this can be a real concern, since support pedestal to deckplate/structural member weld stresses can get high. If the heavy-duty baseplate criterion was achieved by furnishing a special pump casing, bracket support and a heavy-duty baseplate, there is a chance that as the pump comes up to temperature, it will slide across the top of the support pedestals and move out of alignment. If this does not occur, there can be yielding in the pedestal structure, which also can affect shaft alignment.

*Subsection 2.4.7*

2.4.7 If specified by the purchaser, the vendor shall provide *calculations* and/or *available supporting test data* for shaft displacement at the coupling with *forces and moments applied according to 2.4.3* or *2.4.6* [1].

Subsection 2.4.7 was intended to provide the means whereby the purchaser could obtain documentation that the pump-baseplate assembly had been adequately designed to meet the requirements of Subsection 2.4.3 or 2.4.6. Due to the costs associated with testing grouted in place baseplates, most manufacturers supply calculations to demonstrate compliance to the 0.005 in shaft displacement criterion. Besides the cost of a test baseplate, the labor and materials associated with grouting, a loading apparatus is required. A test rig capable of applying forces and moments simultaneously to the pump through each nozzle as required by Subsection 2.4.3 would be more expensive and sophisticated than the pump assembly itself.

As indicated in Subsection 2.4.2, to analytically describe the behavior of a pump-baseplate assembly subjected to external nozzle loadings, many assumptions must be made. This does not include interpretation of the meaning of Table 2 loads. The validity of these assumptions determines the accuracy of the resultant shaft displacement. Unfortunately, many of the assumptions are difficult to validate without experimental test data. For a strength of materials type of analysis, one could expect to find simplifying assumptions that deal with the following unknowns:

What are the stiffness characteristics of the pump casing itself? With respect to the baseplate and pump support pedestals consider the pump casing:

- Rigid in all directions and for all loadings,
- Rigid in some directions and for some loadings,
- Flexible in all directions and for all loadings.

How does the bolted pump foot to support pedestal interface introduce flexibility into the system? Possible assumptions about the behavior of these bolted joints include:

- The pump feet are integral with the tops of the support pedestals and no flexibility is introduced into the system.
- The pump feet are integral with the tops of the support pedestals, but have some degree of fixity. This means that some flexibility is introduced into the system, but the pump feet cannot slide across the tops of the support pedestals.
- The behavior of the zolted pump foot to support pedestal joint is a complicated interface whose behavior is dependent upon:
  - fastener size, number and preload,
  - coefficient of friction between the bearing surfaces,
  - magnitude, direction and type of external nozzle loading,
  - magnitude of pump casing thermal growth and support pedestal flexibility,
  - thickness of the foot and the mounting pad.

How will a bearing housing support or wobble plate affect shaft displacement? As discussed in Subsection 2.4.3, the effectiveness of these devices is difficult to predict, due to the number of variables involved.

What are the stiffness characteristics of the support pedestals?

- Assume some degree of fixity at the deck plate or structural member to support pedestal interface,
- Assume support pedestal bending and torsional flexibilities can be described using long beam theory neglecting end effects,
- Assume support pedestals move together and can therefore be idealized as a single equivalent beam whose moment of inertia can be evaluated using the parallel axis theorem.

What are the stiffness characteristics of the deckplate and structural member assembly?

- Rigid in all directions and for all loads,
- Rigid in some directions and for some loadings,
- Flexible depending upon:
  - the effect of grout,
  - size and weight of structural members,
  - size, weight and location of cross members,
  - weld size,
  - thickness of deck plate and means of attachment.

*Subsection 3.3.1—Baseplates for Horizontal Pumps*

In the INTRODUCTION, it was mentioned that proper design, manufacturing, handling and installation are key ingredients necessary to provide reliable service of rotating equipment. There are a few areas in Subsection 3.3.1 where clarification of existing text, through added or revised criteria, result in better design and simplify the handling and installation of baseplates. Subsections of 3.3.1 that appear inadequate are printed with key phrases highlighted by *italics*.

> *Subsection 3.3.1.3*
>
> 3.3.1.3 All mounting pads shall be fully machined flat and parallel to receive the equipment. Corresponding surfaces shall be in the same plane within .002 inch per foot of distance between pads, *as machined* [1].

The phrase, *as machined*, indicates that the installed baseplate shall have mounting pads that are in the same plane, without subsequent machining. What is not indicated is that manufacturers frequently handle baseplates differently than contractors or users, forklifts vs. cranes, and that fabricated steel baseplates before grouting are flexible; the main structural members are not flat and the shape of a baseplate can change as it is handled. Prior to grouting, the baseplate is to be leveled to the 0.002 in/ft criterion, using the machined pads as the reference. Vertical leveling screws, or less preferably, wedges and shims, are used to reposition the baseplate to its as-machined configuration, so that the mounting pads are in plane. Recognizing the characteristics of fabricated steel baseplates and to simplify leveling operations, baseplates can be machined with shims installed between the main structural members and the machine bed, to reduce distortion as it is clamped down. If provisions for lifting are not shown on the outline drawing, chances are the manufacturer has not considered how the baseplate will be handled by others. Depending upon the length of the baseplate, the size of the structural members, the weight and position of the mounted equipment, which may need to be removed, and how the unit is rigged for lifting, permanent distortion can take place, making it difficult to level the baseplate to the 0.002 in/ft criterion.

> *Subsection 3.3.1.4*
>
> 3.3.1.4 Baseplate and pump supports shall be constructed and the pumping unit mounted *to satisfy the requirements of 2.4 for external forces and moments and to minimize misalignment caused by other mechanical forces, such as internal differential thermal expansion and hydraulic piping thrust.* The underside of fabricated baseplates beneath the pump and driver supports shall be *welded in order to reinforce crossmembers* and shall have members shaped to lock positively into the grout to resist upward movement of the baseplate [1].

The first portion of the first sentence of this subsection merely alerts the reader that the structural design criteria for baseplates can be found in Section 2.4—External Nozzle Forces and Moments. The second portion of the first sentence indicates that mechanical loads other than those produced by the attached piping can cause pump to driver shaft misalignment and should be accounted for in some manner. Although the intent of this phrace is not clear, it may have been included primarily for hot multistage pumps, where there can be significant axial growth of the pump casing. For these applications, provisions (dowel and sliding key arrangements) are normally made that allow the pump casing to expand away from the driver.

The last sentence of this subsection makes little sense the way it is written. The obvious intent was to require angle or channel cross bracing under the pump and driver supports. These structural members are to be securely welded in place to provide reinforcement for the pump and driver supports.

> *Subsection 3.3.1.7*
>
> 3.3.1.7 When specified by the purchaser, the pump, baseplate, and pedestal support assembly shall be *sufficiently rigid to be mounted without grouting and satisfy the requirements of 2.4.3* [1].

This optional requirement is usually invoked for pumping equipment used in the oilfield or on offshore platforms. It may, however, be applicable for spring-mounted or free baseplates that are occasionally used in the refinery when there is not adequate space for a flexible piping configuration [7, 8]. To comply with the requirements of this subsection, the main structural members and cross members must be substantially larger and heavier than those found on a groutable baseplate. The James and Horrell articles should be reviewed if spring-mounted or free baseplates are to be used [7, 8]. Meeting the requirements of Subsection 2.4.3 without grout may not result in an adequate spring-mounted baseplate.

> *Subsection 3.3.1.8*
>
> 3.3.1.8 When specified by the purchaser, a *heavy-duty grouted-type baseplate per 2.4.6* shall be furnished [1].

This subsection should be deleted, since the stiffness requirements of Subsection 2.4.6 are generally unrealistic and inconsistent with the scope of this Standard. Refer to the comments concerning Subsection 2.4.6.

*Appendix C - Effect Of External Nozzle Forces and Moments On Piping Design*

As mentioned in the INTRODUCTION, the API 610 Standard provides two nozzle loading criteria, one for the pump vendor (Table 2) and another for the piping designer (Appendix C). Although Appendix C is not a binding portion of the 610 document, it is usually invoked by the piping designer, since being an Appendix to the API Standard, it is easy to justify and equally important and it permits higher nozzle loadings than normally allowed by the pump vendor. As with Section 2.4 and Subsection 3.3.1, there have been problems interpreting the meaning of the text in Appendix C. Rather than underlining ambiguous phrases, a concice flowchart, Figure 7, which was prepared by the C.F. Braun Company and ruled by the American Petroleum Institute as a correct interpretation, is provided. With this flowchart, the procedure to determine if the design piping loads are acceptable becomes straightforward. To estimate the magnitude of the maximum component nozzle loads that can be imposed on API 610 pumps with standard baseplates by the

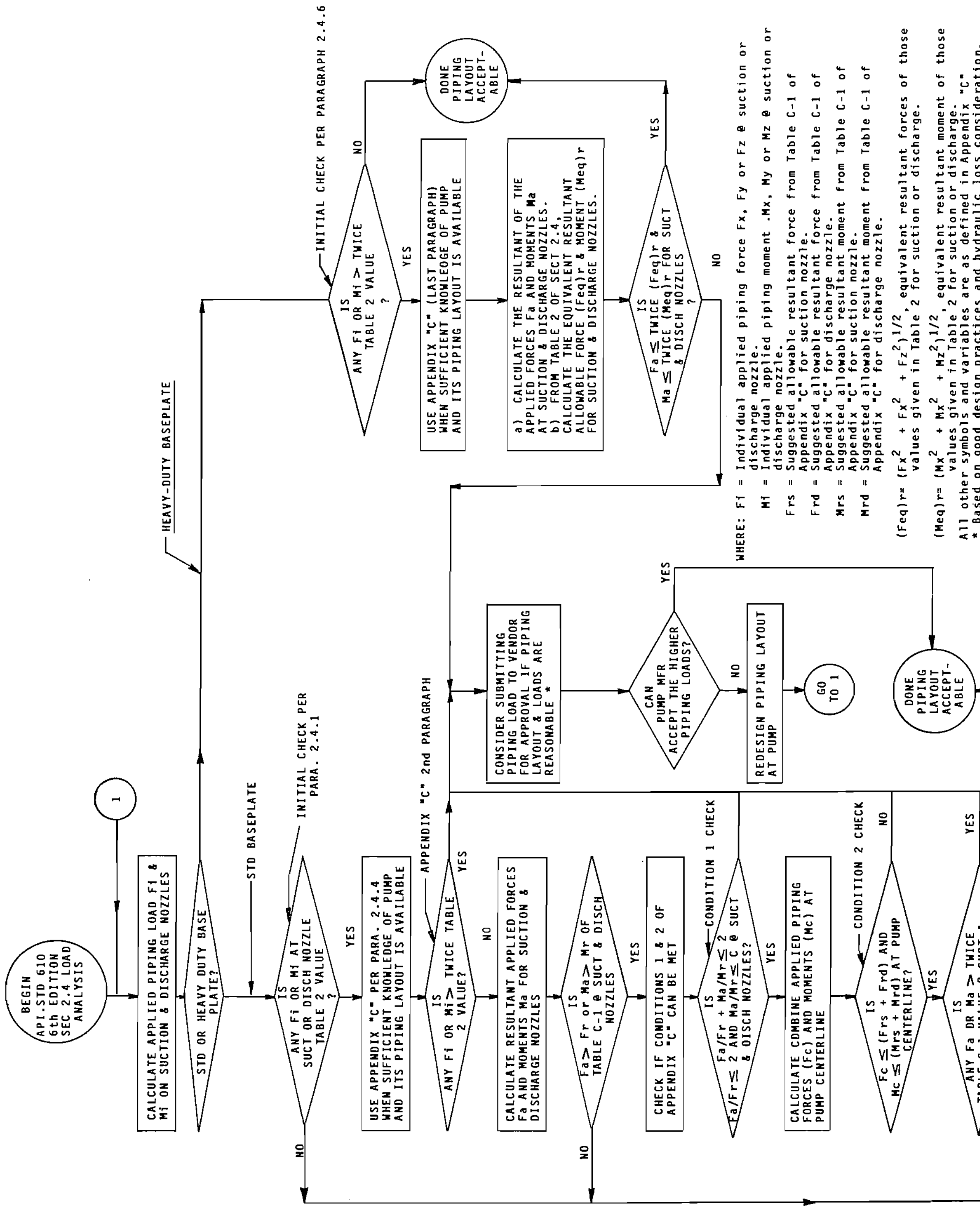



*Figure 7. Flowchart for APPENDIX C [1].*

TABLE 3

COMPARISON OF ALLOWABLE PUMP AND ALLOWABLE PIPING LOADS, Ref. 1

| | Nominal Size of Nozzle Opening (inches) | | | | | | |
|---|---|---|---|---|---|---|---|
| | 2 | 3 | 4 | 6 | 8 | 10 | 12 |
| $\frac{FA}{FR} = \frac{FR_{2XT2}}{FR_{C-1}}$ | 1.35 | 1.34 | 1.32 | 1.34 | 1.36 | 1.63 | 1.80 |
| $\frac{MA}{MR} = \frac{MR_{2XT2}}{MR_{C-1}}$ | 1.33 | 1.36 | 1.33 | 1.32 | 1.34 | 1.52 | 1.49 |
| FA/FR + MA/MR | 2.68 | 2.70 | 2.65 | 2.66 | 2.70 | 3.15 | 3.29 |
| C | 2.00 | 2.00 | 2.00 | 2.00 | 1.75 | 1.60 | 1.50 |

attached piping, Table 3 has been compiled. Studying the parameters shown in Table 3 with Figure 7, it is found that loads on the order of double Table 2 (Figure 1) values can generally be justified. For heavy-duty baseplate construction, the flowchart indicates that component loads can be greater than double Table 2 values, but when combined on a nozzle must not exceed twice the resultant value using Table 2 values as a basis. For either type of baseplate construction, Appendix C piping loads should not affect pump reliability, providing the pump vendor is using the range concept for Table 2 values and meeting the criteria found in Section 2.4. As will be shown in the next section (GROUTED TEST DATA), the pump baseplate assembly stiffness criteria are not always satisfied in this manner.

## GROUTED TEST DATA

In May of 1984, the API 610 Seventh Edition Task Force, in conjunction with the API 610 Manufacturers Subcommittee on Centrifugal Pumps, decided to embark on a rigorous experimental test program to determine the stiffness characteristics of overhung process pumps mounted on grouted-filled baseplates. With this test data, the Task Force believed that they could establish realistic pump and baseplate stiffness criteria that would be acceptable to all concerned parties. The program was designed around the horizontal overhung process pump, since it is the workhorse of the refining industry, and believed to be most susceptible to piping loads. The results of this test program are illustrated in Figure 8. Before testing was underway, the Task Force had anticipated that shaft displacements would increase with pump nozzle size and that a linear acceptance criterion could be established. Although the required stiffness of the pump-baseplate assembly increases with pump size due to increased nozzle loading with a constant allowable shaft displacement, test data did not indicate this trend. Instead, the program indicated that there was considerable variance in the stiffness among the equipment tested. The shaft displacement values are scattered between 0.0066 in and 0.0398 in, indicating "apparent violations" to the 610 Standard, and inconsistency with the published sales literature. These are referred to as "apparent violations" since one does not know how each manufacturer interprets Section 2.4 of API 610 6th Edition, nor have the guidelines for the Task Force program been stated.

If the manufacturers have assumed a literal interpretation of the phrase, "*the forces and moments shown in Table 2*," then all pumps shown in Figure 8, with shaft displacements less than 0.010 in could be construed to be in compliance. If bracket supports are also considered an acceptable means of providing stiffness, then all of the pumps tested with shaft displacements less than 0.015 in are probably adequate.

Needless to say, the 610 Task Force adopted the "range" interpretation of "*the forces and moments shown in Table 2*," and elected not to take credit for stiffness achieved by bearing housing supports or wobble plates. (Note that this does not mean that pumps cannot be equipped with bearing housing supports or wobble plates.) Since the Task Force was interested in determining the maximum shaft displacement for each pump to be tested, it was only necessary to consider the upper (positive) and lower (negative) bounds for the loads shown in Table 2. Intermediate values, although they represent possible loading conditions, need not be considered when evaluating a maximum

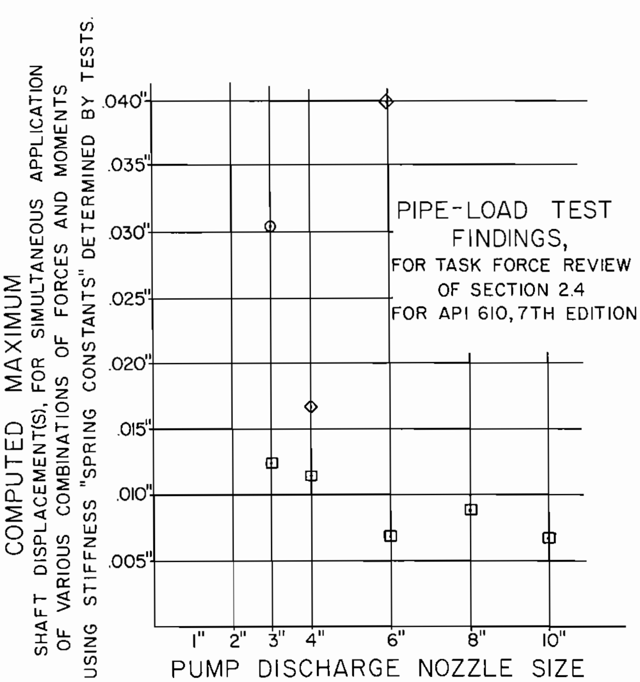



*Figure 8. Pump Shaft Displacement vs. Nozzle Size.*

```
                    ***JOB IDENTIFICATION: 10 X 12 X 19 TC***

THE MAXIMUM SHAFT DISPLACEMENT @ COUPLING = 6.6 MILS

THE COMBINATION OF FORCES AND MOMENTS TO PRODUCE MAXIMUM DISPLACEMENT ARE:

       NOZZLE:     FX      FY      FZ      MX      MY      MZ
      SUCTION:   1800.  -1200.   1500.   4500.   3400.  -2200.
    DISCHARGE:   1200.    750.   1000.   3700.   2800.  -1800.

THE SUMMATED FORCES AND MOMENTS @ THE CENTROID ARE:

                   FX      FY      FZ      MX      MY      MZ
                 3000.   -450.   2500.   8684.   6582.  -7217.

=========================================================================

                             ***INPUT DATA***

THE GIVEN FORCES AND MOMENTS ARE:
       NOZZLE:     FX      FY      FZ      MX      MY      MZ
      SUCTION:   1800.   1200.   1500.   4500.   3400.   2200.
    DISCHARGE:   1200.    750.   1000.   3700.   2800.   1800.

THE DISPLACEMENT CONSTANTS USED IN THE CALCULATIONS ARE:

         FXX=  3.59E-07       FXY=  3.59E-07       FXZ=  1.25E-07
         FYX=  4.19E-09       FYY=  3.35E-07       FYZ= -6.25E-08
         FZX=  0.0            FZY=  0.0            FZZ=  8.13E-07
         MXX=  0.0            MXY=  3.75E-08       MXZ=  4.88E-08
         MYX= -6.41E-08       MYY= -5.00E-08       MYZ=  1.16E-07
         MZX= -1.07E-07       MZY= -5.63E-07       MZZ= -6.67E-08

NOZZLE COORDINATES RELATIVE TO THE CENTROID:

            NOZZLE:         X            Y            Z
           SUCTION:      12.125       -0.250        0.0
         DISCHARGE:      -3.875       18.000       15.750
```

*Figure 9. Computer Printout for the Ten Inch Pump (Grouted Baseplate).*

shaft displacement. With twelve events (six component forces and six component moments, three each per nozzle), and two possible combinations for each event (plus or minus), there are $2^{12}$ or 4096 different ways in which the upper and lower bounds of the components loads may be applied. Knowing this, the Task Force realized that it was not practical nor economically feasible to load and directly measure maximum shaft displacements. Instead, maximum shaft displacements were evaluated by means of a computer program that uses 18 experimentally determined flexibility factors (displacement constants) and iterates through the 4096 loading conditions [9]. A computer printout for the ten inch pump is shown in Figure 9. The input data for the computer printout is shown in the lower portion of the figure. The given (component) forces and moments are extracted from Table 2 of the 610 Standard (Figure 1) for the appropriate suction and discharge nozzle size. The nozzle coordinates relative to the centroid (center of the pump) are used in conjunction with the given forces and moments (appropriate signs must be added) to generate 4096 different summated loading conditions. The six equations used in this transformation are:

$$FXC = FXS + FXD \tag{1}$$

$$FYC = FYS + FYD \tag{2}$$

$$FZC = FZS + FZD \tag{3}$$

$$MXC = MXS + MXD - [(FYS)(ZS) + (FYD)(ZD) - (FZS)(YS) - (FZD)(YD)]/12 \tag{4}$$

$$MYC = MYS + MYD + [(FXS)(ZS) + (FXD)(ZD) - (FZS)(XS) - (FZD)(XD)]/12 \tag{5}$$

$$MZC = MZS + MZD - [(FXS)(YS) + (FXD)(YD) - (FYS)(XS) - (FYD)(XD)]/12 \tag{6}$$

It is important to note that the summated forces and moments and not the component loads are used to determine the shaft displacement. Piping designers concerned with the strength of the piping are typically only concerned with component loads or their resultants. Shaft displacements in the X, Y, Z directions are evaluated by multiplying the summated loads by the appropriate displacement constants and summing. The resultant displacement is found by the square root of the sum of the squares method. These processes are best described in Equations 7, 8, 9 and 10.

$$SDX = (FXC)(FXX) + (FYC)(FYX) + (FZC)(FZX) + (MXC)(MXX) + (MYC)(MYX) + (MZC)(MZX) \tag{7}$$

$$SDY = (FXC)(FXY) + (FYC)(FYY) + (FZC)(FZY) + (MXC)(MXY) + (MYC)(MYY) + (MZC)(MZY) \tag{8}$$

$$SDX = (FXC)(FXZ) + (FYC)(FYZ) + (FZC)(FZZ) + (MXC)(MXZ) + (MYC)(MYZ) + (MZC)(MZZ) \tag{9}$$

$$SDT = [(SCX)^2 + (SDY)^2 + (SDZ)^2]^{1/2} \tag{10}$$

The computer program iterates through the 4096 loading conditions, storing and printing only the loading condition that produces the largest value of SDT. For clarity in describing the calculation process, nomenclature has been changed slightly from that shown on the printout. SDX, SDY and SDZ (not shown on the printout) denote the shaft displacements in the X, Y and Z directions, respectively, while SDT represents the total shaft displacement associated with a particular loading. The maximum value of SDT for any of the 4096 loading conditions is equal to "THE MAXIMUM SHAFT DISPLACEMENT @ COUPLING" (Figure 8).

The displacement constants are the critical parameters in the shaft displacement evaluation. Since these are experimentally determined, they eliminate the many assumptions needed for a purely analytical solution to this type of problem. To ensure credibility, user or contractor personnel were present to witness the experimental determination of the system displacement

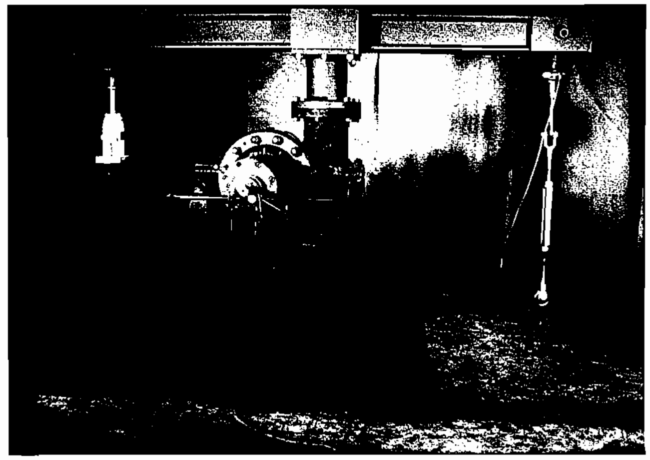


*Figure 10. Typical Test Set-Up for the Applied Moment about the X-Axis.*

*Figure 11. Typical Test Set-Up for the Applied Moment about the Y-Axis.*

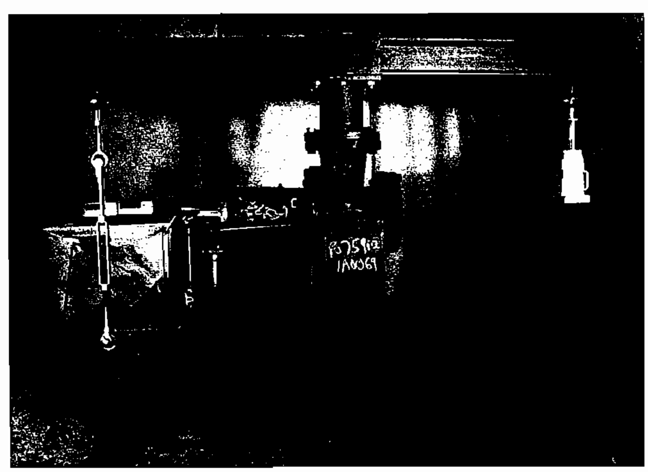


*Figure 12. Typical Test Set-Up for the Applied Moment about the Z-Axis.*

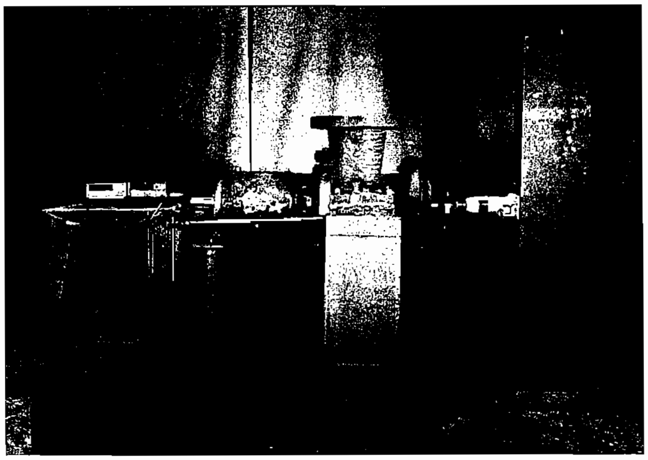


Figure 13. Typical Test Set-Up for the Applied Force in the X-Direction.

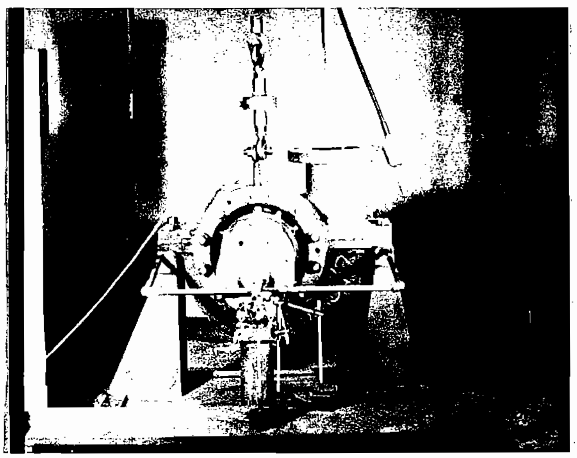


Figure 14. Typical Test Set-Up for the Applied Force in the Y-Direction.

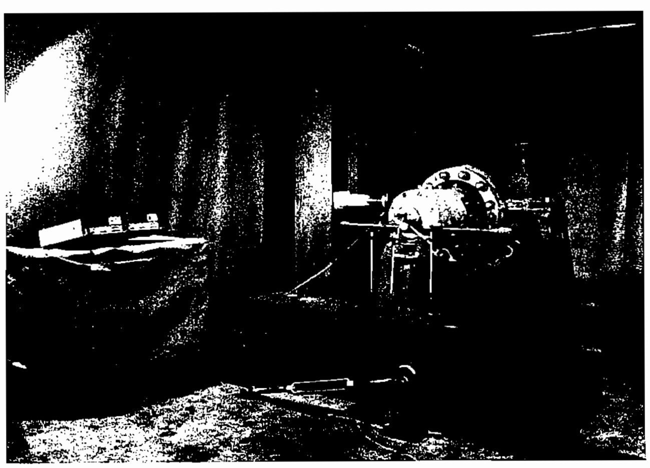


Figure 15. Typical Test Set-Up for the Applied Force in the Z-Direction.

constants. Test rigs similar to the one shown in Figures 10 through 15 were used to apply six loads (FX, FY, FZ, MX, MY and MZ) to each pump casing. As shown, the loads were applied singly, while displacements were measured at the coupling end of the shaft in three orthogonal directions (X, Y, Z) (Figure 16). Plots of shaft

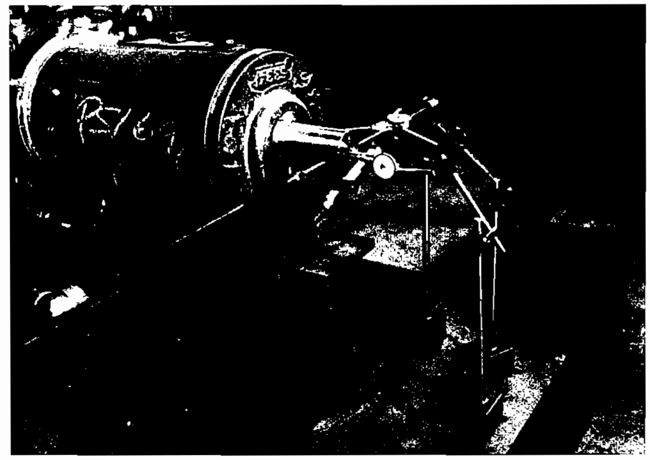


Figure 16. Dial Indicators for Measuring Pump Shaft Displacements in the X, Y, Z Planes.

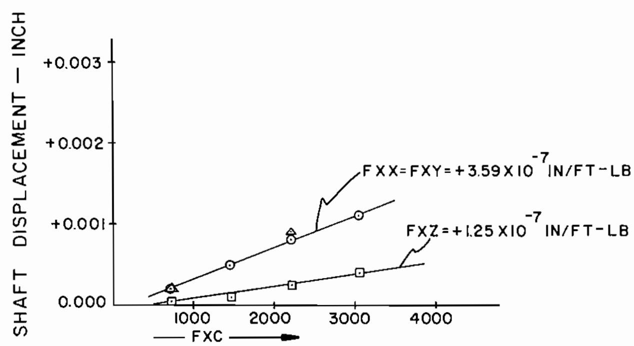



Figure 17. Force Displacement Constants for the Ten Inch Pump.

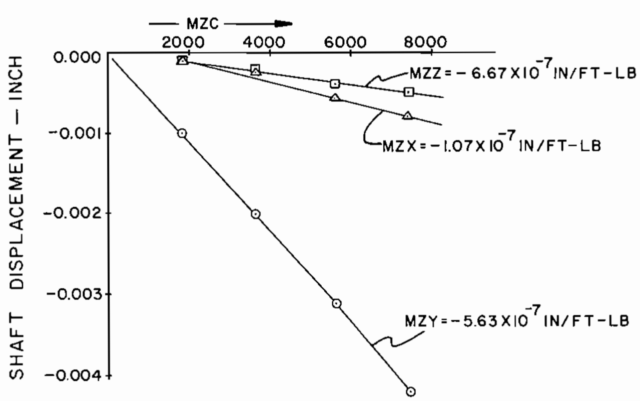



Figure 18. Moment Displacement Constants for the Ten Inch Pump.

displacement vs. applied load (Figures 17 and 18 being typical) provide the experimentally determined flexibility factors. The first two letters of the symbols representing the flexibility factors (displacement constants) define the loading orientation, while the last letter denotes the direction in which the shaft displacement was measured.

As a result of the grouted test data, the 7th Edition Task force has made significant changes to the baseplate stiffness criteria. The revised criteria will result in pump-baseplate assemblies whose maximum shaft displacement due to the application of API 610 Table 2 nozzle loads (6th or 7th Edition) will be 0.010 in

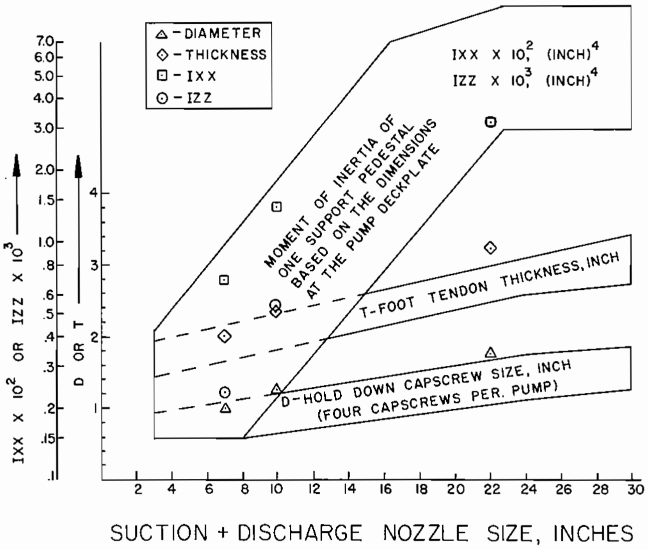
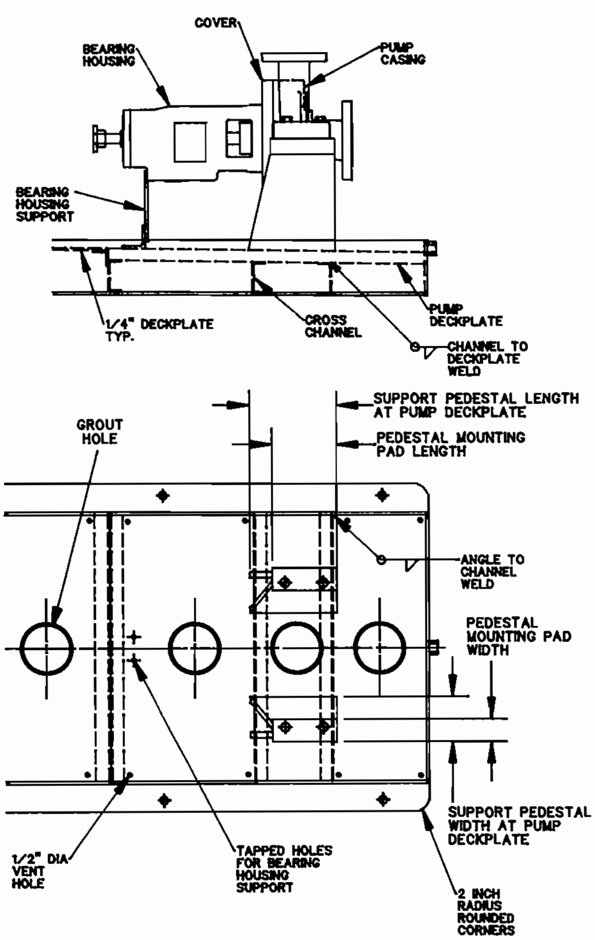
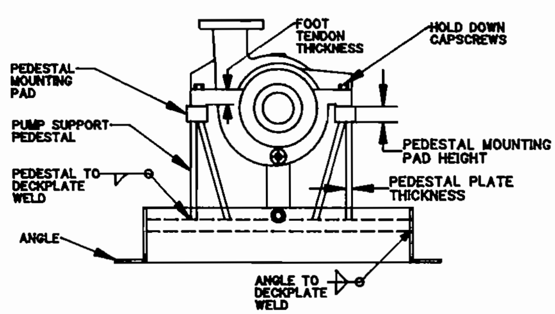



*Figure 19. Pump-Baseplate Assembly Evaluation Chart.*

when grouted in place without a bearing housing support. As previously stated, the pump-baseplate assembly can be no stiffer than its most flexible component. An attempted correlation of three parameters that have a significant effect on the stiffness of pump-baseplate assemblies is represented in Figure 19. Included in Figure 19 are data points for three 6th Edition pump-baseplate assemblies that easily comply with the requirements of the proposed 7th Edition ungrouted baseplate test criteria. It should be noted that these three baseplate assemblies have their pump support pedestals attached to a one or one and one-half inch thick deckplate. Typical construction is shown in Figures 20, 21 and 22. The relatively narrow width of the support pedestals (Figure 20) significantly reduces the moment of inertia about the X-X axis (API 610 coordinate system) as shown in Figure 19 (compare Ixx with Izz). Since the stiffness of the support pedestals is proportional to the moment of inertia, the reduced value of Ixx allows the support pedestals to absorb casing thermal expansion with lower loads on the hold-down capcrews and the support pedestal to pump deckplate attachment welds [4, 10], It is hoped that Figures 19 and 20 can be used to help manufacturers optimize pump-baseplate assemblies and by purchasers and/or users to know when to specify the new optional ungrouted baseplate stiffness test. The following comments must be made before Figure 19 can be used:




| | STRUCTURAL MEMBERS | | | FILLET WELD LEG DIMENSIONS | | | |
|---|---|---|---|---|---|---|---|
| MAXIMUM NOZZLE SIZE (INCH) | ANGLE SIZE & THICKNESS (INCHES) | CROSS CHANNEL SIZE & WT/FT (INCH & LB/FT) | PUMP DECKPLATE THICKNESS (INCH) | PEDESTAL TO DECKPLATE (INCH) | ANGLE TO DECKPLATE (INCH) | CHANNEL TO DECKPLATE (INCH) | ANGLE TO CHANNEL (INCH) |
| 6 | L7X4 – 3/8 | C4 – 5.4 | 1" | 1/4 – 1/2 | 3/8 | 1/4 | 1/4 |
| 8 | L7X4 – 1/2 | C4 – 7.25 | 1" | 1/2 – 3/4 | 1/2 | 1/4 | 1/4 |
| 12 | L8X4 – 1/2 | C4 – 9.0 | 1 1/2" | 3/4 – 1 | 1/2 | 1/4 | 1/4 |

PUMP SUPPORT PEDESTAL DIMENSIONS

| MAXIMUM NOZZLE SIZE (INCH) | PEDESTAL GROUP NUMBER | PLATE THICKNESS (INCHES) | PEDESTAL MOUNTING PAD LENGTH X WIDTH X HEIGHT (INCHES) | BOTTOM AT PUMP DECKPLATE LENGTH X WIDTH (INCHES) |
|---|---|---|---|---|
| 4 | 1 | 1/4 – 1/2 | 8 X 2-5/8 X 2 | 11 X 5-1/4 |
| 4 | 2 | 3/8 – 1/2 | 10 X 3-1/4 X 2 | 12-5/8 X 5-7/8 |
| 6 | 2 | 3/8 – 3/4 | 10 X 3-1/4 X 2 | 12-5/8 X 5-7/8 |
| 8 | 3 | 1/2 – 1 | 13 X 4 X 2 | 17-1/2 X 5-5/8 |
| 10 | 4 | 3/4 – 1-1/8 | 15 X 4 X 2 | 25 X 6-1/8 |
| 14 | 5 | 3/4 – 1-1/4 | 17-3/4 X 4-1/4 X 2 | 23-1/2 X 6-1/8 |

*Figure 20. Typical Baseplate Construction for the Three Pump-Baseplate Assembly Parameters Shown in Figure 19.*

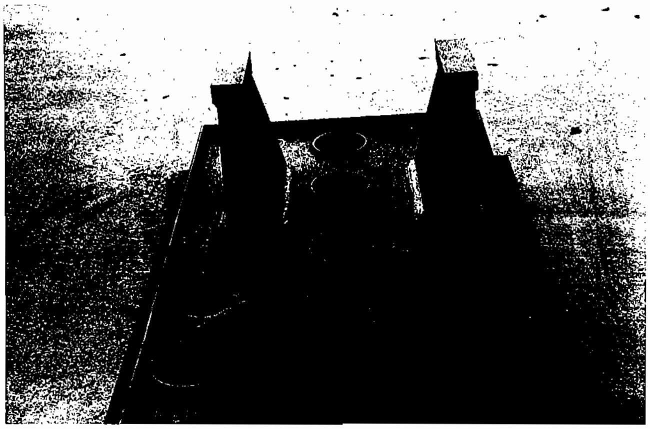


*Figure 21. Pump-Baseplate Assembly Constructed in Accordance with Figure 22.*

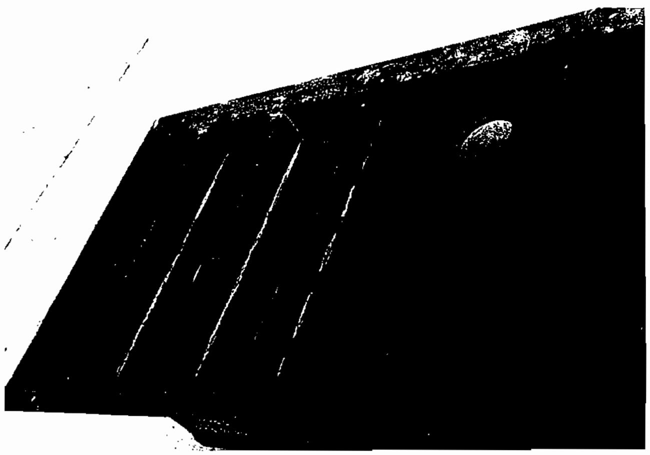


*Figure 22. Underside of a Baseplate Constructed in Accordance with Figure 20.*

• Figure 19 is applicable for radially split one stage pumps that are mounted on two support pedestals, which in turn are welded to a groutable baseplate structure.

• If all of the key parameters are on the high side of their respective tolerance bands, then, the design will easily meet the 7th Edition ungrouted baseplate stiffness criteria.

• If all of the key parameters are near the middle of their respective tolerance bands, the design is probably adequate.

• If all of the key parameters are near the low side of their respective tolerance band, then the design will not meet the 7th Edition ungrouted baseplate stiffness criteria.

## THE EFFECT OF GROUT

Based on field experiences with 5th Edition pumps and baseplates subjected to abnormally high piping loads, one pump company has found that a good grout job was necessary to retain pump and driver shaft alignment. The Murray study supports this statement and also indicates that poor grouting is a common problem associated with pump baseplates [11]. If a baseplate has been properly grouted, there will be no voids between the concrete foundation and the underside of the baseplate. Realizing that the allowable nozzle loadings for the piping designer (Appendix C of the 6th Edition) can be significantly greater than the values recommended by the pump manufacturer (Table 2 of the 6th Edition), and that grouting was sometimes inadequate, it was decided to design 6th Edition baseplates that would not be so sensitive to the influence of grout. Furnishing a one and one-half inch thick pump deckplate appeared to be a simple method of providing an adequate substructure for the pump support pedestals, even if foundation support was lacking. To evaluate the effectiveness of this type of design, the ten inch pump previously discussed was first stiffness tested on its baseplate in the ungrouted condition. It was thought that if there was only a small change in overall shaft displacement, the baseplate could be considered insensitive to the effect of grout. The computer printout for this test is shown in Figure 23. The grout increased the overall stiffness of the pump-baseplate assembly, by a factor of 1.65 (10.9/6.6). A tabulation of the ratio of the ungrouted to grouted displacement constants, indicates the individual changes in stiffness, as shown in Table 4. A ratio greater than 1.0 indicates that the grout increased the stiffness of the baseplate assemzly while a value of 1.0 indicates that the grout had no effect. A ratio of 0.0 indicates that the grout decreased the stiffness of the baseplate assembly or the shaft displacement values were too small to be accurately read with the dial indicators. In hindsight, larger loads should have applied to decrease the error in displacement measurements. Although the ungrouted baseplate assembly performed well, it is believed that the design could be improved by reinforcing the main structural members with a few well placed gussets.

TABLE 4
RATIO OF UNGROUTED TO GROUTED DISPLACEMENT CONSTANTS FOR 10 X 12 X 19 TC PUMP

| | | |
|---|---|---|
| $R_{FXX} = 1.64$ | $R_{FXY} = 2.09$ | $R_{FXZ} = 0.00$ |
| $R_{FYX} = 3.82$ | $R_{FYY} = 1.68$ | $R_{FYZ} = 0.00$ |
| $R_{FZX} = 1.00$ | $R_{FZY} = 1.00$ | $R_{FZZ} = 1.41$ |
| $R_{MXX} = 1.00$ | $R_{MXY} = 2.37$ | $R_{MXZ} = 1.00$ |
| $R_{MYX} = 1.10$ | $R_{MYY} = 1.06$ | $R_{MYZ} = 1.84$ |
| $R_{MZX} = 3.22$ | $R_{MZY} = 1.48$ | $R_{MZZ} = 2.34$ |

```
                   ***JOB IDENTIFICATION: 10 X 12 X 19 TC***

THE MAXIMUM SHAFT DISPLACEMENT @ coupling = 10.9 MILS

THE COMBINATION OF FORCES AND MOMENTS TO PRODUCE MAXIMUM DISPLACEMENT ARE:

       NOZZLE:      FX      FY      FZ      MX      MY      MZ
       SUCTION:   1800.  -1200.   1500.   4500.   3400.  -2200.
     DISCHARGE:   1200.    750.   1000.   3700.   2800.  -1800.

THE SUMMATED FORCES AND MOMENTS @ THE CENTROID ARE:

                    FX      FY      FZ      MX      MY      MZ
                  3000.   -450.   2500.   8684.   6582.  -7217.

===========================================================================

                          ***INPUT DATA***

THE GIVEN FORCES AND MOMENTS ARE:
       NOZZLE:      FX      FY      FZ      MX      MY      MZ
       SUCTION:   1800.   1200.   1500.   4500.   3400.   2200.
     DISCHARGE:   1200.    750.   1000.   3700.   2800.   1800.

THE DISPLACEMENT CONSTANTS USED IN THE CALCULATIONS ARE:

         FXX=  5.88E-07      FXY=  7.50E-07      FXZ=  0.0
         FYX=  1.60E-07      FYY=  5.62E-07      FYZ=  0.0
         FZX=  0.0           FZY=  0.0           FZZ=  1.15E-06
         MXX=  0.0           MXY=  8.88E-08      MXZ=  4.88E-08
         MYX= -7.03E-08      MYY= -5.31E-08      MYZ=  2.13E-07
         MZX= -3.45E-07      MZY= -8.33E-07      MZZ= -1.56E-07

NOZZLE COORDINATES RELATIVE TO THE CENTROID:

           NOZZLE:          X           Y           Z
           SUCTION:      12.125      -0.250        0.0
         DISCHARGE:      -3.875      18.000       15.750
```

*Figure 23. Computer Printout sor the Ten Inch Pump (Ungrouted Baseplate).*

Before jumping to any conclusions concerning the effect grout has on pump-baseplate assembly stiffness, it must be recognized that it is very much a function of the pump casing and baseplate design. For instance, if the pump casing itself is flexible (thin pump feet and small hold down bolting) relative to the baseplate, then the placement of grout will have a small effect. If the pump casing is relatively stiff and the baseplate assembly flexible, grout can make a significant contribution.

## THE PROPOSED 7TH EDITION STANDARD (December 1985 Draft)

Section 2.4 of the proposed API Standard 610 (7th Edition) has been shortened to one paragraph of text. Subsections that deal with casing design are now found in Section 2.2 (Pressure Casings), while the requirements for baseplates are found in Section 3.3 (Mounting Plates). Section 2.4 is shown herein along with the related figures, table and referenced subsections. Table 2 (Figure 24) incorporates the same component nozzle loadings that appear in the 6th Edition Standard (Figure 1). An allowable resultant force and moment for each nozzle size is also provided. These resultant values are used by the piping designer and are calculated from the tabulated component loads. A note has also been added to Table 2 (proposed 7th Edition), clarifying that each of the loads shown represent an allowable range that varies from plus to minus the specified value. Significant changes found in Section 2.4 and subsection 2.2.8 have been *italicized*. Due to the previous remarks concerning the shortcomings of the 6th Edition Standard, these changes are believed to be self explanatory. Subsections 3.3.1.5 and Appendix F are essentially new and therefore comments follow the text.

*Section 2.4—External Nozzle Sorces and Moments (7th Edition)*

*Pumps with nozzles 16* inches and smaller in size and with casings constructed of steel or alloy steel shall be capable of satisfactory operation when subjected to the forces and moments shown in Table 2 (Figure 24 herein). The vendor shall submit nozzle loadings for pump flanges larger than *16* inches and for pump casings constructed of materials other than steel or alloy steel. The *range* of allowable forces and moments given in Table 2 (see Figure 1 through 5 for orientation, i.e., Figure 25 herein) are to be used by the vendor (see section 2.2.8 and 3.3.1.5) for pump and baseplate design and by the piping designer (see Appendix M) to establish acceptable piping configuration. Two effects of nozzle loading are considered: *distortion* of the pump casing (see 2.2.8) and misalignment of the pump and driver shafts (see 3.3.1.5). Vertical and vertical inline pumps driven by integrally mounted electric motors whose shaft alignment is not affected by external nozzle loads may be subjected to forces and moments that are double the values shown in Table 2 [5].

NOTE: Each value shown below indicates a range from minus that value to plus that value; for example, 160 indicates a range from -160 to +160.

| | Nominal Size of Nozzle Flange (inches) | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| Force/Moment[A] | 2 | 3 | 4 | 6 | 8 | 10 | 12 | 14 | 16 |
| Each top nozzle | | | | | | | | | |
| FX | 160 | 240 | 320 | 560 | 850 | 1200 | 1500 | 1600 | 1900 |
| FY | 200 | 300 | 400 | 700 | 1100 | 1500 | 1800 | 2000 | 2300 |
| FZ | 130 | 200 | 260 | 460 | 700 | 1000 | 1200 | 1300 | 1500 |
| FR | 290 | 430 | 570 | 1010 | 1560 | 2200 | 2600 | 2900 | 3300 |
| Each side nozzle | | | | | | | | | |
| FX | 160 | 240 | 320 | 560 | 850 | 1200 | 1500 | 1600 | 1900 |
| FY | 130 | 200 | 260 | 460 | 700 | 1000 | 1200 | 1300 | 1500 |
| FZ | 200 | 300 | 400 | 700 | 1100 | 1500 | 1800 | 2000 | 2300 |
| FR | 290 | 430 | 570 | 1010 | 1560 | 2200 | 2600 | 2900 | 3300 |
| Each end nozzle | | | | | | | | | |
| FX | 200 | 300 | 400 | 700 | 1100 | 1500 | 1800 | 2000 | 2300 |
| FY | 130 | 200 | 260 | 460 | 700 | 1000 | 1200 | 1300 | 1500 |
| FZ | 160 | 240 | 320 | 560 | 850 | 1200 | 1500 | 1600 | 1900 |
| FR | 290 | 430 | 570 | 1010 | 1560 | 2200 | 2600 | 2900 | 3300 |
| Each nozzle | | | | | | | | | |
| MX | 340 | 700 | 980 | 1700 | 2600 | 3700 | 4500 | 4700 | 5400 |
| MY | 260 | 530 | 740 | 1300 | 1900 | 2800 | 3400 | 3500 | 4000 |
| MZ | 170 | 350 | 500 | 870 | 1300 | 1800 | 2200 | 2300 | 2700 |
| MR | 460 | 950 | 1330 | 2310 | 3500 | 5000 | 6100 | 6300 | 7200 |

[A]F = force, in pounds; M = moment, in foot-pounds; R = resultant. See Figures 1 thru 5 for orientation of nozzle loads (X, Y, & Z).

*Figure 24. Table 2—Nozzle Loadings* [5].

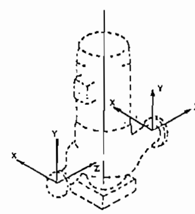



Figure 1
Coordinate system for the Forces and Moments in Table 2
VERTICAL INLINE PUMPS

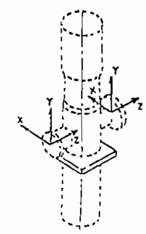



Figure 2
Coordinate system for the Forces and Moments in Table 2
VERTICAL DOUBLE CASING PUMPS

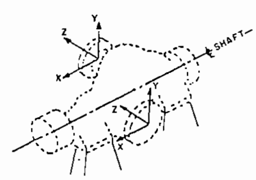



Figure 3
Coordinate system for the Forces and Moments in Table 2
HORIZONTAL PUMPS
End suction-Top Discharge Nozzles

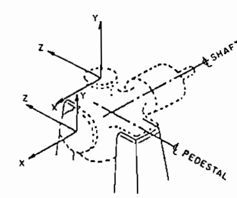



Figure 4
Coordinate system for the Forces and Moments in Table 2
HORIZONTAL PUMPS
End suction-Top Discharge Nozzles

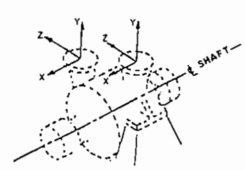



Figure 5
Coordinate system for the Forces and Moments in Table 2
HORIZONTAL PUMPS
Top Nozzles

*Figure 25. Figures 1-5* [5].

*Section 2.2—Pressure Casings 7th Edition*

*Subsection 2.2.8*

2.2.8 The pump pressure casing shall be capable of withstanding double the forces and moments in Table 2, applied simultaneously to the pump through each nozzle, plus internal pressure, without *distortion which would impair operation of the pump or seal* [5].

| | Pump Shaft Displacement | | |
|---|---|---|---|
| Loading Condition | Inches | Micrometers | Direction |
| $\underline{MC}_C$ | 0.003 | 76 | $\underline{Z}$ |
| $\underline{MZ}_C$ | 0.007 | 178 | $\underline{Y}$ |

NOTE: $\underline{MY}_C$ and $\underline{MZ}_C$ equal the sum of the allowable suction and discharge nozzle moments from Table 2.

$$\underline{MY}_C = (\underline{MY})_{SUCTION} + (\underline{MY})_{DISCHARGE}$$
$$\underline{MZ}_C = (\underline{MZ})_{SUCTION} + (\underline{MZ})_{DISCHARGE}$$

*Figure 26. Table 10—Stiffness Criteria for Ungrouted Baseplates.*

*Subsection 3.3.1—Baseplates for Horizontal Pumps*

*Subsection 3.3.1.5*
3.3.1.5 To minimize misalignment of pump and driver shafts due to the piping load effects, baseplates shall be constructed with sufficient structural stiffness to limit pump shaft displacement at the drive end of the shaft or at the register fit of the coupling hub to the values shown in Table 10 (Figure 26, herein). Grout or bearing housing supports (wobble plates) shall not be used as a means of obtaining the required stiffness. (It is recognized that grout can significantly increase the stiffness of the baseplate assembly; by neglecting this effect, baseplate construction can easily be verified by testing at the vendor's shop (see 3.3.1.6) [5].
*Subsection 3.3.1.6*
3.3.1.6 If specified, the vendor shall test to demonstrate that the pump and its baseplate assembly, when anchored at foundation bolt hole locations with any bearing housing support disconnected, are in compliance with the requirements of 3.3.1.5. The pump casing shall be subjected to moments $MY_C$ and $MZ_C$, so that the corresponding Z and Y shaft displacements can be measured and recorded. The shaft displacement measurements shall be absolute (not relative to the baseplate). For record purposes, the vendor's test data shall include a schematic drawing of the test setup, the calculated moment loads ($MY_C$ and $MZ_C$), the applied moment loads, and their corresponding displacements at the drive end of the pump shaft [5].

As can be seen, Subsections 3.3.1.5 and 3.3.1.6 represent significant and needed departures from the pump support assembly stiffness criteria found in the 6th Edition. This is due to the fact that the design and compliance test criteria do not take credit for grout, even though it will be placed when the baseplates are installed at the jobsite. With this departure, the criteria become economically enforceable. A simple test rig that can be used to measure pump shaft displacement as required by Subsections 3.3.1.5. and 3.3.1.6 is illustrated in Figures 27 and 28. The weight hanging from the suction nozzle piping creates the applied moment (force X distance). Notice that the shaft displacements are being measured with respect to the floor and not with respect to the baseplate structure. Ten 3/4-in concrete anchors have been used to hold the baseplate (API Standard Number 8) to the shop floor. Plotted test data for this ungrouted

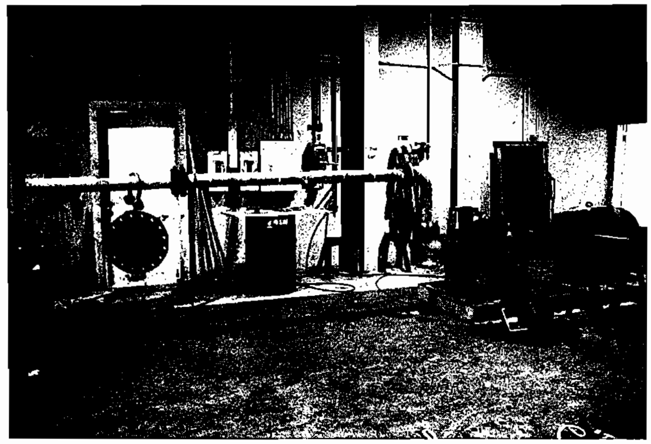


*Figure 27. MZC Loading Technique for the Ungrouted Baseplate Compliance Test.*

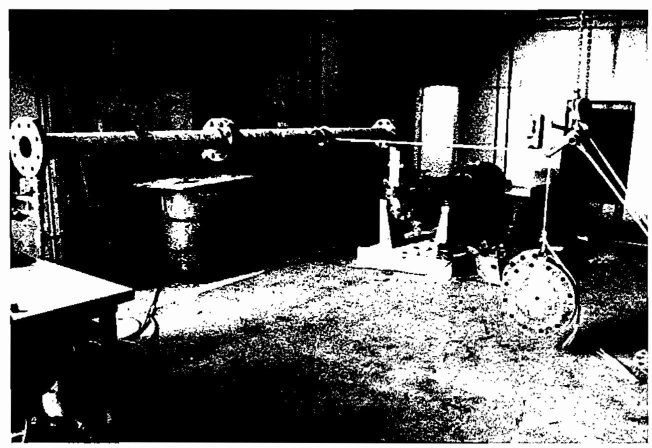


*Figure 28. MYC Loading Technique for the Ungrouted Baseplate Compliance Test.*

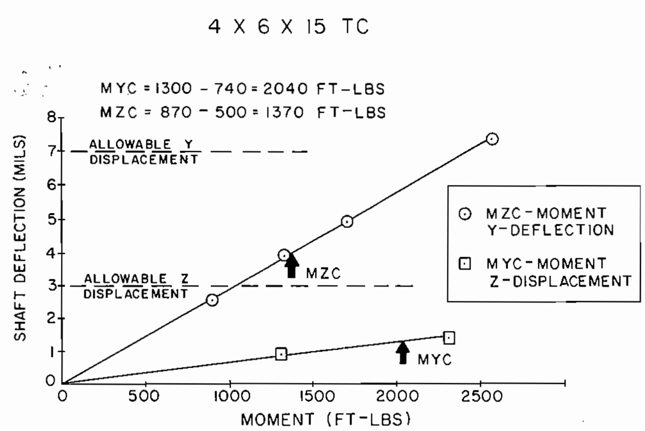



*Figure 29. Plotted Data for an Ungrouted Baseplate Compliance Test.*

compliance test are shown in Figure 29. Approximately four to twelve hours time is needed to set-up, conduct and document this type of a test. The more sophisticated apparatus shown in Figures 11-12 could also be used to apply loads to pump casing. Dial indicators or proximity probes can be used to measure shaft displacement.

The design and acceptance criteria for 7th Edition pump-baseplate assemblies are based on two (MYZ and MZY) of the eighteen previously discussed displacement constants (Figure 26). The first criterion (MY loading and Z displacement) provides a measure of the torsional rigidity of the pump pedestal support assembly while the second criterion (MZ loading and Y displacement) reflects the bending stiffness of the support pedestals and the torsional stiffness of the pump feet. It is interesting to see how these criteria vary with pump size (Figure 30). Stiffness constants (KYZ and KZY) which are the reciprocals of the displacement constants (MYZ and MZY) increase the pump size. As indicated, a 12x14 pump denoted by a value of 30 on the horizontal axis must be nine times stiffer than a 2x3 (abscissa value of five) due to the variation in allowable piping loads (Table 2). Also indicated on Figure 30, are test points for four 6th Edition pumps. These test points demonstrate that the new requirements are indeed realistic and achievable.

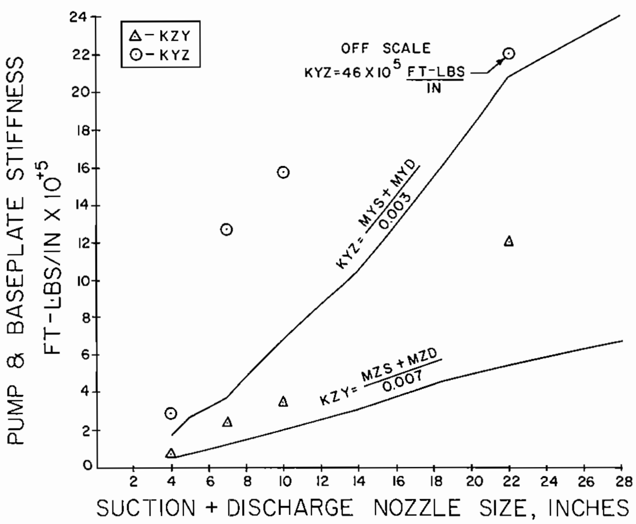



*Figure 30. Ungrouted Pump-Baseplate Stiffness Criteria [5, (Table 10)].*

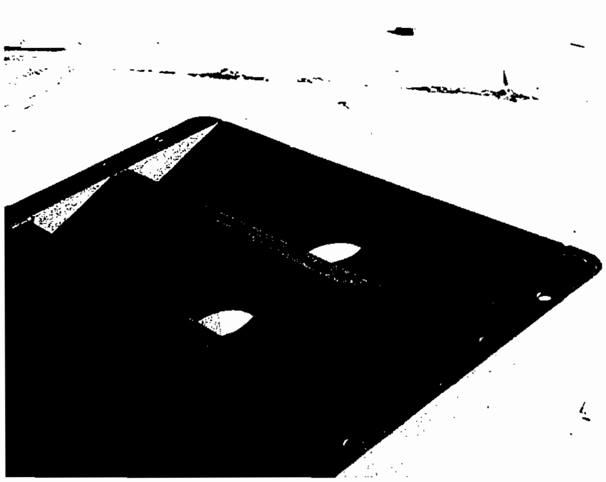


*Figure 31. Drip Pan Type Baseplate with Two Inch Rounded Corners.*

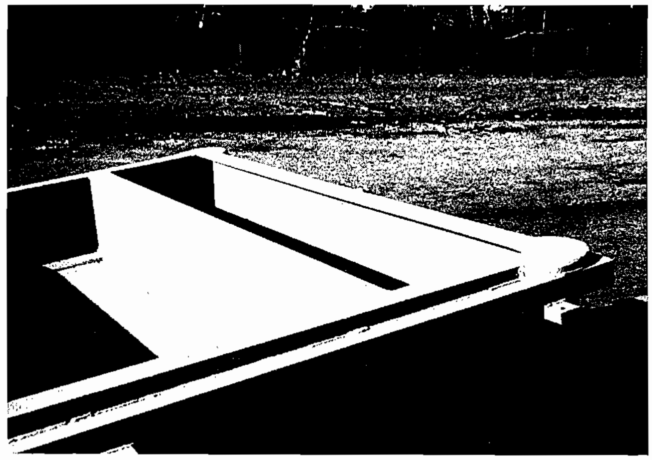


*Figure 32. Drip Rim Type Baseplate with Two Inch Rounded Corners.*

*Subsection 3.3.1.9*

3.3.1.9 The outside corners of the baseplate in contact with the grout shall have at least 2-inch (50-millimeter) radii (in the plan view) [5].

This new requirement has been added to reduce the stress concentration associated with square corners and thereby prevent grout crackage. Baseplates with rounded corners are illustrated in Figures 31 and 32.

*Appendix F—Criteria For Piping Design*

APPENDIX F (called APPENDIX B herein) consists of three main sections: 1) F.1—Horizontal Pumps with Grouted Baseplates, 2) F.2—Vertical Inline Pumps, 3) F.3—Sample Problems (included to demonstrate the intent and use of the equations contained within the Appendix). Section F.2 and F.3 will not be presented nor discussed herein. Section F.1 was rewritten, taking into account the relaxation of the pump-baseplate stiffness criteria. The stiffness requirements (shown in Figure 26) result in pump equipment that is approximately 50 percent of what was specified for 6th Edition standard baseplates. To clarify to the piping designer and the user the effects of piping loads greater than Table 2 values, upper bounds for anticipated shaft displacement have been provided. Subsection F.1.2 is very similar to the piping load criteria found in Appendix C for standard baseplates. Subsections F.1.2.1 and F.1.2.2 are equivalent to the old Appendix C criteria for nozzle sizes eight inches and smaller. Above eight inches, Subsection F.1.2.1 permits larger resultant piping loads on the individual nozzles. Subsection F.1.2.2 controls stress and strain in the pump casing. Subsection F.1.2.3 is new and limits the magnitude of load that can be imposed on the pump itself. This criterion controls the magnitude of the shaft displacement due to flexibility in the pump-baseplate assembly. Without Subsection F.1.2.3, pump shaft displacements in the 0.020 in range would be possible. It should be understood that the shaft displacement values cited in Appendix F assume that a bracket support is not present. With bracket supports, shaft displacements would be substantially less.

## CONCLUSIONS

API 610 6th Edition baseplate and nozzle loading criteria have been examined and found to be vague and subject to interpretation. The grouted baseplate test program developed by the API 610 7th Edition Task Force demonstrated that the standard and heavy-duty baseplate stiffness criteria found in the 6th Edition of Standard 610 were not being achieved and that there was a need for more realistic criteria that could be enforced with a simple, inexpensive compliance test. The proposed 7th Edition of API Standard 610 (December 1985 Draft) addresses this problem and the other major shortcomings found in the 6th Edition.

## APPENDICES

### *APPENDIX A*

APPENDIX 1 – TABLE 1 : EFFECT OF EXTERNAL FORCES AND MOMENTS MEASURED AS DISPLACEMENTS AT PUMP COUPLING

SUBJECT: THE ABILITY OF PUMP MANUFACTURERS TO MEET THE NEW REQUIREMENTS FOR EXTERNAL NOZZLE FORCES AND MOMENTS, CALLED FOR IN SECTION 2.4 OF API 610. SPECIFICALLY ASKED IF ALL OF THEIR PUMPS CAN MEET THE CRITERIA OF PARAGRAPHS 2.4.3 AND 2.4.6 AS SHOWN BELOW

| QUESTIONS | AMERICAN PUMP MANUFACTURERS (ANSWERS) | | | | | |
|---|---|---|---|---|---|---|
| | COMPANY A | COMPANY B | COMPANY C | COMPANY D | COMPANY E | COMPANY F |
| PARAGRAPH 2.4.3 A MAXIMUM DEFLECTION OF 0.005 INCH AT THE COUPLING WITH THE SIMULTANEOUS APPLICATION OF THE LOADS SHOWN IN TABLE 2, (WITH VENDORS STANDARD BASEPLATE) | MODELS A, B, C, ETC WILL ACCOMODATE LOADS IN GROUTED CONDITION. OTHER MODELS X, Y, & Z REQUIRE SPECIAL LAGGING & BASE CONFIGURATION | STD. BASEPLATE COMPLIES WITH THE REQUIRE-MENTS FOR A MAX. DEFLECTION OF 0.005 INCH AT THE COUPLING | API 610 PUMPS CAN MEET CRITERIA SET OUT IN PARAGRAPH 2.4.3 | ALL PROCESS PUMPS ARE DESIGNED TO MEET API 610, 6TH EDITION AND PAR. 2.4.3 | PUMP WERE REDESIGNED AND STRENGTHENED TO MEET PARAGRAPH 2.4.3 | ALL OF THEIR API PUMPS WILL COMPLY TO PAR. 2.4.3 |
| PARAGRAPH 2.4.6 A MAXIMUM DEFLECTION OF 0.005 INCH AT THE COUPLING WITH THE SIMULTANEOUS APPLICATION OF TWO TIMES THE LOADS SHOWN IN TABLE 2, (PAGE 6, API 610), (WITH VENDORS HEAVY DUTY BASEPLATE) | MODELS A, B, C, ETC WITH HEAVY DUTY BASEPLATE WHEN GROUTED WILL TAKE TWICE THE LOADS SHOWN IN TABLE 2.–OTHER MODELS AS SHOWN ABOVE REQUIRE SIMILAR MODIFICATION. | HEAVY DUTY BASEPLATE COMPLIES WITH THE REQUIREMENT OF PARAGRAPH 2.4.6 | API 610 PUMPS CAN MEET CRITERIA SET OUT IN PARAGRAPH 2.4.6 | EXTRA HEAVY DUTY BASEPLATE WITH ADDITIONAL COST CAN MEET THE REQUIREMENT OF PARAGRAPH 2.4.6 | WITH 200# LOAD & EXTRA HEAVY DUTY BASEPLATE, THEIR CALC. OF SHAFT DEFL. MEET PAR. 2.4.6 & IN SOME CASES MIGHT BE LESS. RESULTS WILL BE AVAIL. ONLY AFTER PURCH. OF PUMP | ALL OF THEIR API PUMPS WILL COMPLY TO PAR. 2.4.6 & WILL ACCEPT DOUBLE THE API LOADS WITH A MAXIMUM OF 0.005 INCH DEFLECTION AT THE COUPLING |
| PARAGRAPH 2.4.6 WHICH OF YOUR PUMPS COULD MEET THE CRITERIA OF PARAGRAPH 2.4.6, IF THE MAXIMUM ALLOWABLE DEFLECTION WERE 0.002 INCH THIS WOULD BE WITH TWICE THE LOADS SHOWN IN TABLE 2. | ALL PUMPS REQUIRE SPECIAL TREATMENT SUCH AS CASE LAGGING, EXTRA HEAVY BASE OR USE OF GUIDE KEYS. EACH PUMP NEED TO BE EVALUATED ON AN INDIVIDUAL BASIS | NONE OF THEIR STANDARD AND HEAVY DUTY BASEPLATES, CURRENTLY DESIGNED COMPLY WITH THIS REQUIREMENT | NONE OF THEIR PUMPS WILL MEET THE CRITERIA OF PAR. 2.4.6 IF THE MAXIMUM ALLOWABLE DEFL. WERE 0.002 INCH. | COULD ONLY MEET FOR LARGER PUMP SIZES AFTER SPE-CIAL ENGR. OF CASE PATTERNS, BASEPLATE & MA-CHINING, ALSO PUMP & DRIVER NEED HOT ALIGN. IN THE FIELD | NO RESPONSE TO THIS QUESTION | THEY CAN NOT MEET THE MAX. ALLOWABLE DEFLECTION OF 0.002 INCH. |

APPENDIX 1 – TABLE 1 : EFFECT OF EXTERNAL FORCES AND MOMENTS MEASURED AS DISPLACEMENTS AT PUMP COUPLING

SUBJECT: THE ABILITY OF PUMP MANUFACTURERS TO MEET THE NEW REQUIREMENTS FOR EXTERNAL NOZZLE FORCES AND MOMENTS, CALLED FOR IN SECTION 2.4 OF API 610. SPECIFICALLY ASKED IF ALL OF THEIR PUMPS CAN MEET THE CRITERIA OF PARAGRAPHS 2.4.3 AND 2.4.6 AS SHOWN BELOW

| QUESTIONS | EUROPEAN AND JAPANESE PUMP MANUFACTURERS (ANSWERS) | | | |
|---|---|---|---|---|
| | COMPANY G | COMPANY H | COMPANY I | COMPANY J |
| PARAGRAPH 2.4.3 A MAXIMUM DEFLECTION OF 0.005 INCH AT THE COUPLING WITH THE SIMULTANEOUS APPLICATION OF THE LOADS SHOWN IN TABLE 2, (WITH VENDORS STANDARD BASEPLATE) | ALL API 610 PUMPS WITH CAST CARBON STEEL CASING AND STD HEAVY DUTY BASEPLATES WILL MEET THE CRITERIA OF PAR. 2.4.3 AND PAR. 2.4.6 | HEAVY DUTY RANGE OF PROCESS PUMPS COMPLY WITH PAR. 2.4.3 WITH STD. BASEPLATES | ALL THE API PUMPS THEY PRODUCE WILL MEET THE DEFLECTION REQUIREMENTS OF PAR. 2.4.3 | YES ALL OF THEIR API 610 PUMPS CAN MEET THIS CRITERIA. |
| PARAGRAPH 2.4.6 A MAXIMUM DEFLECTION OF 0.005 INCH AT THE COUPLING WITH THE SIMULTANEOUS APPLICATION OF TWO TIMES THE LOADS SHOWN IN TABLE 2, (PAGE 6, API 610), (WITH VENDORS HEAVY DUTY BASEPLATE) | SEE ABOVE REMARK: ALL CS PROCESS PUMPS CAN TAKE UNLIMITED F&M WITH A MAX. DEFL. OF 0.005 INCH IF THE BASEPLATES ARE MOUNTED ON SPECIALLY DESIGNED VIBRATION DAMPERS. | HEAVY DUTY PUMPS CAN MEET THE STIFFNESS REQUIREMENT OF PAR. 2.4.6 WHEN USING A SPECIAL HEAVY DUTY BASEPLATE DESIGN | ALL THE API PUMPS THEY PRODUCE WILL MEET THE ALLOW-ABLE DEFL. OF 0.005 INCH WITH DOUBLE THE LOADS SHOWN IN TABLE 2 WITH THEIR STD PEDESTALS & BASEPLATES | YES ALL OF THEIR API 610 PUMPS EXCEPT OVERHUNG TYPES CAN MEET THIS CRITERIA. OVERHUNG TYPES ARE UNABLE TO MEET THIS CRITERIA BECAUSE OF THE CASING THICKNESS. |
| PARAGRAPH 2.4.6 WHICH OF YOUR PUMPS COULD MEET THE CRITERIA OF PARAGRAPH 2.4.6, IF THE MAXIMUM ALLOWABLE DEFLECTION WERE 0.002 INCH THIS WOULD BE WITH TWICE THE LOADS SHOWN IN TABLE 2. | HIGH PRESS. PUMPS WITH OVERHUNG IMP. & WITH IMP BETWEEN BRG. DESIGN & WITH PUMP CASING MADE UP OF CAST CS CAN MEET AN ALLOWABLE MAX. DEFLECTION OF 0.002 INCH | DEFLECTION OF SHAFT CAN NOT BE MET WITH STD HEAVY DUTY PUMP RANGE (EXCEPT MODEL D) WITHOUT ADJUSTMENT | FOR ALL API PUMPS IT IS ENTIRELY POSSIBLE TO KEEP SHAFT DISPLACEMENT WITHIN 0.002 INCH WITH TWICE THE LOADS SHOWN IN TABLE 2 | ALL OF THEIR API 610 PUMPS EXCEPT OVERHUNG TYPES CAN MEET THIS CRITERIA WITH SOME ADDITIONAL COST. |

### *APPENDIX B*

*Appendix F*
*Criteria For Piping Design*

Section 4.1—Horizontal Pumps with Grouted Baseplates
NOTE: These criteria do not apply to pumping equipment purchased in accordance with 1.1.2 of this standard.

*F.1 Horizontal Pumps with Grouted Baseplates*

F.1.1 Acceptable piping configurations shall not cause excessive distortion of the pump casing and shall not cause excessive misalignment between the ends of the pump and driver shafts. Piping configurations that produce component nozzle loads that lie within the ranges specified in Table 2 will limit casing distortion to one-half the vendor's design criterion (see 2.2.8) and will ensure pump shaft displacements of less than 0.010 inch.

F.1.2 Piping configurations that produce loads outside the ranges specified in Table 2 are also acceptable without consultation with the pump vendor, provided the conditions specified in F.1.2.1 through F.1.2.3 are satisfied. Satisfying these conditions will ensure that any pump casing distoriton will be within the vendor's design criteria (see 2.2.8) and that the displacement of the pump shaft will be less than 0.015 inch.

F.1.2.1 The individual component forces and moments acting on each pump nozzle flange shall not exceed thae range specified in Table 2 by a factor of more than 2.

F.1.2.2 The resultant applied force ($FRS_A$, $FRD_A$) and the resultant applied moment ($MRS_A$, $MRD_A$) acting on each pump nozzle flange shall satisfy the appropriate interaction equation (Equations F-1 and F-2).

$$(FRS_A/1.5FRS_{T2}) + (MRS_A/1.5MRS_{T2}) \leq 2 \qquad \text{(F-1)}$$

$$(FRD_A/1.5FRD_{T2}) + (MRD_A/1.5MRD_{T2}) \leq 2 \qquad \text{(F-2)}$$

F.1.2.3 The applied component forces and moments acting on each pump nozzle flange must be translated and resolved to the center of the pump. The magnitude of the resultant applied force ($FRC_A$), the resultant applied moment ($MRC_A$), and the applied $Z$ moment ($MZC_A$) shall be limited by Equations F-3, F-4, and F-5. (The sign convention shown in Figures 3-5 and the right-hand rule should be used in evaluating these equations.)

$$FRC_A < 1.5(FRS_{T2} + FRD_{T2}) \qquad \text{(F-3)}$$

$$MZC_A < 2.0(MZS_{T2} + MZD_{T2}) \qquad \text{(F-4)}$$

$$MRC_A < 1.5(MRS_{T2} + MRD_{T2}) \qquad \text{(F-5)}$$

where

$$FRC_A = [(FXC_A)^2 + (FYC_A)^2 + (FZC_A)^2]^{1/2}$$
$$FXC_A = FXS_A + FXD_A$$
$$FYC_A = FYS_A + FYD_A$$
$$FZC_A = FZS_A + FZD_A$$
$$MRC_A = [(MXC_A)^2 + (MYC_A)^2 + (MZC_A)^2]^{1/2}$$
$$MXC_A = MXS_A + MXD_A - [(FYS_A)(zS) + (FYD_A)(zD) - (FZS_A)(yS) - (FZD_A)(yD)]/12$$
$$MYC_A = MYS_A + MYD_A + [(FXS_A)(zS) + (FXD_A)(zD) - (FZS_A)(xS) - (FZD_A)(xD)]/12$$
$$MZC_A = MZS_A + MZD_A - [(FXS_A)(yS) + (FXD_A)(yD) - (FYS_A)(xS) - (FYD_A)(xD)]/12$$

F.1.3 Piping configurations that produce loads greater than those allowed in F.a.a or F.1.2 shall be mutually approved by the purchaser or the vendor.

*Nomenclature*

- $F$ force, in pounds.
- $FR$ resultant force. ($FRS_A$ and $FRD_A$ are calculated by the square-root-of-the-squares method using the applied component forces acting on the nozzle flange. $FRS_{T2}$ and $FRD_{T2}$ are extracted from Table 2 using the appropriate nozzle size.)
- $M$ moment, in foot-pounds.
- $MR$ resultant moment. ($MRS_A$ and $MRD_A$ are calculated by the square-root-of-the-squares method using the applied component moments acting on the nozzle flange. $MRS_{T2}$ and $MRD_{T2}$ are extracted from Table 2 using the appropriate nozzle size.)

C center of the pump, as defined by the intersection of the pump shaft centerline and the support pedestal centerline (see Figures 3-5).
S suction nozzle.
D discharge nozzle.
Subscript A applied loads.
Subscript T2 loads extracted from Table 2.
$x, y, z$ location coordinates of the nozzle flanges with respect to the center of the pump.
$X, Y, Z$ direction of the loads (see Figures 3-5).

APPENDIX C

Baseplate Checklist [5]

1. Will the baseplate assembly be lifted with all equipment mounted?
   a. Are the structural members properly sized?
   b. Are attachment welds adequate?
   c. Are lifting lugs provided and properly located?
   d. Are spreader bars required for lifting?
   e. Will any piping that runs the length of the baseplate be damaged during lifting?
2. Will the baseplate be grouted in place?
   a. Will the footprint of the baseplate conform to the standard dimensions found in Appendix H?
   b. Are grout holes of adequate size and properly located?
   c. Are 1/2 inch vent holes properly located?
   d. Are cross members properly located, attached and shaped to lock positively into the grout?
   e. Have vertical leveling screws been specified? Are there they properly located and of adequate size?
   f. Are the outside corners of the baseplate rounded to prevent grout cracking due to stress concentration?
   g. Will epoxy grout be used? Is a suitable primer being applied to all groutable surfaces of the baseplate?
3. Should the optional stiffness test be specified?
   a. What is the severity of the service?
   b. What are the construction features of the pump-baseplate assembly?
      1. Thickness of tendon connecting pump casing to foot pad?
      2. Size and number of pump hold-down capscrews?
      3. Size, shape, height and thickness of support pedestals?
      4. How and where are the support pedestals attached to the baseplate?
   c. What are the construction features of installed pump-baseplate assemblies in similar service? Do they have good maintenance records?
   d. Does the vendor have witnessed stiffness test data for similar equipment?
   e. What will the vendor charge to perform the optional stiffness test?
   f. What is the operating speed and the length of the coupling spacer?
   g. What is the magnitude of the calculated piping loads?
   h. Are the calculated piping loads realistic?
   i. Is the pump handling hot product? Will there be operating conditions that cause temperature swings?
   j. Will the equipment be realigned when it is hot?
4. Have alignment positioning screws been provided?
5. Does the baseplate drain rim or pan have adequate slope?
6. Does the baseplate extend under the pump and drive elements to contain leakage?

## REFERENCES


1. American Petroleum Institute, "Centrifugal Pumps for General Refinery Services," *API Standard 610*, Six Edition (1981).
2. American Petroleum Institute, "Centrifugal Pumps for General Refinery Services," *API Standard 610*, Fifth Edition (1971).
3. Payne, J.R., "PVRC Pump-Piping Interaction Experience Survey," Pressure Vessel Research Committee-Welding Research Council (February 1985).
4. Steiger, J.E., "Horizontal Process Pump Modifications to Comply with API 610 Six Edition Forces and Moments," Proceedings of the First International Pump Symposium, Texas A&M University, College Station, Texas, pp. 47-55 (May 1984).
5. American Petroleum Institute, "Centrifugal Pumps for General Refinery Services," *Proposed* API Standard 610, Seventh Edition (December 1985 Draft).
6. "Boiler and Pressure Vessel Code," Section VIII—Pressure Vessels Division 1, p. 187 (1983).
7. James, W.H., "Try Spring-Mounted Pumps," Hydrocarbon Processing, pp. 247-250 (September 1978).
8. Horrell, W.R., "Free Baseplates Reduce Cost and Increase Reliability," Hydrocarbon Processing, pp. 177-180 (April 1981).
9. Jones, W.B. and Hayrapetian, V., "API Nozzle Loading Data Reduction Program," Pacific Pumps Division, Dresser Industries (September 12, 1984).
10. Bussemaker, E.J., "Design Aspects of Baseplates for Oil and Petrochemical Industry Pumps," The Institution of Mechanical Engineers C45/81, pp. 135-641 (1981).
11. Murray, Jr., M.G., "Better Pump Grouting," Hydrocarbon Processing, pp. 100-104 (February 1974).