

# HORIZONTAL PROCESS PUMP MODIFICATIONS TO COMPLY WITH API-610 SIXTH EDITION FORCES AND MOMENTS

by

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## ABSTRACT

The forces and moments stiffness criteria established by API-610, Sixth Edition has resulted in a new generation of horizontal process pumps and baseplates, each with increased structural rigidity. Pictures and text explaining the structural modifications are presented along with equations for establishing the size of the integral pump feet and the hold-down capscrews. Supporting experimental stiffness test data for three sizes of Sixth Edition overhung process pump casings are illustrated graphically. A computer generated pedestal selection chart is presented as a means of optimizing the design of the pump support pedestal assembly, which must accommodate casing thermal expansion and Sixth Edition stiffness requirements.

## INTRODUCTION

The revised forces and moments criteria established by API-610, Sixth Edition [1] has had a significant impact on the structural design of process pump cases and baseplates. The purpose of this paper is to acquaint the reader with the modifications necessary to comply with the Sixth Edition. The analytical procedures used to predict the design changes are presented along with supporting experimental test data.

According to API-610, there are two criteria to be met with the specified nozzle loadings (Table 2 of API-610). Paragraph 2.4.2 requires the pump to be capable of withstanding double the loads in Table 2, without adversely affecting normal pump operation due to internal rubbing or mechanical seal problems. Paragraph 2.4.3 requires that the pump and baseplate assembly have adequate structural stiffness to limit shaft displacement at the coupling hub to 0.005 inch in any direction when subjected to the loads in Table 2.

Past field experience and conservative hand calculations have indicated that casing modifications made to meet the requirements of paragraph 2.4.3 would result in designs that could withstand double the loads in Table 2 with no internal rubbing or adverse operation of the pump or mechanical seal. Realizing that all subcomponents of the pump assembly are flexible and to provide a systematic analysis procedure, we elected to divide the permissible 0.005 inch shaft displacement (2.4.3) among three subcomponents:

1. The pump casing—0.002 inch.
2. The pump to support pedestal capscrews—0.0015 inch.
3. The support pedestal assembly—0.0015 inch.

By dividing the 0.005 inch maximum displacement in this way, the pump casing and capscrews can be analyzed and modified once, while the baseplate and support pedestal assembly is customized for each particular installation.

## DESIGN MODIFICATIONS

To achieve this "in-house" stiffness criteria, significant changes have been made to the size of the pump to support pedestal attachment welds. The thickness of the pump feet, the baseplate structural members and the deckplate under the support pedestal have been increased substantially. A picture of a Sixth Edition 6 × 14 TC pump and baseplate is presented in Figure 1. For this particular pump, the hold-down capscrews were increased from 1 inch to 1<sup>3</sup>/<sub>8</sub> inches in diameter, while the thickness of the pump feet and the deckplating under the support pedestals were increased from 1 inch to 2<sup>1</sup>/<sub>4</sub> inches and <sup>1</sup>/<sub>4</sub> inch to 1 inch, respectively. To prevent slippage at the foot to pedestal pad interfaces, hold-down capscrew preloads have been increased, and hardened steel washers are now being furnished.

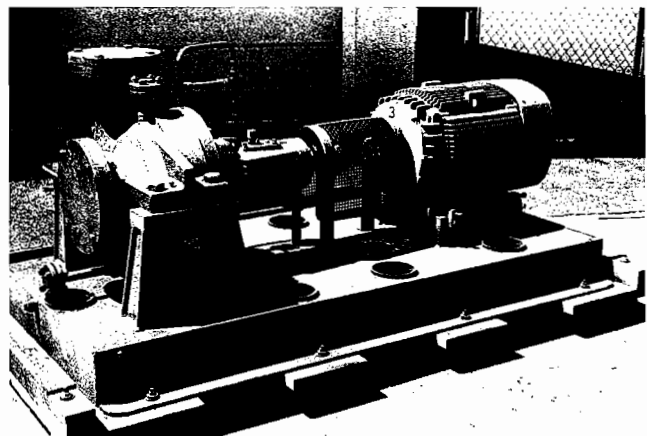


Figure 1. Sixth Edition Overhung Process Pump (6 × 14 TC).

To further illustrate the changes in construction, Figures 2, 3, 4 and 5 have been included. Figure 2 is a picture of a 1<sup>1</sup>/<sub>2</sub> × 11 TC case pattern modified to show the differences in 5th and 6th Edition pump casings. The foot attached to the discharge nozzle side of the casing has not been modified, and represents 5th Edition proportions. The foot on the opposite side has been thickened to satisfy our case stiffness requirement (0.002 inch). Pump feet have been increased in thickness by a factor ranging

between 2 and 5. Pumps with large nozzles relative to impeller diameter (case size) require thicker feet. Reinforcement of the volute wall in the vicinity of the pump feet has been provided so that the local case thickness is never less than one-third of the foot thickness.

The change made in the hold-down capscrews to achieve the 0.0015 inch shaft displacement criteria is also illustrated in Figure 2. Capscrews generally increase by about 4 sizes, i.e., from  $\frac{3}{4}$  inch-10 to  $1\frac{1}{4}$  inch-7. To accommodate the larger capscrews, the pump foot pads are frequently increased in length and/or width. For duplicate 5th Edition pumps, this added material is milled away and the original smaller diameter capscrew holes are drilled. Longer capscrews must be furnished due to increased foot thickness.

Pictures of a 6th Edition baseplate are featured in Figures 3 and 4. These pictures illustrate the revised method of attaching the support pedestals to the baseplate. As can be seen, the support pedestals are welded directly to a 1 inch to  $1\frac{1}{2}$  inch thick deckplate that is in turn attached to the baseplate structural members by continuous fillet welds  $\frac{3}{8}$  inch to  $\frac{1}{2}$  inch in size. The heavy deckplate with no pedestal "cut-outs" significantly reduces any local plate deformation and permits the use of large support pedestal attachment welds ( $\frac{3}{8}$  inch to  $\frac{3}{4}$  inch). The degree of "fixity" at the support pedestal to baseplate interface increases with weld size. Support pedestals on 5th Edition process pump baseplates (Figure 5) were attached to the structural members (short sides only) and to the quarter inch thick deckplate. The attachment weld size ( $\frac{1}{8}$  inch- $\frac{3}{8}$  inch) was limited by the thickness of the structural members and the deckplate.

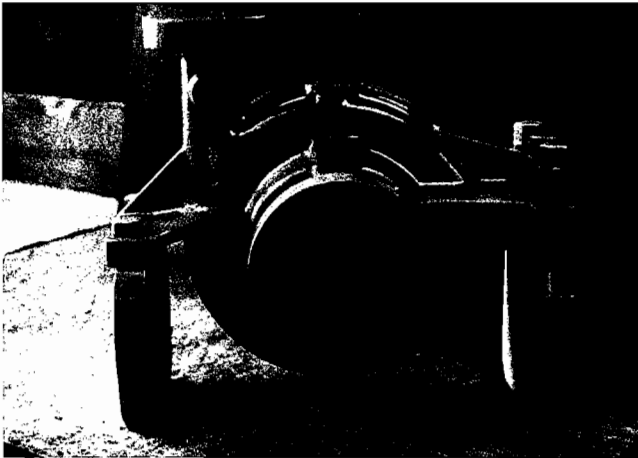


Figure 2. Modified Fifth/Sixth Edition Overhung Process Pump Case Pattern ( $1\frac{1}{2} \times 11$  TC).

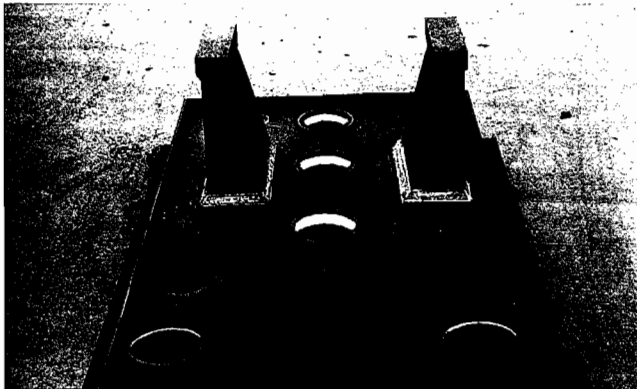


Figure 3. Topside View of a Sixth Edition Angle Type Baseplate.

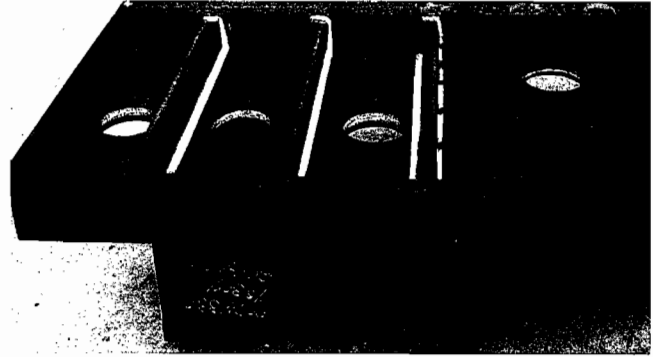


Figure 4. Underside View of a Sixth Edition Angle Type Baseplate.



Figure 5. Underside View of a Fifth Edition Angle Type Baseplate.

## ANALYTICAL WORK

The design modifications previously described are the consequence of our stiffness criteria and the associated analytical sizing procedures. Changes to either the stiffness criteria or the sizing procedures could result in different equipment modifications. For this reason, it is important to understand the equations and inherent assumptions made in the sizing procedures.

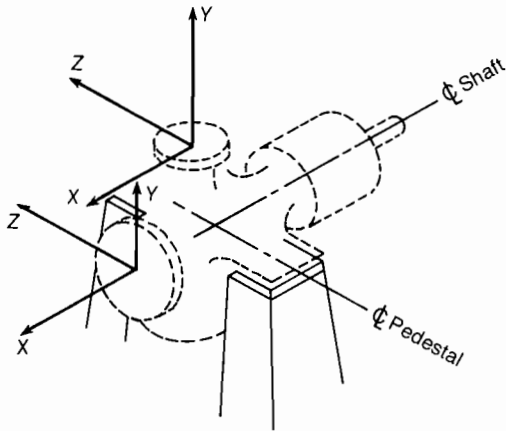
The sizing equations shown on the following pages have been derived for AP-610 Table 2 Nozzle Loads (Figure 6). These component loads are unsigned and, therefore, define direction and magnitude range. This means that each nozzle can be subjected to an infinite number of loading conditions. This obstacle has been overcome by defining a worst case condition for each of the three major subcomponents. The following text describes the equations and assumptions inherent in each subcomponent sizing procedure.

### The Pump Casing

Previous research has demonstrated that overhung process pumps are most sensitive to moments about the Z axis [2]. This is attributable to a lack of torsional rigidity in the pump feet which transmit piping loads from the relatively stiff pump casing to the support pedestals. By comparing pump case geometry, the same statement can be made of one and two stage double bearing process pumps. Knowing this fact and by making the following simplifying assumptions, one can develop three equa-

Table 2—Nozzle Loadings

Force/Moment <sup>a</sup>	Nominal Size of Nozzle Flange (inches)								
	≤2	3	4	6	8	10	12	14 <sup>b</sup>	16 <sup>b</sup>
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



—Coordinate System for the Forces and Moments in Table 2

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.

<sup>a</sup> 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.

<sup>b</sup> These values are for guidance only and are subject to negotiation between the purchaser and the vendor for the specific application.

Figure 6. Nozzle Loadings and Associated Coordinate System (Extracted from API-610 Sixth Edition).

tions which can be used to predict pump casing design modifications which will limit the shaft displacement at the coupling to 0.002 inch.

ASSUMPTIONS

1. The body of the casing is rigid relative to the attachment feet.
2. The pump casing material has a modulus of rigidity equal to  $11.5 \times 10^6$  PSI (Cast Steel).
3. All loads other than those moments about the Z axis cause negligible shaft displacement and can be ignored.
4. The maximum resultant Z moment,  $(MZC)_{MAX}$ , due to Table 2 Piping Loads, produces a deflection at the pump shaft coupling hub equal to 0.002 inch.

$$J_{MIN} = \frac{(5.2 \times 10^{-4}) \times (MZC)_{MAX} \times L_2 \times W}{\left(\frac{L_2}{L_1} + 1\right)} \tag{1}$$

$$MZC = \frac{MZS + MZD - [(FXS)(YS) + (FXD)(YD) - (FYS)(XS) - (FYD)(XD)]}{12} \tag{2}$$

$$J_{ACT} = B_f L_f T_f^3 + B_r T_r^3 \tag{3}$$

In establishing the value of  $J_{MIN}$ , it is necessary also to establish  $(MZC)_{MAX}$ . This value is obtained from Equation (2) using API-610 Table 2 Loads (Figure 6) with appropriate signs to max-

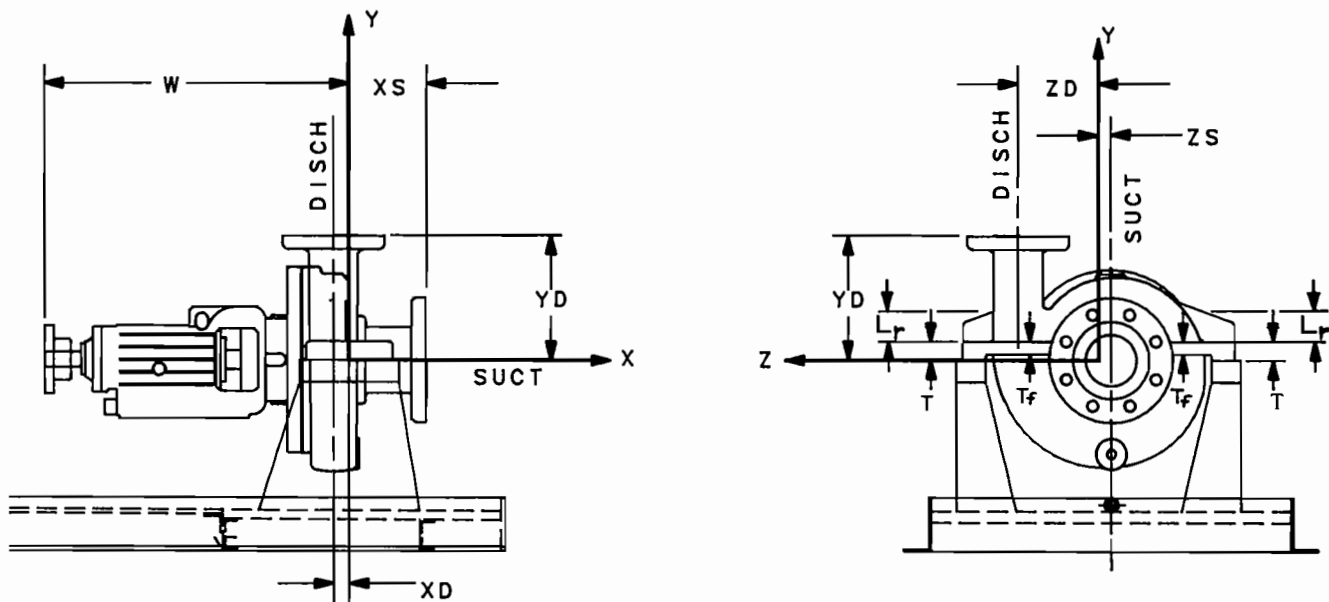


Figure 7a. End Suction Overhung Process Pump Nomenclature.

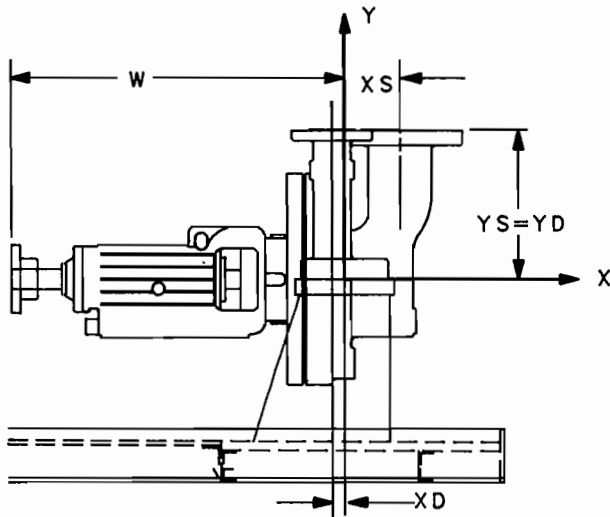


Figure 7b. Top Suction Overhung Process Pump Nomenclature.

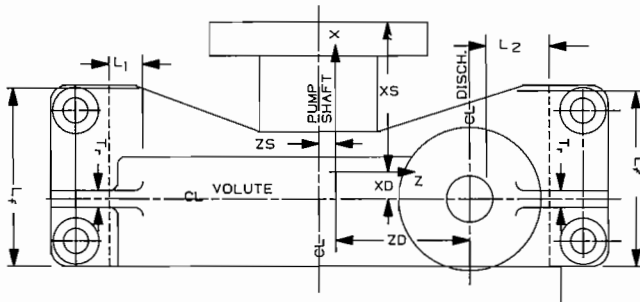
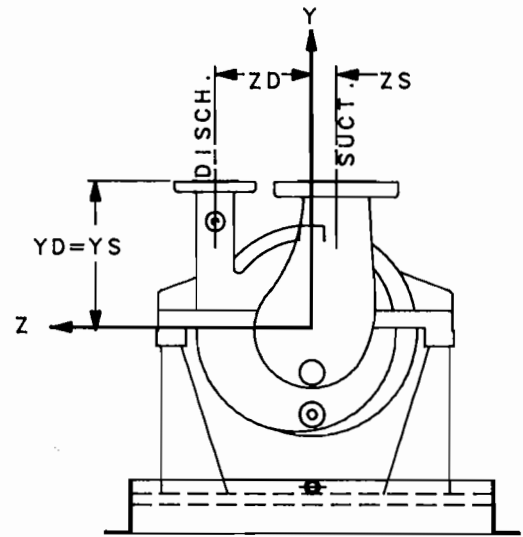


Figure 8. Top View of an End Suction Overhung Process Pump with Foot-Sizing Nomenclature.

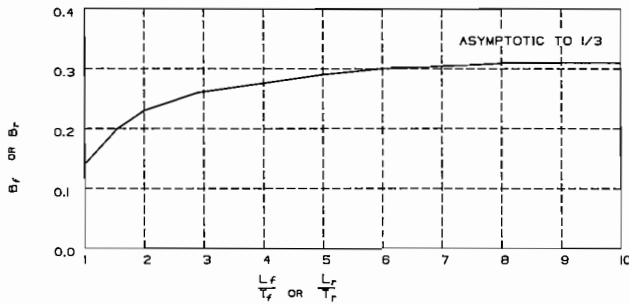


Figure 9. Pump Foot or Rib Torsional Coefficient Curve.

imize MZC. The actual torsional constant of an attachment foot,  $J_{ACT}$  should be slightly larger than  $J_{MIN}$  so that the calculated shaft displacement is less than 0.002 inch. The meaning of the variables used in Equations (1, 2 and 3) is clarified in Figures 7, 8 and 9.

**The Pump to Support Pedestal Capscrews**

The cross-sectional area of the hold-down capscrews must be sufficient to limit shaft displacement at the coupling to 0.0015 inch. It is assumed that any elongation of the capscrews is amplified at the coupling due to rigid body rotation of the

pump casing. The permissible capscrew elongation is a function of pump geometry and can be evaluated by Equation (4).

$$D_c = \frac{0.0015 D_2}{0.5 D_2 + W} \tag{4}$$

Having an expression for the permissible capscrew elongation, an equation for the cross-sectional area has been derived. Equation (5) must be solved using a "trial and error" technique since the cross-sectional area is a function of the capscrew diameter and assumes:

1. Capscrew preload can be neglected.
2. The capscrew material has a modulus of elasticity of  $2.9 \times 10^7$  PSI (steel).
3. The effective length of the capscrew is equal to the foot pad thickness plus 80 percent of the capscrew diameter ( $T + 0.8d$ ).

$$A_{MIN} = \frac{3.45 \times 10^{-8} (FY)_{MAX} (T + 0.8d)}{D_c} \tag{5}$$

For a four hold-down capscrew configuration as shown in Figure 10, the maximum tensile force,  $FY$ , can be determined by equation (6).

$$(FY)_{MAX} = \frac{(FYC)_{MAX} - PUMP\ WT}{4} + \frac{6(MXC)_{MAX}}{D_1 + \frac{D_o^2}{D_1}} + \frac{6(MZC)_{MAX}}{D_2 + \frac{D_3^2}{D_2}} \tag{6}$$

where

$$FYC = FYS + FYD \tag{7}$$

and

$$MXC = MXS + MXD - [(FYS) (ZS) + (FYD) (ZD) - (FZS) (YS) - (FZD) (YD)]/12 \tag{8}$$

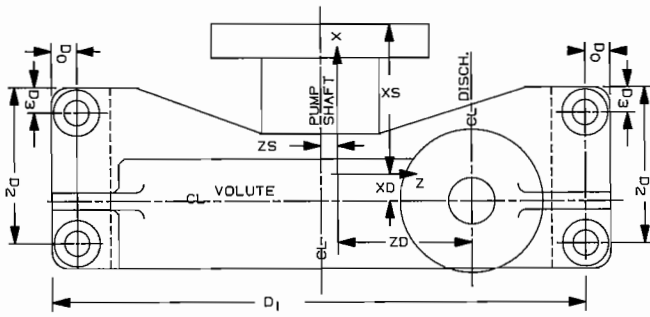


Figure 10. Top View of an End Suction Overhung Process Pump with Hold-Down Capscrew Nomenclature.

Each of the resultant loads shown in Equation (6) are individually maximized by selecting appropriate signs for the component loads. This simplifying technique produces conservatively large values for  $(F_Y)_{MAX}$  due to the inconsistent assignment of component load directions.

Besides meeting the stiffness criteria, the hold-down capscrews must be properly sized and tightened to prevent any sliding at the foot pad to support pedestal interface. Calculations indicate that casing thermal growth can cause horizontal shear forces ( $F_{ZT}$ ) that are significantly larger than the piping loads. Friction forces generated by the hold-down capscrews must be

sufficient to provide non-sliding foot pad to support pedestal interfaces. Equation (9) can be used to confirm whether the capscrew size determined by Equation 5 is sufficient. Equation 9 assumes that the capscrews are preloaded to 75 percent of their proof load by tightening and that the static coefficient of friction between the bearing members is 0.74.

$$A_{MIN} = \frac{1.33}{PL} [0.675 (F_{XP})^2 + (F_{ZP} + \frac{2F_{ZT}}{N})^2 + (F_Y)_{MAX}] \tag{9}$$

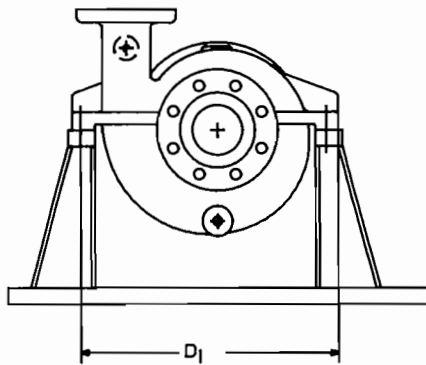
where

$$F_{XP} = \frac{(F_{XC})_{MAX}}{4} + 6(MYC)_{MAX} (D_1 - D_o) / [(D_2 - D_3)^2 + (D_1 - D_o)^2] \tag{10}$$

$$F_{ZP} = \frac{(F_{ZC})_{MAX}}{4} + 6(MYC)_{MAX} (D_1 - D_o) / [(D_2 - D_3)^2 + (D_1 - D_o)^2] \tag{11}$$

*The Support Pedestal Assembly*

Due to the increased stiffness of 6th Edition process pump casings, we believe that the support pedestals must be designed to absorb casing thermal growth in addition to meeting the



UNITED PUMP TYPE AND SIZE	D1 DIMENSION INCHES	CASING THERMAL GROWTH (INCHES) WHEN HANDLING 500°F PRODUCT			CASING THERMAL GROWTH (INCHES) WHEN HANDLING 700°F PRODUCT		
		CARBON STEEL	HIGH CHROME STEEL	AUSTENITIC STAINLESS STEEL	CARBON STEEL	HIGH CHROME STEEL	AUSTENITIC STAINLESS STEEL
3 X 7 TC	16-1/8	.034	.029	.048	.052	.044	.071
4 X 9 TC	21-7/8	.047	.039	.065	.070	.060	.096
6 X 13 DVSHF	35-3/8	.076	.063	.105	.113	.097	.155
8 X 16 DSTHF	41	.088	.073	.122	.131	.113	.180
8 X 21 TC	38-3/8	.082	.069	.114	.123	.105	.168
10 X 22 TC	42-3/8	.091	.076	.126	.136	.117	.186
10 X 27 DVSHF	53	.113	.095	.158	.170	.146	.233
14 X 29 (32) DVSHF	63	.135	.113	.187	.202	.173	.277

Figure 11. Tabulation of Calculated Horizontal Process Pump Thermal Expansion.

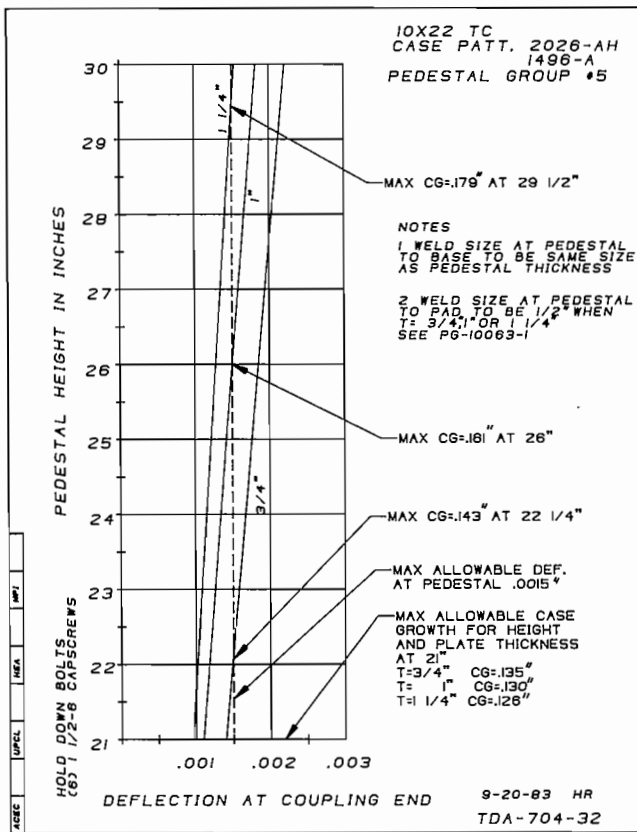


Figure 12. Computer Generated Pedestal Selection Chart.

.0015 inch subcomponent stiffness criteria.

If casing thermal growth, as illustrated in Figure 11, is not absorbed in an acceptable manner, pump operability may be affected due to shaft misalignment [2]. Shaft misalignment can be caused by: 1) unsymmetrical support pedestals that twist or do not absorb casing thermal growth equally, 2) sliding of the pump across the support pedestal pads as a result of insufficient hold-down capscrew preload, 3) permanent distortion (yielding) of the load carrying members due to insufficient support pedestal flexibility.

Our computer generated pedestal selection charts (Figure 12) are used to ensure that support pedestals are stiff enough to

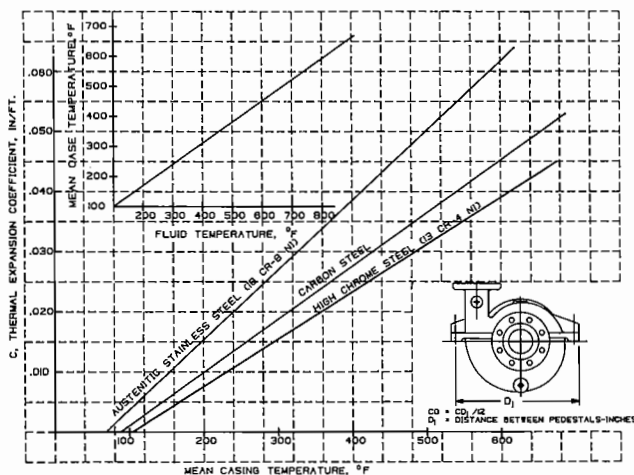


Figure 13. Horizontal Process Pump Thermal Expansion Chart.

meet the .0015 inch deflection criteria, yet flexible enough to absorb the casing thermal growth without causing undue shaft misalignment. The solid lines identified in Figure 12 as (3/4, 1, 1 1/4) represent pedestal plate thickness while CG is the allowable casing thermal growth which must be greater than the calculated thermal growth established by Figure 13 and Equation (9). To meet the subcomponent stiffness criteria, the deflection at the coupling end must be less than the allowable value which is nominally 0.0015 inch and is indicated by the dashed vertical line.

### EXPERIMENTAL TESTING

Experimental testing has been conducted to verify that the analytical procedures previously described produce equipment that meets API's design criteria. Loads up to two times (MXC)<sub>MAX</sub>, (MYC)<sub>MAX</sub>, (MZC)<sub>MAX</sub> were applied to assembled pumps by hanging known weights to piping attached to the suction and discharge nozzles (Figure 14). For each load incre-

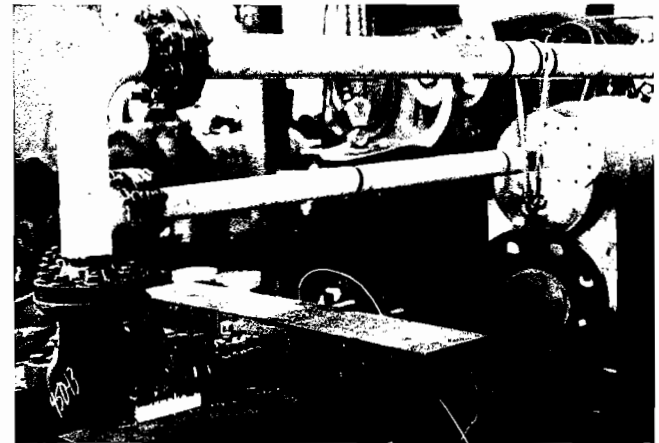


Figure 14. Side View of an Overhung Process Pump Case Stiffness Test Set-Up.

ment, an operability test (paragraph 2.4.2) and a stiffness test (paragraph 2.4.3) were performed. By neglecting the effect of internal pressure (radial hydraulic load), a simple test for operability was devised: To detect any internal contact of rotating and stationary components due to the imposed piping loads, the rotor was rotated by hand. To date, no perceptible rubbing has been felt. Assembly stiffness was measured in terms of shaft displacement at the coupling. A digital voltmeter and three noncontacting eddy current probes were used to measure shaft movement in the X, Y, Z directions (Figure 15).

Factory tests were conducted with assembled pumps bolted to large rigid shop pedestals secured to a bedplate. The proximity probes were attached to a 9-inch channel which was supported by the shop pedestals. With this test arrangement, the effect of support pedestal flexibility has been eliminated. This means that the measured shaft deflection at the coupling will be due to flexibilities in the pump casing (0.002 inch) and the hold-down capscrews (0.0015 inch). Test results for three overhung process pumps are presented graphically in Figures 16-18. All graphs are plotted to the same scale to illustrate that the pump casing is most sensitive to moments about the Z axis. To demonstrate "trends" as a function of pump size and load orientation, displacements acting in the same direction have been connected by solid, dashed and dotted lines in these figures. This experimental data indicates that the attachment foot and hold-down capscrew sizing procedures are satisfactory for predicting design modifications. The data also indicated that

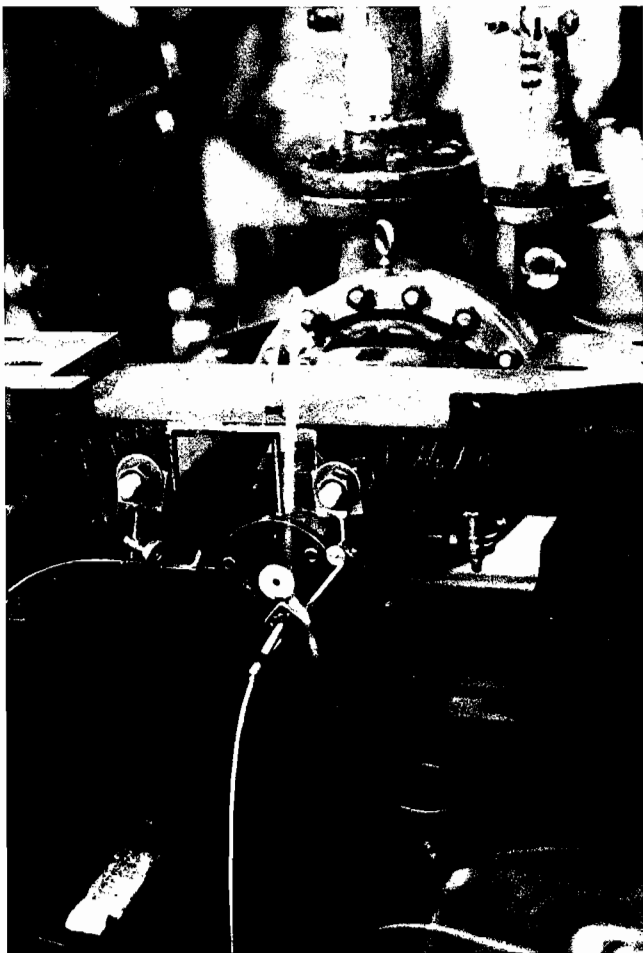


Figure 15. End View of an Overhung Process Pump Case Stiffness Test Set-Up.

the Z displacement due to moments about the Y axis may not be negligible and that these pump casings would be unacceptable for a heavy-duty baseplate application.

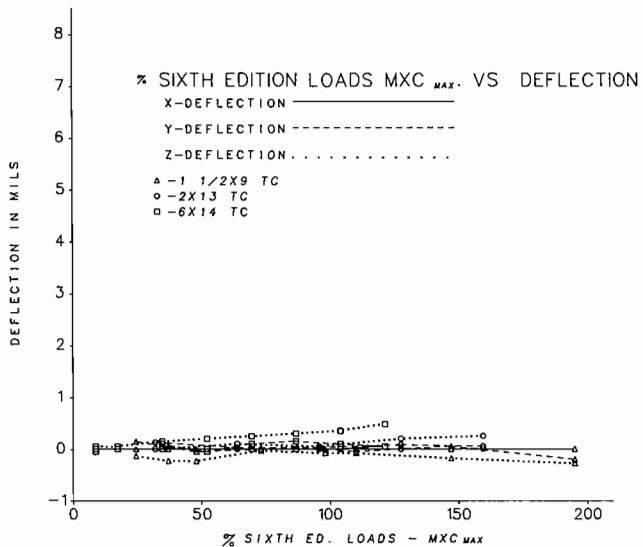


Figure 16. Experimental Pump Case and Hold-Down Capscrew Stiffness Data for MXC Moments.

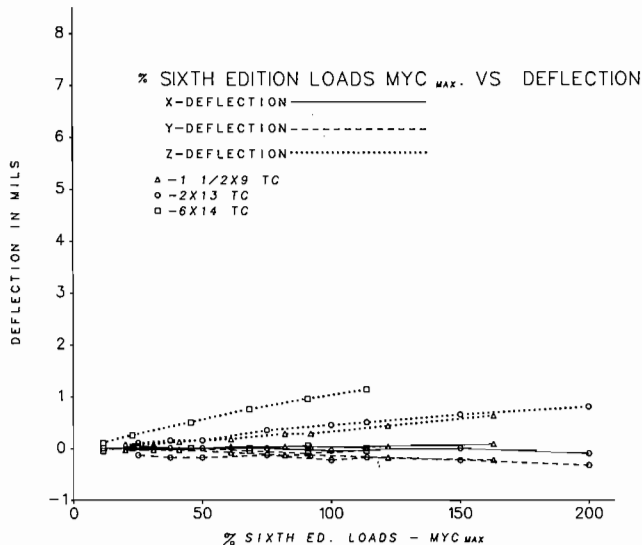


Figure 17. Experimental Pump Case and Hold-Down Capscrew Stiffness Data for MYC Moments.

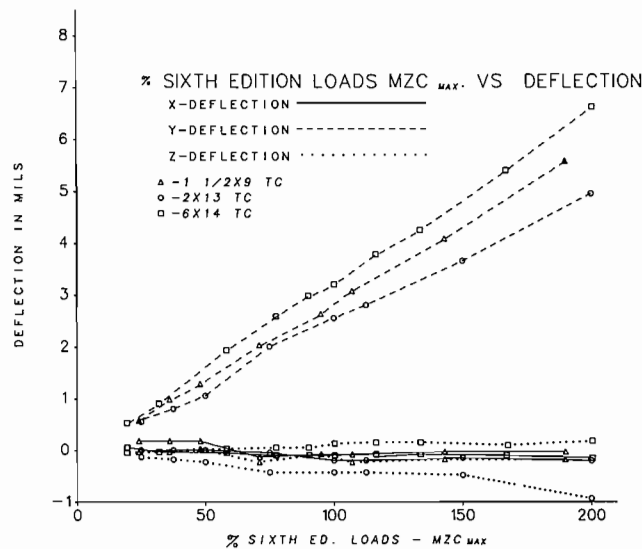


Figure 18. Experimental Pump Case and Hold-Down Capscrew Stiffness Data for MZC Moments.

The results of field tests conducted on a 3 × 9 TC pump mounted on its 6th Edition baseplate are illustrated in Figure 19. The test set-up was similar to that previously described, except shaft displacements at the coupling hub were measured relative to the concrete foundation. This was done to take into account support pedestal and baseplate flexibilities. The magnitude of the applied moments was also limited by the available working space. The field data shown in Figure 19 are remarkably similar to the case stiffness test data illustrated in Figures 16, 17 and 18. The calculated overall displacement at the coupling hub assuming that  $(MXC)_{MAX}$ ,  $(MYC)_{MAX}$ ,  $(MZC)_{MAX}$  moments occur simultaneously is 3.63 mils. This verifies that the 3 × 9 TC pump, when mounted on its grouted-in baseplate, meets API-610 nozzle loading requirements.

HEAVY-DUTY BASEPLATES

Paragraph 2.4.6 states that an optional heavy-duty baseplate can be supplied which will double the stiffness of the pump, baseplate, and support pedestal assembly. Calculations

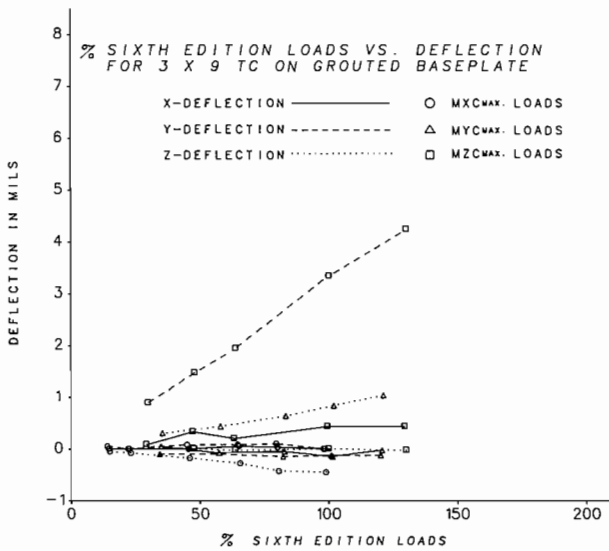


Figure 19. Experimental Pump and Baseplate Stiffness Data.

and test data indicate that a heavy-duty baseplate alone cannot meet this requirement. Assuming that a perfectly rigid baseplate and pedestal support structure (cold product applications) could be designed and manufactured, the pump casing and hold-down capscrew deflection criteria would need to be changed from 0.0035 inch to 0.0025 inch based on Table 2 piping loads. Special thick-walled pump casings would probably be required since the standard 6th Edition attachment feet are stiffer than the pump casing.

According to API-610, the purpose of the heavy-duty baseplate was to simplify piping layouts by allowing higher loads from the attached piping. In most instances high piping loads occur when hot product is being handled. In these applications, the support pedestals must have flexibility in order to absorb pump casing thermal expansion. For this reason, we do not feel that it is advisable to furnish stiffer support pedestals. Assuming that the support pedestal stiffness is not increased, the deflection criteria for the pump casing and hold-down capscrews must be changed from 0.0035 inch to 0.001 inch using Table 2 loads as a basis. This means that special thick-walled pump casings would need to be about 3½ times stiffer than the standard 6th Edition casings previously described. Obviously, this is not a practical solution to the heavy-duty baseplate requirement.

We believe that contractors and users would be better off to specify standard 6th Edition baseplates and submit component piping loads when they exceed Table 2 (Figure 6) values. In most instances higher component piping loads can be imposed and still meet the 0.005 inch deflection criteria. This is particularly true when  $MZC_{MAX}$  based on actual piping loads is less than  $MZC_{MAX}$  based on API-610 Table 2 loads. If the component piping loads are too large to meet the 0.005 inch deflection criteria, we can supply the contractor with pump nozzle and pedestal support stiffnesses which can be input into the piping flexibility analysis. Pumps are normally modelled as rigid pipe anchors; introducing nozzle and pedestal flexibilities into the computer model generally results in smaller component piping loads.

NOMENCLATURE

$A_{MIN}$  Required hold-down capscrew tensile stress area (in<sup>2</sup>) to limit shaft displacement at the coupling to 0.0015 inch and to prevent slid-

- $B_i, B_r$  Torsional coefficient from Figure 9 for pump feet or ribs.
- $d$  Nominal diameter (in) of the pump hold-down capscrews.
- $D_c$  Hold-down capscrew displacement (in) that will produce a 0.0015 in deflection at the pump shaft coupling hub due to rigid body rotation of the pump casing.
- $D_0, D_1, D_3$  Dimensions (in) used to define the hold-down capscrew pattern (Figure 10).
- $FXC, FYC, FZC$  Resultant forces (lbs) acting at the center of the pump in the X, Y or Z direction due to suction and discharge nozzle component forces.
- $FXD, FYD, FZD$  Component forces (lbs), extracted from Table 2 of API-610, 6th Edition, acting on the discharge nozzle flange facing in the X, Y or Z direction.
- $FXP, FZP$  X or Z component of the resultant horizontal shear force (lbs) per hold-down capscrew due to Table 2 piping loads.
- $FXS, FYS, FZS$  Component forces (lbs), extracted from Table 2 or API-610, 6th Edition, acting on the suction Nozzle flange facing in the X, Y or Z direction.
- $(FY)_{MAX}$  Maximum axial force (lbs) per hold-down capscrew due to API-610 Table 2 nozzle loads.
- $FZT$  Horizontal shear force (lbs) per support pedestal due to casing thermal expansion. Evaluated with computer program base-load.
- $J_{ACT}$  Actual torsional constant (in<sup>4</sup>) of a pump foot.
- $J_{MIN}$  Minimum acceptable torsional constant (in<sup>4</sup>) for a pump foot which will limit shaft deflection at the coupling hub to 0.002 inches for a given  $(MZC)_{MAX}$  moment.
- $L_f$  Effective length (in) of a pump foot parallel to the pump shaft (Figure 8).
- $L_r$  Effective length of a pump foot rib (Figure 7).
- $L_1$  Effective length (in) of a pump foot opposite the discharge nozzle and perpendicular to the pump shaft (Figure 8).
- $L_2$  Effective length (in) of the pump foot on the discharge nozzle of the casing and perpendicular to the pump shaft (Figure 8).
- $MXC, MYC, MZC$  Resultant moment (ft/lbs) about the X, Y or Z axis based on component piping loads (Table 2 of API-610) resolved to a coordinate system located at the center of the pump.
- $MXD, MYD, MZD$  Components moments (ft/lbs), extracted from Table 2 of API-610, acting on the discharge nozzle flange facing about the X, Y and Z axis.
- $MXS, MYS, MZS$  Component moments (ft/lbs), extracted from Table 2 of API-610, acting on the suction nozzle flange facing about the X, Y or Z axis.
- $N$  Total number of hold-down capscrews.



PL	Hold-down capscrews proof load (psi).
T	Thickness (in) of the pump foot in the vicinity of the hold-down capscrews (Figure 7).
$T_f$	Thickness (in) of the pump foot at the foot to casing junction (Figure 7).
$T_r$	Thickness (in) of the pump foot ribs (Figure 8).
W	Distance (in) from the center of the pump foot to the face of the pump shaft coupling hub (Figure 7).
XD,YD,ZD	Location coordinates (in) of the discharge nozzle flange facing (Figures 7 and 8).
XS, YS, ZS	Location coordinates (in) of the suction nozzle flange facing (Figures 7 and 8).

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