A Study of Allowable Piping Loads on Double-suction Centrifugal Pumps*
(2nd Report, Shaft Displacement Due to Pump Casing Deformation)

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Shaft displacement due to pump casing deformation which occurs when external piping forces and moments are applied was investigated. The test pump was an axially split, single-stage double suction volute type with foot support. An analysis by FEM(MSC/NASTRAN) was performed and considerably good agreement was obtained with actual measurement. It was found that shaft displacement due to pump casing deformation can not be neglected for pumps with foot support. In API 610-6th standard, limit of shaft displacement is 0.127 mm for pumps which have suction bores 300 mm or less. But according to this investigation, it is reasonable with pumps having suction bore diameters larger than 300 mm to enlarge this limitation. Effects of pump size and casing thickness on shaft displacement were investigated by FEM. The possibility of a galling of the casing wearing ring and a considerable leakage from mechanical seal caused by piping loads was also investigated.

Key Words: Hydraulic Machine, Structural Analysis, Centrifugal Pump, Allowable Load, Nozzle, Shaft Displacement, Casing Wearing Ring, Mechanical Seal, Piping, Finite Element Method.

1. Introduction

When piping loads are applied to pump nozzles, stress and strain on the pump casing increase, and shaft displacement occurs due to deformation of the pump casing or baseplate. An increase in the shaft displacement causes a misalignment of the coupling between a pump and driver, which will increase pump vibration, and may cause a failure of the coupling or damage to pump casing. In an extreme case, a failure of a critical pump could cause shutdown of an entire plant.

API-610 standard, 6th edition(1) (hereinafter referred to as API) specifies various elements of the structural design of pumps. Recently, International organization for standardization is also preparing draft of regulations for structural design(2). Although there are various problems of interpretation when applying API(3), the standard is very useful practically, and its influence is far reaching.

Shaft displacement is caused by deformation of both casing and baseplate. According to Bussemaier(4), in small center-supported single-stage centrifugal pumps, shaft displacement due to the deformation of pedestal or baseplate is predominant and larger than that due to pump casing deformation. However, in foot-supported pumps which are the most commonly used and have a flexible casing, it is considered that shaft displacement due to pump casing deformation is too large to be neglected. Another unsolved problem in structural design is that the allowable shaft displacement in API is the same regardless of pump size (5). The effect of pump size on shaft displacement is still not clear. On the other hand, with the appearance of a super heavy-duty pump of single-stage double-suction type with hydraulic pressure to 20 MPa (204 kgf/cm²)(6), demand for the establishment of a suitable structural design related to piping loads has become urgent.

In this report, shaft displacement due to pump casing deformation is investigated by both actual measurements and structural analysis by the finite element method (hereinafter referred to as FEM) under various load conditions. The possibility of a galling of rotor elements due to deformation of casing wearing ring and a leakage or breakage of mechanical seals caused by piping loads is very important for rotating machinery, and it is also taken into consideration.

2. Nomenclature

\[ D_r : \text{pump discharge diameter} \quad \text{mm} \]
\[ E : \text{modulus of longitudinal elasticity} \quad \text{kN/\text{mm}^2} \]
\[ F : \text{force} \quad \text{kN} \]
Test pumps used were four foot-supported single-stage double-suction centrifugal pumps. These were the same pumps as used in a previous report(7). Pump dimensions and coordinates are described in that report.

Fig. 1 shows an outline of the test apparatus. View A - A in Fig. 1 shows load condition when torsional moment $M_\alpha$ is applied to the pump discharge nozzle. In addition to the four cap screws, two dowel pins located diagonally were used to fix the feet and prevent a slip on the steel base which would cause errors in measurement of shaft displacement. Shaft displacements were measured in $x, y, z$ directions by displacement detectors of cantilever strain gauge type. The detectors were placed at the measuring points so as to take readings in the middle of their scale to permit measurement of shaft displacement in both directions. Fig. 2 shows the pump outline when shaft displacements were measured.

In addition to the shaft displacements, relative movements between stuffing boxes and shaft were also measured to investigate the effects of piping loads on sealing of the mechanical seals. Actual mechanical seals were assembled to stuffing boxes, and leakage and damage of the mechanical seals due to piping loads were checked when internal pressures were applied to the casing.

4. Experimental Results

4.1 Shaft displacement due to a single load

First, investigation was made for shaft displacement, when a single piping force or moment was applied to the pump nozzles. Fig. 3 shows an example of the results when a single force was applied to pump 4. It is observed in the figure that, for each piping force, the shaft displacement increases approximately in proportion to the loads. This verifies that the main cause of shaft displacement is an elastic deformation of the pump casing and effects of the slip of pump foot are small. The figure also demonstrates that, when a piping force $F_p$ was 2.5 times the piping load specified in API (hereinafter referred to as API load), shaft displacement exceeded the limit of the allowable shaft displacement specified in API. Then, it is clear that in foot-supported pumps, shaft displacements due to casing deformation are so large that they cannot be neglected.

Fig. 4 and Fig. 5 show the results of analysis of the casing deformation and stress distribution by FEM when a moment $M_\alpha$ is applied to the discharge nozzle. Highly deformed parts show high stresses, demonstrating that casing deformation and stress correspond to each other even in such a complex shape.

Fig. 6 shows a comparison between analysis of shaft displacement by FEM and actual measurement when a single piping load is applied to pump 1. Good agreement of analytical results with measured ones for shaft displacement is obtained as was for casing stresses in the previous report. According to Simon (9), moments $M_\alpha$ and $M_\beta$ cause a much larger shaft displacement than other piping loads with center-supported single-suction centrifugal pumps. With foot-supported double-suction pumps, each piping force or moment produces almost the same amount of shaft displacement. From this it is clear that magnitude of the effect of each piping load greatly depends on the type of pump support. As the largest shaft displacement occurs in the $z$ direction for all loads except $M_\alpha$ or $M_\beta$, care must be taken with pipe support in the $z$ direction.
(the flow direction of fluid) for this type of pumps.

4.2 Shaft displacement caused by plural loads

In the previous report, it was stated that casing stress greatly depends on the directions of pipe loads. In the present study the same investigation was carried out for shaft displacement. Fig. 7 shows the results about shaft displacement by both FEM and actual measurement when API forces and moments were applied to pump 1 at the same time. In all cases, the magnitudes of each set of loads are the same but the directions are different for each case as shown in Table 1. In the same way as with casing stresses in the previous report, measured values were calculated by integrating shaft displacements of each single piping load. As can be seen from Fig. 7, the magnitude of shaft displacement varied greatly depending on the direction of load. In case 1, very small shaft displacements were caused due to cancelling effect of opposed loads. But in case 3, as the result of integration of each load, very large values of shaft displacement occurred.

Table 2 shows shaft displacements when plural API loads were applied to

![Fig. 2 Pump outline during shaft displacement measurement](image)

![Fig. 3 Measured results of shaft displacement of pump 4](image)

![Fig. 4 Casing deformation due to \( -M_e \)](image)

**Stress Contour Level**

1: 0 Mpa, 6: 12.3 Mpa
2: 2.5 Mpa, 7: 14.7 Mpa
3: 4.9 Mpa, 8: 17.2 Mpa
4: 7.4 Mpa, 9: 19.6 Mpa
5: 9.8 Mpa, 10: 22.1 Mpa

![Fig. 5 Casing stress distribution due to \( -M_e \)](image)
pumps 1-4, where the directions of loads were the same as those in case 3 in Table 1. Comparing two pumps of different sizes but with the same \( N_o \) and the same maximum casing pressure (pumps 1 and 3 or pumps 2 and 4), shaft displacement increased approximately in proportion to the pump suction diameter. API uniformly limits shaft displacement to 0.127 mm for pumps with suction nozzle diameters of 300 mm or less, although the reason for this is not clear. But as the above table indicates, it is reasonable to increase the allowable limit of shaft displacement in proportion to the suction nozzle diameter, at least for foot-supported pumps with suction nozzle diameters of 350 mm or more.

4.3 Effect of pump size and casing thickness

Double suction centrifugal pumps range in size of suction bore from 100 mm to 1500 mm. The effect of pump size on shaft displacement was investigated. Its aim was to develop a method to obtain shaft displacement in pumps of different sizes and casing thicknesses from data on a similar shaped pump with known shaft displacement.

First, for simplification of analysis, a pump is assumed to be a circular cross section, and its displacement is investigated. Supposing that there are two cantilevers of the same shape but different sizes, we put the dimension ratio as \( m \). That is, when the length of one cantilever is \( l \) and its diameter is \( d \), the length and diameter of the other cantilever are \( ml \) and \( md \), respectively. When the same magnitudes of external forces \( F \) are applied at the ends of the two cantilevers, the deflections \( \delta \) and \( \delta' \) can be expressed as follows.

\[
\delta = \frac{F l'^2}{3 E I} \propto \frac{F l'^2}{d^4} \quad \text{(1)}
\]
\[
\delta' = \frac{F (ml)'^2}{3 E I} \propto \frac{F (ml)'^2}{m^4 d'^4} \quad \text{(2)}
\]

From these, the following equation can be obtained.

<table>
<thead>
<tr>
<th>Load (kN)</th>
<th>Direction (kN-m)</th>
<th>Case 1</th>
<th>Case 2</th>
<th>Case 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>7.1</td>
<td>+</td>
<td>-</td>
<td>+</td>
<td></td>
</tr>
<tr>
<td>5.8</td>
<td>-</td>
<td>-</td>
<td>+</td>
<td></td>
</tr>
<tr>
<td>8.9</td>
<td>+</td>
<td>+</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>6.4</td>
<td>-</td>
<td>-</td>
<td>+</td>
<td></td>
</tr>
<tr>
<td>4.7</td>
<td>-</td>
<td>+</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>3.1</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>6.7</td>
<td>+</td>
<td>+</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>5.3</td>
<td>-</td>
<td>+</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>8.0</td>
<td>-</td>
<td>+</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>6.1</td>
<td>+</td>
<td>-</td>
<td>+</td>
<td></td>
</tr>
<tr>
<td>4.6</td>
<td>+</td>
<td>+</td>
<td>-</td>
<td></td>
</tr>
<tr>
<td>3.0</td>
<td>+</td>
<td>-</td>
<td>-</td>
<td></td>
</tr>
</tbody>
</table>

Table 2 Shaft displacement of pump 1-4 by API loads

<table>
<thead>
<tr>
<th>Items</th>
<th>Pump 1</th>
<th>Pump 2</th>
<th>Pump 3</th>
<th>Pump 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sd (mm)</td>
<td>350</td>
<td>350</td>
<td>500</td>
<td>500</td>
</tr>
<tr>
<td>Dd (mm)</td>
<td>300</td>
<td>250</td>
<td>400</td>
<td>350</td>
</tr>
<tr>
<td>Ns</td>
<td>375</td>
<td>200</td>
<td>375</td>
<td>200</td>
</tr>
<tr>
<td>Material</td>
<td>FC25</td>
<td>FCD40</td>
<td>FC25</td>
<td>FCD40</td>
</tr>
<tr>
<td>( \delta )</td>
<td>191</td>
<td>143</td>
<td>314</td>
<td>217</td>
</tr>
</tbody>
</table>
\[ \delta = \frac{d_0}{m} \tag{3} \]

In the same way, the following equation can be obtained for external moment.

\[ \delta = \frac{d_0}{m^2} \tag{4} \]

Consequently, it is theoretically recognized that larger pumps are more rigid against the same magnitude of external loads. However, actual pumps of the same size have various casing thicknesses due to the difference in the pressure ratings or manufacturing considerations such as castability. Therefore, when shaft vibrations due to casing deformation are calculated, it is necessary to separate the effects of casing deformation into two factors, that is, the effect of pump size and that of casing thickness.

For a pump with known shaft displacement due to piping loads and a nominal casing thickness of \( d_0 \), shaft displacement at a certain piping load is expressed by \( \delta \). For a pump of similar shape, the nominal casing thickness is expressed by \( t \) and the dimension ratio by \( m \). When the same piping loads are applied, shaft displacement is expressed by the following equation.

\[ \delta = \delta_s \left( \frac{d_0}{t} \right)^{\frac{a}{m}} \tag{5} \]

Here, index \( a \) represents the effect of dimension ratio and index \( b \) that of the thickness ratio on shaft displacement. Table 3 shows the calculated results of indices \( a \) and \( b \) using the FEM analysis model of pump 1. The values of \( a \) and \( b \) in Table 3 are limited to the values in the direction where the shaft displacement is predominant for each load. From Table 3, it is recognized that the magnitude of the effect of pump size or casing thickness on shaft displacement depends on the direction of piping forces and moments. For example, pump size or casing thickness has almost no effect on moment \( M_{90} \) and \( M_{90} \), but casing thickness has a large effect on forces \( F_{90} \) and \( F_{90} \). On the other hand, when the thickness ratio is the same as the dimension ratio, equation (5) becomes as follows.

\[ \delta = \frac{d_0}{m^2} \tag{6} \]

As shown in Table 3, the sum of the indices \( a \) and \( b \) is one (1st power) for piping forces and two (2nd power) for piping moments. Consequently by using FEM analysis, it is generally made clear for the first time that there is no problem practically in expressing the proportional relation of shaft displacement by the affine equations (3) and (4).

5. Deformation of Casing Wearing Ring

(a) Radial thrust and other factors can be causes of contact between pump rotating parts, such as the impeller wearing ring and the casing wearing ring, but the deformation of the casing wearing ring caused by casing deformation due to piping loads can also be a large factor. The casing shape of a double-suction centrifugal pump is similar to a sphere and most sections are generally circular. Relative movements of casing wearing ring and rotating elements were analyzed by FEM to investigate how they are deformed by the internal pressure or external loads and to investigate the possibility of the contact of rings. The results are shown in Table 4. From Table 4, it is recognized that the relative movements between the casing wearing ring and the shaft were very small, that is, below 0.01

<table>
<thead>
<tr>
<th>Load</th>
<th>Displacement of casing wearing ring (( \mu m ))</th>
<th>Displacement of shaft (( \mu m ))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fxs</td>
<td>( y ) 4 ( z ) 16</td>
<td>4 ( z ) 16</td>
</tr>
<tr>
<td>Fys</td>
<td>6 ( z ) 15</td>
<td>1 13</td>
</tr>
<tr>
<td>Fis</td>
<td>-8 ( z ) 42</td>
<td>0 -5</td>
</tr>
<tr>
<td>Mxs</td>
<td>-8 ( z ) 42</td>
<td>0 -5</td>
</tr>
<tr>
<td>Mys</td>
<td>5 ( z ) 1</td>
<td>5 -1</td>
</tr>
<tr>
<td>Mzs</td>
<td>0 ( z ) -2</td>
<td>1 -2</td>
</tr>
<tr>
<td>Mxd</td>
<td>1 -14</td>
<td>1 14</td>
</tr>
<tr>
<td>Myd</td>
<td>-11 38</td>
<td>-3 38</td>
</tr>
<tr>
<td>Mzd</td>
<td>-4 -4</td>
<td>0 -4</td>
</tr>
<tr>
<td>Mzd</td>
<td>-2 1</td>
<td>5 1</td>
</tr>
</tbody>
</table>
mm against the single loads corresponding to 1.1 - 3.3 times the API loads. On the other hand, as specified in API, the clearance in the wearing parts of this pump size is normally 0.25 - 0.3 mm in radius. Consequently, there is almost no possibility of a galling between rings for the double-suction centrifugal pumps meeting API standards.

6. Leakage of Mechanical Seals

It has often been experienced that a leakage or a breakage of mechanical seals occurs due to piping loads. This possibility was investigated with double-suction centrifugal pumps. Mechanical seals were assembled on pumps 1 - 4 and a leakage or a breakage of mechanical seals was checked when maximum casing internal pressure in addition to piping loads was applied. The liquid used was clean water at normal temperature. A leakage detector was provided at the mechanical seal end covers. The results show that no leakage from mechanical seals occurred when plural loads of 3 - 7 times the API loads were applied to the pumps.

For double-suction centrifugal pumps, it is assumed from their construction that radial displacements of stuffing boxes and the shaft are almost the same. Relative movements in radial direction between stuffing boxes and shafts were measured by displacement detectors when single loads of 3 - 14 times the API loads were applied to pumps 1 - 4. The results were that in all load conditions, the relative movements were 0.01 mm or less. From these results, it is recognized that in foot-supported double-suction centrifugal pumps, the deformation of mechanical seal parts is so small that no leakage of mechanical seal will occur in the above range of piping loads.

7. Conclusions

Shaft displacement due to casing deformation of foot-supported single-stage double-suction centrifugal pumps was investigated by actual measurements and FEM analysis, and the following results were obtained.

(1) Shaft displacements due to casing deformation are so large that they cannot be neglected for foot-supported pumps. The magnitude of shaft displacements due to the piping loads of various directions are almost the same. This is significantly different from the case of center supported pumps, where direction of load affects magnitude of shaft displacement.

(2) In foot-supported pumps, shaft displacements in z direction are the most predominant.

(3) Against API loads, larger size pumps show a larger shaft displacement. These displacements are approximately in proportion to the suction nozzle diameters when the $N_c$ and maximum casing internal pressure are the same.

(4) The influence of pump size and casing thickness on shaft displacements varies greatly depending on the directions of piping forces and moments.

(5) The FEM results of shaft displacements by FM agreed well with actually measured results.

(6) As the casing wearing rings and rotor elements deform uniformly under piping loads and a large local deformation does not occur, there are less possibilities of a galling between these two.

(7) In foot-supported double-suction centrifugal pumps, there is little possibility of a leakage or damage of mechanical seals due to piping loads.

References


