Computational study of the effect of pump mounting to foundation compliance on pump natural frequencies

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Abstract. This article deals with the development of more accurate models of high-performance pump machines, which are mounted onto the foundation by means of compliant supports. These models demonstrate dependence between support foundation compliance and dynamics of pump elements such as rotor, pump casing and casing of pump bearings. After application of these approaches, the method of vibrational calculation with variable values of support compliance taken into account is provided. This task has been performed by using the multifunctional program complex ANSYS Workbench. In particular, the investigation employs the Modal Analysis subprogram, which implements the finite element method at the tasks of vibration analysis of constructions. The degree of effect of pump baseplate compliance on the rotor dynamics behaviour has been determined. Data charts of dependencies of rotor, pump casing and bearing housings natural frequencies on pump mounting to foundation compliance have been obtained.

1. Problem description

Centrifugal pump units are desirable machines utilized by various organizations in any engineering industry. At the same time, it is equally important to mention the current trend within the pump industry of ever-increasing performance characteristics of pump units. The best strategy for achieving characteristic increases, especially for head, is acceleration of the speed of rotation of the rotor. The rotating rotor, in and of itself, is the primary vibration source. Therefore, an increase in its speed of rotation, in turn, leads to rise in forces generated by the rotor, particularly the amount of unbalance, which is proportional to rotational speed squared, hydrodynamic force, various mechanical forces etc.

Vibration level is one of the most important factors for reliable performance of the pump units because excessive vibrational activity leads to, if not accidents, then at least accelerated wear of individual machine components and damage as a result.

Currently there are many approaches of determining the vibration behaviour of the rotor itself [1-3] and its balancing on special test stands [4, 5]. Thereby, after installation of the rotor into casing the latter receives periodical forced loads through the bearings. The cause of these loads is not only unbalance, but also such excitation forces as blade, inducer, electromagnetic forces etc. The casing, in turn, passes these loads through mount components of the unit to the foundation and building structures, which initiates noise and vibration in those structures and could result in local resonance which is again impacts the life of these structures.
While investigating the vibration behavior of the pump unit as a whole, the calculation of the natural frequencies of its casing and rotor must be conducted at the highest priority. Compliance of the pump unit’s base plate has a significant effect on pump casing dynamics [7]. The rotor, bearing housings and pump casing are crucial for analysis of dynamic system of the whole pump unit. Thus the purpose of this study, firstly, is to determine to what extent a change in mount compliance affects the natural frequencies of the pump casing, bearing and rotor, and secondly to describe the nature of this effect.

2. Description of the calculation method for dynamic characteristics of a pump

Object of vibration characteristics research is a pump unit intended to feed water supply from a deaerator into a nuclear power plant unit steam generator [8]. The main pump of the pump unit is a four-stage barrel casing pump that consists of a suction casing, impellers and diffusers.

Analysis of the feed pump unit system was conducted in the multiphysics program complex ANSYS [9], which utilizes the finite elements method (FEM). The major and most vital benefit of said method is that it includes both universal algorithms of solutions for a wide range of engineering problems and high effectiveness of computer-assisted realization.

Provided the equilibrium of model nodes or with the aid of variational methods, methods of residuals that can be applied to the whole finite element model, a global system of equilibrium equations can be formulated for the stress-strain model [10]. In static problems this equation system is presented as a matrix equation:

\[ [K] \{u\} = \{P\} + \{P\} + \{P\} + \{P\}, \]

where \([K]\) – global stiffness matrix of the model; \(\{u\}\) – vector of nodal displacements of the model; \(\{P\}\) – global vector of external nodal forces; \(\{P\}\), \(\{P\}\), \(\{P\}\) – global vectors of internal nodal forces equivalent to distributed surface and body forces, and forces equivalent to initial strain and stress respectively.

In solving dynamics problems, calculation of natural frequencies and their respective mode shapes in particular, the D’Alembert’s principle is employed [11]. Thus inertial forces are added into the calculation as well. Because these forces are expressed through accelerations that are second-order derivatives of displacements, matrix expression (1) turns into a global differential motion equation:

\[ [M] \{\ddot{u}\} + [C] \{\dot{u}\} + [K] \{u\} = \{F\}, \]

where \(\{u\}\), \(\{u\}\), \(\{u\}\) – vectors of displacements, velocities and accelerations respectively; \([M]\), \([C]\), \([K]\) – global mass, damping and stiffness matrices of the design; \(\{F\}\) – vector of equivalent nodal forces.

While applying FEM in ANSYS Workbench, the system is assumed to be linear. That is how the program avoids all types of non-linear conditions, such as non-linear material properties, boundary contact conditions or finite displacements [12]. It is also assumed that the damping and external forces are zero. Therefore, the vibration equation (2) in matrix expression is as follows:

\[ [M] \{\ddot{u}\} + [K] \{u\} = 0. \]

With the modal analysis, which is performed after the preliminary calculation of the stressed structure, taken into account, equation (3) is changed to:

\[ [M] \{ \ddot{\mathbf{s}} \} + [K] \{ u \} = 0 \]

where \([K]\) – linear stiffness matrix; \([K]\) – so called geometric stiffness matrix, which is obtained from the prestress tensor and the non-linear part of the strain tensor [13, 14].
For any system of linear vibration, a vibratory displacement is traditionally [11] rendered as a harmonic function:

\[ u_i = \{\phi\}_i \cos(\omega t) \]  

(5)

where \( \{\phi\}_i \) – \( i \)-th mode eigenvector, corresponding to \( i \)-th natural frequency; \( \omega_i \) – \( i \)-th natural frequency. Substitution of (5) into (4) gives us

\[ (-\omega^2_i [M] + [K] + [K]^T)\{\phi\}_i = 0 \]  

(6)

Provided a non-trivial solution is determined, that is \( \{\phi\}_i \neq 0 \), the determinant of the first bracket in (6) should be equal to zero.

\[ \det(-\omega^2_i [M] + [K] + [K]^T) = 0 \]  

(7)

Therefore, expression (7) works out to a generalized problem of eigenvalues, which is solved by \( n \) pairs of natural frequencies \( \omega^2_i \) and eigenvectors \( \{\phi\}_i \), where \( n \) – the number of structural degrees of freedom.

3. Description of the computational model of a pump

For the purpose of this article it was necessary to determine the natural frequencies of the design and their respective modes. Modal analysis was employed to calculate natural frequencies and their respective modes.

A 3-D model of the pump was imported into the ANSYS complex. For further computational analysis of dynamic characteristics of the pump, a finite element model was generated on the basis of geometric model.

![Figure 1. Finite element computational model of a pump with compliant spring elements instead of stiff support elements.](image)

The finite element model of the feed pump consists of 450,000 tetrahedral elements. This type of element has four nodes, and an equilibrium equation is created for each of this node. The system of equilibrium equations of one element is local. Adding all the local systems of each element to a single one, we obtain a global system of design equations, on the basis of which ANSYS performs user-
defined analysis. Since we use modal analysis in this study, the applicable systems of equations are used, as described in Section 2.

The model shown in Figure 1 represents a system, where the rotor is supported by two plain bearings and several gap seals, which are marked in ANSYS as a “Bearing” element. These elements are simulating a connection between the rotor and the rotating assembly. The rotating assembly is made of cartridge type. It is rigidly bounded with a barrel mechanically. The pump belongs to pump type BB5 according to [6]. The barrel is supported by four feet on the base plate, which in turn is fastened to the foundation with anchor bolts and grouted in concrete. In the given article the base plate and foundation are replaced by springs with variable compliance. There are three springs per foot.

In the computational model, a bonded contact was specified at the contact points of the casing parts. The sustainability of this solution is explained by the fact that first, the pump must be leak tight. This means that the contact spot between the barrel and its suction and delivery covers must always be constant. Secondly, in this way we can optimize the calculation process, thus simplifying the calculation model and thus reduce the machine calculation time.

The main characteristics that affect the accuracy of the results of this computational analysis are: accuracy of the geometry of the feed pump, i.e. adequately inertia of the rotor, bearing housings and pump casing (stator), as well as hydrodynamic characteristics of gap seals and plain bearings. Calculations of gap seals and plain bearings are given in [7].

4. Calculation of dynamic characteristics, plotting the dependence graphs of natural frequencies on pump mounting compliance

This investigation is based on findings of a previous study [7], so the computational model of a feed pump is identical to the one given there. The number of computational points for pump mounting compliance has been increased in range from 5∙10^{-10} m/N to 1∙10^{-8} m/N.

The value of pump mounting springs compliance is set to 5∙10^{-10} m/N. This value is chosen because compliance below this value is negligible for dynamics of pump elements [7]. As a result of the computational analysis the dynamic characteristics of the pump have been obtained. Figures 2-4 show the first three modes of the pump rotor.

![Figure 2](image-url)  
*Figure 2. The first rotor mode at pump mounting to foundation compliance of 5·10^{-10} m/N.*
Figure 3. The second rotor mode at pump mounting to foundation compliance of $5 \cdot 10^{-10}$ m/N.

Figure 4. The third rotor mode at pump mounting to foundation compliance of $5 \cdot 10^{-10}$ m/N.

At this point pump modes are as follows (Figures 5-7).
Figure 5. The first and second rotor modes of feed pump as a dynamic system at pump mounting to foundation compliance of $5 \times 10^{-10}$ m/N.

Figure 6. The third and fourth rotor modes of feed pump as a dynamic system at pump mounting to foundation compliance of $5 \times 10^{-10}$ m/N.

Figure 7. The fifth and sixth rotor modes of feed pump as a dynamic system at pump mounting to foundation compliance of $5 \times 10^{-10}$ m/N.

Figure 8 shows the dependence of natural frequencies of a feed pump (as a whole dynamic system) on mounting compliance.
Since the central focus of the previous study [7] was on the ordinal number of natural frequency, each line in a graph similar to Figure 8 was matched with No. of the natural frequency of the feed pump mode. In this study, the lines of Figure 8 are not matched with No. but with the pump part which vibrates with specific frequency depend on compliance of barrel support. Lines Nos 1, 2 and 3 represent first, second and third rotor mode shapes. Lines Nos 4 and 5 are vibrations of the bearing housing, vertical and horizontal respectively. Lines Nos 6 and 7 represent vibration of the pump casing in a vertical plane, lines Nos 8 and 9 – horizontal plane. Axial vibration of the casing is shown as both linear at line No. 10 and torsional at line No. 11.

It can also be observed that lines such as 8 and 1, 6 and 8, 6 and 10 and some others are overlapping at some points. Physical substance of such data is that at some casing mounting compliance values the elements in the dynamic system are vibrating in superposition with the same natural frequency.

Figure 8 shows that the change in the compliance of the pump mounting to the foundation has minimal effect on the dynamics of the rotor, but we can see little effect at certain compliance when there is a coincidence with the natural frequencies of the pump casing. At the same time, the natural frequencies of the bearing housings decrease if the compliance of pump mounting to the foundation increases to a value of $0.5 \cdot 10^{-8}$ m/N, and then the compliance of the pump mounting to the foundation does not effect on the natural frequencies of the bearing housings. It should be noted that all the natural frequencies of the pump casing tend to zero when compatibility of the mounting to the foundation increases.

5. Findings and conclusions
In this article an analysis of dynamic characteristics of a feed pump was conducted by the finite elements method. As a result of performed calculations the values of natural frequencies and modes of vibration of a pump on compliant bases have been obtained.
It is established that when compliance of the pump mounting to foundation changes, the natural frequencies of the rotor almost do not change within the covered range of compliance of pump mounting to the foundation.

However, the natural frequencies of the bearing housings change only in a certain range of compliance of the pump mounting to the foundation and reduce the natural frequency values not as drastically as the pump casing.

The natural frequencies of the pump casing are unchanged up to a certain value of the compliance of the pump mounting to foundation. But if the value of the compliance continues to increase, the values of the natural frequencies of the pump casing approach zero. In other words, there is an almost complete dependence of the natural frequencies of the pump casing on the pump mounting to the foundation.

Based on the results of the study, we can make a hypothesis. The hypothesis is as follows that there is a certain zone within which the compliance effect on the pump natural frequencies and this zone is shifted to the left if the values of natural frequencies increase.

Therefore, the main conclusion of this study is that the calculations of the natural frequencies of the dynamic system must be performed for the pump as a whole (including the rotor). And when calculating the dynamics of the rotor, it is not necessary to take into account other elements of the pump.

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