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Structural Re-design of Seawater Pump Impeller Shaft
해수펌프 임펠러 샤프트의 구조 재설계

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Abstract : Critical response of seawater pump impeller shaft structure to various exciting loads is a fundamental factor in re-designing of the structure after its functional failure. In this paper, a typical case of the shaft structure's failure is investigated for re-designing purposes. Failure causes of interest are excessive bending moment, fatigue loads and dynamic resonance due to relevant motor rotation and unbalancing of the rotation loads. Static analyses of shaft structure under the conditions of concerned loads are carried out, followed by a dynamic investigation of the effects of resonance between the shaft and the motor on the structure. The relevant structural analyses are carried out using the Finite Element Methods combined with ANSYS code. Based on these, the primary cause for the shaft's structural failure is obtained. It is found that the change of the bending stiffness of the shaft is the primary concern in the re-designing process. A guideline for the re-design process of the seawater pump shaft structure is established, and a re-design scheme of the structure is proposed.

Keywords : impeller shaft, stress concentration, re-design, unbalancing loads, resonance

요 지 : 해수 임펠러샤프트의 각종 하중에 의한 파단과 이를 개선하기위한 재설계는 정적, 동적 해석을 통한 원인분석과 유한요소법을 이용하여 효과적으로 수행할 수 있다. 본 논문에서는 전형적인 임펠러 샤프트의 파손에 대한 원인 분석을 수행하고 관련된 재설계기법을 제시하였다. 일차적으로 정적구조해석을 수행하였고 다음으로 구조물의 외력과의 공진문제를 포함한 동적해석을 수행하였다. 구조해석은 ANSYS코드를 사용하였으며, 결과적으로 파단원인을 찾아 분석하였다. 주된 파단원인은 과도한 굽힘모멘트의 발생과 응력집중, 구조물의 외력과의 공진에 의한 것으로 분석되었다. 해수 임펠러샤프트의 파단과 관련된 재설계기법의 이론적 배경을 정립하였으며, 재설계기법의 적용성과 정적, 동적 샤프트 재설계에 대한 유용성을 제시하였다

핵심용어 : 임펠러샤프트, 응력집중, 재설계, 비균형하중, 공진

1. Introduction

A typical failure phenomenon of a seawater pump shaft structure is observed because of an excessive bending moment, fatigue loads and dynamic resonance due to relevant motor rotation and unbalancing of the rotation loads. A method of investigating the reason for failure is not well-defined, and providing an adequate method for further re-design of the original structure remains difficult. To solve this problem, static analysis of the structure with the loads of concern, as well as dynamic analysis accounting for the effect of resonance between the shaft and the motor are necessary.

Structural re-design based on the change of the mass and stiffness of the original structure have been introduced by Stetson(1981). For structures with poor dynamic response, excitation forces can be minimized or a dynamic re-design can be performed. As change or reduction of the exciting loads is impossible in this case, it is necessary to re-design the structure with such limitations in mind.

Although these re-design approaches using stiffness changes are very simple overall, these alternatives are hardly applicable in this case. Finite Element analysis of the relevant structure is needed to assess the stress concentration phenomenon, as well as to provide the magnitude of stress concentration factor. Grinding of the shaft surface may be the optimal choice, but it requires a considerable cost, making this approach unrealistic as well.

Dynamic re-design problem has been studied and several typical procedures have been proposed by Sandstrom (1982), Hoff(1986). A penalty function method with a minimum weight or minimum mass condition as objective

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functions and a set of residual force errors as the penalty term was employed by Kim(1983). Cho formulated the problem using constraints components in the nonlinear dynamic equilibrium perturbation equation(1989, 1997). Investigation to the shaft failure in this paper, however, shows that such re-design processes are not applicable due to the various constraints imposed in the manufacturing process. The re-design associated with this structure is the simple change of the shaft diameter to avoid the resonances, which allows the problem to be only confined to the re-design of the proper shaft diameter change. If this approach does not work properly, the alternative is changing the exciting frequency.

2. Mathematical Development for Re-designing Procedure

2.1 Specifications on Structural Changes

Structural re-design problem can be interpreted using an equilibrium equation of the structure. The equilibrium equation without damping term, for the structural analysis can be expressed as:

$$[m]\{\ddot{x}\}+[k]\{x\} = \{F(t)\} \tag{1}$$

where $[k]$, $[m]$ are the stiffness and mass matrices of the structures. Changes in the system give a new solution to the system. We can give the structural changes $[\Delta k]$, $[\Delta m]$ to the original structures stiffness and mass matrix$[k]$, $[m]$. Relationships between the baseline system and the objective system can be defined in terms of perturbation of the baseline system.

$$\begin{aligned} [k'] &= [k]+[\Delta k] \\ [m'] &= [m]+[\Delta m] \\ [\phi'] &= [\phi]+[\Delta\phi] \\ [\lambda'] &= [\lambda]+[\Delta\lambda] \end{aligned} \tag{2}$$

where $[\phi]$, $[\lambda]$ are the eigenvector and eigenvalues of the system.

The equilibrium equation for the changed structure can be expressed as:

$$[[m]+[\Delta m]]\{\ddot{x}\}+[[k]+[\Delta k]]\{x\} = \{F(t)\} \tag{3}$$

This equation gives the desired structural responses, including displacements and stresses, etc.

A practical interpretation can be given to the structural changes. By decomposing the system changes into l element changes, the structural changes are expressed as:

$$\begin{aligned} [\Delta k] &= \sum_{e=1}^{l}[\Delta k_e] \\ [\Delta m] &= \sum_{e=1}^{l}[\Delta m_e] \end{aligned} \tag{4}$$

Furthermore, each element change can be expressed as a fractional change from the original stiffness and mass as follows:

$$\begin{aligned} [\Delta k_e] &= [k_e]\alpha^k \\ [\Delta m_e] &= [m_e]\alpha^m \end{aligned} \tag{5}$$

where α^k and α^m represents fractional change in the stiffness and the mass of the element, respectively. This way, the structural changes in the system can be expressed in terms of elemental structural changes. In practical industrial level, a practical definition of element changes is necessary. For a common beam element, stiffness of the beam can be the combination of the several stiffness effects.

$$[k_{beam}] = [k_{axial}]+[k_{bending}]+[k_{torsional}] \tag{6}$$

Structural characteristic, in association with axial effects, is the cross-section area; for the bending effects, it is the moment of inertia; for torsional effects, it is torsional stiffness. Changes in elemental mass are related to cross-sectional area changes.

We can express beam element changes in the eigenvalue problem of the structure in terms of element changes:

$$\begin{aligned} &[\Delta k_{beam}]-\omega^2[\Delta m_{beam}] \\ &\quad = [k_{axial}-\omega^2 m_{beam}]\alpha^{area}+[k_{bending}]\alpha^I+[k_{torsional}]\alpha^J \end{aligned} \tag{7}$$

The eigenvalue problem in the dynamic analysis of structure can be expressed by setting the right side of the equations (1),(3) equal to zero. (Choi 2002) For the baseline system, it can be written as:

$$[k][\phi] = [m][\phi][\lambda] \tag{8}$$

If the masses and stiffness are changed, the eigenvalues and eigenvectors also change. We may call this system as objective system in comparison with the baseline system expressed in equation (8). The equilibrium equation for such a perturbed eigen system is:

$$([k]+[\Delta k])[\phi+\Delta\phi] = [m+\Delta m][\phi+\Delta\phi][\lambda+\Delta\lambda] \tag{9}$$

Equations (6),(7),(9) are the basic equations in the re-design process.

2.2 Application to Re-design of Impeller Shaft

As mentioned previously, a penalty function method where the objective function is a minimum weight or minimum mass condition, and the penalty term is a set of residual force errors, can be employed to find re-design characteristics. However, investigation to the shaft failure in this paper shows that these kinds of re-design processes are not applicable, as various constraints are imposed in this case due to the manufacturing process. Equations (6), (7) show that the change of the stiffness in association with bending is the primary factor: failure of the shaft is thought to be due to an excessive stress from the bending of the shaft. The effective structural characteristic in this case is the change in diameter, which consequently results in a change in the area and the moment of inertia. Thus, a very simple re-design parameter is suggested for this structure: a change in shaft diameter allows avoiding the stresses concentration phenomenon, as well as the resonance problem. The changed stiffness and mass provide a new system, and the solution of equation (3) gives the desired stress. This approach overall is based on the concepts of perturbation of the parameters such as stiffness and mass. As a result of this method, the shaft failure problem can be confined to a simple problem of re-designing.

3. Static Failure Causes and Relevant Analyses

3.1 Failure Particulars and Material Cases

Seawater pump shaft failed during the operation and several possible causes are pointed out. A relevant static analysis is carried out to defines causes for the failure. Investigations of the causes of the impeller shaft provide several specific phenomena for observation:

1) Failure of the shaft is confined to the vertical direction only, and this is not a typical case of failure due to the twisting moment that occurs during shaft rotation. (Gere 2009) A typical failure due to the excessive shear stress by the torque would show a 45 degree directional failure (Fig. 1).

2) Position of the observed failure along the shaft length is the point where a sudden diameter change occurs. This point is the maximum bending moment point as well as a drastic shape change point (Fig. 2).

3) The section of the failed shaft shows no wave pattern generally observed due to fatigue propagations. Thus this failure is mainly caused by a shortage of the ultimate strength of the member (Fig. 3).

4) Unbalanced loads due to the eccentricity of the impeller loads are observed, and this may be the another cause of the failure (Fig. 4).

Fig. 1. Failure of the shaft.

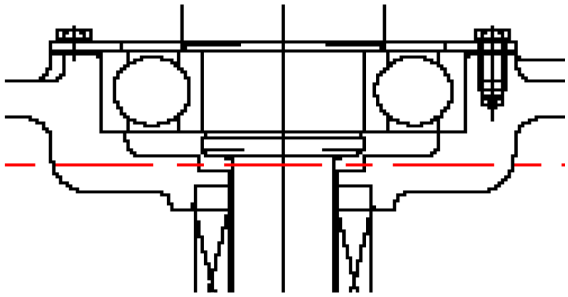

Fig. 2. Position of failure.

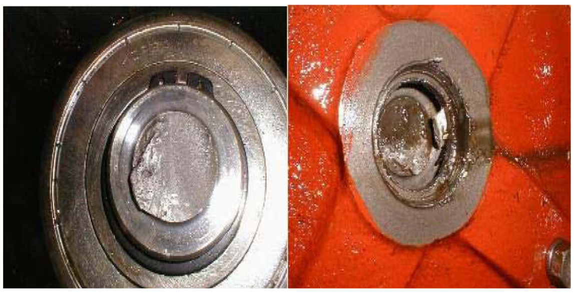

Fig. 3. Cross section of the failed shaft.

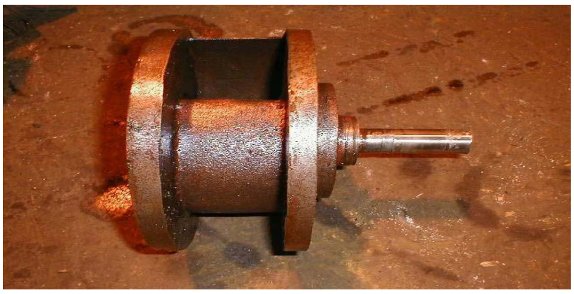

Fig. 4. Impeller shape.

3.2 Relevant Static Analysis

3.2.1 Analysis on Stress Concentration using Solid Element

Finite Element Analysis for the shaft was carried out by using ANSYS Program. This covers static structural analysis, in which qualitative structural behaviors of the structure and relevant stress concentration phenomenon are observed. The element used is a solid element, and the material

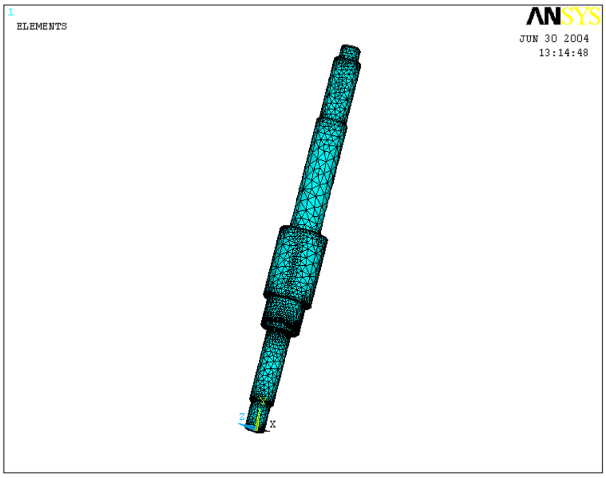

Fig. 5. FEM model of shaft.

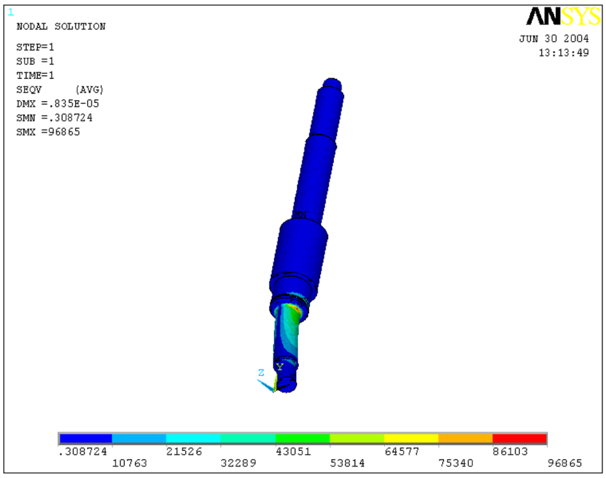

Fig. 6. von Mises stress.

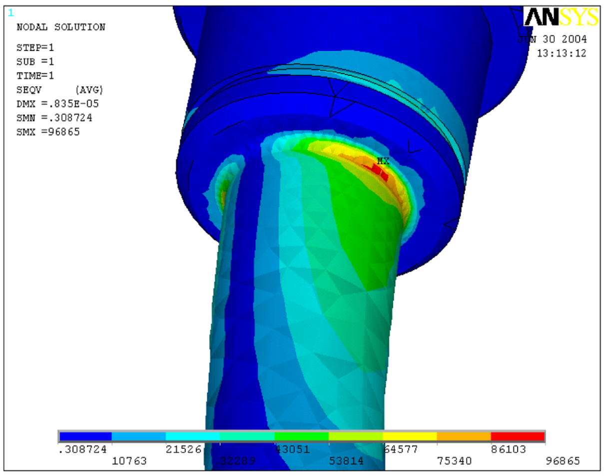

Fig. 7. Details for the part of max stress.

property is from SUS410 steel (Fig. 5). Bearing position is assumed to be supported, loads are simulated by implying the distributed standard value along the contacting points with the impeller. Fig. 6 and Fig. 7 show the stress concentration area in the model as anticipated. The area is closely coincided with the area where a geometrically abrupt change of the diameter is observed. This point is the same point where the red line is indicated in Fig. 2.

3.2.2 Sensitivity Analysis Based on the Re-design Scheme

Based on Equation (6), (7), the first parameter to be considered for the re-design of this shaft is the moment of inertia I in the bending situation. Thus based on this fact, change of the diameter is chosen to be the first step considered in the re-design process. A sensitivity analysis using different shaft diameters is carried out with the same boundary conditions and loads for verification.

Changes of stress at the hot spot with varying diameters are tabulated in Table 1, together with changes of the [$k_{bending}$]. In this case, it is clearly indicated that the decrease of diameter affects the structural behavior of the given structure. It is found that the decrease in diameter is one of the efficient re-design factors in this case. The inverse value of the design parameter I changes with the diameter change, and it is shown in Fig. 8. This curve is well correlated with the stress change curve.

3.3 Relevant Dynamic analysis

3.3.1 Analysis on Natural Frequencies using Beam Model

Dynamic analysis of the shaft was carried out using

Table 1. Change of the stress

Diameter (mm)	Stress (N/m^2)
21	302,949
23	230,592
25	179,559
27	142,540
29	115,036
31	94,176

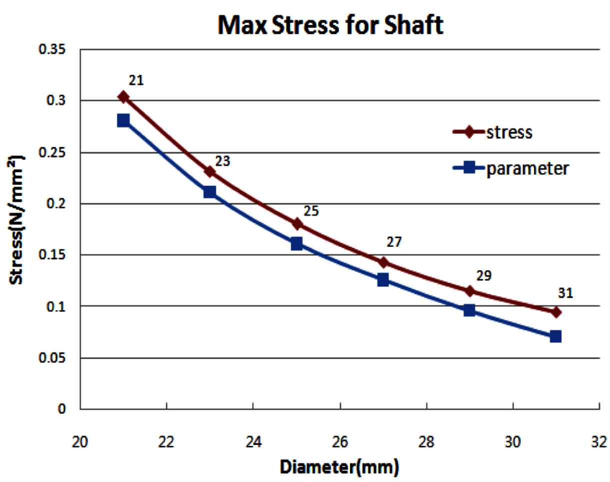

Fig. 8. Change of stress with diameter change.

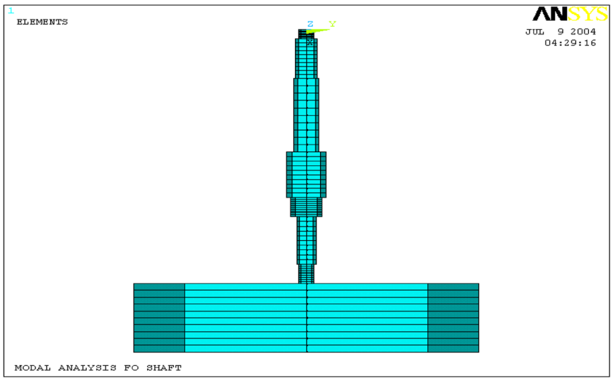

Fig. 9. Shaft-impeller modelling.

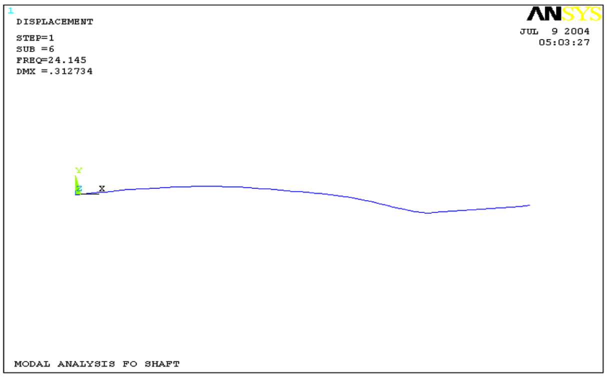

Fig. 10. 6th mode shape of the model.

ANSYS Program. This covers dynamic structural analysis, from which natural frequencies and corresponding mode shapes of the structure are obtained. This analysis gives the possibility of resonance between a natural frequency and exciting frequency from the shaft rotation. The element used is the same solid element with the material property from SUS410 steel (Fig. 9). Fig. 10 shows the mode shapes along with the corresponding natural frequency. Primary concern is the resonance of the structure, and a dynamic structural failure may also occur due to the eccentricity loads by the impeller rotation in resonance condition. The most plausible resonance frequency of the structure is the 6th one, 24.145 Hz, while the exiting frequency due to the shaft rotation is 29.167 Hz. Thus it is necessary to avoid the resonance by changing the relevant structural characteristics.

Equation (9) shows that the change in natural frequency obtained after the change in stiffness and mass of the system. We see that the change in diameter, which is the only possible choice in this case, definitely results in the change in natural frequency as expected. Changing the rate of the 6th natural frequency of the shaft is not drastic compared with the changing rate of the stress as discussed in 3.2.2. Thus, finding the desired natural frequency by

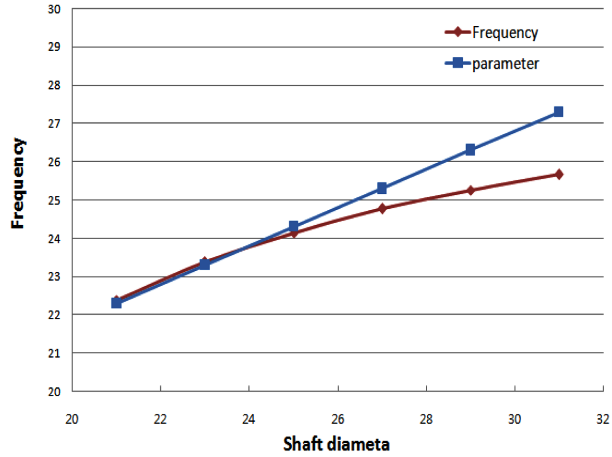

Fig. 11. 6th frequency change with diameter change.

altering one of the structural characteristics, diameter of the shaft, is expected to be difficult (Fig. 11).

3.3.2 Alternative in Dynamic Re-design

For the poor dynamically responded structure, one of the ways to avoid the resonance situation is changing of the excitation frequency that is close in value to the resonant natural frequency of the structure. We may alter the rpm of the shaft motor in specific regions. One can change this to be under the 1080 rpm or over the 3240 rpm, which are in close proximity to the 6th and 7th natural frequency values, respectively.

The design parameter value of root A, that changes in association with the natural frequency is shown in Fig. 11. This curve is relatively well correlated with the frequency changes. It is noted that the frequencies of modes 1,2,3 in Table 2. are those correspond to the rigid body modes of the structure.

Table 2. Frequencies of shaft-impeller model

Mode	Frequency
1	0.0
2	$0.36502{\times}10^{-4}$
3	$0.47177{\times}10^{-4}$
4	4.1965
5	12.076
6	24.145
7	84.588

Table 3. 6th frequency with diameter change

Diameter (mm)	Frequency (Hz)
21	22.389
23	23.368
25	24.145
27	24.766
29	25.265
31	25.664

4. Re-design of Impeller Shaft Based on Static and Dynamic Analyses

Among several ways for structural improvement to avoid the failure of the shaft, the first parameter obtained for re-designing in this case is the moment of inertia I in bending situation. Structural change in response to I in stiffness is the change in diameter square. Based on this fact, change in diameter is chosen to be the first parameter considered in the re-designing process. In this case, it is clearly indicated that the decrease in diameter affects the structural behavior of the given structure. It is found that the decrease in diameter is one of the prominent re-design factors in this case. This re-designing approach is very simple in some sense, however, this can be applicable to other cases without any difficulties. Beside this, maintaining precision of the shaft layer, changing position of the bearing support point also can be alternative methods.

Dynamic re-designing problem has been studied regarding this shaft problem. However, it is shown that such re-design processes are not applicable here due to various constraints imposed in the manufacturing process. The simple re-designing associated with this structure can involve a simple change in stiffness and in mass matrix in association with shaft in order to give the desired natural frequency and to avoid resonance. However, because this approach is proved to be not efficient through structural analysis, the alternative must be considered; that is, the change of the resonant exciting frequency. The most plausible resonance frequency of the structure is the 6th one, 24.145 Hz, and the exciting frequency due to the shaft rotation is 29.167 Hz. Thus, it may be necessary to avoid resonance by changing the relevant environments, rather than changing the corresponding natural frequency.

5. Re-design Procedure Using Structural Characteristics

A scheme for re-designing the impeller shaft is proposed in this section. This can be applicable to similar structures in principle. Structural characteristics regarding behaviors of the corresponding structure are considered first, and the changes in stiffness and mass of the structure can be expressed mathematically with the structural characteristics. The design parameters can be chosen based on the object structure, and the re-design procedure can be carried out consequently. The procedure in relation to the problem of shaft re-design is shown in the flow chart in Fig. 12.

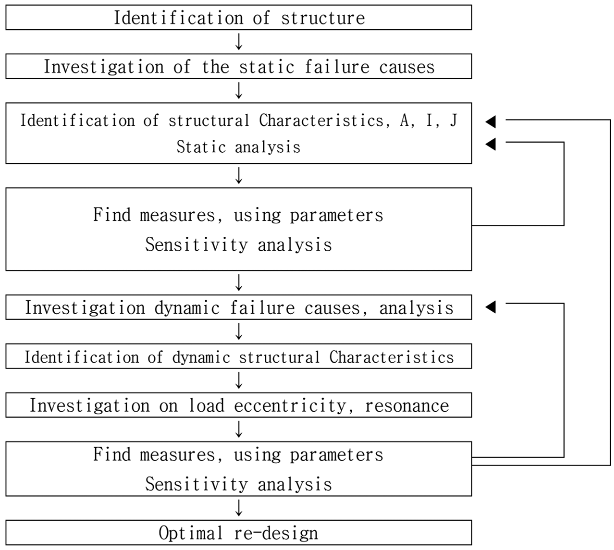

Fig. 12. Flow chart for re-design.

6. Conclusions

The main contribution of this paper is the development of a procedure for re-designing the structural system of an impeller shaft. Here, an investigation of phenomena of typical failure of a shaft structure is carried out. The causes are excessive bending moment, fatigue loads and dynamic resonance due to relevant motor rotation and unbalanced rotation loads. A re-designing method is developed by applying the concept of perturbation of stiffness and mass of the baseline structure. Relationships between the baseline system and the desired system are defined in terms of the fractional changes from the baseline structure. Structural characteristics are defined in terms of the stiffness of the structure and the parameters are categorized. Structural re-design based on the change in mass and stiffness of the original structure is carried out.

In the re-designing problem of impeller shaft, the first parameter is found to be the moment of inertia I in bending. Based on this fact, a degree of change in diameter is chosen, and the re-design process is carried out consequently. A sensitivity analysis is als o carried out for verification. Dynamic re-designing problem is also studied, and several re-designing recommendations are made.

Finally, a guideline for the re-designing process of the shaft structure is established, and a re-design scheme of the structure is established. These are applicable to similar structures in principle.

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