Hydraulic Characteristics and Structural Stability of the Rotor of a Molten-salt Pump

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Abstract

To evaluate the operation stability of the centrifugal pump that transports the high-temperature molten salt, the numerical coupling of flow, thermal and structural fields was implemented. Flow characteristics of the pump were investigated using the computational fluid dynamics technique. The stress, deformation and vibration of the pump rotor were solved using the finite element method. Effects of the flow rate of the medium were considered. The results show that the static pressure increases continuously from the impeller inlet to the volute outlet and this trend is remained with the variation in the flow rate. Along the shaft, temperature decreases gradually from the impeller to the bearings. The maximum Von Mises stress in the impeller arises at the connection between the blade and the impeller shroud and decreases as the flow rate increases. The impeller outer edge is responsible for the largest deformation. The deformation of the shaft attenuates from the impeller to the bearings end. The overall deformation of the rotor tends to be mitigated with the increase in the flow rate. The difference is manifested between dominant natural frequencies of the rotor and the blade passing frequency and its harmonics. At low vibration modes, bending of the rotor is predominant. As the mode order increases, torsional vibration patterns of the pump rotor are evidenced.

Key Words: Centrifugal Pump, Molten Salt, Fluid-thermal-structure Coupling, Static Pressure, Stress, Deformation

1. Introduction

The pump that transports the high-temperature molten salt is usually installed vertically and equipped with a long shaft [1]. The temperature of the molten salt arrives at as high as 540 °C [2]. The application of such a special type of pump has penetrated into chemical industry, nuclear energy utilization and projects of thermal power conversion [3,4]. The pump assumes the function of providing the liquid medium energy to overcome various flow resistance. Featured by high temperature and solid particles, the operation of the molten-salt pump differs considerably from that of common centrifugal pumps [5]. The core part of the pump is the rotor, which is composed of the impeller, the shaft and the other rotating components. As the rotor rotates, hydraulic loads are exerted on the rotor and non-axisymmetric hydraulic loads might incur the vibration of the rotor, impairing the operation stability. Considering the structure of the molten-salt pump, the long shaft is appreciably affected by the variation of the hydraulic loads as it rotates. Therefore, the evaluation of the structural stability is of special significance for the molten-salt pump.

Two aspects, flow and structure, influence to a large extent the operation performance and stability of the molten-salt pump. Nevertheless, practical performance data of the molten-salt pump are difficult to obtain due to harsh application environments. In laboratories, it is difficult to build a loop circulating high-temperature liquid medium. Moreover, both measurement techniques and...
instruments applying to high-temperature liquids require special design. In practice, the measurement of the flows in the molten-salt pump and the structural parameters has rarely been reported heretofore. Under the circumstances, numerical simulation has developed into a useful and feasible tool for the study of the molten-salt pump.

The rotor of the molten-salt pump is exposed to the effects of flow, heat and structure. The method of coupling between multiple physical fields has been attempted in the investigation of pumps [6]. Moreover, elucidating the fluid-structure interaction and vibration modes contributes to the evaluation of the structural performance of the pump [7]. Therefore, hydraulic loads obtained using the computational fluid dynamics (CFD) technique, and the rotor deformation can be analyzed in combination. Besides, the strength of the pump rotor can be quantified, providing a reference for structural optimization [8]. Nevertheless, thermal fluid flows have seldom been considered in the centrifugal pump. Thus far, only the influence of temperature on individual blades has been described [9].

The natural frequency of the pump, which is related to structural properties of the pump is of practical significance. The finite element method (FEM) enables the calculation of the natural frequency [10]. Regarding the impeller machinery, as the excitation frequency approaches the natural frequency of the rotor, high risk of resonance is predictable [11]. Such a fundamental knowledge has been widely recognized in pump industry but the cases as the hydraulic loads are imposed onto the pump rotor have rarely been investigated.

The purpose of the present study is to investigate the operation stability of a centrifugal pump that transports high-temperature molten salt. Considering both flow and structure, the coupling between flow, temperature and structure was implemented using the commercial software ANSYS. The thermal fluid flows in the pump were simulated using unsteady CFD techniques and instantaneous flow parameter distributions were described. Hydraulic forces imposed on the pump rotor and the temperature distributions were analyzed. Hence, the finite element method was used to simulate stress distributions in the rotor with the input conditions transplanted from the CFD results. Stress and deformation of the rotor were quantified. With the modal analysis, natural frequencies and vibration amplitudes of the rotor were obtained. For the simulations, effects of the flow rate were considered. It is expected to deepen the understanding of the flow and structural features of the molten-salt pump.

2. Molten-salt Pump Model

Both the schematic and the prototype of the centrifugal pump transporting molten salt are presented in Figure 1. During its operation, the pump is installed vertically in a tank which is filled with molten salt. Only the pump components above the supporting plate are visible. The impeller is completely submerged in the molten salt and the medium enters the pump from the bottom, ensuring continuous flow in the pump. Due to temperature difference, heat is transferred upwards along the pump shaft. It is perceivable that the two pairs of bearings at the upper part of the rotor suffer from large radial hydraulic forces. The lower end of the shaft is connected with the impeller; moreover, the only support for the rotor is provided by the two pairs of bearings. The deviation of the rotor from the central axis can trigger the interference between the inner race and the outer ring of the bearings [12]. As

Figure 1. The centrifugal pump transporting molten salt.
the temperature of the bearings is high, such a risk will be improved. In this case, the cooling system for the bearings can lend its important support [13]. At present, primary factors that influence the operation of the rotor have not been elucidated.

3. Numerical Set-up

3.1 Computational Models

The computation model constructed in the present work is parallel to the prototype of a centrifugal molten-salt pump, which operates at the design flow rate, \( q_V \), of 200 m\(^3\)/h, the rotational speed, \( n \), of 1450 r/min, and then yields a pump head, \( H \), of 70.3 m. The pump efficiency under the design condition reaches 75%. The pump impeller has six blades. Regarding the fluid-flow simulation, the computational domain is composed of the inlet, impeller, volute and outlet parts.

3.2 Flow Simulation Preparations

3.2.1 Liquid Medium Properties

The molten salt with dual elements, namely NaNO\(_3\) of 60% and KNO\(_3\) the rest, was chosen as the medium for the flow simulation. Such a medium has been used in the solar photovoltaic power generation [14]. Physical properties of the molten salt are listed in Table 1.

3.2.2 Governing Equations and Turbulence Model

Considering the medium properties, the molten salt was assumed incompressible; then the governing equations take the following form [15]:

\[
\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0
\]  

(1)

\[
\frac{\partial (\rho \mathbf{u})}{\partial t} + \nabla \cdot (\rho \mathbf{u} \otimes \mathbf{u}) = -\nabla p + \nabla \cdot \mathbf{\tau} + \rho \mathbf{g} + \mathbf{F}_i
\]  

(2)

\[
\frac{\partial (\rho T)}{\partial t} + \nabla \cdot (\rho \mathbf{u} T) = \nabla \cdot \left( \frac{k}{c_p} \nabla T \right) + S_T
\]  

(3)

The turbulence model was required to accomplish the closure of the governing equations. Hitherto, no generalizable conclusions have been established for the suitability of any turbulence model. Here, the re-normalization group (RNG) \( k-\varepsilon \) model was selected. The RNG \( k-\varepsilon \) turbulence model proved to be feasible for the simulation of flows in impeller passages [16,17]. Moreover, effects of the blade curvature on flows bounded by the blades can be described with the RNG \( k-\varepsilon \) turbulence model.

3.2.3 Boundary Conditions

The velocity inlet boundary condition was set at the inlet of the whole computational domain. The volume flow rate was related to the inlet velocity and could thereby be adjusted. Flow rates of 0.8 \( q_V \), 1.0 \( q_V \), and 1.2 \( q_V \) were selected. Because exact static pressure as well as velocity distributions at the outlet of the computational domain was unknown prior to the simulation, the outflow boundary condition was set at the outlet section. Except the static pressure, all the other parameters were associated with zero normal gradients at the outlet section of the computational domain. A turbulence intensity of 5% was defined at the inlet of the computational domain and no-slip boundary conditions were imposed on all solid walls. A surface roughness of 0.025 mm was predefined for all surfaces wetted by the molten salt, as complied with practical situation of the prototype pump. Near-wall regions were treated with the scalable wall function, which is more effective relative to the standard wall function. The scalable wall function accommodated various \( y^+ \) values and facilitated the grid refinement. During the unsteady flow simulation, the time intervals between neighboring simulation steps were set equal to the time span with which the impeller rotated for one degree. Ten full cycles of impeller rotation were covered in the simulation and the data associated with the tenth cycle were processed.

3.3 Structural Simulation Scheme

3.3.1 Structural analysis

Derived from the Newton’s second law, the conser-

<table>
<thead>
<tr>
<th>Medium</th>
<th>Density (kg/m(^3))</th>
<th>Dynamic viscosity (Pa*s)</th>
<th>Temperature (°C)</th>
<th>Specific heat capacity J/(kg*K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Molten salt</td>
<td>1804</td>
<td>0.00147</td>
<td>450</td>
<td>1520</td>
</tr>
</tbody>
</table>
viation equations for the solid part are expressed as:

\[ \rho \ddot{d}_s = \nabla \cdot \sigma_s + f_s, \]

where \( \rho \) denotes the density of the solid part, \( \sigma_s \) is the Cauchy stress tensor, \( f_s \) is the volumetric force vector, and \( \ddot{d}_s \) is the acceleration vector.

### 3.3.2 Thermal Analysis

The static thermal analysis was conducted in the present simulation and the result of the thermal analysis was associated with the thermal stress. The sum of the heat streamed into the system and the heat generated in the system equals the heat flowing out of the system. The governing equation of the static thermal analysis is given by:

\[ [K(T)]\{T\} = \{Q(T)\} \]

where \([K(T)]\) is the heat transfer matrix, \(\{T\}\) is the temperature vector of the node and \(\{Q(T)\}\) is the heat flow rate vector of the node.

### 3.3.3 Stress Analysis

According to the Von Mises yield criterion, the yield failure of a material is highly dependent of the dilatational strain energy density. Here, the stress analysis was performed based on this principle. The Von Mises stress, complying with the well-known fourth strength theorem, is given as:

\[ \sigma_v = \sqrt{\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2 / 2} \]

### 3.3.4 Modal Analysis

Modal characteristics of the pump rotor serve as an important reference for the vibration analysis, failure diagnosis and optimal design of the rotor. Based on the Newton’s second law and mechanical vibration principles, the modal equation of a structural system with multi-degrees of freedom is given as:

\[ [M] \ddot{\{x\}} + [C] \dot{\{x\}} + [K] \{x\} = \{F(t)\} \]

where \([M]\) is the mass matrix, \([C]\) is the damping matrix, \([K]\) is the stiffness matrix, \(\ddot{\{x\}}\) is the acceleration vector, \(\dot{\{x\}}\) is the velocity vector, \(\{x\}\) is the displacement vector, and \(\{F(t)\}\) is the force vector.

The excitation force and effects of structural damping on the structural mode and vibration pattern were neglected; therefore, Eq. (7) is simplified as:

\[ [M] \ddot{\{x\}} + \{K\} \{x\} = 0 \]

Structural vibration is generally signified by harmonic vibration; thus free modes of the rotor can be solved with the characteristic equations.

### 3.3.5 Loads and Constraints

The pump rotor was supported at the two positions where the bearings were mounted. Therefore, the constraints of the rotor were imposed at two cylindrical surface segments, as shown in Figure 2.

The impeller and the shaft were exposed to the gravitational force, centrifugal force, thermal stress and hydraulic loads. The centrifugal force was produced in the simulation as the angular speed was specified for the pump rotor. The hydraulic loads were obtained from CFD results. Both the hydraulic and thermal loads are transferred to the solid parts via the interface between the liquid and solid parts; then the coupling could be implemented.

### 3.4 Grid Independence Examination

The commercial code ICEM CFD was used to discretize the computational domain of the molten-salt pump with unstructured grids; moreover, local areas were refined to improve the spatial resolution of the simulation. To reduce the uncertainty caused by the grid number, a grid independence examination was performed at 1.0 \( q_y \) using five sets of grids with different grid numbers. Based on numerically obtained velocity and static pressure at the pump inlet and outlet, the pump head was calculated. With the torque generated in the impeller, the pump efficiency was obtained. The results are displayed in Figure 3. As the grid number exceeds 2.7 \( \times 10^6 \), the relative deviation between the neighboring results was less than 3%. Therefore, the grid scheme with a grid number of

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**Figure 2.** Loads and constraints.
2.706 × 10^6 was finally selected in subsequent simulations. The grid scheme used in subsequent simulations is illustrated in Figure 4.

To illustrate the physical reliability of the numerical scheme, a comparison between numerical and experimental results of the pump considered was performed. In Figure 5, two performance quantities, the pump head and pump efficiency, were used to illuminate the comparison. As is clearly evident, for the pump head or pump efficiency, numerical and experimental results are in good accordance. Numerical results are consistently higher than their counterparts but the deviation between the two results is small. Relatively large deviation arises at low flow rates; as qV increases, the two results tend to approach each other. This is reasonable since the volumetric loss was neglected in the numerical scheme. Even so, for the pump head and pump efficiency, the maximum deviations in Figure 5 are less than 5% and 3%, respectively, providing a sound proof for high fidelity of the numerical results.

4. Results and Discussion

4.1 Static Pressure Distribution

Cross-sectional static pressure distributions at 0.8 qV, 1.0 qV and 1.2 qV are displayed in Figure 6. It is seen that the three pressure distributions are analogous. In practice, flow rate is a critical parameter and the change in flow rate might disturb the normal operation of the whole process [20]. In Figure 6, the flow quality is not impaired by the variation of flow rate, demonstrating good flow-rate adaptability. The static pressure increases gradually from the impeller inlet to the outlet and the symmetry of static pressure distribution over individual blades is well maintained, which agrees with previously obtained re-
sults [18]. The maximum pressure is attained near the volute tongue instead of at the pump outlet. This is attributed to a sharp increase in hydraulic loss owing to high viscosity, which gives rise to a rapid decrease in the static pressure. It is noticeable that local flows surrounding the impeller are uneven in circumferential direction, as is related to two factors; one is the cross-sectional area of the spiral volute is not identical, the other one lies in the volute tongue, which hinders the circumferential motion of the molten salt.

To clarify flow patterns near the volute tongue, local flows are focused and illustrated in Figure 7, where $\theta_0$ denotes an initial rotation angle. As the blades rotate periodically through the volute tongue, local pressure distribution alters transiently. Nevertheless, variation of the flow occurs exclusively within a small area and overall flow patterns in the flow passage remain unaffected. Meanwhile, the periodic impeller-volute interaction induces the variation in the radial hydraulic force acting on the impeller. Considering the instantaneous flow patterns exhibited in Figure 7, a dramatic change in the radial hydraulic force exerted on the impeller is not expected. As the impeller blades sweep through the volute tongue, high static pressure at the blade outlet is well remained, which is in sensitive to the interaction between the blade and the volute tongue.

4.2 Temperature Distribution

The temperature distribution in the pump rotor is illustrated in Figure 8. Since the impeller is completely submerged in the molten salt, it is reasonable that the impeller is dominated by the temperature equivalent to that of the molten salt. In the impeller hub, since the heat is transferred towards the adjacent shaft, which bears relatively low temperature, local temperature gradients are remarkably high. In this case, heat conduction proves to be the major means of heat transfer at the conjunction between the impeller and the shaft [19].

The temperature distribution along the pump shaft is shown in Figure 9. It is observable that temperature varies continuously from one end to the other end. Moreover, temperature arrives at its maximum at the connection of the pump shaft and the impeller. As the axial distance from this position is augmented, temperature decreases gradually. With the concentration of high temperature at the lower end of the shaft, high thermal stress might be induced in the shaft. Therefore, the effects of temperature deserve a comprehensive consideration during the design of the molten-salt pump.

4.3 Stress Distribution

Distributions of the Von Mises stress at different flow rates are displayed in Figure 10. The similarities among the three distributions are explicit, although the maximum Von Mises stress decreases with increasing flow rate. Furthermore, the maximum Von Mises stress

![Figure 7. Instantaneous static pressure distributions at 1.0 $q_V$.](image)

![Figure 8. Temperature distribution over the impeller.](image)

![Figure 9. Temperature distribution over the pump shaft.](image)
arises at the connection between the blade inlet and the front shroud, as is shared by the three flow-rate conditions. At this position, the intersection angle between the two curved surfaces is relatively small, so high risk of damage due to high stress is expected. Similar situation has been found experimentally in the vertically installed pump-turbine [20]. Therefore, an optimal design of the impeller geometry is necessitated.

Regarding the two impeller shrouds, the distributions of Von Mises stress in the circumferential direction exhibit clear periodicity. Corresponding to each blade passage, there are two noticeable stress distribution zones. Considering the pressure distributions displayed in Figures 6 and 7, the hydraulic loads exerted on the impeller shrouds are uneven. Meanwhile, velocity at the middle part of the blade passage is high, as depends on wall effects; thus the lowest hydraulic pressure is caused at the middle part of the blade passage.

Distributions of the Von Mises stress along the shaft for the three flow rates are presented in Figure 11. A joint action of radial hydraulic loads and the thermal stress yields high Von Mises stress, which is seen at the shaft segment between the impeller and the lower bearings. Meanwhile, at the shaft shoulder, stress concentration is reinforced. On the whole, high flow rate promotes the uniformity of the distribution of the Von Mises stress. Additionally, the maximum Von Mises stress decreases as the flow rate increases.

4.4 Deformation of the Rotor

Distributions of the deformation magnitudes in the impeller are illustrated in Figure 12. These instantaneous distributions were constructed based on unsteady simulation results. As can be seen, the entire impeller is characterized by the radial displacement towards one single direction, which is determined by the resultant radial hydraulic force acting on the impeller. The deformation magnitude increases continuously along a single direction. The distribution pattern is common for the three flow rates. It is observable that the maximum deformation magnitude reduces with increasing flow rate.
At 0.8 \( q_V \), the maximum deformation magnitude of 12.2 mm is attained and it occurs at the impeller outer edge. Such a magnitude is rather small for the pump rotor considered. Meanwhile, the deformation of the impeller blades is relatively weak. In view of the operation parameters of the pump, small blade width should be selected; moreover, the blades are fixed by the front and rear shrouds. Therefore, the deformation of the impeller falls within a safe range for all the three flow rates.

In the same manner, distributions of deformation magnitudes over the shaft for different flow rates are exhibited in Figure 13. It is seen that the shaft is considerably distorted, which is ascribed to the radial hydraulic forces acting on the impeller. The maximum deformation is seen at the connection between the impeller and the pump shaft. Then deformation attenuates towards the lower bearings. Similar to that indicated in Figure 12, the maximum deformation magnitude varies inversely with the flow rate.

### 4.5 Variations of Stress and Deformation with the Impeller Rotation

Flow patterns in the pump passages vary periodically with the rotation of the impeller and the hydraulic loads imposed on the rotor fluctuate accordingly. Nevertheless, the response of the solid parts to the variation in the flow pattern is not that fast due to the differences in physical properties. With the coupling strategy, the response of the rotor to instantaneous pressure distributions is described. Here, six rotation moments, temporally even in the rotation of a single blade passage, were selected. Correspondingly, stress and deformation associated with the rotor at the six rotation moments were calculated. As a reference, the impeller position at the initial moment is sketched in Figure 14.

Variations of the maximum Von Mises stress in the impeller and the shaft with the rotation of the impeller are diagramed in Figure 15. As the blades sweep through the volute tongue, the maximum Von Mises stress in the impeller varies accordingly. The design flow rate is responsible for the smoothest variation of the maximum Von

![Figure 12. Distributions of deformation magnitudes in the impeller.](image)

![Figure 13. Deformation magnitude distribution along the shaft.](image)
Mises stress, as is shared by the impeller and the shaft. Meanwhile, at $1.2q_V$, overall V on Mises stress reaches its minimum. It is extrapolated that the tangential component of the hydraulic force increases with the flow rate and then counteracts a portion of the centrifugal force [21]. In Figure 15(a), fluctuations of the maximum V on Mises stress with the rotation of the rotor are remarkable except at the design flow rate. This is dependent of the impeller geometry and the flow patterns in the impeller passage. Although the impeller can be assumed as a disc-type object in view of its mechanical characteristics, the mass distribution in the impeller is uneven. As the liquid medium passes through the impeller, the position where the maximum V on Mises stress is produced varies with the rotation of the impeller. At the design flow rate, flow patterns are smooth in the impeller passage, so the magnitude of the maximum V on Mises stress changes slightly with the rotation angle. The off-design conditions should be avoided in consideration of the intense fluctuations of the Von Mises stress. As a matter of fact, stress fluctuations impair the fatigue life of the components and pose a threat to the normal operation of the whole rotor.

Time-dependent variations of deformations in the impeller and the shaft are plotted in Figure 16. At a given flow rate, the highest deformation emerges at $0.8q_V$, as is shared by the impeller and the shaft. Meanwhile, the flow rate of $1.2q_V$ is associated with the most alleviated deformation, which is compatible with the situation shown in Figure 15. Nevertheless, the overall deformation magnitude of the shaft is slightly smaller than that of the impeller. It is noteworthy that the rotation angle of $50^\circ$ is associated with the peak magnitude of the maximum deformation. At this position, the blade just passes through the volute tongue and the interference between the two solid parts is considerably severe.

### 4.6 Modal Analysis

From experimental aspect, the modal analysis can be performed based on the data of impact test [22]. Nevertheless, for the molten-salt pump considered in the present work, it is impractical to implement such a strategy. Instead, the modal performance of the rotor was obtained based on FEM results. As the rotor rotates, the natural frequency of the rotor might be influenced by the centrifugal force; thus the inspection of the natural frequency of the rotor under conditions with and without prestress was performed.

In Figure 17, a comparison of the natural frequency and the vibration amplitude of the rotor between the conditions with and without prestress at $1.0q_V$ is diagramed. According to the test data of the long-shaft pump considered in [22], multiple frequencies were excited but the natural frequency of the rotor and its harmonics are pre-
dominant. Therefore, the frequencies shown in Figure 17(a) are of explicit significance for the vibration of the rotor. With the introduction of prestress, the natural frequency of the rotor becomes slightly high. It is thereby concluded that the rotational speed of the rotor is sufficiently low to avoid the impact of prestress. At low-order modes, the rotor behaves as a rigid body; therefore, the natural frequencies are low. At the sixth-order mode, the natural frequency is prominently high, as is related to the vibration pattern of the rotor.

During the operation of the pump, multiple frequencies will be induced. Among them, the blade passing frequency and the doubled blade passing frequency are predominant. The impeller considered has six blades and the rotational speed is 1450 r/min. Therefore, the blade passing frequency and the doubled blade passing frequency are 145 and 290 Hz, respectively. As is evidenced in Figure 17, the obtained natural frequencies deviate apparently from the two frequencies; hence, the resonance of the pump rotor is avoided.

Vibration patterns of the rotor at the six modes are shown in Figure 18. For different vibration modes, the vibration patterns are apparently different. For the first- and second-order modes, the rotor swings around the rotor axis and the total deformation decays from the impeller to the lower bearings. In practical applications, the motor was located above the pump. Therefore, the two bearings at the upper part of the shaft play a significant role in the support frame of the pump. In [23], the vibration of the motor of a vertically-installed pump was specifically investigated and it indicated that at low-order modes, bending of the motor was typical. It can thereby be extrapolated that a combination of the motor and the pump considered here will incur a complex bending pattern at low-order modes. The third-mode is dominated by the torsional vibration, and the total deformation is symmetric with respect to the rotor axis. The impeller outer edge is associated with the maximum total deformation. Regarding the fourth- and fifth-order mode, the swing of the rotor is quadratic rather than linear. The total deformation of the shaft is enhanced relative to that of the impeller. Moreover, the supporting manner for the rotor determines in a large part the vibration pattern. For the between-bearings rotor, it is shown in [24] that the primary fracture arises near the keyway through which the impeller is fixed. Here, there is no lateral constraints for the impeller, thus the freedom of the impeller end is high. For the sixth-order mode, the cubic vibration pattern of

Figure 17. Modal performance of the pump rotor.

Figure 18. Total deformation for the first six modes of the rotor.
the rotor is explicit and the shaft is distorted in two radial directions. In this case, the maximum total deformation arises at the impeller outer edge.

5. Conclusions

(1) As thermal fluid flows through the molten-salt pump, static pressure increases continuously from the impeller inlet to the volute outlet. This tendency is unaffected by the flow rate. The hydraulic loads imposed on the rotor vary as the impeller blades periodically sweep through the volute tongue.

(2) High Von Mises stress is produced at the junction of the blade inlet edge and the front impeller shroud. The largest deformation arises at the impeller outer edge. Both the maximum Von Mises stress and the largest deformation decline with the increase in the flow rate. The deformation of the shaft attenuates from the impeller to the bearings end.

(3) The dominant natural frequencies of the rotor deviate considerably from the blade passing frequency and its harmonics, ensuring stable operation of the pump rotor. At low vibration modes, bending of the rotor is representative. As the mode order increases, the torsional vibration patterns of the pump rotor are evidenced.

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