Transient thermal-mechanical coupling behavior analysis of mechanical seals during start-up operation

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Abstract. A transient thermal-mechanical coupling model for a contacting mechanical seal during start-up has been developed. It takes into consideration the coupling relationship among thermal-mechanical deformation, film thickness, temperature and heat generation. The finite element method and multi-iteration technology are applied to solve the temperature distribution and thermal-mechanical deformation as well as their evolution behavior. Results show that the seal gap transforms from negative coning to positive coning and the contact area of the mechanical seal gradually decreases during start-up. The location of the maximum temperature and maximum contact pressure move from the outer diameter to inside diameter. The heat generation and the friction torque increase sharply at first and then decrease. Meanwhile, the contact force decreases and the fluid film force and leakage rate increase.

1. Introduction

The quality of mechanical seals, which plays a significant role as a vital component in the marine equipment, always directly affects their reliability and safety. Especially under the unsteady conditions, mechanical seals are pressured by instantaneous load, including the mechanical load and the thermal load. Both temperature distribution and face deformation change over time during transient condition, either causing high local temperature so as to heat cracking failure, or oversize leakage. Therefore, it makes a claim for the optimal design of mechanical seals. And accurately predicting the face deformation appears to be particularly important.

Although most of thermal-mechanical models concentrate on steady state conditions [1-3], there are still some theoretical studies dealing with transient conditions. Harp [4] developed a mathematical model to study transient behavior of hydrostatic mechanical seals during the deceleration and acceleration, including fluid film pressure, film thickness and leakage rate variation. Green [5] studied the influence of the balance ratio and thermal time constant on the film thickness, leakage and face deformation etc., under time-varying condition. However, this model overlooked the frictional heat generation and mechanical deformation. Danos [6] developed a numerical model of the THD behavior in transient regime during start-up, which was used to study the viscous heat generation and thermal transfer. It showed that viscous heat generation and axial temperature gradient would result in
irreversible thermal instabilities. Tournerie [7] improved a TEHD transient model during start-up, using the method of influence coefficients, to study how the heat generation and minimum film thickness were changing with time. Salant [8] developed an unsteady model of a contact mechanical seal using Duhamel’s method, which was used to realize the maximum temperature, average film thickness and leakage rate variation with time.

Most of all the above studies only focused on the time history of the leakage rate, maximum temperature or average film thickness. However, the evolution laws of face deformation and temperature distribution in the interface are neglected, which need further investigation.

This paper presents a transient thermal-mechanical coupling model during start-up, taking into consideration the non-linear and strong coupling relationship among thermal-mechanical deformation, fluid film properties and heat generation. The time history of thermal-mechanical deformation and temperature distribution in the interface are analyzed depending on finite element method and multiple iteration technology.

2. Theoretical Model

![Diagram](image)

**Figure 1.** The geometrical configuration and thermal-mechanical boundary conditions of the sealing rings

Basically a mechanical seal is composed of a pair of rings as shown in figure 1. The stator is fixed to the chamber, while the rotor is spring loaded and can float axially. Two rings keep closed by the spring and fluid pressure. It operates in the mixed lubrication regime and the seal gap is full of lubrication film between the sealing faces.

To simplify the calculation process, the following assumptions have been made: (1) the temperature field and stress field of the sealing rings are axially symmetric; (2) the fluid film temperature and viscosity change along the radial direction, which are changeless along the axial direction, and the fluid density remains constant; (3) the fluid film is Newtonian fluid; (4) the influence of thermal lag and inertia is ignored; (5) a quasi-steady-state approach is used.

The character of rotational angular velocity during start-up is
\[ \omega = 314.2 \times (1 - e^{nt/2}) \]  

(1)

where \( \omega \) is rotational angular velocity at any time.

Film temperature distribution and temperature field in the sealing rings are influenced by the heat generation produced by fluid film viscous shear and asperity frictional contact.

\[
k_i \frac{\partial^2 T}{\partial r^2} + k_i \frac{\partial T}{r \partial r} + k_i \frac{\partial^2 T}{\partial z^2} = \rho c \frac{\partial T}{\partial t}
\]

(2)

where \( T \) is temperature of the rings, \( k_i \) is heat conduction coefficient, \( i \) represents the rotor and the stator respectively, which is 1 or 2. Besides, Heat transfer by convection is considered on faces \( S_1-S_7 \), \( S_9-S_{12} \) and \( S_{14}-S_{15} \) [9], as shown in figure 1, governed by the boundary condition,

\[-k_i \frac{\partial T}{\partial n} \bigg|_i = h_{ij} (T - t_f) \]

(3)

where the convection coefficient \( h_{ij} \) is referenced from Ref. [10], and \( j \) represents convection with medium and air respectively which is equal to 1 and 2; \( t_f \) is the medium temperature entering the seal chamber.

Heat flux generates by friction is transferred by conduction to the rings entirely,

\[-k_i \frac{\partial T}{\partial n} \bigg|_i = q \]

(4)

where \( q \) is heat flux on the sealing face.

The heat flux in the interface is determined by viscous shear \( q_v \) and asperity frictional contact \( q_f \),

\[ q = q_v + q_f = \mu \frac{r^2 \omega^2}{h} + f_p r \omega \]

(5)

Where \( \mu \) is dynamic viscosity of the fluid film, and it is modified exponentially with the temperature which is refered to Ref. [11], \( h \) is the thickness of the fluid film, \( f \) is frictional coefficient, \( p_c \) is the contact pressure on the sealing face and can be determined by a plastic contact model[1],

\[ p_c = H \cdot 2 \pi k \int_{\alpha_d(r)}^{e} \frac{1}{\sigma_s \sqrt{2\pi}} e^{-z^2/2\sigma_s^2} \, dz \, r \, dr \]

(6)

where \( H \) is yield stress of the softer face material, \( \sigma_s \) is combined standard deviation of the roughnesses.

Furthermore, the time domain of the heat conduction equation is discreted by the Galerkin difference scheme, the basic equation is as following:

\[
\left[ \int_0^1 [2k_i \frac{\partial T}{\partial r} + 3r \partial T/\partial r + k_i \frac{\partial^2 T}{\partial z^2}] ds + \int [k_r \frac{\partial T}{\partial r} + k_r \frac{\partial^2 T}{\partial z^2} - 3 \Delta t \rho c T N] ds \right. \\
\left. + \int_0^1 h_i r N (T - T_n) dl_i + \int_0^1 h_{id} r N (T_{id} - T_n) dl_i + \int_0^1 r N (q_{id} + 2q) dl_i = 0 \right]
\]

(7)
where $N$ is the interpolation function; $\Delta t$ is the time step; $\rho_i$ is the density of rings; $c_i$ is the specific heat capacity of rings; $q_{old}$, $h_{old}$, $T_{old}$ are the heat flux, the convection coefficient and the temperature distribution at previous time, respectively.

Within each time step, two rings are always in force equilibrium,

$$F_{open} - F_{close} = 0$$  \hspace{1cm} (8)

and,

$$F_{open} = 2\pi \int_{R_i}^{R_o} p_r r dr + 2\pi \int_{R_i}^{R_o} p_r r dr$$  \hspace{1cm} (9)

$$F_{close} = P_o \pi \left(R_o^2 - R_i^2\right) + P_i \pi \left(R_b^2 - R_i^2\right) + p_{sp} \pi \left(R_o^2 - R_i^2\right)$$  \hspace{1cm} (10)

where $p$ is the fluid film pressure, $P_o$ is the sealed pressure, $P_i$ is the atmospheric pressure, $p_{sp}$ is the spring ratio pressure, $R_b$ is the balance radius.

The opening force is affected by the fluid film pressure distribution, which is determined by the Reynolds equation,

$$\frac{\partial}{\partial r} \left( \frac{r h^3}{12 \mu} \frac{\partial p}{\partial r} \right) = r \frac{\partial h}{\partial t}$$  \hspace{1cm} (11)

Also, the time domain of the Reynolds equation is discreted by the Galerkin difference scheme, the basic equation is as following,

$$2\int_{R_i}^{R_o} \left( \frac{r h^3}{12 \mu} \frac{\partial p}{\partial r} \frac{\partial N}{\partial r} \right) dr + \int_{R_i}^{R_o} \left( \frac{r h_{old}^3}{12 \mu_{old}} \frac{\partial p_{old}}{\partial r} \frac{\partial N}{\partial r} \right) dr - \frac{3}{\Delta t} \int_{R_i}^{R_o} \left(r N \left(h - h_{old}\right)\right) dr = 0$$  \hspace{1cm} (12)

where $h_{old}$, $\mu_{old}$ and $p_{old}$ are the film thickness, dynamic viscosity and fluid film pressure at previous time.

The fluid film pressure and the contact pressure are decided by the fluid film thickness, which could be described by the face deformation,

$$h(r) = h_m + \left(\delta_1(r) + \delta_2(r)\right)_{max} - \left(\delta_1(r) + \delta_2(r)\right)$$  \hspace{1cm} (13)

where $h(r)$ is the film thickness at different radius; $h_m$ is the minimal film thickness; $\delta_i(r)$ is the deformation of sealing faces and compared with original shape, it is positive if the sealing face is stretched, so that negative if compressed; $(\delta_1(r) + \delta_2(r))_{max}$ is the maximum algebraic sum of the face deformation.

Moreover, face deformation is composed of thermal deformation and mechanical deformation. According to the principle of minimum potential energy, the finite element equation of differential equation of elastic mechanics is as follow [12],

\begin{align*}
\end{align*}
\[
\frac{1}{2} \int_S r \{e\}^T [D] \{e\} \, dS - \int r \{u\}^T \{Q\} \, dr - \int_S r \{e\}^T [D] \{e_T\} \, dS = 0
\]  
(14)

where \( \{e\} \) is strain component; \([D]\) is elastic matrix; \(\{u\}\) is the displacement vector; \(\{Q\}\) is surface pressure including sealed pressure, atmospheric pressure, fluid film pressure and contact pressure; \(\{e_T\}\) is strain component generated by thermal stress and \(\{e_T\} = \{\alpha(T-t_b), \alpha(T-t_b), \alpha(T-t_b), 0\}^T\); \(\alpha\) is the thermal expansion coefficient; \(t_b\) is the room temperature.

3. Numerical procedure

To obtain the sealing performance at each time step, the finite element method and Galerkin difference scheme are used to discrete the spatial domain and time domain respectively and all above equations are solved with each other using multiple iteration technology. The numerical procedure is shown in figure 2 and includes four iteration loops. Firstly, it needs to carry out force equilibrium analysis to obtain the initial film thickness and the temperature filed. Then, the thermal-mechanical coupling analysis under quasi-steady-state would be executed at each instant of time, including the film thickness relaxation iteration, minimal film thickness adjustment and temperature field update[13]. When the fluid film pressure, contact force, thermal-mechanical deformation and temperature distribution in the interface are convergent at the moment, the temperature field at present time will be compared with that at previous time. If it’s not convergent, next time step will be done. If it’s convergent, the computation is ended and all the parameters changing with time can be obtained.

**Figure 2.** Numerical procedure

4. Results Analysis
The physical parameters and the operating parameters of the computing instance are outlined in table 1 to table 4.

**Table 1.** material properties of the seal rings

| Parameters                        | Rotor | Stator |
|-----------------------------------|-------|--------|
| Yong’s modulus E/ GPa            | 25    | 380    |
| Poisson’s Ratio ν                 | 0.25  | 0.27   |
| Thermal conductivity k/ W/(m·℃)  | 45    | 150    |
| Thermal expansion coefficient α / 1/℃ | 6.2×10⁻⁶ | 4.3×10⁻⁶ |
| Yield stress H/ MPa              | 125   | —      |
| standard deviation σ/ μm         | 0.15  |        |
| friction coefficient f           | 0.1   |        |

**Table 2.** fluid properties

| Parameters                        | Values |
|-----------------------------------|--------|
| Thermal conductivity k/ W/(m·℃)  | 0.6256 |
| Density ρ/ kg/m³                  | 994    |
| Dynamic viscosity at 35℃ µl/ Pa·s | 0.719×10⁻³ |
| Specific heat c/ J/(kg·℃)        | 4178.5 |
| Thermoviscosity coefficient β/ 1/k | 0.0175 |

**Table 3.** operating parameters

| Parameters                        | Values |
|-----------------------------------|--------|
| Fluid pressure P/ MPa            | 3.45   |
| Atmospheric pressure P/ MPa      | 0.101  |
| Fluid inlet temperature t/ °C    | 35     |
| Rotation velocity n/ rmp         | 3000   |
| Spring pressure pₛ/ MPa          | 0.3    |
| convection coefficient h/ W/(m²·℃) | ID 528ω²/3 | 176ω²/3 |
|                                   | OD 1.43×10⁻³ω²/3 | 4.47×10⁻²ω²/3 |

**Table 4.** geometric parameters

| Parameters                        | Values |
|-----------------------------------|--------|
| Balance Ratio B                   | 0.85   |
| External diameter R₁/ mm          | 53.02  |
| Internal diameter R₂/ mm          | 48.26  |
| Balance diameter R₃/ mm           | 49     |

The film thickness and the contact pressure distribution are illustrated in figure 3 and 5, respectively. And figure 4 shows the maximum and minimum film thickness variation during start-up operation. As shown in figure 3, the seal gap appears to be negative coning and the film thickness distribution is relatively uniform along the radial direction at the beginning of the start-up. Due to the influence of the thermal deformation, the location of the minimum film thickness moves gradually from the outer diameter to the inside one with the increase of the time. In result, positive coning of the seal gap appears to be forming and the film thickness distribution is extremely non-uniform, which is contrary to the initial distribution. Besides, the minimum film thickness increases quickly at the beginning and then quickly reduced to a fixed value which is close to that under steady-state. However, the maximum film thickness is consistently increasing until approaching to a fixed value. It follows that the positive coning is exacerbated, due to the effect of thermal-mechanical coupling. Figure 5 indicates that the location of the maximum contact pressure is gradually moving toward the inside...
diameter. At the instant of a half second, contact pressure is close to 0 MPa at the outer diameter. Since then, contact area gradually reduced and the maximum contact pressure increases gradually near the inside diameter which can reach 2.5 MPa in the end. As we can see from the film thickness and contact pressure variation, the thermal-mechanical deformation and heat generation also change over time, which will inevitably lead to the higher local temperature and partial wear, even drastically rubbing.

![Figure 3. Film thickness distribution variation](image1)

![Figure 4. Maximum and minimum film thickness variation](image2)
Force variation and heat generation variation over time are shown in figure 6. Frictional heat generation and viscous heat generation rapidly increase to a maximum value at the instant of start-up, and frictional heat generation always dominates during start-up. The step change of heat generation is easy to cause large instantaneous temperature and temperature gradient, leading to a higher thermal stress and ultimately resulting in heat cracking failure. In addition, the contact force decreases while the fluid film force increases gradually, and fluid film force always dominates. This is because both the contact area and fluid film pressure drop near the outer diameter gradually decreases with the development of the positive coning.

Figure 7 displays the temperature distribution along the radius direction at the different time. Gradually the face temperature increases and the location of highest temperature moves towards the inside diameter which shows the same change law as the minimum film thickness in figure 4. The main reason lies in the higher heat flux at the location of the minimum film thickness than other locations. Besides, the temperature rise near the outer diameter is much smaller than that near the inside diameter, on account of the better convective heat transfer effect near the outer diameter.
Figure 7. Face temperature distribution variation  

Figure 8. Thermal and total deformation variation  

Figure 8 shows the thermal deformation and thermal-mechanical deformation of the rotor face in the leakage direction at different instants of time. The thermal deformation increases gradually with time, and causes the seal gap to be positive coning all the time. Similarly, the net deformation increases with time and the seal gap converts from the negative coning into positive coning gradually. This can be explained that the negative coning generated by mechanical deformation is offset by thermal deformation. Thus, the results will have larger deviation when ignoring the influence of temperature field and thermal deformation. Particularly during start-up, seal gap variation caused by the thermal deformation will directly affect the change of film thickness, which gives rise to the redistribution of fluid film pressure, contact pressure and heat generation and finally, the instability of sealing rings.

Figure 9. Friction torque and leakage rate variation  

The friction torque and leakage rate variation over time are shown in figure 9. Contact friction torque is always greater than viscous friction torque during the start-up and both have a step change and then decrease with time, which is same with the variation in figure 6. The major reasons are that the contact pressure distribution is relatively uniform and the seal gap is negative. The excessive friction torque at the beginning can cause a difficult start-up, and even lead to serious sealing rings fracture. In addition, leakage rate, affected by the seal gap and fluid film viscosity, increases rapidly
after the instant of start and then flattens out.

5. Conclusions
(1) The transient thermal-mechanical coupling model of mechanical seals for start-up operation is applies to predict transient behavior during start-up, which can be used to analyze the time history of temperature distribution, thermal and mechanical deformation.

(2) With the increase of the time, the seal gap transforms from the negative coning to the positive coning and the contact area gradually decreases. The location of the maximum temperature and maximum contact pressure move from outer diameter to inside diameter.

(3) The heat generation and torque take on step change and then decrease. Meanwhile, the contact force decreases while fluid film force and leakage rate increase.

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