Research Article

Theoretical and Experimental Study of the Torque Impact and Shaft Response during the Rapid Engaging Process of a Gas Turbine

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Received 25 November 2021; Revised 18 April 2022; Accepted 27 April 2022; Published 17 May 2022

Academic Editor: Francesco Bucchi

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Gas turbines produce torque impacts during the rapid engaging process, which could obviously affect the safety of the propulsion shafting. To analyze the impact load and the corresponding response of a shaft, a calculation method is proposed for the torque impact load, based on a propulsion shafting model and the motion characteristics of a synchro-self-shifting (SSS) clutch. In addition, the time-domain integration method was used to analyze the shaft response under an impact load. Finally, the engaging process was reproduced in an experimental plant, and the proposed method was verified with measured transient torque responses. The results show that an obvious torque impact is produced in the shafting under the action of the damping force when the SSS clutch meshes during the engaging process, with the torque response amplitude fluctuating within a certain range. Adjusting the engaging parameters of the working engine and the engaging engine can reduce the amplitude of the shaft response.

1. Introduction

Combined gas turbine and gas turbine (COGAG) plants are widely used in current warships because they have the advantages of high power density and flexibility within changing working conditions. COGAG plants can produce higher total output power and require simpler logistic maintenance than combined diesel and gas turbine (CODAG) plants. Ships equipped with COGAG plants usually use one gas turbine during cruising conditions to increase the endurance of the ship. When the ship needs to accelerate, another gas turbine is started and merged into the system.

At present, to ensure the stability of the engaging process, the engaging gas turbine gradually increases the output torque and adjusts the rotational speed over a period of time [1]. However, to improve the maneuverability of ships in emergency situations, rapid engaging methods and their corresponding impacts during the transition process should be studied.

A large amount of research has been conducted on the performance of gas turbines [2, 3]. In this research, the linear or nonlinear characteristics of each gas turbine component were converted into a modular digital simulation model to simulate the steady-state or transition conditions of the gas turbines. Additionally, gas turbines have been combined with propulsion shafting to simulate the operational processes of COGAG and CODAG plants [4, 5]. These simulations could be used to optimize the operation parameters.

The synchro-self-shifting (SSS) clutch is the key component for achieving reliable engagement and disengagement between a gas turbine and a combined power plant. Hendry introduced the operation and maintenance experience of the SSS clutch on a ship in detail [6–8]. The study of the dynamic characteristics of SSS clutches primarily involves either numerical calculation [9, 10] or multibody dynamics calculation methods. The former focus on the kinematic characteristics of the clutch meshing process and are often nested into modular simulations of COGAG plants. The latter is generally used to calculate the dynamic
characteristics of the clutch engagement process, including the collision and impact phenomena [11]. Luneburg described the impact phenomenon in a simulation of an SSS clutch engagement process using a numerical method [12].

The impact phenomenon caused by the engaging process of a COGAG plant has not been given much attention, and research regarding the dynamic response of shafting caused by such an impact is rare. The transient torsional vibration of propulsion shafting, however, has been thoroughly studied, and the literature includes studies regarding polar ship shafting under ice loads. The ice loads could produce significant drops in rotational speed and torque response. The modelling methods used for transient torsional vibration of propulsion shafting include the lumped parameter method and the bond graph method [13, 14]. In addition, the influence of control parameters on the torsional vibration and speed fluctuations on shafting can be analyzed in the time domain by coupling the speed control system of an engine with a shafting model [15].

The experimental studies regarding gas turbines focus on improving the combustion performance and the efficiency of the turbines [16, 17]. Few test benches have been built for studying the engaging process of a COGAG plant. Tian developed a digital detector for monitoring the parameters and fault predictions of SSS clutches [18].

This paper, focusing on the problem of little available research regarding rapid engagement and the impact phenomenon in COGAG plants, proposes a rapid engaging method and an impact load calculation method to investigate the mechanisms governing these processes. The method of using lumped parameter modelling combined with the Newmark method to calculate the dynamic response of shafting under an impact is also described. Finally, the proposed engaging working conditions were reproduced in an experimental plant, and the transient torque of the shaft and the instantaneous rotational speed of the SSS clutch were measured. The theoretical calculation method was verified using experiments. The calculation method proposed in this paper can provide a reference for safety evaluations and analyses of COGAG plants.

2. Analysis of the Torque Impact during the Engaging Process of a Gas Turbine

2.1. Gas Turbine Engaging Process. A COGAG plant is normally composed of two gas turbines, SSS clutches, a parallel gearbox, a propeller, and corresponding shafting. When one gas turbine (hereafter referred to as the running turbine) in the system drives the load and the other gas turbine (hereafter referred to as the engaging turbine) engages into the system, the two turbines must be synchronized. Then, the SSS clutch of the engaging turbine is driven to mesh, and the loads of the two gas turbines are adjusted to the set values to complete the engaging operation.

In a COGAG plant, the SSS clutch is a key component for achieving engagement of the gas turbines and changing the torque transmission path in the system. As shown in Figure 1, an SSS clutch is primarily composed of a driving unit, a sliding unit, and a driven unit [6]. It operates according to the principles described next. When the rotational speed of the driving unit is synchronized with the driving unit, and when the rotational speed of the driving unit begins to surpass that of the driving unit, the torque from the driving unit is transmitted through the spiral spline to the sliding unit. The sliding unit remains relatively stationary with respect to the driving unit because of the ratchet and pawls, but relative rotation occurs between the driving unit and the sliding unit. Under the action of the helical spline, the sliding unit axially slips until it reaches the end surface. Then, the main gear of the sliding unit and the ring gear of the driven unit are fully meshed, and the SSS clutch starts to transmit torque.

In a normal engaging process, the system parameters change smoothly and the impacts on the system are not significant. However, in emergency conditions, to improve the maneuverability of the power plant, the engaging turbine directly raises the throttle, bypassing the speed synchronization process, after reaching its self-sustaining speed. In this way, the engaging turbine can be rapidly connected to the system. This method can shorten the engagement time; however, the relative motion and collisions between the components during the engagement of the SSS clutch are more severe, which may generate a corresponding significant torque impact.

2.2. External Conditions of the SSS Clutch Engagement Process. The driving torque, \(M_{\text{in}}\), in the driving unit of the SSS clutch is determined by the engaging gas turbine connected to it, while the external torque, \(M_{\text{ext}}\), at the driven unit is determined by the state of the load and the running gas turbine. According to Figure 2, the propulsion system can be divided into two parts with the SSS clutch as the boundary (hereinafter the front part and the rear part). The kinetics equation for the rear shafting connected to the gearbox can be expressed by the following equation:

\[
(M_{\text{ext}} + M_T)i - M_{\text{pro}} = J \frac{d\omega_{\text{out}}}{dt},
\]

(1)
where $M_F$ represents the output torque of the running gas turbine, $i$ is the gearbox reduction ratio, $M_{pro}$ is the resisting torque of the propeller, and $J$ is the equivalent moment of inertia of the rear part. $J$ can be calculated by the following equation:

$$J = J_{\text{front}} + J_{\text{rear}},$$

(2)

where $J_{\text{front}}$ represents the moment of inertia of the non-reduced part of the shafting (including the pinion gear in the gearbox) and $J_{\text{rear}}$ is the moment of inertia of the reduced part of the shafting (including the big gear in the gear box).

$M_{pro}$ in equation (1) can be obtained from the following equation:

$$M_{pro} = K_O p_w n_p^2 D_h^5,$$

(3)

where $K_O$ represents the torque coefficient of the propeller, $p_w$ is the density of seawater, $n_p$ is the rotational speed, and $D_h$ is the diameter of the propeller.

2.3. Kinetic Analysis of the Meshing Process of the SSS Clutch.

The kinetic equations for the driving unit can be expressed by the following equation:

$$\begin{cases}
M_{in} - M_{hr} - M_{fr} = J \frac{d\omega_{in}}{dt}, \\
F_{fa} + F_R - F_{ha} - F_{b1} = 0,
\end{cases}$$

(4)

where $J_{in}$ represents the moment of inertia of the driving unit, $\omega_{in}$ is the angular velocity of the driving unit, and $M_{in}$ is the external torque loaded on the driving unit. $M_{fr}$ and $F_{fa}$ are the circumferential moment and axial force, respectively, produced by the tangential and axial components of the helical tooth contact force, $F$. $M_{fr}$ and $F_{fa}$ are the circumferential friction moment and axial friction force, respectively, produced by the tangential and axial components of the helical tooth friction force, $F_f$. $F_{fa}$ is the axial force that keeps the driving unit axially fixed.

When the sliding unit approaches the end of the spiral spline, under the action of the damping oil chamber, it is subjected to a damping force to avoid a strong rigid collision. Therefore, the driving unit will be subjected to a reaction force, $F_R$, from the sliding unit. The forces acting on the driving unit are shown in Figure 3.

The helical tooth contact force, $F$, can be obtained from the following equation:

$$F = \frac{2M_{hr}}{D_h \cos \beta \cos \alpha_h}$$

(5)

where $D_h$ represents the pitch of the helical spline, $\beta$ is the helix angle of the helical spline, and $\alpha_h$ is the pressure angle of the helical spline.

The axial force, $F_{ha}$, can be expressed by the following equation:

$$F_{ha} = \frac{2M_{hr} \tan \beta}{D_h}$$

(6)

The gear tooth friction force, $F_f$, can be obtained from the following equation:

$$F_f = F_f \beta$$

(7)

where $f_b$ is the tooth friction coefficient of the helical spline.

Then, the axial friction force, $F_{fa}$, and the circumferential friction moment, $M_{fr}$, can be obtained from the following equation:

$$\begin{cases}
F_{fa} = \frac{2M_{hr} f_b}{D_h \cos \alpha_h} \\
M_{fr} = \frac{M_{hr} \tan \beta f_b}{\cos \alpha_h}
\end{cases}$$

(8)

$F_R$ can be calculated using the empirical formula in the following equation [19]:

$$F_R = \frac{c}{2} \left( \frac{D_h}{2 \tan \beta} \right)^2 (\omega_{in} - \omega_s)^2 \forall x_s \geq L_R$$

(9)

where $x_s$ represents the axial displacement of the sliding unit, $L_R$ is the position where the damping oil chamber starts to produce the damping force, $\omega_s$ is the angular velocity of the sliding unit, and the difference between $\omega_s$ and $\omega_{in}$ represents the axial velocity of the sliding unit. $c$ is the damping coefficient, which can be calculated by the following equation:
\[ c = \frac{\rho A_c^3}{2 \mu^2 A^2} \]  

where \( \rho \) is the oil density, \( \mu \) is the flow coefficient of the damping oil hole, \( A_c \) is the cross-sectional area of the damping oil chamber, and \( A \) is the cross-sectional area of the damping oil hole.

The kinetic equations for the sliding unit are expressed by the following equation:

\[
\begin{aligned}
M_{hr} + M_{fr} - (M_p + M_g) &= J_s \frac{d\omega_s}{dt} \\
F_{ha} - F_{fa} - (F_{fp} + F_{fg}) - F_R &= m_s \frac{dv_s}{dt}
\end{aligned}
\]  

(11)

Here, \( J_s \) represents the moment of inertia of the sliding unit, \( \omega_s \) is the angular velocity of the sliding unit, \( m_s \) is the mass of the sliding unit, and \( v_s \) is the axial velocity of the sliding unit. \( M_p \) is the circumferential moment and \( F_{fp} \) is the axial friction force, both of which are produced by the ratchet and pawl. \( F_{ha} \), \( F_{fa} \), \( F_{hr} \), and \( M_p \) are reaction forces and reaction moments from the driving unit. During the meshing process, the ratchet and pawl gradually disengage, and the main gear and the ring gear start to engage. The circumferential moment, \( M_g \), and the axial friction force, \( F_{fg} \), begin to act on the sliding unit, both of which are produced by the main gear and the ring gear. The forces acting on the sliding unit are shown in Figure 4.

The axial friction force, \( F_{fp} \), can be obtained from the following equation:

\[ F_{fp} = \frac{2M_p f_p}{D_p \cos \alpha_p} \]  

(12)

where \( f_p \) represents the friction coefficient between the ratchet and pawl, \( D_p \) is the diameter of the ratchet, and \( \alpha_p \) is the equivalent pressure angle of the contact surface between the ratchet and pawl.

Similarly, the axial friction force, \( F_{fg} \), can be obtained from the following equation:

\[ F_{fg} = \frac{2M_{fg} f_g}{D_g \cos \alpha_g} \]  

(13)

where \( f_g \) is the friction coefficient between the main gear and ring gear, \( D_g \) is the diameter of the main gear, and \( \alpha_g \) is the pressure angle of the main gear.

The kinetic equations for the driven unit can be expressed by the following equation:

\[
\begin{aligned}
M_g + M_p - M_{ex} &= J_{out} \frac{d\omega_{out}}{dt} \\
F_{b2} - (F_{fp} + F_{fg}) &= 0
\end{aligned}
\]  

(14)

where \( J_{out} \) represents the moment of inertia of the driven unit and \( \omega_{out} \) is its angular velocity. \( M_{ex} \) is the external torque loaded on the driven unit, which is determined by the shafting of the rear part. \( F_{fp} \), \( F_{fg} \), \( M_p \), and \( M_g \) are the reaction forces and reaction torques from the sliding unit, and \( F_{b2} \) is the axial force that keeps the driven unit axially fixed. The forces acting on the driven unit are shown in Figure 5.

During the meshing process, since the circumferential rotation of the sliding unit is constantly restricted by the driven unit, the following equation is applicable:

\[ \omega_s = \omega_{out} \]  

(15)

The axial velocity of the sliding unit can be calculated by the following equation:

\[
\begin{aligned}
v_s &= \frac{D_p}{2 \tan \beta} (\omega_{in} - \omega_s) ; \forall \omega_{in} \geq \omega_s \\
0 &< \omega_{in} < \omega_s
\end{aligned}
\]  

(16)

Combining equations (1) and (14) yields the following equation:

\[ M_r \left( J_{out} + \frac{1}{i} \right) \frac{d\omega_s}{dt} = \frac{M_{pro} - iM_r}{i} \]  

(17)
where $M_r$ is equal to $M_p$ before the ratchet and pawl disengage and is equal to $M_g$ after the ratchet and pawl disengage.

By combining equations (4), (11), and (17), the kinetic equation for the SSS clutch engagement process can be written in a matrix form, as shown in equation (18). By solving equation (18), the motion and force states of the components in the SSS clutch during the engagement process can be obtained.

\[
\begin{bmatrix}
1 + \tan \frac{\beta f_h}{\cos \alpha_g} & 0 & 0 \\
1 + \tan \frac{\beta f_h}{\cos \alpha_g} & -1 & 0 \\
2(\tan \beta \cos \alpha_g - f_h)D_h \cos \alpha_g & \frac{2f_p}{D_p \cos \alpha_p} + \frac{2f_g}{D_g \cos \alpha_g} & -m_l D_h \tan \beta & m_l D_h \tan \beta & 0 & -\left(J_{in} + \frac{J}{i}\right)
\end{bmatrix} \begin{bmatrix}
J_{in} \\
0 \\
0 \\
-1 \\
0 \\
0
\end{bmatrix} = \begin{bmatrix}
M_{hr} \\
M_t \\
\frac{d\omega_{in}}{dt} \\
\frac{d\omega_s}{dt} \\
0 \\
F_R \\
M_{pro} - iM_T
\end{bmatrix}
\tag{18}
\]

The engaging process in a COGAG plant is defined next. First, the working gas turbine drives the stably working system with a certain output torque, then the engaging gas turbine begins to increase the output torque within the system with a certain output torque, then the engaging gas turbine starts and finishes the engagement process, respectively. During the loading process, the rotational velocity of the engaging gas turbine increases until it is equal with that of the working gas turbine, thereby driving the SSS clutch to mesh. The rotational velocity of the propeller continues to increase to a stable value. During the gas turbine loading process, the output torque climbs approximately linearly. Therefore, as shown in equation (19a 19b), it is assumed that the output torque, $T_B$, of the working turbine is constant before and after the engagement, and the output torque, $T_E$, of the engaging gas turbine linearly increases during the engagement process to simplify the loading process and better control the variables.

\[
T_R = N_R T_{rated}, \quad (19a)
\]

\[
T_E(t) = \begin{cases}
0 & \forall t < t_s \\
N_E T_{rated} \frac{t - t_s}{t_c - t_s} & \forall t_s \leq t \leq t_c \\
N_E T_{rated} & \forall t > t_c
\end{cases} \quad (19b)
\]

Here, $T_{rated}$ represents the rated torque of the gas turbine. $N_R$ and $N_E$ are the torque percentages of the gas turbines set before engaging process starts. $t_s$ and $t_c$ are the moments at which the engaging gas turbine starts and finishes the loading process, respectively.

MATLAB was used in carrying out programming, and the experimental bench in Figure 6 was taken as the object. The torque percentage, $N_R$, was set to 70%, and the loading time, $t_l$, was set to 9 s. The engaging turbine torque percentage, $N_E$, was set to 50%, 70%, and 90%. The torque on the spiral spline and the displacement of the sliding unit were then obtained, as shown in Figure 7.

Figure 7 shows that the axial velocity of the sliding unit is significantly reduced at the moment when the damping force acts, and obvious torque impacts are produced on the spiral spline, with amplitudes of 75.73 N·m, 70.31 N·m, and 59.17 N·m, for $N_E$ values of 90%, 70%, and 50%, respectively. In addition, the greater the torque percentage of the engaging turbine, the greater the torque impact and the shorter the time required for the SSS clutch to finish meshing.

### 3. Dynamic Response Analysis of the Shafting under an Impact Load

The torque impact obtained from the analysis presented above is located in the spiral spline and acts on the rear and front parts of the shafting simultaneously. The torque dynamic response is generated in these two multiple-degree-of-freedom systems. The amplitude of the torque response is very important for a safety evaluation of the engaging process, so it is necessary to analyze the dynamic response.

#### 3.1 Calculation Model

Figure 8 takes the shafting in an experimental plant in Figure 6 as an example, simplifying it into a chain lumped model, in which the left branch includes the engaging engine. The front part consists of units #1–#3, and the rear part consists of units #4–#10. Torque impact loads with the same sizes and opposite directions were applied to units #3 and #4, then the dynamic response of the shafting under the impact loads could be obtained. Vibration differential equations for the two parts can be expressed by the following equations:
Taking the front part model as an example, the uncondi-
tional convergence Newmark-β method is used to solve its dynamic response in the time domain. According to equations (21) and (22),

\[
\begin{align*}
J_{\text{in}} \ddot{\theta} + C_{\text{in}} \dot{\theta} + K_{\text{in}} \theta &= T_s(t). \\
J_{\text{ex}} \ddot{\theta}' + C_{\text{ex}} \dot{\theta}' + K_{\text{ex}} \theta' &= -T_s(t). 
\end{align*}
\]  

Here, \( J_{\text{in}} \), \( C_{\text{in}} \), and \( K_{\text{in}} \) are the inertia, damping, and stiffness matrices for the front part model, and \( J_{\text{ex}}, C_{\text{ex}}, \) and \( K_{\text{ex}} \) are the inertia, damping, and stiffness matrices for the rear part model. \( \theta, \dot{\theta}, \) and \( \ddot{\theta} \) represent the angular dis-
placement, angular velocity, and angular acceleration vectors for the front part model, and \( \theta', \dot{\theta}', \) and \( \ddot{\theta}' \) are the angular displacement, angular velocity, and angular acceleration vectors for the rear part model. \( T_s(t) \) is the vector defined by the impact load.

Equation (23) expresses the vibration differential equation at time \( t + \Delta t \).

\[
\begin{align*}
J_{\text{in}} \ddot{\theta}_{t+\Delta t} + C_{\text{in}} \dot{\theta}_{t+\Delta t} + K_{\text{in}} \theta_{t+\Delta t} &= T_s(t + \Delta t). \\
J_{\text{ex}} \ddot{\theta}'_{t+\Delta t} + C_{\text{ex}} \dot{\theta}'_{t+\Delta t} + K_{\text{ex}} \theta'_{t+\Delta t} &= -T_s(t + \Delta t). 
\end{align*}
\]

According to equations (21) and (22),

\[
\begin{align*}
\dot{\theta}_{t+\Delta t} &= \dot{\theta}_t + \left[ (1 - \delta) \dot{\theta}_t + \delta \dot{\theta}_{t+\Delta t} \right] \Delta t. \\
\dot{\theta}'_{t+\Delta t} &= \dot{\theta}_t + \dot{\theta}_t \Delta t + \left[ \frac{1}{2\beta} \right] \frac{1}{\beta} \Delta \theta_t. 
\end{align*}
\]

Substituting equations (24) and (25) into equation (23) yields the following equation:

\[
\tilde{K} \theta_{t+\Delta t} = \tilde{T}_{t+\Delta t}. 
\]
Equations (27) and (28) define the variables in equation (26).

\[
\mathbf{K} = \mathbf{K}_\text{in} + \frac{\delta}{\beta \Delta t} \mathbf{C}_\text{in} + \mathbf{J}_\text{in}. \tag{27}
\]

\[
\mathbf{T}_{t+\Delta t} = \mathbf{T}_t(t + \Delta t) + \mathbf{J}_\text{in} \left[ \frac{1}{\beta \Delta t} \dot{\theta}_t + \frac{1}{\beta \Delta t} \dot{\theta}_t + \left( \frac{1}{2 \beta} - 1 \right) \ddot{\theta}_t \right] + \mathbf{C}_\text{in} \left[ \frac{\delta}{\beta \Delta t} \dot{\theta}_t + \left( \frac{\delta}{\beta} - 1 \right) \dot{\theta}_t + \left( \frac{\delta}{2 \beta} - 1 \right) \Delta t \dddot{\theta}_t \right]. \tag{28}
\]

\[\dot{\theta}_{t+\Delta t}\] can be obtained by solving equation (26), and then \[\dot{\theta}_{t+\Delta t}\] can be obtained from equations (24) and (25). The dynamic response of the rear part model can also be solved in a similar fashion.

After obtaining the instantaneous angular displacement, angular velocity, and angular acceleration of each element, the instantaneous torque, \(T_i(t)\) of the shaft between elements can be obtained from the following equation:

\[T_i(t) = k_{t,i+1} \left[ \theta_i(t) - \theta_{i+1}(t) \right] (i = 1, 2, \ldots, n - 1). \tag{29}\]

Here, \(n\) is the total number of elements in the model and \(k_{t,i+1}\) represents the torsional stiffness of the shaft between element \(i\) and element \(i + 1\).

The torque response of the shaft between the driven unit and the pinion gear and the torque response of engaging engine shaft, under impact loading was obtained by programming the model in MATLAB. The results for the engaging parameters \(N_E = N_R = 70\%\) and \(t_1 = 9\ s\) are shown in Figure 9.

Figure 9 shows that under impact loading, obvious torque responses occur at the shafting in both the front and rear parts. The amplitude for the driven unit and the pinion gear is 19.14 N-m, and the amplitude for the engaging engine shaft is 36.76 N-m.

3.2. Effect of the Initial Relative Angle. When the velocity of the driving unit of the SSS clutch exceeds that of the driven unit, the ratchet pawl is usually not engaged properly, but rather is offset by a certain angle. This means that the driving unit must turn by another certain angle, \(\varphi_r\), relative to the driven unit before the ratchet and pawl engage. The maximum value of \(\varphi_r\) is related to the number of ratchet teeth and pawls, and it can be calculated using the following equation:

\[\varphi_{r, \max} = \frac{2\pi}{b z_p}. \tag{30}\]

Here, \(b\) is the number of ratchet teeth and \(z_p\) is the number of pawls.

The larger the initial relative angle, \(\varphi_r\), the longer the driving unit will accelerate before the ratchet and pawl engage. This indicates that the sliding unit has a higher initial velocity when it begins to slip, thus aggravating the torque impact. Since the relative position between the ratchet and pawl before the SSS clutch engages is uncontrollable, the initial relative angle, \(\varphi_r\), is a random variable for the system. This randomness could cause the torque impact and shafting response to be random within a certain range.

4. Experimental Verification

4.1. Experimental Bench Design. To verify the correctness of the calculation method for the impact loading and shafting response during the engagement process, an experimental bench with a servo motor as the prime mover and an electric dynamometer to simulate the propeller load was built. The experimental bench included two servo motors with the same parameters (hereafter referred to as the running motor and the engaging motor), one electric dynamometer (hereafter referred to as the loading motor), one SSS clutch, one reduction gear box, and corresponding connecting shafts. Since the running gas turbine is continually connected to the system during the engagement process, the SSS clutch at the running turbine was eliminated to simplify the system. The parameters of the related equipment are listed in Table 1.
Table 1: Parameters of the experimental equipment.

| Equipment           | Parameter   | Value       |
|---------------------|-------------|-------------|
| Servo motors        | Rated power | 4 kW        |
|                     | Maximum speed | 3000 rpm   |
| Electric dynamometer| Rated power | 10 kW       |
|                     | Maximum speed | 2000 rpm   |
| SSS clutch          | Rated torque | 800 N·m     |
| Gear box            | Speed ratio  | 3:1         |
| Torque sensor       | Sampling rate | 20 Hz      |
| Transient torque sensor | Sampling rate | 1 000 Hz |

Figure 6 shows the layout of the equipment on the experimental bench.

During the experiments, the average torques and average rotational speeds of the running motor, the engaging motor, and the loading motor were monitored by three torque sensors. The transient torque of the shaft between the driven unit of the SSS clutch and the gearbox was collected by a transient torque sensor, and the transient rotational speed of the driven and driving units of the SSS clutch were measured by a hall sensor and a built-in gear mounted in the transient torque sensor and the torque sensor, respectively.

4.2. Experimental Results. In the experiments, the two servo motors operated according to the loading method described in Section 2.1. The engaging parameters used in the experiments are shown in Figure 10, which indicates there were 45 working conditions.

Figure 11 shows the average rotational speed and the average torque during the engaging process, with the engaging parameters $N_E = 70\%$, $N_R = 70\%$, and $t_1 = 9$ s. Figure 11 shows that before engagement, the rotational speed of the running motor is 1,553.34 r/min. After the engaging motor starts to work, its output torque reaches 16.80 N·m within nine seconds, and its rotational speed begins to rise rapidly and synchronizes with the running motor at 47.64 s. After this, the rotational speed of the loading motor increases and stabilizes at 2,454.02 r/min at 56.57 s, and its torque reaches 91.26 N·m at 56.57 s. Due to the resistance in the system, the torque of the loading motor after stabilization is less than the theoretical value of 100.80 N·m, so the transmission efficiency of the system is 90.54%.

Figure 12 shows the transient torque measured during the meshing process of the SSS clutch. Figure 12 shows that the rotational speed difference between the driving and driven units of the clutch reaches a maximum value of 60 r/min at 0.15 seconds after the meshing initiation, and then rapidly decreases. At the moment when the rotational speed difference decreases to 0, an obvious torque response with an amplitude of 25.46 N·m is observed.

Repeated experiments were conducted for a single engaging condition. Some of the results for the $N_E = 50\%$ and $t_1 = 9$ s engaging parameters are shown in Figure 13.
Figure 13 shows that the amplitudes of the torque responses for each engaging condition are evenly distributed between the maximum and minimum values and that the ratios of the maximum value to the minimum value are 1.62, 1.57, and 1.56, for NR values of 50%, 70%, and 90%, respectively.

To overcome the inconvenience of the result analyses caused by the randomness of the torque response, average values were used for analysis, as shown in Figure 14.

Figure 15 presents the theoretically calculated results for the corresponding parameters. The following four primary observations were drawn from Figures 14 and 15:

1. The average values of the torque response amplitude obtained from the experiments are consistent with the theoretical calculation results. The minimum error is 1.08%, and the maximum error is 15.33%. This result indicates that although the torque response of the shafting fluctuates within a certain range, the correctness of the calculation method was verified by the repeated experiments.

2. The torque response amplitude decreases with increases in the loading time of the engaging motor, and the decreasing trend gradually slows. Taking the engaging motor torque percentage of 50% in Figure 14(a) as an example, as the loading time increases from 1 s to 5 s to 9 s, the torque response amplitude drops from 40.63 N·m to 24.62 N·m to 20.04 N·m, respectively, which represent decreases of 39.40% and 50.67%, respectively.

3. The torque response amplitude increases as the engaging motor torque percentage increases, and the increasing trend gradually slows. Taking the loading time of 1 s in Figure 14(a) as an example, as the torque percentage of the engaging motor increases from 50% to 70% to 90%, the torque response amplitude increases from 40.63 N·m to 45.62 N·m to 48.55 N·m, respectively, which represent increases of 12.28% and 19.49%, respectively.
Comparing the results of the different running motor torque percentages laterally shows that the torque response amplitude increases slightly with increases in the running motor torque percentage. Taking the engaging motor torque percentage of 50% and the loading time of 1s as an example, as the running motor torque percentage increases from 50% to 70% to 90%, the torque response amplitude increases from 40.63 N·m to 41.99 N·m to 44.18 N·m, respectively, which represent increases of 3.38% and 8.74%, respectively.

Although real COGAG plants have more complex structures and working conditions than the experimental plant, both share the same critical components and torque transmission paths. To apply the calculation method proposed in this paper to a real COGAG plant, it is necessary to simplify the key component SSS clutch and obtain the torque characteristics of the gas turbine during the loading process to evaluate the torque impact. It is also necessary to simplify the shafting part in the plant to obtain an accurate dynamic model, which includes the equivalent moment of inertia, the stiffness, and the damping, to calculate the torque response of each component.
5. Conclusions

In this study, the torque impact and torque response that may be generated by a COGAG plant during the engaging process were calculated and analyzed by simulating an SSS clutch engagement process and using a time-domain response calculation method. The research is based on a shafting model on an experimental bench that contained the same critical components as a real COGAG device. From the theoretical analyses and experimental results, the following four primary conclusions can be drawn:

(1) COGAG plants can realize the engagement of the SSS clutch and complete the engaging operation using the rapid-engaging method proposed in the article. The moment when the SSS clutch meshes, a significant torque impact and a torque response can be produced in the shafting.

(2) The torque response calculation method proposed in the article can be used to calculate the torque impact and the shaft response during the engaging process, and it was verified by experiments.

(3) Engaging parameters have significant influences on the torque response amplitude under torque impact. Reducing the torque percentages of the running and engaging motors and extending the loading time of the engaging motor can all reduce the torque response amplitude, with the loading time extension being the most effective.

Figure 15: Calculated torque response amplitudes for different engaging parameter values: (a) $N_R = 50\%$, (b) $N_R = 70\%$, and (c) $N_R = 90\%$. 

Shock and Vibration 11
(4) The response amplitude of the shaft system fluctuated within a certain range for the same engaging parameters. Therefore, to evaluate the safety of the engaging process more accurately, the torque response at the maximum relative angles should be considered.

It should be noted that, in order to simplify the working conditions, the torque was assumed to be linearly loaded in this study. In further research, combined with a gas turbine simulation model, the engaging process and the impact load will be simulated for a real COGAG plant.

Data Availability

The data used to support the study are included in the paper.

Conflicts of Interest

The authors declare that there are no conflicts of interest.

Acknowledgments

This study was supported by the National Major Science and Technology Projects (2017-IV-0006-0043) and the National Natural Science Foundation of China (Grant No. 51839005).

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