UNBALANCE RESPONSE CONTROL IN HIGH-SPEED GAS TURBINE SYSTEM USING MAGNETO-RHEOLOGICAL FLUID BASED SEMI-ACTIVE SQUEEZE FILM DAMPER

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ABSTRACT

The high-speed gas turbine system, consisting of dual (high- and low-pressure spools) rotors, supporting ball bearings, and an oil-lubricated squeeze film damper (SFD), is becoming a standard layout for aircraft propulsion system. In this work, a magneto-rheological fluid based SFD (MR-SFD) is introduced, in lieu of the conventional SFD, in order to investigate its capability of attenuating the unbalance responses as the rotor system passes the critical speeds. The finite element model of the dual rotor equipped with the MR-SFD is established, and then, an input-scheduling algorithm based on the singular value analysis is utilized to determine the optimal input current level so that the maximum relative stability of the rotor system is attained. It is found that the MR-SFD is very effective in attenuating the unbalance response over the critical speeds associated with the low-pressure spool rotor, not with the high-pressure spool rotor, because the MR-SFD directly supports the low-pressure spool rotor only.

Keywords: Magneto-rheological fluid, squeeze film damper, unbalance response control, dual rotor, singular value analysis

NOMENCLATURE

C, $\mathbf{C}$	damping
D	diameter
E	Young's modulus
$\mathbf{f}$	external force vector
F_X	force
I_c	input current
j	imaginary number
J-I	difference between polar and diametrical moments of inertia
k, $\mathbf{K}$	stiffness
l	shaft length
m, $\mathbf{M}$	mass
$\mathbf{q}$	state vector
t	time
X	displacement
Γ, $\mathbf{\Gamma}$	hysteresis damping
ρ	density
$\sigma(\cdot)$	singular value
Ω	rotational speed

INTRODUCTION

Recently the trends in turbomachinery seeking for a greater power output per unit of weight, and for higher speed and lighter weight power transmission shaft, demand a need for the development of high-speed flexible rotor and shaft technology. The implementation of lightweight flexible shaft, particularly in high-speed gas turbine system, brings about the apparent advantages such as fewer parts, a less complex mechanism, and more efficiency [6]. Usually the high-speed gas turbine system adopts a dual spool structure that is interconnected by an intershaft bearing to enhance the load capacity. In such a system, the vibrations of the inner and outer shafts are coupled through the intershaft bearing. Consequently, the unbalance of each rotor depends upon not only its own unbalance but also the other's. Furthermore, the shaft flexibility of the dual rotor system may contribute to increased unbalance response and thus lead to excessive vibrations particularly near critical speeds. It requires integration of an additional damping element such as SFD into the rotor system such that the excessive vibration could be effectively attenuated. However, it has been known that the conventional SFD is limited in the effective vibration control of the flexible rotor over several critical speeds, mainly due to its passive nature in dynamic characteristics. Such limitation has stimulated the research works on use of controllable dampers working in active or semi-active way [1,2,4,7,10], including the semi-active SFDs using controllable fluids, namely electro-rheological (ER) and magneto-rheological (MR) fluids.

In this work, the semi-active MR fluid based SFD (MR-SFD) introduced in [4] is applied to a high-speed dual rotor gas turbine system and its unbalance response is evaluated, using an empirical dynamic stiffness model for the MR-SFD and a finite element (FE) dual rotor system model. For the semi-active control of MR-SFD, a simple, yet effective, control algorithm for unbalance response control is proposed, that determines the proper control current level based on the singular value analysis, and the effectiveness of the algorithm in vibration attenuation is investigated.

1. SYSTEM DESCRIPTIONS

1.1 High-Speed Gas Turbine System

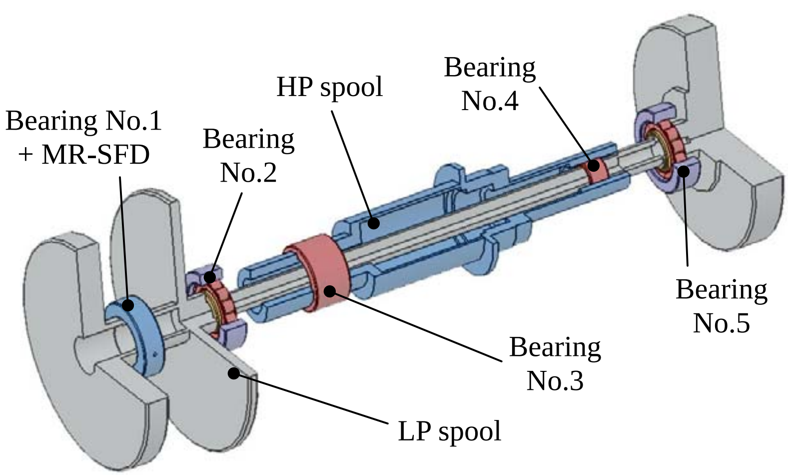


Fig. 1. Schematic of the high-speed gas turbine system [6].

Figure 1 shows the schematic view of the isotropic high-speed gas turbine system of interest, where the high pressure (HP) and low pressure (LP) spools are interconnected by an intershaft bearing (bearing No.4). Table 1 gives the specifications of the system along with the designed operating speed ranges for each spool.

Table 1. Specifications of the high-speed gas turbine system [6].

LP spool data	
Shaft	ρ=7.83×10^3 kg/m^3, E=1.79×10^{11} N/m^2, l=638 mm, m=11.7 kg $D_{o,max}$=93 mm/$D_{o,min}$=20 mm, $D_{i,max}$=50 mm/$D_{i,min}$=0 mm
Disk	No.1: m_d=3.02 kg/$(J\text{-}I)_d$=0.0069 $kg\text{-}m^2$ No.2: m_d=1.60 kg/$(J\text{-}I)_d$=0.0050 $kg\text{-}m^2$ No.3: m_d=3.93 kg/$(J\text{-}I)_d$=0.0101 $kg\text{-}m^2$
Bearing	No.1: MR-SFD No.2: k=2.05×10^8 N/m (roller bearing) No.5: k=1.87×10^8 N/m (roller bearing)
Speed	35,600 rpm at full speed
HP spool data	
Shaft	ρ=7.83×10^3 kg/m^3, E=1.79×10^{11} N/m^2, l=363.6 mm, m=15.39 kg $D_{o,max}$=100 mm/$D_{o,min}$=21 mm, $D_{i,max}$=70 mm/$D_{i,min}$=26 mm
Disk	No.1: m_d=5.60 kg/$(J\text{-}I)_d$=0.010 $kg\text{-}m^2$ No.2: m_d=6.54 kg/$(J\text{-}I)_d$=0.013 $kg\text{-}m^2$
Bearing	No.3: k=1.00×10^8 N/m (ball bearing) No.4: k=0.50×10^8 N/m (roller bearing, interconnected with LP spool)
Speed	42,000 rpm at full speed

Note that the LP spool has a slender shaft so that the rotor has to pass through several flexible critical speeds. The oil-lubricated conventional SFD that is originally installed at bearing No.1 is usually designed to effectively attenuate the unbalance response associated with, not the whole modes of interest, but the fundamental mode. Thus, the unbalance response of the rotor may not be properly controlled by a single conventional SFD beyond the first critical speed. Hence, a semi-active MR-SFD capable of changing its dynamic characteristics is attempted to improve the stability of the flexible rotor.

1.2 MR-SFD

Unlike the conventional SFDs, MR-SFDs are utilizing MR fluids of which the apparent viscosity varies in response to the applied magnetic field: contrary to Newtonian fluid, a yield stress develops and monotonically increases as the strength of applied magnetic field increases. Kim, *et al*. [4] presented the design procedures for the MR-SFD as well as the experimental method to derive its empirical dynamic stiffness model.

Figure 2 depicts the prototype design of the MR-SFD that was presented in [4]. The magnetic circuit analysis was performed to determine a proper design avoiding magnetic saturation and its feasibility was proved via FEM analysis. Optimal values for some crucial design parameters, such as radial clearance and seal's gap width, were experimentally explored and reflected in the prototype design. Bellows-shaped silicone sealing was also used to enhance the practicality of the MR-SFD.

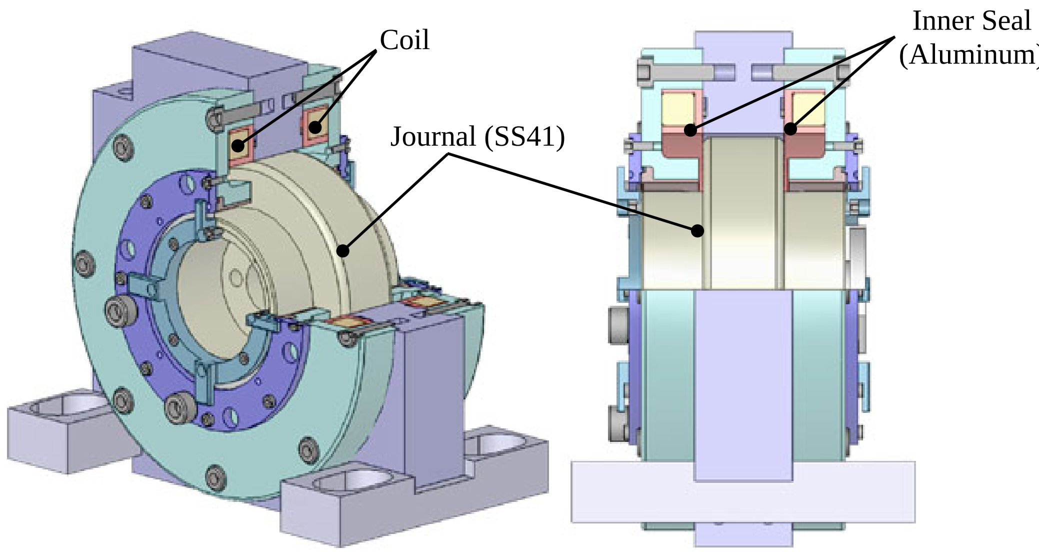


Fig. 2. Cross-sectional view of the MR-SFD [4].

Using an active magnetic bearing (AMB) system [3] as an exciter, experiments were extensively carried out, and the dynamic stiffness model for the MR-SFD was obtained and modeled as:

$$\begin{aligned}\mathrm{Re}\left(F_X / X\right) &= K\left(I_\mathrm{c}\right) - M\left(I_\mathrm{c}\right)\Omega^2 \\ &= 0.056 + 1.498 \cdot I_\mathrm{c}^2 - \left(7.19\times10^{-6} - 3.13\times10^{-6}\cdot I_\mathrm{c}\right)\Omega^2 \quad [N/m]\end{aligned} \tag{1}$$

$$\begin{aligned}\mathrm{Im}\left(F_X / X\right) &= \Gamma\left(I_\mathrm{c}\right) + C\left(I_\mathrm{c}\right)\Omega \\ &= 0.075 + 2.625 \cdot I_\mathrm{c}^2 + \left(1.26\times10^{-3} + 3.98\times10^{-3}\cdot I_\mathrm{c}\right)\Omega \quad [N/m]\end{aligned} \tag{2}$$

Here, M, C, and K denote the mass, damping, and stiffness coefficients, respectively. Γ is the hysteresis damping. I_c and Ω, respectively, represent the input current and rotational speed. Referring to Eqs. (1) and (2), one can find that only direct terms are considered since the MR-SFD is characterized by its non-rotating journal. It is also found that the dynamic characteristics of the MR-SFD can be changed by the input current level I_c.

2. FINITE ELEMENT MODEL FOR DUAL ROTOR

As shown in Fig. 1, the high-speed gas turbine system is composed of several mechanical components including rigid disks, bearings, damper, and shaft segments. Two procedures are commonly used for the FEM analysis of free and forced response of the rotordynamic system. These are the direct stiffness method (DSM) and the transfer matrix method (TMM). In this work, the FE model of the dual rotor system is made by means of the DSM.

Referring to [5], the system equations of motion including gyroscopic effects at a rotational speed Ω, which is equal to the whirl frequency, can be expressed as:

$$\mathbf{M}\ddot{\mathbf{q}} + \mathbf{C}(\Omega)\dot{\mathbf{q}} + \mathbf{K}\mathbf{q} = \mathbf{f}(t) - \mathbf{f}_\mathrm{c}(t, I_\mathrm{c}) \tag{3}$$

Here, the state vector **q** includes the translational and angular displacements at each of the nodes for the HP and LP spools together, and the states at the cross-coupled nodes are also defined in the vector. The symmetric system mass matrix **M** includes the inertias of the rigid disks and flexible shaft elements. The damping matrix **C**(Ω) includes the dissipation effect from the bearings or damper and the rotational-speed-dependent gyroscopic effect. The stiffness matrix **K** consists of the stiffnesses of the undamped shaft elements and the bearings. The force vector **f** and $\mathbf{f}_c$ represent the unbalance forces and control inputs to the MR-SFD, respectively. For the FE model of the target dual rotor system, a total of 68 degrees of freedom with 32 elements is considered (18 sub-elements for the LP spool and 14 sub-elements for the HP spool).

3. UNBALANCE RESPONSE CONTROL ALGORITHM

Based on the FE model (3), we want to determine a control force in order to attenuate the synchronous rotor vibrations caused by unknown distributed imbalance force along the length of the flexible rotor. Here, without loss of generality, Eq.(3) can be reduced, in the frequency-domain, to:

$$\left\{\mathbf{K}-\Omega^2\mathbf{M}+j\Omega\mathbf{C}(\Omega)\right\}\mathbf{Q}(j\Omega)=\mathbf{F}(j\Omega)-\mathbf{F}_\mathrm{c}(j\Omega,I_\mathrm{c}) \tag{4}$$

where

$$\mathbf{F}_\mathrm{c}(j\Omega,I_\mathrm{c})=\left[\mathbf{K}_\mathrm{c}(I_\mathrm{c})-\Omega^2\mathbf{M}_\mathrm{c}(I_\mathrm{c})+j\left\{\mathbf{\Gamma}_\mathrm{c}(I_\mathrm{c})+\Omega\mathbf{C}_\mathrm{c}(I_\mathrm{c})\right\}\right]\mathbf{Q}(j\Omega) \tag{5}$$

which means the control force vector exerted by the MR-SFD.

According to Eqs.(4) and (5), unbalance responses can be easily predicted as:

$$\begin{aligned}\mathbf{Q}(j\Omega)&=\left[\mathbf{K}+\mathbf{K}_\mathrm{c}(I_\mathrm{c})-\Omega^2\left\{\mathbf{M}+\mathbf{M}_\mathrm{c}(I_\mathrm{c})\right\}+j\mathbf{\Gamma}_\mathrm{c}(I_\mathrm{c})+j\Omega\left\{\mathbf{C}+\mathbf{C}_\mathrm{c}(I_\mathrm{c})\right\}\right]^{-1}\mathbf{F}(j\Omega)\\&\triangleq\mathbf{R}(j\Omega,I_\mathrm{c})\cdot\mathbf{F}(j\Omega)\end{aligned} \tag{6}$$

where, $\mathbf{R}(j\Omega,I_c)$ is the receptance matrix. Here, if *a priori* knowledge on unbalance force $\mathbf{F}(j\Omega)$ is available, one can determine the level of input current, which can minimize the unbalance response $\mathbf{Q}(j\Omega)$ without difficulty. However, it is impractical to assume that the magnitude and phase of mass imbalance distribution of a rotor are completely given and, moreover, it is also impossible to identify the mass imbalance distribution with a semi-active actuator.

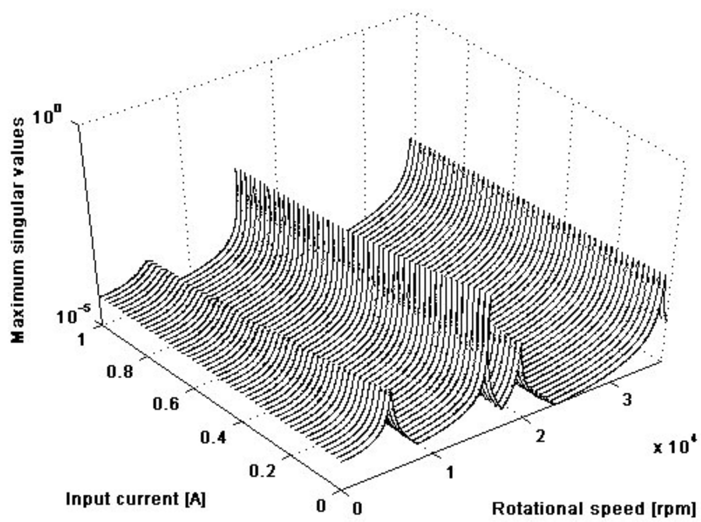
In order to resolve this problem, a robust design concept is utilized [8,9]. If one can imagine a matrix as a system with $\mathbf{F}(j\Omega)$ as its input and $\mathbf{R}(j\Omega,I_c)\cdot\mathbf{F}(j\Omega)$ as the output, then 2-norm can be interpreted as a gain. Moreover, the largest singular value among $\sigma(\mathbf{R})$ corresponds to the maximum gain. Therefore, the control strategy, which can make the total system less sensitive to unbalance forces, can be found in minimizing the largest singular value as:

$$I_\mathrm{c}^\mathrm{opt}(\Omega)=\min_{I_\mathrm{c}}\left\{\max\left(\sigma(\mathbf{R})\right)\right\},\quad \forall I_\mathrm{c}\in\left[I_\mathrm{min},I_\mathrm{max}\right] \tag{7}$$

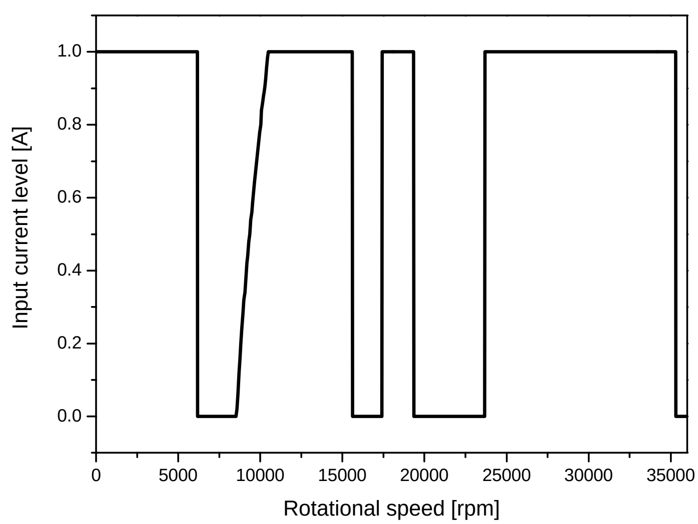
This control algorithm is so mathematically simple and physically reasonable that it can be applied to other semi-active control problems.

4. NUMERICAL ANALYSIS

Based on the FE modeling method and the proposed control algorithm, unbalance responses of the dual rotor are numerically investigated. To determine the optimal input current level at each rotational speed of interest, the maximum singular value analysis is carried out with the FE model of the dual rotor system. Figure 3a shows the distribution of the calculated maximum singular value from which four distinctive peaks are observed, especially for the input current equal to zero, over the operating speed range. It indicates that the dual rotor is characterized by the presence of four critical speeds associated with four lightly damped modes. From the singular value analysis results, the optimal input current levels, which can improve the dual rotor's unbalance responses, can be scheduled according to the control law (7) as plotted in Fig. 3b.



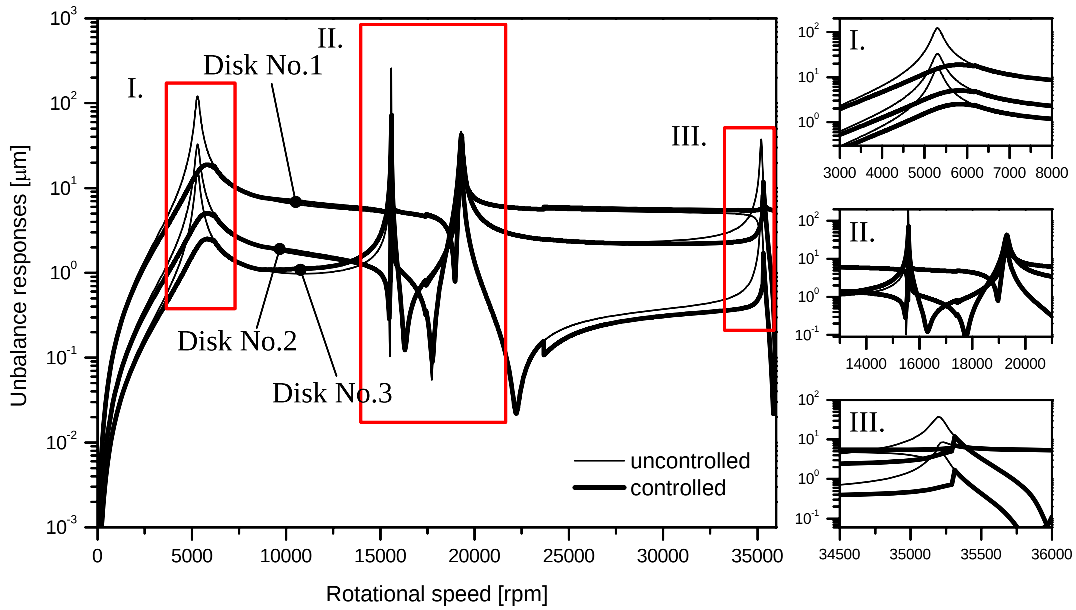
(a) Maximum singular value distribution



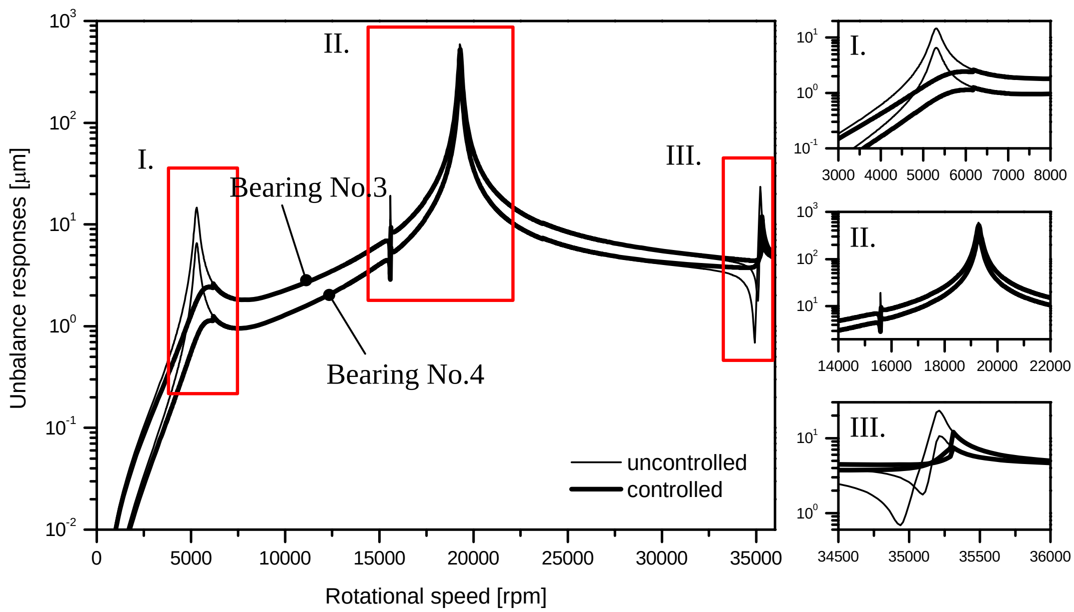
(b) Scheduled input current level

Fig. 3. Scheduling input current level for the high-speed gas turbine system.

With the scheduled input current levels to the MR-SFD, the controlled unbalance responses at the disk locations of the high-speed gas turbine system are predicted assuming that the mass imbalances only exist at the disks.



(a) LP spool



(b) HP spool

Fig. 4. Controlled unbalance responses for the high-speed gas turbine system.

In Fig. 4, the unbalance responses when the MR-SFD is turned off (i.e. the input current equals to zero) are plotted with thin lines. As the singular value analysis implies, there are four distinctive peaks in each unbalance response curve. Note that the third highest mode near 19,000 rpm is found to be not a flexible mode of the LP spool but it reflects one of the flexible modes of the HP spool.

Figure 4 also shows the results of the controlled unbalance responses. When applying the scheduled input currents, the unbalance response levels predicted at each disk are reduced by, at least, 70%. Therefore, it can be concluded that the stability of the dual rotor system is significantly improved by the introduction of the MR-SFD. However, the vibrations caused by the HP spool's flexible mode are not effectively attenuated by the control action of the MR-SFD.

CONCLUSIONS

In this paper, the unbalance responses of the high-speed gas turbine system are analyzed and attenuated by applying the semi-active MR-SFD in lieu of a conventional SFD. Based on the empirical dynamic model of the MR-SFD, the FE model of the dual rotor system is derived and analyzed. Considering the dynamic characteristics of the semi-active actuator, a simple yet effective control law is proposed from the singular value analysis and the optimal input current levels are scheduled. Simulation results clearly demonstrated that the proposed MR-SFD can significantly improve the unbalance responses of the flexible rotating machinery and the effectiveness of the proposed control algorithm is also proved.

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