Selection of the hydrodynamic damper type for the turbomachine rotor

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Abstract. This work discusses a number of issues related to selection of the hydrodynamic damper type widely spread today for damping turbomachinery rotor oscillations. The analysis of works dedicated to this issue has been performed and it has been shown that today there is a large variety of damper designs. Therefore, the method allowing to reasonably select the type of dampers and their parameters is required. It contains the methodology of designing such dampers consisting of the stages of parameter optimization of the "rotor-damper" system, damper structural optimization and selection of optimal parameters of dampers of the type selected.
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Introduction

Today it is considered to be proved that most oscillation defects can be successfully eliminated through proper damping of oscillations of aircraft engine parts and mounts. These mounts include rotors, pipelines, wheel blades and compressor and turbine stators, containments of bodies and equipment [1]. Dry friction dampers [1,2] and hydrodynamic dampers [3,4] are used today. A large number of works are devoted to hydrodynamic dampers the detailed analysis of which is performed in a monograph [4]. The especially notable monograph is the one by Sergeev S. I. [5] in which all the basic formulas for hydrodynamic forces have been obtained. It is also necessary to mention the work [6] in which the possibility of existence of the failure mode in operation of a rigid rotor with hydrodynamic dampers in supports was first shown. Rip flows in a clearance and operation of a damper with various seals have been studied in works [7-9]. Efficiency of damper operation is provided through optimization of it parameters in the "rotor – support" system. Otherwise it can worsen the rotor oscillation condition. To perform design works it is necessary to have relevant means – design methodology, software and database with information about dampers of various types. Therefore, the problem of selection of the hydrodynamic damper type for the rotor depending on its operating conditions is described in this article. The diagram of a damper with indications is shown in Fig. 1.

Hydrodynamic force components - radial - F_R and tangential - F_t – are directed along the center line and along normal to it. The type of these forces depends on the damper type and the assumed calculation method. The value of forces depends on damper geometry – length L, diameter D and clearance b, operating mode – precessional frequency w and damper process fluid properties - viscosity m .

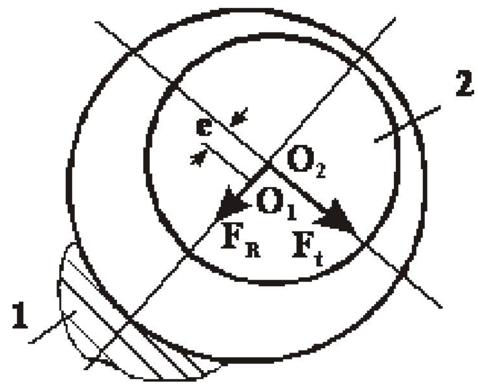

Fig. 1. Hydrodynamic damper diagram:

1 – stator; 2 – oscillator; O_1 – stator geometric center;

O_2 – oscillator geometric center

If we consider the linear problem the tangential force can be equated to the damping force and expressed as follows

$F_t = dV$,

where d - damping coefficient, $V=ew$ – linear speed of oscillations, e- precession radius or oscillation amplitude.

Methodology for designing hydrodynamic dampers.

The methodology consists in choosing necessary methods for damper calculation and forming the sequence of actions leading to the main result – determining the optimal design with its parameters.

There are three tasks (Fig. 2). The first one is determining the damping level in supports. At that, it is necessary to perform the analysis of dynamics of the rotor with hydrodynamic dampers in supports. This stage can be referred to parameter optimization of the "rotor – support" system. The second one is selecting the damper type.

This task refers to the field of structural design. And finally, the third task is determining parameters of the selected damper type.

To select the required damping value it is necessary to calculate the amplitude-frequency characteristic the type of which essentially depends on actuating forces. The calculation method used depends on the stage of work on engine designing. At early stages it is recommended to use the simplified calculation method for a rigid rotor described in works [4,6,10-12].

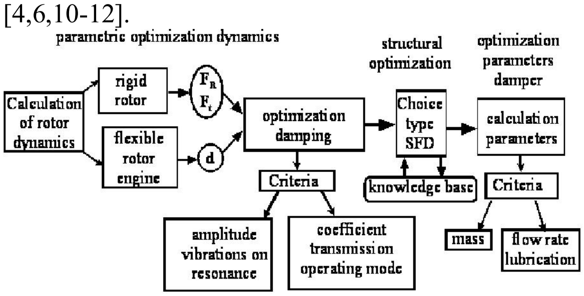

Fig. 2. Methodology for damper designing

As a result, damper characteristics are defined – hydrodynamic force components F_R and F_t. In detailed designing at the detailed design stage and finishing it is necessary to use a more precise model of the rotor described, for example, in work [3]. To estimate actuating forces it is necessary to conduct an analysis of the design and power diagram of a rotor. In case of a composite rotor it is necessary to conduct an analysis of its components. As a result, places of disbalances introduction are determined and for composite rotors possible combinations of disbalances are estimated in order to assess the phase of actuating forces.

To optimize the parameters of the "rotor – support" system it is necessary to select the relevant criteria. As known [4], damping reduces the oscillation amplitude at resonance but after the resonance damping increases loads transferred to the body that are characterized by the transmission coefficient T which results in growth of loads on bearings and reducing their life.

Transmission coefficient T is equal to the ratio of the force transmitted to the body through the damper to forces from rotor unbalance. Therefore, we can review several damping optimization criteria.

The first one is introduction of such damping that provides the set oscillation amplitude at resonance determined by clearances in labyrinth sealing. The second criterion relates to selection of damping with minimum transmission coefficient T in the operating mode with the oscillation amplitude at resonance not exceeding the permitted value. It is also possible to select damping that ensures the permissible amplitude at resonance and permissible value of the transmission coefficient in the operating mode.

In this case the transmission coefficient for the set damping level d is defined using the known formula of the linear theory of oscillations [4]

$$T = \frac{g\sqrt{(ec)^2 + (ewd)^2}}{Gsw^2},$$

where G – rotor weight on the support, g – gravity acceleration, s – residual rotor unbalance, c – damper spring stiffness.

The force transmission coefficient can be calculated for any rotor point and any support. However, mounts where large dynamic disturbances are most likely to occur are of greatest interest. Coefficients of force transmission to the body at operating frequencies should be preferably less than one. In critical frequencies the transmission coefficients can reach the significant value which is quite acceptable as transition over the critical frequency is performed rather fast. The stage of parameter dynamics optimization finishes with defining the parameters of damping for the rotor system (Fig. 2). The approaches used to select the parameters of hydrodynamic devices have been successfully applied for face seals [13-15].

Selection of the damper type and identification of parameters

Selection of the damper type is the stage of structural optimization – it is necessary to select the device structural diagram. For this purpose it is necessary to use the knowledge base that should be constantly filled in the process of work which is reflected in Fig. 2 with double arrows. This stage is currently insufficiently formalized and expert estimates can be used when taking a decision. To improve the knowledge base it is necessary to develop the principles of damper designing.

The third one, methodology final stage, relates to identification of damper parameters – geometry, lubricant properties and operating modes ensuring the damping level set earlier. Optimization criteria at that can be, for example, minimum weight, dimensions or lubricant consumption.

The analysis conducted in work [4] identifies the variety of damper types but the question of selection of the damper type remains open. We consider that solution of this problem should be started with the issue of using the spring element as it significantly complicates the damper design. Spring elements are used for the following factors:

- frequency detuning
- rotor axial force sensing in case of a radial-axial bearing
- perception of the rotor weight

To define spring stiffness in the first case it is necessary to calculate the rotor natural frequency spectrum and select stiffness so that natural frequencies are taken from the working range.

In the second case (when using a radial-axial bearing) it is recommended to mandatory use the spring element. This allows to stabilize damper characteristics in operation. Otherwise due to the axial force at damper faces the dry friction force appears that depends on the axial force value and can greatly change depending on the engine operating condition. The stiffness value is also selected from the condition of providing frequency grading.

In the third case it is required to prove the need of using the spring element through solution of the task of oscillations of a rigid unbalanced rotor at dampers taking into account the rotor weight. The method for this calculation is described in work [16]. Calculations start with zero spring stiffness (Fig.3). If after solution it emerges that the rotor weight is too large and the movement orbit is unstable, stiffness is introduced – the spring element or unloading device from the rotor weight. Simultaneous use of the unloading device and the spring element can also take place.

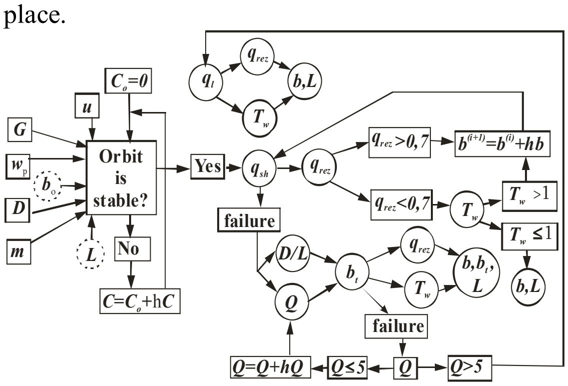

Fig. 3. Algorithm of selection of the hydrodynamic damper type

Basic types of hydrodynamic dampers used today are short dampers (with or without seals), long dampers and dampers with face gaps. For proper selection of the damper type it is necessary to assess its impact on rotor system dynamics. For this purpose it is necessary to calculate the rotor dynamics by the simplified method described in works [4,10,11]. Such method is used for a rigid rotor. Simplification at this stage is justified as it allows to significantly accelerate preparation of initial data for calculations. At that the set values are rotor weight *G* on the support, disbalance *u*, stiffness *c* of the support spring element defined at the previous stage, working rotation speed w_p, oscillator diameter *D*, lubricant dynamic viscosity *m*. It is necessary to define the type of a damper and its geometry – length and clearance.

First, it is necessary to assess the possibility of use of a short damper without seals as it is the one with the simplest design. The damper length should be maximally admissible for this support design. The damper clearance value *b* should be defined using the method of sequential approximations. First, taking the clearance b_0=0,1mm (minimum clearance that can be technically provided) we shall calculate the relative oscillation amplitude q_{sh}, where index *sh* means "short". If after calculations it emerges that the damper operates without failures, after optimization the length and clearance in the damper are finally defined. The algorithm shown in Fig. 3 presents optimization by a permissible amplitude at resonance ($q_{rez}<0,7$) and ensuring conditions $T_p<1$.

The case is possible when the damper works without failure but the transmission coefficient *T*>1, i. e. the damper does not reduce forces transferred to the body. This fact indicates the high level of relative disbalance. It can be reduced through damper clearance increasing. Therefore, having increased it by the value of a clearance step *hb* it is necessary to repeat calculation. If the value $T_p<1$ is not reached it is necessary to change the damper type. The damper type should be also changed in case of failure modes, i. e. when damping in the system is insufficient.

The next damper type is the short one with seals. It has damping 4 time larger than the damper without seals [17]. Therefore, having increased damping by 4 times it is necessary to repeat calculations based on the method described.

If use of a short damper with sealing rings also does not create a sufficient level of damping it is possible to use the damper with face gaps which gives 10...15 time larger damping than the short one. We consider that the short damper with seals is less successful in design than the damper with face gaps which is simpler in design and has larger damping. It is characterized by the non-dimensional parameter Q

$$Q = \frac{1}{2}\left(\frac{b}{b_t}\right)^3 \ln\left(\frac{1}{1-2L_t/D}\right),$$

where b_t and L_t – axial clearance and face gap length respectively.

Retaining the same value of the damper clearance as in the short damper but reducing the value of the axial clearance b_t and the gap length L_t, i. e. increasing the parameter *Q*, it is possible to eliminate the failure and then to optimize the damper. The parameter *Q* changes with its step *hQ*. However, it is necessary to consider that in gas turbine engine supports $Q<5$ [18] . With $Q>5$ it is necessary to proceed to calculation of a long gas turbine engine, i. e. to set seals without using a feeding groove. Complication is justified in this case as it provides significant increasing in damping (100...300 times). Optimization stages are not reflected in Fig.3 in branches relating to calculation of a damper with a face gap and the long hydrodynamic damper, as they are similar to calculation stages of a short hydrodynamic damper.

Conclusion

The analysis of hydrodynamic damper designs have been conducted and it has been shown that today there is a large variety of such devices. Therefore, the method of justified selection of a damper type is required.

The methodology for designing such dampers has been developed which consists of the stages of parameter optimization of the “rotor-damper” system, structural damper optimization and selection of optimal parameters of dampers of a selected type.

The selection method for four types of dampers has been suggested – a short damper with and without seals, a long damper and a damper with face gaps. It has been shown that for all the above dampers there is a failure operating mode depending on the value of disbalance in the system.

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