Для поглинання потужності, що виробляється вільною турбіною, при наземних випробуваннях авіаційних турбовальних газотурбінних двигунів застосовуються гідрогальмові установки різних конструкцій. Наземні випробування авіаційних турбовальних газотурбінних двигунів із застосуванням таких гідрогальм можуть призводити до нерозрахованих режимам роботи системи автоматичного управління двигуном в області роботи регулятора обертів вільної турбіни. Основною причиною не розрахункової роботи системи автоматичного управління є невідповідність завантажувальних характеристик гідрогальм завантажувальним характеристикам повітряного гвинта, що приводиться в обертання вільною турбіною двигуна.

Представлені експериментальні завантажувальні характеристики гідрогальма і несучого гвинта вертольоту показують суттєву різницю даних характеристик за величиною коефіцієнта посилення. Для усунення даної відмінності розглядається можливість моделювання динамічних параметрів несучих або гребних гвинтів простими засобами автоматизації. Для рішення даного завдання було розроблено лінійну математичну модель та структурну схему автоматизованої системи управління гідрогальмом, призначеної для наземних випробувань турбовальних газотурбінних двигунів. Обґрунтовано закон регулювання завантаженням гідрогальма. Представлена структурно-динамічна схема автоматизованої системи управління, яка розробляється, і наведені розрахункові формули для визначення параметрів регулятора. Виконано розрахунки перехідних характеристик гідрогальмової установки без засобів автоматизації та із застосуванням автоматизованої системи управління завантаженням. Результати розрахунків, що представлені, показують, що застосування засобів автоматизації дозволяє повністю емулювати характеристики несучих гвинтів вертольотів

Ключові слова: автоматизація гідрогальм, завантажувальна характеристика, динамічні параметри, закон регулювання, перехідні характеристики

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DEVELOPMENT OF AN AUTOMATED HYDRAULIC BRAKE CONTROL SYSTEM FOR TESTING AIRCRAFT TURBOSHAFT GAS TURBINE ENGINES

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1. Introduction

Unsatisfactory operation of systems for automatic regulation of free turbine rotation speed is often observed in ground tests of turboshaft gas turbine engines. It manifests itself as steady-state fluctuations of engine parameters. In some cases, continuous increase in fluctuation amplitude up to triggering of automatic protection systems is observed.

When engine is running as a component of a helicopter power system, a free turbine drives the helicopter rotor (HR). In this case, the free turbine (FT) with a connected rotor (FT+HR) is the object of control.

Ground tests of helicopter turboshaft gas turbine engines (GTE) with a free turbine are most often performed on hydraulic brake units to absorb (utilize) power generated by the FT. In this case, a free turbine with a connected hydraulic brake (FT+HB) is the object of control.

Hydraulic brake units are designed for a certain consumed power and loading characteristics. Inertial properties of the hydraulic brake are determined by its design features.

It should be noted that loading characteristics and inertial properties significantly differ from loading characteristics and inertial properties of the HR. This difference can lead to unsatisfactory operation of the free turbine speed regulator (FTSR) during ground tests of the engine in conjunction with HB.

It is a very laborious and expensive task to change HB design in order to change its characteristics. Therefore, it is relevant to search for solutions that will provide emulation of specified characteristics with simple automation tools.

2. Literature review and problem statement

Continuous improvement of aviation turboshaft gas turbine engines requires the same continuous improvement of HB units for ground testing both by developing and introducing new units [1] and upgrading existing ones.

The issue of development and use of air brakes for ground testing of gas turbine engines is considered in [1]. Calculations of power absorbed by an air brake are presented. However, an air brake is considered not as a way to change dynamic parameters of the brake unit but as a means of reducing the cost of developing brake units by using low-pressure compressors of existing gas turbine engines. The problem of improving dynamic characteristics of the brake unit through the use of a compressor remain unsolved.

The use of automated control systems in ground tests of aircraft gas turbine engines is one of new ways to improvement of test benches. When developing automated control systems for ground testing of aircraft gas turbine engines, main attention is paid to automated data acquisition and processing systems [2–4].

Issues of automating tests of aircraft gas turbine engines based on LabView SCADA system are considered in [2]. Possibility of integrating the test results obtained using the LabView SCADA system and DVIGwp simulation system is shown. Capabilities of a new automated information acquisition and measuring system described in [3] make it possible to conduct real-time processing and transmission of data. Results of development and application of an intelligent test bench for testing aircraft gas turbine engines are presented in [4]. Practical application of the results obtained in [2–4] to automation of test benches can significantly reduce time and material expenditures in ground engine tests. However, possibilities of automating tests of aircraft gas-turbine engines are considered in these works as a means of measuring parameters in steady-state engine operating conditions. Capabilities of automation to control braking torque have not been considered.

Automation of test benches taking into account dynamics of the processes occurring in testing engines on hydraulic brake units is the subject of works [5–7]. Methods for controlling stability of measurement systems during excitation of shaft oscillations are considered in [5]. Objective difficulties when measuring reactive moment of a hydraulic brake are shown. General approaches to provision of stability of the system for measuring reactive moment of a HB under dynamic loads are considered. One of the ways to overcome difficulties in measuring reactive moment of a hydraulic brake with low-frequency shaft oscillations was proposed in [6]. This paper discusses an adaptive system for measuring reactive moment of a hydraulic brake with low-frequency shaft oscillations by introducing time delays into the measurement system. Design of a hydrodynamic dynamometer for measuring reactive moment of a hydraulic brake under impact loads is considered in [7].

The HB shaft dynamics (natural and forced vibrations) is considered in these papers. Automation tools are used there to ensure the measuring system stability under dynamic loads. Dynamics of automated system s for regulation of FT shaft speed near the equilibrium point was not considered.

Causes of low-frequency oscillations taking place in testing engines with the use of hydraulic brake units are considered in [8, 9]. A model of calculating excitation of low-frequency oscillations in a system for measuring reactive moment by high-frequency oscillations of fluid flow pressure is presented in [8]. However, this model does not explain occurrence of self-oscillations of the engine parameters or unstable operation of the engine during operation of automatic FT rotational speed control system and absence of fluctuations in operation of the turbocompressor rotational speed regulator. Experimental and theoretical studies of the oscillatory processes arising in tests of high-power turboprop engines on a hydraulic brake unit are presented in [9]. It was shown that one of the causes of fluctuations in the FT rotational speed consists in entering of a non-stationary or two-phase (cavitation) flow in the gas turbine. It is also noted in [9] that a possible cause of occurrence of low-frequency fluctuations in the FT rotational speed consists in interaction of the engine and the HB but these causes could not be fully identified.

Issues of design and manufacture of hydraulic brake test benches for testing various types of engines were considered in [10, 11]. Recommendations are given in [10] for choosing parameters of newly designed hydraulic brake units. However, methods for changing loading characteristics and their adaptation to characteristics of real objects are not considered. Paper [11] raises problems of creating latest hydraulic brake units with ability to emulate characteristics of ship and aircraft screw propellers. However, concrete data on hydraulic brake control methods, calculation results or experimental data are not provided.

As the published data analysis has shown, issues of modeling dynamic characteristics of real objects using hydraulic brake units by automation means have not been given a due attention. Objectively, this is caused by the fact that different objects are tested on hydraulic brake units including those that are not prone to development of oscillations. For such complex objects as turbo-shaft gas turbine engines, issues of interaction of an automatic engine control system with a hydraulic brake unit are extremely acute. The problem consists in that parameters of the free turbine speed regulator (FTSR) are selected in accordance with the HR parameters and do not correspond to parameters of the hydraulic brake used in ground tests. The problem can be solved by means of a system of automated HB rotor torque control which provides full emulation of the HR dynamic parameters.

The above-mentioned allows us to conclude that it is advisable to conduct studies on development of automated systems for controlling loading of hydraulic brake units.

3. The aim and objectives of the study

The study objective: development of an automated control system (ACS) for a hydraulic brake unit designed for ground testing of turboshaft gas turbine engines. Automation of hydraulic brake units will eliminate discrepancy between dynamic parameters of hydraulic brakes and helicopter rotors and ensure correct operation of the system for automatic regulation of speed of free turbines during ground engine tests.

To achieve this objective, it is necessary to solve a series of tasks:

– develop a block diagram of the developed ACS;

– substantiate the law of regulation of HB loading;

– provide mathematical description of the study object;

– conduct computational studies of dynamics of the developed HB ACS.

4. Mathematical description of the study object

The study object: an automatic regulation system (ARS) for regulation of rotational speed of a free turboshaft engine turbine including:

– a dynamic model of an FT rotor with HB (FT+HB);
– a dynamic model of an FT rotor with HR (FT+HR);
– a dynamic model of the HB regulator.

Let us consider an equation of dynamics of a working rotor [12] of a free turbine with an HB:

$$\frac{\pi I_{(FT+HB)}}{30}\frac{dn_{FT}}{dt}=M_{FT}-M_{HB}; \qquad (1)$$

where $I_{(FT+HB)}$ is reduced moment of inertia of the free turbine rotor and the hydraulic brake rotor; n_{FT} is the rotational speed of the FT rotor; M_{FT} is FT torque; M_{HB} is the moment of resistance of the HB.

Equation of dynamics of a working rotor of a free turbine with HR:

$$\frac{\pi I_{(FT+HR)}}{30}\frac{dn_{FT}}{dt}=M_{FT}-M_{HR}; \qquad (2)$$

where $I_{(FT+HR)}$ is the reduced moment of inertia of the free turbine rotor and the helicopter rotor; M_{HR} is the moment of resistance of the HR.

In general terms, torques of the free turbine, HB and HR are complex functions:

$$M_{FT}=M_{FT}\left(n_{FT};n_{TC};P_I;T_I\right); \qquad (3)$$

$$M_{HB}=M_{HB}\left(n_{FT};\alpha_{FT};t_W\right); \qquad (4)$$

$$M_{HR}=M_{HR}\left(n_{FT};\phi_{HR};P_I;T_I\right); \qquad (5)$$

where n_{TC} is the rotational speed of the engine turbocompressor (TC) rotor; P_I is the total air pressure at the engine inlet; T_I is the total air temperature at the engine inlet; t_W is the water temperature at the HB inlet; t_I is the position of the valve that controls water flow through the HB; φ_{HR} is the HR blade angle.

Assume for simplicity that the engine ground tests are conducted in a standard atmosphere. This will allow us to neglect influence of variation of external conditions $\left(P_I;T_I;t_W\right)$ at the initial stage.

Taking into account the above assumptions, equation of the working FT rotor in a linear statement and in relative variables for (FT+HB) will be as follows [4]:

$$T_{FT}\dot{\bar{n}}_{FT}+\bar{n}_{FT}=K_{FT/n_{TC}}\bar{n}_{TC}+K_{HB/\alpha_{HB}}\bar{\alpha}_{HB}; \qquad (6)$$

where T_{HB} is the time constant of the rotor (FT+HB), s; $\bar{n}_{FT}$ is the relative rotational speed of the FT rotor;

$$\dot{\bar{n}}_{FT}=\frac{d\bar{n}_{FT}}{dt}$$

is relative rate of variation of rotational speed of the FT rotor;

$$K_{HB/n_{TC}}=\frac{\Delta\bar{n}_{FT}}{\Delta\bar{n}_{TC}}$$

is coefficient of effect (gain) of the turbocompressor rotational speed on the FT rotational speed;

$$K_{HB/\alpha_{HB}}=\frac{\Delta\bar{n}_{FT}}{\Delta\bar{\alpha}_{HB}}$$

is coefficient of effect (gain) of the HB control valve on FT rotational speed.

Equation in a linear object statement (FT+HR):

$$T_{HR}\dot{\bar{n}}_{FT}+\bar{n}_{FT}=K_{HR/n_{TC}}\bar{n}_{TC}+K_{HR/\phi_{HR}}\bar{\phi}_{HR}; \qquad (7)$$

Values of the coefficients of gain $K_{HR/n_{TC}}$ and $K_{HR/\phi_{HR}}$ can be determined from loading characteristics of the HR and the HB (Fig. 1):

$$K_{HR/n_{TC}}=\frac{\Delta\bar{n}_{FT}}{\Delta\bar{n}_{TC}},\quad K_{HB/n_{TC}}=\frac{\Delta\bar{n}_{FT}}{\Delta\bar{n}_{TC}}. \qquad (8)$$

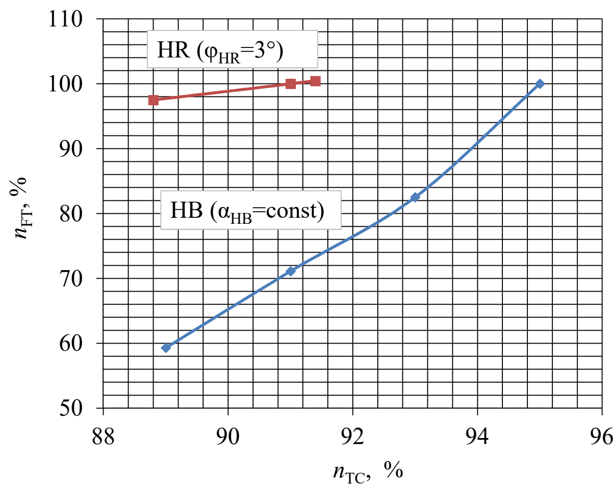

Fig. 1. Variation of rotational speed of the TV3-117 free turbine when working with HR and HB

Fig. 1 shows that the (FT+HB) object reacts much "sharper" to the change of the engine rotational speed (power) than the (FT+HR) object. That is, increment of the FT revolutions at the same variation of the TC revolutions is greater in the hydraulic brake than that in the HR:

$$K_{HB/n_{TC}}>K_{HR/n_{TC}}. \qquad (9)$$

As the analysis has shown, equations of dynamics of (FT+HB) and (FT+HR) are identical in structure but their dynamic parameters differ. Therefore, we can conclude that to maintain similarity of dynamic processes, it is enough to observe equalities:

$$T_{HB}=T_{HR}, \qquad (10)$$

$$K_{HB/n_{TC}}=K_{HR/n_{TC}}. \qquad (11)$$

5. Substantiation of the law of regulation by HB loading

Based on mathematical description of elements of the automated system for controlling HB loading, it is possible to compose a structural and dynamic diagram of the ARS(n_{HB}), Fig. 2.

It is necessary to find transfer function of the regulator that provides dynamic parameters of the ARS(n_{HB}) close to dynamic parameters of the object (FT+HR).

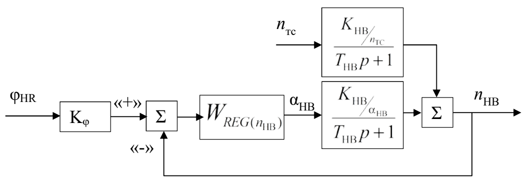

Fig. 2. Structural and dynamic diagram of the automated regulation system ARS(n_{HB})

When choosing transfer function of the regulator, the following should be considered. General form of equation of ARS(n_{HB}) dynamics should correspond to the general form of equation of the HR dynamics. In addition, to model dynamic parameters of various propellers, it is necessary to provide independent effect both on the gain coefficient and the time constant oчf the ARS(n_{HB}).

When frequency of the TC rotation changes, transfer function of ARS(n_{HB}) has the form:

$$W_{ARS(n_{\text{HB}})} = \frac{K_{\text{HB}/n_{\text{TC}}}}{(T_{\text{HB}}p+1)+W_{REG(n_{\text{HB}})}\cdot K_{\text{HB}/\alpha_{\text{HB}}}}. \qquad (12)$$

In general case, transfer function of the regulator takes the form:

$$W_{REG(n_{\text{HB}})} = \frac{A_{REG}(p)}{L_{REG}(p)}. \qquad (13)$$

where $A_{REG}(p)$ is polynomial of numerator of transfer function of the regulator; $L_{REG}(p)$ is polynomial of denominator of transfer function of the regulator; p is Laplace operator.

It follows from analysis of the transfer functions $W_{REG(n_{\text{HB}})}$ and $W_{ARS(n_{\text{HB}})}$ that:

a) in order to preserve form and order of the ARS(n_{HB}) dynamics equation corresponding to the HR equation, characteristic transfer function polynomial of denominator of the regulator should tend to unity and numerator polynomial should not exceed the first order;

b) in order to provide an independent effect on the coefficient of gain and time constant of the ARS(n_{HB}), numerator of the transfer function of the regulator should have two independent coefficients.

Thus, transfer function of the regulator in a standard form should correspond to the real boosting corrector:

$$W_{REG(n_{\text{HB}})} = \frac{K_F(T_F p+1)}{T_{REG}p+1}, \qquad (14)$$

where K_F is the coefficient of gain of the regulator; T_F is the time constant of the boosting corrector; T_{REG} is the time constant of the regulator.

As noted above, time constant of the regulator should be a small quantity $T_{REG}\approx 0$. In this case, inertia of the regulator itself can be neglected. With this in mind, determine transfer function of the ARS(n_{HB}) with a boosting corrector when speed of the TC rotation changes:

$$W_{ARS(n_{\text{HB}})} = \frac{\dfrac{K_{\text{HB}/n_{\text{TC}}}}{1+K_F K_{\text{HB}/\alpha_{\text{HB}}}}}{\left(\dfrac{T_{\text{HB}}+K_F K_{\text{HB}/\alpha_{\text{HB}}}T_F}{1+K_F K_{\text{HB}/\alpha_{\text{HB}}}}\right)p+1}. \qquad (15)$$

Denote:

$$K_{ARS_{\text{HB}}} = \frac{K_{\text{HB}/n_{\text{TC}}}}{1+K_F K_{\text{HB}/\alpha_{\text{HB}}}}, \qquad (16)$$

$$T_{ARS_{\text{HB}}} = \frac{\left(T_{\text{HB}}+K_F K_{\text{HB}/\alpha_{\text{HB}}}T_F\right)}{1+K_F K_{\text{HB}/\alpha_{\text{HB}}}}. \qquad (17)$$

Transfer function is obtained in a standard form:

$$W_{ARS(n_{\text{HB}})} = \frac{K_{ARS_{\text{HB}}}}{T_{ARS_{\text{HB}}}p+1}. \qquad (18)$$

Proceeding from the above, it is possible to determine the regulator parameters that provide dynamic parameters of the stabilized HB the same as for the HR:

$$\frac{K_{\text{HB}/n_{\text{TC}}}}{1+K_F K_{\text{HB}/\alpha_{\text{HB}}}} = K_{\text{HR}}, \qquad (19)$$

$$\frac{\left(T_{\text{HB}}+K_F K_{\text{HB}/\alpha_{\text{HB}}}T_F\right)}{1+K_f K_{\text{HB}/\alpha_{\text{HB}}}} = T_{HR}. \qquad (20)$$

Having solved the resulting system of equations (20), (21), values of the regulator parameters for a concrete HR are obtained:

$$K_F = \left(\frac{\dfrac{K_{\text{HB}/n_{\text{TC}}}}{K_{\text{HR}/n_{\text{TC}}}}-1}{K_{\text{HB}/\alpha_{\text{HB}}}}\right), \qquad (21)$$

$$T_F = \frac{T_{\text{HR}}\left(1+K_F K_{\text{HB}/\alpha_{\text{HB}}}\right)-T_{\text{HB}}}{K_F K_{\text{HB}/\alpha_{\text{HB}}}}. \qquad (22)$$

Block diagram of an automated control system ACS(n_{HB}) for controlling the helicopter rotor speed is shown in Fig. 3.

The hydraulic brake is controlled by changing water flow rate at the inlet, $G_{\text{I FLOW}}$. In equilibrium modes, the HB rotation speed is stabilized by a closed automatic regulation system ARS(n_{HB}) according to the program:

$$\bar{n}_{\text{HB}} = \left(\bar{n}_{\text{HB}}\right)_{SET}. \qquad (23)$$

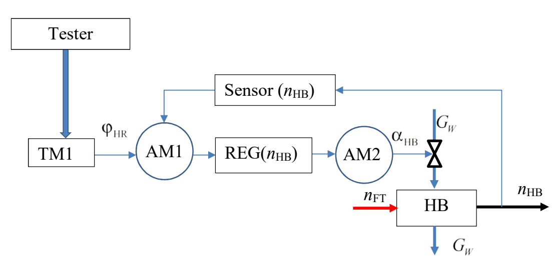

Fig. 3. Block diagram of the HB ACS

In this case, the tester controls the hydraulic brake not directly but through setting of the ARS(n_{HB}) simulating the specified HB blade angle.

6. Results obtained in the computational studies

Dynamics of the developed system of automated control of loading the hydraulic brake unit was studied for dynamic parameters of the HR and the HB close to real values.

Fig. 4 shows variation of rotational speed of the FT when working with the HR and non-stabilized HB with the following parameters:

$$K_{HR/n_{TC}} = 1.5, \quad T_{HR} = 2, s, \quad K_{HB/n_{TC}} = 5.9, \quad T_{HB} = 0.8, s.$$

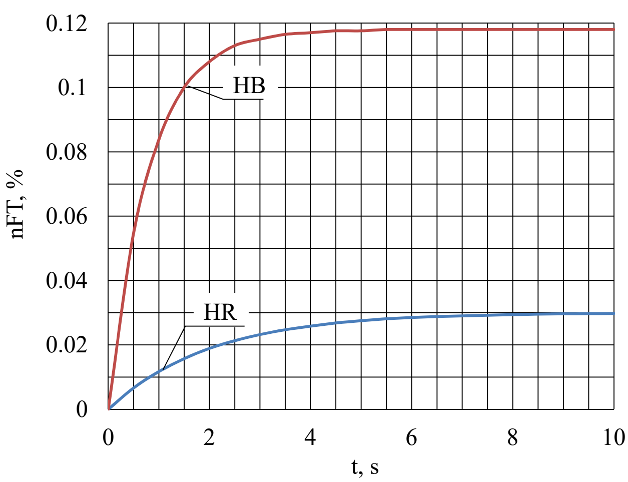

Fig. 4. Transitive process of the FT when working with the HR and non-stabilized HB

Fig. 4 shows that dynamics of the FT with non-stabilized HB does not correspond to dynamics of (FT+HR). A 4-fold difference between the HB and the HR rotational speeds in the transitive process can be reached.

Fig. 5 shows variation of rotational speed of the FT when working with the HR and stabilized HB. The regulator parameters calculated from formulas (22), (23) were taken as follows:

$$K_F = 0.978, \quad T_F = 2.41, s, \quad T_{REG} = 0.08 T_{HB}.$$

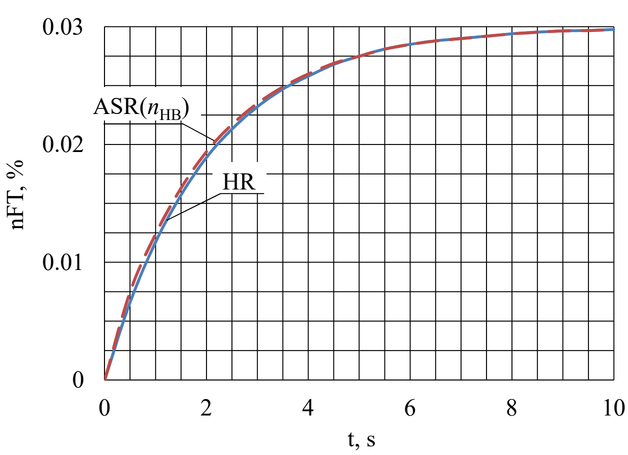

Fig. 5. Transitive process of the FT when working with HR and stabilized HB

As can be seen from Fig. 5, with correct selection of the ARS(n_{HB}) regulator parameters, automated hydraulic brake adequately reflects the HR dynamics. Difference between the stabilized HB and the HR rotational speeds did not exceed 1.5 % in the transitive process.

When changing dynamic parameters of the regulator, it is possible to ensure resemblance of dynamic tests with the HRs taken from different helicopters.

In a general case, it should be noted that equation of the HR dynamics depends also on external conditions. In order to take into account effect of external conditions on the HR characteristics, it is necessary to introduce appropriate corrections to the automated brake control system.

7. Discussion of results obtained in the computational study of transient characteristics of a hydraulic brake unit with automated loading control

When developing an ACS for the HB loading, correspondence of transient characteristics of the automated HB to transient characteristics of a real object (a helicopter rotor) is a criterion for fulfilling the task.

Studies of transient characteristics have shown that dynamics of the stabilized hydraulic brake corresponds quite well to the rotor dynamics (Fig. 5). The greatest difference was observed at the initial stage of the transitive process. It was decreasing over time. This is due to the peculiarity of dynamics of the boosting regulator and its inertia. It should be noted that difference between the transitive process increases continuously for the non-stabilized HB with respect to that for the HR (Fig. 4).

Thus, computational studies confirm correctness of the chosen method of hydraulic brake stabilization during ground tests of turboshaft engines.

The method proposed for the HB stabilization features application of the boosting law of load regulation (15). Typically, P, PI, or PID regulation laws are applied to stabilize object parameters. However, application of such laws contradict the condition (a) from Section 6.

Limitations of the method application:

1. From analysis of stability of the hydraulic brake ACS, dynamic parameters of the hydraulic brake itself have the following limitations:

$$K_{HB} > K_{HR}; \tag{13}$$

$$T_{HB} < T_{HR}\left(1 + K_F K_{HB/\alpha_{HB}}\right). \tag{14}$$

Failure to comply with these restrictions will inevitably lead to unstable operation of the ARS(n_{HB}).

2. In addition, it was assumed in the studies that the regulator is fast-acting with time constant T_{REG}=0.08T_{HB}. Structurally, this requirement is achievable. However, when designing automatic control systems for gas turbines, it should be taken into account that a decrease in the regulator speed will lead to a deviation of dynamic parameters of the ARS(n_{FT}) from dynamic parameters of the HR. That is, the desired effect from the use of an automated control system for the hydraulic brake will decrease.

Disadvantages of the method include the fact that influence of changes in external conditions on dynamic parameters of the HR such as atmospheric pressure and ambient temperature was not taken into account. On the one hand, this influence is not so significant and if it is necessary to take it into account, it is sufficient to use a regulator with variable settings or an additional loop for regulation of water discharge from the HB.

Further development of this method of controlling the hydraulic brake loading may consist in adapting the regulator parameters to dynamic parameters of the rotors and propellers of various types and taking into account effect of external conditions on them.

8. Conclusions

1. A block diagram of the SAR(n_{FT}) for a hydraulic brake unit has been developed. Basic elements necessary for its practical implementation were shown. The developed diagram features a regulator with a proportionally differentiating regulation law and a high-speed actuating mechanism for the HB loading. Application of other regulation laws may not have a positive effect or even lead to unstable ARSn_{FT} operation.

2. Application of the proportionally differentiating law of regulation of a hydraulic brake loading was substantiated. It was shown that to ensure similarity of dynamic parameters for (FT+HB) and (FT+HR), the regulator time constant should be small, i. e., the regulator's actuator must be fast-acting.

3. A linear dynamic model of a system for automated control of the hydraulic brake loading has been constructed. An equivalent transfer function of ARS(n_{FT}) for the hydraulic brake unit was obtained which makes it possible to calculate the regulator parameters (K_F, T_F) for specified dynamic parameters of a real object.

4. Computational studies of transients of the helicopter rotor and the hydraulic brake without stabilization and with an automated loading control system have been performed. These studies have shown that with a selected structure of the HB ACS and a correct choice of the regulator parameters, transient processes of the ARS(n_{HB}) are close to the transient processes of the HR. As can be seen from Fig. 5, difference between the stabilized HB and the HR rotational speeds does not exceed 1.5 % in the transient process.

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