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Research of the Thermal State of Aviation Engine Supports

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Abstract: This study discusses the main sources of heat transferred to the support of an aircraft engine; explores ways to identify and give them a quantitative comparison. A technique for determining the thermal state of supports of aircraft engines was developed this makes it possible to select a desired cooling system as well as to evaluate and adjust the basic parameters of the engine oil system.

Key words: Heat flow, temperature, heat transfer coefficient, friction, seal, bearing, cooling

INTRODUCTION

Apart from the main units of the Gas Turbine Engine (GTE) that generate heat that flow to the supports of the power plant there are a significant number of friction (rubbing) parts (rotor bearings, drive gear of aggregates and speed reduction gear systems, seals, gear meshes of drives and gear box, splines, friction pairs of aggregates, etc.) that also generate heat. Friction causes wear of parts and also generates a significant amount of heat which together with other thermal streams from body parts flows to the support of the engine. However, it is necessary to maintain target levels of the thermal state of engine support assemblies to meet performance and reliability requirements. The oil system as well as various design methods of thermal protection of various elements of GTE maintain the required thermal level and ensure proper performance of the engine.

At the same time, providing an acceptable thermal state of friction units of modern GTE is a much more important function of oil systems as opposed to ensuring lubrication of friction surfaces. For example, a thermally stressed angular ball bearing may be sufficiently lubricated with a very small amount of oil (no >0.1 kg h^{-1}) and the removal of the amount of heat generated in it (10-20 kW) requires up to 0.15 kg sec^{-1} supply of oil (Tryanov *et al*., 2009).

Statistical data is often used to calculate heat emission since exact calculation of heat transfer by the lubricating oil is associated with great difficulties. For turbojet engine, heat emission is 3-6 kJ sec^{-1} at 10 kN bench thrust; for turboprop 15-25 kJ/sec/1000 kW equivalent bench power. In turbofan engines, heat emission is less dependent on the thrust as part of the heat is transferred to the bypass air (Kuz'michev *et al*., 2014). For medium and large engines, heat emission is 35-55 kJ sec^{-1}.

Currently, justification of the amount of oil supply needed for sufficient cooling requires specific calculations of engine heat transfer (these quantities are directly proportional). The approximate rate of oil in L/min needed to be pumped through the engine can be calculated through Eq. 1:

$$W_{eng} = 6.10^4 \frac{q_{eng}}{C_{oil}.\rho_{oil}.\Delta t_{oil}} \quad (1)$$

Where:

q_{eng} = Heat emitted by engine or heat flow from it to oil (kW)
C_{oil} = Specific heat capacity of oil under arithmetical mean value of temperature sat the entrance and exit (kJ/(kg.K))
ρ_{oil} = Density (kg/m^3)
Δt_{oil} = Temperature difference between the entrance and the exit, κ (Beach *et al*., 1979)

Heat transferred from the engine to the oil can be found by analyzing the heat balance of the engine. This is what has been done in this research.

Analysis of heat sources in the support OF GTE: There are six main sources of heat in the support of a GTE: Q1: heat entering the support from the air-gas tract; Q2: heat entering through the walls of the support; Q3: heat entering through the shaft; Q4: heat from friction in seals; Q5: heat from friction in bearings, gears, splines, etc.; Q6: heat brought in with air through seals. Heat sources

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like oil effervescence, cavitation and others were not considered in this research due their relative insignificance and difficulty in calculating them (Fig. 1).

The solution of the problem of finding the heat flow from friction units could be difficult. If the time for the engine to get to steady-state mode is comparable with the time of flight in this mode then it is necessary to take into account the non stationarity of the heat transfer process.

Generally, the heat generated from the zone of friction is distributed between contacting bodies and also between them and the surrounding medium. This kind of heat exchange occurs by means of thermal conductivity, convection and or radiation (heat exchange by radiation). Thermal conductivity plays an important role. An exact assessment of the amount of heat emitted by friction is difficult to make. For this reason, the following assumption is made: all the research done by friction is converted into heat energy. In other words, the intensity of heat emission q_{fr} (J/(m^{2}sec)) can be defined by Eq. 2:

$$q_{fr} = \mu.P.V \qquad (2)$$

Where:
μ = Friction coefficient
P = Contact pressure
V = Sliding velocity

There are two types of frictions that take place in bearings; rolling friction and sliding friction. Rolling friction is usually connected with elastic hysteresis, adhesion, plastic deformation and micro slip. Rolling friction is much lesser than sliding friction. This is why main energy losses are defined by sliding friction in the rolling body-raceways contacts due to the elastic deformation of the surfaces in contact (Myshkin and

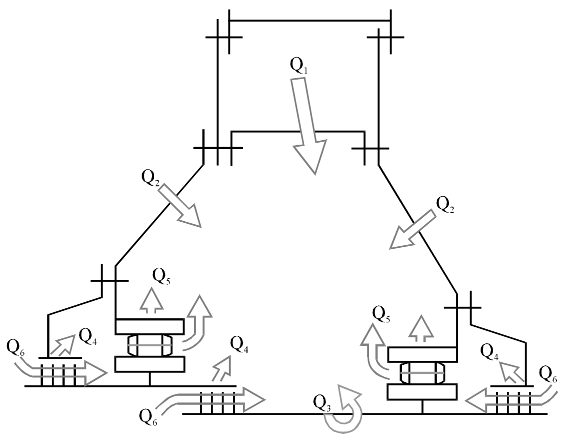

Fig. 1: Sources of heat transferred to the GTE support

Petrokovets, 2006). From Eq. 2 we have an expression for finding the quantity of heat generated by friction in bearings:

$$Q_{bear} = N_{fr} = F.\mu.\pi.n.d \qquad (3)$$

Where:
F = Educed load on bearing
μ = Friction coefficient (0.0025-0.01 for roller bearings; 0.001-0.004 for ball bearings)
d = Diameter of shaft
n = Rotation speed

From a tribological point of view, seals are very similar to bearings (Falaleev, 2014). In essence, all modes of friction can be achieved in seals from dry friction to friction under conditions of hydrodynamic lubrication (Lebeck, 1991). In particular, the amount of heat generated from the friction of end-face radial lip seals can be described by Eq. 4:

$$Q_{seal} = N_{fr} = \Delta P.S.\mu.\pi.n.d_{med} \qquad (4)$$

Where:
ΔP = Pressure difference
S = Area of contact
μ = Friction coefficient
d_{med} = Medium diameter
n = Rotation speed

Friction in gear systems differ in some aspects due to the geometry and kinematics of teeth. During a mesh cycle, we have a motion which includes rolling and sliding. The following expression is used to assess the quantity of heat generated by friction in splines and gear systems:

$$Q_{gear} = (1-\eta).N \qquad (5)$$

Where:
η = Efficiency of gear system
N = Transmitted power (W)

It is important to note that experimental formulae are used for more accurate calculations (Glukharev and Zubarev, 1983). The quantity of heat which enters to get her with air depends on the quantity of air or gas that passes through the seal, the heat capacity of the air or gas and the temperature difference between the mediums separated by the seal. In the case, we use Eq. 6 (Meyer, 1978):

$$Q_{gear} = G_{air}.C_p.(t_{air} - t_{oil}) \qquad (6)$$

Where:
G_{air} = Rate of air flow through seal
C_p = Specific heat capacity of gas
t_{air} = Air temperature
t_{oil} = Oil temperature

The process of heat transferred through the walls of the support is complex; the enclosing wall is a heat conductor through which heat is transmitted by conduction and the heat is transmitted from the wall to the surrounding medium by convection and radiation. The main means by which heat is transferred from the walls to the surrounding medium is convection (Montenay *et al*., 2000; Evans *et al*., 2004) and this is not considered in subsequent calculations of this study. Generally the quantity of heat transferred through the wall from air to oil can be described by the law of Newton-Richman (Kirpichev *et al*., 1940):

$$Q_{wall} = h.S.(t_{air} - t_{oil}) \quad (7)$$

Where:

- h = Heat transfer coefficient (W/(m^{2}K))
- S = The area of the external surface of the wall (m^2)
- t_{air} = External air temperature (K)
- t_{oil} = Oil temperature (temperature of the oil chamber) (K)

A series of assumptions are made when studying the process of heat transfer through the wall of the support. The air tract of the engine core around the support zone is divided by reinforcing ribs into a series of parts in which the flow of gas could be liken to flow through short canals (Fig. 2).

Since, the temperature in the bypass of the engine is lower than it is in the engine core, we could consider that the maximum temperature of the rib is at the middle of its height.

In other words, only half of the height of the rib takes part in the process of heat transfer to oil. Since, the ratio of the outer and inner diameters of the cylindrical wall is close to one, the calculation of heat transfer through it could be done with a sufficient degree of accuracy using formula for a flat wall.

Heat transfer at the side walls of the middle support is done by heat transfer from the air driving the compressor disks to oil which gets to the walls from the oil chamber. For the tapered walls, due to the complexity of the flow model, we used expressions for determining the heat in the cylinder walls through partition of the conical sections.

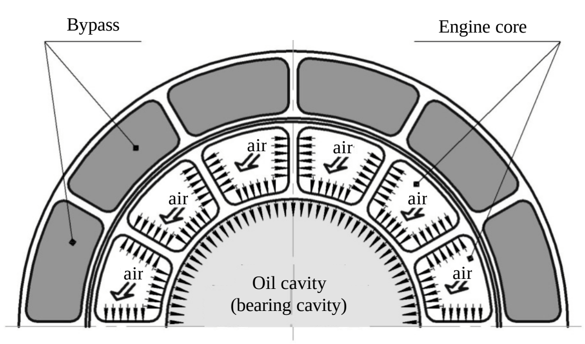

Fig. 2: Process of heat tranfer through walls of GTE support

CALCULATION OF THE THERMAL STATE OF GTE SUPPORT

For calculation of the quantity of heat transferred to the middle support from the tract and through the walls of the shaft it is necessary to determine the temperature of parts of the support and convective heat transfer coefficients.

From known results of thermogasdynamic calculations on the entrance and exit points (P^*, T^*, G_{air}) obtained in the design of the engine, we used the values of temperature and pressure at the points of bleed and venting of air in the system of the middle GTE support (Fig. 3).

According to known geometry of the canals and the values of gas dynamic parameters (pressure and temperature) it is necessary to determine the parameters of flows in the entire network. Meanwhile the calculation has to consider heating of the flow since the temperature of elements are significantly not uniform.

The calculation method is based on presentation of the system as graph from which basic chords are chosen and a minimum tree diagram is built (Fig. 3).

The mathematical model is described by the relations derived from Kirchhoff's laws and trailing relation characterizing the relationship between pressure, hydraulic resistance and flow in the branches of the graph. After a number of transformations, we obtain a system of equations for the increments on the chords of the graph. The number of equations equal to the number of linearly independent contours. This substantially reduces the time of calculation (Kapinos *et al*., 2004).

The results of the calculations in the program give the values of convective heat transfer coefficients in the support as well as the average temperature of the channel walls (Fig. 4).

Thus the value of heat transfer coefficient in the seal ring at the maximum operating mode of the engine reached h = 5048 W/(m^{2}K) and the maximum temperature of the walls of the support was at 353°C.

The model must be solved by successive approximations. In the first approximation, the temperature is given based on the prototype used or based on previous experiences of designing cooling systems.

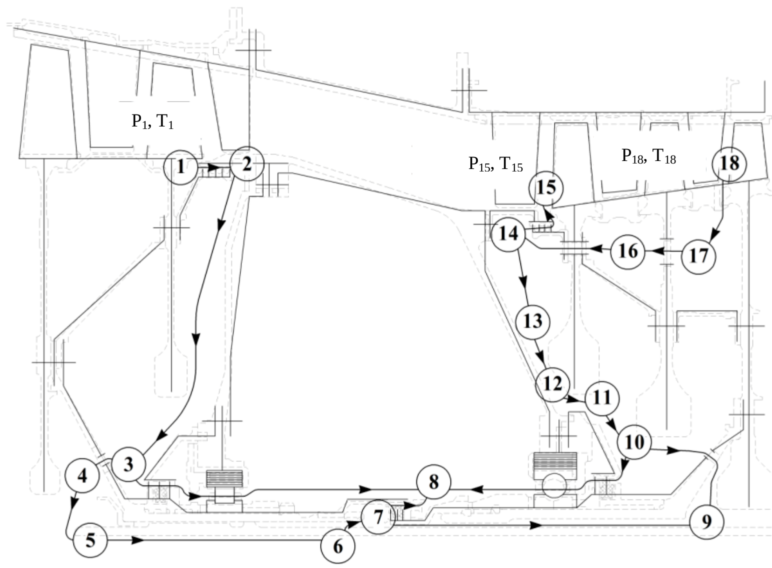

Fig. 3: System of pressurization and cooling of middle support

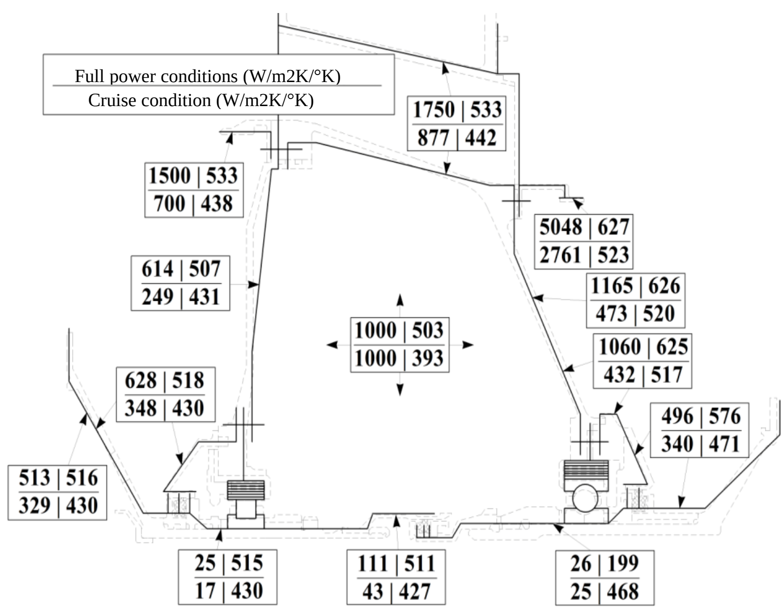

Fig. 4: Distribution of heat transfer coefficient and temperatures in the engine support

ANALYSIS OF HEAT FLOWS IN GTE SUPPORT

According to the derived values of the temperatures of elements and parts of the middle GTE support, it is possible to calculate the amount of heat transmitted to the GTE support. The calculation was done using the Eq. 2-7 outlined above (Table 1).

Table 1: Quantity of heat transmitted to the GTE support

Heat source	Quantity of heat (W)	Percentage
Q1: from the engine core	2142	12
Q2: from the support walls	4427	24
Q3: through the shaft	299	2
Q4: friction in the seal	2522	14
Q5: friction in the bearings	5161	28
Q6: leakage through the seals	2809	15
Q7: friction in gears	930	5
Total	18290	100

The total sum of heat transmitted to the support from all sources is equal to 18290 W. With knowledge of the quantity of heat transmitted to the middle GTE support, we can now find the necessary amount of oil to be oil to be C_{oil} = 1680 kJ/(kg.κ) under arithmetic mean of temperatures at the entrance and exit (we will consider its value at a temperature of 373°K); ρ_{oil} = 820 kg/m^3; Δt_{oil} = 50°K. Thus the required rate of pumping oil through the GTE support from Eq. 1 is about W_{eng} = 15.9 L min^{-1}.

The obtained results are with significant inaccuracies due to simplifications in the calculation method which was warranted by the necessity to reduce the time of calculation. In this connection, many of the parameters were taken as the average of the most formally mentioned.

CALCULATION PROCEDURE OF THE REQUIRED COOLING

A detailed presentation and analysis of the calculation of the quantity of heat transmitted to the engine support has been shown above which allows qualitative assessment the effectiveness of support cooling and thermal protection. This technique allows the calculation of cooling options and air pressurization of the supports. It allows us to examine and consider the effect of the cooling air not only on the cooling parameters but also on specific parameters of the engine. Following the procedure, you can calculate the necessary pumping oil as well as examine heat sources distribution, depending on the operation mode of the engine (Fig. 5).

According to Fig. 5 the most rapidly increasing emission of heat is in gears since, they are directly connected to the engine shaft, hence their speed increases with increasing rotor speed. Due to the large thermal inertia of the engine components the heat is transferred through the wall of an oil cavity which increases smoothly and has a nonlinear relationship.

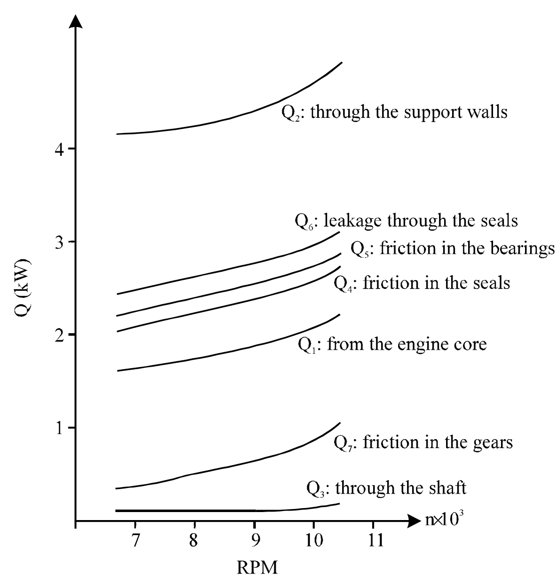

Fig. 5: Heat transfer rate by source, depending on the engine mode

Similarly, the heat transfer is increased through the engine passage. The heat introduced by the air through the seal, increases in proportion to the pressure and temperature of the air which is pressurizing the oil chamber.

CONCLUSION

Continuous improvement of aircraft engine design is directly related to the improvement of its cycle parameters. The temperature rise in the engine air-gas channel leads to considerable complications of the operating conditions of the aircraft engine supports (Vinogradov, 2014). This particularly applies to the friction units located in the support that requires forced cooling (Tryanov *et al*., 2009; Klingsporn, 2004). In order to evaluate the thermal state of support, it is necessary to identify the main sources of heat and to estimate their share in the total value of the amount of heat which is introduced into the support. The ratio between the heat amounts may vary within a wide range. However, the calculations conducted by these studies have shown that the main heat is produced by friction in bearings and seals which goes to the support through the walls and the hot gas through the seal (Flouros, 2004). The total amount of the heat from the sources is at 80-90%. The share of seals is 20-30%. Therefore, to evaluate the thermal state of the support it is necessary to estimate the amount of heat coming from each of these sources and to improve the cooling efficiency of support to improve the design of the seals.

The most difficult step in evaluation of the thermal state of the support is to determine the coefficient of convective heat transfer. In this study these coefficients are determined by hydraulic calculation in which the cavity surrounding the support is modeled as a specific kind of channel. At the maximum mode, the temperature of the support walls ranged from 234-353°C and the convective heat coefficient transferred from the same surfaces varied from 614-1165 W/(m^{2}K). To get the temperature distribution in all parts of the engine support, the results of the hydraulic calculation were the input for the next structural calculation (Saunders *et al*., 2007). Thus, on the basis of the created models, a method of calculating of the thermal state of the support was created and implemented.

The creation of the method which is used to calculate the thermal support state will allow you to select the method of cooling the support based on the allowable amount of heat contributed as well as to design a sealing system that implements the selected method. Further application of the developed technique is possible to design elements supply to the friction units within the support as well as elements of the oil discharge

(Tryanov *et al*., 2009). Implementation of the above calculations ultimately will allow adjustment of the basic parameters of the engine oil system (Hart, 2008) as they are in direct proportion to the total amount of heat which is transferred to the support.

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