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STUDY ON ACCURACY OF HEAT TRANSFER COEFFICIENT DETERMINATION IN THE BEARING CHAMBER FOR GAS TURBINE ENGINE

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ABSTRACT

The heat transfer coefficient (HTC) is one of the key parameters that should be known at the stage of the bearing chamber design. This ensures safe temperature conditions for the lubrication oil and reliable operation of the gas turbine engine. The temperature gradient method is commonly used in experimental practice to determinate the HTC. The accuracy of the HTC determination is sensitive to changing of the bearing chamber operating conditions and should be analyzed at the stage of experimental studies planning. This paper presents a study on the accuracy of HTC determination when the external cooling of the bearing chamber is used to obtain the temperature difference sufficient for measurement. Three ways to reduce the relative error of the HTC determination in the bearing chamber were analyzed: i) decreasing the temperature measurement error; ii) decreasing the temperature of external cooling medium; iii) increasing the external heat transfer coefficient and contribution of wall thermal resistance optimization. For different operating conditions of the bearing chamber, the temperature of the outer wall that ensures the specified accuracy of the experimental HTC and the required parameters of the cooling medium were determined and recommended for practical implementation.

Keywords: bearing chamber, heat transfer coefficient, accuracy, relative error, measurement, temperature gradient method, external cooling conditions.

NOMENCLATURE

A_{chm}	bearing chamber surface area (m^2)
c_p	specific heat capacity (J/kg-K)
D_h	hydraulic diameter (m)
h	enthalpy (J/kg)
h_c	heat transfer coefficient (HTC) (W/m^2-K)
k	thermal conductivity coefficient (W/m-K)
K	error coefficient (1/K)
$\dot{m}$	mass flow rate (kg/s)
Nu_{Dh}	Nusselt number (dimensionless)
Q_{br}	heat generation in the bearing (W)
q_w	heat flux density (W/m^2)
q_{ce}	external convective heat flux density (W/m^2)
$q_{\varepsilon e}$	external radiant heat flux density (W/m^2)
R_w	wall thermal resistance (m^2-K/W)
Re	Reynolds number (dimensionless)
Re_u	circumferential Reynolds number (dimensionless)
T_{nw}	fluid near wall temperature (K)
T_{wi}	internal wall temperature (K)
T_{we}	external wall temperature (K)
T_e	temperature of the external (cooling) flow (K)
u	velocity (m/s)
Y	mass fraction (dimensionless)

Greek Symbols

Δx_j	measurement error of parameter x_j
ΔT_{TC}	measurement error of thermocouple (K)
δ	thickness (m)
δx_j	relative error of parameter x_j (dimensionless)
ν	kinematic viscosity (m^2/s)

Subscripts

a	air
e	external (cooling) flow
f	oil film
i	internal flow
in	inlet
l	oil
m	mixture
out	outlet
w	wall

1. INTRODUCTION

Gas turbine industry development is guided by the instantly growing parameters of the thermodynamic cycle as well as by the reduction of engine overall dimensions and more compact design of its components, e.g. bearing chambers. The above-mentioned trends lead to a heavier thermally stressed state of the engine, higher rotational speeds, higher temperatures, pressures, and velocities of airflow moving through the gas path [1]. Some challenges should be solved on the way to fuel-efficient gas turbine engines. Among them are minimization of power consumption to lubricate the friction zones of the bearings and minimization of air consumption to pressurize and protect the bearing chambers from the hot gases. One more challenge that designers of new engines must tackle is oil protection against the temperature impact.

The design of a reliable, robust and efficient bearing chamber is considered to be the most challenging task related to the gas turbine engine lubrication system. The published results of the EC-funded FP7 projects ATOS (Advanced Transmission and Oil System concepts) [1-4], ELUBSYS (Engine Lubrication System Technologies) [5-8], E-BREAK (Engine Breakthrough Components and Subsystems) [9-10] as well as results of other research programs (SILOET [11], SAMULET [12], etc.) indicate the increased interest of scientific and industrial community to the complex processes in the bearing chamber.

The bearing chamber is filled with a mixture of supplied oil and air that leaks through the seals. The oil droplets are formed as a result of the oil splash back from the bearing chamber wall, as well as a result of the oil interaction with the swirled air flow within the chamber. When droplets and streams of oil get to the walls of the bearing chamber, they form an oil film. The film is moved under the action of the forces in the interface, gravity, viscosity as well as impact of the oil droplets continuous interaction with the film. Hence, the thickness of the oil film is non-uniform along the bearing chamber wall (see Fig. 1).

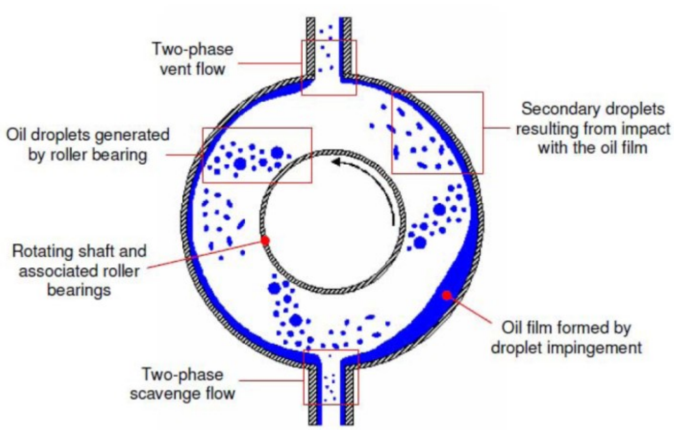

FIGURE 1: MULTIPHASE FLOW STRUCTURE IN THE BEARING CHAMBER [5]

Non-uniform thickness of the oil film makes an influence on the heat exchange processes between the bearing chamber wall and the air/oil mixture available in the bearing chamber. This, in turn, affects safe temperature conditions for both the lubrication oil and bearing chambers. A reliable design of the compact bearing chambers and safe and efficient operation of the gas turbine engine as a whole can be ensured only by a deep understanding of the thermal and hydraulic processes related to the complex behavior of oil-air mixture within the entire range of operating conditions.

On the other hand, it is very difficult to develop the mathematical models that cover complex thermal and hydraulic processes in the bearing chamber. The behavior of the multiphase fluid (droplet-film-air) is influenced by oil and airflow rates, rotation speed and direction, bearing chamber design, internal pressure, variation of the thermodynamic properties of the phases due to heat transfer, the roughness of the chamber walls and by many other factors. Therefore, to better understand the heat transfer and flow distribution phenomena in the bearing chamber, it is necessary to combine the CFD simulation methods with intensive experimental studies.

Being dependent on many factors mentioned above, the heat transfer coefficient (HTC) also varies due to changing of engine operating conditions. Experimental investigations performed in the University of Karlsruhe provided a great contribution to understanding of the bearing chamber flow distribution and heat transfer phenomena [13-18]. However, the experimental correlation obtained for the HTC determination applicable only for bearing chambers of simplified geometry and limited range of operating conditions. The aim of this work, being performed in the frame of the AMBEC[1] project, is to overcome these limitations and to develop a reliable experimentally validated methodology able to calculate fluid distribution and heat transfer coefficients in different zones of real bearing chamber.

2. METHODOLOGY

The complexity and interrelation of thermal and hydraulic processes in the bearing chamber do not allow obtaining sufficiently universal ratios, even for the HTCs in the bearing chamber. This is indicated, in particular, by significantly different values of the local internal HTCs (1) that are reported by authors of [18-21]

$$h_{ci} = \frac{q_w}{\left(T_{nw} - T_{wi}\right)}, \tag{1}$$

where T_{nw} is the fluid near wall temperature, T_{wi} is the internal wall temperature, q_w is the wall heat flux density.

[1] "Advanced Modelling Methodology for Bearing Chamber in Hot Environment (AMBEC)" project has received funding from the Clean Sky 2 Joint Undertaking under the European Union's Horizon 2020 research and innovation programme under grant agreement No 785493

The accuracy of the heat flux density determination by the gradient of the temperature in the wall, which thickness is equal to δ_w and thermal conductivity is equal to k_w, is beyond doubt. For a flat wall, the heat flux density is determined by equation:

$$q_w = \frac{T_{wi} - T_{we}}{R_w}, \tag{2}$$

where

$$R_w = \delta_w / k_w, \tag{3}$$

is the thermal resistance of the bearing chamber wall and T_{we} is the external wall temperature.

For typical designs of the bearing chamber, the ratio between the wall thickness and the chamber inner diameter is less than 12%. With this geometry, the difference in heat flux density calculated for a flat and cylindrical walls is less than 1%, that allows us to use the formulas of a flat wall.

At steady state, the internal HTC can be calculated based on equations (1) and (2) resulting in:

$$h_{ci} = \frac{1}{R_w} \cdot \frac{T_{wi} - T_{we}}{T_{nw} - T_{wi}}. \tag{4}$$

Temperature differences, involved in expression (4), highly depend on internal conditions in the bearing chamber and external cooling. Variations of the supplied oil and air flow rates, their temperatures, shaft rotational speed, etc., which represent the bearing chamber operating conditions, influence the bearing chamber internal conditions. In parallel, parameters of the cooling medium such as temperature and external HTC define the external conditions. Thus, the accuracy of internal HTC experimental determination depends on the accuracy of temperatures T_{nw}, T_{wi} and T_{we} measurement, the level of their difference and experimental methodology used, that is shown below.

2.1 Indirect determination of the internal HTC

The main reason of the internal HTC experimental determination error is associated with its indirect determination using the relations (1) and (2) based on the parameters measured during the experiment. The general equation for calculating the root mean square error of the indirect variable f is represented below:

$$\Delta f = \sqrt{\sum \left(\frac{\partial f}{\partial x_j} \Delta x_j \right)^2}, \tag{5}$$

where Δx_j is the error of direct measurement of the parameter x_j, which influences the value of f. The relative error is determined through dividing (5) by the value of f. Taking into account equations (1) and (2), the HTC relative error is equal to:

$$\delta h_{ci} = \sqrt{(\delta R_w)^2 + (\frac{\Delta T_q}{T_{wi} - T_{we}})^2 + \frac{(\Delta T_{nw})^2 + (\Delta T_{wi})^2}{(T_{nw} - T_{wi})^2}} \tag{6}$$

The relative error of the heat flux density is defined in this case as:

$$\delta q_w = \sqrt{(\delta R_w)^2 + (\frac{\Delta T_q}{T_{wi} - T_{we}})^2} \tag{7}$$

Here ΔT_q is the measurement error of temperature difference $(T_{wi} - T_{we})$ measured by a differential thermocouple; ΔT_{nw}, ΔT_{wi} are the measurement errors of temperatures T_{nw} and T_{wi}, respectively; δR_w is the relative root mean square error of thermal resistance of the bearing chamber wall, which is defined as:

$$\delta R_w = \sqrt{\left(\frac{\Delta k_w}{k_w}\right)^2 + \left(\frac{\Delta \delta_w}{\delta_w}\right)^2}. \tag{8}$$

If all the temperatures are measured independently, the relations for calculating the relative error of heat flux density and internal HTC will change as compared to (6) and (7) resulting in:

$$\delta q_w = \sqrt{(\delta R_w)^2 + \frac{(\Delta T_{wi})^2 + (\Delta T_{we})^2}{(T_{wi} - T_{we})^2}} \tag{9}$$

$$\delta h_{ci} = \sqrt{\begin{aligned}&(\delta R_w)^2 + \frac{(\Delta T_{nw})^2}{(T_{nw} - T_{wi})^2} + (\frac{\Delta T_{wi}}{T_{wi} - T_{we}} + \\ &+ \frac{\Delta T_{wi}}{T_{nw} - T_{wi}})^2 + \frac{(\Delta T_{we})^2}{(T_{wi} - T_{we})^2}\end{aligned}} \tag{10}$$

The relative error of thermal resistance of the bearing chamber wall by itself is determined mainly by the accuracy of thermocouple location $\Delta\delta_w$ and equals to about 1%. The level of relative error of other terms in equations (6), (7) and (9), (10) is much higher. Thus, the error of thermal resistance of the wall may be neglected. Such an assumption does not affect the reliability of the results, but allows us to obtain expressions that are more convenient for analysis.

If the individual temperature measurement errors are the same, then the relative error of the HTC determination is almost proportional to the temperature measurement error as it follows from the formula (10). In this case, the minimum relative error of the internal HTC is achieved provided that temperature differences are equal

$$T_{nw} - T_{wi} = T_{wi} - T_{we}\,. \tag{11}$$

This corresponds to the equality of thermal resistances

$$1/h_{ci} = R_w \tag{12}$$

Since the internal HTC varies around the circumference of the bearing chamber and depends on the operating mode, it is impossible to implement such conditions for all cases.

The resulting error of the internal HTC calculated with the use of formula (10) is much higher than in case of using formula (6). This is due to the separate measurement of all temperatures that determine the heat flux density and the internal HTC value instead of using the differential thermocouple for measurement of the temperature difference between the internal and external wall surfaces. Mathematically, this is explained by the fact that the expression (6) contains three components of the temperature measurement error, while the expression (10) contains six ones. Thus, the relative error of the internal HTC is increased by about 1.4 times.

The more complex and clear impact of the main factors on the error of the internal heat transfer coefficient are characterized by the expression

$$\delta h_{ci} = \Delta T_{TC}\sqrt{\begin{matrix}\left[\left(\dfrac{k_w}{\delta_w h_{ci}}\right)^2 + 1 + \left(\dfrac{k_w}{\delta_w h_{ci}} + 1\right)^2\right]\times \\ \times\dfrac{\left(\dfrac{\delta_w h_{ci}}{k_w} + 1 + \dfrac{h_{ci}}{h_{ce}}\right)^2}{\left(T_{nw} - T_e\right)^2}\end{matrix}} = \tag{13}$$

$$= \Delta T_{TC} \cdot K$$

It is obtained from equation (6) and takes into account convective cooling of the outer wall of the bearing chamber with the same error ΔT_{TC} of individual temperature measurement by thermocouples. The temperature differences included in (6) are expressed using relations (1) and (2) through the corresponding thermal resistances, taking into account the equation for external heat transfer with the external HTC h_{ce}

$$q_w = h_{ce}(T_{we} - T_e)\,. \tag{14}$$

Similarly, the following relation is obtained from the equation (9)

$$\delta h_{ci} = \frac{\Delta T_{TC}}{T_{nw} - T_{we}} \cdot \sqrt{2 \cdot \left[\begin{matrix} 4 + \left(\dfrac{h_{ci}\cdot\delta_w}{k_w}\right)^2 + \\ + \left(\dfrac{k_w}{h_{ci}\cdot\delta_w}\right)^2 + \\ + 3\cdot\left(\dfrac{k_w}{h_{ci}\cdot\delta_w} + \dfrac{h_{ci}\cdot\delta_w}{k_w}\right)\end{matrix}\right]} \tag{15}$$

This relation can be used for any external cooling conditions. A generalizing factor of its influence is the temperature T_{we} of the outer wall of the chamber. The lower it is, the lower the internal HTC error will be. Since the temperature T_{nw} is determined by the operating modes of the bearing chamber, it is usually possible to determine the temperature T_{we}, which will ensure the required relative error of the internal HTC determination.

2.2 Near wall temperature determination

Another problem in the calculation of the internal HTC is associated with the determination of the fluid near wall temperature. In the classical theory of convective heat transfer it is considered as the temperature of the unperturbed flow at a certain heat flux direction. In addition, heat transfer is determined by the movement of the homogeneous (single-phase) medium. In the bearing chamber, the heat is transferred by two phases. Moreover, for the bearing chambers the mass flow rate of the air can exceed the oil mass flow rate on 75%, and the heat equivalent of air $(c_{pa}\dot{m}_a)$ can be up to 75% of the oil heat equivalent. Due to the significantly higher density, the volume fraction of oil droplets in the bearing chamber is less than 0.5%, while air is a continuous phase, which, in principle, realizes the heat transfer to the chamber wall. However, in this case the values of the internal HTC are significantly (50 times) lower than the values observed in experimental studies and numerical simulations represented in [13-21]. They do not correlate with the heat flux density in such conditions. This result is quite expected. It confirms that the main transfer of heat from the chamber central part to the wall is carried out by oil droplets formed during the rotation of shaft and bearing elements.

Moving oil film, formed by precipitated oil droplets, contacts directly with the chamber wall. Definite contribution to its motion and heat exchange is made by interfacial interaction with the air. But the temperature near the wall in the area where the oil becomes a continuous phase is determined by the droplets and, therefore, is quite close to the temperature of the adjacent air-droplet flow. The heat transfer is carried out not by the classical air convection, but by the movement of oil droplets. Therefore, the thermal resistance in the classical sense is absent on this boundary.

The heat exchange with the chamber inner wall is determined by the moving oil film. The temperature on the inner (along the radius) surface of the film is maintained by precipitated oil droplets, which additionally tubulise the film. This improves heat transfer conditions. However, the main thermal resistance for liquids with high Prandtl numbers, like an oil, is associated with a laminar sublayer that is formed near a solid wall.

Such results are most widely represented in the research that deals with film condensation [22, 23], where film moves under the action of friction, gravity, and resistance at the interface with the vapour. For thin films, the HTC (1) is in good agreement with the value calculated for a stationary liquid as:

$$h_c = \frac{k_l}{\delta_f}. \tag{16}$$

Here, δ_f is the film thickness, and k_l is the thermal conductivity coefficient of oil. For thicker films, the HTC is calculated taking into account the laminar speed and temperature profile. At the same time, when the film thickness and the Reynolds number

$$\mathrm{Re}_f = \frac{u_f \delta_f}{\nu_l} \tag{17}$$

increase, the HTC decreases, but becomes noticeably higher than the value calculated on the basis of equation (16) for stationary medium. Such variation of the HTC is explained by the predominant contribution of increasing the boundary layer thickness compared with increasing the film velocity. Even for single-phase flow along the plate, the Nusselt number always increases with increasing the Reynolds number, while the HTC decreases. Mathematically, this occurs due to the fact that in the similarity equations the exponent for Reynolds number is less than 1.

It is interesting to note that the HTC values obtained by the calculation of oil flow in the annular gap are close to those observed in experiments with the bearing chamber [18, 19]. Under the same conditions (geometry and velocity), the HTC values are 1500...4350 $W/(m^2{\cdot}K)$ for IPM-10 oil and 1300...3700 $W/(m^2{\cdot}K)$ for ETO 2380 oil. This fact can be explained by a plausible description in the criterial equations of the predominant contribution of the thermal resistance in a viscous sublayer.

Thus, for reliable determination of the internal HTC given by equation (1), it is necessary to select correctly the temperature T_{nw} of unperturbed flow near the wall. It should be the temperature outside the boundary layer. At least, it should be the temperature that is measured in the area of its linear variation over the film thickness, i.e. out of logarithmic velocity profile in turbulent mode. Moreover, it is also important for presenting the results of numerical simulation.

Therefore, even with absolutely accurate direct measurements of the internal and external temperatures of the bearing chamber wall, the accuracy of the internal HTC experimental determination will depend on what variable is used as the temperature of the unperturbed flow.

In this research, we selected for this purpose the temperature T_m of the oil/air mixture that is generated after the bearing. The temperature of the mixture can be determined by knowing its enthalpy:

$$h_m = h_l Y_l + h_a(1 - Y_l) + Q_{br}/(\dot{m}_l + \dot{m}_a), \tag{18}$$

where the oil mass fraction is

$$Y_l = \frac{\dot{m}_l}{\dot{m}_l + \dot{m}_a}, \tag{19}$$

where $\dot{m}_l, \dot{m}_a$ are mass flow rates of oil and air, respectively.

The enthalpies of each phase are calculated for the corresponding parameters at the bearing chamber inlet. Since the oil heat capacity depends on the temperature, the method of successive approximations should be used to determine the mixture temperature T_m by its enthalpy. This temperature is a temperature near the wall T_{nw}, considered above.

It should be taken to account that the bearing is the main source of heat within the bearing chamber. The heat generation Q_{br} in the bearing can be determined for a given geometry of bearing elements, revolutions, and radial load [24-26]. For example, the heat dissipation of the considered bearing under a load of 500 N is 0.5...8 kW in the range of rotational speed of 5,000...20,000 rpm.

2.3 Expected value of internal HTC

To determine the relative error of internal HTC using the formulas (13) or (15) it is need to know the expected value of the internal HTC that depends on bearing chamber operating conditions.

One of the ways to find the expected value of the internal HTC is to use some available correlations for bearing chambers, e.g. correlation of Busam et al [19]. In this research the Nusselt number is expressed as a function

$$Nu_{D_h} = 0.35 D_h^{1.46}\, Re_a^{0.48}\, Re_l^{0.32}\, Re_U^{0.35}, \tag{20}$$

of dimensionless complexes that quantitatively determine the effect of chamber geometry, oil and air flow rates, and shaft speed. Here D_h is the bearing chamber hydraulic diameter, Re_a, Re_l are the Reynolds numbers of sealing air and lubrication oil flows, Re_U is the circumferential Reynolds number.

According to [19], the application area of relation (20) is limited by the oil consumption of 24.4...175.8 l/h; sealing air consumption of 5.3...20.5 g/s; shaft speed of 4,000...16,000 rpm; hydraulic diameter of 0.012...0.034 m.

Another way to find the expected value of the internal HTC is to use the energy conservation principle based on bearing chamber inflow and outflow energies and corresponding assumptions. In this case, the enthalpy of the cooled air-oil mixture at the bearing chamber outlet $h_{m\ out}$ is used in the following form to define temperature of the cooled air-oil mixture $T_{m\ out}$

$$h_{m\ out} = h_{m\ in} - q_w \cdot A_{chm} /(\dot{m}_l + \dot{m}_a), \quad (21)$$

where A_{chm} is the bearing chamber surface area, $h_{m\ in}$ is the inlet enthalpy of the mixture calculated by (18), which determines the temperature T_{nw}.

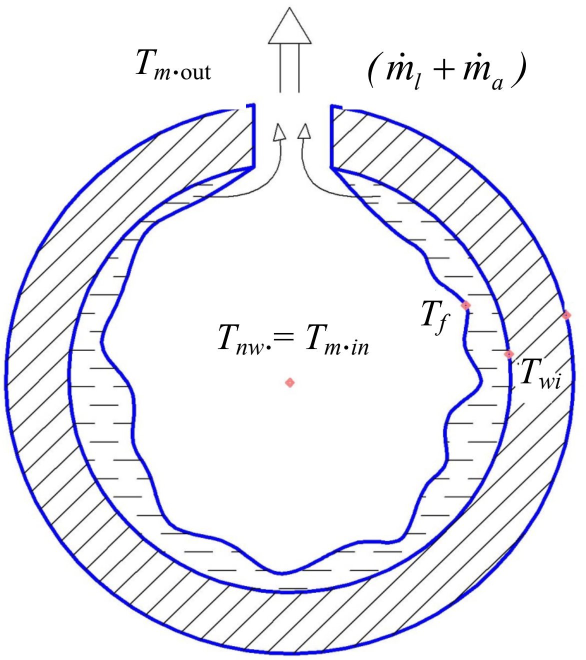

FIGURE 2: TEMPERATURE OF THE COOLED MIXTURE

2.4 Radiation heat transfer

If the external thermal resistance has the highest value, then the outer wall temperature will be close to the temperature in the bearing chamber. In this case, the contribution of radiation heat transfer is quite noticeable in this case.

Temperature differences in the denominators of expressions (6) and (10) are small and depend on the heat flux density. In the absence of radiation heat transfer this heat flux density is defined as follows:

$$q_w = \frac{T_{nw} - T_e}{1/h_{ci} + \delta_w / k_w + 1/h_{ce}}. \quad (22)$$

Here, the temperature T_{nw} is determined by the bearing chamber operating mode and T_e is the temperature of the external cooling flow. The total thermal resistance includes wall resistance δ_w/k_w and internal $1/h_{ci}$ and external $1/h_{ce}$ thermal resistances of heat transfer.

Taking into account the radiation heat exchange $q_{\varepsilon e}$, the total heat flux density

$$q_w = q_{ce} + q_{\varepsilon e} \quad (23)$$

is determined as a result of successive approximations by temperature T_{we} of the bearing chamber outer wall. In this case the convective heat flux density from outer wall is

$$q_{ce} = \frac{T_{we} - T_e}{1/h_{ce}}, \quad (24)$$

and total heat flux density is

$$q_w = \frac{T_{nw} - T_{we}}{1/h_{ci} + \delta_w / k_w}. \quad (25)$$

3. RESULTS AND DISCUSSION

The work described is performed in the frame of the AMBEC project, as mentioned in the Section 1 above. One of the key tasks of the AMBEC project is to validate experimentally the methodology proposed for calculation of the fluid distribution and heat transfer coefficients in different zones of real bearing chamber. For this purpose, the model bearing chamber will be designed, manufactured and subjected to intensive experimental tests. The external cooling will be applied to increase the accuracy of the HTC experimental determination. To specify the conditions for external cooling of bearing chamber and to develop an appropriate cooling system, it is important to analyze how the intensity of external and internal heat transfer as well as the temperature of the cooling medium and medium in the bearing chamber influence on the internal HTC error at the experiment preparation stage.

The base operating mode of the model bearing chamber is characterized by the internal HTC, calculated by Busam et al correlation (20), equal to 2000 W/(m^{2}K) and the temperature of oil-air mixture (medium in the bearing chamber) equal to 120 °C. An airflow with temperature 20°C and external HTC 32 W/(m^{2}K) is considered for external cooling of the bearing chamber. The temperature measurement error is 1 °C.

Relation (13) allows analyzing the influence of the main factors on the internal HTC error. Square root in the expression (13) is an error coefficient K that reflects the error of the internal HTC. For the considered basic mode of the bearing chamber operation it is equal to 1.17. It means that the error in measuring the temperature of 1 K causes an error in determining the HTC equal to 117%.

Figures 3-6 illustrate the influence of the main factors (such as the intensity of external and internal heat transfer as well as the temperature of the cooling medium and medium in the bearing chamber) on the error coefficient K.

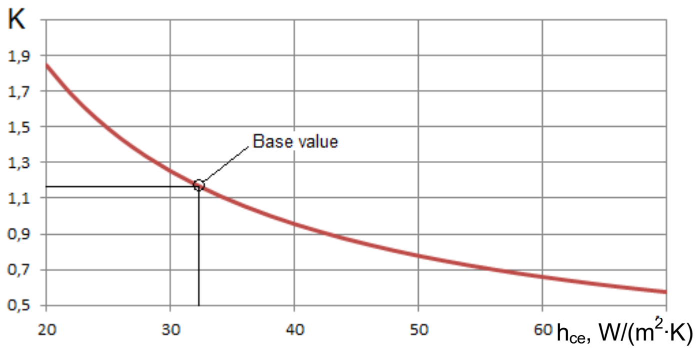

FIGURE 3: THE DEPENDENCE OF THE ERROR COEFFICIENT ON THE INTENSITY OF EXTERNAL HEAT TRANSFER

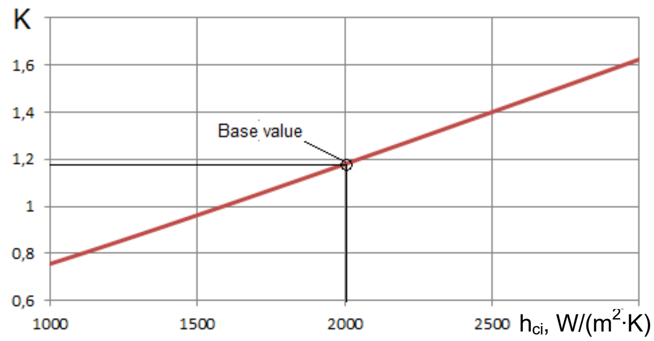

FIGURE 4: THE DEPENDENCE OF THE ERROR COEFFICIENT ON THE INTENSITY OF INTERNAL HEAT TRANSFER.

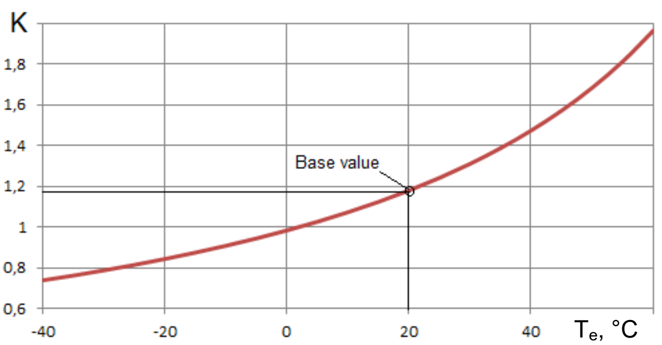

FIGURE 5: THE DEPENDENCE OF THE ERROR COEFFICIENT ON THE TEMPERATURE OF THE COOLING MEDIUM

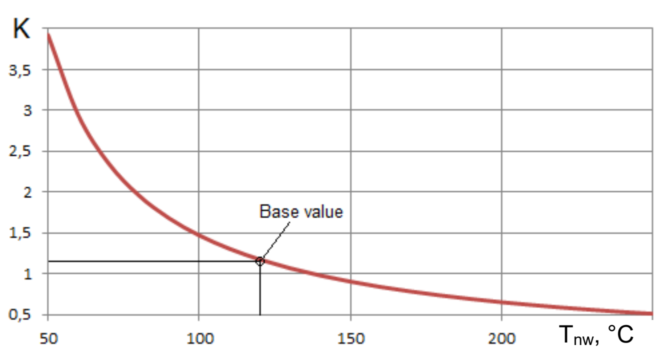

FIGURE 6: THE DEPENDENCE OF THE ERROR COEFFICIENT ON THE TEMPERATURE OF THE MEDIUM IN THE BEARING CHAMBER

In the case of independent measurement of three temperatures and using formula (15), the nature of the influence of the temperature T_{nw} and the value of h_{ci} on the error of the internal HTC is the same. This measurement approach is the most widespread, but it leads to the 40% higher error comparing to measurement of the temperature difference between the internal and external wall surfaces with the differential thermocouple and the following calculations in accordance with formula (10). In this regard, the issue of reducing the error of the internal HTC is even more acute. For fixed parameters of the bearing chamber and operating mode, the only way is to decrease the outer chamber wall temperature T_{we}.

The contribution of radiation heat transfer is quite noticeable for air cooling of the bearing chamber external surface. Figure 7 shows the nature of its influence on the temperature differences for the base operating mode of the bearing chamber. In this case, the radiation heat flux density for steady-state conditions is about 430 W/m^2 at the outer wall temperature of 112.2 °C and a total heat flux density of 3410 W/m^2. The radiation heat transfer contribution is always represented in decreasing the bearing chamber outer wall temperature (red line in Fig.7), compared with the value calculated with the use of convection only. Thanks to the increase of the temperature gradient, the errors of determining the heat flux density and the internal HTC are approximately 13% lower.

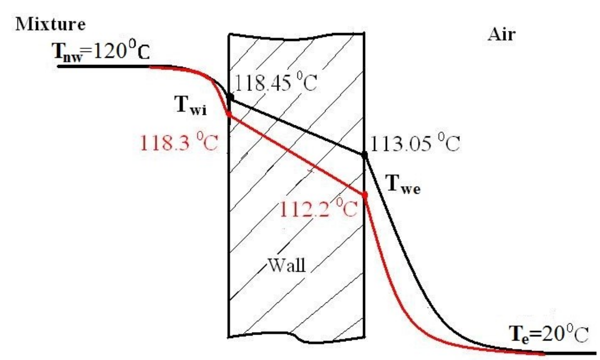

FIGURE 7: TEMPERATURE DISTRIBUTION

Any intensification of external cooling leads to increasing the heat flux density. However, the heat flux density in local heat transfer calculated by using the Busam et al correlation given by equation (20) may be excessively high. Due to the heat rejection, the calculated temperature of the cooled air-oil mixture $T_{m\,out}$ in the outlet of the bearing chamber (scavenge and vent lines) will be lower than the temperature of the bearing chamber inner wall T_{wi}, (see Fig.2) which is physically impossible. In this regard, when analyzing the accuracy of experimental HTC, it seems rational to set the internal HTC maximum value from the condition that the temperature of the cooled air-oil mixture $T_{m\,out}$ is equal to the temperature of the inner wall T_{wi}. Taking into account equations (1) and (21), the expected value of the internal HTC h_{ci} is determined from the equation

$$\frac{h_{ci} \cdot A_{chm}}{c_{pm} \cdot (\dot{m}_l + \dot{m}_a)} = 1, \qquad (26)$$

where c_{pm} is the mean heat capacity of the mixture.

The HTCs obtained according to expression (26) are significantly lower than those calculated using equation (20). For example, for considered bearing chamber operating mode with oil mass fraction 60% and inlet mixture temperature T_{nw} 120 °C the value of the internal HTC obtained from equality (26) is 400 W/(m^{2}K) instead of 2000 W/(m^{2}K) obtained on the basis of the Busam et al correlation. By using equation (15), a 20% relative error of the internal HTC is achieved at temperature of the chamber outer wall T_{we} equal to 92.5 °C. Following expression (14), if the cooling air temperature is equal to 20 °C, such conditions are realized with an external HTC of about 105 W/(m^{2}K), which can be created with the use of a 30 m/s airflow.

4. CONCLUSION

The relative error of the internal HTC experimental determination depends on the bearing chamber operating mode and predefined by the following parameters: (i) the temperature T_{nw} of the oil-air mixture, (ii) the value of the internal HTC, (iii) the thermal resistance of the chamber wall, (iv) the accuracy and approach used for measurement of specific temperatures, (v) the outer wall temperature determined by the external cooling conditions. High errors are observed for bearing chamber operation conditions, which characterized by low temperature T_{nw}, when the temperature difference decreases.

The maximum error of the internal HTC (15) occurs as a result of application of a typical scheme of independent measurement of three temperatures T_{nw} , T_{wi}, T_{we} when determining the heat flux density (1) and the internal HTC (2). It is 40% higher in comparison with the scheme, which uses a differential thermocouple for independent determination of the heat flux density (1).

Other important parameters are thickness and material of the analyzed bearing chamber. For the independent temperature measurement scheme, the minimum relative error of the internal HTC is achieved in the case of equality of thermal resistances of internal heat transfer and thermal resistance of the wall (12). Since the internal HTC varies, it is impossible to implement such conditions for all bearing chamber operating conditions.

For the fixed chamber design and operating conditions, the temperature measurement scheme and accuracy, the error of the internal HTC (15) is determined by the temperature T_{we} of the chamber outer wall. It can be reduced by decreasing the external cooling media temperature or increasing the external HTC. When using atmospheric air, heat exchange by radiation makes a significant contribution to external cooling.

For experimental determination of the internal HTC, it is recommended to use the temperature T_{nw} of the oil-air mixture as the determining temperature of the flow core. When analyzing the error of the internal HTC for different bearing chamber operating modes, the expected value of the internal HTC should be limited by the condition (26), that the temperature of the cooled air-oil mixture $T_{m\ out}$ is equal to the temperature of the inner wall T_{wi}.

Additional errors can be caused by the non-uniformity and non-stationary temperature field in the experiment.

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