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Application of additive laser technologies in the gas turbine blades design process

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Abstract. An emergence of modern innovative technologies requires delivering new and modernization existing design and production processes. It is especially relevant for designing the high-temperature turbines of gas turbine engines, development of which is characterized by a transition to higher parameters of working medium in order to improve their efficient performance. A design technique for gas turbine blades based on predictive verification of thermal and hydraulic models of their cooling systems by testing of a blade prototype fabricated using the selective laser melting technology was presented in this article. Technique was proven at the time of development of the first stage blade cooling system for the high-pressure turbine. An experimental procedure for verification of a thermal model of the blades with convective cooling systems based on the comparison of heat-flux density obtained from the numerical simulation data and results of tests in a liquid-metal thermostat was developed. The techniques makes it possible to obtain an experimentally tested blade version and to exclude its experimental adjustment after the start of mass production.

1. Introduction

The operating life of a gas turbine engine is primarily determined by the operating life of the HPT blades. Development of the blade with an effective cooling system is a complex and time-taking process that includes selection of a cooling path design and its optimization, the gas-dynamic, thermal and strength calculations, experimental tests, the manufacturing technology development and reliability assessment.

Hydraulic and thermal models of the cooling path, allowing to determine the boundary conditions from the gas and cooling air sides, are used for calculation of a blade thermal state under operating conditions. An accuracy of obtained results that are based on hydraulic resistance values of the channels with intensifiers and the criterion dependencies for calculation of heat transfer from the channel walls to the cooling air is mostly effected by efficacy of the hydraulic and thermal model of the cooling path. Possible failure of the thermal and hydraulic model, the actual fabricated blade, is due to the fact that hydraulic resistances and criterion equations used while designing are generally obtained on the models with a constant cross section having a size that is larger than the channels of an actual blade and under simulation conditions that deeply vary from actual conditions. Data on calculation of the heat transfer



on transition sections of channel connections, corners and the channel branching is limited. This all impairs the computational accuracy of blade thermal fields and thus the strength coefficients.

Verification of the thermal and hydraulic model is carried out by testing an actual blade manufactured in batches by the lost-wax casting. The major non-compliances of thermal and hydraulic characteristics identified during the tests with calculated values require changes in design and technological documentation, further development or even production of new moulds for manufacturing of ceramic cores that form the internal cavity of the blades being cast. This fact leads to significant additional costs and additional time for design of blades and an engine in whole.

The use of additive laser technologies, especially, selective laser melting technology (SLM-technology) makes possible to produce the prototypes of a blade and a model of separate cooling channels made of metal in 1:1 real size scale. Manufacturing accuracy of the blade internal cavity elements using a selective laser melting technology is close to accuracy of a precision investment casting.

The use of this technology provides, during the selection stage of the channel and heat transfer intensifiers geometry, performing of experimental studies to determine local heat transfer coefficients to the cooling air on the channel models having geometry corresponding to actual feather internal cavity, and during the final design stage – performing of predictive verification of the thermal and hydraulic model for the blade cooling system in whole.

2. A design technique for gas turbine blades

Excluding the experimental adjustment stage of the blade during the test stage of a prototype engine is the main task of design technique being developed. This may be achieved through the predictive verification of thermal and hydraulic models for the blade cooling system prior to development of the design documentation.

During designing of blades, after selection of cooling system, not only the blade thermal condition shall be taken into consideration but also its stress condition together with the effect of thermal loads associated with a nonuniform temperature field in the blade cross sections.

During design it is reasonable to use the relative air flow for blade cooling and the minimal local strength coefficient as efficiency functions. In this case, a design task solution is limited to seeking an extremum for one of the mentioned design tasks while restricting another within certain limits or finding a compromise solution.

A flowchart for the proposed design technique of the blade cooling system is given in figure 1 [1].

It consists of two interconnected modules, thermal and strength design modules, each of which includes a number of blocks. During the cooling system design the change in varied parameters shall not result in fundamental changes of the selected cooling scheme, and also in significant changes of geometric characteristics of blade cross sections. Varied parameters are: geometric dimensions of channels, means of heat transfer intensification in channels and barrier cooling elements. It is also possible to vary the center-of-gravity position of different blade sections in order to compensate stresses caused by the bending loads. It is necessary to keep in mind that the bending loads act in blade sections due to aerodynamic forces, which also need to be considered. Geometry of the feather external surface is determined by the calculation results for three-dimensional flow process and turbine cascade tailoring, and is not changed during the cooling system design. The thermal module consists of five blocks, each of which performs an independent function.

Block 3 is designed to calculate the air flow distribution in the cooling path channels. The following values are calculation results: flow rate value G_{ai} , Reynolds number Re_{ai} , air temperature T_{ai} , total air flow through the blade G_a , and relative air flow for blade cooling $F_1 = G_{cool}/G_c$ (G_{cool} – total air flow for blade cooling; G_c – air flow through the engine compressor).

Results from block 3 are used in blocks 4 and 5 for calculation of the cooling air flow through the blade and heat transfer boundary conditions on the blade external surface, while in block 6 – for calculation of heat transfer boundary conditions in cooling system channels.

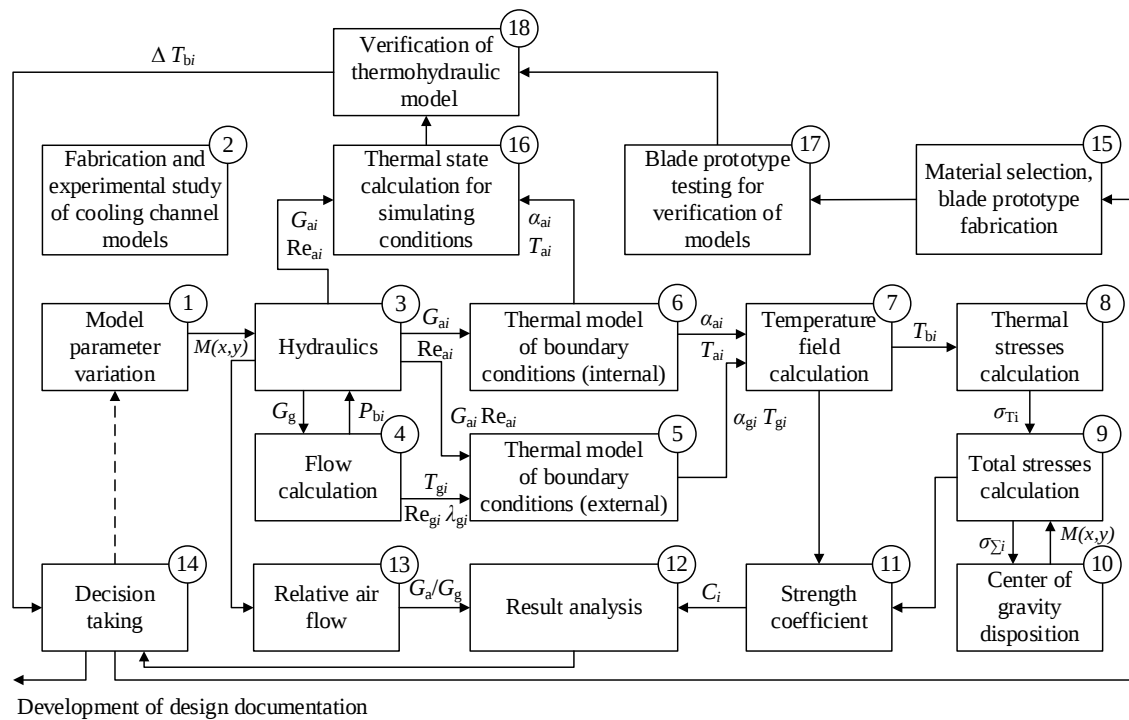


Figure 1. Flowchart of blade design technique.

Geometric parameters of channels and heat transfer intensifiers which a designer selects from the heat transfer study materials relying on own experience are initial data for block 1. An additional experimental study of models for particular areas of cooling channels (block 2) is proposed at this stage. Channel models are fabricated using SLM-technology and fully corresponds to the internal cavity geometry of the blade cooling path section being studied. Obtained experimental data allows a designer to use accurate values of hydraulic resistances and criterion dependencies for drawing the hydraulic model (block 3) and thermal model (block 6). This approach allows to improve significantly the quality of developed models of the cooling system.

Block 7 is designed for calculation of temperature fields in blade sections for different operating conditions. The strength design module consists of four blocks. Block 8 is used for determination of thermal stresses in blade sections caused by temperature field nonuniformity. Block 9 for calculation of total stresses allows to determine stresses caused by centrifugal loads and loads from gas forces, and also to find the total stresses from all actual factors. There, in addition to the information delivered from block 8, the gas loads, rotor speed, material characteristics and blade geometrical parameters are used as input information. Block for compensation of bending loads (block 10) allows to vary the center-of-gravity position in each blade cross section in order to compensate the active bending stresses from the gas and thermal loads.

Block for calculation of strength coefficient (block 11) allows to determine the distribution of strength coefficient C_i in the blade cross sections and its minimum value which is the efficiency function F_2 . Information from strength design module 9 is delivered in this block in the form of total stresses $\sigma_{\Sigma i}$ and from thermal block 7 – in the form of temperature values T_{bi} , that allows to determine the characteristics of a long-time strength of the blade material.

Having the efficiency function values F_1 , F_2 and information on blade temperature field, a designer makes decision in block 14 on a conformance of the cooling system with technical requirements. In a case of negative findings in block 14, a designer analyzing the strength coefficient fields in blade cross sections makes changes in geometry of cooled channels for correction of the temperature field in blade sections.

Once the decision on conformance of the developed blade with specified criteria has been taken, experimental tests for verification of the hydraulic and thermal model of a blade is carried out prior to the preparation of structural documentation.

In accordance with the proposed techniques the material is selected and the blade prototype is made for performing the thermal and hydraulic tests (block 17). 3-D geometric model of the blade obtained following the design is used for fabrication of its prototype by SLM-technology. Prototype is fabricated in 1:1 real size scale. Depending on the test method and program, a blade prototype may be fabricated with holes in the channel walls to measure the static pressure, with flanges for connection of the blade to the working part of a testing bench and etc. [2, 3, 4]. A prototype made in such way does not require performing additional works on dissection.

Calculation of the coolant flow distribution and blade thermal state for simulative conditions of prototype tests is carried out in blocks 3, 6 and 16. A steady or non-steady temperature field as well as distribution of thermal flux density along the feather surface can be a calculation result. Selection of the calculation result is determined by the blade design and test method based on the results of which the verification of blade thermal model will be carried out.

Verification of the thermohydraulic model is carried out in block 18 by the comparison of prototype test results with calculation results. When there are irregularities, their effect on efficiency functions F_1 , F_2 is determined. If necessary, the hydraulic and thermal blade model is rectified in order to determine required structural changes in a cooling system. The whole cycle of calculations, fabrication and prototype tests is repeated.

3. Experimental verification

It will be useful to select parameters reliably measured during the blade prototype tests as criteria for evaluation of the hydraulic model efficiency. As the main criteria it is reasonable to use the following: total air flow through the blade G_a measured at isothermal conditions (temperature of cooling air is equal to blade temperature – cold blowdown); total air flow at heating conditions $G_{a,h}$ (cooling air temperature is lower than blade temperature – hot blowdown).

Blowdown of the blade prototype is performed in the pressure difference range of $\pi = P_b/P_0$ from 1 to $\pi_{\max} = P_{b,\max}/P_0$, where P_0 – atmospheric pressure. Maximum pressure value $P_{b,\max}$ is set on the basis of a working pressure difference value on the blades in operation conditions π_w , usually $\pi_{\max}$ is approximately equal to $1.5 \cdot \pi_w$. Calculation curves $G_a = f(\pi)$ are drawn on which the experimental points are plotted. Deviation of the total design air flow from measured in an experiment is considered an acceptable within $\pm 5\%$.

Measurement of static pressures in feather points corresponding to nodal points of the design hydraulic circuit is performed for blades with complex cooling schemes. Pressure values P_i measured in the experiment are compared with design pressure values P_{id} , obtained for set pressure difference $\pi = P_b/P_0$.

Determination of total pressure distribution heightwise of the trailing edge slit is an additional parameter for examination of the hydraulic model efficiency. Obtained total pressure distribution is compared with the flow characteristics on the branches simulating the trailing edge slit. Dependencies $G_i/G_x = f(H)$ and $P_i/P_x = f(H)$ are plotted for comparison (H – slit height, x – section heightwise of the trailing edge, usually equals to $0.5 \cdot H$). Differences between calculated and observed values shall not be more than $\pm 5\%$. This deviation in flow rates will result in a change of heat transfer coefficients not exceeding $\pm 3.6\%$.

Analysis of methods for determination of thermal characteristics of blades with a convective cooling showed that the calorimetric measurement method in a liquid-metal thermostat is most informative and not requiring the significant material and time costs [3].

At given engine parameters that determine temperatures of gas flow through the turbine blades T_g^* and cooling air T_a , the heat transfer rate in cooled blades is characterized by dimensionless relative temperature of a given point of the feather (equation (1)) [3].

$$\Theta = \frac{T_g^* - T_b}{T_g^* - T_a} = \frac{C_{sh} \cdot \alpha_a / \alpha_g}{(C_{sh} \cdot \alpha_a / \alpha_g) + 1}, \quad (1)$$

where T_b – blade temperature;

α_g – coefficient of heat transfer to gas;

α_a – coefficient of heat transfer to cooling air;

C_{sh} – shape factor.

Shape factor takes into account difference of blade temperature from temperature of a thin flat wall. A specific characteristic of the shape factor is that it depends very little on heat transfer conditions and can be easily determined analytically for those simple geometric shapes.

At constant values of local heat transfer coefficients from the gas to the external surface α_g , for each feather point the equation (2) is correct.

$$\frac{\Theta_{bc}^{-1} - 1}{\Theta_{bp}^{-1} - 1} = \frac{C_{sh} \cdot \alpha_{ap}}{C_{sh} \cdot \alpha_{ac}} = \frac{q_c}{q_p}, \quad (2)$$

where Θ_{bp} – dimensionless temperature of blade surface obtained based on calculation results using the verified thermal model;

Θ_{bc} – dimensionless temperature of blade surface obtained based on the test results in a liquid-metal thermostat for determination of local heat transfer coefficients α_a ;

q_p – thermal flux density calculated for test conditions using the verified thermohydraulic model;

q_c – thermal flux density obtained by experiment during the prototype tests in a liquid-metal thermostat.

Given that the values of experimental and calculated air flow rates are matching than the equation (3) is correct.

$$\frac{\alpha_{ap}}{\alpha_{ac}} = \frac{\lambda_{ac}}{\lambda_{ap}} \cdot \left(\frac{\mu_{ac}}{\mu_{ap}} \right)^{-m}, \quad (3)$$

where λ_a – air thermal conductivity;

μ_a – air viscosity coefficient;

m – index of a power in criterion equation for calculation of α_{ap} .

Using the equation (4) for calculation of changes in thermal conductivity λ_a and air viscosity μ_a and taking into account representative values of index of a power at Reynolds number of $m = 0.6-0.8$ the equation (5) is correct.

$$\lambda_a = f_1(T_a)^{0.64}, \quad \mu_a = f_2(T_a)^{0.76}, \quad (4)$$

$$\frac{\alpha_{ac}}{\alpha_{ap}} = \left(\frac{T_{ac}}{T_{ap}} \right)^{0.64-0.76m}. \quad (5)$$

If $m = 0.8$, than a difference in temperatures by 1.5 times will result in a difference in $q_c/q_p = 1.013$. Using equation (2) and being given Θ_{bp} for this feather section the allowed deviation $(q_c/q_p)_d$ of q_p from q_c will be evaluated by allowed deviation of $\Delta\Theta$ (equation (6)).

$$\left(\frac{q_c}{q_p} \right)_d = \frac{(\Theta_{bc} - \Delta\Theta)^{-1} - 1}{\Theta_{bp}^{-1} - 1}. \quad (6)$$

If the obtained parameter value $q_c/q_p \leq (q_c/q_p)_d$, than we can consider that thermal model sufficiently describes the internal heat-transfer processes at this feather section.

It is advisable to do comparison of q_c and q_p for several test modes, in other words for different pressure differences. q_c/q_p value shall not be essentially changed depending on pressure difference P_b/P_0 . In this case the main error in criterion dependencies of the thermal model will be related to a coefficient value at Re. If comparison of experimental and calculated heat fluxes q_c/q_p obtained for different differential pressures give the same value of deviation on the concerned sections, than it may be concluded that power m in used criterion dependence $Nu_i = A_i \cdot Re_i^m$ corresponds to the flow pattern of cooling air, by this it is meant index of a power m.

4. Conclusion

The design technique for gas turbine blades has been improved, the distinguishing feature of which is the experimental verification of the thermohydraulic model at the early design stages by testing the blade prototype manufactured using SLM-technology.

The technique makes it possible to obtain an experimentally tested blade version and to exclude its experimental adjustment after the start of mass production.

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