Technical solutions for increasing efficiency of gas turbine engines

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***Abstract*—A method for studying the thermal state of turbomachine blades is presented. A numerical investigation method takes into account the phenomenon of gas-dynamic temperature stratification. The possibility of increasing the efficiency of cooling turbine blades due to the phenomenon of gas-dynamic temperature stratification is considered. The possibility of improving the accuracy of the calculated forecasting of the thermal state of the blades by obtaining reliable data by developing a mathematical model and a unique software and information complex for modeling is discussed.**

***Keywords*— mathematical modeling, numerical methods, thermal protection, film cooling, software and information complex, dispersed flow**

I. INTRODUCTION

According to the Energy Strategy of Russia for the period until 2020, the wear of the active part of fixed assets in the electric power industry is 60-65%. On most turbines, the blades have exhausted their own resource and, therefore, their fast, high-quality and cheaper production is required. The traditional technology for manufacturing turbine blades is very expensive and takes a lot of time to prepare the production: from 6 months to a year. Therefore, providing thermal protection of the working surfaces of the turbine blades is a significant reserve for improving the technical and tactical characteristics of turbomachines, increasing the time of their operation, and reducing the cost of their maintenance. The development of effective thermal protection of the turbine blades is a complex and time-consuming process involving gas dynamic, thermal and strength calculations, the selection of rational cooling systems and their optimization.

The development of advanced gas turbine plants should ensure their operation in conditions of increasing the temperature of the working fluid in order to increase the efficiency while ensuring reliable and economical operation. A continuous increase in the parameters of the working fluid provides more heat-stressed states of gas turbine elements, especially turbine blades. Under such conditions, the elasticity limit of the blade material is sufficiently sensitive to temperature change, and with an increase in the shaft temperature by 10 K, the design life is reduced by half [1]. Therefore, knowledge of a more accurate distribution of the temperature field in the blade itself and the coolant temperature fields is required. The application of numerical simulation (CFD) allows modernizing the interaction of inviscid and viscous stationary flows with velocities, currents and heat transfer in rotating channels, turbulent heat transfer under conditions of favorable and unfavorable pressure gradients. The flows of liquids and gases play a decisive role in the gas turbine working processes and they are accompanied by non-stationary effects. Therefore it is necessary to take into account the influence of non-stationary phenomena on the operation of the scapula. The absolute error of the calculated forecasting of the blade temperature at the stage of its design should not exceed 20-30 K, which increases the requirements for CAD [5,7,9-10].

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II. FEATURES OF HEAT EXCHANGE AND GAS DYNAMICS IN TURBOMACHINES AND RESERVES OF IMPROVEMENT OF THERMAL PROTECTION

The formation of a laminar boundary layer begins with flow around the inlet edge by a gas flow. Under the influence of various factors in a laminar interface, there are vortex indignations, which are called waves of Tollmien–Schlichting. They may be generated by a streamlined surface roughness or as a result of external influences. These external influences may be the acoustic waves or turbulence flow. Moreover, the cause of a laminar boundary layer perturbation eddy vibration may also serve as the rigid surface. At a low Reynolds number, laminar flow remains stable. When a certain point is reached, the critical Reynolds number, vortex perturbations are beginning to grow and there is a loss of stability of the laminar boundary layer. Determining the extent of areas on the blade profile with laminar, transitional and turbulent flow, it is important to determine flow patterns and heat transfer in each region, which determine the distribution along the blade profile of local heat transfer coefficients from the gas to the blade surface. The gas turbine engines with moderate Reynolds number of initial temperature of the gas turbine blades are typically $10^5…10^6$. In high-temperature gas turbine engines, the Reynolds numbers reach $9{\cdot}10^6$. The average value of the perimeter of the profile of the heat transfer coefficient on the gas side is 200 – 1200 W/(m^2·degrees), and in the high-temperature gas turbine engine – up to 5000 W/(m^2·degrees).

The boundary layer not only passes from the laminar flow to the turbulent flow, but an inverse process, called relaminarization, is also possible. Inverse transitions occur only a sufficiently large negative pressure gradient, with longitudinal nonisothermicity etc. In addition, an important impact on the processes of heat exchange provides the boundary layer separation from the aerodynamic surface. This phenomenon is currently poorly understood. Therefore, the simulation is performed approximately.

The temperature factor should be considered when moving from wall to flow. G. Lipman and G. Phil show that the Reynolds number decreased by half compared to the isothermal boundary layer with the difference in temperature between the wall and the flow $\Delta T = 100°C$. The temperature profile for the boundary layer of the air flow flowing around the plate is obtained without a convex inflection point. Consequently, a negative pressure gradient leads to stabilization of the flow around the cooled surface, and a positive pressure gradient leads to destabilization. When the wall temperature is less than the critical temperature Reynolds number of air flow is 1.5 times more as compared with the process in which these temperatures are equal.

Because of the complex structure of the boundary layer on the blade surface, it is advisable to use the concept of the effective viscosity of the boundary layer ν_ef; laminar boundary layer ν_ef = ν, i.e. normal kinematic viscosity coefficient, which depends on its temperature and pressure; in a turbulent boundary layer, as stated above, ν_ef = ν + ν_T; in a transitional boundary layer - ν_ef = ν + γ ν_T, where ν_T – turbulent viscosity, and gamma – intermittency factor in the transition zone. Thus, to calculate the local heat transfer on the profiles of turbine blades by solving differential equations of the boundary layer using the mixing path length model, determine the coordinates of the beginning and end of the transition of the laminar boundary layer into a turbulent one, and then find the length of sections of the profile with various flow regimes, to determine the coefficients intermittency in the transition zone, and after solving a system of differential equations using dependencies to find the effective viscosity of the distribution of velocities and temperatures in the boundary layer, the local values of the specific heat flux at the surface and the corresponding heat transfer coefficients.

By using different types of penetrating cooling cooler blowing on the surface of the profile, it is necessary to consider the effect of blowing on the boundary layer characteristics. Injection can be arranged in various ways, including when penetrating the porous cooling.

A method for calculating boundary layers on permeable surfaces in a continuous distribution of injection is largely developed, when the problem can be solved within the framework of the approximate boundary layer theory. Such problems are solved in the framework of the Navier – Stokes or their turbulent analogue – Reynolds equations. In addition, the heat transfer in the working blades is affected by additional factors due to the rotation of the rotor: centrifugal force and coriolis force.

For analysis of hydrodynamic and thermal processes in ducts of complex shape necessary to consider two–dimensional and even three–dimensional equations of motion and energy, since in such channels can transfer the amount of heat flows through the secondary channel in cross–section. Feature hydrodynamic and thermal processes in ducts of complex shape is that the coolant flow in the channel cross section substantially uneven, resulting in significant changes in the shear stress, heat flux and surface temperature over the cross section of the channel perimeter. The temperature in the coolant flow field on the surface is determined by the heat flux and surface interaction, so it is necessary to analyze the conjugate task. To determine the friction and the distribution of coolant flow can be used for calculating the approximate engineering method. Numerous experiments have shown that the stabilized developed turbulent fluid flow in channels of different shapes depending on the nature of the frictional resistance coefficient Reynolds number is the same as in the flow in the cylindrical channel[6,8].

III. MATHEMATICAL MODEL

High-precision research in nonlinear formulation is performed using numerical methods, taking into account the dependence of the thermophysical properties of the blade material on the temperature of the three-dimensional unsteady temperature field of the blade. Therefore, it is necessary to solve the nonstationary nonlinear spatial problem of the thermal conductivity of the blade [4]:

$$c_l \rho_l \frac{\partial T}{\partial \tau} = \frac{\partial}{\partial x}\left(\lambda_l \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(\lambda_l \frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(\lambda_l \frac{\partial T}{\partial z}\right) \quad (1)$$

where x, y, z are coordinates (m); T is temperature (K); τ is time (s); λ_l is coefficient of thermal conductivity of the blade material (W/(m·s)); c_l is specific heat (J/(kg·K)); ρ_l is the density of the blade material (kg/m^3); magnitudes λ_l, c_l, ρ_l depend on temperature.

For the transition from the differential equation of the thermal conductivity of the blade to the finite-difference method of thermal balances. In Fig. 1, the scheme to the calculation of a two-dimensional boundary layer for flow past the surfaces of the back and trough of shovels and the scheme for scanning the boundary layer in a Cartesian coordinate system are shown.

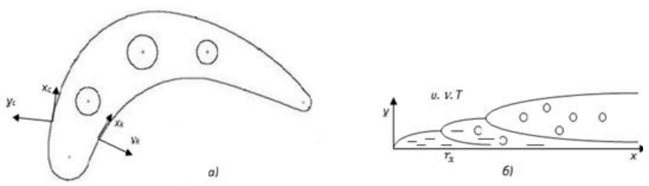

Fig. 1. Scheme for the calculation of a two-dimensional boundary layer in flow around the surfaces of the trough and the blade back in the curvilinear coordinates {x_c, y_c} and { x_k, y_k} (a) and the boundary layer scanning scheme in the Cartesian coordinate system{x,y} (б): r_x – is the radius of curvature; $r_x > 0$ for the backrest; $r_x < 0$ for the trough.

The density q of the heat flux during heat transfer from the working medium in the subsonic tract to the working body in the supersonic tract is expressed by the equation [2]:

$$q = k(T_{r1} - T_{r2}) = k\Delta T \, , \tag{2}$$

where:

$$k = \left(\frac{1}{a_1} + \frac{1}{a_2} \right)^{-1} \tag{3}$$

The density of the heat flux, and hence the intensity of the thermal stratification, increases with increasing temperature pressure ΔT_r, and with increasing heat transfer coefficient k. The temperatures T_{r1}, T_{r2} re expressed through the thermodynamic flow temperatures T_1, T_2 and the temperature recovery coefficients r_1, r_2 [2]:

$$T_{r1} = T_1 + r_1(T^* - T_1) \, ; \tag{4}$$

$$T_{r2} = T_2 + r_2(T^* - T_2) \, ; \tag{5}$$

$$T^* = T + u^2 / 2c_p \, ; \tag{6}$$

Using the phenomenon of gas-dynamic temperature stratification will reduce the consumption of cooling air as a result of intensification of heat exchange processes, which will reduce the cooling air outflow to the flowing part of the turbine and increase its efficiency [2].

For equation (1), the boundary conditions of the third kind in the analytical form are written as follows:

– for the blade surface from the gas side:

$$-\lambda_l \left(\frac{\partial T}{\partial n} \right)_w = a_g (T_r - T_{w1}); \tag{7}$$

– for the blade surface from the side of the cooler:

$$-\lambda_l \left(\frac{\partial T}{\partial n} \right)_w = a_{co} (T_{w2} - T_{co}); \tag{8}$$

– for the blade surface from the gas side in the flow separation channel:

$$-\lambda_l \left(\frac{\partial T}{\partial n} \right)_w = a_g (T_{w3} - T_{r2}); \tag{9}$$

The boundary condition (9) is applied to blades in which cooling channels are made in which a subsonic gas flow passes. For the transition from the differential equation (1) to the finite-difference equation, the method of thermal balances is used. The boundary conditions are determined by numerically solving the differential equations of the boundary layer. The system of equations describing the stationary heat transfer process on the surface of a flat blade can be represented in the form:

$$a_r = -\frac{\lambda}{|Tr - Tw|} \left(\frac{\partial T}{\partial y} \right)_{y=0} ; \tag{10}$$

– differential energy equation:

$$\rho c_p \left(\frac{\partial T}{\partial \tau} + u \frac{\partial T}{\partial x} + \upsilon \frac{\partial T}{\partial y} \right) = \frac{1}{r^n} \frac{\partial}{\partial y} \left[r^n (\lambda + \lambda_T) \frac{\partial T}{\partial y} \right] + + (\mu + \mu_T) \left(\frac{\partial u}{\partial y} \right)^2 + \frac{\partial p}{\partial \tau} + u \frac{\partial p}{\partial x} + q_v ; \tag{11}$$

– differential equation of motion:

$$\rho\left(\frac{\partial u}{\partial \tau}+u\frac{\partial u}{\partial x}+\upsilon\frac{\partial u}{\partial y}\right)=\frac{1}{r^n}\frac{\partial}{\partial y}\left[r^n\left(\mu+\mu_T\right)\frac{\partial u}{\partial y}\right]- -\frac{\partial p}{\partial x}+s_v; \quad (12)$$

– differential equation of continuity:

$$\frac{\partial \rho}{\partial \tau}+\frac{1}{r^n}\left[\frac{\partial}{\partial x}\left(\rho u r^n\right)+\frac{\partial}{\partial y}\left(\rho \upsilon r^n\right)\right]=0; \quad (13)$$

– equation of state:

$$\rho=\rho(T) \quad (14)$$

Additional terms q_v and s_v characterize the thermal and aerodynamic effects of particles on the carrier medium. The direct influence of particles on the structure increases with increasing longitudinal pressure gradient [4]. The flow of a dispersed flow in the subsonic path is characterized by the absence of a transverse particle motion in the boundary layer. This allows us to determine the coefficient of temperature recovery from the dependence for a homogeneous gas flow:

$$r_1=\sqrt[3]{\Pr}\,; \quad (15)$$

In the supersonic flow there is a transverse movement of the particles, so the dependence for determining the temperature recovery coefficient is given by:

$$r_2=\frac{\sqrt[3]{\Pr}}{1+28{,}6^{0{,}8}}, \quad (16)$$

where G – generalized variable has the meaning of a similarity criterion, which characterizes the effect of condensed particles:

$$G=\frac{|s_v|\mu_0}{\left(u_{sm}-u\right)\rho_0^2 u_0^2} \quad .(17)$$

The intensity of internal heat sources q_v and the amount of motion s_v applied to the dispersed boundary layer:

$$s_v=\frac{0{,}75\rho_s\rho c_{fs}}{\rho_v d_s}\left|u_s-u\right|\left(u_s-u\right), \quad (18)$$

$$q_v=\frac{6\alpha_s\rho_s}{\rho_v d_s}\left(T_s-T\right); \quad (19)$$

In the area of the blade surface with a laminar boundary layer, let us take $\lambda_T=\mu_T=0$.

On the surface area with a turbulent boundary layer, turbulent heat transfer coefficient λ_T can be determined from the relation:

$$\lambda_T=\frac{\mu_T c_P}{\Pr_T}\approx\frac{\mu_T c_P}{0{,}9}. \quad (20)$$

Turbulent momentum transfer coefficient μ_T in accordance with the Prandtl model of the mixing path is determined by the dependence:

$$\mu_T=\frac{\rho l^2\partial u}{\partial y} \quad (21)$$

where the length of mixing path l is calculated by the expression:

$$l=\text{æ}y\{1-exp[-\rho v y/(26\mu)]\}, \quad (22)$$

where ν – the dynamic velocity at the point under consideration.

Coefficient $\text{æ}=\text{æ}_T$ is determined by the dependence proposed by N.N. Kovalnogov [4], taking into account the influence of the dynamic non-stationarity factor, the longitudinal pressure gradient and the curvature of the streamlined surface:

$$\text{æ}=0.4\frac{\sqrt{1-\frac{4.9}{\rho_\infty u_\infty\left(\partial u/\partial y\right)_{y=0}}\left[\frac{\partial p_\infty/\partial \tau}{u_{0\infty}}+\left(\frac{\partial p}{\partial y}\right)_{\max}\right]}}{1-21.4\frac{\partial p_\infty/\partial x}{\rho_\infty u_{0\infty}\left(\partial u/\partial y\right)_{y=0}}} \quad (23)$$

where index ∞ characterizes the flow parameters in the analyzed section beyond the boundary layer.

In determining the heat transfer coefficients on the surfaces of the blades, the surface is divided into a number of characteristic regions, for each of which a similarity equation of the form $\overline{Nu}=c\,\mathrm{Re}^n$ is written.

For the input edge zone, the heat transfer coefficient is determined by the similarity equation:

$$\overline{Nu}_{ie} = 0{,}635\,\mathrm{Re}_{ie}^{0{,}5}. \qquad (24)$$

For the outlet edge zone, the similarity equation is used:

$$\overline{Nu}_{oe} = 0{,}0325\,\mathrm{Re}_{2oe}^{0{,}93}. \qquad (25)$$

The third definable characteristic area is the zone of the concave part of the profile - the trough. In active gratings, the average value of the heat transfer coefficient on the concave side:

$$\overline{\alpha_{gcs}} = (0{,}85....0{,}95)\overline{\alpha_g}\ . \qquad (26)$$

In jet gratings:

$$\overline{\alpha_{gjs}} = (1....1{,}2)\overline{\alpha_g}\ . \qquad (27)$$

The fourth characteristic area is the zone of the convex part of the profile - the back. In active gratings, the average value of the heat transfer coefficient at the back:

$$\overline{\alpha_{gcb}} = (1....1{,}1)\overline{\alpha_g}\ . \qquad (28)$$

In jet gratings:

$$\overline{\alpha_{gjb}} = (1{,}3....1{,}5)\overline{\alpha_g}\ . \qquad (29)$$

At the entrance of the back (0.6 - 0.7 length of the back):

$$\overline{\alpha_{geb}} = (0{,}75....0{,}95)\overline{\alpha_g}\ . \qquad (30)$$

On the output (remaining) part of the backrest:

$$\overline{\alpha_{gob}} = (1{,}2....1{,}4)\overline{\alpha_g}\ . \qquad (30)$$

In the channels of the working blades, the heat transfer coefficient is determined by the equation [1]:

$$\alpha_{\mathrm{cov}} = \alpha_{co} K_v.. \qquad (31)$$

The adequacy of the model of a turbulent dispersed boundary layer was verified by comparing the calculations of the heat transfer coefficients of a disperse flow in nozzles with experimental data. The comparison results given in [4] indicate that the proposed methodology as a whole adequately reflects the features of the exchange processes in the dispersed boundary layer.

In figure 2 an example of a thermal picture obtained with the help of a set of programs for calculating the thermal state of a turbomachine blade was developed on the basis of the Power Engineering and Fuels Department of Ulyanovsk State Technical University. From figure 2 that the inlet and outlet edges of the blade are the most heated.

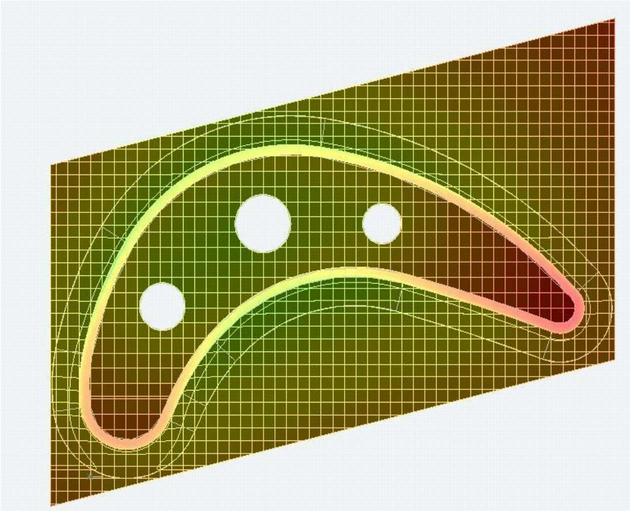

Fig. 2. An example of a thermal picture of a turbomachine blade for a warm-up time of 30 s.

IV. Results of the Study

At the Department of «Thermal and Fuel Energy» of Ulyanovsk State Technical University with the help of the developed program complex, the effectiveness of gas dynamic temperature stratification of dispersed flows in turbine units was studied. Variants with co-cooling of gas and air, with cooling of air and with heating a working body were considered [2]. The highest efficiency is achieved in the latter case, and this method of increasing the efficiency of the turbine installation is implemented in the RF patent for the invention (patent №2557793 Russian Federation: Gas-turbine engine / VN. Kovalnogov, D.A. Generalov, E.V. Shkolin. - Declared. No. 2014110128/06, March 14, 2014; Publ. 07/27/2015. Bul.№21). The gas turbine installation was calculated, taking into account the expected increase in temperature due to the effect of the gas-dynamic temperature stratification in the Leontiev tube. The increase in the efficiency of the thermal power installation was evaluated by the change in the effective efficiency. As a result of the calculation, the dependence of the efficiency gain on the temperature increase is shown (figure 3).

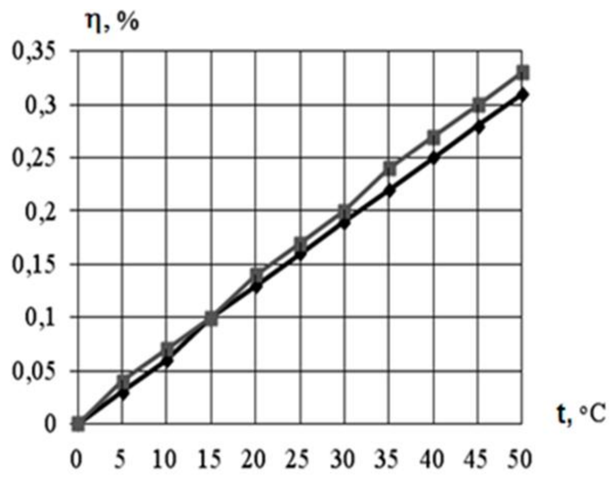

Fig. 3. Dependence of the increase in internal and effective efficiency on the increase in the temperature of the working fluid: ♦ – internal efficiency, ■ – effective efficiency

With an increase in the gas temperature by 50 ° K, the increase in effective efficiency is practically 0.35%.

Taking into account the positive results realized in patent (patent №2557793 Russian Federation: Gas-turbine engine / VN. Kovalnogov, D.A. Generalov, E.V. Shkolin. - Declared. No. 2014110128/06, March 14, 2014; Publ. 07/27/2015. Bul.№21), two more technical solutions were developed, increasing the technical and tactical characteristics of the engines, to which the decision was made to issue a patent for the invention of the Russian Federation (the numbers of positive decisions of applications for invention №2015155123 and №2015155124).

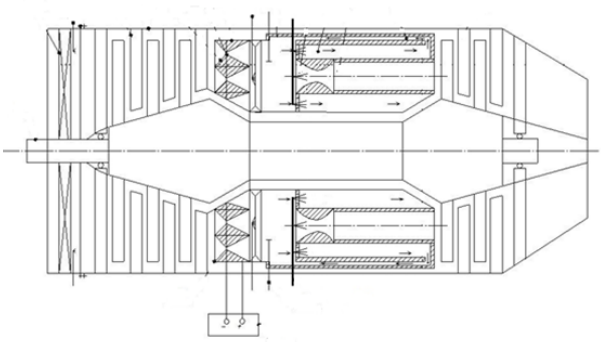

Fig. 4. Gas turbine engine with steam nozzles (first technical solution, patent for the invention №2015155123)

The second technical solution is to use the heat of the subsonic flow in the Leontief pipe (Figure 5). The temperature drop formed in the pipe leads to the appearance of a heat flow from the subsonic part of the current to the supersonic one. Subsonic flow, giving heat to supersonic flow, is directed through the channel of recirculation of subsonic current to the heat exchanger. The supersonic flow at the exit from the Leontief tube, more heated and with higher pressure, enters and expands in the turbine, rotates the motor shaft and is discharged into the atmosphere. Thus, the use of the heat of the subsonic flow in the heat exchanger will reduce fuel consumption and increase the economy of the engine.

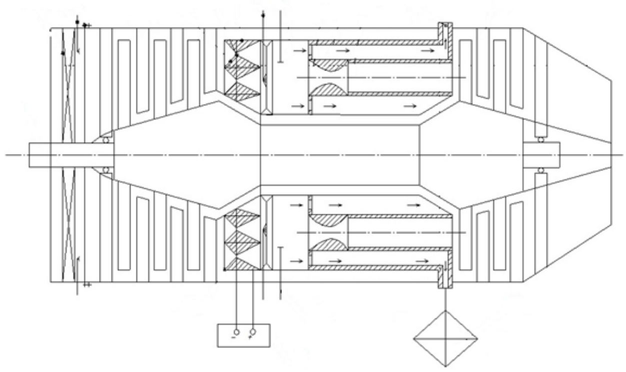

Fig. 5. Gas turbine engine with steam nozzles (second technical solution, patent for the invention №2015155124)

V. Conclusions

In order to develop new methods for thermal protection of the turbine machine's blades, to improve the technical and tactical characteristics of advanced gas turbines, the Ulyanovsk State Technical University has developed and introduced into the educational process a set of programs for calculating the thermal state of the blades of turbomachines. With the help of this software package, the above technical solutions for increasing the efficiency of gas turbine engines have been developed.

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