Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-46821

CONVECTIVE COOLING OPTIMIZATION OF A BLADE FOR A SUPERCRITICAL STEAM TURBINE

Grzegorz Nowak
Silesian University of Technology
Institute of Power Engineering and Turbomachinery
ul. Konarskiego 18, 44-100 Gliwice, Poland

Włodzimierz Wróblewski
Silesian University of Technology
Institute of Power Engineering and Turbomachinery
ul. Konarskiego 18, 44-100 Gliwice, Poland

Iwona Nowak
Silesian University of Technology
Institute of Mathematics
ul. Kaszubska 23, 44-100 Gliwice, Poland

ABSTRACT

This paper discusses the problem of blade cooling system optimization connected with Conjugate Heat Transfer (CHT) analysis for reliable thermal field prediction within a steam cooled component. Since the full CHT solution, which involves the main flow, blade material and the coolant flow domains is computationally expensive from the point of view of optimization process, it was decided to reduce the problem by fixing the boundary conditions at the blade surface and solving the task for the interior only (both solid material and coolant). Such assumption, on one hand, makes the problem computationally feasible, and on the other, provides more reliable thermal field prediction than it used to be with the empirical relationships.

The analysis involves shape optimization of internal cooling passages within an airfoil. The cooling passages are modeled with a set of four Bezier splines joined together to compose a closed contour. Each passage is fed with cooling steam of constant parameters at the inlet. In the present study the airfoil profile is taken as aerodynamically optimal. The search problem is solved with evolutionary algorithm and the final configuration is to be found among the Pareto optimal cooling candidates.

NOMENCLATURE

a, *b* – floating point number
C_x – axial chord, *m*
I – individual
M – Mach number
$\dot{m}$ - coolant mass flow, *kg/s*
p – pressure, *MPa*
q – constant parameter
T – temperature, *K*
Tu – turbulence level
ε - disturbance
x,y – global coordinates

Subscripts
max – maximum
i – nodal value

Superscripts
k – iteration (generation) number

INTRODUCTION

The continuous increase in the demand for energy is the key stimulus for the development of power engineering technologies. Not only is it now necessary to produce more power, especially electricity, but to produce it with due respect for the environment. Owing to the global warming effect, a great emphasis is put on the reduction in the emissions of greenhouse gases, CO_2 in particular. Therefore, the development of power engineering technologies aims at the improvement in the efficiency of thermal cycles which, with the same amount of fuel burnt and, consequently, the same amount of pollution generated, makes it possible to increase the capacity of facilities, or, at a fixed level of energy production, reduces fuel consumption and pollution.

One of the most dynamically developed technologies is the coal technology for supercritical steam parameters, where facilities with steam turbines constitute one of the major components. Due to the rise in steam parameters, it is possible to obtain an efficiency of electrical power generation exceeding 50%. This means that in many countries a coal-fired condensing unit will remain the basic source of electricity for many years to come. Power generation efficiency in a condensing unit depends mainly on the parameters of live steam and reheated steam at the inlet to individual components of the turbine, and grows as they rise. However, raising steam parameters is limited by the properties of the materials used in the manufacture of the machinery and equipment of power units (boilers, turbines, pipelines) which come into contact with the hot working agent. With the present state of the art, the temperature and pressure values usually range from 600-630°C and 26-30MPa, respectively.

A steam turbine which operates in cycles with supercritical steam pressure and with ever higher temperature is more exposed to heavy thermal loads. Meanwhile, the anticipated increase in steam temperature in planned facilities for supercritical parameters is by 50-100K higher as compared to those already in service.

A further growth in the efficiency of cycles for supercritical parameters involves finding a solution to a number of tasks concerning the structure of steam turbines. At present, main efforts to develop steam turbines are focused on the following problems:

- to increase the temperature before the blade system of the turbine,
- to introduce a cooling technology of the flow cycle at the inlet of the high- (HP) and intermediate pressure (IP) part,
- to reduce the number of the turbine cylinders (combining the IP and LP parts),
- to improve the life and reliability of those joints and fragments of the facility which are under heaviest loads,
- to introduce new solutions to the blades of the final stages (by lengthening and/or shrouding them).

Obtaining at the inlet the steam temperature and pressure of 920K and 30MPa, respectively, involves an application of cooling in the flow cycle. Such a solution allows the use of materials which are now being used in modern structures.

The problem of cooling turbine blades can therefore be taken into consideration while designing steam turbines working with high steam parameters. This concerns mainly the first stages of the high- and intermediate pressure parts. Convection cooling seems to be the most appropriate, as it is similar to the case analyzed in this paper. The technique of convection cooling of gas turbine blades can be used in steam turbines for ultra-supercritical steam parameters. Besides, steam used as a cooling agent has much better properties than air. However, modelling flow phenomena in these cases is much more complex, mainly due to the need to use the real gas model for steam.

This paper deals with the problem of optimization of the location and shape of the inner cooling passages of the blade of a steam turbine for supercritical steam parameters. It is assumed that the blade is cooled with straight passages of a complex shape, where the shape of the passage section is modelled by means of Bezier curves.

Because there is no experimental data for cooled blades of a steam turbine, an example geometry of the stator blade is assumed for the analysis and assessment of the developed numerical method.

The problem of optimization of the cooling system of turbine blades was confronted by many authors but mainly with reference to gas turbines. Few works concern the cooling of steam turbine components. Despite the similarities in the methodology of solving the optimization task for blades used in both types of turbines, the problem of the design of steam turbine blades is much more challenging from the point of view of individual solutions. It results from the fact that the optimization of the cooling system of gas turbine blades cooled with air was very often based on numerical simulations of the temperature field and/or the stresses in the solid body only, with simplified boundary conditions of the heat exchange in the passages. Due to very high intensity of the heat exchange on the external surface of the profile (steam with very high thermodynamic parameters), it is necessary to use steam with relatively high parameters for the cooling. For this reason, in order to make a proper thermal-flow assessment of the system of the cooled steam turbine blade, it is necessary to take into account the specificity of the working agent characterized by a non-linear real gas law equation and by substantial changes in the material properties. This is why the problem has to be solved as a conjugate task. Obviously, the solution of a single conjugate heat transfer (CHT) task does not pose significant problems at the moment. But adopting this approach in an optimization task, which entails an assessment of several thousand potential configurations of the cooling system, does create serious difficulties.

In reference literature there are examples of works which deal with the use of CHT analyses for the optimization task of the cooling system of gas turbine blades (Verstraete at al. 2008, Nowak and Wroblewski, 2009). One of the first works concerning an extension of the CHT algorithm to cover gas turbines cooled with steam was presented by Bohn et al. (2002). The use of the CHT algorithm for a closed steam cooling system in a modern steam turbine was described by Bohn et al. (2005).

The fundamental problem, connected with a cooling system and regarding turbine effectiveness, can be formulated in the following manner: for the allowable temperature distribution within turbine components it is necessary to build a cooling system causing the lowest possible energy dissipation. There are also other criteria which should be taken into account (i.e. lifetime, reliability, costs etc.) in present applications, and which determine the turbine performance. A precise solution to this task is nowadays rather vague because it calls for a rapid solution to Navier-Stokes and conduction equations for complex geometries and boundary conditions (Bohn et al., 2001). Some attempts especially in the field of Conjugate Heat Transfer (CHT) solution for cooled geometries, were presented by Facchini et al. (2004), Montomoli et al. (2004) and a similar task but for the case of rib-roughened cooling channels, is discussed by Fedruzzi and Arts (2004).

An appropriate selection of cooling system requires some optimization, which aims to search for such a configuration of the system that will provide fulfilling the fundamental economical, safety and reliability criteria. Among these criteria we can find:

- keeping component maximum temperature at allowable level

- reduction of coolant flow rate
- keeping component stresses at safe level
- providing sufficient durability of cooled component (machine)
- providing proper stiffness of a component

An internal cooling system, due to its discreet nature (local cooling with a number of passages), is a source of enhanced thermal gradients within the component, which in consequence produce higher thermal stresses and decrease the component's lifetime. So, a great care should be taken to keep the temperature variations as low as possible. To provide more uniform cooling of a component, the passages should be located in the areas of highest temperature, close to the external wall. Such an action leads towards more rational coolant usage by cooling only the spots that require it.

The problem of cooling system optimization seems to be quite new and has been the point of interest only in the last several years mainly in the area of gas turbine blades. An optimization of cooling system for steam turbine blades has hardly been considered in literature. This is due to the fact that most of the steam turbines have been working with steam temperature below the material's limits. Only the development of supercritical systems brought on the needs of blade cooling. Scientific research undertaken so far which has aimed to find the optimal cooling system has dealt with the internal passage convective cooling only. It seems to be the only cooling technique that can be, at the moment, analyzed as a whole. Dulikravich with his team (Martin and Dulikravich, 2001, Jeong et al., 2003, Dennis et al., 2003a, b) for several years have studied different problems concerning gas turbine blade optimization from the point of view of flow, thermal and structural criteria. The outcome of this research is the Multidisciplinary Computer Automated Design Optimization (MCADO) system. They dealt mainly with optimization of internal cooling passages (Dennis et al., 2003, Martin and Dulikravich, 2001) and possible coolant outflow at the trailing edge of the blade (Dulikravich et al., 1999). The works mentioned reveal that although they operate on many design variables, the search process was significantly restrained in terms of the geometry changes (i.e. optimization undergone passage fillets or passage distance from the wall etc.). In 2003 Dennis et al. (2003a) showed a work dealing with optimization of a large number of cylindrical cooling passages. All the passages were located close to the external wall and they could move only within a narrow strap along the wall. If a channel was unnecessary, on the basis of the criteria posed, it vanished. The aim of the research was to find such a passage distribution that would, by keeping the blade temperature at an allowable limit, minimize the heat flux and also the coolant usage. Very interesting seems to be another work of Dennis et al. (2003b), where the optimization of serpentine-like cooling passage was presented. This task was realized in 3D with the thermo-mechanical criteria involved. The results showed large potential hidden in a cooling system; appropriate size and location of channels had a great influence on the coolant usage, thermal stress level and in consequence lifetime of the component.

Recently Verstraete et al. (2008) presented a technique which implements a meta model of CHT based on Artificial Neural Network and Radial Base Functions. This meta model is a substitute for full CHT analysis for the optimization problem. It is however iteratively fed and improved with CHT solution data. Next year Nowak and Wroblewski (2009) showed a methodology of reduced CHT solution applied to a cooling system optimization within a gas turbine blade fed with air.

Most of the papers dealing with optimization problem used fixed thermal boundary conditions for the external surface of the blade. In the case of internal passages the heat transfer conditions were assessed on the basis of experimental relationships (usually Dittus-Bolter equation). Such estimation of thermal boundary conditions seemed to be very approximate but good enough to demonstrate the optimization techniques themselves. However, accuracy of heat transfer determination has a critical influence on the heat amount transmitted from the working medium to the coolant, which in turn determines temperature distribution within the cooled airfoil. If so, more precise estimation of thermal boundary conditions, both on the external surface of the blade and the cooling passages will influence the optimization as well. The final result will depend on the boundary conditions applied and their reliability will produce a robust solution. This paper demonstrates a possibility to implement the reduced CHT analysis mentioned for the cooling system optimization of a steam turbine blade, which is a step forward comparing to the approach used so far in gas turbines and one of the first solutions for a steam turbine.

The optimization problems dealing with turbine blade cooling systems usually involved quite a high number of design variables as well as a complex, implicit and susceptible to slight parameter changes, response (function) of the system in question. This response in a form of temperature, stress or other parameters was not monotonic but usually included a large number of local extremes, which was a significant problem for many gradient-based optimization methods (Mueller, 2002). In such cases, the stochastic methods were preferred, which was observed in almost all works mentioned before.

The most common was the evolutionary approach which apart from some disadvantages, was suitable to solve complex technical problems.

EVOLUTIONARY OPTIMIZATION

The optimization based on the evolutionary algorithm imitates real life with its evolution process of living organism. This method, which is based on a probabilistic search and an imitation of biological evolution, is able to overcome the settlement in local extremes, where many traditional optimization methods would finish their running.

The evolution process in a population comprised of abstract individuals, prepared for the optimization problem needs, takes place to produce better individuals in the successive populations till the optimal solution (best fitted individual) is reached. So the first step in the evolutionary search is the adequate preparation of the individual, which should include all the design parameters. In the optimization of a cooling structure the design parameters define all the variables which describe a specific configuration. The design variables in the evolutionary algorithm are usually binary or real coded to compose the chromosome of a single individual. In the case of floating point representation it has a form of a vector built of the design parameters:

$$I = [a_1, a_2, a_3, ... a_n] \quad (a_i \to \Re) \tag{1}$$

Similar to biology, each individual reacts somehow to the external (surroundings) conditions. This reaction is quantified and represents the level of adoption to the conditions. This value is called fitness. In other words it shows to what extend particular individuals fulfill the optimization criteria.

The population composed of a number of individuals, where the size of the population depends on the problem in question, is then subject to the evolutionary operations. Such operations change the genotype (some of the design parameters) of the members, which in consequence changes their fitness. The main genetic operations were already mentioned and they mimic the actual biological behavior of organisms:

- *recombination* – process in which two individuals share their features. For the real number representation this operation is based on weighted averaging.

$$\begin{matrix}[a_1, a_2, \ldots] \\ [b_1, b_2, \ldots]\end{matrix} \xrightarrow{\text{recombination}} \begin{matrix}[m_1 a_1 + (1-m_1)b_1, m_2 a_2 + (1-m_2)b_2, \ldots] \\ [(1-m_1)a_1 + m_1 b_1, (1-m_2)a_2 + m_2 b_2, \ldots]\end{matrix}$$

$$(m_i \to \{0,1\})$$

- *mutation* - introduction of a random disturbance (ε) is into the design variables.

$$[a_1, a_2, \ldots] \xrightarrow{\text{mutation}} [a_1 + \varepsilon_1, a_2 + \varepsilon_2, \ldots]$$

- *selection* – process which selects individuals for reproduction and creates the succeeding population. In this procedure we care for the best fitted individuals to be copied to the new population, and the worst members are eliminated. The process is controlled by reproduction operators.

The way of design variable coding can have an effect in the search process performance. Michalewicz (1996) proves that in technical applications where the domains are quite complex and the system response for the design variable change is very susceptible, the floating point coding is preferable.

For the purpose of this research, the Multi Objective Evolutionary Algorithm (MOEA) is used for the optimization search with use of the Pareto approach.

The Pareto optimization consists in searching for a set of what is referred to as non-dominated solutions.

To this set of non-dominated solutions (which is called Pareto set) belong solutions non-dominated in the sense of the domination relation $\prec$ defined as follows:

$$\boldsymbol{x} \prec \boldsymbol{y} \Leftrightarrow \forall k=1,\ldots m \; f_k(\boldsymbol{x}) \leq f_k(\boldsymbol{y}) \text{ and } \exists k \; f_k(\boldsymbol{x}) < f_k(\boldsymbol{y}) \qquad (2)$$

where notation $\boldsymbol{x} \prec \boldsymbol{y}$ means that $\boldsymbol{y}$ is dominated by $\boldsymbol{x}$, and $f_i(\boldsymbol{x})$ are numerical values of the i-th criterion in the problem being considered. If any two solutions belong to the Pareto set, it means that one is better than the other with relation to at least one criterion, and at the same time - worse with relation to another criterion. As a result, in the Pareto sense of optimization a group of solutions is obtained that cannot be compared (not being in any relation) to one another in the sense of the above mentioned definition.

Non-dominated solutions are arranged along what is referred to as the Pareto front, which is part of the boundary of the set containing the partial objectives (Fig 1).

The solution, defined eventually as optimal, can be chosen from this set for example on the grounds of other additional premises.

Because of too great a number of non-dominated solutions, in practice it is usually very difficult to find them all. Usually, only a finite group of representatives of the Pareto front is found on the grounds of whose location its approximate shape can be determined.

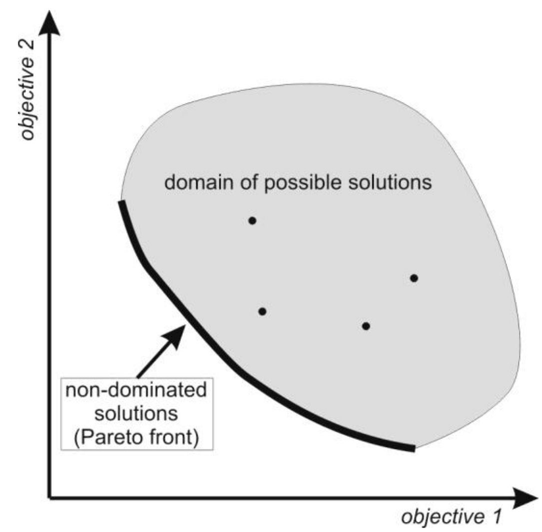

Fig. 1 Pareto domination

The optimization process is automatically coupled with the ICEM CFD and Ansys CFX software utilized for the computation of each cooling configuration.

Many solutions to the cooling of gas turbine blades are characterized by passages with non-circular sections whose shape corresponds to the shape of the profile, and the passages are separated with fins which ensure rigidity and strength of the whole structure. In blades of this type, although they are characterized by a high demand for the cooling agent, the volume of the solid material that needs cooling is much smaller, and in the case of rotating blades, reducing their weight is also important. On the other hand, it is worth remembering that the relatively thin layer of material between the passage and the outer wall results in great differences in temperature, which generates high thermal stresses.

In the analyzed method Bezier curves were adopted to model the cooling holes of non-circular shapes. A Bezier curve is built of Bezier segments, each of which is defined unequivocally by four control points. Although the number of the control points may vary, and although it affects the segment properties, it is usually assumed that a single segment is built on the basis of four control points according to the following formula

$$B(u) = P_1 u^3 + P_2 u^2 (1-u) + P_3 u (1-u)^2 + P_4 (1-u)^3 \qquad (3)$$

The value $B(u)$ denotes a point on the Bezier curve, P_1, P_2, P_3 *and* P_4 denote control points, whereas u is a real number from interval [0,1]. Adopting definition (3) cubic Bezier curve, i.e. smooth curve is obtained. Bezier segments can be closed and joined, which is done by defining the first and the last control point (of the same segment, or two different ones) at the same location.

In order to obtain holes properly "matching" the section of the blade, four Bezier segments were used to model each of them. Two of them (referred to as upper and lower) were to reproduce the shape of the blade profile, whereas the other two (right and left) were to close the contour of the cooling hole (as shown in Fig. 2).

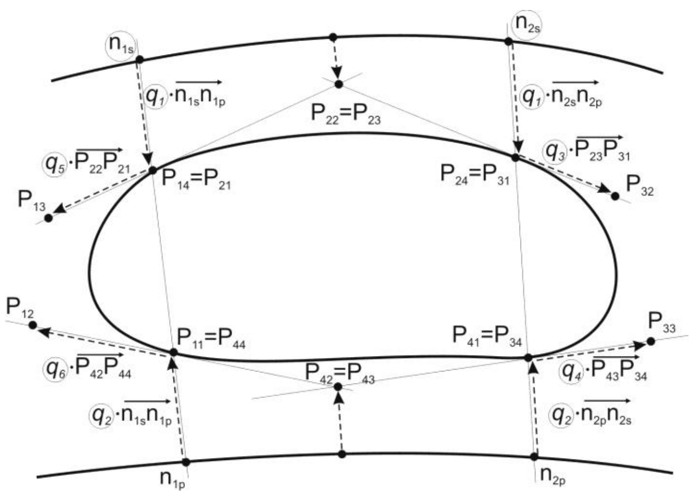

Fig. 2 Passage shape modelling.

As it was mentioned before, a single segment is based on four control points, but because of the joining of individual segments into a closed curve, some of the points coincide. In the presented approach, in order to reduce the number of variables, and to simplify the procedure, it was additionally decided that the second and the third point of the upper and lower segment coincide as well.

Eventually, the modelling of a single cooling hole requires the determination of 10 control points, and consequently 20 coordinates. In this paper the coordinates of all points are determined on the basis of 8 values ($n_1,n_2,p_1,p_2,p_3,p_4,p_5,p_6$), which in this case are estimated parameters. A sample cooling passage with marked control points and estimated values is shown in Fig. 2.

CALCULATION PROCEDURE

The aim of the present optimization process is to find optimal size and location of circular cooling passages. The optimization is to be approached with objectives formulated on the basis of an airfoil's thermal field. This requires specification of boundary conditions both on the external profile and the cooling passages. Since the optimization changes the cooling structure and in consequence the cooling conditions it would be necessary to adjust the boundary conditions to current cooling configuration. To take the changes into consideration a CHT problem should be involved. However full CHT is computationally expensive from the point of view of the optimization where usually many cooling candidates need to be analyzed. This paper tries to implement the CHT problem into the optimization task.

Model for CHT problem

The problem is solved for a 3D model for the assumed blade vane profile. The vane profile is assumed to be fixed during the computation process. The aim to be considered is to introduce a convective cooling for a homogeneous solid blade. The blade was made from the material used for the working steam parameters about 873K. The assumption when an increase of the life steam temperature to 923K takes place is to remain the same material for the blade and to reduce thermal load by means of cooling channels. The blade vane was investigated within a linear cascade supplied with life steam. Cooling of the 42 mm high vane was provided with steam supplied from the cool part of the superheater pipe. The cooling steam is flowing radially through four shaped channels.

The whole computational domain is 3D and consists of a periodic domain with a single vane. It is divided into three sub domains: life steam domain, solid domain of the vane and the cooling steam domain. The life steam domain was extended in the axial direction upstream of the blade leading edge and downstream of the trailing edge to eliminate influence of the uniform boundary conditions on the flow structures.

The vane material is martensitic stainless steel, which has a thermal conductivity specified as

$$k = = 0.01801\,(T - 273) + 23.88\ \text{W/m/K} \qquad (4)$$

a constant density of ρ=7900 kg/m^3, a molar mass of 55.85 kg/kmol and a specific heat capacity of:

$$c = 0.46+0.000177(T-273.15)+4.67e\text{-}7(T-273.15)^2\ \text{J/kg/K}.$$

The flow conditions were assumed from preliminary assumptions about the expansion line in the HP turbine, stage number and reaction. The power unit operates in the sliding pressure mode. All those factors enable to determine an enthalpy drop in the vane as about 15kJ/kg and as a consequence to assume static pressure at the outlet of the vane as 28.82MPa. The total pressure p_t=30MPa and total temperature T_t=923K were specified at the inlet,. For the life steam flow at the inlet, the turbulence intensity T_{u1} was specified to be Tu_1=10%

Both the life steam and the cooling steam are calculated as a real gas using IAPWS standard [26]. It concerns as well molecular viscosity, conductivity and specific heat of the water steam.

The flow conditions for the coolant flow in the passages were specified by boundary conditions independently on the cooling channel position and it's geometry. For all cooling channels equal boundary conditions were assumed. At the inlet the total pressure p_{tc}=5MPa and total temperature T_{tc}=600K were taken. The flow at the inlet had a high turbulence level Tu_{1c}=10%. At the outlet the static pressure was p_{2c}=4.2MPa.

The end walls at the hub and shroud were assumed as symmetry. Simplification of the model made it possible to avoid a boundary layer modeling in those regions.

Numerical solution and turbulence model

The flow solver was the commercial CFD package ANSYS CFX. It solves 3D RANS equations on unstructured meshes of different cell types and a mixed structure. The solution strategy is based on the algebraic multi-grid method. For Conjugate

Heat Transfer analysis, the energy equations for the fluid and solid are solved simultaneously. The energy equation for the solid is a degenerate form of the energy equation for fluids and is solved using the same numerical algorithm.

The turbulent eddy viscosity was obtained from the SST turbulence model. The SST turbulence model was modified using Kato-Launder formulation and curvature correction which are recommended for such a problem. The Gamma-theta transition model is used to simulate boundary layer transition, which is one of the key features of blade heat transfer. In the Gamma-theta model two additional equations for the intermittency and transition onset Reynolds number are solved.

Mesh

The model geometry was created and meshed using the ICEM CFD package. Since the C3X vane had a constant cross section, a 2D mesh for all domains was generated first. For the external life steam path region the quad dominated mesh was chosen with elements evenly distributed in the layers near the wall boundary. It ensures nearly orthogonality of the mesh near the vane surface. In the solid region and in the cooling channels the unstructured quad dominated mesh was also used. The cooling channels. The merged 2D mesh was then stretched up to form the 3D mesh. The 3D grid had 25 layers to reduce the total mesh size, because the reduction in time consumption was strongly preferred in the optimization process.

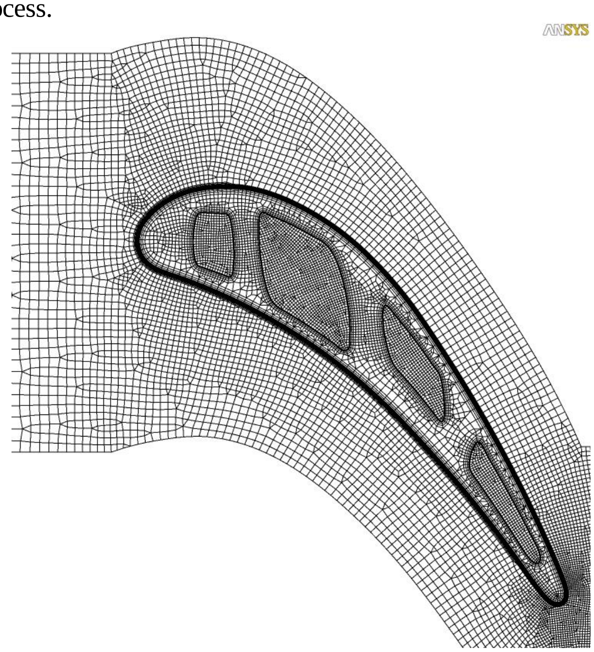

Fig 3. Computational meshes for all domains

Totally, the mesh consisted of about 900000 cells for the entire domain, with about 590000 cells for the hot gas domain, 270000 for the solid domain and 8000-25000 for the each cooling passage depends how long the perimeter of the hole is.

The sample of the computational grids on the one layer are shown in Fig. 3. The near-wall spacing of the life steam mesh is about y+=1.

CHT solver protocol

The optimization procedure calls the CHT solver many times. Implementation of the full CHT problem for the optimization seems to be irrational in terms of computational cost. In order to make the problem feasible to be considered it was decided to reduce its size. There assumed that instead of full CHT solution it would do, at this stage, to take into account the CHT problem within the airfoil only (airfoil material and cooling passages), with fixed boundary conditions at the blade wall. Those boundary conditions are obtained from the full CHT prediction for the initial configuration (Fig.3). The algorithm for the CHT solution consists of three steps. In the first step the cooling passages generated by the evolutionary algorithm are automatically meshed using a script file for ICEM CFD. Next, new meshes for the solid and the cooling channels are reloaded into the simulation file in the pre-processing of the ANSYS CFX and the solver is started. When the solver reaches the prescribed residual level (set to 1e-4 for the maximum residual) results in the third step are loaded to the postprocessor and necessary data for optimization procedure are exported. All steps are controlled by user defined procedures and proceed automatically.

Cooling structure coding

Each individual of the genetic population represented the cooling system configuration, which was equipped with four non-circular channels and during the optimization the number was fixed. Each passage required 8 parameters to be uniquely defined. Taking into account the whole cooling system it made the optimization problem formulated in 32 dimensional design space. All the design parameters were stored within a design vector.

The objectives were defined in the domain of the blade cross-section, so the constraints needed to be properly determined. The cooling passages might freely move within the blade domain, but care should be taken to prevent passages from overlapping and crossing the boundary. Even a single passage which violated the boundaries made the individual discredited for further calculations. This was because of an automatic model generation for CHT computations. To meet the modeling requirements both minimum mutual distance between neighboring passages and wall distance was restrained to 2mm. Additionally, the minimum and maximum passage size in the chord-wise direction were specified.

Due to a complex shape of the design parameters domain it was necessary to introduce a local coordinate system connected with the airfoil shape and to perform any modifications of an individual within this system. The local system was curvilinear, where the abscise axis followed the suction side of the blade and the ordinate was always perpendicular to it. Both coordinates were normalized to interval (0:1).

Evolutionary search

At the beginning of the calculation process the base (initial) population was sampled with the constraint criteria being satisfied. So the whole base population was composed of a set of feasible cooling configurations. To determine the fitness function value for each cooling system candidate the chromosome was decoded (specific values were transferred from local to global system) in order to build the computational model. Then a command file for ICEM software was prepared for the flow and thermal model grid generation. In the next step the CHT computations, which involved the blade domain with cooling channels were performed giving as a result the temperature distribution within the airfoil and the flow field in the passages. On the basis of the results obtained the fitness of the specific solution was calculated. This process was repeated for each population member.

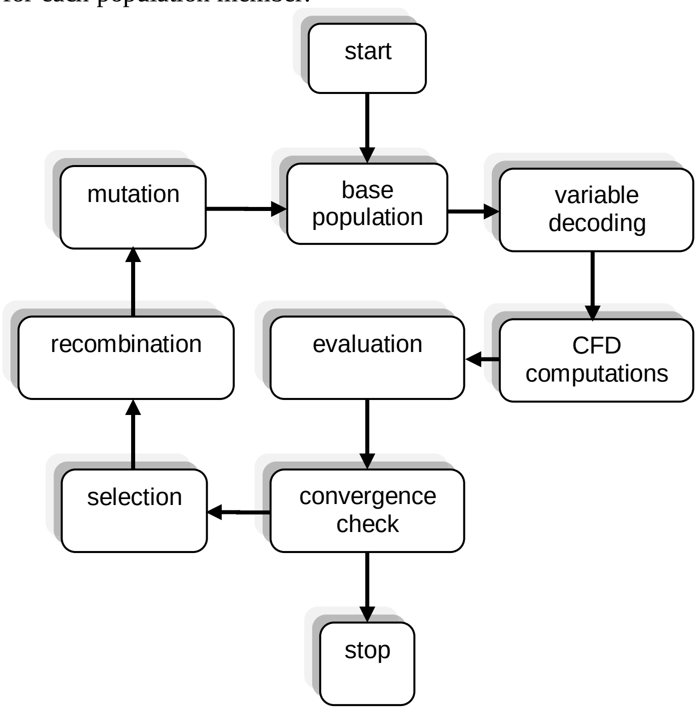

Fig. 4 Evolutionary algorithm flow chart

After that, the population was subject to the genetic operations and in consequence a new offspring population was obtained. Then the constraints for each individual of the children population were checked and the process was repeated (Fig. 4).

Some preliminary calculations have shown that the population of about 60-100 individuals was sufficient to solve the problem posed. Individuals that belong to the non-dominated set are stored within a pool and their copies undergo evolutionary operations to produce successive population. The evolution takes place until the whole population consists of non-dominated solutions only.

The individuals were subject to recombination with probability of 60% and the probability of mutation was 5% for each design variable.

The range of mutation operation shown above was after Michalewicz (1996) changeable (diminishing) with generation number in order to provide better tuning of individuals along with the algorithm advance. A variable (x_i) in generation k of the maximum generation number K undergone the mutation was modified by the following manner:

$$x_i^{k+1} = x_i^k + (UB - LB) \cdot \left(1 - RND^{\left(1-\frac{k}{K}\right)^q} \right). \qquad (5)$$

UB and LB defined the upper and lower domain boundary of the x_i, respectively, and q was a system parameter (constant). One could have seen that for a relatively small number of generation the disturbance range covered the whole domain, whereas while k approached the K, it narrowed.

The iteration process was terminated when the convergence criteria were met, usually if fitness function variations adhered to a certain limit for an assumed number of generations.

NUMERICAL INVESTIGATIONS

The methodology described above was used to find an optimal shape and distribution of a cooling configuration for the steam turbine blade in question. There was decided to build the cooling system based on four non-circular cooling passages whose shape was modeled with the technique discussed previously.

The optimization was carried out for two objectives only in order to present clearly the set of non-dominated solutions in the objective space. Both objectives (maximum temperature and maximum thermal gradient) were determined on the basis of the thermal field within the solid material which resulted from the reduced CHT analysis. All the computations were performed for set cooling conditions within the passages and for constant boundary conditions on the profile. After convergence one of the non-dominated solutions was selected and verified with the full CHT analysis.

Results

The steam flow in the vane blade-to-blade channel is characterized by relatively low Mach numbers. Fig.5 presents a Mach number contour in the main flow. The maximum value does not exceed 0.3. It is a consequence of the assumption of enthalpy drop in the vane.

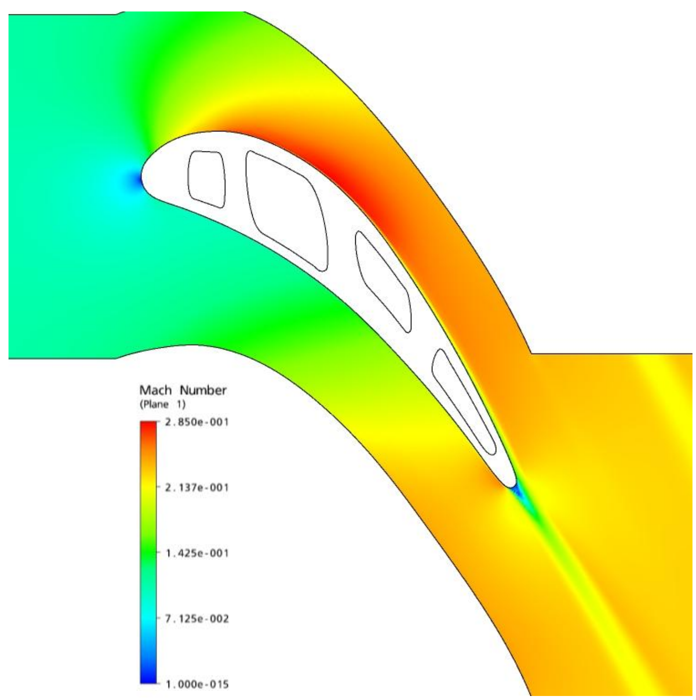

Fig.5 Mach number contour

The relative pressure distribution on the blade wall is shown in Fig.6 The CHT analysis carried out for the initial cooling configuration enabled proper formulation of the boundary conditions for the optimization purposes. The result of the CHT calculation was the heat transfer coefficient on the blade wall calculated with assumption of bulk temperature T_b=923K. The distribution of heat transfer coefficient is shown in Fig.7. The values change in a very wide range, from 10000 to 50000 W/(m^{2}K). The distribution exposes a sudden increase on the suction side for *x/Cx*=0.35, where *Cx* means the axial chord. This phenomena is caused by the laminar-turbulent transition.

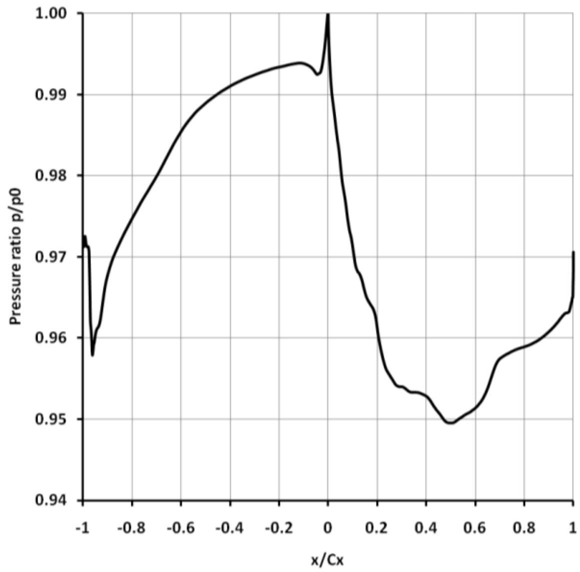

Fig.6 Pressure distribution on the blade

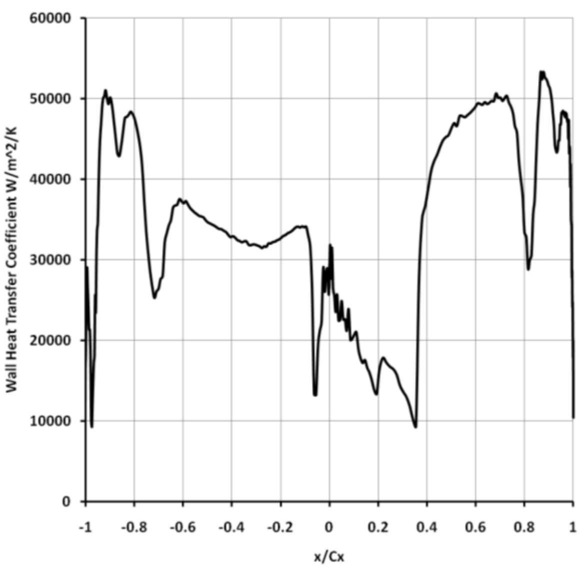

Fig.7 Wall heat transfer coefficient assumed for the optimization

The results of the optimization are presented in the objective space in Fig. 8. The non-dominated solutions form a Pareto front in the *maximum temperature–maximum thermal gradient* (T-gradT) coordinates. The solutions are additionally marked with colors indicating the mass flow rate of cooling steam obtained for each configuration. It has to be stresses that the steam usage was not optimized and its magnitude is only informative. We can see that the front in almost whole temperature range is fairly flat. Only when the lowest possible temperature is achieved a very steep increase of thermal gradient is observed with negligibly small decrease (a fraction of a degree) of maximum temperature. This is due to small distance between the exposed to hot steam external profile and the cooled passage wall.

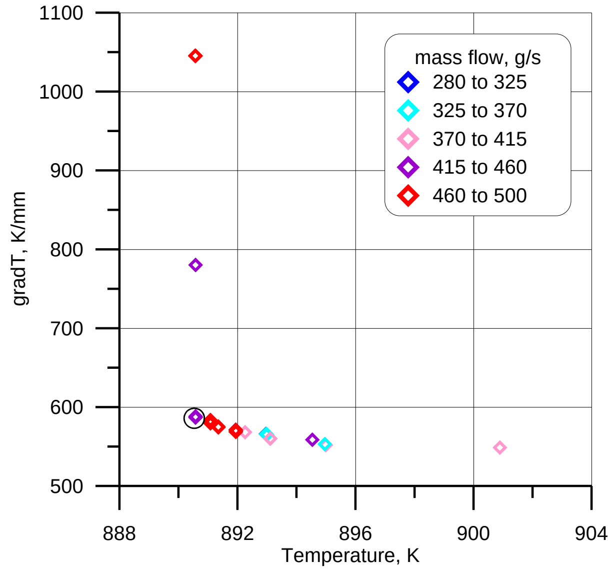

Fig.8 Pareto optimal solutions in the objective space

As we can see from the Pareto front the maximum cooling gain in terms of temperature reduction comparing to the bulk temperature is rather small and does not exceed 33K for the cooling conditions specified. One of the p-optimal cooling configurations depicted in Fig. 8 is shown in Fig.9.

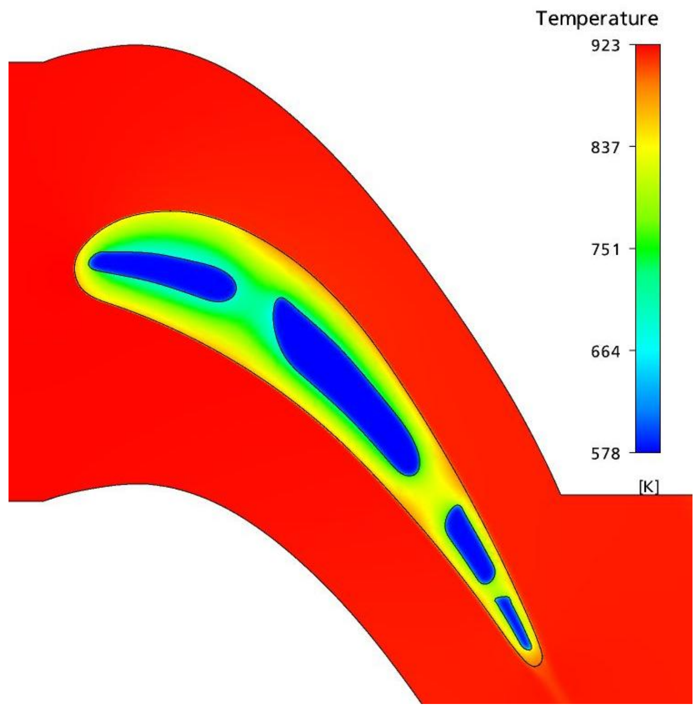

Fig.9 Temperature distribution for the optimized cooling configuration.

This cooling system consists of four elongated passages evenly distributed along the blade. This is due to very small drop of temperature within the external flow in the axial direction. Such elongated channels provide also an increased area for heat transfer and from this point of view are preferable. The highest temperature appears at the trailing edge of the blade and further cooling is not possible because of the minimum thickness constraint of the solid.

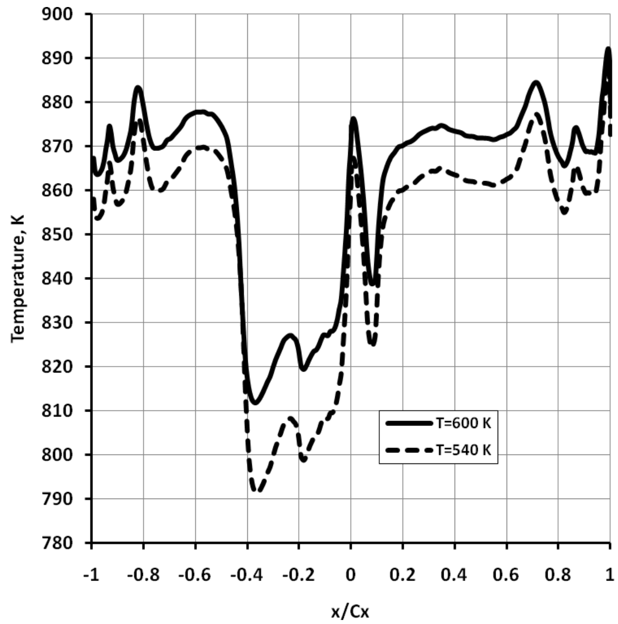

Fig.10 Temperature distribution on the profile for the optimized cooling configuration for two cooling steam temperatures 600K and 540K .

The problem of cooling the steam vane is relatively hard due to the very high temperature of the life steam around the blade profile. The most difficult part for reducing the temperature is the trailing edge, where a very small space for an introduction of cooling passage occurs. Fig. 10 shows how small is the influence of a drop of the cooling steam temperature on the blade wall temperature. Temperature reduction by 60K causes decrease of blade wall temperature in the front part of the blade about 12K and in the trailing edge region about 7K only.

The optimization problem of a cooled blade with use of the CHT analysis is very expensive computationally. Prediction of the thermal field for a single cooling candidate requires about 2 minutes if a parallel processing with 8 CPUs was applied. That makes the whole process to run for more than 2 months.

CONCLUSIONS

The paper presents the application of Conjugate Heat Transfer prediction for optimization of a cooling system configuration. The search procedure utilized the evolutionary algorithm. The evaluation of a particular solution was done on the basis of CHT analyses within a cooled airfoil. For the purpose of this research a convectively cooled airfoil was assumed and the optimization consisted in the relocation and diameter changes of the cooling passages. The search procedure was run as a Multi Objective Evolutionary Algorithm with use of the Pareto approach.

As a result of the optimization problem a set of cooling configurations was obtained which form a front of non-dominated solutions within the objective space. On this basis the final configuration can be selected with some further premises.

The optimization showed that the effect of cooling of steam turbine blades, especially those for supercritical steam parameters seemed to be a complex problem. Due to a very high intensity of heat transfer on the blade surface maximum temperature reduction within the solid is not satisfactory. It seems that with the shape optimization only a required temperature drop cannot be achieved and some more search for different coolant thermodynamic parameters would be necessary.

Applied optimization strategy is said to be one of the best tools for the global optimum search especially in the case of problems with complex geometries and a large number of design variables.

The main improvement worked out in the area of airfoil cooling optimization was the inclusion of CHT analysis for the blade interior, instead of empirical formulae for heat transfer conditions, which were widely used in such problems. This project showed the direction which should be followed to obtain more reliable designs.

One of the most painful disadvantages of the evolutionary algorithm together with the CHT analysis is their high computational cost, so a parallel computing is required while using this approach.

The next step in the development of a cooling system optimization should be the application of the full CHT prediction, which would make the whole process more reliable. Also in the future work the flow should be considered more precisely: including more layers in the spanwise direction and hub and tip boundary layer, using finer grids in the holes, analyzing the different turbulence models, calibrating a transition model. Nowadays it is a task for the single case CHT analysis and therefore extension of the optimization procedure could be done stepwise, depending on the computer power.

Acknowledgements

The results presented in this paper were obtained from research work co-financed by the Polish National Centre of Research and Development in the framework of Contract SP/E/1/67484/10 – Strategic Research Programme – Advanced technologies for obtaining energy: Development of a technology for highly efficient zero-emission coal-fired power units integrated with CO2 capture.

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