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INSTYTUT MASZYN PRZEPŁYWOWYCH

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

TRANSACTIONS
OF THE INSTITUTE OF FLUID-FLOW MACHINERY

70-72

W A R S Z A W A – P O Z N A Ń 1976
P A Ń S T W O W E   W Y D A W N I C T W O   N A U K O W E

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

poświęcone są publikacjom naukowym z zakresu teorii i badań doświadczalnych w dziedzinie mechaniki i termodynamiki przepływów, ze szczególnym uwzględnieniem problematyki maszyn przepływowych

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THE TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machinery

REDAKCJA – EDITORIAL OFFICE
Instytut Maszyn Przepływowych PAN
ul. Gen. Józefa Fiszera 14, 80-952 Gdańsk, skr. pocztowa 621, tel. 41-12-71

Printed in Poland

PAŃSTWOWE WYDAWNICTWO NAUKOWE – ODDZIAŁ W POZNANIU

Wydanie I. Nakład 630+90 egz. Ark. wyd. 63,25 Ark. druk. 49,25 Papier druk. sat. kl. V. 65 g., 70×100. Oddano do składania 4 VII 1975 r. Podpisano do druku w czerwcu 1976 r. Druk ukończono w czerwcu 1976 r. Zam. 528/118. H-16/245.
Cena zł 190,–

DRUKARNIA UNIWERSYTETU IM. ADAMA MICKIEWICZA W POZNANIU

III KONFERENCJA NAUKOWA

na temat

TURBINY PAROWE WIELKIEJ MOCY

Gdańsk, 24 - 27 września 1974 r.

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IIIrd SCIENTIFIC CONFERENCE

on

STEAM TURBINES OF GREAT OUTPUT

Gdańsk, September 24 - 27, 1974

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III НАУЧНАЯ КОНФЕРЕНЦИЯ

на тему

ПАРОВЫЕ ТУРБИНЫ БОЛЬШОЙ МОЩНОСТИ

Гданьск, 24 - 27 сентября 1974 г.

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

1976 Zeszyt 70 - 72

JAROSŁAW MIKIELEWICZ, ANDRZEJ KONORSKI, MARIAN TRELA, EUGENIUSZ IHNATOWICZ, STANISŁAW GUMKOWSKI, MARIUSZ ADELT, JERZY ZEMBIK

Poland*

Heat Transfer Problems of Hollow Guide Vanes Heating in Steam Turbines

List of symbols

d – diameter,
$\varphi(\theta_0)$, $\phi(\theta_0)$ – functions of the contact angle
G_f – mass flow rate of the film,
h – film thickness,
h_0 – film thickness without evaporation,
h^+ – dimensionless thickness of the broken film,
t – temperature,
Z – co-ordinate,
α – mean heat transfer coefficient,
β – vane angle (measured from horizontal),
θ_0 – angle of contact,
μ – dynamic viscosity,
ρ – density,
σ – surface tension,
τ – shear stress on the film interface.

Subscripts

e – experimental values,
0 – outer surface of the blade,
s – saturation,
t – theoretical values,
w – value of the wall,
$'$ – denotes properties of the liquid.

1. Introduction

The method applied for drying steam in the final stages of the steam turbine by means of inner heating is based on an entire or partial evaporation of the fluid phase from the guide vane surfaces. The liquid phase upon the vane may appear in a form of the continuous film or non-continuous one formed by drops or rivulets produced by the separation of large drops which have a greater dynamic inertia than the flow of steam. The inner heating is based on the application of guide vanes, referred to as hollow vanes [1]. The heating steam taken from intra-stage area preceding the heated stage or from one of the former stages of the turbine is supplied into the hollow vane.

The steam condensing inside the heating spaces heats and evaporates the liquid phase upon the vane surface.

This paper presents the results of the investigations of this subject which have been carried out at the Institute of Fluid-Flow Machinery, Polish Academy of Sciences in Gdańsk.

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* Institute of Fluid-Flow Machinery, Polish Academy of Sciences, Gdańsk.

2. Survey of heat transfer studies

The process of the heat transfer in a hollow vane Fig. 1 is composed by the heat transfer due to the heating steam condensation, the heat conduction through the vane wall and the heat transfer caused by the evaporation of the liquid phase upon the outer surface of the vane.

The complex heat transfer problem (the interconnection of condensation with evaporation occurs through the metalic wall of the vane) is a three-dimensional one. In order to

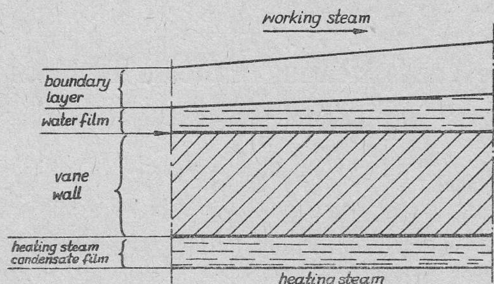

Fig. 1. Cross-section of the blade wall element

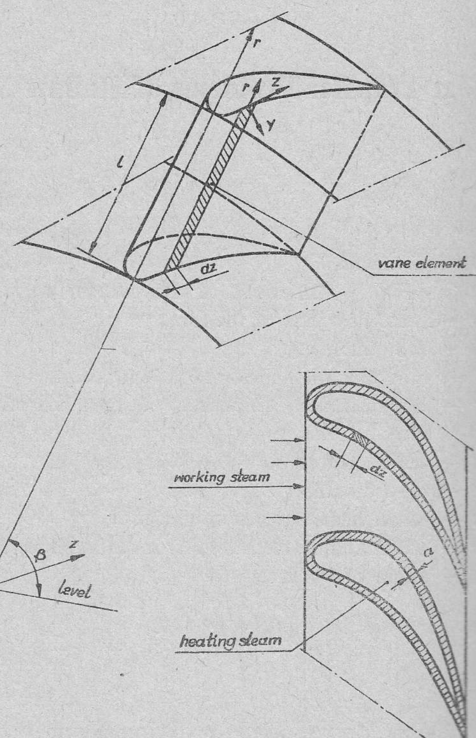

Fig. 2. Scheme of the blade-ring with heating blades

simplify the considerations the problem has been treated as a two-dimensional one, neglecting the variability along the vane axis. It has been assumed that the cross-section of the vane remains constant along the vane length.

2.1. Models of heat and mass transfer processes

In order to estimate the effectiveness of the system and its influence upon the performance of a turbine, a method of analyzing the heat and mass transfer processes has been developed and the influence of these processes on the evaporation of the liquid film and on the changes of the droplet structure in the wet steam has been investigated [2, 3].

To avoid complications which could make the computation method of no practical value, a number of simplifications has been assumed; a simplified sketch of a hollow vane has been shown in Fig. 2. The main principles of the model under consideration can be seen from Fig. 1.

The condensation of a heating steam inside the hollow guide vanes is being considered under the assumptions of Nusselt's condensation model. The obtained heat flux value is being averaged with regard to the vane height and with regard to the slope angle of the vane. Standing upon these assumptions the formulae for a heat flux density transferred from the heating steam to the inner surface of the wall have been derived.

The phenomenon of heat conduction along the circumference of the profile has been taken into account when considering the heat transfer across the wall. This phenomenon creates an additional perturbation of the process in question.

The steam flowing round a profile contains two kinds of the water drops:

a) very small drops which have been created spontaneously and which affects significantly the thermodynamic state of the medium,

b) large drops originating in the preceding stages of the turbine which gather on the surface of the profile (mainly on its concave part) creating a liquid film. This fluid flows along the vane surface in the direction of the outlet edge mainly under the influence of the shear stresses exerted by the working steam.

Within this water layer we deal with three phenomena: 1) heat conduction in a perpendicular direction to the surface, 2) the convection in the direction of the flow, and 3) the evaporation on the interface.

For the flow of the working fluid the balance equations which take into account the transport phenomena for various forms of the liquid phase will be valid. Additional equations are formed by equations of the energy and momentum transfer in a boundary layer. The laminar and turbulent boundary layers have been considered separately. As a result

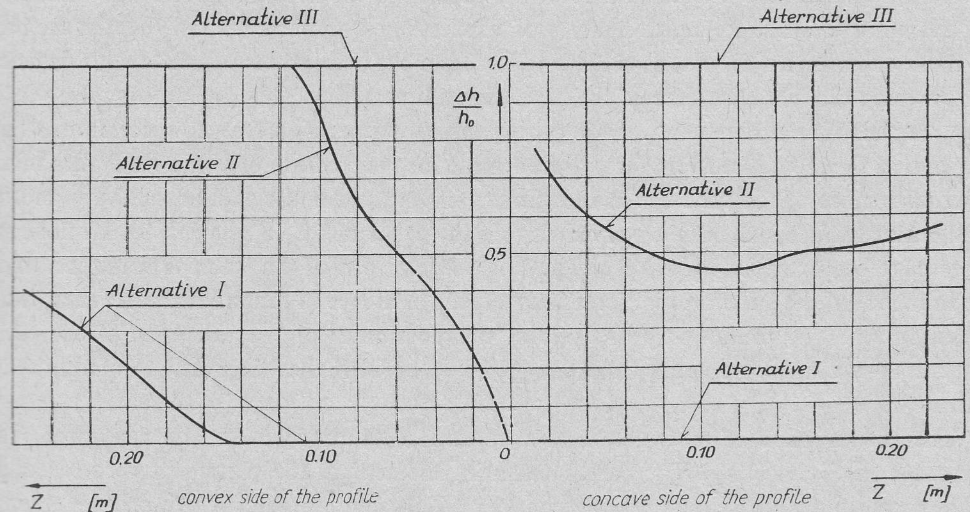

Fig. 3. Relative change of the water film thickness on the heating blade surface (50 MW turbine, stage 22)

Alternative I – stage 22 heated by steam tapped before stage 22
Alternative II – stage 22 heated by steam tapped before stage 21
Alternative III – stage 22 heated by steam tapped before stage 20

we obtain a complicated set of differential equations describing all phenomena regarding the heating vane. This of set equations may be used in two versions: for the external vane surface covered with water film and for a dry external vane surface. In each of these cases a method of solving numerically the appropriate sets of equations has been developed.

A simplified method of the calculations permitting estimation of the expected results of the heating has also been developed.

Using the above mentioned method it was possible to analyze [2] the transport processes and film evaporation on the guide vanes. The results of one of these computations are shown in Fig. 3 where the variations of the thickness of the film (assumed as formed by droplets which had been separated out of the steam flow) due to the heating of guide-vanes at various heating steam conditions are presented.

Another simplified model of the heat transport, based on the assumption that the heat transport at the outer of the profile is determined mainly by the resistance of the laminar film, has been described in [4]. Using the same assumption a method of determining the two-dimensional temperature field in the wall of the vane has been proposed.

2.2. Studies on hydrodynamics and evaporation of liquid phase moved by the shear stresses on interface

The process of fluid film evaporation from a heated surface is composed by a number of phenomena. One of them concerns the hydrodynamics of the film flow. In order to obtain some information on this problem some preliminary qualitative experimental investigations of the water film flow in the gravitational field have been carried out [5].

The experiments have been performed on a vertical copper tube of 35 mm diameter (d=35 mm) and 1500 mm length. The water film was produced by means of a special device for wetting the outside surface of the tube. The experiments have been carried out for a laminar film, the Reynolds number of which ranged between 8.6 and 784. On the basis of these experiments it was possible to determine the minimum unit flow rate sufficient for covering the whole surface of the tube with a liquid film. This rate amounted to $2.15 \cdot 10^{-3}$ kg/ms. The observations made during the experiments in free flow conditions have shown that the most important factor responsible for the creation of a thin water film on the external surface of the tube lies in the surface smoothness and cleanliness. A wave motion of the film surface has been observed during the experiment. In relation to the flow rate there have been various kinds of waves. The observations of the water film motion under the influence of the gravitational forces suggest the existence of longitudinal as well as transversal waves. The preliminary analysis of the possibility of the existence of the stable transversal waves is presented in paper [6] together with the observations of thin water film flows.

The investigation of the water film evaporation was carried out on a research plant presented in Fig. 4 [7, 8]. The basic measuring element of the plant is a vertical copper tube placed in a glass steam-carrying channel of 104 mm diameter which enables to carry out the optical observations of the phenomena taking place during the water film flow on the external surface of the tube.

The steam flow through the ring channel, consisting of a heated tube (over which the film was flowing) and a surrounding transparent one, is directed downwards. The influence

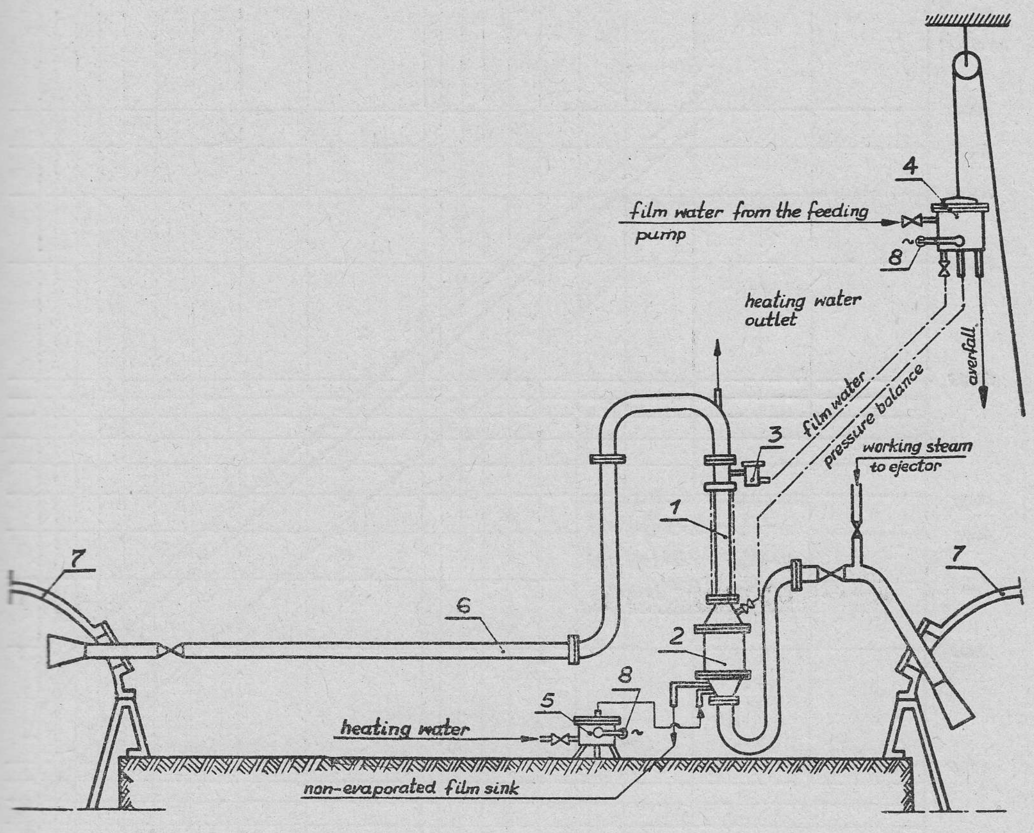

Fig. 4. Scheme of the experimental installation for the investigation of the water film evaporation
1 – film evaporating pipe, *2* – measurement tank for non-evaporated film, *3* – film water temperature measurement tank, *4* – film water heater, *5* – heating water heater, *6* – steam pipeline, *7* – turbines, *8* – electric heaters

of the steam flow was the main factor of the film motion. The water film flow rate was controlled in order to obtain the thinnest possible fluid layer entirely covering the test section. The investigations were performed in the following ranges of parameters:

- saturation pressure $(0.04 \div 0{,}09)$ kG/cm^2,
- steam flow velocity $(118 \div 161)$ m/s,
- film flow-rate $(291 \div 540)$ kg/mh,
- heat flux density $(799 \div 20\,750)$ kcal/m^{2}h.

The experiments regarding the film evaporation in above ranges of the parameters values have proved that the coefficient of the heat transfer for the evaporated film may be determined with a fair approximation on the basis of the heat conductivity of the liquid layer. This fact indicates that under the conditions at which the experiments were carried out the influence of the waves upon the film surface, and also the influence of the mass transfer on the interface, were negligible.

The experimentally determined values of the heat transfer coefficient of the liquid film plotted against the temperature difference $\Delta t_f = t_{w0} - t_s$ are shown in Fig. 5 [9].

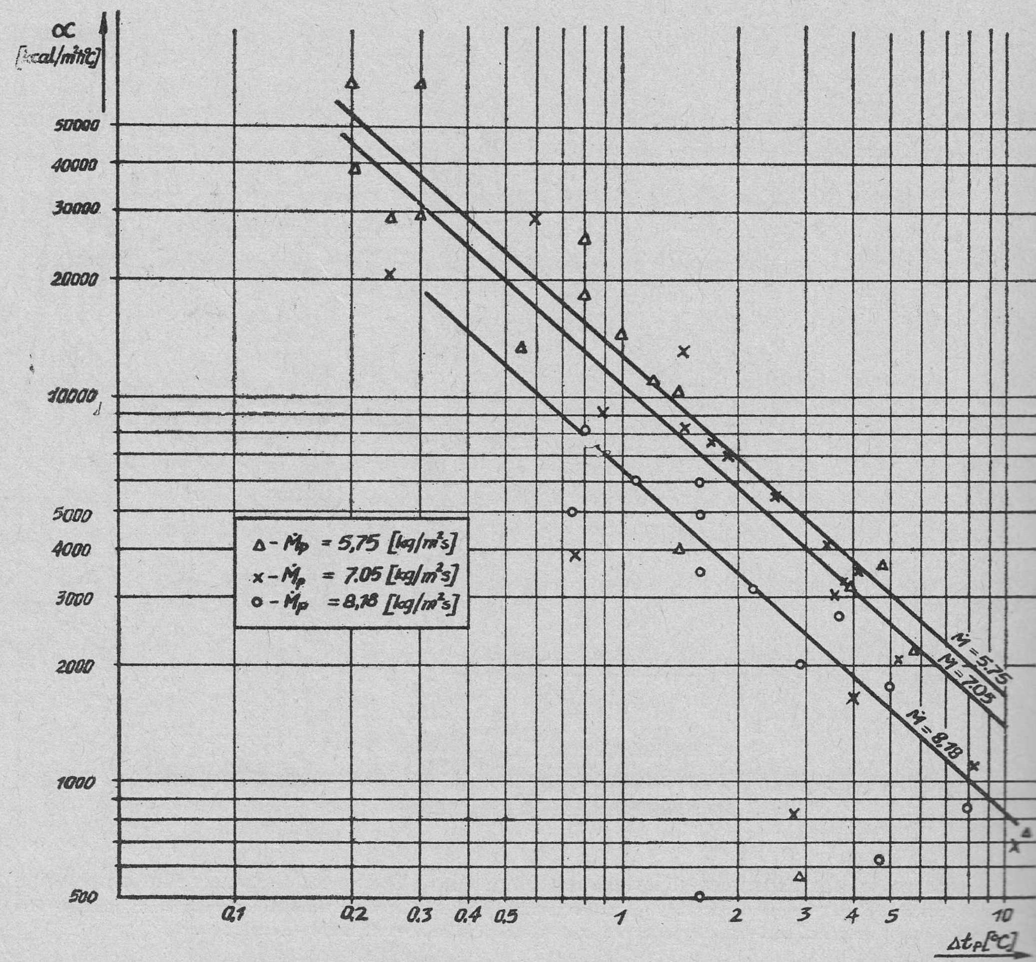

Fig. 5. Experimental heat transfer coefficient versus temperature difference for various steam mass flow rates

An essential problem which arose during the investigation consisted in the break down of the liquid film into separate rivulets. This problem has been already discussed in [10] with regard to steam turbines guide vanes.

The problem of the shear driven liquid film stability on the surface under the influence of shear stresses constituted the subject of our theoretical and experimental investigations. The analysis of the interface stability based on the theory of small perturbations has been presented in [4]. This kind of analysis gives however approximate results only with regard to the conditions of the film breaking, as it neglects an important parameter, viz: the contact angle responsible for the stream formation. An improved analysis of the liquid film stability which takes into account the contact angle influence is presented in [11].

In [11] a new parameter – a ratio of the wetted area to the total one has been introduced. The energy of the post-break down liquid rivulets proved to depend upon this parameter. This fact can be described by an appropriate set of equations resulting from:

– the principle of energy conservation of the film before break down and of the rivulets after the film is broken,

– the principle of rivulet stability after the film is broken, consisting in the exigence that the energy of the rivulet flow be minimum,

– the principle of mass conservation before and after breaking.

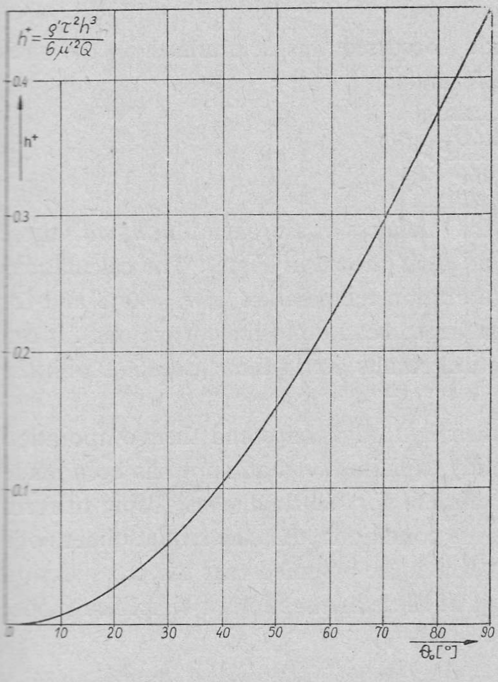

Fig. 6. Dimensionless thickness of the broken film versus the angle of contact

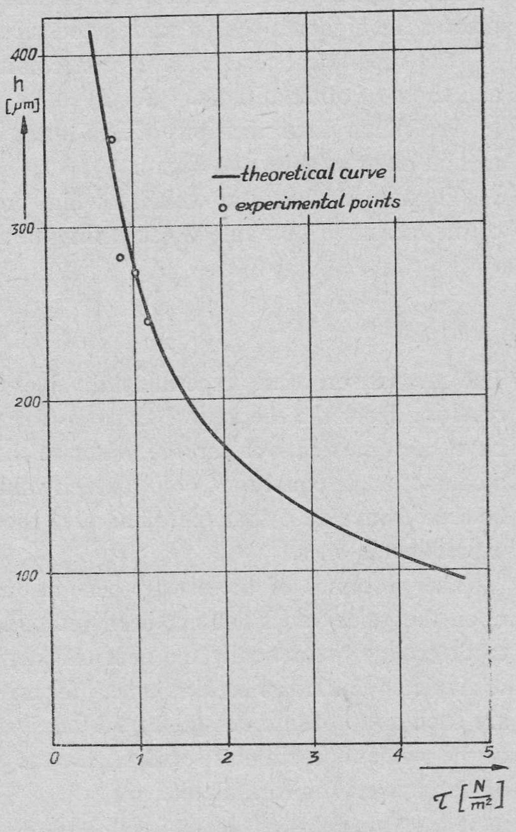

Fig. 7. Broken film thickness versus shear stress on the film interface

During the analysis the both the kinetic and the surface energy were considered. It can be easily proved that the cross-section of the shear driven liquid rivulets is similar in shape to a sector of the cylinder. Taking this into account the following equation for determining the dimensionless thickness of a breaking film has been obtained (Fig. 6):

$$h^+ = 3\cdot 2^{\frac{2}{3}}\left(\frac{\theta_0}{\sin\theta_0}-\cos\theta_0\right)^{\frac{1}{3}} h^{+\frac{2}{3}} q(\theta_0)^{\frac{2}{3}}\,\frac{\sin\theta_0}{\phi(\theta_0)}-(1-\cos\theta_0), \tag{1}$$

where

$$h^+(\theta_0)=\frac{\rho'\tau^2h^3}{6\mu'^2\,\sigma}\,. \tag{2}$$

In the course of the theoretical investigation of the film stability the conditions under which the continuous form of liquid phase changes into non-continuous rivulets have been defined.

It results from the analysis that the film breakdown depends on the magnitude of the shear stresses on the interface, on the saturation pressure and on the contact angle. The contact angle depends in turn on the kind of liquid, on the material of the wall as well as on the quality of its surface. The probability of thin film existence on the surface is the greater the higher is the saturation pressure and the larger shear stresses on the interface.

The theoretical results have been confirmed by our own experimental investigations and those of other authors.

Preliminary investigations of film break down have also been carried out in the experimental plant as shown in Fig. 4.

The film thickness at which the film breaking occurred was determined on the basis of the measured flow rate G_f according to the formula:

$$h_e = \sqrt{\frac{2\mu' G_f}{\rho' r}}. \tag{3}$$

The comparison of the experimental values of the thickness of a broken film h_e with those resulting from the theoretical analysis (formula (2)) is shown in Fig. 7. The calculations have been carried out for the water at the saturation temperature of $t_s = 40°C$ and for a contact angle equal to $\theta_0 = 40°$ (which value seems to be close to the correct one). It can be seen from Fig. 6 that the qualitative theoretical results are in fair agreement with the experimental ones.

The processes of the rivulet generation, their hydrodynamics and their evaporation upon the guide vanes has not been yet sufficiently explained. An attempt has been made to determine theoretically the heat transfer coefficient for the fluid phase being in form of rivulets [12]. It has been assumed, for any given conditions, that the rivulet dimensions are such as to ensure its stability. Moreover it had to be assumed that the cross-section of the rivulet is a circular segment. On the basis of these assumptions a simplified model of the rivulet was formulated.

2.3. The problem of an inner condensation in a hollow vane

The water steam condensation in a laminar condensate flow over a non-circular cylindrical surface is analysed in [13] from the viewpoint of the inner heating problems. The flow trajectories of the condensate upon an arbitrary non-circular cylinder and the local heat transfer coefficients resulting from the local thickness of the condensate have been calculated using the known Nusselt assumptions.

In order to determine the condensate trajectories the planar development of the non-circular cylinder surface has been examined. Since the motion of the condensate upon this plane is due to the influence of the variable gravitation component, the streamline of the condensate is curved. The computation of the thickness of the condensate layer allowed to determine the trajectories of the condensate. Further on, taking into account the influence of the wave motion of the condensate [14], the dependence of the mean value of the heat transfer coefficient on the condensate trajectory has been obtained.

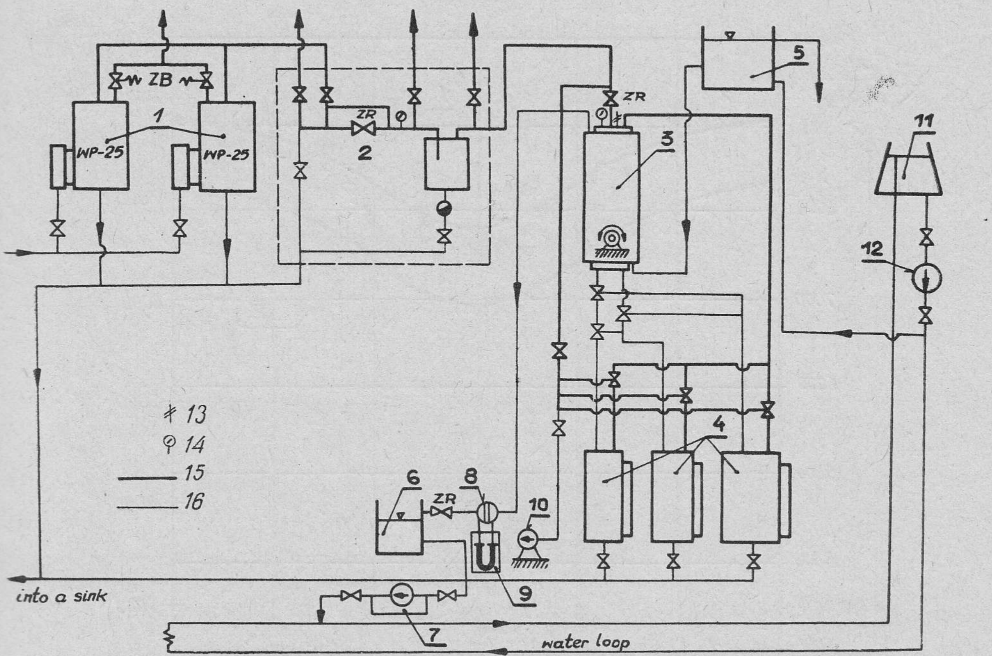

Fig. 8. Scheme of the experimental installation for steam condensation in the hollow vane
1 – steam generation units, *2* – control station, *3* – test section, *4* – measurement tanks, *5* – upper water tank, *6* – bottom water tank, *7* – water pump, *8* – flow-meter, *9* – differential pressure gauges, *10* – vacuum pump, *11* – water cooler, *12* – circulating pump, *13* – temperature measurement, *14* – pressure measurement, *15* – steam pipeline, *16* – water pipeline, *ZR* – control valve, *ZB* – safety valve

On the basis of this dependence numerical calculations of the heat transfer coefficient for a non-circular cylindrical surface of the hollow vane have been carried out. For the same purpose the experimental investigations have also been performed. The investigations have been carried out at pressure $p_s = 0.8$ bar for various blade angle values β, lying within the range of $\beta = 15° \div 165°$. The more detailed data concerning the experiments and the method of determining the heat transfer coefficients can be found the reference [13]. The scheme of the test stand is shown in Fig. 8. The comparison of the computed values of the heat transfer coefficient with the experimental data is shown in Fig. 9, indicating a very good agreement between them.

It has been indicated that the wave motion of the film influences the heat transfer coefficient values. This effect may be taken into account by introducing a correction factor in a way similar to that used for analyzing the condensation on vertical surfaces. The validity of thus derived equations have been proved for the forms considered geometrical here.

2.4. Investigations of the heat conduction in the blade shell

The temperature field in the blade wall is a three-dimensional one. This is due to the variable working steam temperature around the circumference and along the vane length as well as due to the variability of the coefficients of heat transfer inside and outside the

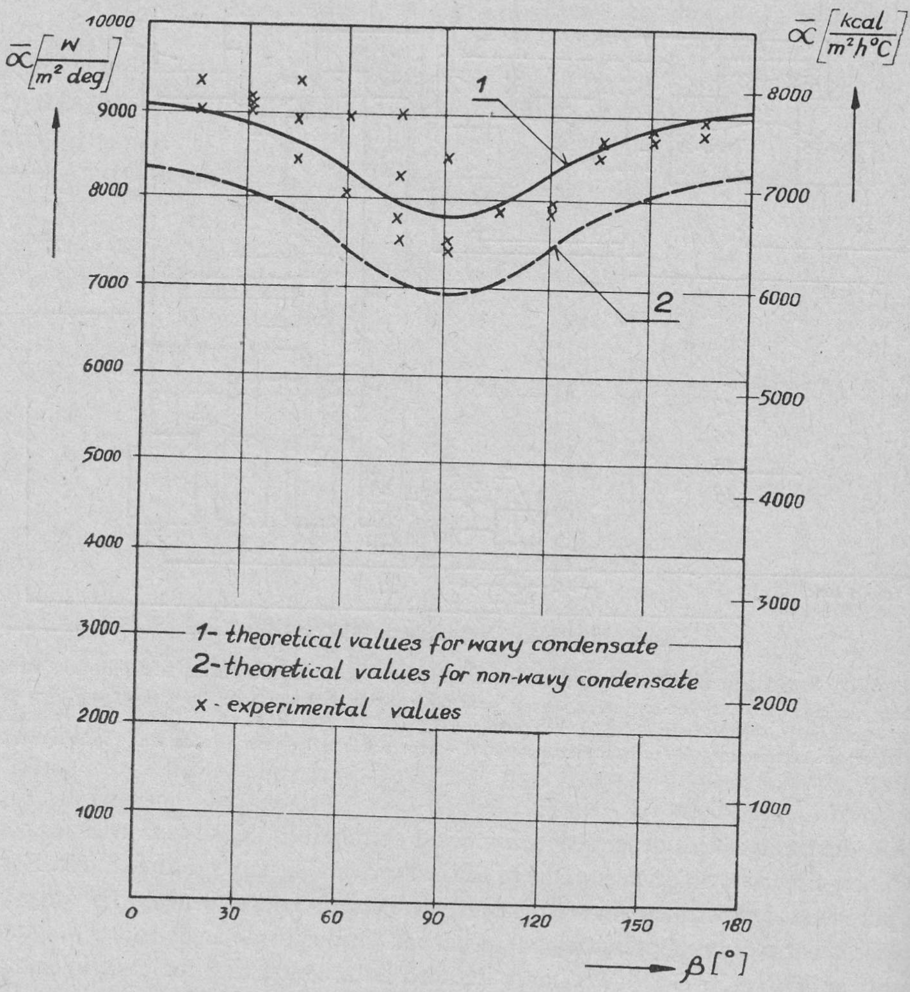

Fig. 9. Theoretical and experimental condensation heat transfer coefficients for a hollow vane

vane shell. To simplify the problem the temperature has been averaged along the vane length thus making possible a two dimensional analysis. The boundary conditions are of III type, while the heat transfer coefficients are the power functions of temperature differences. This problem has been solved using analog and numerical methods. An electrical analogy has been used to perform an experiment on a continuous model made of a conducting paper [15]. For the application of the analog method it is necessary to make step-by-step approximations when modelling the boundary conditions (heat transfer coefficients) since they depend on the temperature difference, which itself is the object of the investigation.

On the basis of the analogous studies it has been proved that the heat conduction in the vane circumference direction considerably levels the distribution of the temperature in a shell and therefore should not be neglected.

The applied numerical method was based on averaging the temperature across the thickness of the wall. This method was discussed in [16]. The method is relatively simple and fast convergent which facilitates carrying out the calculations on a digital computer. The appropriate dependence of the heat transfer coefficients on the temperatures of respective surface may be obtained regardless of the temperature averaging by introducing an additional "function of unevenness" on account of which both surfaces of the wall will show the different temperatures.

The problem which remains for the further consideration consists in connecting the description of the thin-walled shell (where the method presented above may be used) with that of the "tail" part of the blade, which may be regarded as a rib.

3. Thermodynamic analysis of the heating effects

Using the method described in [2] an analysis of the thermodynamic effects of steam heating for several types of condensing turbines has been carried out assuming in each type of the turbine the various alternatives of tapping the heating steam (thus various values of the thermal parameters of the heating steam).

It has been shown that guide vane heating leads to most essential results (amounting to the evaporation of the whole film or a considerable part of it) if the heating steam is tapped from one of the former stages where higher steam temperatures prevail. On the contrary, if heating steam is tapped from the stage immediately preceding the heated quide vane, the evaporation is only slight (and occurs on the convex side of the profile only) although this layout is very favourable from the technological point of view. Such outcome is due to both the character of the working steam flow around the profile and to the typical development of the phenomena of the large droplets separation.

The influence of guide vane heating on the turbine performance and on the energy balance of the unit has been considered in [17, 18, 19, 20]. The energy effects consist in a decrease of the thermal efficiency of the cycle (because of tapping the heating steam from the turbine stages and because of the heat supply to the working steam) and, in a certain increase of the internal efficiency due to the increase of working steam dryness.

The positive or negative sign of the resulting change of everall efficiency will depend upon the choice of the thermal cycle including the choice of points from where heating steam is to be tapped (hence the thermal parameters of the heating steam) and the choice of the guide vanes which are meant to be heated. The absolute value of the changes of the turbine output will depend on the heating intensity, i.e., on the heating steam fluxes and will be limited by the heat transfer in the guide vanes.

The most essential components which are responsible for the overall change of the turbine output power are:

– power gains in stages, following the heated one, due to the increase of working steam enthalpy and its dryness (this includes, among others, the effect of internal efficiency increase),

– the power loss caused by the heating steam tapping (occuring in stages following the point from which heating steam is being tapped),

– the change of the output power caused by changing the tapped steam flow-rate (relevant for turbines with regeneration system),

– the change of the power needed for driving the cooling water pump.

Calculation of the energy balance in the several types of the turbines have been carried out in [17, 18, 21].

An essential effect of the steam heating consists in the decrease of the size of the damaging droplets in the working steam. It leads to economic profits due to the prolongation of the life-time periods of LP blading systems and preserving their internal efficiency from the decrease caused by the erosion damage of their surfaces.

From the viewpoint of the energy balance the more advantageous are such layouts in which the temperature differences between heating and working steam are as small as possible. In this case, however, the decrease of the content of the damaging moisture is small. In the reverse case i.e. when the intensity of the heat transfer increases, the drying of the working steam becomes more effective; hence the lifetime of the LP turbine blading increases but the certain energy loss has to be expected. An essential indication for choosing the layout of the heating system consists in avoiding too complicated constructions.

4. Conclusions

In the above mentioned reference papers the fundamental phenomena connected with the hydrodynamics of working and heating steam flow as well as with the problems of heat transfer occurring in inner heating systems has been explained. Among them the heating steam condensation inside the hollow vane, the stability and evaporation of the shear driven water films moving at high velocities of the steam phase and some problems concerning the heat conduction in a vane have been considered.

The theoretical models of the transfer processes and the methods of computation of energy-balance processes have also been described.

The calculations of the effects of the liquid film evaporation from the guide vane surface and of the changes of the energy balance caused by the steam heating have been carried out for two types of turbines: a 50 MW unit and a 200 MW reheat turbine. It has been shown that the application of the guide vane heating leads to alterations of the energy balance which can be reduced to the negligible amounts, thus ensuring the possibility of decreasing the content of damaging moisture in the working steam and therefore increasing the life-time of the blading.

From the authors' point of view it appears necessary to carry out both the theoretical and experimental investigations of the above problems. These investigations should be followed by a full-scale experiment on a turbine stage in which the guide vane heating is applied.

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Problemy wymiany ciepła w podgrzewanych kierowniczych łopatkach powłokowych turbin parowych

Streszczenie

Uszkodzenia erozyjne niskoprężnych stopni turbin kondensacyjnych, przez które przepływa para o dużej wilgotności, są spowodowane obecnością stosunkowo dużych kropel, pochodzących z filmu wodnego powstającego na powierzchni łopatek kierowniczych. Aby ograniczyć formowanie się filmu

wodnego na powierzchni łopatek i tym samym zmniejszyć ilość dużych kropel niesionych przez parę, zaproponowano metodę polegającą na ogrzewaniu parą od wewnątrz łopatek kierowniczych. Para grzewcza kondensuje się wewnątrz łopatki oddając ciepło parowania poprzez ściankę filmowi powodując jego częściowe lub całkowite odparowanie.

W artykule krótko omówiono prace badawcze poświęcone tym problemom, wykonane w Instytucie Maszyn Przepływowych PAN w Gdańsku.

Zbudowano ogólny model sprzężonych procesów wymiany ciepła i masy, uwzględniający główne procesy składowe: kondensację pary grzewczej wewnątrz łopatki, przewodzenie ciepła przez ściany łopatki oraz przepływ pary w kanale łopatkowym z oddziaływaniem na powierzchnię filmu. Na podstawie tego modelu opracowano odpowiednią metodę obliczeń.

Wykonano badania eksperymentalne w celu wykrycia najważniejszych czynników wpływających na wymianę ciepła i masy. Zbadano kondensację pary wewnątrz łopatki powłokowej przy różnych kątach nachylenia względem poziomu. Na modelu analogowym przeprowadzono badania dwuwymiarowego pola temperatur w ściankach łopatki. Przeprowadzono także eksperymentalne i teoretyczne badania stabilności filmu wodnego na zewnętrznej powierzchni łopatki. Ustalono teoretycznie wartości krytyczne związane z przepływem filmu.

Przeanalizowano wpływ podgrzewu wewnętrznego na obieg termodynamiczny i sprawność turbiny oraz przeprowadzono obliczenia sprawności dwóch typów turbin kondensacyjnych wyposażonych w podgrzewane łopatki. Wyniki obliczeń wykazały, że podgrzew wewnętrzny, prowadzący do istotnego zmniejszenia szkodliwej zawartości wody w parze roboczej, jest termodynamicznie i ekonomicznie uzasadniony.

Проблемы теплообмена в подогреваемых оболочковых направляющих лопатках паровых турбин

Резюме

Эрозионные повреждения ступеней НД конденсационных турбин, через которые протекает сильно увлажненный пар, происходят за счет присутствия соответственно больших капель, возникающих из водяной пленки, образующейся на поверхности направляющих лопаток. Чтобы ограничить формирование водяной пленки на поверхности лопаток и тем самым уменьшить количество больших капель, транспортированных паром, предлагается метод, основанный на внутреннем подогреве направляющих лопаток паром.

Греющий пар конденсируется внутри лопатки, отдавая тепло испарения через стенку пленке, которая таким образом частично или полностью испаряется.

В статье дается короткое обсуждение исследовательских работ в этой области, выполненных в Институте проточных машин ПАН в г. Гданьске.

Построена общая модель сопряженных процессов тепломассообмена, учитывающая основные составляющие процессы: конденсацию греющего пара внутри лопатки, теплопроводность через стенки лопатки, а также течение пара в лопаточном канале с воздействием на поверхность пленки. На основе этой модели разработан соответствующий расчетный метод.

Проведены экспериментальные исследования с целью обнаружения главных факторов, влияющих на процесс тепломассообмена и изучения процесса конденсации пара внутри оболочковой лопатки при различных углах наклона относительно горизонтального положения. На аналоговой модели изучалось двухмерное поле температур в стенке лопатки. Проведены также теоретические и экспериментальные исследования устойчивости водяной пленки на внешней поверхности лопатки. Теоретически устанавливаются критические значения, связанные с течением пленки.

Анализируется влияние внутреннего подогрева на термодинамический цикл и к.п.д. турбины и выполняются расчеты к.п.д. двух типов конденсационных турбин, оснащенных подогреваемыми лопатками. Из результатов расчетов следует, что внутренний подогрев, способствующий значительному уменьшению количества вредной влаги в водяном паре, является обоснованным как термодинамически так и экономически.