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III НАУЧНАЯ КОНФЕРЕНЦИЯ

на тему

ПАРОВЫЕ ТУРБИНЫ БОЛЬШОЙ МОЩНОСТИ

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Some Problems of a Flow Through the Low-Pressure Section of a Large-Output Steam Turbine

The low-pressure section is, without doubt, the most complicated section of a steam turbine of large output, both in respect of the flow of steam and also from some other angles of view. The flow cross-sectional area of the last stage imposes considerable limita-

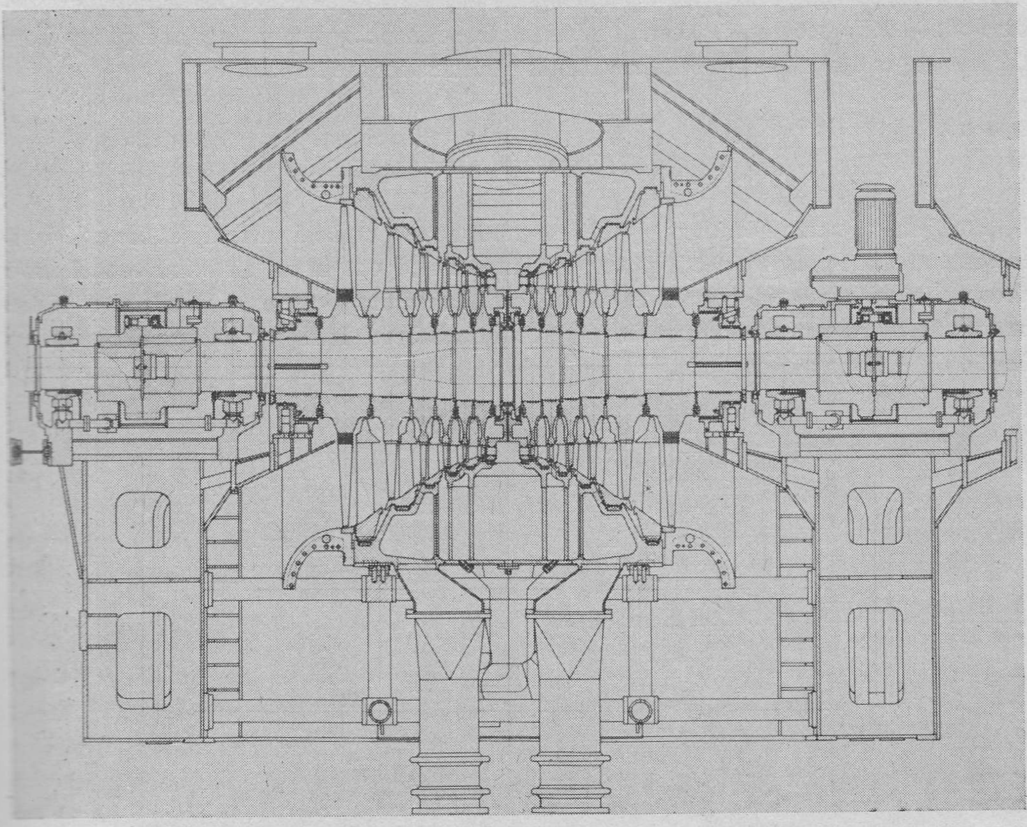

Fig. 1. Sectional view of a low-pressure section of a 500 MW turbine

* ŠKODA National Corporation, Plzeň.

tions on the output obtainable by the low-pressure section and, consequently, by the whole turbine. Arising and movement of the liquid phase in the blading bring about the erosion to the rotating blades of the last stage, which can reduce the service life of the engine. It is likewise desirable that the low-pressure section should show a high efficiency in the region of outputs applied for long times, especially in case of saturated-steam turbines which work with a less enthalpy drop.

The requirements for operational flexibility of large-output steam turbines, for their work within a wider region of outputs, and for frequent shutting-down and starting up are ever increasing. The flow conditions in the low-pressure section under working regimes departing from the design can thus give rise to high dynamic and temperature stresses in some highly-stressed portions, and cases of heavy erosion can also occur.

In the following text attention is given especially to the scope of problems relating to a three-dimensional flow of the steam phase, wherever possible under all working conditions of the turbine, from an idle run to the full output. Some secondary effects caused by the flow under working regimes departing from the design are considered too.

The experiments presented in the following relate mostly to the low-pressure section with last stage having blades of 840 mm length. For the arrangement of this section for a turbine of 500 MW see Fig. 1. Steam is admitted from above by two pipings and passes from the last stages down to a condenser, which is placed under the turbine.

1. Cross-over pipes

Problems of steam flow through the low-pressure section begin already at the exit from the intermediate-pressure section. Experiments carried out in an air wind tunnel have shown that with an axial admission of the flowing medium to the exhaust chamber of the intermediate-pressure section the flowing medium is brought into rotation which is then kept on in the cross-over pipes.

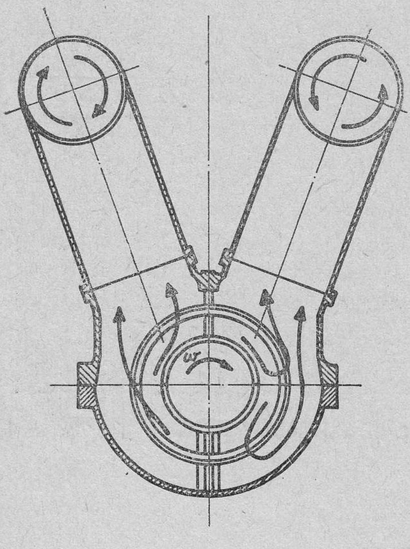

Fig. 2. Diagram of flow through the exhaust chamber of an intermediate-pressure section

In an actual turbine the flow entering the exhaust chamber of the intermediate-pressure section posssses always a tangential component of the velocity, which is imparted by the last rotating blade cascade. Under such conditions the left-hand piping and the right-hand piping are filled up differently – see streamline diagram in Fig. 2. On the left-hand side the dynamic pressure of the stream is utilized better, and therefore more steam is flowing off through the LH piping than through the RH piping. Measurements taken on an actual engine have shown that at all outputs of the engine about 54% of steam pass through the LH piping and 46% through the RH piping.

These facts can take place even in case of a flow through the inlet chamber of the low--pressure section; they can influence the conditions at the entry into the first stationary cascades, and can contribute to a non-uniform heating-through of the inlet chamber whenever the output of the turbine varies.

2. Last stage

From among the blade stages of the low-pressure section the most complicated flow conditions prevail in the last stage. The cascades of the last stage have comparatively long blades, work at high circumferential velocities which at the tip exceed the speed of sound, and handle large pressure drops with which the flow at transonic speeds is associated.

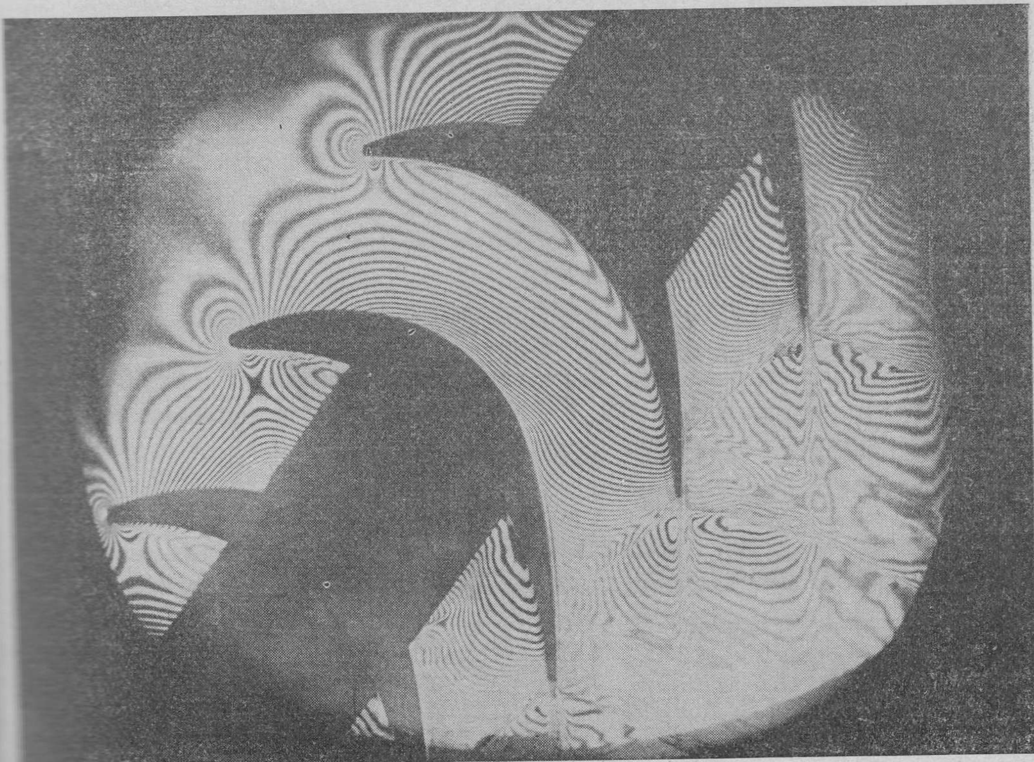

Fig. 3. Interferogram of flow through a blade cascade at the root of a rotating wheel at the Mach number of 1.10

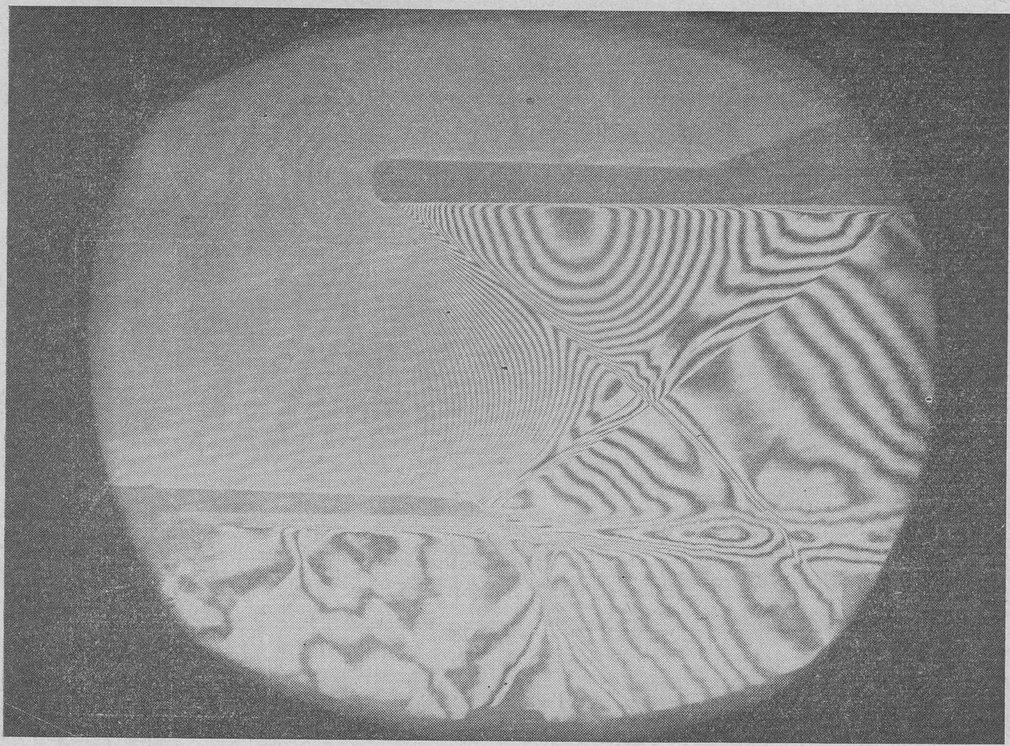

Fig. 4. Interferogram of flow through a blade cascade at the tip of a rotating wheel at the Mach number of 1.41

They work on wet steam, and the steam conditions there vary mostly with changes in the turbine output or in the condenser pressure.

An intricate transonic flow arises in the blade cascades at the root and tip of the rotating blades. The root cascade, as a rule, is distinguished for its high subsonic speed at the entry and low supersonic speeds at the exit. The flow through the cascade is in the strict sense of word transonic and cannot by simply broken down into subsonic and supersonic flows. The sonic line does not run across the channel in its narrowest point as one would conclude from a single-dimensional conception. It commences at the trailing edge, turns forwards in the channel, and terminates on the suction side of the neighbouring profile far ahead, sometimes as far as in the proximity of the leading edge, as it can be seen in the flow interferogram in Fig. 3. In this figure the arrangement of the shock waves and the wake at the exit from the cascade at Mach number $M_2=1.10$ are clearly visible.

The cascade at the tip of the rotating blades has thin profiles with a large pitch-chord ratio. It works under the designed working regime with a subsonic inlet velocity and with a high supersonic exhaust velocity, where the value of Mach number M_2 varies usually between 1.4 and 1.8. There are considerable problems in obtaining low losses with these cascades within the wide region of the Mach number. Fig. 4 presents and interferogram of a flow through such a cascade of the thin flat plate type, with a Mach number of 1.41.

Evident are some faults in the flow at the leading edge and the configuration of the shock waves and wakes at the exit.

With an actual rotating wheel the blades are made to untwist in rotation [1] on account of centrifugal forces, and the stagger angle of a profile is caused to change in the cascade at the tip of the rotating blades, as it is evident from Fig. 5. The situation at a standstill is evident from the top picture. Besides the blade profiles also damping wires can be seen. The lower picture illustrates the tip profiles at a speed of 3338 r.p.m. A considerable turning of the profiles relative to the damping wires is clearly evident. By taking measurements in five stages with a length of the blade of 840 mm it was ascertained that the degree of untwisting of the tip profiles at a speed of 3000 r.p.m. amounts to $5.75° \pm 0.25°$. This change in the stagger angle of the profiles should be taken into account, otherwise the flow conditions in an actual transonic cascade would differ entirely form the designed ones.

A problem analogous to that concerning the attempts to obtain a low loss coefficient within the wide limits of the Mach number does exist even for the root of the stationary cascade where the Mach number under designed conditions reaches a value of 1.4 to 1.7. Fig. 6 shows the loss coefficient ζ plotted against the Mach number M_2 for three types of stationary cascades. One cascade has been designed in a standard manner for a subsonic flow. In the case of this cascade high losses are encountered, caused by the flow separation when the shock wave and the boundary layer are interacting, provided that the Mach number is larger than the critical one. The second cascade has rear portion of the suction side of profiles designed for a supersonic shock-free flow; its shape caused, however, the

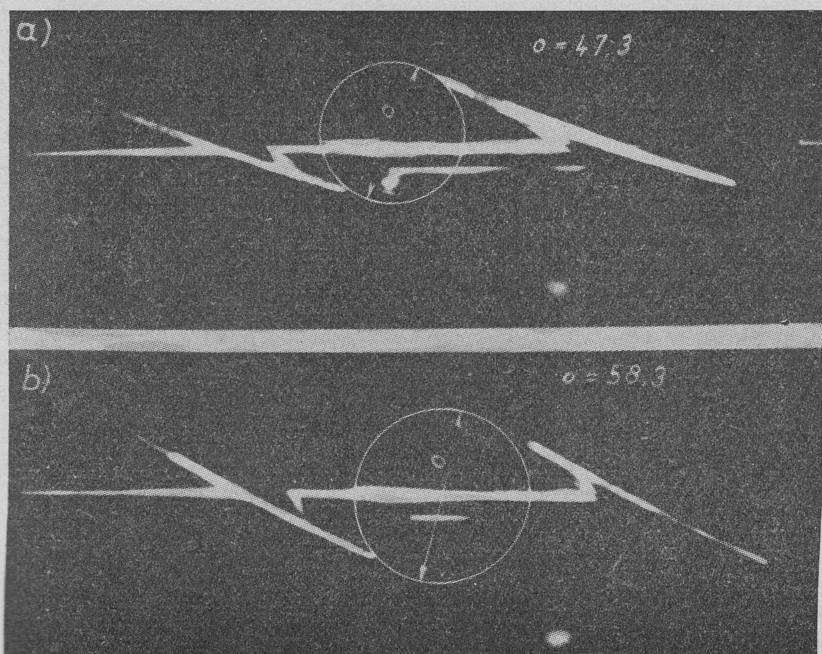

Fig. 5. Photograph of tip profiles of rotating blades; a) at a standstill, b) at a speed of 3338 r.p.m.

increase of losses if the flow conditions were near the Mach number of 1.0. The third cascade has been designed in such a way so as to ensure that the flow separation while the shock wave is interacting with the boundary layer will be put off and reduced.

For turbine cascades working at supersonic flow velocities at the exit the phenomenon of interaction of the shock wave with the boundary layer is of great importance, and is, as a rule, decisive for the losses. In experiments in aerodynamic wind tunnels the boundary

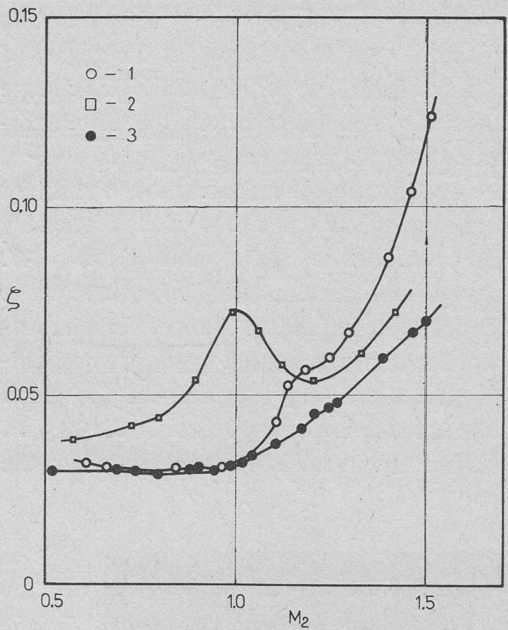

Fig. 6. Loss coefficient plotted against the Mach number for stationary cascade

1 – standard subsonic cascade, *2* – cascade for supersonic exit, *3* – special subsonic cascade

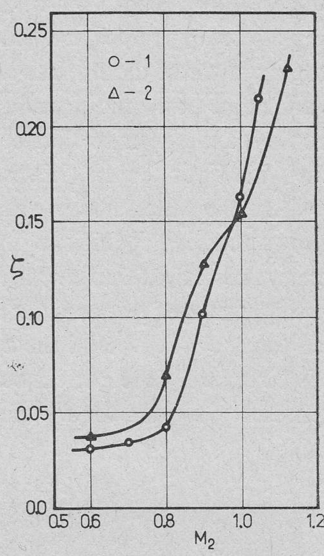

Fig. 7. Loss coefficient plotted against the Mach number for stationary cascade of subsonic type

1 – without turbulence, *2* – with an artificial turbulization of boundary layer

layer in the point of interaction is usually laminar, whilst in an actual engine this need not necessarily be the case. In order to judge the difference in the interaction of a normal shock wave with both the laminar and turbulent boundary layers, a turbulent boundary layer was created artificially on the suction side of the profile in a cascade of a stationary type, similar to the previously described cascade designed for the subsonic flow around. On a blade with a profile having a chord of 50 mm a wire with 0.05 mm in diameter was attached in a suitable point so as not to affect the configuration of shock waves and to prevent relaminarization of the boundary layer past the wire. A comparison of the loss coefficient ζ plotted against the Mach number M_2 for cases without and with an artificial turbulization of the boundary layer is evident from Fig. 7. At lower subsonic velocities the loss coefficient gets, as expected, increased due to larger frictional losses in the turbulent boundary layer. Within the region of the Mach number where the flow becomes separated as a result of an interaction of the shock wave with the boundary layer, the turbulization of the boundary layer gives rise to a small improvement only, which is obviously to be attributed to the geometry of the blade cascade. With a more suitable shape of the profile, a more considerable reduction in the region of the separated flow could apparently happen.

The blade cascades at the tip of the nozzle ring work within the limits of subsonic velocities and their design does not present any difficulty in this respect. The problem lies in that the three-dimensionality of flow through the stage exerts its influence sharply. Therefore it is necessary to take into account the radial component of the flow and the change in the thickness of the meridional streamlayer.

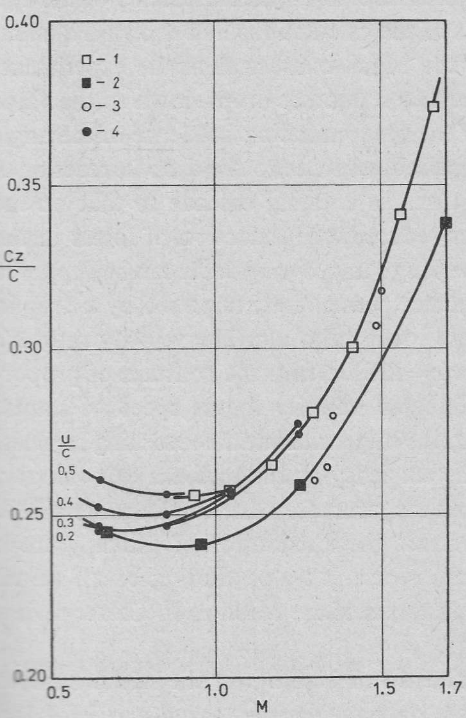

Fig. 8. Ratio of axial component to medium exit velocity plotted against the Mach number
calculated: *1* – actual stage, *2* – model stage, measured: *3* – actual stage, *4* – model stage

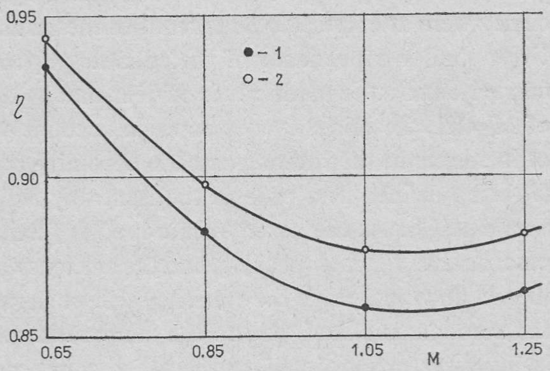

Fig. 9. Optimum efficiency of model stage plotted against the Mach and Reynolds numbers
1 – Re = $1 \cdot 10^5$, *2* – Re = $2 \cdot 10^5$

There arises a question, what flow conditions do exist in the last stage under non-rated aerodynamic regimes that correspond to conditions existing when reducing the output of the turbine and when changing the pressure in the condenser. It is advisable to prove this question on a turbine stage as a whole, irrespective of whether the stage is of the model type or is an actual one [2, 3, 4].

The variation in properties of the flow through the last stage can be judged from the ratio of the axial component of the medium exit velocity c_z/c (where c designates isentropic velocity from the enthalpy drop for the whole stage). In Fig. 8 it is plotted against the Mach number for the whole stage M (M is defined for the velocity c). In the graph there are plotted results of experiments done on an actual turbine stage, results obtained from a model stage when superheated steam was flowing, and results of calculations carried out with the help of a mathematical model of the flow through the last stage. The model stage showed certain geometrical deviations from the actual one, as is also evident from the different calculated values. The measurements on the model were carried out independently for various values of the velocity ratio u/c. It is clear that with the aerodynamic relieving of the stage the ratio of the axial component of the medium exit velocity up to the Mach number $M=1$ decreases rapidly to remain afterwards nearly constant.

The dependence of the efficiency η of the stage on the aerodynamic loading of the stage expressed again by the Mach number M for the whole stage is evident from Fig. 9. Here results obtained on the model stage when superheated steam was flowing are given. During these experiments also the Reynolds number Re underwent changes. The velocity ratio taken was the optimum one. On an actual stage in a turbine no measurements of the efficiency were attempted to be taken, but the results of the calculations had shown a qualitatively similar dependence of the efficiency upon the Mach number as in the experiments carried out on the model. It is evident that as the Mach number drops down to the value of $M=1.1$, the efficiency decreases on account of the predominating influence of the angle of incidence to the rotating cascade becoming impaired, and due to the drop in reaction R (R being defined as a ratio of the isentropy drop on the rotating cascade to that on the whole stage). As the aerodynamic load is further reduced, the reduction of losses in the cascades has a greater bearing, and the efficiency begins to improve again. It can be supposed that in an actual stage the efficiency will be in practice permanently impaired by a drop in the aerodynamic load, due to the increase in the velocity ratio u/c. The velocity ratio u/c is inversely proportional to the Mach number $u/c=K/M$, where the constant of proportionality K is represented by a ratio of circumferential velocity to the speed of sound.

If desired to clarify the flow through the last turbine stage with variable working regimes, the knowledge of integral parameters, such as the efficiency or the medium exit velocity, and the like, is insufficient. It is necessary to learn the structure of a flow in more detail. For that purpose an investigation was undertaken into the distribution of enthalpy drops between the stationary and rotating cascades at the root and tip of blading of an actual stage, where the ratio of gradients was expressed in terms of the reaction R. Under review was also the flow field at the exit from the stage.

The dependence of the reaction at the root and tip of blading on the Mach number is shown in Fig. 10. The measurements were taken on both the right-hand and left-hand sides of the engine. In experiments the designed working regime was not attained, the Mach number reached being 1.28 at the most. At this value the average reaction at the root and tip is $-$ 0.08 and 0.57 to 0.65, respectively. For the designed working regime of the stage, values were plotted that had been calculated on the basis of various presumptions about the curvature of meridional streamlines. It is obvious that in extrapolation of the measured dependence rather a good agreement with the results of calculations would be obtained. As the aerodynamic load of the stage decreases, the reaction at the root increases to reach zero at a value of the Mach number of about 0.65. The reaction at the tip, on the contrary, decreases at first to increase afterwards. Indicated are some differences between the right-hand and left-hand sides of the stage, resulting from the non-uniformity of the flow through the stage in the circumferential direction; they are primarily due to the effect of the exhaust chamber.

Within the limits of low aerodynamic loads the course of reaction is substantially variable. Negative values of the Mach number correspond to a higher pressure downwards of the stage as against the pressure upwards of the stage. They indicate that the stage works under these working conditions as a fan to compress the flowing medium, or that a recirculation flow through the whole stage takes place. In Fig. 10 the beginning of windage of the stage is also shown, defined by a value of the velocity ratio $u/c=1.1$. Under working conditions with a Mach number lower than 0.5 the last stage operates hence under a windage regime with the steam flowing but little at the same time. In Fig. 10 there is also shown the beginning of the recirculation flow at the root which will still be dealt with later on.

The investigation into the flow field at the exit from the stage has also brought about with it interesting information. For instance, from Fig. 11 the distribution of the exit angle of stream from the rotating blade cascade along the blade can be observed. Plotted are values that were measured on both the right-hand and left-hand sides, and, for the purpose of comparison, also the value of the exit angle drawn according to a sine rule. It is possible to observe marginal regions affected by the secondary flow, within which the values of the exit angle considerably diminish as compared with the values given on the basis of the sine rule. Within the region of the blade tip also act the radial components of the velocity enforced by the exhaust chamber. Within the middle portion of the blade the measured values of the exit angle are in an acceptable agreement with the sine rule, which does well justice to

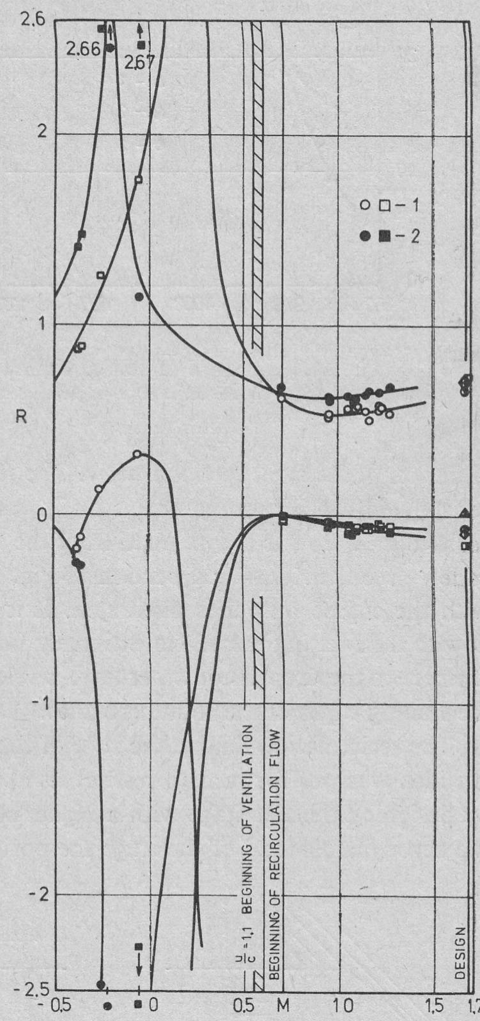

Fig. 10. Reaction at the root and tip of blading plotted against the Mach number: *1*—right-hand side, *2*—left-hand side

the exit angle at a two-dimensional subsonic flow through a cascade. In the experiment under review the flow at the exit from the rotating cascade was likewise subsonic so that one can suggest that within the middle portion of the rotating blade cascade the flow closely approaches in two-dimensional flow.

The investigation into the velocity field at the exit has likewise shown that under low aerodynamic loads on the stage there occurs a recirculation flow at the root of the blading and, under certain conditions, even at the tip of the blading. The recirculation flow is represented diagrammatically in Fig. 12. In the left-hand portion of Fig. 12 one can see that

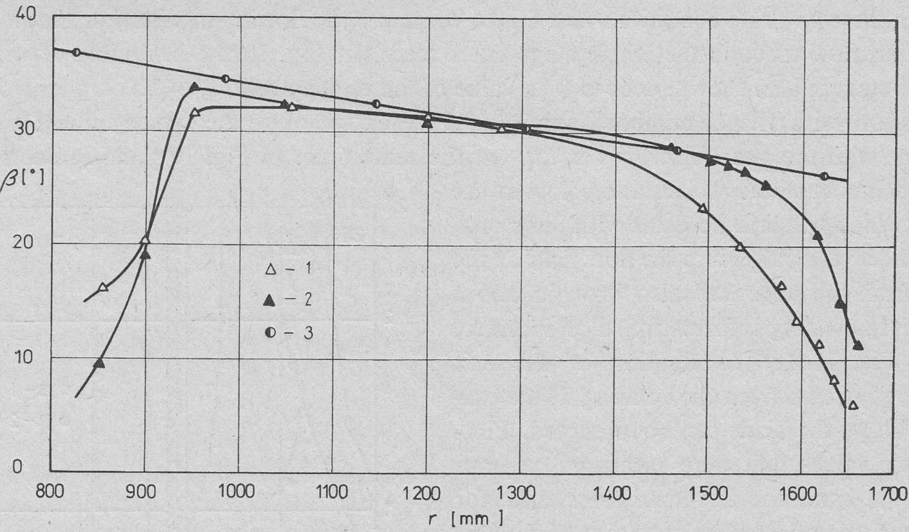

Fig. 11. Exit angle of stream from a rotating cascade plotted against radius: measurement *1* – right-hand side, *2* – left-hand side, *3* – sine rule

situation in the axial sectional view. The recirculation flow at the root occupies a characteristic width S. From the analysis of the measured exit stream angles it follows that the recirculation flow is a continuation of the secondary flow at the root of the rotating cascade – see the right-hand portion of Fig. 12, where the change in the exit stream angle with the engine output is illustrated. It is evident that the direction of the recirculation flow to the rotating cascade is extremely unfavourable. If a wet steam is the flowing medium, then the droplets of water have a velocity high enough to give rise to the erosion of the trailing edges of the rotating blades.

The recirculation flow at the root of the rotating blade is an undesirable phenomenon. An idea was put forward to restrict its appearance and limit to the least possible extent by designing the last stage with a higher reaction at the root, in order that in decreasing the aerodynamic load of the stage the positive pressure drop and the flow in the required

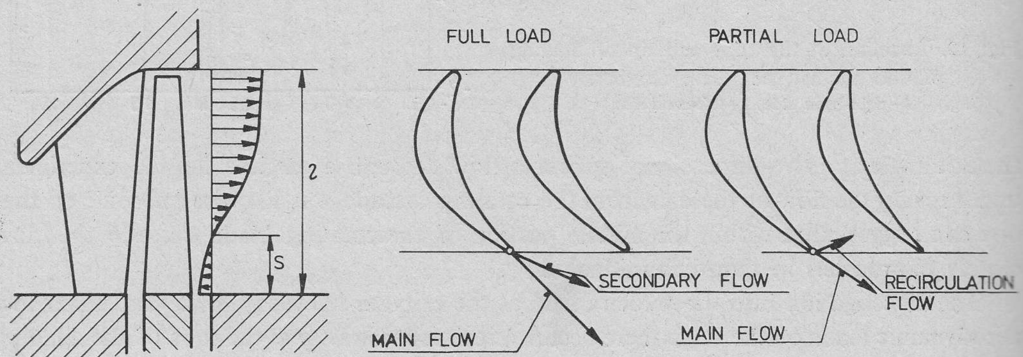

Fig. 12. Diagram of the recirculation flow at the root of rotating blade

direction may be maintained on the rotating cascade for a longer time. Therefore, a version of the last stage was examined, in which the stationary cascade had its blades inclined from the radial line in direction of the exit stream. The inclination of the stationary blades increases the reaction at the root and decreases the reaction at the tip, which is favourable for the flow through the stage. The dependence of the reaction at both the root and tip of the blades on the aerodynamic load applicable to this version is shown in Fig. 13. Here again are plotted values that have been ascertained on both the right-hand and left-hand sides of the stage. The measurements were carried out to a Mach number of 1.45, the reaction at the root and tip amounting 0.39 and 0.435, respectively. As the Mach number decreases, the reaction at both the tip and root decreases rapidly to reach zero within the limits of the Mach number $M = 0.5$ to 0.75. Within these limits of the Mach number there also appears the beginning of the recirculation flow and the beginning of windage operation of the stage. In Fig. 13 there are also plotted calculated values of the reaction under the designed working regime. It is evident that the actual reaction at the tip is rather lower than it should be, and at the root it is, on the contrary, considerably higher. The difference in the calculated and measured values of reaction has been caused by an effect of inclination of the stationary blades stronger than determined by the present simplified theory of this phenomenon.

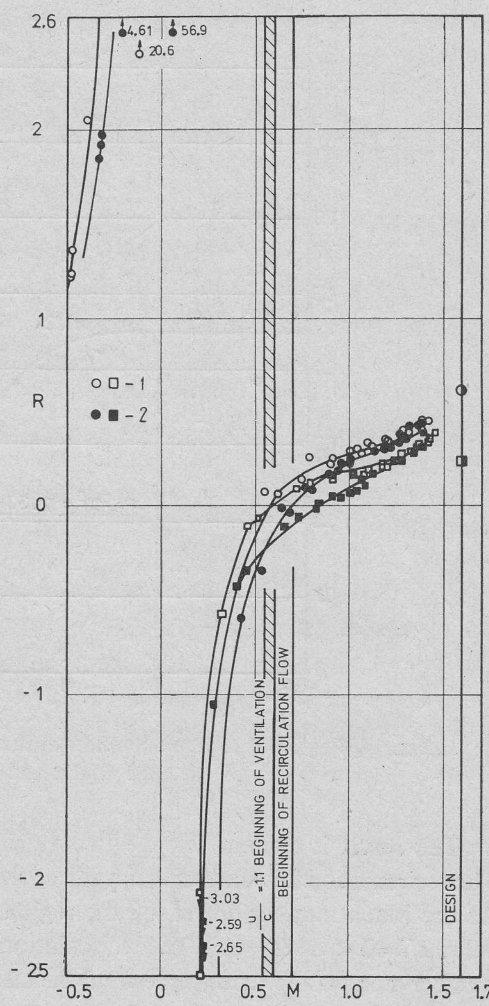

Fig. 13. Reaction at the root and tip of blading plotted against the Mach number for a stage with inclined stationary blades
1 – right-hand side, *2* – left-hand side

The appearance of recirculation flow at the root of blading associated with a drop in the aerodynamic load foreshadows a transition of the stage to a windage operation at which the stage does not give the output any more and appears with turbine stages to be inevitable. It has been ascertained with the last stages even by other authors – R. Ecker [5], W. Hauke [6], V. P. Lagun and L. L. Simoyu [7]. From the experiments carried out so far it follows that the recirculation flow through the last stage depends in the first place on its aerodynamic load. To express this phenomenon quantitatively, the dependence of the width-to-blade length ratio of the region s/l upon the Mach number and Reynolds number of flow through the stage was selected, within which the recirculation flow arises.

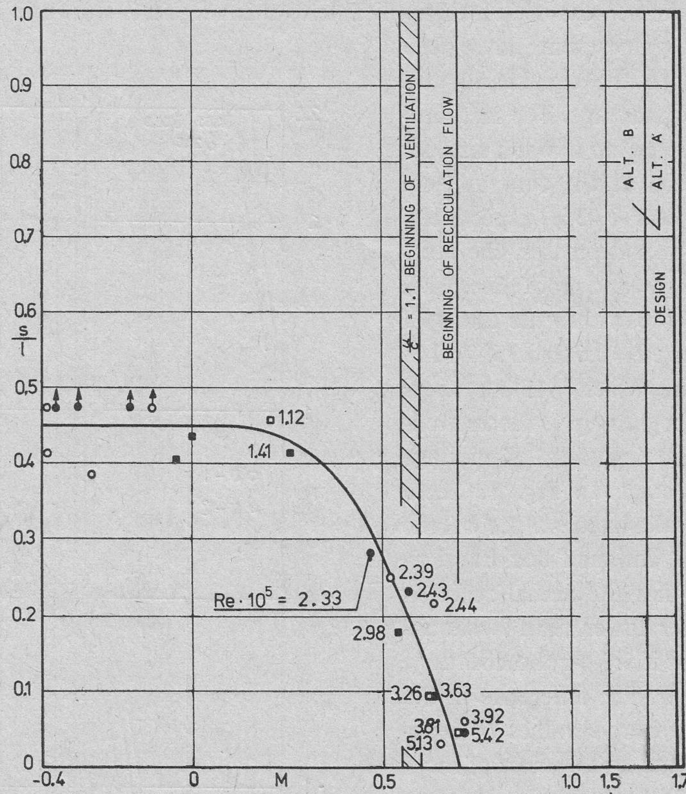

Fig. 14. Region of recirculation flow at the root of blade of the last stage plotted against the Mach and Reynolds numbers

The result is evident for the two versions of the stage under review from Fig. 14. The recirculation flow begins to appear at the root of the blade at the Mach number of 0.7. As the Mach number decreases, the region of the recirculation flow extends to reach the value $s/l=0.45$ at $M=0.2$. The recirculation flows does not depend neither on the Reynolds number nor on the arrangement of the stationary cascades by which the two versions of stages differ from each other. Fig. 14 also depicts designed and windage conditions. It can be seen that the appearance of the recirculation flow closely adjoins to the beginning of windage of the stage and will obviously depend but little on the flow conditions in the stage and on the reaction at the root under the designed working regime.

The connection between the recirculation flow at the root of the last stage and the working conditions of a turbine will become more evident if we change the dependence according to Fig. 14 to a dependence of the width-to-blade length ratio of the region of recirculation flow and pressure at inlet p_a and exit p_e from the low-pressure section. The result is presented in Fig. 15. The pressures are related to their design values, while the ratio of pressures upstream of the low-pressure section corresponds approximately to the ratio of the turbine outputs N/N_n. It is evident that the recirculation flow at the root of the stage begins to appear at about one-third of the turbine output, if the pressure in the condenser is as designed, or also at a designed output, if the pressure in the condenser is three times the designed one.

In the case of large steam turbines the requirements for the operational range of outputs are gradually increasing. If until the recent time the minimum output mostly required amounted to 60 to 70% of the nominal one, now there appear requirements for the least long-time turbine output equalling 30% of the rated one. To meet this requirement, if desired to avoid the operation with a recirculation flow through the last stage, is very difficult. Requirements for tightness of the condensation plant and for its maintenance are ever growing.

Based on the measurements taken, an attempt was made to diagrammatically represent the flow through the last stage under low aerodynamic loads.

Let us pay our attention first to the conditions of flow, under which the recirculation flow begins to appear at the root of blading. This situation in the original stage is shown in Fig. 16 in the left-hand portion. In addition to the recirculation flow at the root the same flow is caused to also appear at the tip of the blade. The recirculation flow at the tip is due to the diffuser effect of two deflectors of the exhaust chamber. In the right-hand portion of Fig. 16 one can see analogous conditions in a stage which includes a stationary nozzles with inclined blades. Besides the recirculation flow at the root even here does exist a vortex at the tip of the blading, this time between the stationary cascade and the rotating cascade.

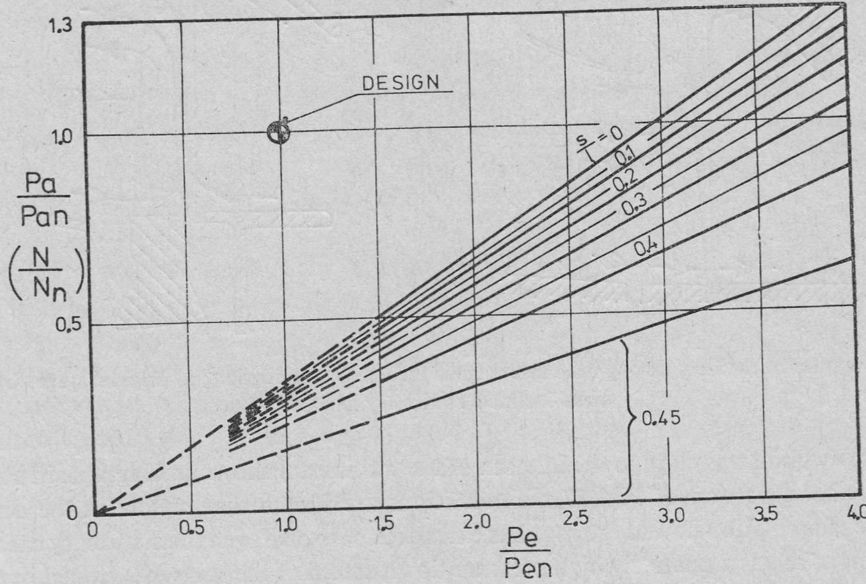

Fig. 15. Region of recirculation flow at the root of blade of the last stage plotted against the pressure in the condenser p_e and the pressure upstream of the low-pressure section p_a or output N

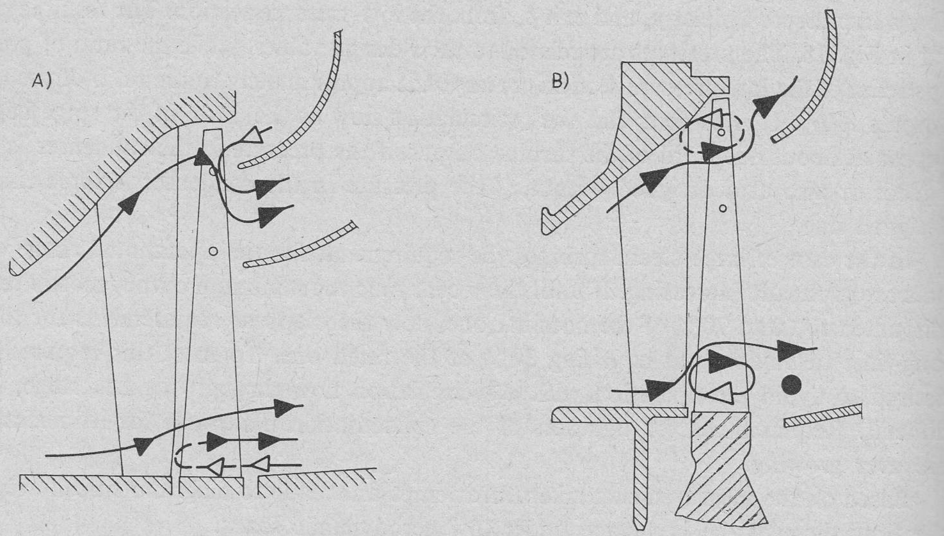

Fig. 16. Diagram of the flow through the last stage when recirculation flow appears at the root of rotating blade; a) original stage, b) stage with inclined stationary blades

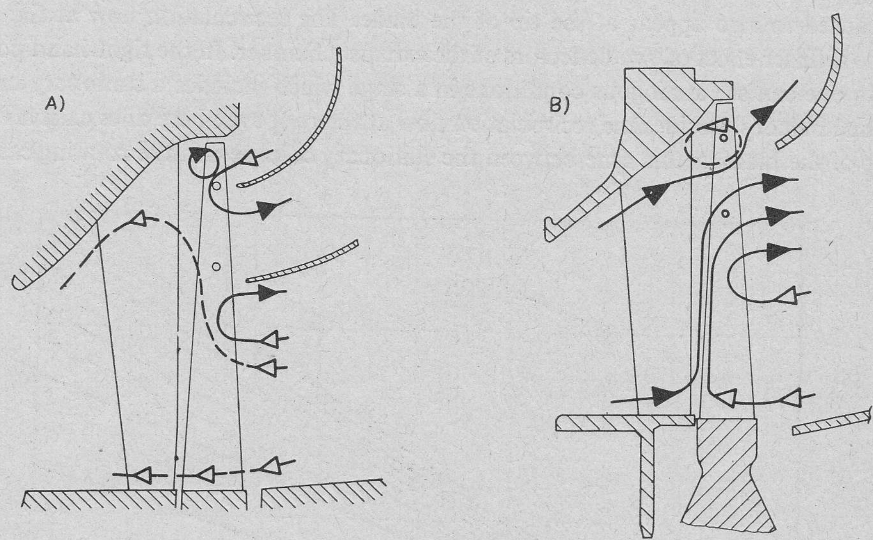

Fig. 17. Diagram of the flow through the last stage if turbine is running idle; a) original stage, b) stage with inclined stationary blades

The flow conditions close to the idle run of the turbine are shown in diagrams in Fig. 17. In the left-hand portion of Fig. 17 the conditions are shown that prevail in the original stage. The recirculation flow at the root has extended to involve even the stationary cascades. The reason of this is a relatively high pressure downstream of the stage, developed in transferring large quantities of steam to the condenser by means of bypass stations. A certain amount of steam passes through the last stage back to the first regenerative steam extraction point. The recirculation flow at the tip is supplemented by a vortex between the stationary cascade and the rotating cascade, which, however, rotates in the direction opposite to that in the version with inclined stationary blades. The appearance of the vortex between the stationary cascade and the rotating cascade is due to the meridional extension of the stationary cascade, which at a low speed manifests itself by a strong diffuser effect at which a flow separation from the wall takes place. In the version with inclined stationary blades this vortex appears sooner as the flow separation from the wall is furthered by the inclination of the stationary blades. The appearance of the vortex at the tip between the stationary cascade and the rotating cascade under low aerodynamic loads is for the last stages characteristic. It was also ascertained by V. P. Lagun *et al.* [8].

On the right-hand side of Fig. 17 one can see analogous flow conditions for the version with inclined stationary blades. The recirculation flow at the root does not pass through the stationary cascade and at the tip it is missing, since there is no second deflector to form a diffuser together with the present one.

The recirculation flow at the root of the stage is hardly dangerous from the viewpoint of exciting forces. This, however, cannot be claimed in respect of the recirculation flow at the tip or in respect of the vortex at the tip between the stationary cascade and the rotating cascade, especially if these conditions happen to come true under a higher pressure in the condenser. The turbine should pass through these conditions as rapidly as it can, and in no case it should be operated under them for a prolonged time under an elevated pressure in the condenser.

3. Exhaust chamber

A continuation of the last stage of a steam turbine is an exhaust chamber. The flow through the last stage and the exhaust chamber affect each other. An example of the influence of deflectors on the appearance of recirculation flow at the tip of blading of the last stage was described above. Let us notice furthermore the influence of the stage on the cooling of the exhaust chamber in idle run.

The recirculation flow at the root of the last stage has a positive significance for the cooling of the exhaust chamber in idle running. Through its medium the last stage suck-in the cool steam that is bypassed to the condenser, in order to cool with its help the whole low-pressure section. Fig. 18 shows the distribution of temperatures in the flow downstream of the laste stage in two idle runs, which differ from each other by the quantity of steam by-passed to the condenser.

In idle running *1* a sufficient amount of cooled-down steam was admitted, so that the recirculation flow between the blade root and the lower deflector had saturation temperature t_s, and the temperature of steam at the blade tip increased by about 25°C.

Idle running *2* is distinguished for a reduced supply of a cooled-down steam. The steam temperature at the blade root is 30 to 90°C higher than the saturation temperature. Especially in the right-hand portion of exit the temperature rise is enormous. At the tip the steam temperature reached a value 110°C higher than the saturation temperature.

In rather an old type of exhaust chamber a non-uniform cooling of the outer shell of the low-pressure section was observed in a long-time idle run. Under certain conditions after a time of about one hour local heating of front faces of the outer shell on the right-hand side of the chamber top to a temperature of about 130°C took place. Fig. 19 is a view into the inside of the exhaust chamber. Indicated are isotherms on the outer shell and diagrammatically represented are probable streamlines of the cool steam flowing from the

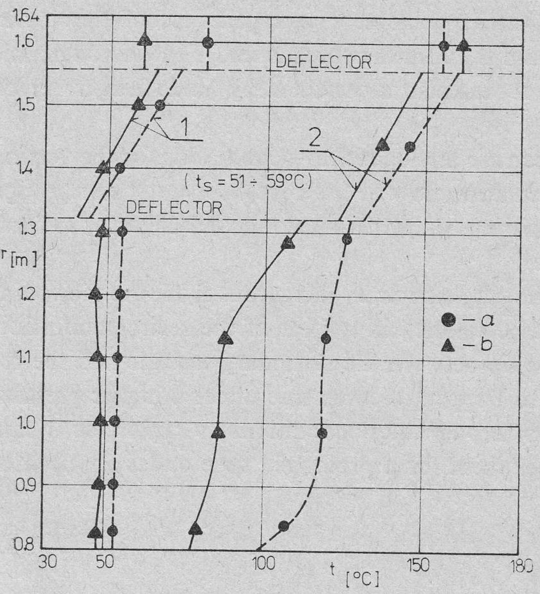

Fig. 18. Steam temperature at the exit from the last stage when turbine is running idle
a – right-hand side, *b* – left-hand side

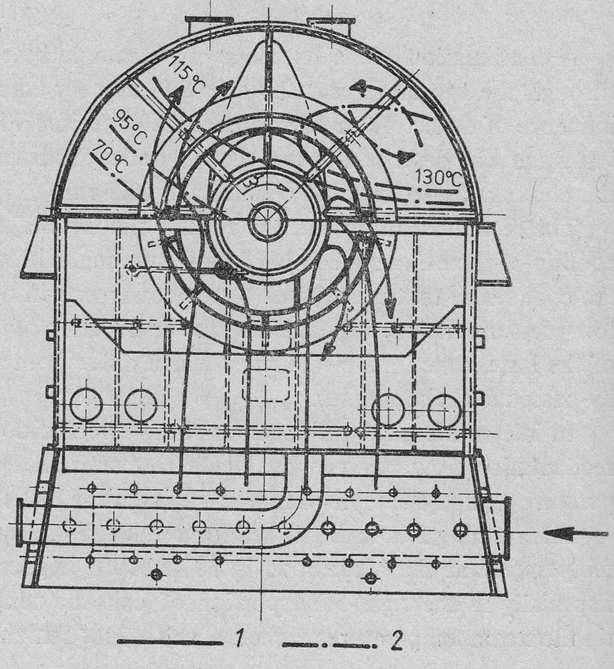

Fig. 19. Diagram of cooling of the exhaust chamber of the low-pressure section when turbine is running idle
1 – streamlines, *2* – isotherms

by-pass stations to the last stage and from there to the surrounding space. Steam outgoing from the stage has in idle running a strong tangential component in direction of rotor rotation, which leads to creation of a wake region downstream of the stiffening rib in the right-hand portion of the chamber top, and this is the very cause of the local heating of the outer shell.

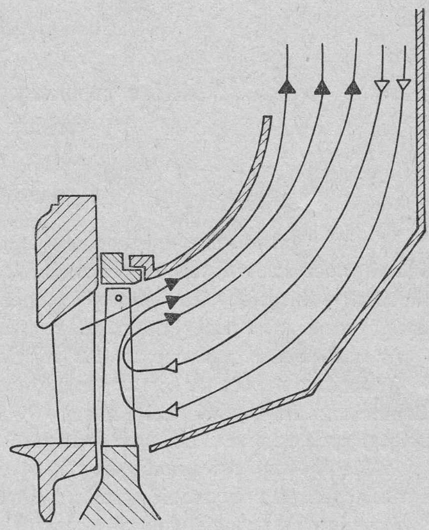

Fig. 20. Diagram of the flow through the diffuser of the exhaust chamber for the turbine running idle

Of interest is the development of recirculation flow in the space downstream of the last stage if an axial-radial diffuser is present. Measurements have shown that at entry into the diffuser the recirculation flow occupies 45% of the channel width, and at exit still 30%. For a diagrammatical representation thereof see Fig. 20.

With the flow through the exhaust chamber a number of other, well known problems are associated. The exhaust chamber should have low losses in the region of outputs employed for a long-time. At the exit from the last stage it is necessary to have as uniform a distribution of pressures on the periphery as possible, in order that the exhaust chamber may act with small exciting forces upon the rotating blades. Under control should also be the uniformity of the stream at entry into the condenser.

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Niektóre problemy przepływu w części niskoprężnej turbiny parowej dużej mocy

Streszczenie

W pracy omówiono wybrane problemy trójwymiarowego przepływu pary wodnej przez niskociśnieniową część dużej turbiny parowej. Specjalną uwagę poświęcono stanom istniejącym w warunkach roboczych odbiegających od przewidzianych w projekcie. Zwrócono również uwagę na pewne wtórne zjawiska związane z przepływem, jak nagrzewanie ścianek, naprężenia dynamiczne, erozja łopatek itp.

W rurociągach zasilających ma miejsce wirowanie strumienia i nierównomierność przepływu.

Wskazano na złożoność przepływu transsonicznego w palisadach łopatkowych ostatniego stopnia. Rozpatrzono zmiany własności przepływu i sprawności ostatniego stopnia w warunkach roboczych odbiegających od przewidzianych w projekcie. Specjalną uwagę zwrócono na rozkład spadków ciśnienia w palisadach ostatniego stopnia oraz na pole przepływu na wylocie. Zdefiniowano własności przepływu recyrkulacyjnego przy stopie wirującej łopatki oraz wirów przy wierzchołkach łopatek.

Przepływ na wylocie wpływa na warunki panujące w ostatnim stopniu, i *vice versa.* Zastosowanie na wylocie owiewek kierujących może doprowadzić do przepływu recyrkulacyjnego przy wierzchołkach łopatek ostatniego stopnia. W artykule omówiono warunki przepływu na wylocie i chłodzenie wylotu podczas pracy turbiny bez obciążenia.

Некоторые проблемы течения в части НД паровой турбины большой мощности

Резюме

В работе обсуждаются избранные проблемы пространственного течения водяного пара через часть НД большой паровой турбины. Особое внимание уделяется состояниям рабочих условий, отличающихся от расчетных. Обращается также внимание на некоторые вторичные явления, связанные с течением, как нагрев стенок, динамические напряжения, эрозия лопаток и т.п.

В паровых трубопроводах выступают завихренности потока и неоднородность интенсивности течения.

Указывается на сложность сверхзвукового течения в решетках лопаток последней ступени. Рассматриваются изменения свойств течения и к.п.д. последней ступени в рабочих условиях, отличающихся от расчетных. Особое внимание обращается на распределение перепадов давления в решетках последней ступени, а также на поле течения на выходе. Определяются свойства рециркуляционного течения у корня вращающейся лопатки и вихрей у вершин лопаток.

Течение на выходе влияет на условия, действующие в последней ступени и наоборот. Применение направляющих обтекателей на выходе может способствовать возникновению рециркуляционного течения у вершин лопаток последней ступени. В статье обсуждаются условия течения на выходе и охлаждения выхода во время холостого хода турбины.