

P O L S K A A K A D E M I A N A U K
INSTYTUT MASZYN PRZEPŁYWOWYCH

PRACE
INSTYTUTU MASZYN
PRZEPŁYWOWYCH

TRANSACTIONS
OF THE INSTITUTE OF FLUID-FLOW MACHINERY

70-72

WARSZAWA-POZNAN 1976
PAŃSTWOWE WYDAWNICTWO NAUKOWE

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

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Warszawa 1976

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.

EDWIN SOMM

Switzerland*

Problems in the Development of Last Stage Blades for Large Steam Turbines

Symbols

c_D — steam velocity,	s — flow path,
c_t — tangential velocity,	S_M — surface,
c_W — water velocity,	t — time,
c_z — axial velocity	u — peripheral velocity,
D_M — mean diameter,	$\dot{V}_M$ — volume loss per unit time,
D_N — hub diameter,	x_2 — specific steam content at stage outlet,
$\dot{e}$ — erosion rate,	β_2 — flow angle,
f — frequency,	η — efficiency,
h — blade height,	ε — standardized erosion rate,
p — pressure,	σ — surface tension.
P — output,	

1. Introduction

The continual rapid increase in the unit output of steam turbines for power stations makes great demands upon the turbine manufacturer in his efforts to develop those components which determine the output limit of the unit. One of the most important limiting components is the last stage rotor blade of the low pressure turbine. The demand for a blade which offers operating reliability and maximum efficiency presupposes the solution of highly complex mechanical and aerodynamic problems. The approach we adopted in attempting to solve the various types of problems was essentially as follows:

- a comprehensive theory was developed in order to understand the physical inter-relationships,
- programmes were evolved to calculate behaviour on the basis of the theory,
- the results arrived at by calculation were verified by means of small scale models and actual size tests.

* Brown Boveri and Co., Baden.

With the assistance of the most up-to-date methods of calculation and testing it is now possible to fulfil all requirements, so that, as far as the last stage is concerned, all required unit outputs in the next few years can be met.

2. The importance of the last stage for large steam turbines

2.1. Maximum attainable output

It is generally known that the specific costs of the installation decline as the unit output increases, and so the aim is to build turbines with the maximum possible unit output. The output limit is, however, largely determined by the size of the last stage blades. It is easy to deduce from this that at a given vacuum there is a correlation between maximum possible output and the total exhaust area. This can be increased either by the number of parallel flows or by means of larger diameters and blade lengths in the last stage. There

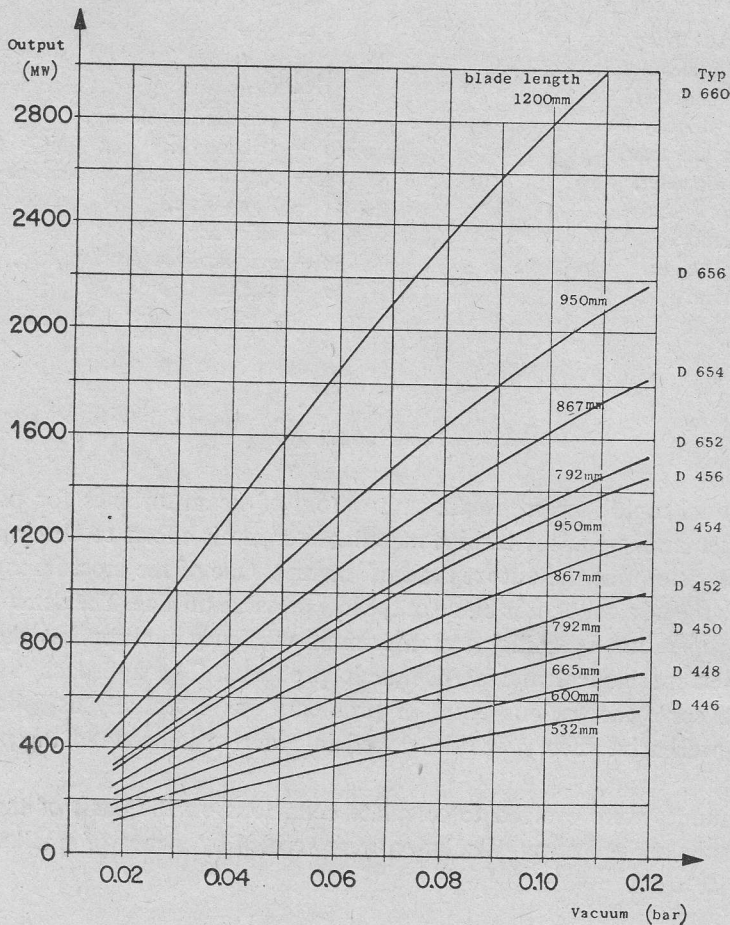


Fig. 1. Output for conv. reheat turbines of 3000 r.p.m.

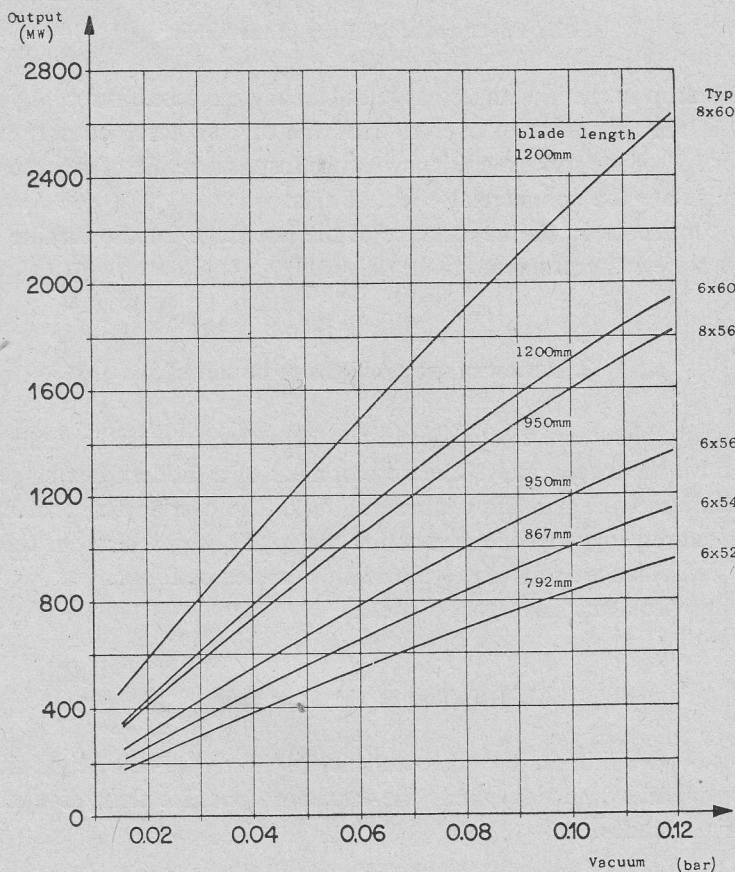


Fig. 2. Output for wet steam turbines of 3000 r.p.m.

are, however, limits to both of these methods. For example, a unit with 6 or 8 double flow low pressure casings would be unusual. On the other hand, at a given rotational speed, blades cannot simply be made of any height, for aerodynamic and mechanical reasons. A possible alternative would be to build two units for a given output or operate at a lower rotational speed, e.g. 1500 r.p.m. However, both of these are special solutions, as both twin-unit installations and 25 Hz units, in comparison to 50 Hz units, are becoming increasingly more expensive (evaluated already with η).

It is therefore a matter of general interest, both for conventional units and especially for atomic power station turbines, to make the exhaust cross-section per flow as large as possible on the 50 Hz unit. Fig. 1 shows the output limits for plants using fossil fuels. For example, at vacuum of 0.10 bar, which corresponds approximately to the exhaust pressure prevailing in cooling tower operation, outputs of 2800 MW are possible with a blade 1200 mm long in the last stage. It is obvious that for a number of other reasons this output figure is still utopian. In the case of turbines using steam from a light water reactor (Fig. 2) the same blade would give an output of 2300 MW, although only in 4 LP-cylinder design. This output is probably achievable in the not so distant future. This is intended merely to illustrate that last stages are already available capable of meeting the requirements of the next 10-20 years.

2.2. Effect on the efficiency of the turbine

The enthalpy drop in the last stages has been steadily increased in recent years, so that this stage may attain an output of over 10% of the turbine total output. Studies have shown that over 1% of the turbine efficiency can be lost as a result of poor blade design. It is easy to calculate the commercial effect of this.

After these comments on the importance of the last stage for the turbine as a whole, we should now review the problems to be solved.

3. Review of the problems to be solved

Although the problems are treated separately hereafter, it must be realized that it is impossible to solve them separately. The development of a modern last stage is a prime example of an extremely demanding engineering task. The overlapping problems must be solved in their entirety, having due regard for the relationships between them. Only in this way can a well-balanced solution to the whole complex be found.

The following problems have to be solved:

3.1. Definition of the basic data

This involves the definition of the characteristic data, such as blade height and diameter, with regard for efficiency, and especially the maximum possible shaft diameter from the point of view of mechanics.

3.2. Mechanics, static

Here it is vital to have effective and reliable calculating methods, in order to predict static stress as accurately as possible so that maximum operating safety can be attained.

3.3. Mechanics, dynamic

If blade failure still occurs today, this is generally a result of dynamic stress. The greater the static stress to which a component is subjected, the smaller the reserve for dynamic stress. It is therefore a question of finding a method of designing blades whose shape will not cause failure due to vibration.

3.4. Mechanics, blade root

The blade root has to transmit the forces of several hundred tons to the shaft. The shaft and the blade root must be co-ordinated carefully with each other. The requirements of reasonable manufacturing cost must also be considered.

3.5. Metallurgy

The principal demands concerning the material are:

- high yield strength,
- high fatigue strength,
- good damping characteristics,
- not susceptible to stress corrosion,
- good erosion resistance or possibility of its protection,
- easy to forge and machine.

3.6. Fabrication

The maximum degree of precision is required in manufacture, otherwise additional, uncontrollable stresses can arise immediately. The efficiency is equally sensitive to poor design and to very rough surfaces. And, last but not least, the manufacturing cost should be reasonable.

3.7. Aerodynamics

On account of the large blade height to diameter ratio, the flow can no longer be treated as one-dimensional, as, for example, in high pressure turbines. Methods of calculation have to be developed for determining, with the required degree of accuracy, the flow patterns at each radius. Profiles must then be designed, ranging from a simple subsonic profile with a high degree of deflection to a transonic profile with a small degree of deflection. The aim is to achieve high efficiency and an accurate outlet angle.

3.8. Erosion resistance

This is essentially a question of defining the erosion hazard quantitatively, and then finding suitable methods of reducing it. It is preferable to take measures which strike at the causes of erosion, rather than those which offer only passive protection.

We shall now discuss in more detail some of the problems mentioned above.

4. Definition of basic data

The development of a modern last stage involves considerable expenditure of time and money. The first studies were concerned with the problem of whether the last stage should be manufactured as an individual item or for each special case or select them from a discrete range of standard types, which is largely a matter of economics.

Individual fabrication has the great advantage that the blade can be adjusted to an optimum degree to the specific operating conditions, i.e. maximum efficiency is attained for every turbine.

Against this advantage one must weigh the following disadvantages: Development costs increase immensely, manufacture becomes more expensive, keeping stock impossible, there is no assurance of interchangeability, etc. These advantages and disadvantages must be weighed carefully. The decision is in favour of standardization. This standardization was, by the way, extended to the entire low pressure turbine.

At the same time, the laws of similarity were used in order to derive blades for various rotation speeds from one blade type for a certain rotation speed. It is easy to demonstrate that families of blades with exactly similar mechanical and aerodynamic characteristic can thus be obtained.

The result of these cost/efficiency calculations provide a matrix of basic data for blade height h and mean diameter D_M and derived from this the important exhaust cross-section, which, as we have seen, is decisive in limiting the output of any type.

The next step is to design the stationary and the rotating blades for each type. We shall only concern ourselves here with the rotating blades. We shall turn first to the mechanics.

5. Mechanical problems

For aerodynamic reasons, different profile shapes must of course be used for each radius. This requirement results in a very complex blade shape, as the example in Fig. 3 shows.

The following principal mechanical problems have to be solved in designing such a blade:

- stress distribution in the blade body,
- vibrational behaviour.

5.1. Stress distribution in the blade body

A modern last stage is subjected to considerable stress. The resulting 3-dimensional state of stress is caused in operation by a variety of stress components [1]. These are:

Centrifugal tensile stress—caused by the mass of the rotating blade over the cross section under observation;

Centrifugal bending stress—attributable to the fact that the centres of mass of the profiles are not radially located, so that bending stresses are caused by the centrifugal forces.

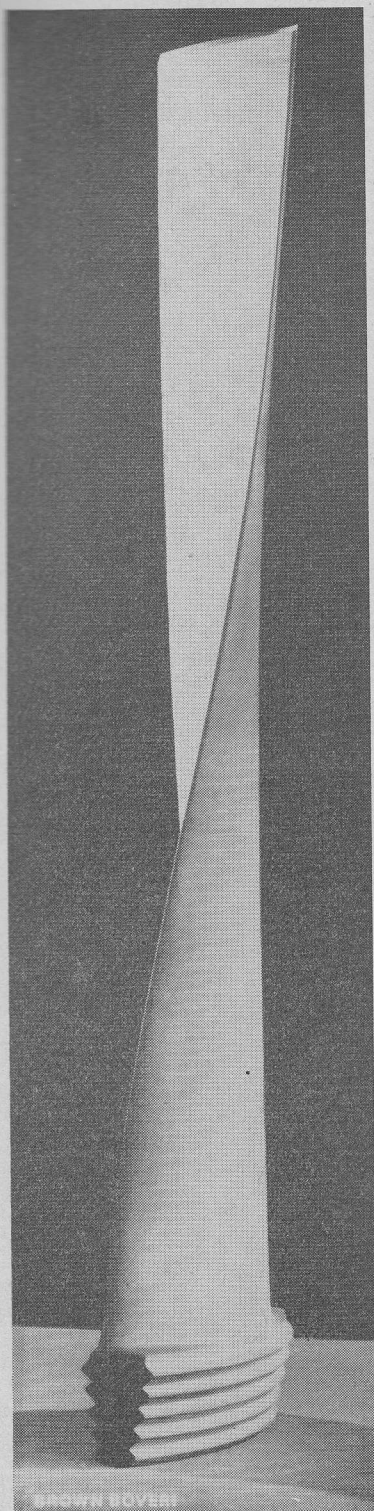
In the case of cylindrical blades or those which are only very slightly twisted, a knowledge of these two stress components is sufficient. This is not so, however, for those which are twisted to a greater degree, where the following additional stress factors have to be taken into account:

Untwisting normal stress and

Untwisting shear stress—both of them resulting from the blade's untwisting as a consequence of centrifugal force.

For the sake of completeness we should also mention the

Steam force bending stress—which is due to the flow forces. In last stages it is very small in relation to the total stress.



	Type of stress	Cause
Static	1 Constant tension due to centrifugal force σ_z	Centrifugal force produced by the blade mass situated above the given cross section
	2 Flexural stress due to centrifugal force σ_B	The departure of the line joining the centre of gravity of the section from the radial
	3 Untwisting normal stress σ_E	The twisting of the blade due to centrifugal force
	4 Untwisting shear stress τ_E	The twisting of the blade due to centrifugal force
	5 Flexural stress due to steam force σ_E	Steam force acting on the blade
Dynamic	6 Alternating flexural stresses σ_W	Steam flow deviations from the preceding stationary blading, detachment, asymmetry (disturbances) in the design (at the horizontal joint), disturbing internals such as probes, critical speeds, short circuit at the generator

Fig. 4. Stresses in the twisted last-stage blades of turbines in service

Fig. 3. Modern LSB

Firstly, a method of calculation based on theoretical considerations is to be developed in order to determine the above stresses at any given point. The basic theory applied is described in [2]. We shall simply repeat here the principal physical assumptions:

Blade geometry

- The blade is considered as a straight beam.
- The radius of curvature of the line connecting all the shear mid-points is very large in relation to the length of the blade.
- Continuous variation of the shape and dimensions of the blade sections along the blade height is assumed, also a slow change in shape and dimensions.
- The blade is a slender rod, i.e. $s_{\max}/h \ll 1$.

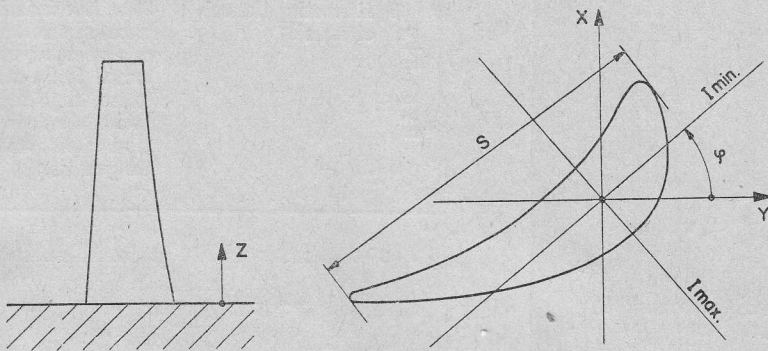


Fig. 5. Geometrical parameters

With respect to deformations

The usual assumptions are made within the scope of the transverse beam theory.

- When a blade is deformed, originally flat blade sections remain flat.
- If torsion occurs, the individual blade sections rotate in their sectional planes around the given torsional centres.
- The deformations of the blade are assumed to be small in relation to the dimensions of the blade.

With respect to the method

- The blade is defined by discrete blade sections.
- Geometrical characteristic quantities such as surface, moments of inertia, etc., can be calculated for any section, using known methods.
- The equations for elastic forces and moments take into account the coupling of bending and torsion.
- It is assumed that the centrifugal forces occurring during the rotation of the blade cause a constant normal stress across a blade section. This assumption simplifies the relationships, especially at the transition point between the blade body and the blade root.

The next step was to programme the extremely complex calculating process for the computer. Fig. 6 shows the result of such a calculation. From such calculation it is then immediately apparent which section of the blade is subjected to the heaviest loading and what influence the additional stresses have.

In accordance with the basic principles of our approach we then had to verify the accuracy of these calculations before proceeding to determine the dimensions of an actual blade.

The more accurately the calculation predicts the actual stress, the more effectively can the material be used.

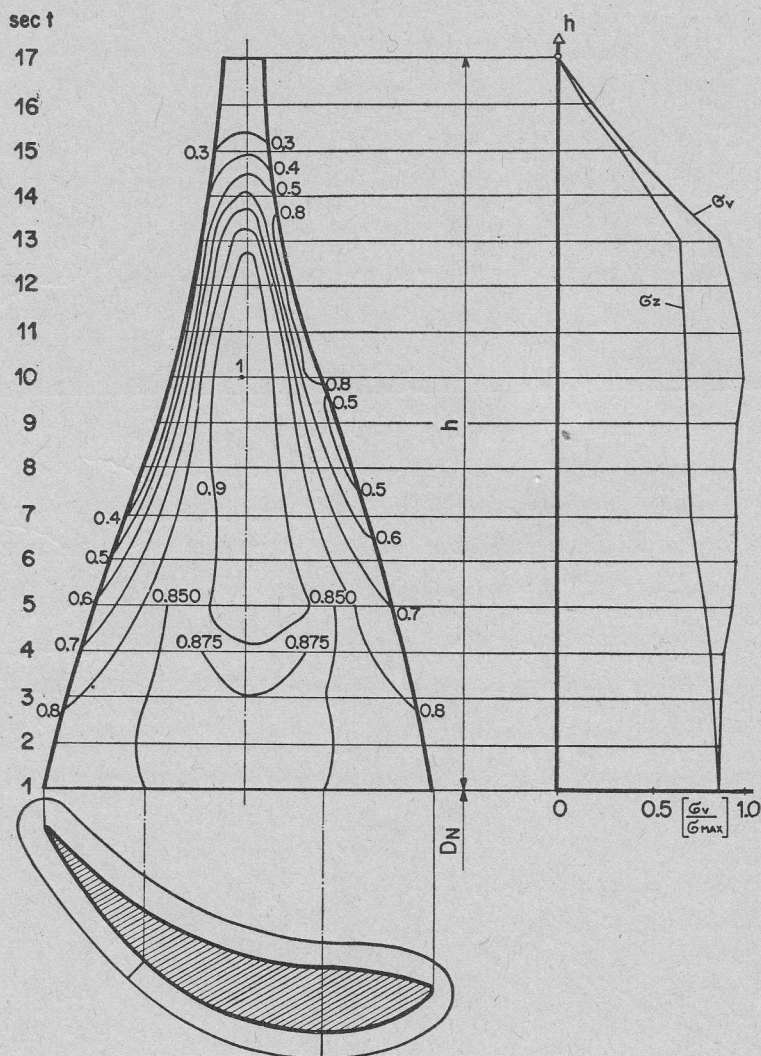


Fig. 6. Distribution of the combined stress across blade, showing the relationship between the sum σ_v of all stress components according to Fig. 4

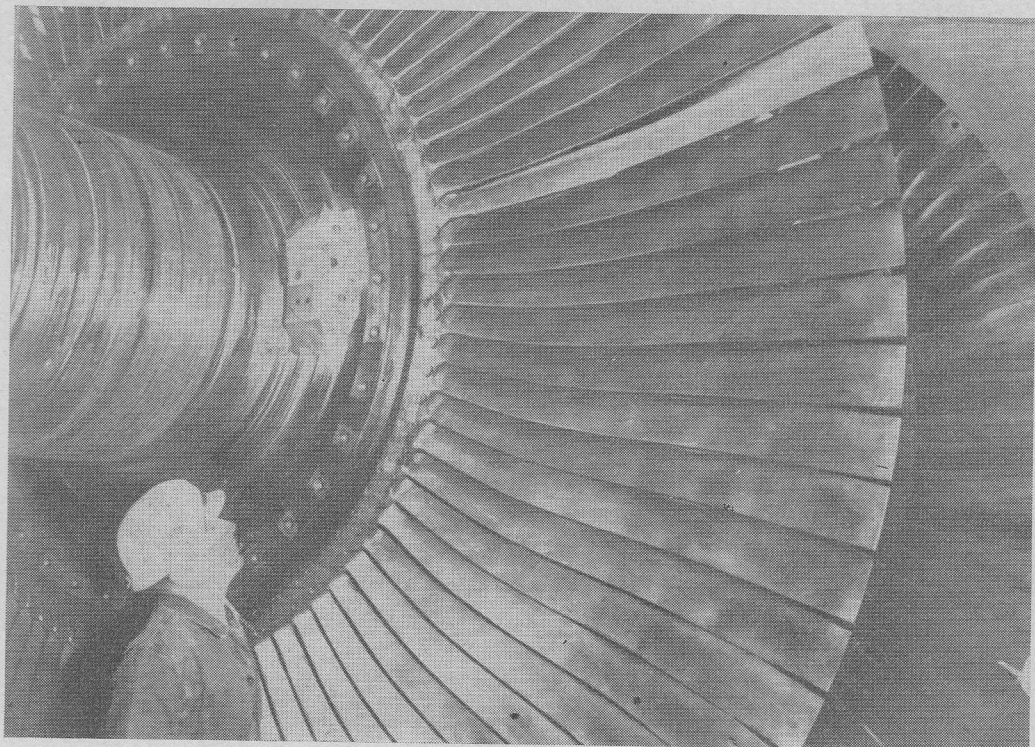


Fig. 7. Instrumentation on a 600 MW LP Rotor

We decided to take measurements on the rotating blade in order to gain as much information as possible from the data. These measurements can be made in a separate testing unit or during the overspeed test, or in the turbine itself. We used all three methods.

- For example, as set up in a 600 MW unit.
- Blade for measurement of the same unit.

It seemed almost impossible to find a satisfactory method for attaching the strain gauge to the blade in the moist atmosphere.

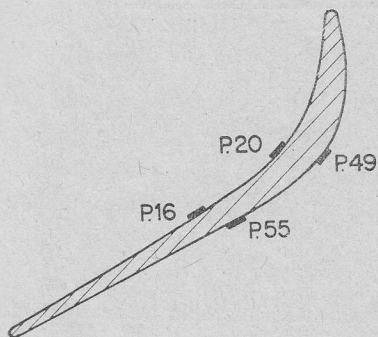


Fig. 8. Test blade on an LSB for a 600 MW turbine

We simply chose an example (Fig. 9) to illustrate the comparison between calculation and measurement. The result was very satisfactory, demonstrating that we are able to predict the stress with sufficient accuracy.

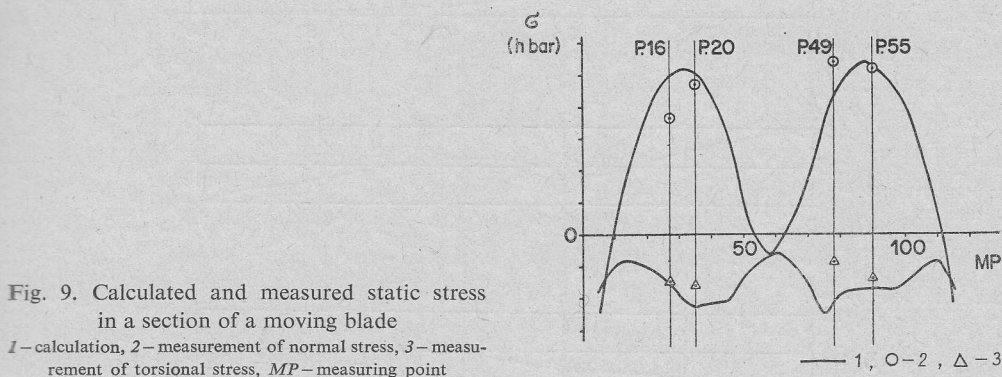


Fig. 9. Calculated and measured static stress in a section of a moving blade

1 — calculation, 2 — measurement of normal stress, 3 — measurement of torsional stress, MP — measuring point

5.2. Vibrational behaviour

The determination of dynamic loading is actually to be expected as the next point. The prediction of dynamic stress presupposes a knowledge of the exciting forces in operation. But these are still largely unknown, and the dynamic deflections of the blade can thus not be calculated in advance. It is, however, possible at any time to calculate the distribution of dynamic stress at a given standard deflection for the various forms of vibration. For this purpose the natural frequencies of the blade are first calculated.

The vibrating blade can be described by a system of ten homogeneous differential equations. The natural frequencies are the eigen values of this system of equations.

After this first step, the system of equations can be integrated from the root to the tip of the blade, so that the characteristic form of the blade for each natural frequency is obtained. As the deflection due to the dynamic steam forces is unknown, a standard deflection at the tip is assumed.

The local deformations of the blade, and thus also the local dynamic stresses, are derived from the characteristic form.

These calculations provide valuable information as to where dynamic stresses can become dangerous. Fig. 10 shows the result of such a calculation.

This does not, however, offer a satisfactory solution to the problem of dynamic stress. In order to obtain a blade which is safe in operation in spite of this, we choose its dimensions so that the natural frequencies of the blade should under no circumstances coincide with the potential frequencies of the oscillating exciting forces. Once again a suitable method of calculation to predict the natural frequencies of these complex blades had to be found. Of the many possible theories available that one which could describe the physical behaviour most accurately was selected. The most important assumptions and the boundary conditions of this theory are:

Concerning blade geometry

The same assumptions as for calculating the static stress.

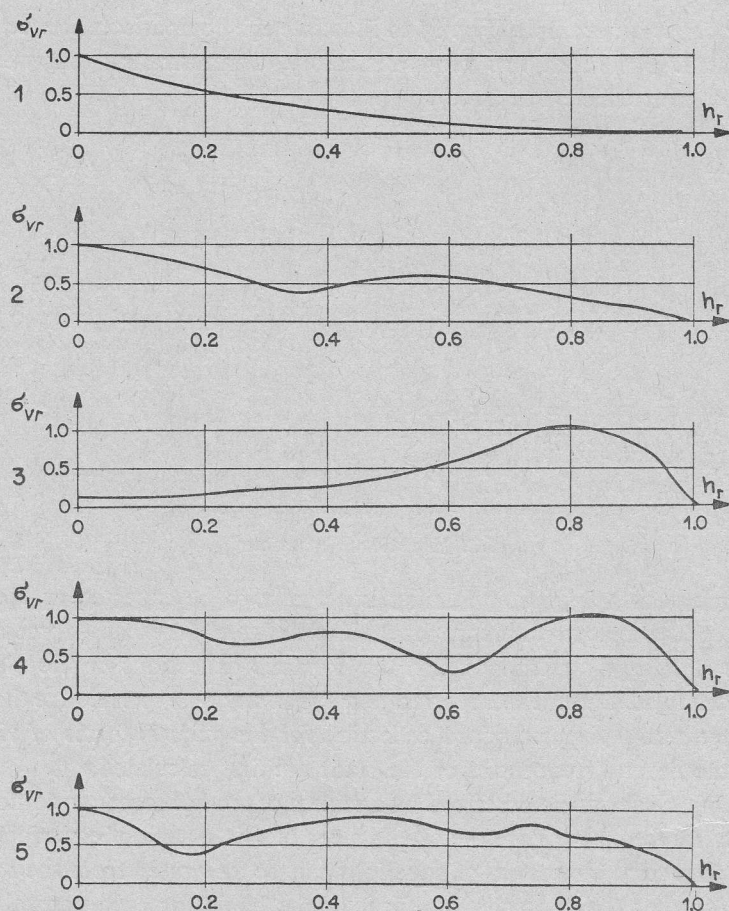


Fig. 10. Calculated maximum relative dynamic stress σ_{vr} at natural frequency 1st-5th order, at 3000 r.p.m., plotted against relative blade height
 h_r —relative blade height

Concerning deformations

- Flat blade sections remain flat even during deformation of the blade by mass and flow forces.
- The natural vibration frequencies of the blade are taken to be those of small amplitude around a point of equilibrium of the blade. Untwisting is taken into consideration during rotation.

With respect to the method

- The number of flat sections must be selected so that the characteristic vibration forms of the blade have sufficient points to define them.
- The boundary conditions for the integration are: rigid or elastic blade roots; free blade tips; no mutual binding of the blades.

– The integration of the system of equations (with the eigen values of the system) furnishes the characteristic forms and the alternating stresses of the blade.

The method was programmed and repeatedly verified experimentally for a variety of conditions. As in the case of the stress tests, the various testing possibilities were fully utilized; we concentrated our efforts on the testing of large units (300 and 600 MW). All conceivable operating conditions of the final blade were examined, e.g.

The blades were measured during start up at exhaust pressures of 0.030–0.250 bar to check damping characteristics.

– Vibrational behaviour was examined under completely different flow conditions with various loadings and exhaust pressures (0.030–0.250 bar).

– Investigations similar to those made during start up. were made at shut-off with partial and full vacuum failure.

– Measurements were made with a load rejection of 20–100% of the rated output in order to observe the effect of a sudden decrease in load.

– The blade frequencies were measured at various exhaust pressures during operation above and below normal frequencies.

– Finally, frequency measurements were made during bypass operation and while checking the valves.

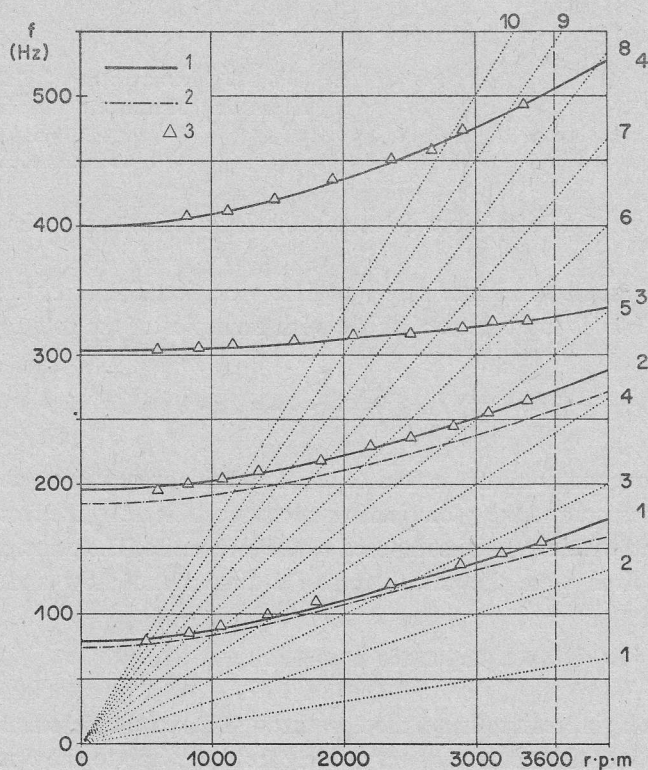


Fig. 11. Normalized natural frequencies as function of the rotating speed

1 – smoothed curve of measured values (normalized to 20°C), 2 – smoothed curve of calculated values (20°C), 3 – measured values (normalized to 20°C)

The results obtained can be summarized as follows:

- The theoretical calculations of the lower natural frequencies are in good agreement with the measured results.
- It is now possible to decide on the basis of a static vibration test alone whether or not a blade can be used. A 100% check is to be recommended.

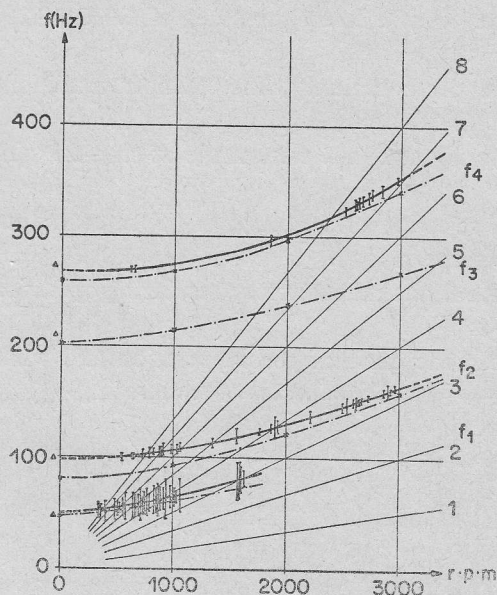


Fig. 12. Measurement of blade vibration and alternating stresses at various operating conditions

I — alternating stresses

- Only under very bad conditions do the alternating stresses attain a level which can no longer be ignored.
- In order to attain maximum safety against vibrational failure, the lower natural frequencies of the blades must not coincide with multiples of the rotation speed.

6. Aerodynamic problems

The problems encountered in the field of aerodynamics were just as complex as those in the field of mechanics, and their solution is made more difficult by the fact that the blades can only be designed within the limits imposed by the mechanics. The principal aerodynamic problems to be solved were the calculation of 3-dimensional flow and profile design.

6.1. Calculation of 3-dimensional flow

In contrast to a high pressure stage, the flow through the last stage is markedly 3-dimensional, i.e. all flow measurements vary very widely from the hub to the tips. This variation is all the more pronounced, the larger the ratio of blade height to mean diameter (h/D_M). It is well known that no major faults occur to a ratio of $h/D_M < 0.15$, if the flow conditions are calculated at the mean section. The ratio is, however, already 0.33 at the last stage.

The variation of the flow across the radius results in the various blade sections having completely different aerodynamic duties. One thus has, for example, an impulse stage with a low degree of reaction at the hub, and a reaction stage with a very high degree of reaction at the tip. A knowledge of these different duties was therefore a supposition for the subsequent development of the profile.

Here, too, we followed the familiar sequence of theory—calculation—experiment.

The basic theory applied is described in [3]. We shall give only the principal assumptions and a brief description of the method here.

The flow—is assumed to be axially symmetrical,

- is calculated at the space between the blades only.
- All flow condition points are on a single polytropic line.

The method

The equations of motion, energy, continuity and state are used to calculate the flow. The process commences with the estimated values of flow conditions (boundary conditions) at the hub and is integrated by means of the Runge-Kutta method. An elastic line is used for the first time as a function of time instead of a function of space, in calculating the radial acceleration.

The results

The results of the 3-dimensional flow calculation are the following quantities as functions of radius at all spaces between blade rows:

- pressure distribution,
- velocity triangles→aerodynamic duty,
- rotor work,
- and a figure of special interest, the exhaust kinetic energy. A skilful choice of angles can, under certain circumstances, gain up to 1% for a turbine here.

The angles must be selected so that the exhaust energy is at a minimum at the point where its application is concentrated, and the increase in exhaust energy is minimal under operating conditions diverging from this design specifications.

The theoretical findings were also verified by experiments in this case, and a number of man-years had to be invested to obtain a satisfactory agreement between calculations and measurements.

Verification consisted in numerous tests on full-scale and laboratory units. The results were finally positive, and the following are intended to summarize the comprehensive comparisons made:

Fig. 13—Tangential speed variation as calculated and measured in a laboratory unit for a blade height of 950 mm in the corresponding full-scale unit.

Fig. 14—Axial speed variation measured in a full-scale unit with a blade height of 867 mm.

In all cases agreement is so good that the duty to be assigned to the various profiles is sufficiently well known.

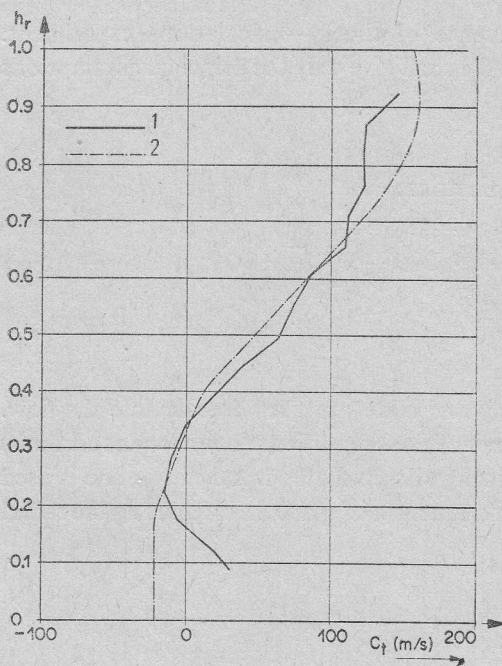


Fig. 13. Calculated and measured tangential velocity in function of the relative blade height
1—measured, 2—calculated, h_r —relative blade height

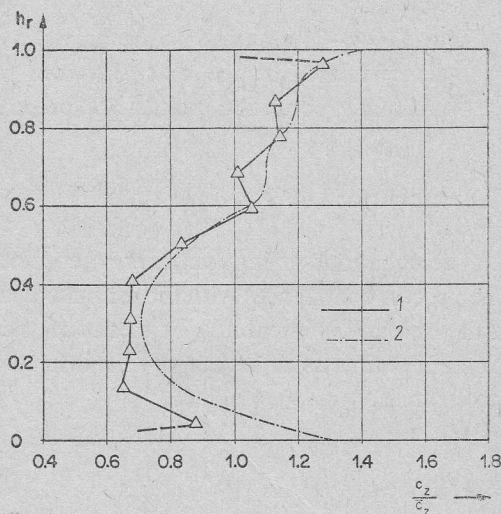


Fig. 14. Calculated and measured axial velocity as a function of the relative blade height
1—measured, 2—calculated, h_r —relative blade height

6.2. Calculation of the cascade characteristics

The corresponding profiles had now to be designed on the basis of the results of the 3-dimensional flow calculations, and their characteristics, namely efficiency and deflection, calculated. A complicating factor in designing the profiles, is the fact that the loading in the last stage varies very widely, either as a result of divergences in vacuum conditions from the ideal design point, or of changes in load. Profiles which are indifferent to the angle of attack must therefore be designed to prevent efficiency loss by the above variations.

The degree of difficulty involved in the problems to be solved increases from the hub to the tip. Conventional methods of calculating the velocity distributions, angle of attack and profile losses could be used for conventional cascades with virtually incompressible flow. The transonic cascades at the outer sections were more difficult to deal with. Characteristic for these cascades is a rapid acceleration of the flow from the purely subsonic range $Ma=0.6$ into the supersonic range at a Mach number of 1.6, the flow being deflected only slightly, if at all. Suitable methods of calculation (impulse or characteristics methods) had to be developed in this case.

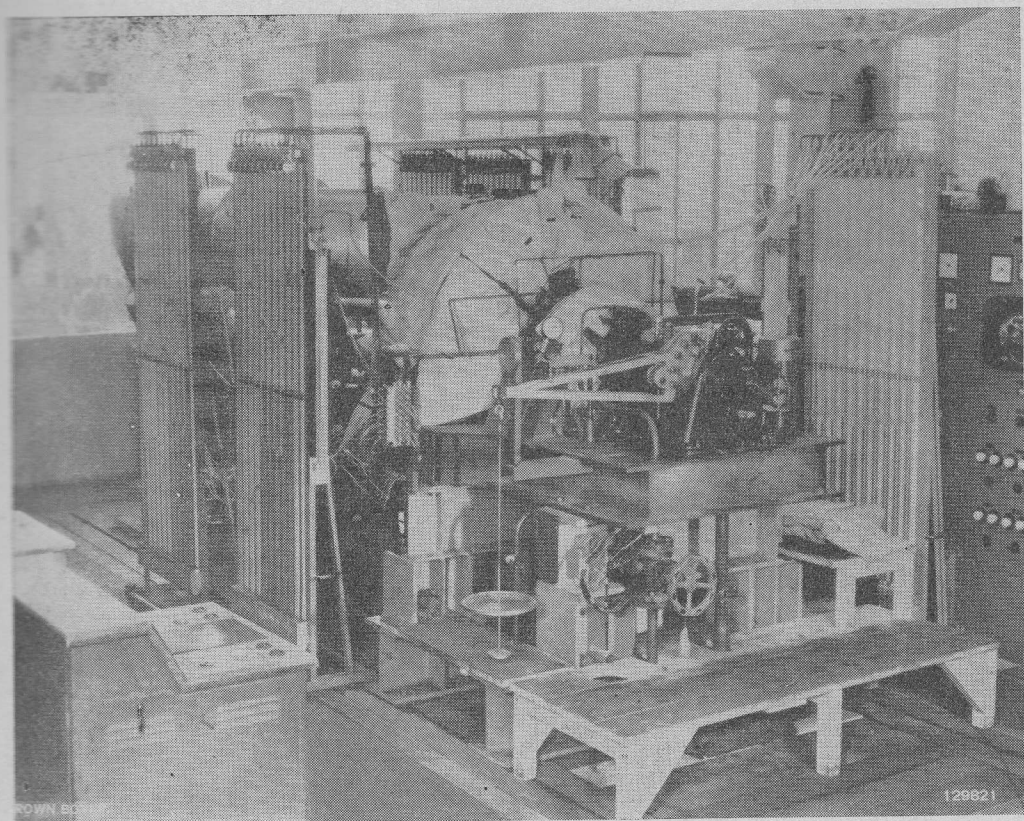


Fig. 15. Test rig "ROSA" for transonic cascade tests

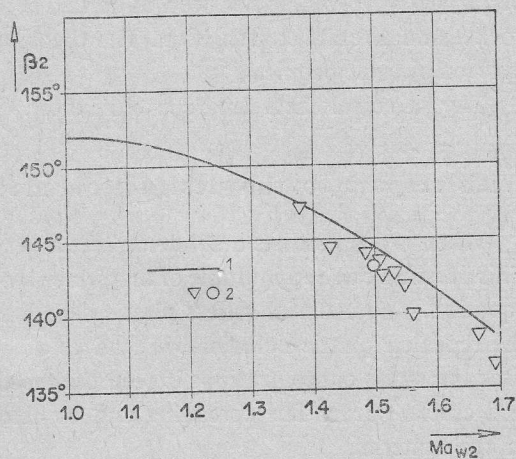


Fig. 16. Calculated and measured downstream flow angle in function of the relative Mach number

1 — calculated, 2 — measured

It became apparent that very good results could be obtained even with quite simple impulse methods [4]. We have succeeded in improving the cascade efficiency in the order of 10%. These incredible results had, of course, to be verified by experiments. These were carried out using air and steam on stationary cascades and using air on rotating cascades. Fig. 15 shows such a unit.

The rotating cascade has the great advantage that the angle of attack is very easy to alter, i.e. by changing the speed of rotation.

The most important quantities to be obtained from calculation are the downstream flow angle and the efficiency. Fig. 16 shows a comparison between calculation and measurement for the flow angle. Although agreement is very good, we believe that effective methods of characteristics must still be developed.

7. Criterion of erosion hazard

Some years ago it seemed that low pressure long rotating blades could not be produced because of the erosion hazard. This conclusion had been reached on the basis of erosion criteria which considered exhaust moisture and peripheral speed u as solely responsible for material erosion. This was taken to be proportional to the exhaust moisture and some power (usually over 3) of tip speed:

$$\dot{e} \sim (1 - x_2) u^n.$$

Thus, for a maximum permissible erosion rate $\dot{e}$, the permissible final pressure can be calculated as a function of tip speed. With a given exhaust steam pressure and a given exhaust cross-section the maximum output of a turbine is, however, fixed. Thus the output limit defined by the above criterion can be calculated (see Fig. 17).

It is obvious that the efficiency of large units would thus have become very questionable. This situation forced the turbine manufacturers to re-examine the problem of erosion. The damage caused by the impact of drops had been investigated at a very early stage in steam turbine manufactures, but systematic studies, taking better account of the aspects of physics involved, have been undertaken more recently, especially in connection with rocket and aircraft construction. The theory we used incorporated the destruction model evolved by these people.

The principal facts are repeated below.

With respect to the destruction model

When a drop strikes a solid surface, a very high impact pressure occurs momentarily. This pressure can be a multiple of that which would be derived from the energy or momentum equation. The material suffers plastic deformation as a result of this impact pressure, and the repeated plastic deformation lead to an increase in internal stresses. After a certain time the stress concentrations exceed the tensile strength of the material at certain points, and cracks occur. The spreading and combination of these cracks then lead to failure in the material.

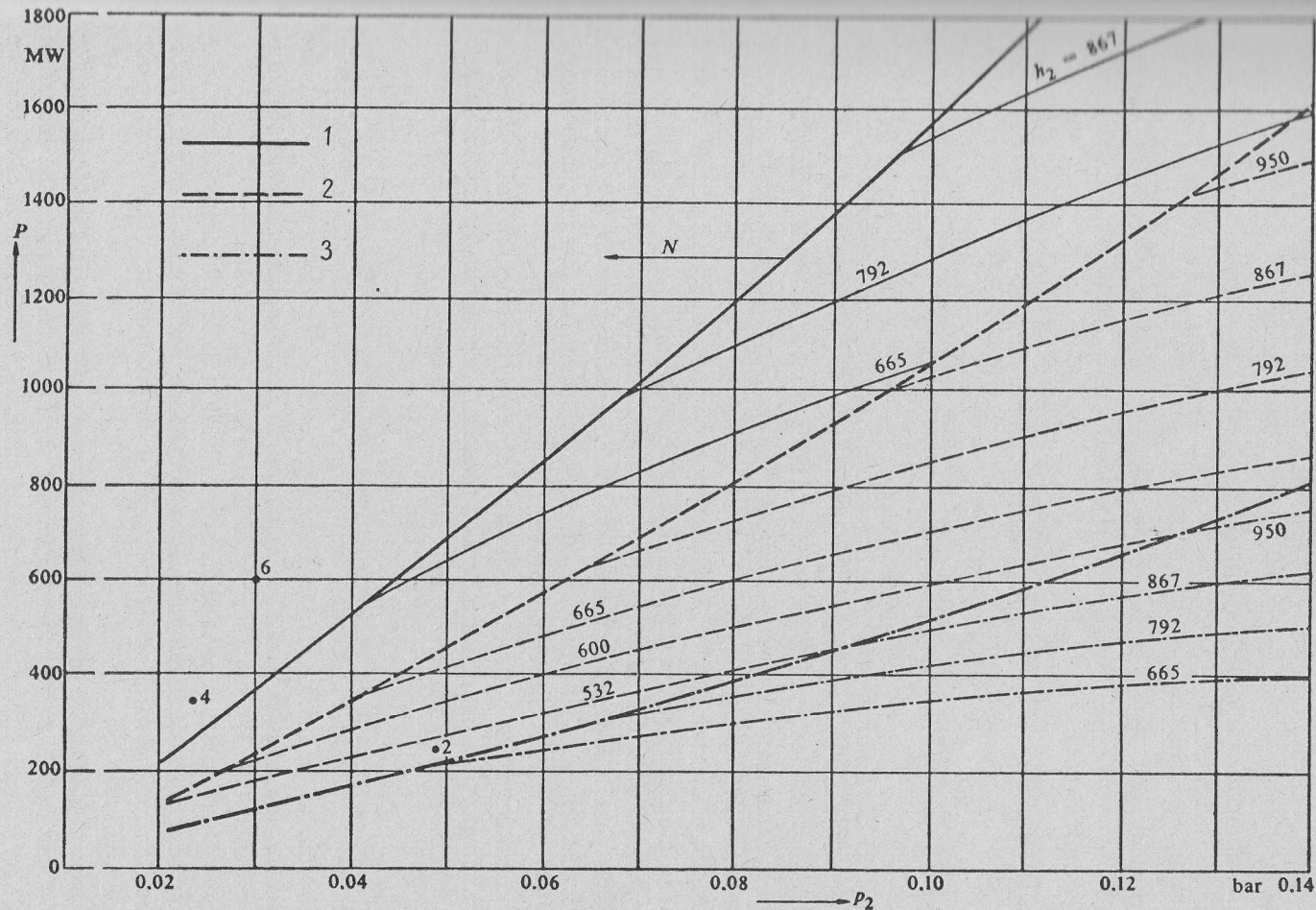


Fig. 17. Limit of maximum permissible output or minimum permissible outlet pressure calculated according to the old erosion criterion for conventional machines with reheat and $n=50$ Hz. The curves show the necessary number of exhausts and the blade length of the last moving row

P —output, p_2 —pressure after last moving row, h_2 —blade height of last moving row (in mm), 1—six exhausts, 2—four exhausts, 3—two exhausts, ●—an actual example of each type of machine. The numbers refer to the number of exhausts, N —no permitted

With respect to the influencing factors

From what has already been said, the following influencing factors are present for a given material:

- total volume of water,
- impingement velocity,
- impingement frequency,
- drop radius.

As a dependent factor

We define the erosion rate as material loss per unit of time and surface, i.e. erosion rate

$$\dot{e} = \frac{\Delta V_M}{\Delta S_M}.$$

In order to calculate this, something must be known about the origin, motion and effect of the drops which cause the erosion.

The drops responsible for erosion are the large ones formed from water which has been deposited on the stationary blades and is brought by sweeping forces into the wake area of the trailing edges.

The water drops increase in size there until they are dragged off by the current of steam and dispersed into smaller drops, as illustrated in Fig. 18.

The movement of these drops must now be traced. For this purpose, their size after dispersion must first of all be known, and this is determined by using the Weber criterion, which states that the ratio between aerodynamic stress and surface tension cannot exceed a certain level. This enables the representative drop size to be defined.

The differential equation of motion of the drops can easily be established using Newton's law. Integration provides the relationship between the displacement s , pressure p , steam velocity c_D and the ratio of water velocity to steam velocity c_W/c_D , i.e.

$$s = f(p, c_D, c_W/c_D).$$

In order to keep erosion to a minimum, the water should flow against the rotor blades in the same direction as the steam, i.e. c_W/c_D should be around 1. This is the more likely to be achieved, the larger p , c_D and s are. There is an optimum combination of the various parameters. The influence of flow route s is particularly interesting (Fig. 19).

The erosion effect must now be calculated on the basis of the known drop sizes and velocities. On the basis of our destruction model we would state that the erosion rate is proportional to the pressure level maintained locally and the impact frequency of the drops on one and the same point on the surface.

The next point is to define the maximum permissible rate $\dot{e}_{\max}$. This is dependent, for a required operating life, on very many operating conditions and design data. Instead of conducting very expensive tests to determine the influence of all these detailed factors, we solved the problem by taking a typical, proven unit as a reference plant in the knowledge that its erosion rate is within permissible limits.

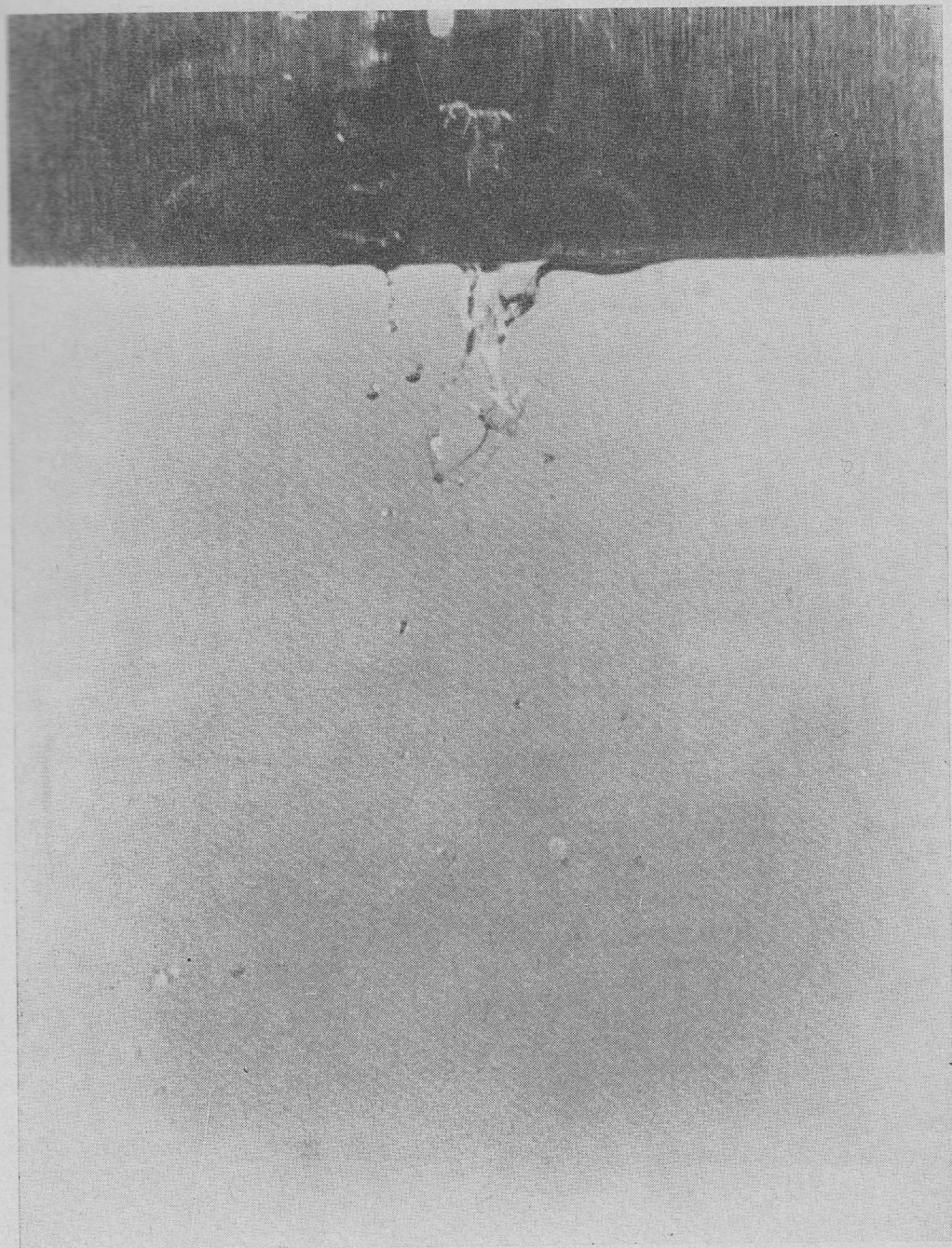


Fig. 18. Detachment and break up of drops from the fixed blade trailing edges in a cascade test with air and water. Air velocity 70 m/s

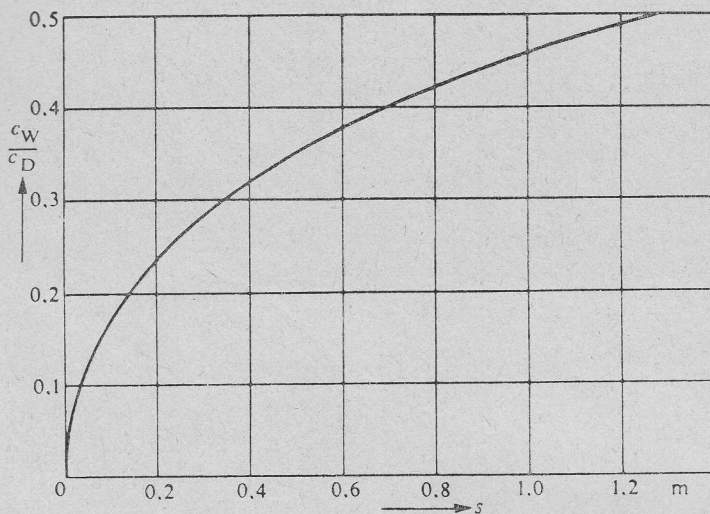


Fig. 19. Ratio of the absolute velocities of water and steam
 s —flow path of water drop, c_D —absolute steam velocity at the instant of break up, c_W —absolute water velocity after a distance s

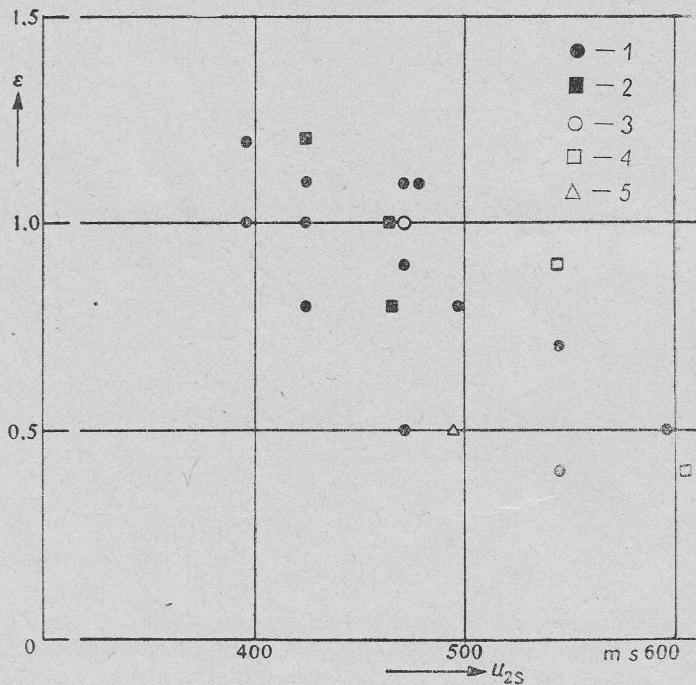


Fig. 20. Standard erosion rate for various machines using the new erosion criterion
 $1-n=50$ Hz (in service), $2-n=60$ Hz (in service), $3-n=50$ Hz (at design stage), $4-n=60$ Hz (at design stage), $5-n=30$ Hz (at design stage), u_{2s} —peripheral velocity at outlet of moving blade tip, ε —standard erosion rate

We define the non-dimensional erosion rate as the ratio between the erosion rate of a unit being assessed and that of a reference unit.

The non-dimensional erosion rates calculated in this manner for a number of units which have already been in operation for some time are shown in Fig. 20. A figure of >1 indicates that the unit in question is more endangered than the reference unit, i.e. values of <1 should be aimed for.

Many subsequent systematic comparisons have confirmed the accuracy of the tendency displayed. For example.

Fig. 21—shows a blade with $h=600$ mm, $u=424$, $t=20,000$ h, $1-x_2=10.7$,

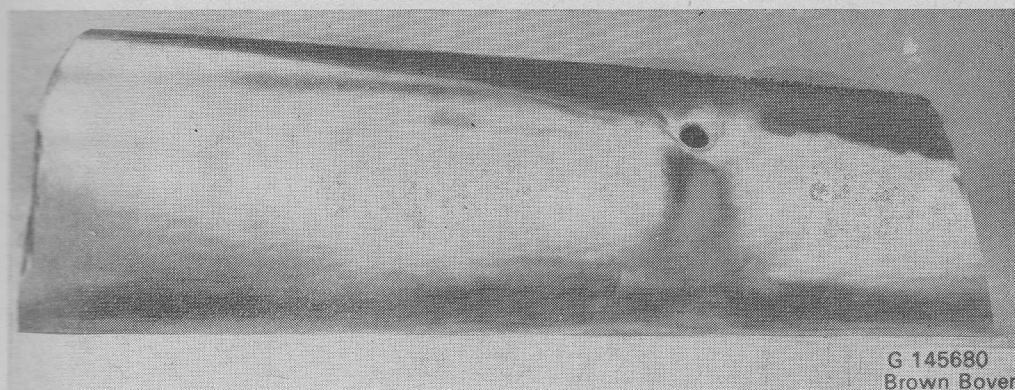


Fig. 21. Eroded tip of a moving blade. Service life=20 000 hours (approx.). Moisture content after moving blade=10.7%. Peripheral velocity at tip=424 m/s

Fig. 22—shows a blade with $h=1000$ mm, $u=596$, $t=7,000$ h, $1-x_2=9.0$.

On the basis of the old criteria, the erosion rate of the 1000 mm blade would have to be 3 times as great, and on the basis of the new criteria only half as much.

This is in good agreement with our observations, as the blade has in the meantime exceeded 30,000 hrs without suffering a dangerous degree of erosion.

Thus we come to the following important conclusion. The large units with very high peripheral speeds are by no means more endangered than the small units operating at low speeds.

The conclusion is plausible, in the first place due to the purely physical fact that the impingement velocity and the volume of water engaged in erosion decline, and in the second place because a number of deliberate measures have been taken to reduce erosion in large units. In contrast to earlier efforts, these measures are all aimed against the cause of erosion. In principle, it is a matter of keeping the number and size of the drops which are whipped off as small as possible.

Although we are well aware that our method still does not cover all processes satisfactorily, we do have very reliable means of assessing the erosion hazard in advance and planning appropriate countermeasures, in order to build the last stage which is reliable also from this point of view.

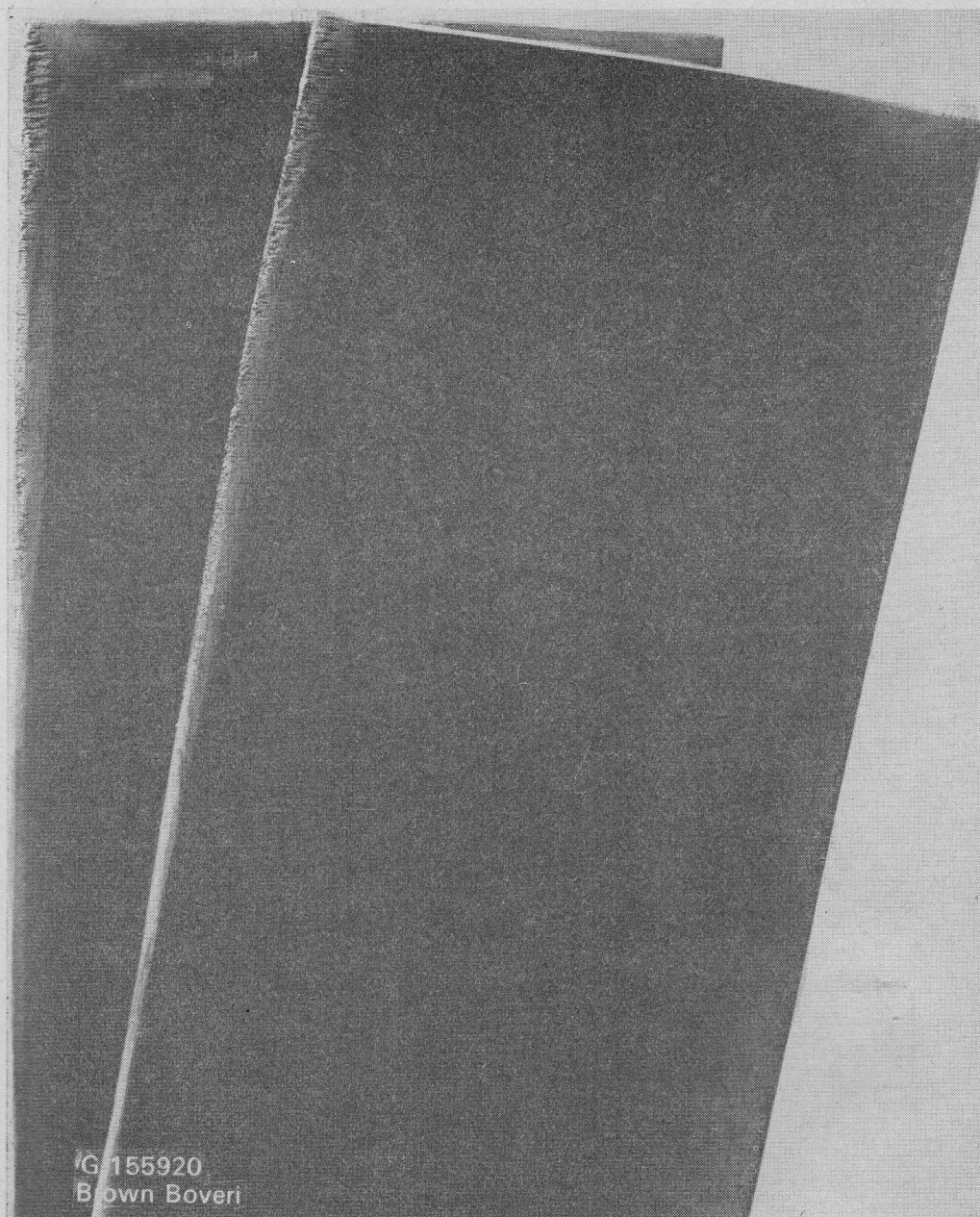


Fig. 22. Eroded tip of a moving blade. Service life=7000 hours (approx.). Moisture content after moving blade 9%. Peripheral velocity at tip=596 m/s

8. Concluding comments

There are a great many interesting problems to be solved in present-day turbine manufacture. Of these the most fascinating is the development of the last stage, because it involves a wide variety of engineering fields, and because in certain areas, one is already confronted with the limits of what is attainable. The state of the art is now at a very high level as a result of the use of the most modern means available, such as computers, complex test installations, precision manufacturing methods, etc. This development must, however, continue, and is now very clearly focused on the search for lighter and yet stronger materials. If we were not to succeed in this, the stresses imposed on the rotor would attain an impermissible level. Parallel with this search, existing materials and methods will and must be constantly refined, in order to achieve steady improvements in operating safety, efficiency and price.

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Problemy rozwoju konstrukcji łopatek wirnikowych ostatnich stopni dużych turbin parowych

Streszczenie

Początek artykułu poświęcono przedstawieniu znaczenia ostatnich stopni dla uzyskania maksymalnej mocy wyjściowej dużych turbin parowych.

Przegląd problemów jakie należy rozwiązać w dziedzinie mechaniki, aerodynamiki, metalurgii i technologii wskazuje wyraźnie, że opracowanie dużej łopatki wirującej wymaga spełnienia niezwykle wysokich wymagań.

Pierwsza część artykułu dotyczy głównie zagadnień mechaniki. Nakreślono w skrócie metodę zastosowaną do obliczania złożonego stanu naprężeń. Praktyczną przydatność przyjętej teorii potwierdzono porównując wyniki obliczeń z wynikami pomiarów. Ten sam sposób zastosowano w odniesieniu do charakterystyk dynamicznych. Szczególną uwagę zwrócono tu na wyczerpujące badania doświadczalne; badania takie pozwalają na przewidywanie zachowania się łopatki w różnorodnych warunkach roboczych.

W drugiej części wymieniono wymagania określone z punktu widzenia aerodynamiki. Przedstawiono sposób obliczania różnych stanów przepływu. Również tu, jak poprzednio, ukazano znaczenie prób dla potwierdzenia teorii.

Na zakończenie podano proste kryterium oceny zagrożenia erozją w stopniu niskoprężnym. Pokazano przede wszystkim, że można bezpiecznie konstruować bardzo duże stopnie końcowe, byle tylko przedsięwzięte zostały odpowiednie środki ostrożności.

Проблемы связанные с проектированием рабочих лопаток для последних ступеней больших паровых турбин

Резюме

Начало статьи посвящено представлению значения последних ступеней для реализации максимальной мощности в больших паровых турбинах.

Просмотр проблем, какие необходимо решить в областях механики, аэродинамики, металлургии и технологии показывает отчетливо, что для разработки большой рабочей лопатки необходимо исполнить чрезвычайно высокие требования.

Первая часть статьи в основном касается проблем механики. Дан эскиз метода, применяемого для расчета сложного состояния напряжений. Практическая пригодность принятой теории подтверждается сравнением расчетных результатов с экспериментальными. Такой же способ принят относительно динамических характеристик. Особое внимание обращено здесь на исчерпывающие экспериментальные исследования. Такие исследования позволяют в настоящее время предвидеть осцилляцию для разнообразных рабочих условий.

Во второй части статьи перечислены требования, определенные с точки зрения аэродинамики. Представлен способ расчета различных состояний течения. Также здесь, как и раньше, указывается на значение эксперимента для подтверждения теории.

В заключении представлен простой критерий оценки эрозионной угрозы в ступени НД. Прежде всего показано, что можно безопасно конструировать очень большие последние ступени, если только предприняты соответствующие меры предосторожности.

Для измерения направления движения использованы металлические шары, покрытые краской. Исследования в турбине показали, что крупные капли, срывающиеся с рабочих лопаток, обладают такой кинетической энергией, которой достаточно для смыва краски с шаров. В результате этого на шаре остается размытое пятно, по которому можно определить направление движения крупных капель. Мелкие капли, которые несутся с паровым потоком, не обладают такой энергией, поэтому от их воздействия следа на шаре не остается.

Для измерения направления движения крупнодисперсной влаги использовалась шаровая гребенка (рис. 15) с шагом 30 мм. Исследования показали, что на верхнем шаре, который был удален от вершины лопатки по радиусу на 45 мм, краска целиком сохранялась. Следовательно влага этого шара не достигала. Следующие 4 шара имели отчетливые размытые пятна. На пятом шаре пятно от капель не резко, но центр пятна удается определить довольно точно. На следующем шаре краска слегка размыта и центр пятна определить невозможно. На остальных шарах, которые расположены на нижней половине лопатки, краска осталась неповрежденной. Следует отметить, что шары с пятнами, которые были в потоке пара, имеют матовую поверхность, в то время как верхний шар, который не был в потоке, имеет блестящую поверхность. Исследования на различных режимах показали, что описанная картина повторяется на всех режимах, но число шаров с отчетливым пятном меняется — оно уменьшается с уменьшением числа оборотов ротора, скорости пара и степени влажности.

На рис. 16 показаны результаты измерений. Видно, что в нижней половине лопатки крупнодисперсной влаги практически нет. В верхней половине капли движутся под углом $\alpha_{2k} = 150^\circ$ и $\gamma_{2k} = 20^\circ$, при этом рассогласование паровой фазы по углу α_2 достигает 50° , по углу γ_2 до 30° .

Следует отметить, что рассогласование по углу γ_2 увеличивается по мере удаления капель от выходных кромок рабочего колеса.

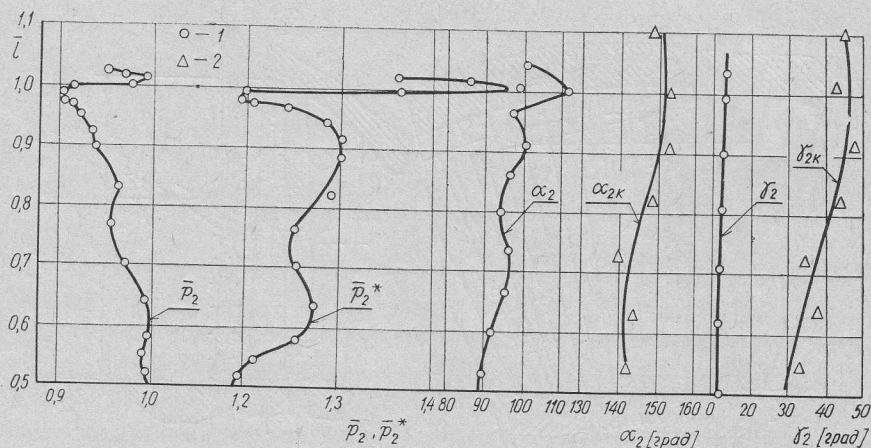


Рис. 16. Распределение газодинамических параметров потока пара и капель крупнодисперсной влаги за рабочим колесом по высоте лопатки

1 — паровая среда, 2 — капли