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PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

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# A Novel Design of Turbine Blading of Extreme Length

The limiting power output of a steam turbine per 1 exhaust to the condenser depends on the exhaust area of the final stage (Fig. 1):

$$A = D\pi l\tau, \tag{1}$$

where $D$ – the mean diameter, $l$ – the length of a rotor blade, $\tau$ – the edge thickness factor at the blade exit.

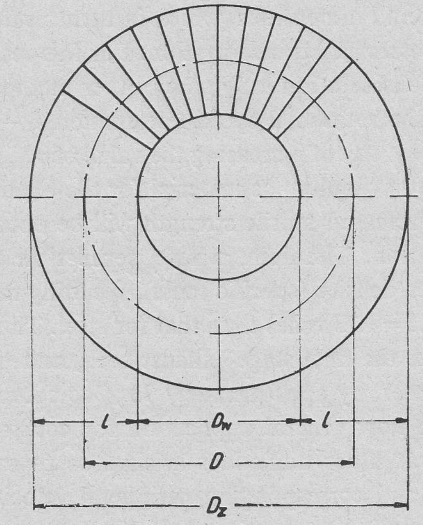


Fig. 1

Both the above used dimensions $D$ and $l$ determine also the value of the tensile stress at the blade root:

$$\sigma = k\sigma_{cyl} = \tfrac{1}{2}kl\rho D\omega^2, \tag{2}$$

where $k$ – the ratio of tensile stress values for identical cross-sections at the roots of two blades: the actual blade, twisted and tapered, with variable cross-sections, diminishing

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from the root to the tip, and another blade, basically a virtual one, cylindrical in shape, that means of constant cross-sections, equal to that at the root, $\rho$ – density of the blade material, $\omega$ – angular velocity of rotation.

By comparing the both formulae one obtains [1]:

$$A=\frac{2\pi\tau\sigma}{k\rho\omega^2}=f\left(\frac{\sigma}{\rho}, \omega, k\right) \tag{3}$$

indicating that the value of the final stage exhaust area attainable depends on 3 parameters: the ratio of the permissible tensile stress to density, understood as a material constant, the square of the angular velocity of rotation and finally the said coefficient ($k$) expressing the effective reduction of mass of the rotating blade at larger radii, as compared to the cylindrical blade. Thus exist also but 3 possibilities of increasing the final stage exhaust area which are:

– reduction of the rotational speed of the turbine, in principle by a half,

– selection of an adequate blade material with higher value of $\sigma_{perm}/\rho$,

– selection of an adequate constructional model of a blade characterized by a lower value of the coefficient $k$.

The well known method of decreasing the rotational speed of a turbine leads compulsorily to the increase of its overall dimensions, which in turn results in more difficult and expensive manufacturing processes and transportation. For this reason the method, however very effective, comes into consideration only when other methods of increasing the final stage exhaust area – with the rotational speed left unaltered – fail.

The next method, acting by way of increasing the value of $\sigma_{perm}/\rho$ as a material constant, is related actually to the use of titanium as material for the blading. However, a possibility that new kinds of steel of increased tensile strength will be obtained in the future is also to be taken into consideration. Titanium, whose permissible tensile stress is identical as that for steel, has a lower density (specific mass), resulting in the permissible value of $\sigma_{perm}/\rho$ – ratio higher $7.8/4.5=1.73$ times than that for steel. Thus titanium blading would allow in principle to increase the final stage exhaust area, and thereby the power output per 1 exhaust to the condenser, by approximately 73% as compared to steel. Obviously, titanium did not find up to now general application for constructing blades of extreme length. This seems to hint at other difficulties arising, so to say, inevitably when new, extreme overall dimensions and corresponding peripheral velocities of final stages are to be taken into account. For blading made of steel the limiting attainable outlet area of the last stage is about $A=8.5$ m² or under the assumption $l/D=1/2.7$, the corresponding dimensions become $D=2.85$ m and $l=1.055$ m while the peripheral velocity at the blade tip attains the very high value $u_e=614$ m/s, and requires undoubtedly the application of effective measures to protect blades against disastrous vibrations as well as against destructive influence of erosion. Corresponding values for titanium blades are: $A=8.5\times1.73=14.7$ m², $D=3.75$ m, $l=1.38$ m, while the peripheral velocity at the blade tip approaches the extreme high value $u_e=806$ m/s! Obviously, under these circumstances, conventional means of blade protection against erosion may become insufficient.

A third method remains, consisting in selecting a new constructional model of a blade with lower value of the coefficient $k$. No doubt, this requires better approximation of the blade model to the shape of uniform strength, no matter what sequence of cross-sectional profiles is used, the profiles being selected mainly in view of flow requirements, particularly in view of spatial flow structure through a blade passage. While in case of a conventional blade the reduction of the cross-sectional areas [2] – root to tip section – may reach values as high as $F_i/F_e=10$, the corresponding coefficient $k$ decreases – as calculations show – down to the value 0.37 – this resulting from a shift of the centre of gravity in the direction of smaller radii, i.e. closer to the root cross-section.

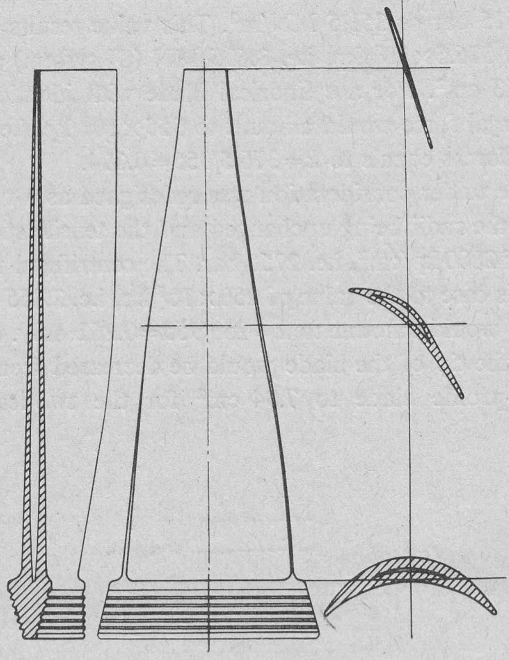

Fig. 2

The object under consideration can be achieved by utilizing a new concept of the blade design consisting in constructing it from two separate profiled sheets (leafs), one of them representing the concave side of the blade and another the convex one, the leaves joined together adequately along the whole length by means of a permanent joint, e.g. by electron welding. Leaf thickness of about 1 to 1.5 mm is sufficient at the blade tip where the tensile stress diminishes in fact to zero. Going down along the blade, in the direction of the root, this thickness may remain constant as far as to the cross-section at which the tensile stress attains the permissible value assumed. Below this cross-section the thickness of the profiled sheets increases according to uniform strength conditions. As a result a hollow space of characteristic shape arises inside the blade, widening at first gradually from the tip down to the root, and then narrowing to a little slit at the root itself (Fig. 2).

Owing to such a type of blade characterized by wall thicknesses chosen according to uniform strength conditions, the centrifugal force of the blade acting on the rotor perimeter as well as the trensile stress in the root of the blade itself decrease as related to the conventional blade design of equal length, while for the load acting on the rotor perimeter and the tensile stress in the blade root being unchanged, the permissible radial dimension, i.e. the blade length considerably increases, to the advantage of the attainable last stage exhaust area.

To illustrate the new concept there have been made calculations for an assumed last stage blading having as dimensions follows: $D_i = 1600$ mm, $D_e = 3440$ mm, $D_m = 2520$ mm, $l = 920$ mm and, correspondingly, $u_e = 538$ m/s and $l/D = 1/2.74$. With the ratio of extreme cross-sectional areas of the blade $F_i/F_e = 8.32$ the maximum tensile stress caused only by the centrifugal force of the blade reaches its maximum value at the root: $\sigma_{max} = 3574$ kp/cm², i.e. 3515 bar or 351,5 MN/m². This value results from the values of the blade centrifugal force $376.5 \times 10^3$ kp, i.e. 3.695 MN (SI system) and the cross-sectional area at the root 105.43 cm². For a cylindrical blade with identical cross-sectional area at the root the centrifugal force would amount to $955 \times 10^3$ kp, i.e. 9.365 MN. The ratio of the two centrifugal forces equals to $k = 376.5/955 = 0.394$.

Thereafter the blade under consideration was redesigned as a two-leaf blade, with the external profiles at all the radii being unchanged and the tensile stress at the root section reduced to the value of 2800 kp/cm², i.e. 2755 bar. The centrifugal force due to the blade would be reduced at this case to the value of $250 \times 10^3$ kp, i.e. 2.455 MN, and the characteristic design parameter would amount to $k = 250/955 = 0.262$ only. At the same time the cross-sectional area at the tip of the blade would be decreased from the value 12.69 cm² for the reference, full-profile blade to 7.94 cm² for the two-leaf one, the reduction amounting to 37.5%.

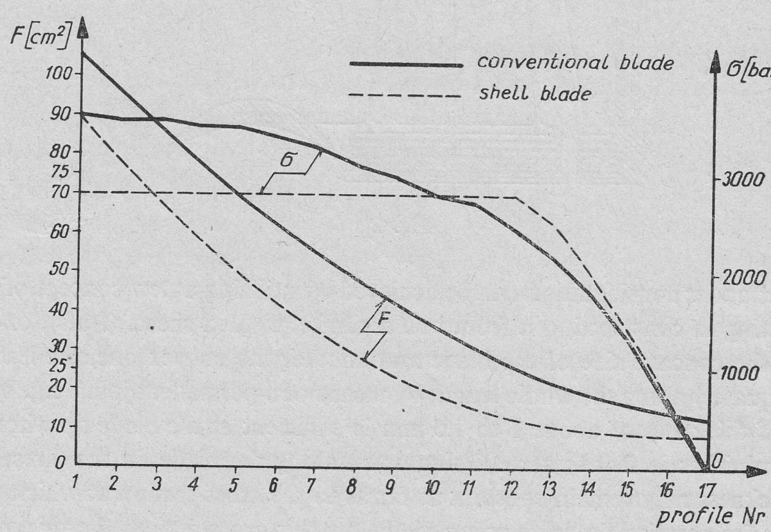


Fig. 3

The variation of the cross-sectional areas and tensile stresses along the blade length for both the full-profile and two-leaf blade structures is shown on the diagram (Fig. 3).

Applying the value $k = 0.262$ to the formula for the final stage exhaust area one obtains a $0.394/0.262 = 1.502$ times increase of the exhaust area as compared to the area attainable for the reference, full-profile blade with $k = 0.394$; so the increase amounts to 50.2%.

For the blades under consideration:

$$(lD)_{2-l}=1.502\cdot(lD)_{\mathrm{conv}}=1.502\times 0.92\times 2.52=3.495\ \mathrm{m}^2.$$

The assumption of an identical value $l/D=1/2.74$ for both blades gives the permissible length of the two-leaf blade equal to:

$$l_{2-l}=\sqrt{(lD)_{2-l}\frac{l}{D}}=\sqrt{\frac{3.495}{2.74}}=1.129\mathrm{m}$$

or in other words the blade is 22.8 percent longer than the reference one. The remaining characteristic quantities are specified in Table 1.

Table 1

| Blade | Conventional | Two-leaf | Rise [%] |
|---|---|---|---|
| $D_i$ [mm] | 1600 | 1966 | 22.8 |
| $D$ [mm] | 2520 | 3095 | 22.8 |
| $D_e$ [mm] | 3440 | 4224 | 22.8 |
| $l$ [mm] | 920 | 1129 | 22.8 |
| $D/l$ | 2.74 | 2.74 | – |
| $u_e$[m/s] | 538 | 664 | 22.8 |
| $lD$ [m²] | 2.325 | 3.495 | 50.2 |

The last mentioned rise of $lD$ refers also to the attainable power output per 1 exhaust of the turbine to the condenser. These results deserve attention.

The two-leaf blade of the novel type is also distinguished – as compared with a conventional blade – by a favourable rise, from the point of view of operation, of the free vibration frequency, which is mainly due to a considerable reduction of the rotating mass. Similarly, bending stresses due to the eccentric action of the centrifugal force can also be reduced considerably to the advantage of the blade strength and general stress conditions.

The novel type of the blade makes it possible to select the profile chord-to-pitch ratio more freely, allowing thereby for higher energy conversion efficiency in the blade passage. This is not the case with the conventional blading where, due to attempts to reduce rotating masses at larger radii as much as possible, blade profile chords are being rather shortened, whereas the pitch increases in proportion to the radius.

The above fact renders the use of larger $l/D$-ratios, up to the value of about 1/2.5. Consequently, increase of the exhaust cross-sectional area of the last stage, indispensable for increasing the unit output of a turbine, can be accomplished without the need to increase the peripheral velocity at the blade root and the rotor perimeter as well as the flow Mach number.

Noteworthy seems also more liberty in selection of blade profiles, particularly in the application of "thick" profiles and those with rounded inlet edges. As for a conventional design this would be accompanied necessarily by an increase of the rotating mass, with simultaneous reduction of permissible blade length and exhaust area of the last stage.

The possibility of obtaining a well-rounded inlet edge is particularly important from the point of view of erosion fighting, as it is known that the normal component of the liquid phase relative velocity which is responsible for the collision pressure and erosion of the blade surface [3], decreases as the inlet edge radius increases.

The assumption of the two-leaf blade design does not impose any restrictions on the material used, and either titanium or its alloys can be used also. From the point of view of the tensile stress in the blade root the superposition of the two effects, the application of titanium and the two-leaf design, would make it possible to produce a very large area of the final stage exhaust, even as large as $A=8.5\times1.73\times1.37=2.37\times8.5=20.16\ \mathrm{m}^2$, which means 2.37 times more than actually attainable. The coresponding blade of extreme length coming up to about 1600 mm for normal speed conditions would then permit to increase the output of a single-shaft steam turbine, with 6 exhausts to the condenser, depending on the vacuum used, to the extreme high value of 2000 to 2500 MW.

Obviously, it is not the tensile stress in the root-section alone determining the permissible extreme length of a blade. From the point of view of erosion hazard rising peripheral velocities could be met either by restriction of steam wetness attained at the end of expansion or by application of more effective protective measures against erosion in the design of the blading itself. The first way could consist for instance in shifting the interstage reheat towards a sufficiently lower pressure or in application of internal heating of the expanding stream by means of guide vanes of the hollow shell type [4] fed with steam extracted from an earlier stage. The second way, covered by a patent on the two-leaf blade design, makes use of the internal blade cavity for sucking water droplets impinging the blade surface into the cavity, through a system of small holes drilled in the blade wall, with following centrifuging them out of the blade and collecting again in a circumferential channel in the turbine cylinder.

Nevertheless many important problems in high speed flow as well as in dynamics and strength of materials are to be solved to make this rapid increase of the blade length possible. On the other hand however the advantages of this novel type of blading may be utilized in the development of much shorter blades in the range of 1200 - 1300 mm, as required actually in progressive turbine technology.

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## Nowa konstrukcja łopatek turbinowych ekstremalnej długości

### Streszczenie

Długie łopatki turbin parowych konstruuje się zasadniczo jako łopatki pełne o przekroju malejącym od stopy do wierzchołka. Dla danego materiału i danej prędkości obrotowej turbiny, dopuszczalna długość łopatki zależy od współczynnika wyrażającego rzeczywistą redukcję masy i związanej z nią siły odśrodkowej wirującej łopatki, w porównaniu z łopatką cylindryczną o tym samym przekroju w stopie. Redukcja ta jest tym skuteczniejsza im bardziej przybliża się model łopatki do postaci równej wytrzymałości. Cel ten można osiągnąć opierając się na nowej koncepcji konstrukcji łopatki, która polega na wykonaniu łopatki jako złożonej z dwóch piór, z których jedno tworzy wypukłą, a drugie wklęsłą powierzchnię łopatki. Ponieważ naprężenie rozciągające przy wierzchołku łopatki maleje do zera, można przyjąć tam bardzo małą grubość obu piór, idąc w kierunku stopy, na przykład 1 do 1,5 mm. Grubość ta popozostaje stała aż do przekroju, w którym naprężenie rozciągające osiąga dopuszczalną wartość, a następnie przyrasta zgodnie z warunkiem zachowania stałego naprężenia. W rezultacie wewnątrz łopatki tworzy się komora o charakterystycznym kształcie, początkowo rozszerzająca się stopniowo od wierzchołka w kierunku stopy, a następnie malejąca do rozmiarów wąskiej szczeliny przy samej stopie. W takim modelu siła odśrodkowa łopatki działająca na obwodzie wirnika oraz naprężenie rozciągające przy stopie łopatki maleją w porównaniu z pełną łopatką o konwencjonalnej konstrukcji; przy takim samym zaś naprężeniu rozciągającym w stopie łopatki dopuszczalna długość w kierunku promieniowym rośnie, co w efekcie pozwala powiększyć powierzchnię wylotową ostatniego stopnia. Nowa konstrukcja długich łopatek daje również inne korzyści, przede wszystkim wyższe częstości drgań własnych oraz większą swobodę w doborze stosunku cięciwy profilu do podziałki przy większych promieniach oraz samych profili łopatki, w szczególności z zaokrąglonymi krawędziami na wlocie.

## Новая конструкция рабочих турбинных лопаток экстремальной длины

### Резюме

Длинные рабочие лопатки паровых турбин конструируются, в основном, как полные с убывающей поверхностью поперечного сечения от корня к вершине. Для данного материала и данной скорости вращения турбины допускаемая длина лопатки зависит от коэффициента, определяющего действительную редукцию массы и связанной с ней центробежной силы относительно цилиндрической лопатки такого же самого поперечного сечения у корня. Эта редукция может быть еще сильнее в случае лучшего приближения модели лопатки к форме лопатки постоянной прочности. Эта концепция сводится к выполнению лопатки из двух оболочек, одна из которых образует выпуклую, в вторая вогнутую стороны лопатки, соответственно связанных с собой вдоль кромок. Учитывая, что растягивающие напряжения у вершины лопатки нулевые, можно принять очень малую толщину оболочки у вершины, примерно 1 до 1,5 мм. Эта толщина остается постоянной на расстоянии от вершины до такого поперечного сечения лопатки, в котором растягивающие напряжения достигают допускаемое значение, а потом увеличиваются согласно условию постоянства напряжений и прочности. В результате внутри лопатки образуется камера характеристической формы, сначала постепенно расширяющаяся от вершины по направлению к корню, а потом суживающаяся до размеров узкой щели у самого корня. В такой модели центробежная сила лопатки, действующая на ободе ротора, а также растягивающие напряжения у корня лопатки, уменьшаются по сравнению с соответствующими величинами для полной лопатки конвенциональной конструкции. Для таких же самых значений растягивающих напряжений у корня лопатки, допускаемая длина лопатки в радиальном направлении растет, что способствует возрастанию выходной поверхности последней ступени. Новая конструкция длинных лопаток приносит также и другие пользы, как напр. в первую очередь повышение собственных частот колебаний, потом, большую свободу в подборе отношения длины хорды профиля к шагу при больших радиусах, а также в подборе самых лопаточных профилей, в особенности с закругленными выходными кромками.