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Poland*

Aerodynamical and Constructional Problems in the Design of Centripetal Double-Flow Inlet Stages for Large Condensing Turbines

Double-flow "butterfly" casings are the most characteristic units used in practically all steam turbines of large output not only as the low-pressure part but more frequently now as the middle- and even the high-pressure part. Such casings are used also in turbines for driving feed pumps, compressors (e.g. for NLG), blast-furnace blowers as well as in top load steam turbines of great compactness.

The 90° direction change of flow from centripetal to axial is the common feature of these constructional solutions. This direction change takes place in the inlet chests before the first stage of multi-stage axial blading. This produces non-uniform inflow along the radius and also in the first axial blade row which causes a serious decrease in efficiency. In addition the present constructions of the steam inlet chests in a double-flow axial steam path are not favourable for introducing the steam governing systems.

Governing and shut-off systems are required before the double-flow part both in wet-steam turbosets and in most of the industrial turbines (Fig. 1, 2). With the increasing output of conventional sets and the use of a double-flow middle-pressure part taking the reheated steam, such a need arose also in these turbines.

The possibility of obtaining the direction change from radial to axial using radial inflow stages and the requirements of governing and shut-off systems were the reasons for beginning the investigations on centripetal inlet stages with adjustable guide vanes. However, for practical applications the advancement of this conception for the inlet turbine stage — as in every new design of a flow path — is dictated by following factors:

1) efficiency,

2) cost of production of blade system as well as of auxiliary elements (e.g. the inlet part of the casing, diaphragms, rotor etc.),

3) adaptability to different load conditions,

4) reliability.

Thus, not only the flow problems but first of all the constructional problems play the decisive role here.

The use of centripetal inlet stage before the double-flow axial blading promises obvious advantages both in the efficiency and construction:

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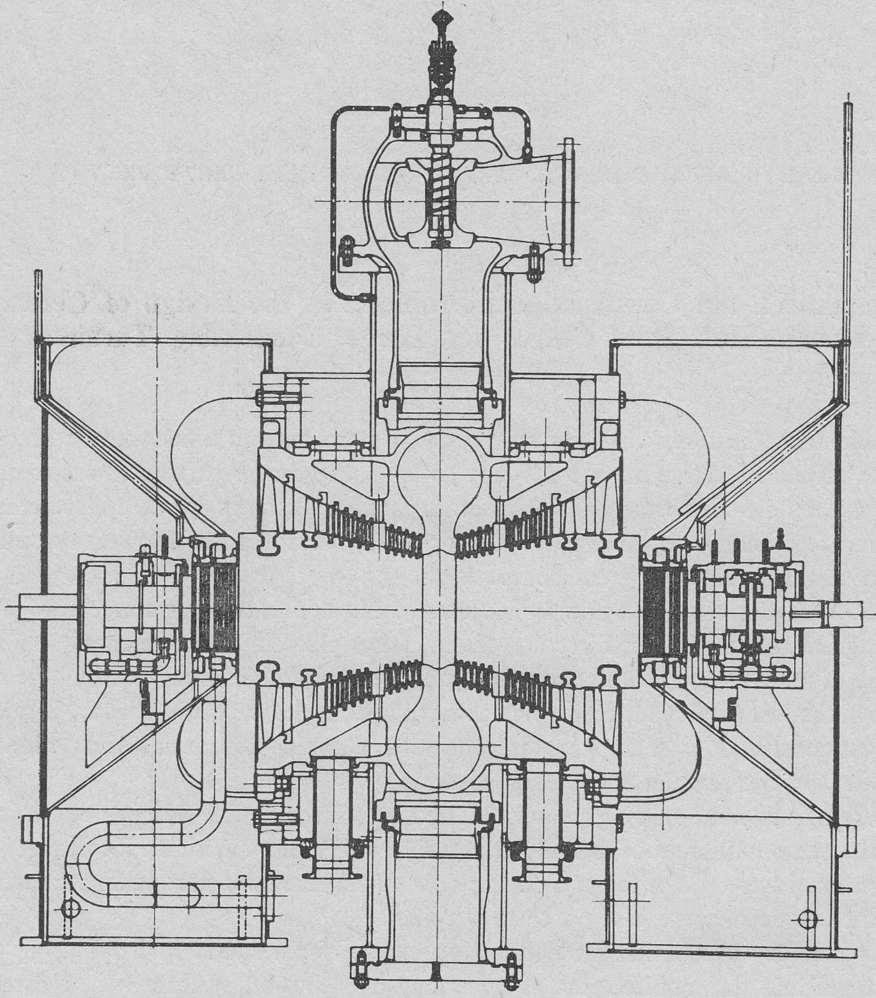

Fig. 1. Industrial "butterfly" type turbine with choking governor (Siemens)

– Firstly it makes possible a more compact shaping of the inlet part, reducing the number of turbine stages. This results in shortening of the rotor length, reduction in the mass and in the dimensions of this turbine part, thus lowering the initial cost of the turboset.

– Secondly it makes possible an increase in the efficiency of the first radial-axial stages and consequently of the whole turbine, thus reducing the operating costs of the turbosets.

These problems were already mentioned during the Conference in Dresden (1966) [1] and during the II Conference on Steam Turbines of Great Output in Gdańsk [2].

The basic problems of flow and efficiency when developing the double-flow centripetal stages, resulting in practical application, will be discussed in turn.

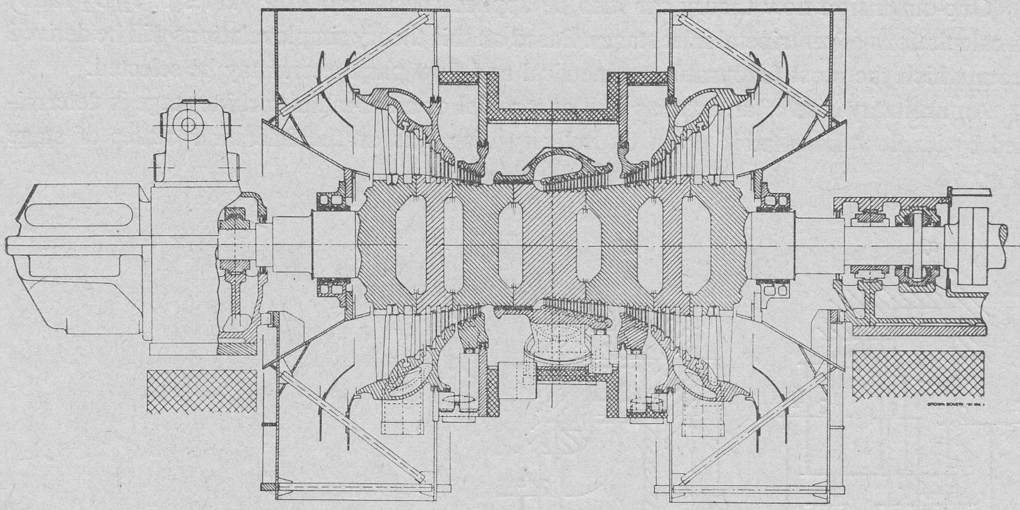

Fig. 2. Variant of "butterfly" type turbine designed to drive the compressor for NLG (Brown Boveri, 150 MW)

The common feature of modern highly efficient steam turbines, both the chamber and drum type, is the sealing of rotating blade rows against the stator and of guide blade rows against the turbine shaft or drum. Only with large volume flows where clearance losses are tolerable (e.g. the last stages of low pressure parts of condensing turbines), are labyrinth sealings not used but free-standing blades with sharpened tips are applied. These stages are generally designed in the same way in the chamber as well as drum turbines namely with a strongly increasing degree of reaction from root to tip of the blade.

Important differences in the kinematics of the flow through the stages of chamber and drum turbines occur only in short blade stages ($l/D<0.15$). In this range of values of l/D, in the both types of turbines, blades of constant profile (cylindrical blades) are used usually.

One-dimension flow theory is generally applied in calculations for stages with cylindrical blades. Non-dimensional parameters, related to circumferencial velocity u_2 and not as usualy in centripetal turbines to $u_{\max}=u_1$, are for practical reasons used in what follows for calculations of the blade system as well as for characterizing the turbine stages. These are:

– meridional velocity factor $$\varphi=\frac{c_{2m}}{u_2}, \tag{1}$$

– specific power factor $$\lambda=2e_u/u_2^2, \tag{2}$$

– enthalpy drop factor $$\psi=2\Delta i_s/u_2^2, \tag{3}$$

– degree of dynamic reaction $$R=\frac{\Delta i_s''}{\Delta i_s'+\Delta i_s''}. \tag{4}$$

One-dimension flow theory can also be applied (because of low values of l_2/D_2 ratio) in calculations of centripetal inlet stages. Based on this theory simple relations will be derived from which the most favourable geometrical and flow parameters may be selected.

An arbitrary case of an axial as well as a radial or axial-radial turbine stage is determined by its flow kinematics given by velocity diagrams at the inlet and outlet of stage (Fig. 3).

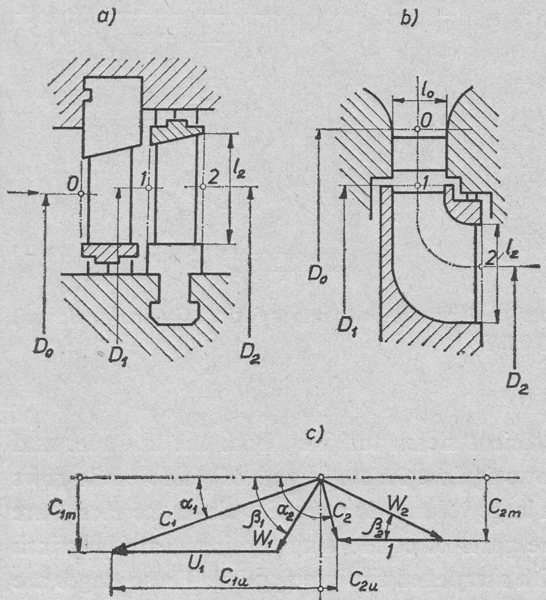

Fig. 3. Explanation of symbols and of considered cross-sections in the turbine stages: a) axial stage, b) radial stage, c) corrected velocity diagrams for an arbitrary turbine stage

The most favourable form thereof results from the correction of all velocities with respect to the circumferential speed at the outlet of the stage u_2, e.g. $C_1 = c_1/u_2$. The outlet velocity diagram is determined, in such a presentation, by two angles: trailing jet angle β_2 in a rotating reference frame and by α_1 in the absolute system. It makes possible to present the meridional velocity factor φ by the relationship

$$\varphi = C_{2m} = \frac{1}{\operatorname{ctg}\beta_2 + \operatorname{ctg}\alpha_2}\,. \tag{5}$$

The non dimensional velocities u_2 and c_2 in the outlet velocity diagram can be presented also as functions of these two angles. The same thing can be done with the velocities in the control inlet velocity diagram taking additionally into account that

$$C_{1m} = K_1 C_{2m}\,. \tag{6}$$

The factor K_1 depends first of all on the ratio of outlet areas and density in considered control sections 1 and 2.

For an arbitrary type of turbine stage the specific power factor λ (2) takes the form of

$$\lambda = 2C_{2m}(K_1 U_1 \operatorname{ctg}\alpha_1 - \operatorname{ctg}\alpha_2) \tag{7}$$

or, after introducing the relation (8) between the angles α_1 and β_1

$$\operatorname{ctg}\alpha_1=\operatorname{ctg}\beta_1+\frac{U_1}{K_1C_{2m}}, \tag{8}$$

it takes the form

$$\lambda=2\left[U_1^2+C_{2m}(U_1K_1\operatorname{ctg}\beta_1-\operatorname{ctg}\alpha_2)\right]. \tag{7a}$$

The relation (7a) and (8) is shown in diagram (Fig. 4) accepting the angle β_1 as the independent variable; the other values are considered as parameters. A case of the pure – axial stage ($U_1=1.0$) with degree of the rotor dynamic reaction of $R=0.5$ being presented by dashed line corresponding to a condition $\beta_1=180-\alpha_2$.

Assuming β_1 in the following as an independent variable the enthalpy drop factor ψ and the degree of dynamic reaction R can be expressed by relations:

$$\psi=C_{2m}^2\left[K_1^2\frac{1-\eta'}{\eta'}(1+\operatorname{ctg}^2\beta_1)+\frac{1}{\eta''}(1+\operatorname{ctg}^2\beta_2)-\right.$$
$$\left.-K_0^2(1+\operatorname{ctg}^2\alpha_2)\right]+\frac{2K_1C_{2m}}{\eta'}U_1\operatorname{ctg}\beta_1+\frac{1+\eta'}{\eta'}U_1^2-1, \tag{9}$$

$$R=\frac{1}{1+\dfrac{\eta''}{\eta'}\dfrac{K_1^2[1+(\operatorname{ctg}\beta_1+U_1(K_1C_{2m})^2-\eta' K_0^2(1+\operatorname{ctg}^2\alpha_2)}{1+\operatorname{ctg}^2\beta_2-\eta''\left[K_1^2(1+\operatorname{ctg}^2\beta_1)-\dfrac{U_1^2-1}{C_{2m}^2}\right]}}. \tag{10}$$

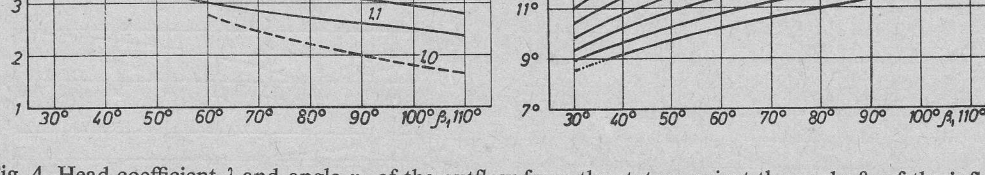

Fig. 4. Head coefficient λ and angle α_1 of the outflow from the stator against the angle β_1 of the inflow to the rotor and the corrected speed U_1

The factor $K_0=C_0/C_{2m}$ is to take into account cases in which either the direction of flow differs essentially from the axial direction, or the stages are the first (inlet), for which $C_0\neq C_2$.

In the relations (9) and (10) appear the efficiencies η' and η'' of stator and rotor respectively. These efficiencies are mainly functions of jet angles at inlet and outlet of the blade rows (i.e. α_0 and α_1 or β_1 and β_2), geometrical parameters of blade rows as t/s, l/s and l/D, and Mach and Reynolds numbers. To determine these efficiencies the following functions

(11) and (12) were assumed based upon different experiences, namely of Filippow [3], Traupel [4], Cordes [5] and our own results

$$\eta' = 1 - \xi' = 1 - (\xi_0' k_{\alpha_1} k_{Re_1} k_{M_1} + \Delta\xi' + \xi_f'), \tag{11}$$

$$\eta'' = 1 - \xi'' = 1 - (\xi_0'' k_{\beta_2} k_{Re_2} k_{M_2} k_{\Delta s''/s''} + \Delta\xi'' + \xi_f''), \tag{12}$$

$$\xi_f = \frac{1}{3}\left(\frac{l}{D}\right)^2. \tag{12a}$$

In relations (11) and (12):

– the values of basic factors ξ_0' and ξ_0'' were defined for trailing jet angles α_1 or $\beta_2 = 20°$, optimum inflow jet angle, optimum pitch-to-chord ratio t/s at $M=0.8$ and $Re \geqslant 6.5 \cdot 10^5$; these values depend on the jet deviation angle within the blade row and on the blade aspect ratio l/s,

– the correction factors k_{α_1}, k_{Re_1}, k_{M_1} and k_{β_2}, k_{Re_2}, k_{M_2} take into account the effect of the trailing jet angle, and the Reynolds and Mach numbers,

– the correction factor $k_{\Delta s''/s''}$ refers only to rotating blade rows of the radial-axial type: it accounts for the effect of lengthened blade chord of radial-axial blade row on flow losses as compared to that of basic profile of axial type blading, and is a function of $U_1 = D_1/D_2$,

– $\Delta\xi'$ and $\Delta\xi''$ account for the rise of tip losses within the stator or rotor due to axial conicity of outer walls of the blade rows,

– ξ_f includes the effect on losses within cylindrical blade rows of varying flow conditions along the radius; the well-known relation (12a) was chosen in this case (for centripetal inlet stages of course $\xi_f = 0$).

The character of the variation of losses ξ_0' and ξ_0'' and of correction factors is shown on diagrams (Fig. 5).

The relations (9) and (10) can be used to develop a whole family of standard turbine stages which can be applied to different conditions, e.g. in the high or the middle pressure part

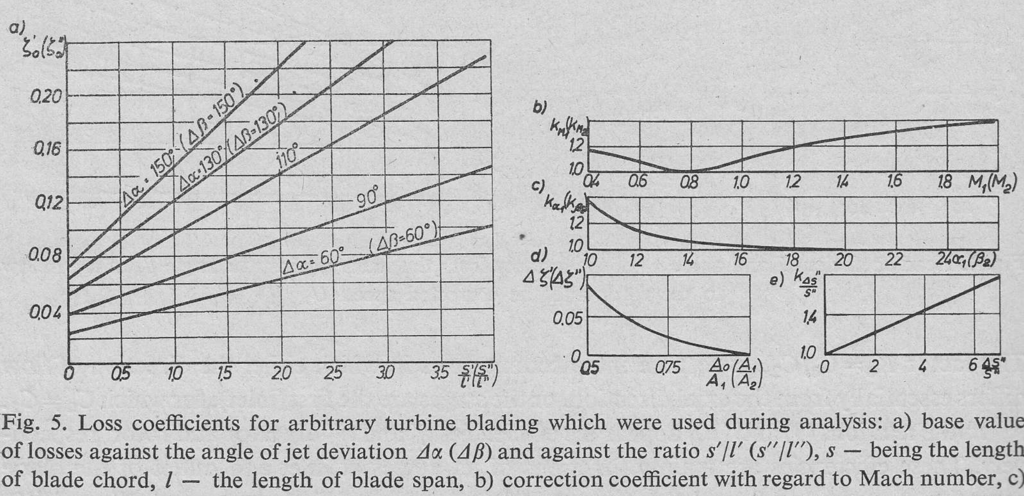

Fig. 5. Loss coefficients for arbitrary turbine blading which were used during analysis: a) base value of losses against the angle of jet deviation $\Delta\alpha$ ($\Delta\beta$) and against the ratio s'/l' (s''/l''), s — being the length of blade chord, l — the length of blade span, b) correction coefficient with regard to Mach number, c) correction coefficient with regard to trailing jet angle, d) additional loss due to axial conicity of the blade tip, e) correction parameter taking in account the aspect ratio of the blade leading edges

as well as the inlet stages in the low-pressure part. To find the optimum solution for a flow path it is possible to compare single stages as well as homogenous blading of different kinematics with regard to efficiency, processed enthalpy drop, etc. There are many ways of proceeding in this case.

One of them will now be briefly described, this one being useful in comparing the efficiency of the axial drum-type blading with $R=0.5$, to that of the mixed-type blading with one centripetal inlet stage and the remained stages of the drum-type.

Assuming some characteristic data for the axial drum-type blading with $R=0.5$ (e.g. main dimensions and meridional velocity factor) the characteristic curves (9), (10) and (8) of standard stages can be easily determined for some range of application (range of pressures and densities).

The blade efficiency of the stage η_u defined by the relation:

$$\eta_u=\frac{\lambda}{\psi+C_0^2-C_2^2}, \tag{13}$$

according to the above takes, by introducing the equation (8), the form:

$$\eta_u=\frac{\lambda}{\psi+C_{2m}^2[K_0^2-(1+\mathrm{ctg}^2\alpha_2)]}. \tag{13a}$$

Detailed comparison of radial centripetal stages with axial reaction stage ($R=0.5$) with regard to efficiency has been carried out on an the example of double-flow low pressure blading, as shown schematically in Fig. 6a. The characteristic geometrical values of all three stages of blading have been presented in Table 1.

Knowing the values of characteristic factors φ, ψ, λ or η_u of separate stages of blading such solutions can be chosen for which the same values of the factor φ are also valid for axial stages. These solutions can be selected from the whole family of standard centripetal inlet stages. The values of the factor ψ of these selected stages should fulfil the simple equations:

$$\left.\begin{aligned} \psi_R&=\psi_{\mathrm{ax\,I}},\\ u_{2\,\mathrm{II}}^2\psi_{R\,\mathrm{II}}&=\psi_{\mathrm{ax\,I}}u_{2\,\mathrm{I}}^2+\psi_{\mathrm{ax\,II}}u_{2\,\mathrm{II}}^2,\\ u_{2\,\mathrm{III}}^2\psi_{R\,\mathrm{III}}&=\psi_{\mathrm{ax\,I}}u_{2\,\mathrm{I}}^2+\psi_{\mathrm{ax\,II}}u_{2\,\mathrm{II}}^2+\psi_{\mathrm{ax\,III}}u_{2\,\mathrm{III}}^2 \end{aligned}\right\}. \tag{14}$$

Table 1

	First stage		Second stage		Third stage	
	stator	rotor	stator	rotor	stator	rotor
Pitch to chord ratio t/s	0.93	0.86	0.98	0.861	1.13	0.826
Chord to span ratio s/l	0.4227	0.4030	0.3019	0.3538	0.1823	0.3069
Span to pitch diameter ratio l/D	0.0562	0.0685	0.0769	0.0933	0.1121	0.1330
Ratio of cross-sectional areas A_0/A_1 or A_1/A_2	0.9043	0.8088	0.8689	0.7982	0.7327	0.9391

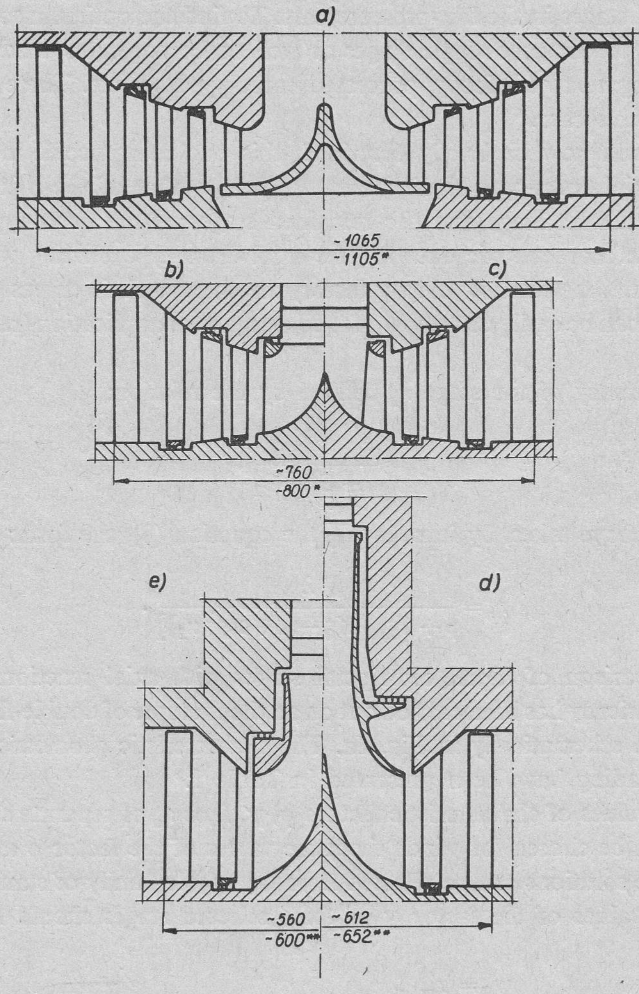

Fig. 6. Comparison of different variants of the double-flow blading of low pressure steam turbine: a) axial blading of reaction type ($R=0.5$), b) mixed type blading containing the centripetal inlet stage with meridional blades, c) variant as b) but in the radial stage only the splitter blades are remained, d) mixed type blading containing the centripetal inlet stage as a substitute for two stages in each flow path of a double-flow turbine – meridional blades, e) variant as d) but with blades bent forward, $\beta_1 \approx 36°$

* with steam extraction for the first and second axial stage, ** with steam extraction for the third axial stage

Fulfilment of these conditions means changing in turn one, two and the whole axial blading into a properly selected radial stage. Two first cases in two variants each have been presented schematically in Fig. 6b, 6c, 6d, 6e. These figures demonstrate at the same time the possibilities of serious shortening of flow path of steam turbines of great output.

Figures 7a and 7b present the characteristic curves of one of the sets of standard centripetal stages ($U_1 > 1$) which is characterised by the same values of flow angles $\beta_2 = 16°$ and $\alpha_2 = 110°$ giving together $\varphi = 0.32$. This value of φ is commonly used in the first drum stages in the low pressure part of steam turbines. For this set it has been assumed additionally that $C_{1m}=C_{2m}$ and $C_0=C_{2m}$, which also means $K_1=1$, $K_0=1$.

The reduced circumferencial speed $U_1=D_1/D_2$ plays the role of a parameter. The case $U_1=1$ and $\alpha_1=\beta_2=16°$ represents an axial stage of the drum-type (Fig. 7a).

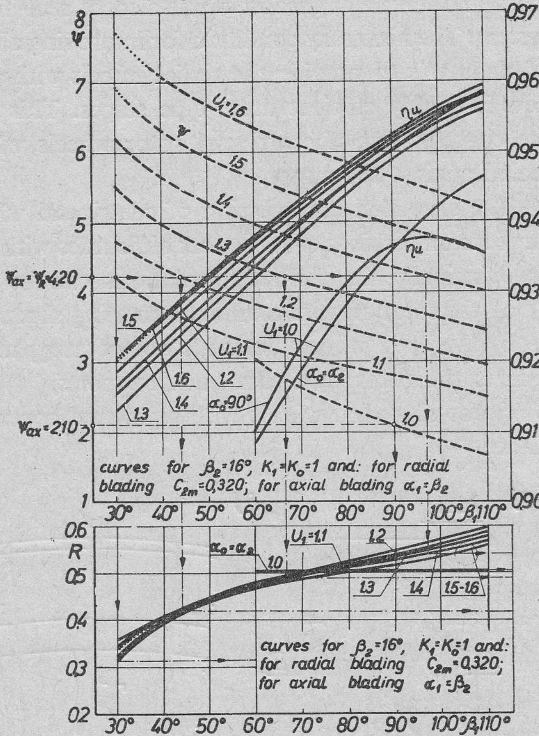

Fig. 7. Characteristic curves for a series of centripetal stages and for comparative axial stages of reaction type: a) enthalpy drop coefficient ψ and circumferential efficiency η_u against the angle β_1 of the inflow and the corrected speed U_1, b) degree of dynamic reaction R of the rotor against the angle β_1 of the inflow and the corrected speed U_1

In Fig. 7a there are for axial stages one common characteristic curve for ψ and two curves for η_u, namely one for the first stage ($\alpha_0=90°$) and the second curve for the second and third stage together as for these two last stages $C_0=C_2$ and $\alpha_0=\alpha_2$. The efficiency of the blade rows of these stages was computed basing on the geometrical data given in Table 1. From the diagram (Fig. 7a) it can be seen that the efficiency η_u and the enthalpy drop factors ψ of the drum-type stages II and III overlap closely one another though the blade length for these stages is obviously different. This is the result of strong rise of tip losses in the III stage because of large axial conicity of outer walls of the blade row.

The centripetal radial stages are characterized by a high efficiency η_u, which can be seen in Fig. 7a. It is especially true for the solutions with meridional blades ($\beta_1=90°$) and for the forward bent blades ($\beta_1>90°$).

The rotors with the backward bent blades ($\beta_1<90°$) demonstrate also high efficiency. The valuable feature of the backward bent blades is a high value of the factor ψ which gives the possibility of making the rotor wheels with much smaller diameter than for $\beta_1=90°$ (see Fig. 6e).

The differences in efficiencies η_u, favourable for radial stage, result first of all from higher efficiencies of radial guide wheels than of axial ones. It becomes more obvious with minor values of the parameter $U_1 = D_1/D_2$.

Summing up, centripetal inlet stages can be applied to all "butterfly" type of flow path in steam turbines, promising not only the possibilities of considerably reducing the mass and length of turbosets but also of real improvement of the efficiency of the blade system. The necessary condition for improving also the internal efficiency of the mixed type of blading is to provide the radial wheels with an equally effective sealing system as used now in highly efficient drum-type stages. This condition determines from the beginning the construction of centripetal rotor wheels which must be designed with shroud and sealed against the stator.

The shrouded construction of centripetal wheels especially for low pressure parts of large steam turbines creates great difficulties with regard to stress and vibration.

Principally the following solutions can be considered:

a) the solution with a uniform rotating shroud but made of fiber reinforced composits,

b) the open wheels but with non-rotating moving and following covers e.g. directed againts the rotor by steam-static supports,

c) the shrouded wheels with segmental covers whose centrifugal forces are carried by blades.

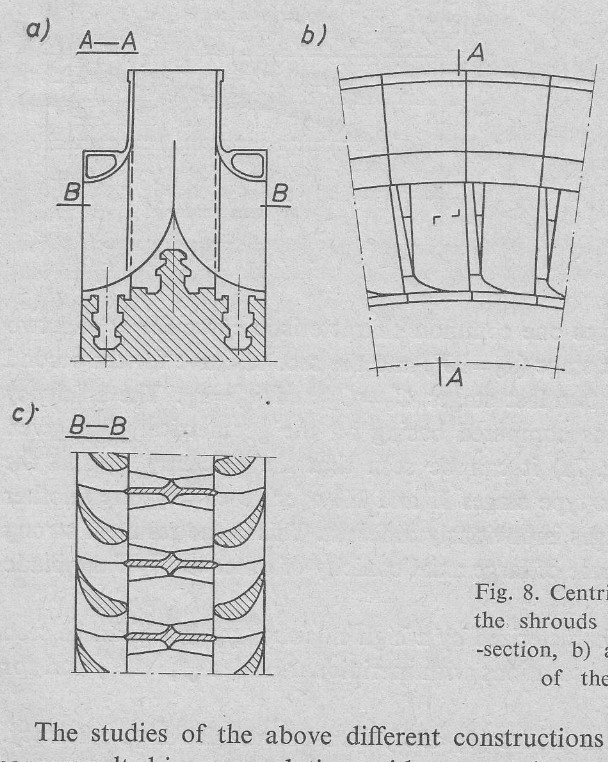

Fig. 8. Centripetal rotor of two-side shrouded type — the shrouds being segmented: a) meridional cross--section, b) as seen from the outlet, c) development of the cylindrical cross-section $B-B$

The studies of the above different constructions carried out by the authors of this paper resulted in some solutions with segmental covers. Fig. 8 shows one of the possible solutions. Their common feature is that the segments of covers are carried by: axial, radial or radial-axial blades and the centrifugal forces pressing one shroud-segment to the other.

In the case under consideration the centripetal rotor wheel consists of a meridional blade row and two axial blade rows. The segments of the cover are made together with axial blades.

The adjustable guide vanes also cause some troubles in designing and particularly their supporting bearings. An attractive solution is perhaps supporting of these blades in steam supports, fed by the process steam.

The use of adjustable guide vanes can bring additional profits when applied in following cases:

a) a correction in flow distribution between multi-flow low pressure parts, first of all with multi-pressure condensing system; such systems will be more frequently widely used in turbosets of large output and in total energy systems,

b) power control in industry steam turbines,

c) "interception", first of all in connection with low pressure parts of wet-steam turbines instead of shutting-off flap valves,

d) active braking by adjusting the guide vanes into inversed position.

The problems mentioned in this paper are recently theoretically and experimentally more widely investigated at the Institute of Fluid-Flow Machinery of the Technical University of Łódź for and in the cooperation with the ZAMECH Turbine Works in Elbląg.

References

[1] W. R. Gundlach, J. Krysiński, *Beitrag zur Entwicklung der Zentripetalturbine*. Maschinenwesen und Technologie der Technischen Universität Dresden, nr 52, 1967.

[2] W. R. Gundlach, *Prace IMP* (Transactions of the Institute of Fluid-Flow Machinery), No. 42 - 44, 1969, p. 570 - 571.

[3] Г. А. Филиппов, В. И. Абрамов, В. В. Фролов, *Тепловой расчет турбин*. Машиностроение, Москва 1974.

[4] W. Traupel, *Thermische Turbomaschinen*. Band I, 2 Auflage. Springer-Verlag, Berlin (Heidelberg) New York 1966.

[5] G. Cordes, *Experimentalle Untersuchungen über den Strömungsverlust in Turbinenschaufelgittern mit konischer Aussenwand*. Zagadnienia maszyn przepływowych (Problems of Fluid-Flow Machines), PWN, Warszawa 1968.

Aerodynamiczne i konstrukcyjne problemy projektowania dośrodkowych, dwustrumieniowych stopni wlotowych dużych turbin kondensacyjnych

Streszczenie

Dalszy wzrost mocy jednostkowej dużych turbin parowych wiąże się ze zwiększaniem wymiarów ostatniego stopnia oraz z poprawą przepływu w części niskoprężnej.

Możliwość praktycznego zastosowania stopnia z dopływem promieniowym jako pierwszego stopnia wlotowego przed dwustrumieniowym, niskociśnieniowym, osiowym układem łopatkowym zależy od dwóch warunków. Po pierwsze, rozwiązanie to powinno zapewnić wzrost sprawności pierwszych stopni mieszanego, promieniowo-osiowego układu łopatkowego w porównaniu z układem wyłącznie osiowym. Dopływ pary do części dwustrumieniowej odbywający się ze zmianą kierunku przepływu pod kątem 90°, powoduje znaczną stratę energii. Drugi warunek dotyczy konstrukcji i sprowadza się do żądania, by część niskociśnieniowa była jak najbardziej zwarta. W zasadzie kilka rzędów osiowego układu łopatkowego można zastąpić dośrodkowym stopniem wlotowym, co daje zmniejszenie

długości i masy części niskociśnieniowych. To rozwiązanie może być specjalnie atrakcyjne w dużych wielokadłubowych turbinach parowych.

W artykule rozważa się wyniki badań nad możliwością zastosowania dośrodkowego stopnia wlotowego w niskoprężnych kadłubach dużych turbin parowych. Warunkiem otrzymania sprawnego stopnia wlotowego jest zastosowanie dwustrumieniowego wirnika z pokrywą segmentową, reagującego jedynie na duże przesunięcia względne spowodowane różnymi szybkościami nagrzewania się wirnika i kadłuba. Zastosowanie typowego wirnika z monolityczną pokrywą (jak w sprężarkach odśrodkowych) napotyka bardzo duże trudności spowodowane dużymi prędkościami obwodowymi i dużymi wymiarami wlotu stopni osiowych.

Przedstawione zagadnienie wymaga rozwinięcia szerokiego programu badań doświadczalnych i teoretycznych. Badania doświadczalnę winny wyjaśnić problemy przepływowe i określić wartości współczynników strat dla omawianych stopn wlotowych.

Na zakończenie podano niektóre wyniki aktualnie prowadzonych badań.

Аэродинамические и конструкционные проблемы проектирования центростремительных двухпоточных входных ступеней больших конденсационных турбин

Резюме

Дальнейший рост единичной мощности больших паровых турбин связан с увеличением размеров последней ступени и с усовершенствованием течения в части НД.

Возможность практического применения ступени с радиальным течением в качестве первой входной ступени, перед двухпоточным аксиальным облопачиванием части НД, зависит от двух условий. Во первых, это решение должно обеспечивать прирост к.п.д. первых ступеней с радиально-аксиальным течением по сравнению с к.п.д. исключительно аксиального облопачивания. Питание паром, происходяшее с отклонением потока под углом 90°, приводит к значительным затратам энергии. Второе условие касается конструкции и сводится к требованию, чтобы часть НД была по возможности наиболее компактной. В основном несколько рядов аксиального облопачивания в части НД можно заместить центростремительной входной ступенью. Это причиняетса к сокращению части НД и к уменьшению ее веса. Такое решение является особенно интересным по отношению к большим многокорпусным паровым турбинам.

В статье обсуждаются результаты исследования возможности применения центростремительной входной ступени в корпусах НД больших паровых турбин. Условием осуществления входной ступени, характеризующейся высоким значением к.п.д., является применение двухпоточного ротора со связывающими кольцеобразными сегментными крышками, реагирующего единственно на большие относительные перемещения, вызванные различными скоростями нагрева ротора и корпуса. Применение типичного ротора с монолитной кольцеобразной крышкой (как в центробежных компрессорах) встречается с очень большими затруднениями ввиду больших скоростей вращения и больших входных размеров аксиальных ступеней.

Проблематика этой задачи требует развития обширной программы теоретических и экспериментальных исследований. Экспериментальные исследования должны быть направлены на выяснение проблем течения и определение значений коэффициентов потерь для обсуждаемых радиальных роторов.

В заключении представлены некоторые результаты актуально проводимых исследований.