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Helium Turbine Design for a 1000 MWe Gas-cooled Fast Breeder Reactor with Closed Gas Turbine Cycle

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by

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Summary

Progress in reactor technology allows to achieve coolant outlet temperatures above 700°C in the case of gas-cooled fast breeder reactors. These high coolant temperatures can be utilized completely through direct connection of a gas turbine. This direct connection of the gas turbine with the reactor is feasible provided that the coolant employed is inert and not susceptible to activation. Helium satisfies this condition. It serves as the reactor coolant and at the same time as the working fluid of the gas turbine and does not give cause for apprehension concerning the contamination of the individual plant components, which is obviously important for the operating safety of the plant.

This report deals exclusively with the preliminary design of a double-flooded helium turbine for a 1000 MWe gas-cooled fast breeder reactor. The influence is studied of several parameters, such as hub ratio, exit angle of the turbine wheel and inlet angle of the guide wheel, on the designed size of the turbine and the centrifugal stress of the blading, in order to get a survey which is helpful in the preliminary design.

Zusammenfassung

Fortschritte auf dem Gebiet der Reaktortechnologie ermöglichen bei gasgekühlten schnellen Brutreaktoren Kühlmittelaustrittstemperaturen des Reaktors von mehr als 700°C. Diese hohen Kühlmitteltemperaturen können durch direkten Anschluß einer Gasturbine vollständig ausgenutzt werden. Diese unmittelbare Verbindung der Gasturbine mit dem Reaktor ist möglich, wenn das verwendete Kühlmittel inert und nicht aktivierbar ist. Helium erfüllt diese Bedingung. Es dient als Reaktorkühlmittel und gleichzeitig als Arbeitsmittel der Gasturbine und läßt keine Kontamination der einzelnen Anlagekomponenten befürchten, was natürlich für die Betriebssicherheit der Anlage wichtig ist.

Dieser Bericht befaßt sich nur mit der Vorauslegung der zweiflutigen Heliumturbine für einen 1000 MWe gasgekühlten schnellen Brutreaktor. Der Einfluß verschiedener Parameter, wie Nabenverhältnis, Laufradabströmwinkel und Leitradzuströmwinkel auf die Baugröße der Turbine und die Fliehkraftspannung der Beschaufelung wird untersucht, um einen Überblick für die Vorauslegung zu erhalten.

Contents

1. Introduction

Progress in reactor technology allows to achieve coolant outlet temperatures above 700°C in the case of gas-cooled fast breeder reactors. These high coolant temperatures can be utilized completely through direct connection of a gas turbine. This direct connection of the gas turbine with the reactor is feasible provided that the coolant employed is inert and not susceptible to activation. Helium satisfies this condition. Moreover, helium possesses very good thermal properties and a very high velocity of sound ($\sim$ 1000 m/sec at 20°C) which are the prerequisites of intense reactor cooling.

Helium serves as the reactor coolant and at the same time as the working fluid of the gas turbine and does not give cause for apprehension concerning the contamination of the individual plant components, which is obviously important for the operating safety of the plant. If helium is used as the working fluid, the pressure losses in the circuit are very low which has a favorable effect on the circuit efficiency /1/. Therefore, helium is very well fitted for the direct connection of the turbine with the reactor. On the other hand, due to its small molecular weight and the high isentropic exponent, helium requires a very great number of stages for the turbine and the compressor. However, this is not a serious drawback, because, on the one hand, the optimum circuit efficiency for the helium turbine corresponds to very low expansion ratios (π_T = 2.2 /1/) and on the other hand the high velocity of sound of helium allows to choose such circumferential speeds that the stability of the turbine blading material is the only factor limiting the maximum circumferential speed. Moreover, the small expansion ratio produces the effect that the maximum and the minimum gas pressures prevailing in the closed cycle process differ but relatively little from each other. This alleviates the consequences of an accidental pressure drop in the reactor tank through the pressure equalization in the cycle process, which constitutes a significant safety factor.

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Also, since the reactor outlet temperature of helium corresponds to the turbine inlet temperature, the closed gas turbine cycle guarantees particularly high efficiencies /1/.

In addition, the direct connection of the reactor with the gas turbine leads to a simple circuit and especially in a fully integrated concept, in which the components of the helium turbine are placed in a pre-stressed concrete vessel together with the reactor, this is equivalent to a very compact and safe plant.

This report deals exclusively with the preliminary design of the helium turbine for a 1ooo MWe gas-cooled fast breeder reactor.

This work aims at examining the influence of different parameters, such as hub ratio, exit angle of the turbine wheel, inlet angle of the guide wheel, on the size of the turbine and at evaluating the number of stages, the hub diameter, the height of the blade and the centrifugal stress, in order to get a survey for the preliminary design.

2. Description of the Plant

To make clearer the underlying principles, a short description will be given of the plant. It is a 1ooo MWe single-shaft system with six units each connected in parallel to serve as heat exchangers and only one turboset provided. The schematic arrangement of the individual components is represented in Figure 1.

At point (Figure 1) the working medium enters the low-pressure compressor, where it is compressed (from 1 to 2), recooled in the intermediate cooler (from 2 to 3) and brought to the maximum process pressure in the high-pressure compressor (from 3 to 4). Subsequently, the gas is preheated in the heat exchanger (from 4 to 5) and heated up to the maximum process temperature in the reactor (from 5 to 6). In the turbine the gas is expanded (from 6 to 7) and in the heat exchanger it releases to the working medium from the high-pressure compressor part of the heat still contained in it (from 7 to 8). Finally, in the precooler the gas is recooled to the lowest process temperature (from 8 to 1). The cycle is closed now.

A possible layout of the plant appears from Figure 2. All the components are fully integrated in a prestressed concrete vessel. In the center of the cylindrical prestressed concrete vessel the reactor cavity is provided which is sealed by a removable lid of about 4 m thickness. In symmetrical arrangement around the core there are 6 penetrations each accommodating a heat exchanger (3), a precooler (4) and an intermediate cooler (5). Underneath the reactor cavity, there are a standby storing room (11) and two rooms for fuel element storage (1o), and in-between the room accommodating the fuel machine is located. Finally, the turboset (2) is installed in the lower part of the prestressed concrete vessel.

3. Design Calculation

The circuit calculation carried out in /1/ yielded the input data summarized in Table 1, such as mass throughput and the variables of state (Pressure and Temperature) at the inlet and outlet of the turbine. The turbine will be designed using these data.

Since the calculation of the flow through a turbogenerator is normally a three-dimensional problem, the description of a mass particle calls for three coordinates, namely axial, radial and circumferential coordinates. The variation of flow with the circumferential direction and with the radius is not taken into account. The flow is calculated according to the so-called mid-plane sectional method which considers the flow in the mid-plane section of the blading as being representative of the total flow over the blade height. This method of computation is well appropriate for the preliminary design of a turbogenerator and can also be applied with sufficient accuracy to a gas turbine, because the blade height is small compared to the mean diameter of the blading.

The first decision to be made in the design of a turbogenerator concerns the choice of the type of blading. Since the costs of blading represent a significant portion of the total cost of a gas turbine, special attention must be paid in design to an economic way of blade fabrication. For this reason the concept with a constant internal diameter is chosen for all stages with the blades twisted according to the spin term

$$C_u \times r = \text{const.}$$

This means that at all stages identical triangles of velocities appear on identical radii, i.e. that all blades of the turbine can be made by mere shortening of the longest blade. So one turbine wheel blade and one guide wheel blade each must be conceived.

Consequently, the curve of the entalpy drop reduction and the axial rate of flow of the medium over the blade height and over the length of the turbine are constant.

For gas turbines this favorable case can always be achieved, since the change of volume of the gas between the inlet and the outlet is relatively small. The homogeneous blades reduce considerably the total cost of the turbine. In the mid-plane section of the last stage a degree of reaction w = 5o % is provided. Constant data for the computation are the properties of the working gas (gas constant R and isentropic exponent K), the values for pressure and temperature at the turbine inlet and outlet, the total gradient of the turbine and the speed. The hub diameter is kept constant over the length of the turbine. Calculations were performed for a reaction turbine. The triangle of velocities in the mid-plane section of the last stage is the determining factor in calculating the reaction turbine.

In Figure 3 a triangle of velocities is represented for a degree of reaction w = 5o %. The axial component of the absolute velocity is

$$C_z = \frac{u_{mA}}{\cot \beta_2 - \cot \alpha_o} \qquad (1)$$

β_2 being the exit angle of the turbine wheel, α_o the inlet angle of the guide wheel and u_{mA} the circumferential speed in the mid-plane section of the last stage. This circumferential speed is derived from the angular velocity ω, the hub ratio $\nu_A = d_i/d_{aA}$ at the turbine outlet and the internal diameter d_i and the outside diameter d_{aA}, respectively, of the blading, viz.

$$u_{mA} = \omega \, d_{aA} \, \frac{1 + \nu_A}{4} \qquad (2)$$

with

$$\omega = \frac{\pi}{3o} \, n$$

where n is the speed.

The continuity equation at the outlet of the turbine reads

$$F_A = \frac{m}{\rho_A \cdot C_z} \qquad (3)$$

m being the throughput and ρ_A the gas density in the outlet sectional area F_A. With $F_A = (d_{aA}^2 - d_i^2)\,\frac{\pi}{4}$ and the gas law $P_A/\rho_A = RT_A$, P_A and T_A being the pressure and temperature, respectively, at the turbine outlet, R the gas constant, equation (3) yields

$$(d_{aA}^2 - d_i^2)\,\frac{\pi}{4} = \frac{m \cdot R \cdot T_A}{P_A \cdot C_z} \qquad (4)$$

If one extends the left side of this equation by d_{aA}^2, if one further considers $\nu_A = d_i/d_{aA}$ and if one substitutes equation (1) for c_z and equation (2), respectively, for u_{mA}, several transformations yield the following expression:

$$d_{aA} = \sqrt[3]{\frac{16 \cdot m \cdot R \cdot T_A(\cot\beta_2 - \cot\alpha_o)}{P_A \cdot \pi \cdot \omega\ (1+\nu_A)\ (1-\nu_A^2)}} \qquad (5)$$

Since the mass throughput m, the outlet temperatures T_A, the outlet pressure P_A and the speed n are constant, it is clearly visible that the external diameter d_{aA} and consequently the circumferential speed u_{mA} and the axial velocity C_z depend exclusively on the hub ratio at the outlet ν_A, the exit angle of the turbine wheel β_2 and the inlet angle of the guide wheel α_o. Varying this quantity yields quite a number of design possibilities.

The number of stages Z is the quotient of the total Entalpy drop ΔH and the stage Entalpy drop $\Delta H_s = u_{mA}\ C_z\ (\mathrm{cotg}\alpha_1 + \mathrm{cotg}\alpha_o)$ (cf. Figure 3).

$$Z = \frac{\Delta H}{u_{mA} \cdot C_z\ (\mathrm{cotg}\alpha_1 + \mathrm{cotg}\alpha_o)} \qquad (6)$$

In this equation α_1, the exit angle of the turbine wheel, is still unknown. Since the degree of reaction is assumed to be w = 5o %, the value for $ß_2$ can be substituted for α_1. Then the equation (6) yields a stage number which in general is not a whole number. Therefore, a rounded off value is used for the number of stages and the value of α_1 is computed from equation (6). This means that the angle α_1 deviates from the angle $ß_2$ which produces the effect that the degree of reaction in the mid-plane section of the last stage is no longer precisely 5o %.

The circumferential speed for u_{aA} at the external radius of the last stage is determined from the diameter d_{aA} and the speed n.

$$u_{aA} = \frac{\pi \cdot n \cdot d_{aA}}{6o} \qquad (7)$$

The blade height h_E at the turbine inlet can then be calculated using the continuity equation, while the blade height h_A at the turbine outlet is found to be

$$h_A = \frac{d_{aA} - d_i}{2} \qquad (8).$$

3.1 Calculation of the Centrifugal Stress

When calculating the centrifugal stress of a blade reference should be made to the following remarks.

The total centrifugal force K of a blade on the radius r is obtained by integration of

$$K = \int_{r}^{r_a} r \cdot \omega^2 \cdot \rho \cdot F(r) dr$$

ρ being the density of the blade material, ω the angular velocity of the blade, F(r) the variable cross-sectional area of the profile and r the variable radius.

This centrifugal force attains its maximum for $r=r_i$, r_i being the internal radius, i.e., the radius in the section of the blade base.

The stress σ_F in the base section with the surface F_i is therefore

$$\sigma_F = \frac{1}{F_i} \cdot \rho \cdot \omega^2 \int_{r_i}^{r_a} F(r) \cdot r \cdot dr \qquad (9)$$

The development of the profile area F(r) can be assumed to be linear with good approximation /1o/.

$$F(r) = F_i - \frac{F_i - F_a}{r_a - r_i} (r - r_i)$$

In this way we are on the safe side. If the ratio of sectional areas of the profiles at the blade head and at the blade base are written

$$f = \frac{Fa}{Fi}$$

the integrated form of equation (9) reads

$$\sigma_F = \rho \cdot u_a^2 \left[\frac{1}{2} (1-\nu^2) - \frac{1}{6} (1-f) (2-\nu-\nu^2) \right] \qquad (1o)$$

Equations (9) and (1o) must be applied both to the last and the first stages. Although the stress is higher at the last stage, the permissible stability values are considerably lower at the first stage on account of the higher temperature. Therefore, the stresses occurring at the first stage must be evaluated. When calculating the tensile stress in the blade base of the first stage it is assumed that the sectional area of the blades decreases by 5o % from the base to the head. In the case of the gas turbine blade, the ratio f varies between 3o % and 7o %.

3.2 Range of Variation of Design Parameters

In the preliminary design of the helium turbine, the hub ratio at the turbine outlet $\nu_A = d_i/d_{aA}$ with the constant hub diameter d_i and the external diameter d_{aA} at the turbine outlet as well as the exit angle of the turbine wheel $ß_2$ and the inlet angle of the guide wheel α_o are varied. The boundary values for the hub ratio are

$$\nu_A = 0.5 \text{ and } \nu_A = 0.9$$

The lower limit is obtained for two reasons. With $\nu_A < 0.5$ there is the risk of dead water formation in the hub, the blade height increases with decreasing hub ratio, and the centrifugal stress in the blade base can no longer be tolerated. The upper limit ν_A is fixed with a view to reasonable tip clearance and blade tip losses, because the blade height becomes too little.

The limit for the exit angle of the turbine wheel was fixed to be $ß_2 = 24^o$ and $ß_2 = 32^o$. The reason is that for a degree of reaction of 50 % $ß_2 = \alpha_1$, and the exit angle of the turbine wheel α_1 influences the circumferential efficiency. The latter attains its maximum for $u/C_1 = \cos\alpha_1$. U/C_1 lies in the range from 0.85 to 0.92 which corresponds to the values for $ß_2$ indicated above.

The choice of the angle α_o is a compromise between the number of stages and the outlet loss. Often α_o is chosen smaller than 90^o in order to increase the stage Entalpy drop and thus decrease the number of stages. On the other hand, the outlet loss increases with decreasing α_o. The cinetic energy $c_{2u}^2/2$ is denoted the outlet loss. The circumferential component of the absolute velocity at the outlet of the last stage is defined to be

$$C_{2u} = C_z/tg\alpha_o.$$

This means that when C_z remains constant, C_{2u} becomes the greater the smaller α_o is and the same applies to the outlet losses.

The lower limit introduced in the calculation is $\alpha_o = 70^o$. In order to restrict the number of design concepts to technically reasonable solutions, the following additional boundary values are fixed: the maximum blade height at the turbine inlet $h_E = 40$ mm, the maximum circumferential speed at the turbine outlet U_{aA} = 45o m/sec, the maximum number of stages Z_{max} = 2o and the maximum centrifugal stress appearing in the blade base of the first stage σ_E = 25 kg/mm^2.

The following considerations led to the indicated boundary values:
In a blade of less than 4o mm length the zone of the undisturbed flow gets too small; the tip zone becomes relatively larger and therefore the blade tip losses increase with decreasing blade height. Moreover, also the tip clearance losses increase and the internal efficiency becomes too low. The number of stages is limited to 2o because of the high cost fraction of the blading in the manufacturing costs of the turbine and because of the length of the turbine.

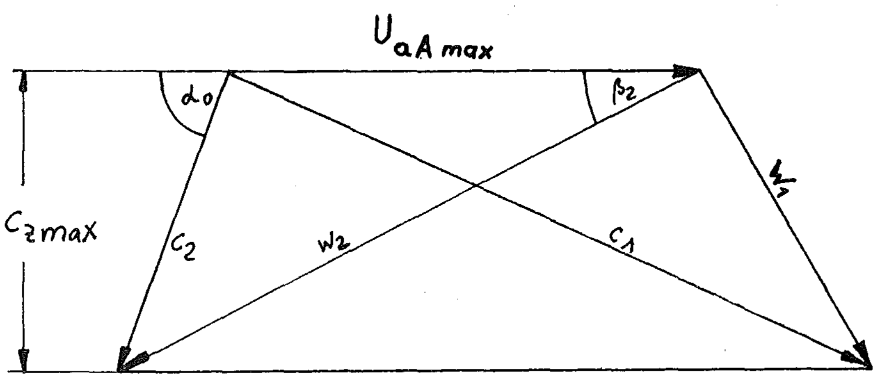
The circumferential speed at the outside radius of the last stage U_{aA} = 45o m/sec corresponds to the maximum permissible stress of the blading material available. It is intended to use G.94 of the English firm Jessop-Saville as blading material. G.94 is a nickel-base austenitic vacuum-fused cast alloy containing 1o % Co, 9 % Cr and 4 % Mo, W, Nb /16/. For the material indicated above $\sigma_{o.1}$ = 5o kg/mm^2 for 1o^{5}h and 7o6^{o}C, and with a safety factor of 1.5 the maximum permissible stress is found to be σ_{perm} = 33 kg/mm^2.

Since, due to the gas forces the bending stress of the blades amounts to about o.3 to o.4 of the centrifugal stress /1o/, the upper limit of the centrifugal stress in the blade base of the first stage is found to be σ_E = 25 kg/mm^2.

3.3 Single- or Double-flooded Turbine

The criterion of choice of a single-or double-flooded turbine is determined by the maximum axial velocity of the gas through the turbine C_{Zmax} on account of the axial shear and the maximum stress of the material. For a helium turbine the maximum permissible axial velocity C_{Zmax} = 2oo m/sec /1o/. The computation is made at the last stage of the turbine where the volume flow is maximum.

The triangle of velocities



with $\beta_2 = 28^{\circ}$ and $\alpha_o = 70^{\circ}$ yields

$$U_{aAmax} = C_{zmax}\ (\cot \beta_2 - \cot \alpha_o) = 304 \text{ m/sec.}$$

The maximum centrifugal stress occurs at maximum blade height. The latter is determined by the minimum value of the hub ratio at the outlet. For potential flow the lower limit of the hub ratio, as explained in 3.2 amounts to $\nu_A = 0.5$.

The centrifugal stress is expressed by

$$\sigma = \frac{10^{-6} \cdot \rho \cdot u^2}{g} \left[\frac{1}{2}\ (1-\nu^2) - \frac{1}{6}\ (1-f)\ (2-\nu-\nu^2) \right] \ \frac{\text{kg}}{\text{mm}^2}$$

With $\rho = 8{,}210 \text{ kg/m}^3$
$U = U_{aAmax} = 304$ m/sec
$g = 9.81 \text{ m/sec}^2$
$\nu = \nu_A = 0.5$
$f = 0.5$

one finds

$$\sigma_{Amax} = 20.95 \text{ kg/mm}^2$$

Since, due to the gas forces, the bending stress of the blade amounts to about o.3 to o.4 of the centrifugal stress, the maximum stress of the blades is found to be

$$\sigma_{max} = \sigma_{Amax} \times 1.4 = 29.33 \text{ kg/mm}^2.$$

This value is inferior to the permissible stress of the material used ($\sigma_{perm.}$ = 33 kg/mm^2 at 7o6^{o}C).

The external diameter of the blading is given by the expression

$$d_{aA} = \frac{U_{aAmax} \times 6o}{\pi \times n}$$

With n = 3ooo rpm we obtain

d_{aA} = 1.936 m and for ν_A = o.5 we obtain

d_i = o.968 m and

h_{aA} = o.484 m and

d_m = 1.452 m

This yields a maximum volume flow of

$$V_{max} = \pi \times dm \times h_{aA} \times C_{zmax} = 441 \text{ m}^3/\text{sec}.$$

Since in our case the volume flow at the turbine outlet is rather high, 71o m^3/sec, a double-flooded turbine must be provided for this plant. Therefore, the entire variation calculation was performed for a double-flooded turbine.

4. Results

The results of the turbine calculations are indicated in Figures 4 - 7. The centrifugal stress σ_E at the turbine inlet, the number of stages Z, the hub diameter d_i, the blade height h_E at the inlet and the circumferential speed U_{aA} at the outlet are represented in these figures as a function of the hub ratio ν_A at the outlet. The parameter is the exit angle of the turbine wheel β_2. All design concepts in the diagram correspond to

$$\alpha_o = \text{const.} = 90^o,\ 80^o,\ 70^o \text{ and } n = \text{const.} = 3000 \text{ rpm.}$$

The diagrams show that the blade height, the number of stages and the tensile stress decrease with increasing hub ratio, while there is an increase in the hub diameter and the circumferential speed.

Since the external diameter d_{aA} is proportional to

$$\frac{1}{\sqrt{(1+\nu_A)(1-\nu_A^2)}}$$

(equation (5)), it is obvious that with increasing hub ratio $\nu_A = di/d_{aA}$ the internal diameter and consequently the circumferential speed increase. This entails an increase in the stage Entalpy drop and, the total Entalpy drop remaining constant, a decrease in the number of stages (stage Entalpy drop proportional to $\Delta C_u \times U$). Since the volume passing through remains constant, the blade heights decrease, because the maximum velocity C_Z increases for constant values of β_2 (Figure 8).

$$V = c_z \times h.$$

Smaller blade heights result in lower centrifugal stresses and consequently the tensile stress decreases with increasing hub ratio.

An increase in the exit angle of the turbine wheel β_2 implies an increase in the axial velocity (Figure 8).

Since the volume flow remains constant, a smaller flow area is obtained.

$$V = F \times C_z$$

Consequently, the internal diameter and the blade height decrease; smaller blade heights imply lower centrifugal stresses.

The influence exerted by the angle α_o is shown in Figure 7. In this design diagram $\beta_2 = 28^o$ and is constant while the angle α_o is a parameter.

With a reduction of the angle α_o the axial speed C_Z increases (Figure 9) and consequently the internal diameters and the blade height become slightly smaller. Also the tensile stress decreases but the internal efficiency gets somewhat lower because of increased outlet losses. On the other hand, the number of stages is reduced, because the circumferential components of the absolute velocity and hence the stage Entalpy drop are enhanced (Figure 9). A lower number of stages also implies lower costs and a shorter length of the turbine. Therefore, the value $\alpha_o = 7o^o$ is adopted.

When defining the design ratio ν_A it must be considered that on the one hand relatively long blades are required to attain a satisfactory efficiency and that on the other hand backflows at the hub should be avoided. Moreover, it must be seen to it that the centrifugal stresses in the blade bases do not become excessive and it should not be forgotten that the length and hence the weight and the cost of the turbine and the turboset, respectively, are considerably influenced by the number of stages.

As a compromise of the requirements indicated a hub ratio of $\nu_A = o.65$ is adopted. Thus the design point is fixed. The major data are listed in Table 2.

5. Conclusion

Considering the results of this first preliminary design calculation it can be stated that the helium turbine can be realized on a very small scale if one bears in mind that the 1ooo MWe turbine has only 3o stages with a maximum blade height of o.37 m and a hub diameter of 1.35 m. A comparison was made between the helium and the steam turbine /17/. The difference relative to the steam turbine is particularly important in the number of stages (steam turbine 9o) and the blade height (steam turbine 1 m) for the same performance. It appears from /17/ that the steam turbine length is greater by about 5o %.

A further advantage of the helium turbine consists in its avoiding the danger of corrosion and water penetration into the circuit.

Moreover, the helium turbine possesses a development potential allowing to increase the helium temperature at the turbine inlet. Evidently, the high temperatures at the turbine inlet offer a great advantage in a closed gas turbine cycle because they bring about a considerable gain in plant efficiency /1/; however, the problems must also be put up with which are connected with high temperatures, such as blade cooling and stability of the turbine rotor.

Symbols

d	mm	diameter
r	mm	radius
m	Kg/sec	mass throughput
n	rpm	speed
F	m^2	flow section
R	$\frac{Kpm}{Kg}$	gas constant
T	^oK	temperature
P	Kp/m^2	pressure
V	m^3/sec	volume flow
h	mm	blade height
f	-	ratio of cross-sectional areas of the profiles
u	m/sec	circumferential speed
C_u	m/sec	components of the absolute speed in the circumferential direction
C_z	m/sec	axial speed of helium
C_1	m/sec	absolute speed of the helium at the inlet of the turbine wheel
C_2	m/sec	absolute speed of the helium at the inlet of the guide wheel
C_{2u}, C_{1u}	m/sec	components of the absolute speed in the circumferential direction
ΔC_u	m/sec	difference of the components of the absolute speed in the circumferential direction
W_1	m/sec	relative inlet speed of the turbine wheel
W_2	m/sec	relative exit speed of the turbine wheel
α_o	0	inlet angle of the guide wheel
α_1	0	exit angle of the guide wheel

β_1	0	inlet angle of the turbine wheel
β_2	0	exit angle of the guide wheel
$\nu = d_i/d_a$	-	hub ratio
ρ	Kg/m^3	helium density
ω	rad/sec	angular velocity
σ	Kg/mm^2	stress
w	-	degree of reaction
ΔH	m^2/sec^2	total entalpy drop in the turbine
ΔH_s	m^2/sec^2	stage entalpy drop
Z	-	number of stages
π_T	-	expansion ratio in the turbine

Indices

a	external
i	internal
m	central
A	exit
E	inlet
F	base

References

/1/ Savatteri, C.: Geschlossener Gasturbinenkreislauf für einen 1ooo MWe gasgekühlten schnellen Brutreaktor - KFK 14o9 - Mai 1971

/2/ Frutschi, H.U.: Gasturbinen für He-gekühlte Kernreaktoren - Technica Nr. 1o (1969) S. 931 - 935

/3/ Savatteri, C.: Der Wirkungsgrad einer Anlage mit Heliumgekühltem Schnellem-Brutreaktor und Gasturbine für 1ooo MWe - INR - interner Bericht - unveröffentlicht

/4/ Thorn, J.D., Prince, K., Coast, G.: Nuclear Gas Turbine Concepts - IAEA - Symposium on Advanced and High Temperature Gas-Cooled Reactors - Jülich C 1-25 October 1968 - SM-111/4o

/5/ Hurst, J.W.: Some Considerations Affecting the Integration of Gas Turbines with Direct Cycle Gas Cooled Reactors - Rolls-Royce Ltd -

/6/ Holliday, J.B., and Hurst, J.W.: The Application of Gas-Turbines to Reactor Systems - Rolls-Royce Ltd -

/7/ Bammert, K. und Twardzick, W.: Kernkraftwerke mit Heliumturbinen für große Leistungen - Atomkernenergie - 9/1o (1967)

/8/ Straupel, W.: Thermische Turbomaschinen - Erster Band (1966), Springer-Verlag, Berlin

/9/ Bammert, K. und Fiedler, K.: Die Strömung in axialen Turbomaschinen - Ingenieur Archiv, Band 33 (1962), S. 322/329

/1o/ Bammert, K.: Theorie der thermischen Turbomaschinen - Teil II - Vorlesung Winter-Semester 1969 - Technische UNI Hannover -

/11/ Pfriem, H.J.: Properties of Helium Gas - Zurich Club Gas Cooled Fast Reactor Heat Transfer Meeting, Würenlingen (197o)

/12/ Malese-d'Hospital, G. - Fortescue, P.: Thermodynamic Comparison of Gas coolants for nuclear reactors - Proceedings of the Institution of Mechanical Engineers, London (1966)

/13/ Bammert, K. - Böhm, E.: High Temperature Gas-cooled reactor with gas turbine - ENEA Symposium on the Technology of integreted Primary Corcuits for Power Reactors, O.E.C.D. Headquarters, Paris (1968)

/14/ Bammert, K.: Zur Frage der Optimierung von nuklearen Gasturbinen-Kraftwerken - Energie und Technik 1 (1965)

/15/ Bammert, K.: Ausblick der Heliumturbine für große Kraftwerksleistungen - Atomkernenergie 12 (1967)

/16/ Jessop-Saville: High Temperature Steels - Data Sleets -

/17/ Bammert, K.: Dampf und Heliumturbinen für Kernkraftwerke mit Hochtemperaturreaktoren - Energie und Technik - 1968 - Heft 1 -

/18/ GHH - Geschlossene Gasturbinenanlage für einen Heliumgekühlten Schnellen Brüter (1967)

Table 1: Basic Data for the Turbine Design

Working medium	-	helium
Temperature at the turbine inlet	^{o}C	7o6
Pressure at the turbine inlet	Kg/cm^2	96.1
Throughput	Kg/sec	1589.1
Expansion ratio	-	3
Speed	rpm	3ooo
Temperature at the turbine exit	^{o}C	388.7
Turbine efficiency	%	91
Mechanical efficiency	%	99.7
Volume throughput at the turbine outlet	m^3/sec	71o

Table 2: Main Data of the Turbine of the 1ooo MWe Plant (Double-flooded Turbine)

Inlet pressure	kg/cm^2	96.1
Power	MW	132o.
Throughput	kg/sec	794.6
Expansion ratio	-	3
Speed	rpm	3ooo
Number of stages	-	15
Internal Diameter d_i	mm	135o.8
External Diameter d_{aE}	mm	1747.7
External Diameter d_{aA}	mm	2o78.2
Hub ratio ν_E	-	o.773
Hub ratio ν_A	-	o.65
Exit angle of the turbine wheel β_2	o	28
Inlet angle of the guide wheel α_o	o	7o
Exit angle of the guide wheel α_1	o	27.o4
Blade height, inlet h_E	mm	198.45
Blade height, outlet h_A	mm	363.69
Tensile stress in the blade base σ_E	kg/mm^2	1o.9
Tensile stress in the blade base σ_A	kg/mm^2	18.87
Circumferential speed U_i	m/sec	212.2
Circumferential speed U_{aE}	m/sec	274.5
Circumferential speed U_{aA}	m/sec	326.4
Axial velocity C_z	m/sec	177.5
Mach number, outlet mid-plane section	-	o.25

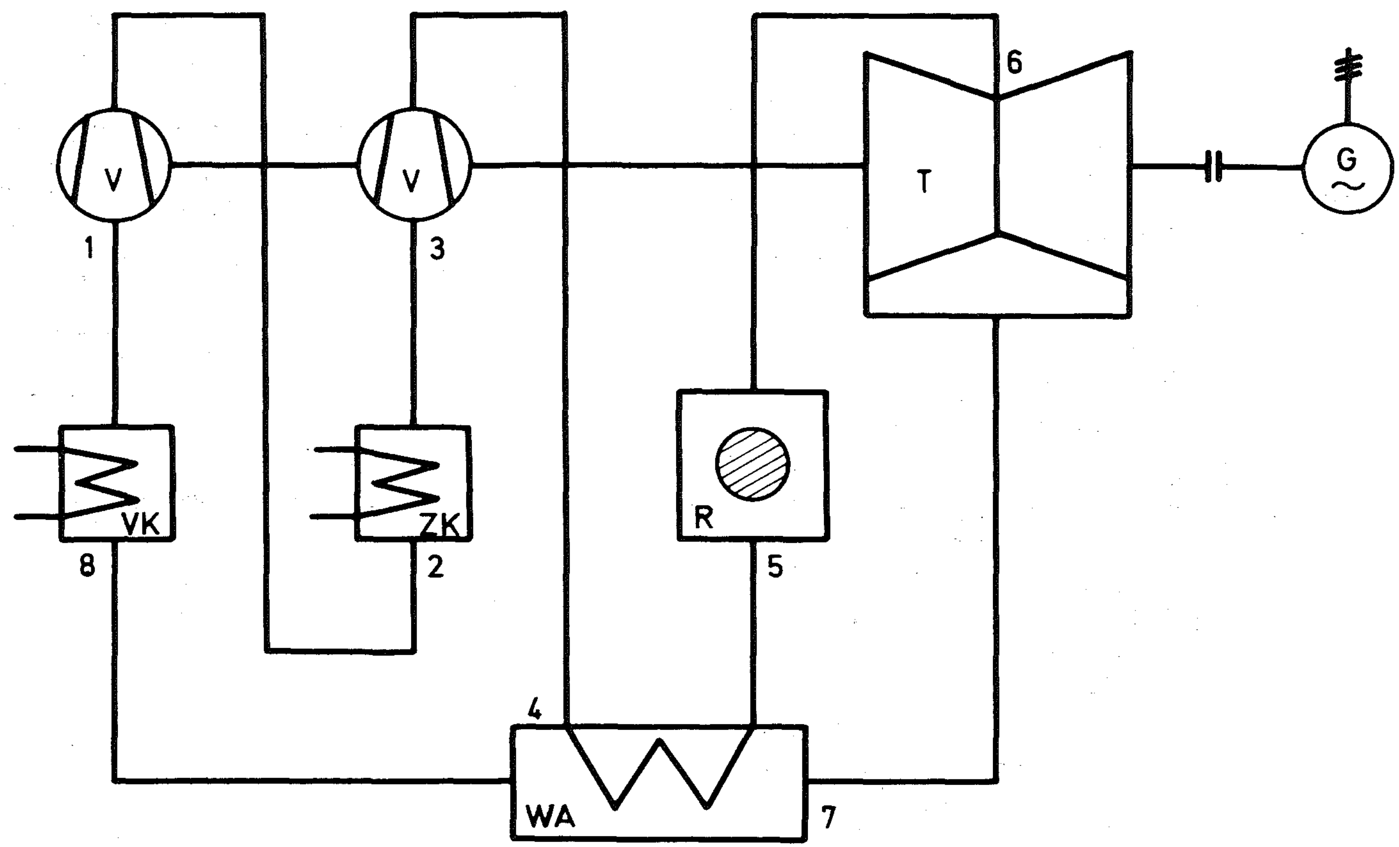
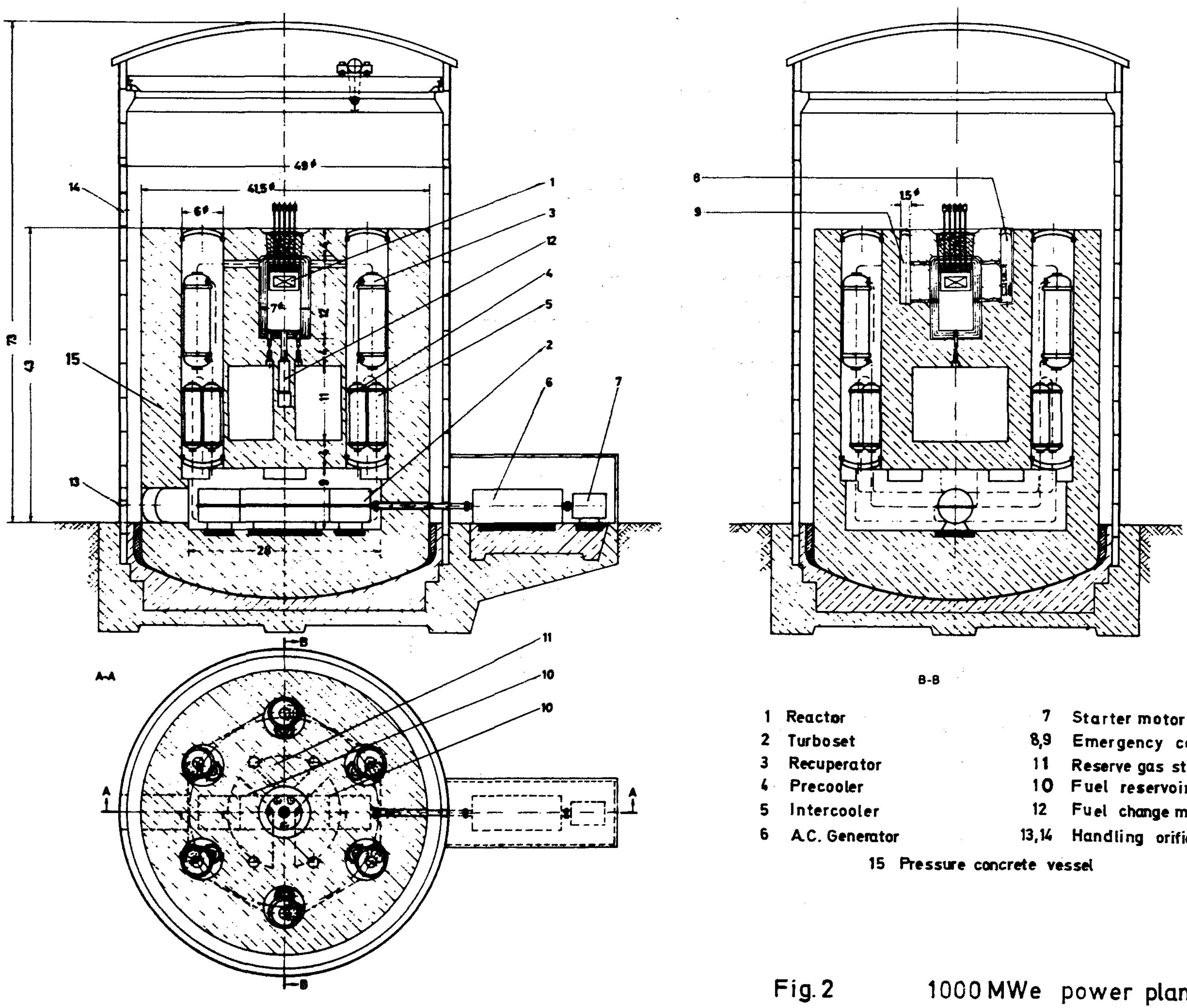


Fig. 1 Circuit diagram for a 1000 MWe helium power plant

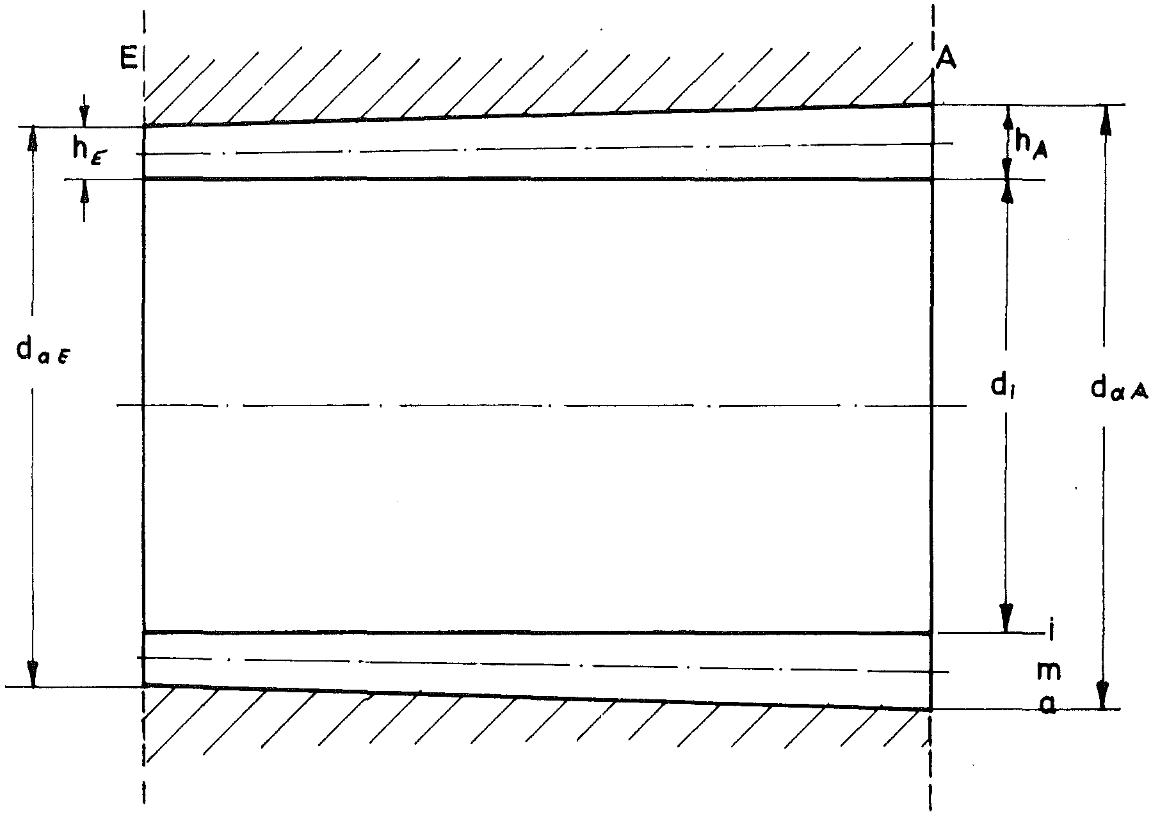
R = Reactor VK = Precooler V = Compressor

T = 2-Flow Turbine ZK = Intercooler WA = Recuperator



1 Reactor
2 Turboset
3 Recuperator
4 Precooler
5 Intercooler
6 A.C. Generator
7 Starter motor
8,9 Emergency cooling system
11 Reserve gas storrage
10 Fuel reservoir
12 Fuel change machine
13,14 Handling orifice
15 Pressure concrete vessel

Fig. 2 1000 MWe power plant



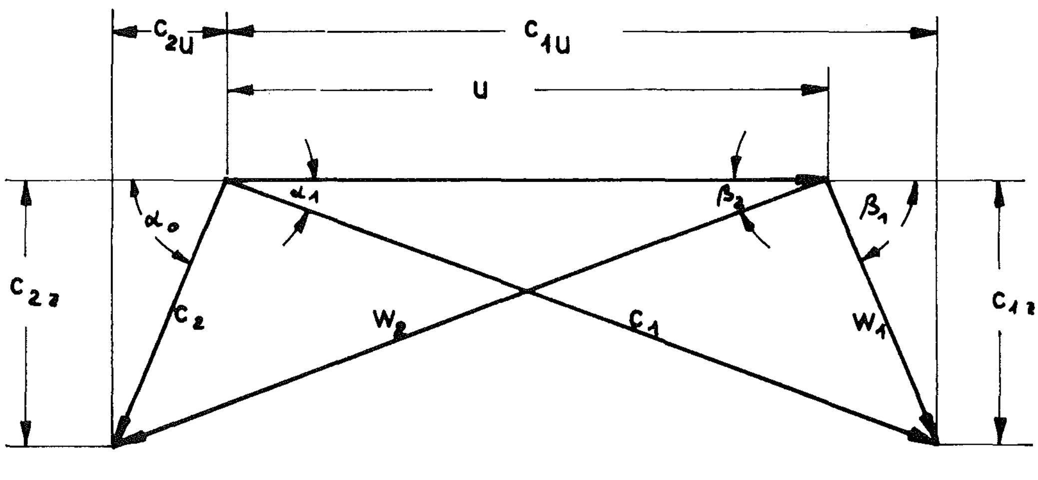


Fig. 3 Schematic longitudinal section of turbine and velocity triangles

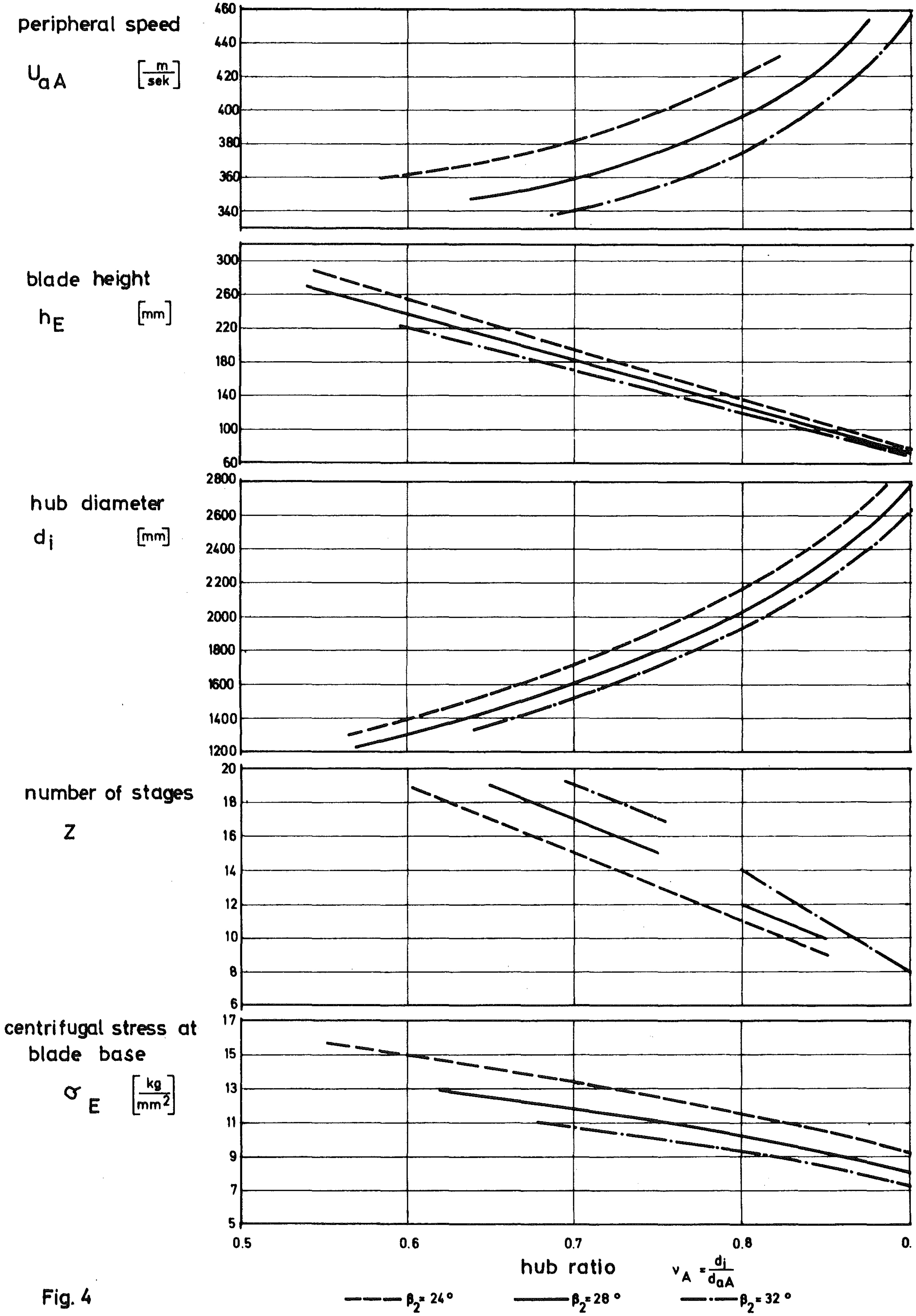


Fig. 4

Design diagram of the turbine for a 1000 MWe power plant

$n = 3000$ rpm ; $\alpha_0 = 90°$

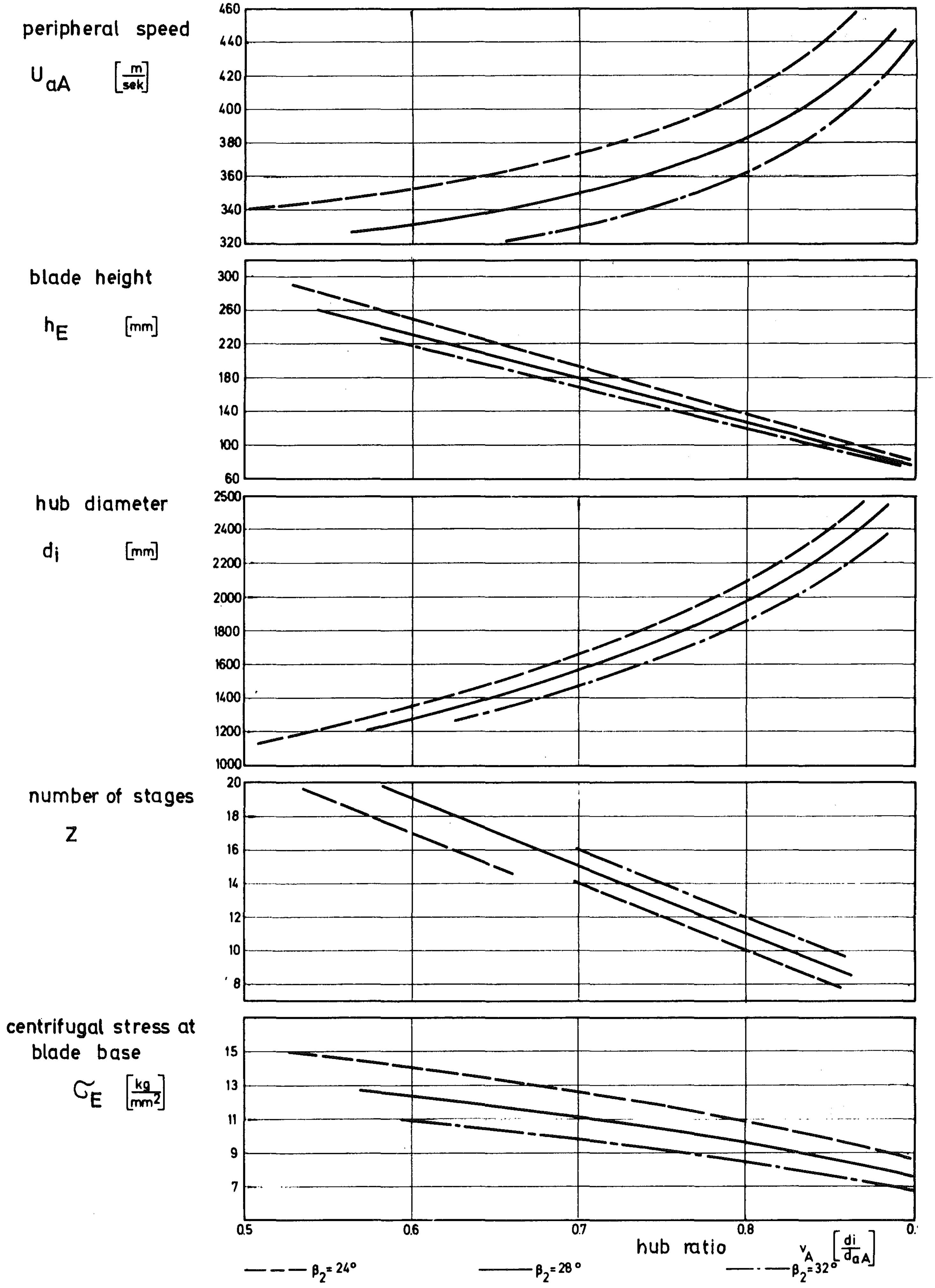


Fig. 5 Design diagram of the turbine for a 1000 MWe power plant

$n = 3000$ rpm , $\alpha_o = 80°$

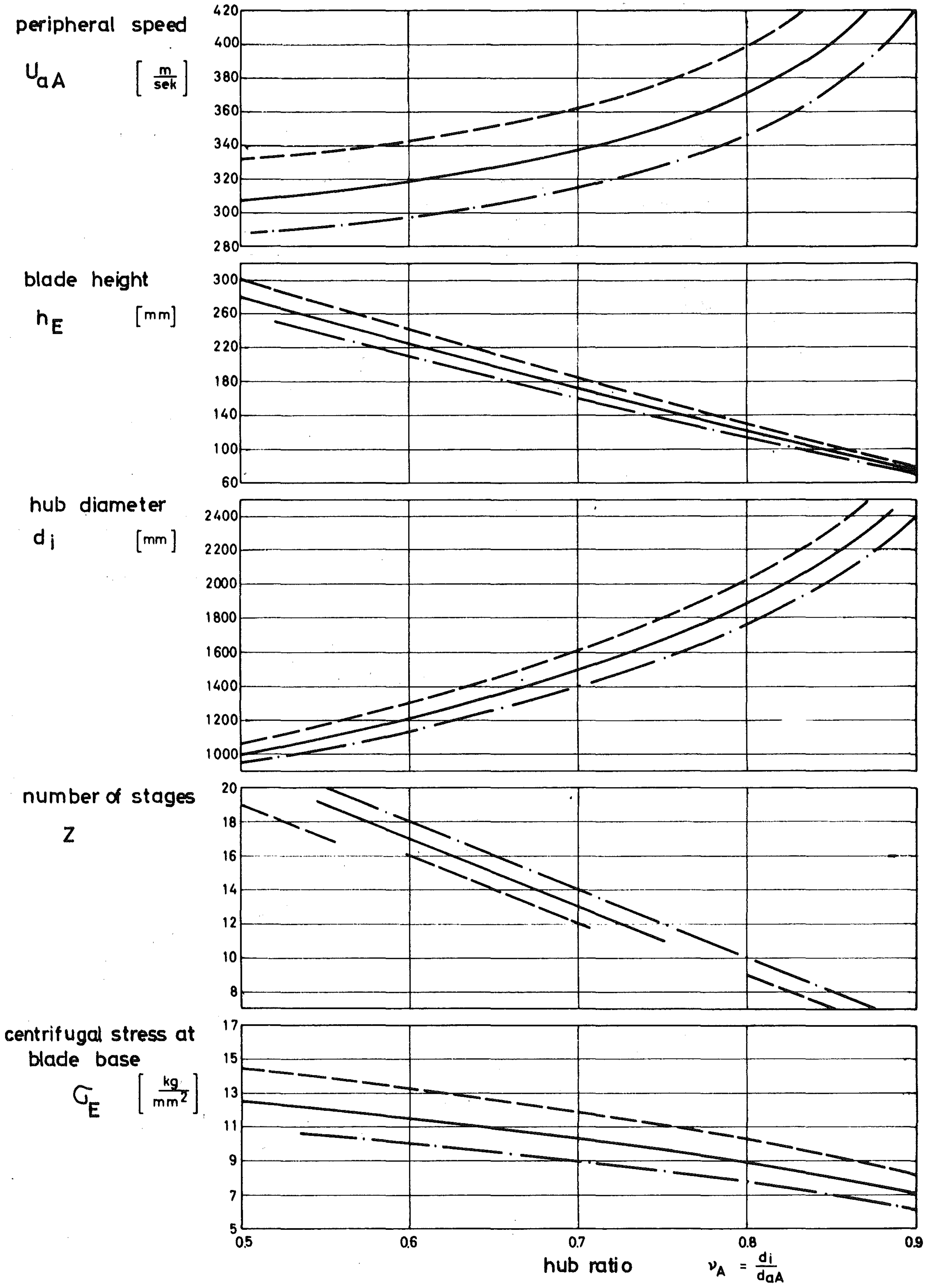


Fig. 6

Design diagram of the turbine for a 1000 MWe power plant

$n = 3000$ rpm ; $\alpha_0 = 70°$

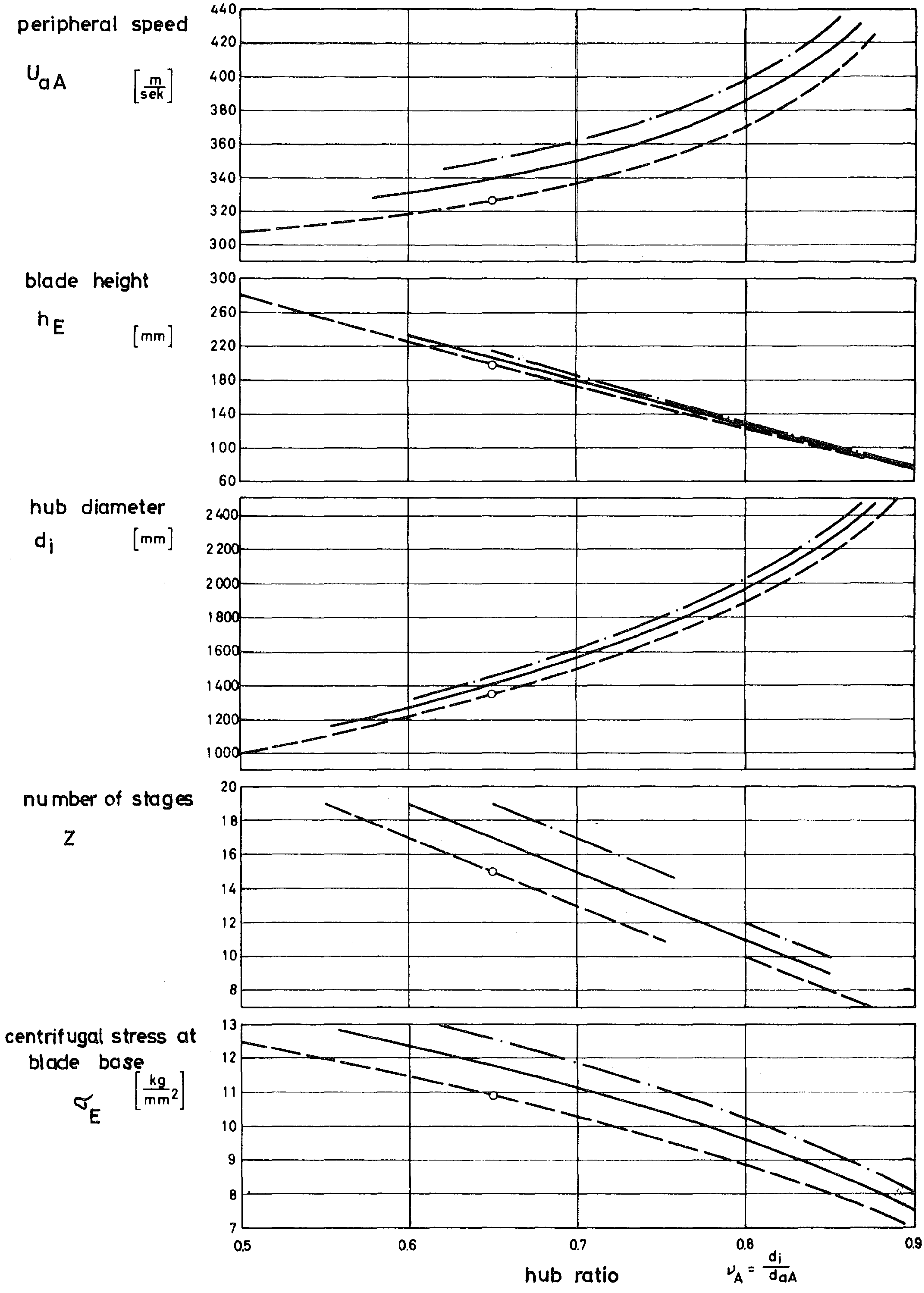


Fig. 7

Design diagram of the turbine for a 1000 MWe power plant

n = 3000 rpm ; $\beta_2 = 28°$; o reference design

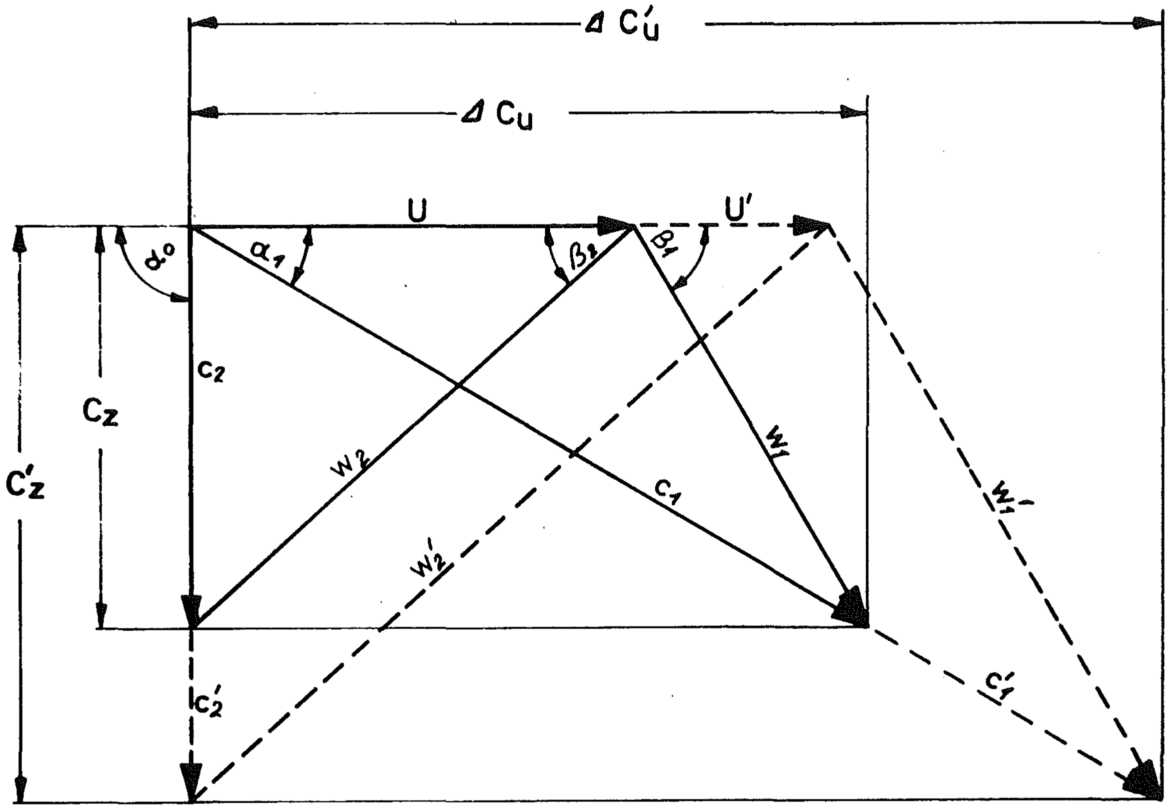


Fig. 8 Velocity triangle

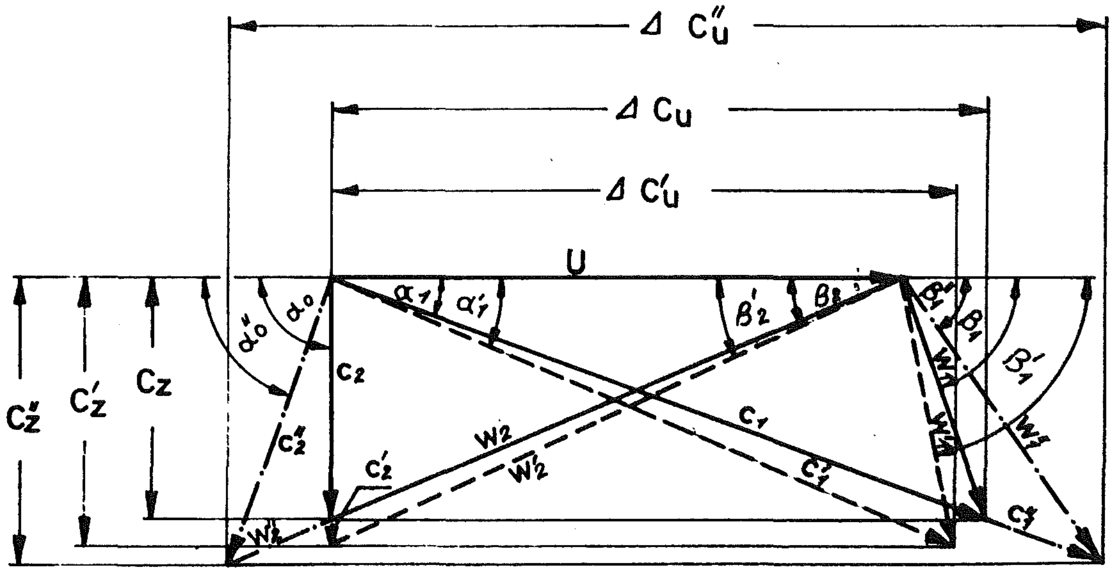


Fig. 9 Velocity triangle