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Hydraulic turbine development for low-output power plants

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Abstract

A model turbine for low-head applications, utilizing a novel flowpath design, was designed and built to meet specific design assumptions formulated for low-output hydraulic power plant design. Power-output and cavitation testing has been carried out, and the universal characteristic for the turbine obtained. The innovative design has been patented and is available as an offer for low-output hydraulic power installations. The design possesses significant advantages – with regard to turbine construction, operation, and emplacement – over the standard designs currently offered by leading world manufacturers. On the basis of velocity fields determined to exist within the rotor chamber of the turbine, a representative optimization procedure has been applied to rotor parameters. The procedure is here proposed as an effective and practical method for optimizing the performance of new rotor designs, as well as existing rotors under actual operating conditions.

Keywords: Water turbine; Design

1 Introduction

The global energy resources stored in watercourses are distributed across a broad range of available hydraulic heads – as constrained by the local terrain conditions (when estimating the gross power capacity for a given stream cross-section, the headwater elevation is functionally equivalent to the head). These resources, however, are allocated in a manner inversely proportional to their available head – the watercourses with the highest exploitable elevation difference account for a relatively small fraction of the total energy capacity. Such sources

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are necessarily rare, as their occurrence is predicated upon the presence of extreme terrain features. As one's attention shifts downrange toward the lower heads, the percentage of available power capacity increases correspondingly; the most plentiful hydraulic energy resources are associated with the smallest drops in elevation.

The utilization of such resources is strictly dependent on the economies of their development. It should be noted that cost-effectiveness of development also stands in correspondence with the available head: in general, the higher the head that a given power plant can utilize, the more economical its operation. The only constraint, aside from the capital available, is the technological state-of-the-art.

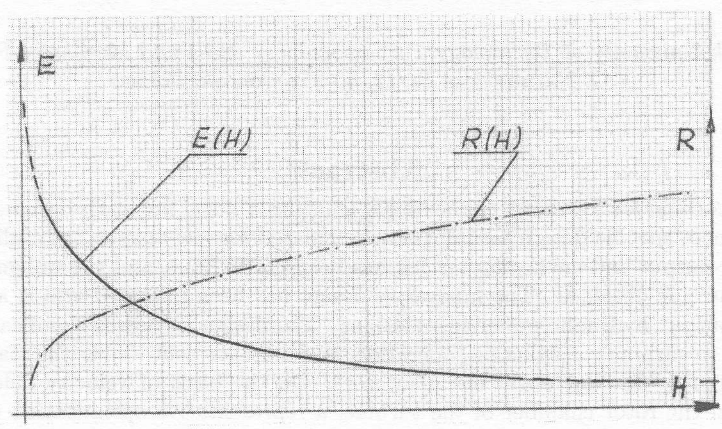


Figure 1. Representative distribution of potential energy E of watercourses and the corresponding economic return R as a function of available head H.

It is likewise the technological barrier that sets the other bound of affordable hydraulic energy utilization, delimiting economically viable access to energy resources at low heads.

The principle of efficiency has dictated that the more attractive high-head sites were the first to be exploited; as their availability lessened, the gradual development of technology, reinforced by the growing preference for ecologically sound renewable energy, pushed the envelope of affordability steadily towards the lower heads. At the current state-of-the-art – although individual cases had best be analyzed separately – it can generally be stated that elevation drops of less than 2 meters are not feasible for economically viable exploitation.

2 Current state-of-the-art and scope of expectations for low-head hydropower generation

In the progress realized so far, the economical utilization of increasingly lower available heads has been realized on the basis of: new developments in machinery, new plant construction designs, the economies of scale inherent in standardization of machinery and mass-production, as well as local preference programs for renewable resource utilization. The turbomachinery mutating in accordance with these factors has developed along the lines of axial-rotor-flow designs characterized by high flow throughput, high rotational velocity, and high efficiency over a broad range of loads.

These modern designs consist mostly of Kaplan and propeller turbines employed – in the course of technological progress – in various configuration culminating in the ubiquitous so-called ducted designs. Such implementations, in a specialized configuration aimed at utilizing low heads, have the additional advantage of simplifying the construction of concrete plant substructure, which can therefore be more cheaply built. Examples of turbomachinery of this kind, in complete installations, are shown in Fig. 2.

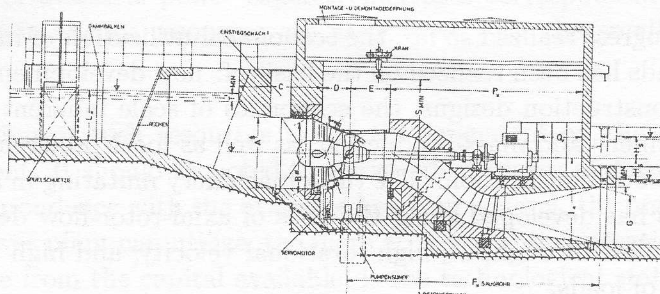
Continuing optimizing research into turbomachinery design, aimed at developing and improving the traits enumerated above, leads to rising economic returns from utilizing watercourses that are already regarded as economically viable, as well as extending the range of viability to lower heads, thus opening for exploitation hitherto unusable locales – areas of opportunity that will continue to expand as the low-head utilization technology is perfected.

3 Proposed turbine design and basic model testing

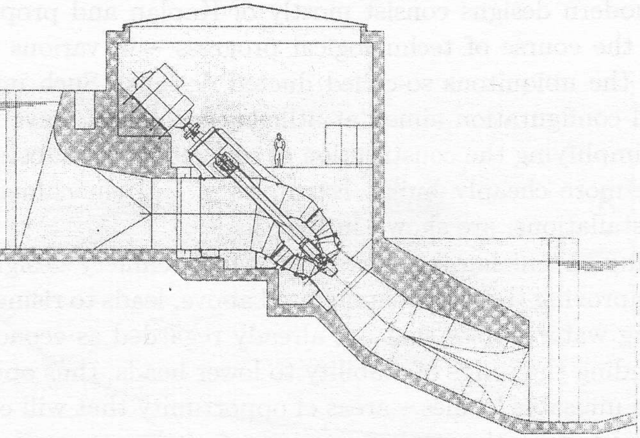
In the context of aforementioned design requirements, a turbine design is proposed with flow ducting configured to realize the principal advantages of installation types presented above. A concept drawing of the system is shown in Fig. 3 [3, 12].

The principal difference of the proposed design with regard to the presented traditional solutions is the means of generating angular momentum within the flow at the entrance to the rotor. A special volute-shaped inlet chamber is employed, formed by a helical flow-directing surface contained between a cylindrical inner wall and a conical outer wall. The volute combines the functionality of an inlet duct and a guide vane ring.

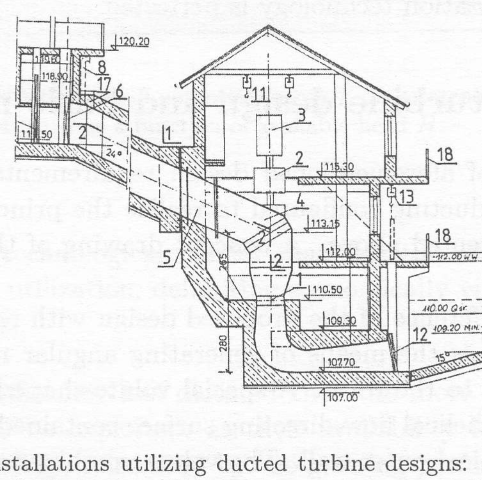
The turbine was constructed as a model, with rotor diameter $D = 265$ mm. Shape of turbomachine elements receiving energy from the flow has been designed on the following assumptions: axisymmetric flow; axial flow within the rotor



a.

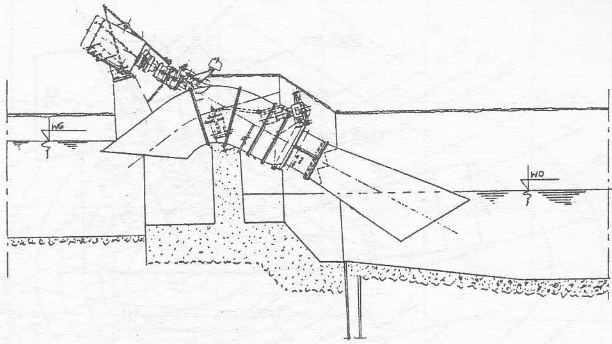


b.

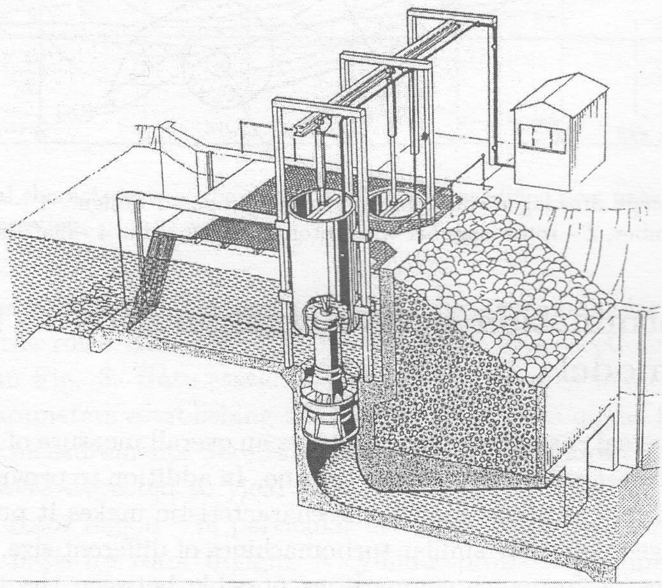


c.

Figure 2. Examples of installations utilizing ducted turbine designs: a. plant with horizontal ducted turbine (Bell, Escher Wyss); plant with inclined ducted turbine (CAT); c. plant with vertical ducted turbine (Myloff).



d.



e.

Figure 2. Examples of installations utilizing ducted turbine designs: d. siphon system (modified); e. submerged hall-less system (Flyght).

region; constant distribution of meridional component of velocity (C_m) along rotor chamber radius.

Flow measurement sections T1, T2, T3 are shown in vicinity of the rotor. The design has been realized as a model, built and installed on a test stand in the lab at Turbomachinery and Fluid Mechanics Department of Gdańsk University of Technology. The model, after initial verification of design assumptions, has been subjected to standard power output and cavitation tests [3] leading to the recording of its universal characteristic (Fig. 4).

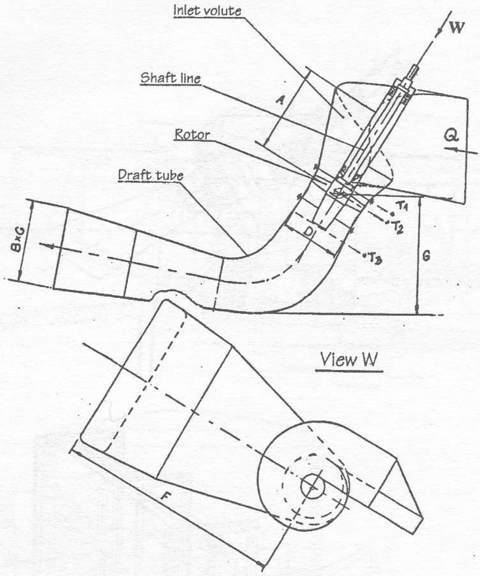


Figure 3. Concept drawing of flow ducting in the proposed design. 1 – volute-shaped inlet chamber, 2 – rotor chamber with rotor, 3 – draft tube, 4 –shaft line.

4 Determination of flow configuration in flowpath of the model

The universal characteristic constitutes an overall measure of combined operation of individual components of the turbine. In addition to providing information on the performance of the model, the characteristic makes it possible to predict behavior of geometrically similar turbomachines of different size, at any loadings. It also makes performance comparisons possible between the tested configuration and other turbomachines for which similar characteristics are available, and enables an overall assessment of design and operational qualities and flaws.

It does not, however, allow for specific analysis of work states for the above-mentioned individual components, which are ultimately responsible for the overall performance; and consequently provides little scope for verification of design correctness as regards both flow shaping within the turbine flowpath and the corresponding blade shaping of the rotor.

In order to carry out such verification it is necessary to determine the actual structure of flow within the flowpath of the turbine, with particular emphasis on flow behavior in the critical region where flow energy is converted into shaft power of the machine.

For the model in question the requisite measurements enabling the determina-

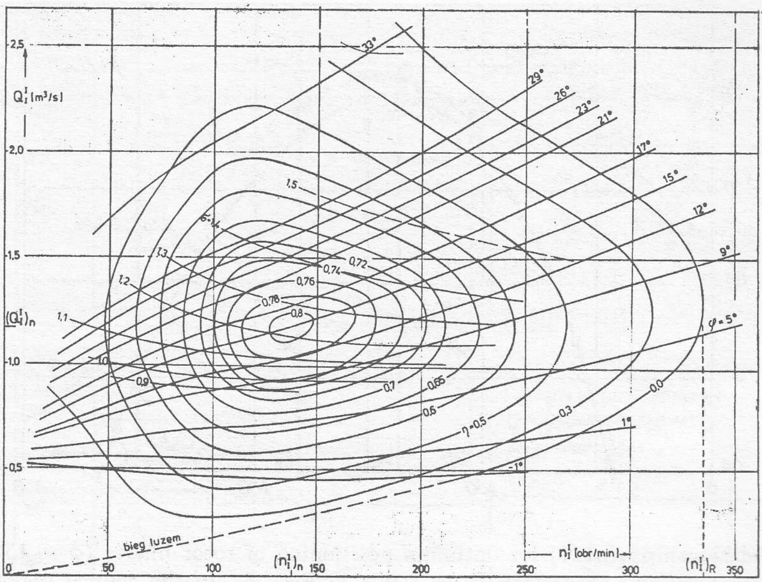


Figure 4. Universal characteristic of the model turbine with modified flow path. Values before rotor correction.

tion of flow structure were carried out in three planes perpendicular to shaft axis located within the rotor chamber. The measurement cross-sections are marked as T1, T2, T3 in Fig. 3. Both static and dynamic pressures were measured, as well as other parameters establishing the spatial kinematics of the flow.

Preliminary measurements were taken with the rotor blades removed. These measurements were expected to yield data on the degree of axial symmetry of flow within the rotor region – in particular at the points where flow kinematics constituted the basis for rotor design. A cylinder probe was employed; at this stage of the measurement procedure, the probe was mounted on the shaft to facilitate measurements at varying angular positions. A 6 mm wide ring-shaped region immediately adjacent to the duct wall, as well as a 10 mm wide ring-shaped region immediately adjacent to the shaft were excluded from measurement for reasons having to do with the shape and mounting of the pressure probe.

The measurements in all three test sections have demonstrated satisfactory uniformity of flow parameters along the angular dimension, for varying measurement radii and varying flow rates. The deviation from mean values in circumferential distributions did not exceed 4.5 % outside of hydrodynamic wake regions; such deviations as were recorded did not exhibit any overall pattern.

For the "rotorless" case, sample distributions of the meridional component of velocity and of flow angle (with respect to the tangential direction) in test plane 2 are shown in Fig. 6 for three radial positions and flow rates.

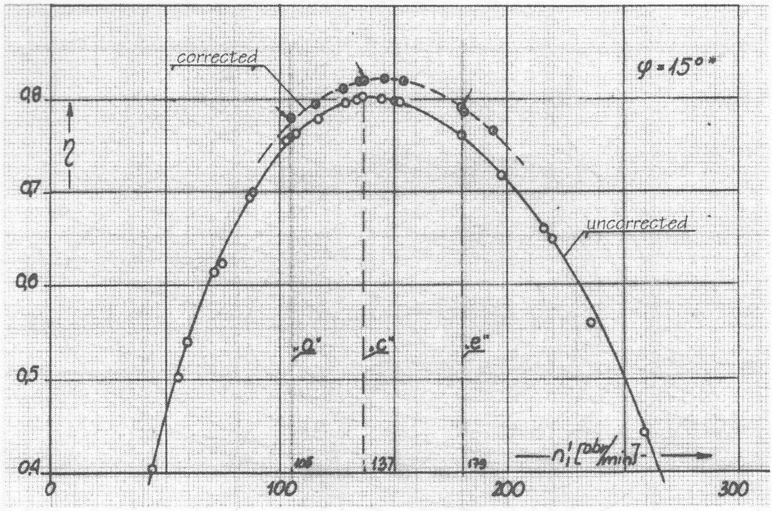


Figure 5. Relationship η(n1/1) for optimum positioning of rotor blades ($\phi = 15^o$) in turbine with universal characteristic as shown in Fig. 4. In the central region of optimal operation a fragment of modified curve is shown, resulting from introduced correction of design angle β_k (cmp. Fig. 9).

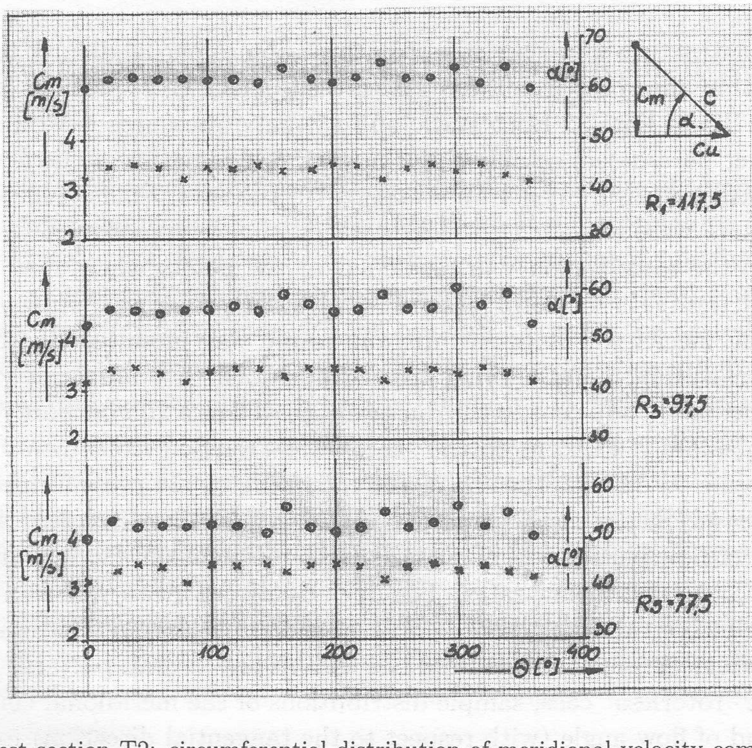


Figure 6. Test section T2: circumferential distribution of meridional velocity component (C_m) and angle (α) of flowline inclination with respect to the tangential direction. Data for optimum throughput ($Q = 0,148$ m^3/s) at selected radii R_1, R_2, R_3. No rotor blades mounted. Mean value of $C_m = 3.31$ m/s.

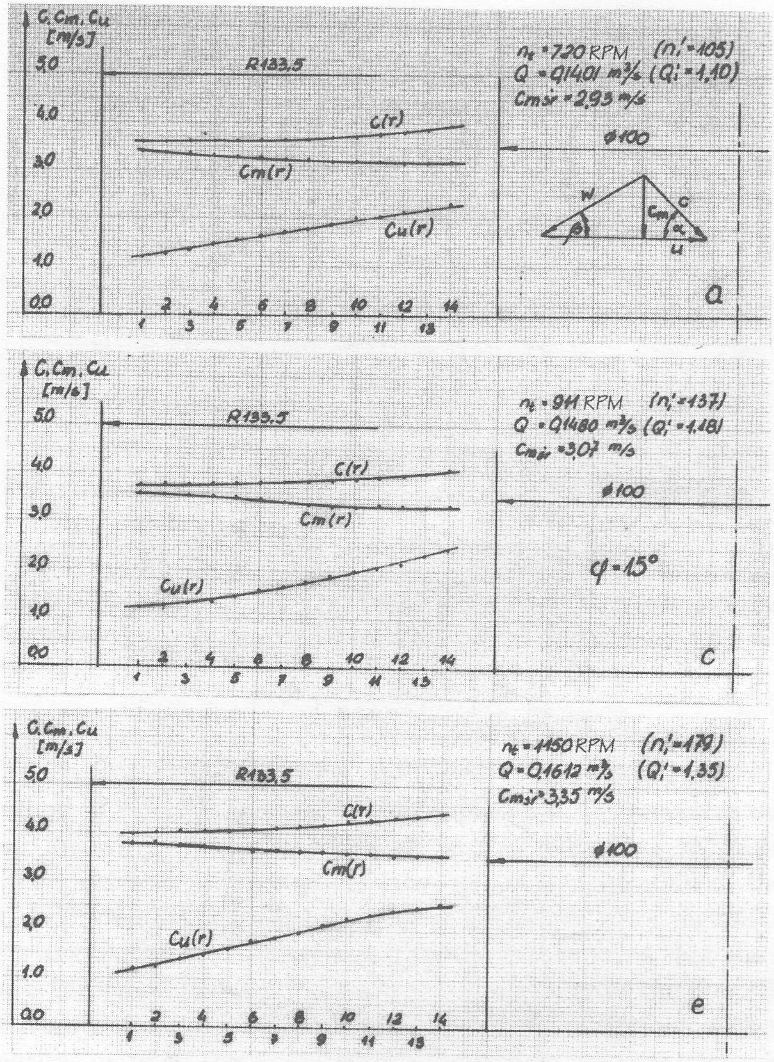


Figure 7. Test section T1: velocity components in flowpath with rotor, for optimal rotor blade position ($\phi = 15°$) and selected rotational speeds n. Vector geometry is shown for a cylindrical surface located behind the axis lying in picture plane.

The circumferential uniformity shown by these measurements has been regarded as sufficient justification for the positioning of a single representative probe (traversed along the radius) in each of the T1 and T3 test sections for the run with rotor blades mounted. A series of measurements were carried out in the resultant setup to determine flow geometry with the rotor blades mounted, with the flow parameters held to specific work points on the characteristic. The resulting flow configuration is shown in Fig. 7 and 8 for near-optimal performance

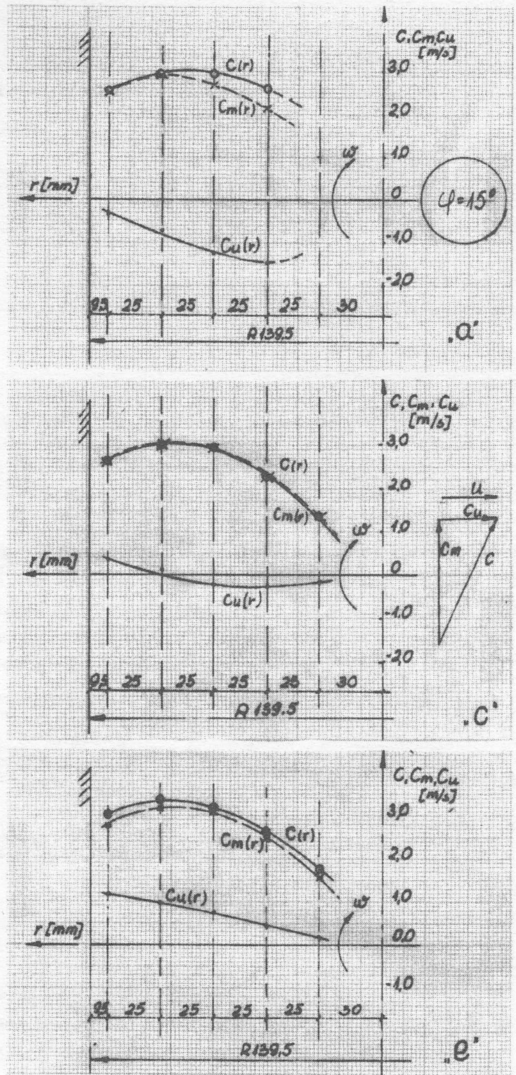


Figure 8. Test section T3: velocity components in flowpath with rotor, for optimal rotor blade position ($\phi = 15°$) and selected rotational speeds n (Fig. 5). Vector geometry is shown for a cylindrical surface located behind the axis lying in picture plane.

states.

The data obtained from flow measurements facilitated a modification (narrowing down) of design assumptions with regard to flow direction at inlet to the rotor region (section T1). The measurements of velocity field in the T3 section have provided valuable confirmation that it is possible to precisely control the

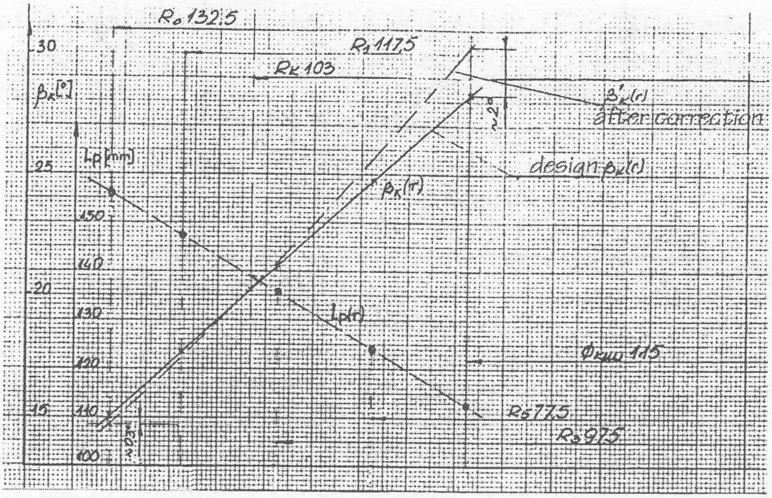


Figure 9. Plot of profile length $L_p(r)$ and profile twist angle $\beta_k(r)$, (inclination of profile with respect to the tangential direction) as function of rotor radius for the designed rotor blade. Corrective modification (shown as dashed line) of angle β_k after experimental determination of velocity field at sections T1 and T3 was carried out for the characteristic optimum work point.

degree to which the generated angular momentum is utilized in the turbine.

The method of velocity field generation is based on flow angular momentum conservation following the Euler equation, as formulated for hydraulic turbomachines. The ideal situation would involve complete utilization of the angular momentum generated in the guide region and entering the rotor region. Any trace of remaining angular momentum at exit from the rotor means that the geometry of the rotor is incorrectly configured to receive the energy in the flow. Such state may result from posited – but unrealistic – design assumptions with regard to flow kinematics, or from the intensity of dissipative processes along individual streamlines, which accounts for local hydraulic efficiency but is unknown *a priori* at the design stage.

Also – as the rotor blade is composed of spatially juxtaposed individual profiles – the influence of the interaction between the set of profiles on the dynamic characteristics of the individual profiles may lead to significant discrepancies between the expected and actual behavior of the blade as a whole. These factors, if estimated incorrectly at the design stage, may result in an efficiency drop in the global performance characteristic, and in local disturbances in the flow structure observed behind the rotor. The above considerations find illustration in the measurement data for the tested rotor operating in the designed flowpath. The presence of significant residual angular momentum (evidenced in the diagrams

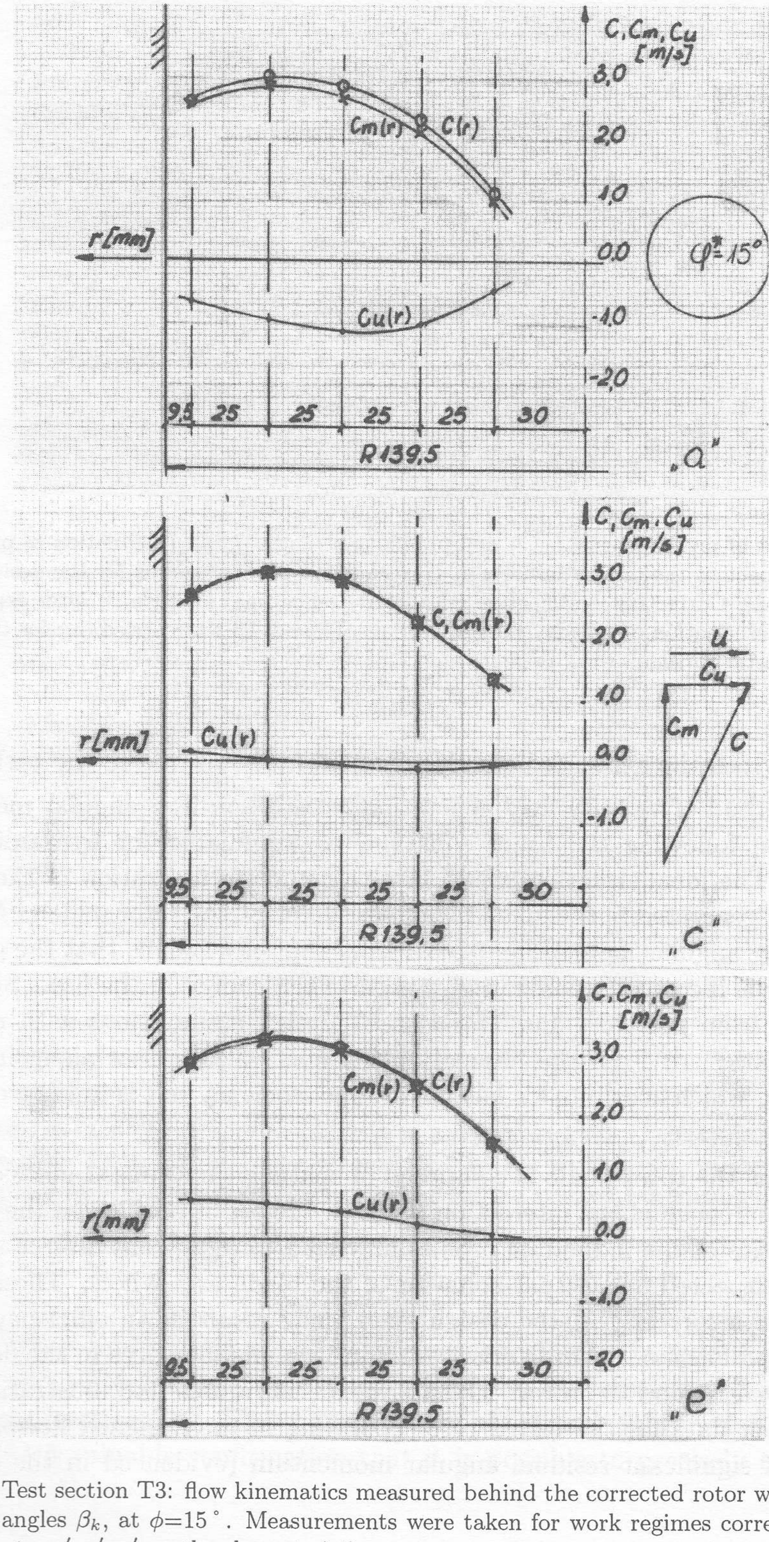


Figure 10. Test section T3: flow kinematics measured behind the corrected rotor with modified angles β_k, at ϕ=15°. Measurements were taken for work regimes corresponding to pts. a', c', e' on the characteristic.

by tangential velocity component C_u) at the exit (Fig. 8) signifies that rotor geometry is not adequate for the actual conditions, despite the choice of flow parameters matching the best point of the characteristic. The distribution of angular momentum in section T3, when referenced to the originally generated angular momentum distribution before the rotor, enables one to introduce specific (with respect to place, quantity, and direction) corrections to the design parameters governing the geometry of the rotor blade. The process of corrective modification is illustrated in Fig. 9 for a representative parameter of blade geometry [3].

Fig. 5 shows the resulting effect of the blade shape correction on the efficiency values in vicinity of the characteristic optimum. Fig. 10 below shows the kinematic state of flow at test section T3 after the correction. The diagram may serve as basis for a decision regarding possible further corrective steps, to be taken either in continuing model research or in designing the working rotor for actual installation.

5 Conclusions

1. A turbine with a novel flowpath design (for which patent protection was obtained) was prepared in concept, built in model, and its universal characteristic was obtained from the model. Operating parameters of the presented design (along with its possible modifications) make it particularly suitable for small-scale powerplants.

2. A method was proposed for precise verification of design assumptions for configuring the shape of rotor blading, as well as a corrective procedure for optimizing the blade shape, on the basis of model measurements of flow kinematics at rotor exit, so as to enable more complete utilization of flow energy.

3. For the designed model, the results of carrying out a one-step optimizing corrective procedure (as above) were demonstrated to markedly improve efficiency as shown on the characteristic plot. The character of attendant changes in the kinematics of the exit flow were also shown.

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