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exist for the publication of theoretical and experimental investigations of all aspects of the mechanics and thermodynamics of fluid-flow with special reference to fluid-flow machines

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poświęcone są publikacjom naukowym z zakresu teorii i badań doświadczalnych w dziedzinie mechaniki i termodynamiki przepływów, ze szczególnym uwzględnieniem problematyki maszyn przepływowych

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Identification of the effects of the flow through the holes in the rotating wheel of an impulse turbine[2]

Some results of the investigations carried out at the Institute of Turbomachinery, Technical University of Łódź, on the test stand of the TM1-3 model turbine, with special attention paid to the side flows (secondary air flow system) in the chamber stages are discussed. In the experimental investigations an approach concerning the division of the main flow through the vanes and blade cascades into three zones was addressed. This approach was developed along with the simulation calculations.

The paper is focused on an attempt to identify the flow through discharge holes in the turbine wheel by means of dynamic measurements of pressure variations in the chamber behind the rotor disc.

Nomenclature

Dpz	–	inlet overpressure,	u	–	circumferential velocity,
h	–	enthalpy of the medium	x/l	–	relative blade length,
m	–	mass flow,	α	–	hole time, angle,
n	–	frequency of rotation,	ν	–	modified Parsons number; $\nu = \frac{u_2}{\sqrt{2\Delta h_s}}$,
Pa	–	atmospheric pressure,	Θ_w	–	dynamic reactivity:
p	–	pressure,			on the inner wall: $\Theta_{WIw} = \frac{h_{Iw2s} - h_{Iw1}}{h_{Iw2s} - h_{Iw0}}$.

Subscripts

c	–	total value (enthalpy, pressure, temperature),	u	–	circumferential component,
			wD	–	inner endwall,
s	–	isentropic, static	z	–	axial component, outer endwall,
o	–	inlet,	I	–	refers to the first stage,
sr	–	average,	II	–	refers to the second stage.

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[2]The work was sponsored by the State Committee for Scientific Research Project No. 9 S603 03506 [Gundlach et al. 1997.]

1. Introduction

Formation of side flows in the stages of the impulse turbine of the chamber type is a very complex phenomenon, to which many theoretical and experimental studies have been devoted [Gundlach 1951, 1984, Dibelius 1983, Owen 1977, 1989, Wittig 1996], and has many common problems with the secondary air-flow systems in the high-temperature gas turbines.

The research conducted in the years 1996-1997 at the laboratory of the Institute of Turbomachinery, Technical University of Łódź, was also devoted to this topic. The tests were carried out on the test stand of the TM1 model turbine [Porochnicki at al, 1989]. The holes of proper dimensions made in the discs have a dominant significance in the process of controlling side flows.

2. Test stand and the turbine stages investigated

The test stand of the TM1 model turbine makes it possible to investigate radial stages, groups of radial and axial stages and groups of axial stages. It is equipped with two shafts in independent bearings, which allow one to measure the rotor moments neglecting the friction losses in the bearings. The torque of the rotors is collected by the disc water brakes which allow one to adjust the frequency of rotation and the moment in wide ranges.

The design properties of the test stand and the instrumentation used made it possible to conduct measurements of the distributions of thermodynamic properties in the rotor disc chambers and to carry out measurements by means of a special spherical microprobe with a thermocouple in the outlet plane of the rotor. Owing to the independent bearing system of both shafts, it was possible to measure the power of each stage separately, and in some cases both rotors could rotate at specified various frequencies of rotation. Such a configuration of the operation of both stages allowed one to obtain the required kinematics of the flow coming to the second stage of the turbine, which was the main object of research and was equipped with a better instrumentation.

The test stand was supplied with the compressed air of the overpressure up to 156 kPa, the mass flow up to 8 kg/s and the temperature up to 140^0C from a laboratory compressor-station. The maximum air mass flow rate did not exceed 4 kg/s during the testing.

The stand measurement system allows one to conduct fully automatic measurements using the remote-controlled ARELS 2 system [Sosin-Łabudź et al, 1991] for data recording and processing, whose task was to control supports moving the probes and to record pressure and temperature from the high-grade transducers. The ARELS 2 measurement system recorded also mechanical parameters, such as the frequency of rotation and the value of the torque.

The operating parameters of the stages under investigation were recorded as time-averaged results of the recordings from tens of samples of a given signal (of pressure, temperature or of mechanical parameters such as the torque and the frequency of rotation of both the rotors).

An electronic miniature pressure transducer with a measuring bridge incorporated in the silicone membrane, built in the stage casing just behind the rotor discharge holes, was a very useful measuring tool, which made it possible to extend the interpretation of the flow through the discharge holes in the rotor. Recordings of the signal from this transducer show the alterations in the unsteady pressure in the chamber behind the rotor and allow one to observe discussed above effects on the flow in the holes in the rotating wheel. This transducer was located flat in the wall in the distance of about 7 millimetres from the rotor disc. The transducer was chosen following the instructions given by Hagel, 1975 . The basic parameters of the signal measured are shown in Fig. 1. The time between passing the successive holes in the rotor disc was marked by T, while the time required by a discharge hole to pass in the front of the pressure transducer was marked by α (both parameters were calculated from the geometry and the frequency of rotation). At a constant frequency of rotation, the ratio α/T depends only on the diameter of the discharge hole. As a trigger was not used during the tests, the exact position of the beginning of the hole time α is not possible to be marked on the diagrams.

The meridional cross-section of the TM1-3 group of stages under investigation

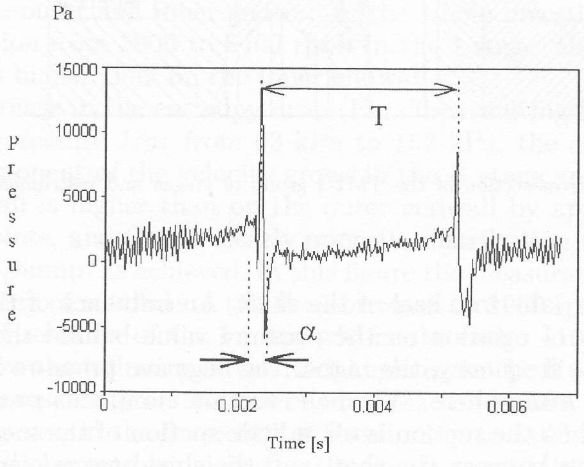


Fig. 1. Unsteady pressure changes behind the rotor discharge holes.

is shown in Fig. 2. The basic geometric data of the stages under investigation are as follows: average blade diameter – 526 mm, average blade height in the group of stages – 25.5 mm, and average stator wheel outlet angle – 14^0.

Ten time-averaged static pressure taps, located along the radius at every 10 mm, were installed on the back wall of the diaphragm and on the wall of the casing behind the rotor. The variable parameters were as follows: supply pressure, rotor frequency of rotation and air suction from the space behind the shield and

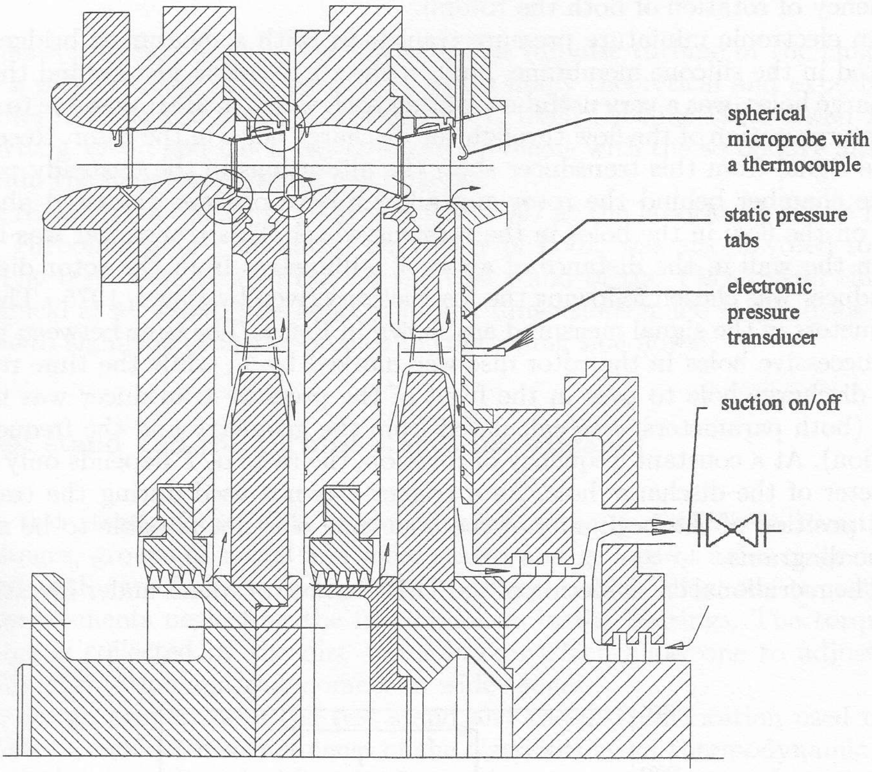


Fig. 2. The meridional cross-section of the TM1-3 group of stages and instrumentation used during investigations.

in front of the outer labyrinth seal of the shaft. An influence of the variations in the rotor frequency of rotation on the pressure value behind the rotor is following: the higher the frequency, the higher the negative pressure in the chamber with respect to the atmosphere. When the suction is on, this pressure difference becomes higher. When the suction is off, a little suction of the medium from outside through the gap between the shaft and the shield takes place. The suction changes the direction of the air flow through this gap to an opposite one.

In the chamber in front of the second disc, the occurrence of the suction or its lack (in this configuration of the test stand) does not exert influence in any noticeable way on the pressure distribution. The dominant influence on the pressure in the chamber in front of the second rotor has, first of all, the supply pressure, and to a smaller degree – the frequency of rotation of the rotor.

On the basis of the recording of the difference of static pressure on the radius of the discharge holes, one can estimate the medium mass flow through the holes and to determine the reference pressure in the region behind the rotor on the hole radius with respect to the atmosphere. During each measurement, 4096 recordings of the signal from the transducer were taken, with the sampling frequency equal to 50 kHz.

The averaged results of the measurement were stored in the computer memory and then processed. An exception from this rule were the dynamic measurements of the pressure with an electronic pressure transducer. In this case, a whole series of signal recordings from the transducer were recorded with the maximum frequency of 50 kHz, which makes it possible to represent the changes of the pressure in the chamber behind the rotor reliably enough.

In the experimental investigations, the TM1-3 blading with a geometry corresponding to the real high rotation steam turbines was used. The vanes and rotor blades were cylindrical. A characteristic feature of the blading under investigation was an apparent trend of the designer to limit maximally the losses in mixing the labyrinth leakage with the main flow by means of special, recommended in many literature sources, modelling of the design parts (marked by O in Fig. 2) and to extend the radial and axial seals [Deicz et al, 1960], which turned out to be disadvantageous. Also the guide vanes with a characteristic „hump" meridional cross-section, whose task is to decrease the pressure gradient along the vane length, were used for this aim. Fig. 3a represents an example of the changes in the dynamic reactivity of the rotors during the operation of both shafts with the same frequency of rotation. In the II stage, the dynamic reactivity is in practice the same on the outer and inner endwall in the whole investigated range of frequency of rotation from 2900 to 5400 rpm. In the I stage, the reactivity on the outer endwall is higher than on the inner endwall.

With an increase in the enthalpy drop (Fig. 3b) resulting from an increase in the supply overpressure Dpz from 63 kPa to 152 kPa, the effect forced by the centripetal component of the velocity grows in the II stage and the reactivity on the inner endwall is higher than on the outer endwall by approximately 1.5 to 2 percentage points, and thus a clearly opposite distribution than in the case of wheels without „hump" is achieved. In this figure the measurement results for the constant frequency of rotation of the II rotor equal to 5900 rpm at the change in the frequency of rotation of the I rotor within the range 4100 to 7100 rpm are presented to show the influence of the inlet angle α_0 in the II stage. Here one can also see that the reactivity on the inner endwall is still higher. It can also be seen that with an increase in the frequency of rotation up to 7100 rpm reactivities on both the endwalls become equal in the I stage. It is an intended characteristic feature of the shape of the used guide vanes, which force the centrifugal component.

The tests were carried out at the frequency of rotation from 7200 rpm to approx. 2700 rpm. The nominal overpressure at the turbine inlet was changed in the range from 156 kPa to 46 kPa. The nominal frequency of rotation was determined by the manufacturer of the stages to be 6000 rpm, and the nominal overpressure, supplying steam to each of the stages, was approx. 55 kPa.

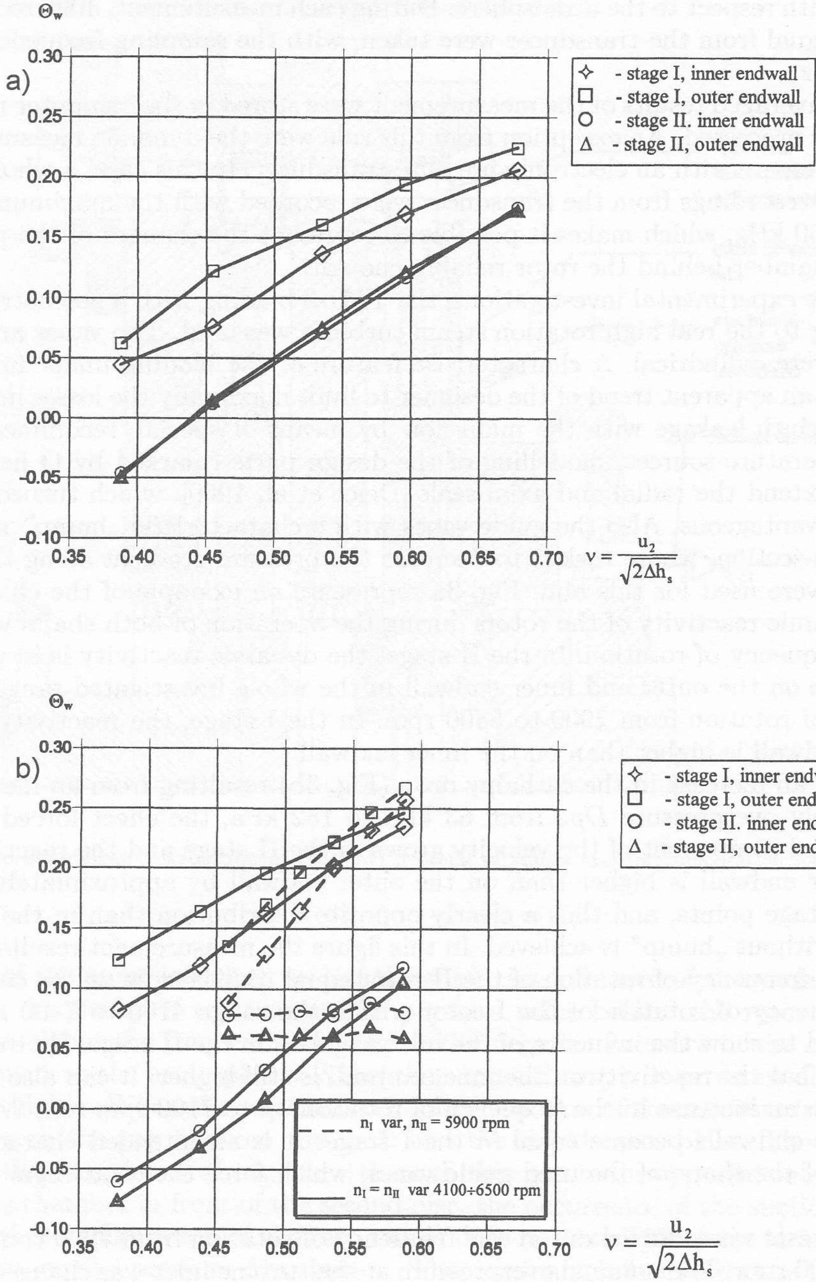


Fig. 3. Characteristics of the thermodynamical reactivity of the rotor $\Theta_w(\nu)$; a) frequency of rotation of both rotors from 2900 to 5400 rpm, Pa=100220 Pa, Dpz = 63 kPa; b) frequency of rotation of the rotor I from 7100 to 4100 rpm, rotor II 5900 rpm, and frequency of rotation of both rotors from 4100 to 6500 rpm, Pa = 100220 Pa, Dpz = 152 kPa.

3. Methods for affecting the side flows in a chamber stage

Side flows in a stage of the chamber turbine can be directed along different ways, depending on the structure and loading of the stage. The design of the TM1-3 model turbine makes it possible to control the propagation of the side flows in the stage by means of changes of some design parameters of the investigated stages (for instance by decreasing the diameter of the discharge holes up to their complete closure, changes of the clearance in the labyrinth seal under the stator of the stage) or changes of the operating point of the stages by means of variations in the frequency of rotation, which affects directly the change of the rotor dynamic reactivity. The diameter of the discharge holes was changed by screwing aluminium sleeves of a specified outer diameter into the threaded holes of the diameter 23.5 mm, already existing in the rotor discs. The minimum diameter of such a discharge hole was 18 mm. The edges of the holes were smoothed only by a small radius of curvature.

The quantity of the suction of the medium from the chamber between the II rotor and the shield was also changed, which simulated the presence of the next stage. The structure of the labyrinth seals in the model turbine allows one to simulate the operating conditions of the second stage as an intermediate stage or the last stage in the turbine casing. By the suction from the chamber behind the rotor, a part of the medium coming from the holes in the rotor disc can flow in the centripetal direction, simulating the existence of the next diaphragm. When the suction is stopped or is minimal, it causes a change in the direction of the flow in the chamber behind the rotor. The rotating wheel operates as a disc-friction centrifugal fan, sucking the medium which mixes with the medium flowing through the holes in the disc from the outside. In the case these holes are lacking, the medium is directed through the circumferential gap at the inner endwall behind the rotating wheel and mixes with the main stream, in such a way as it may occur in the last stage.

The mass flow sucked from the space behind the last seal was changed from zero (then a little suction of the air from outside occurred owing to the suction activity of the rotor disc rotating in a narrow chamber) up to 0.01 kg/s, which was approx. 0.27% of the main flow.

The amount of the leakage under the stator of the investigated II stage was changed by means of an exchange of the cylindrical insert imitating the turbine shaft into another one with a smaller nominal diameter. The size of the labyrinth gap was increased from 0.25 to 0.45 mm. The influence of these changes could be observed on the diagrams of the circumferentially averaged parameters measured with a spherical microprobe behind the rotor of the II stage, especially in the inner and outer endwall.

4. Programme of tests

The tests were carried out on the TM1-3 turbine in five basic measurement series marked from F to L. In the series F-K, two whole stages were investigated. In the series L, the system containing 1.5 stages was tested in order to define the pressure and temperature fields and the velocity vector behind the stator of the II stage. Apart from the basic tests at the rotation frequency $n_I = n_{II}$, the tests of the operation of the second stage, while changing the frequency of rotation n_I of the first stage, were carried out as well.

The dynamic reactivity of the rotor, changing with the frequency of rotation – at the outer and inner endwalls, respectively – exerted a significant influence on a formation of the side flows in the stages, and also on the flow through the discharge holes in the turbine rotors. A simplified programme of the tests for the TM1-3 blading is shown in Tab. 1.

Table 1.

Test series	Blading	Object of tests	Number of series and kind of tests
F	TM1-3	Holes $\phi 18$	$7 \times Dpz$ idem, $n_I = n_{II}$ var;
H	TM1-3	Holes $\phi 23.5$, suction*)	$15 \times Dpz$ idem, $n_I = n_{II}$ var; $2 \times Dpz$ idem, n_{II} idem, n_I var;
J	TM1-3	Holes $\phi 23.5$, increased clearance of the labyrinth	$10 \times Dpz$ idem, $n_I = n_{II}$ var; $1 \times Dpz$ idem, n_{II} idem, n_I var;
K	TM1-3	Without holes increased clearance of the labyrinth, suction*)	$6 \times Dpz$ idem, $n_I = n_{II}$ var;
L	TM1-3	Stator of the stage	Dpz idem, n_I idem

*) In the test series G, J and K the suction from the seal chamber behind the II stage rotor was used, simulating its operation in a group of stages.

5. Test results

Selected results of the tests, focused mainly on the illustration of the dynamic characteristics of the pressure changes in the chamber behind the rotor of the second stage, are shown on the diagrams.

In Figs. 4a and 4b, examplary changes in the dynamic signal from the transducer for two different frequencies of rotation of the turbine rotors with discharge holes with a minimal diameter (test series F) are presented, while Tab. 2 includes examples of the basic operating parameters of the turbine and pressures in the chambers of the stage.

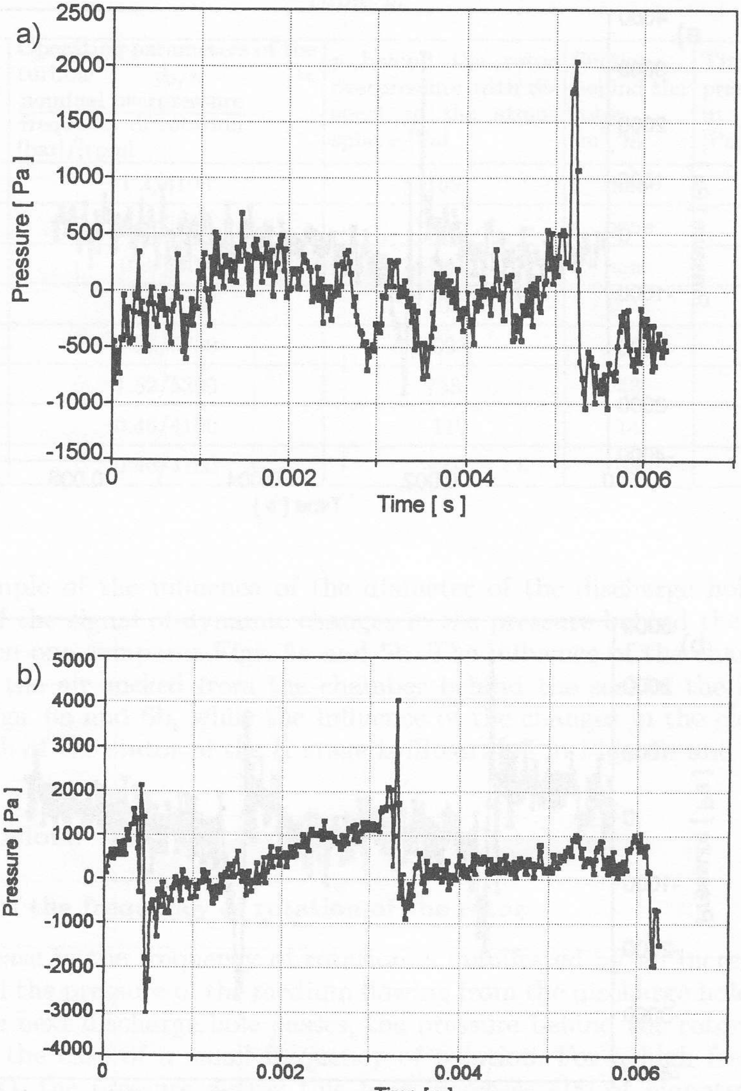


Fig. 4. a) F04-07 series; overpressure at the inlet 0.8 bar; $n = 3500$ rpm, average pressure value behind the rotor with respect to the atmosphere $p = 72$ Pa. b) F04-08 series; overpressure at the inlet 0.8 bar; $n = 6900$ rpm, average pressure value behind the rotor with respect to the atmosphere $p = -342$ Pa.

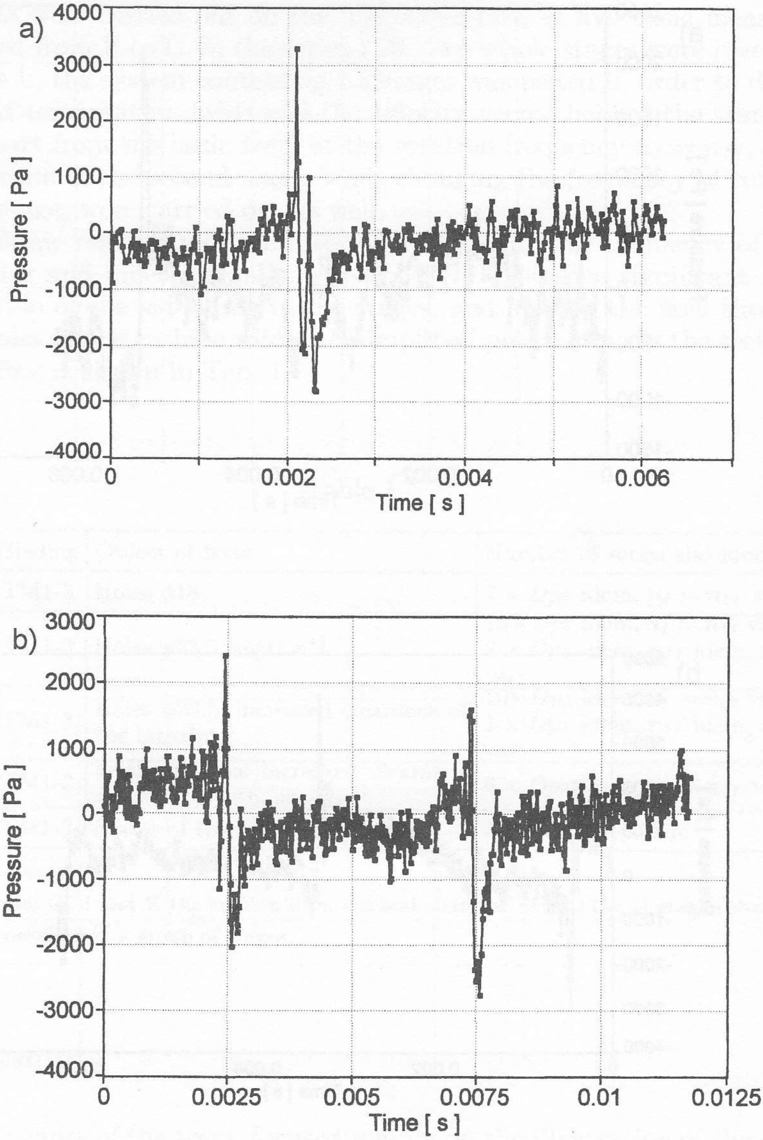


Fig. 5. a) F01-01 series; overpressure at the inlet 1.3 bar; $n = 4100$ rpm, average pressure value behind the rotor with respect to the atmosphere $p = 169$ Pa. b) H01-01 series; overpressure at the inlet 1.3 bar; $n = 4100$ rpm, average pressure value behind the rotor with respect to the atmosphere $p = 104$ Pa.

Table 2.

Name of the series	Operating parameters of the turbine p_0/n = $\frac{\text{nominal overpressure}}{\text{frequency of rotation}}$ [bar]/[rpm]	p behind the rotor overpressure with respect to the atmosphere [Pa]	Suction behind the rotor [m^3/h]	Time-averaged pressure drop in the hole [Pa]
F01-01	1.3/4100	169	none	5321
F04-07	0.8/3500	72	none	3068
F04-08	0.8/6900	-341	none	8572
H01-01	1.3/4100	104	0.75	3636
J04-11	1.52/5300	1023	28	7257
J08-26	1.52/5300	758	43	7288
H13-41	0.46/4100	-119	14	2666
J07-23	0.46/4100	-27	28	2916

An example of the influence of the diameter of the discharge holes on the variations of the signal of dynamic changes in the pressure behind the rotor can be seen when one compares Figs. 5a and 5b. The influence of the change in the quantity of the air sucked from the chamber behind the seal of the II rotor is shown in Figs. 6a and 6b, while the influence of the changes in the clearance in the labyrinth of the stator of the II stage is illustrated in Figs. 7a and 7b.

6. Conclusions

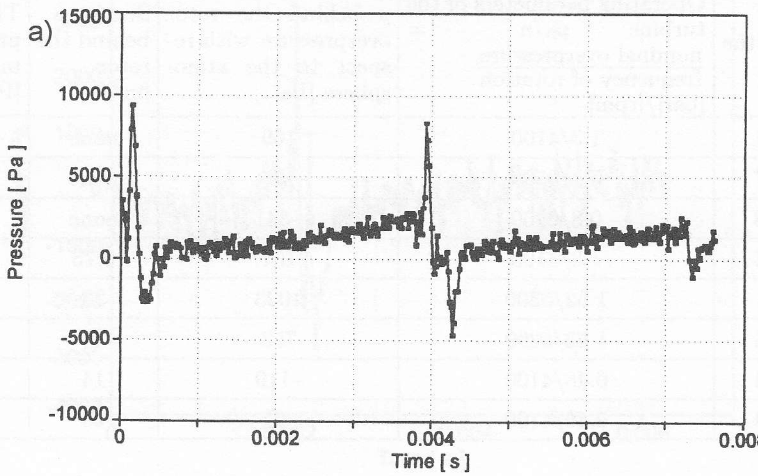
Variations of the frequency of rotation of the rotor

An increase in the frequency of rotation is manifested by an increase in the amplitude of the pressure of the medium flowing from the discharge holes. During the time the next discharge hole passes, the pressure behind the rotor is almost constant in the case of a small frequency of rotation. For a high frequency of rotation (F8), the pressure during this time increases almost monotonically by approx. 3 kPa, and when the hole passes, it rapidly decreases. The average value of the pressure behind the rotor is at the same time by 5500 Pa higher than in the case of a small frequency of rotation.

Variations of the diameter of the discharge holes

The signals from the transducer are similar to one another, both at the diameter of the discharge holes $d_0 = 18$ mm and $d_0 = 23$ mm.

The pressure differences between its maximum value (pressure peaks) and the



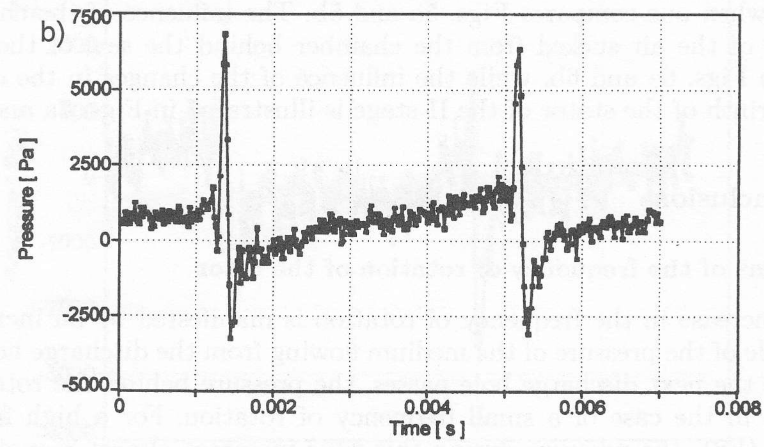


Fig. 6. a) J01-11 series; overpressure at the inlet 1.52 bar; $n = 5300$ rpm, average pressure value behind the rotor with respect to the atmosphere $p = 1023$ Pa. b) J08-26 series; overpressure at the inlet 1.52 bar; $n = 5300$ rpm, average pressure value behind the rotor with respect to the atmosphere $p = 758$ Pa.

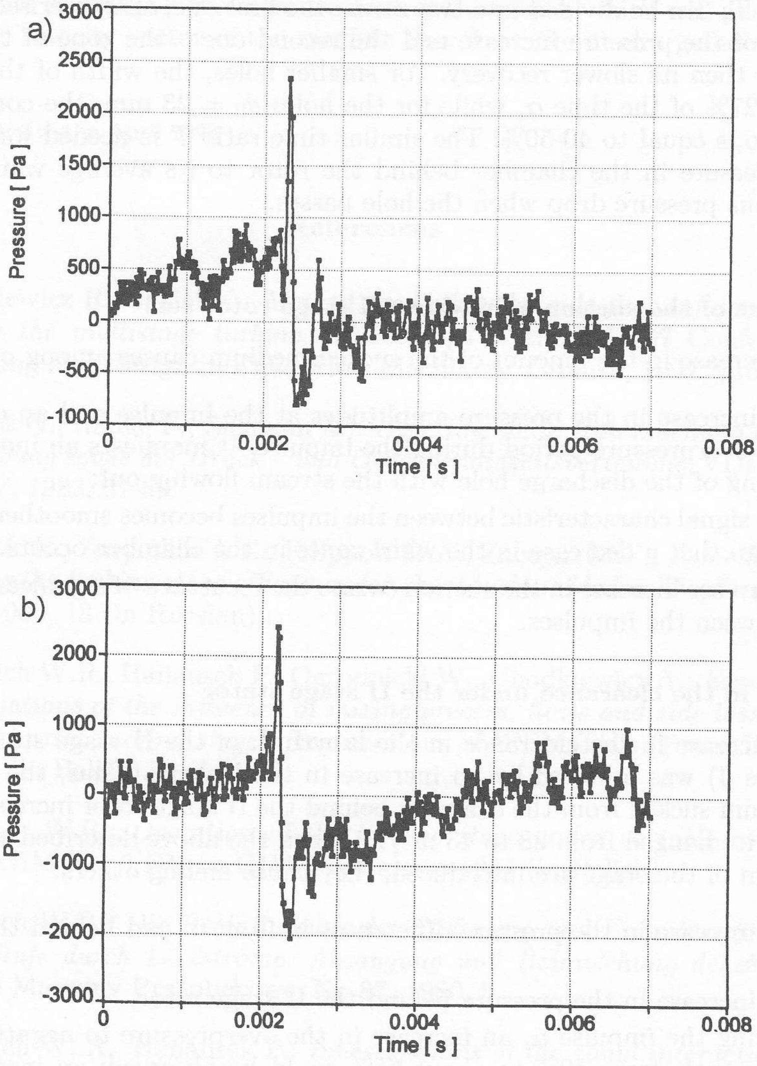


Fig. 7. a) H13-41 series; overpressure at the inlet 0.46 bar; $n = 4100$ rpm, average pressure value behind the rotor with respect to the atmosphere $p = -119$ Pa. b) J07-23 series; overpressure at the inlet 0.46 bar; $n = 4100$ rpm, average pressure value behind the rotor with respect to the atmosphere $p = -27$ Pa.

average value, recorded by the dynamic transducer, are very close to the pressure differences measured as time-averaged values in front of and behind the rotor on the radius of the discharge holes.

The impulse time α, which for the series F is $0.049 \times T$, and for the series H – $0.062 \times T$, can be divided into two zones: the first one, characterised by a large gradient of the pressure increase and the second one – the zone of the pressure drop and then its slower recovery. For smaller holes, the width of the first zone is about 27% of the time α, while for the holes $d_0 = 23$ mm, the corresponding time ratio is equal to 40-50%. The similar time ratio T is needed for a recovery of the pressure in the chamber behind the rotor to its average value after the momentous pressure drop when the hole passes.

Variations of the suction behind the II stage rotor seal

An increase in the amount of the sucked medium causes among others:

- an increase in the pressure amplitudes at the impulse and an extension of the overpressure period during the impulse. It manifests an increase in the filling of the discharge hole with the stream flowing out;
- the signal characteristic between the impulses becomes smoother, which can mean that a decrease in the whirl zones in the chamber occurs.
- a further increase in the suction causes the recurence of significant pulsation between the impulses.

Increase in the clearance under the II stage stator

An increase in the clearance in the labyrinth of the II stage stator (tests in the Series J) was followed by an increase in the leakage. Thus, the quantity of the medium sucked from the chamber behind the II stage rotor increased – in the Series J, it changed from 28 to 43 m^3/h. When the above described variations in the design of the stage are introduced, they cause among others:

- an increase in the pressure differences in front of and behind the discharge holes,
- an increase in the pressure behind the II stage rotor,
- during the impulse α, an increase in the overpressure to negative pressure time ratio from 40% to about 65% could be observed; an increase in the quantity of the leakage causes an occurrence of higher pressure pulsation between the impulses, especially at the end of the period T,
- in both the cases, i.e. before and after the clearance is increased, the value of the pressure between the impulses increases uniformly for large values of the supply pressure and high frequencies of rotation of the rotor; for smaller values of the operating parameters of the rotor, the pressure characteristic between the impulses is nearly smooth.

The analysis of the dynamic pressure changes behind the discharge holes, with reference to the averaged values of the pressure in the chambers in front of and behind the rotor, makes it possible to reach many vital conclusions concerning the formation of the flow through the holes. It seems useful to conduct similar measurements along with the measurement of the phase, which will allow one to evaluate the velocity vector of the medium flow from the discharge holes.

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Identyfikacja oddziaływania przepływu przez otwory w kole wirującym turbiny komorowej

Streszczenie

Przedstawiono niektóre wyniki badań przeprowadzonych w Instytucie Maszyn Przepływowych Politechniki Łódzkiej na stanowisku turbiny modelowej TM1, ze szczególnym uwzględnieniem przepływów ubocznych w stopniach komorowych. W badaniach doświadczalnych nawiązano do koncepcji podziału przepływu głównego przez wieńce stopni na trzy strefy, którą wykorzystano w obliczeniach symulacyjnych.

W artykule skoncentrowano się na próbie identyfikacji przepływu przez otwory odciążające w kole turbinowym za pomocą dynamicznych pomiarów zmian ciśnienia w komorze za tarczą wirnika.