

P O L S K A A K A D E M I A N A U K
I N S T Y T U T M A S Z Y N P R Z E P Ł Y W O W Y C H

P R A C E
I N S T Y T U T U M A S Z Y N
P R Z E P Ł Y W O W Y C H

T R A N S A C T I O N S
O F T H E I N S T I T U T E O F F L U I D - F L O W M A C H I N E R Y

90-91

W A R S Z A W A — P O Z N A Ń 1989

P A Ń S T W O W E W Y D A W N I C T W O N A U K O W E

PRACE INSTYTUTU MASZYN PRZEPLYWOWYCH

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

*

THE TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

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 machinery

RADA REDAKCYJNA — EDITORIAL BOARD

TADEUSZ GERLACH · HENRYK JARZYNA · JERZY KRZYŻANOWSKI
STEFAN PERYCZ · WŁODZIMIERZ PROSNAK
KAZIMIERZ STELLER · ROBERT SZEWAŁSKI (PRZEWODNICZĄCY · CHAIRMAN)
JÓZEF ŚMIGIELSKI

KOMITET REDAKCYJNY — EXECUTIVE EDITORS

KAZIMIERZ STELLER — REDAKTOR — EDITOR
WOJCIECH PIETRASZKIEWICZ · ZENON ZAKRZEWSKI
ANDRZEJ ŻABICKI

REDAKCJA — EDITORIAL OFFICE

Instytut Maszyn Przepływowych PAN
ul. Gen. Józefa Fiszer 14, 80-952 Gdańsk, skr. pocztowa 621, tel. 41-12-71

Copyright
by Państwowe Wydawnictwo Naukowe
Warszawa 1989

Printed in Poland

ISBN 83-01-09017-0

ISSN 0079-3205

PAŃSTWOWE WYDAWNICTWO NAUKOWE - ODDZIAŁ W POZNANIU

Nakład 350+90 egz.	Oddano do składania 27 X 1987 r.
Ark. wyd. 27. Ark. druk. 19,5	Podpisano do druku w sierpniu 1989 r.
Papier offset. IV kl. B-1	Druk ukończono w styczniu 1990 r.
Zam. nr 2/90	K-8/295

DRUK ZAKŁAD POLIGRAFII WSP W ZIELONEJ GÓRZE

HYDROFORUM

KONFERENCJA NAUKOWO-TECHNICZNA

na temat

ZAGADNIENIA HYDRAULICZNYCH MASZYN WIROWYCH

Gdańsk-Władysławowo, 24-27 września 1985 r.

*

HYDROFORUM

SCIENTIFIC-TECHNICAL CONFERENCE

on

PROBLEMS OF HYDRAULIC TURBOMACHINES

Gdańsk-Władysławowo, September 24-27, 1985

*

ГИДРОФОРУМ

НАУЧНО-ТЕХНИЧЕСКАЯ КОНФЕРЕНЦИЯ

на тему

ПРОБЛЕМЫ ГИДРАВЛИЧЕСКИХ ТУРБОМАШИН

Гданьск-Владыславово, 24-27 сентября 1985 г.

ANDRZEJ KORCZAK, JANUSZ LAMBOJ

Institut Maszyn i Urządzeń Energetycznych Politechniki Śląskiej, Gliwice
(Institute of Power Machinery and Equipment of the Technical University of Silesia, Gliwice)

Ribbing of the Internal Front Wall of a Centrifugal Pump Casing as a Method for Axial Thrust Relief

This paper deals with the influence of internal front wall ribbing of a centrifugal pump casing on the axial thrust. The test stand is described and the results of investigations of the relation between the axial thrust and pump parameters on the ribbing of the casing front wall are presented. Also the results of the measurement of the axial thrust and pump parameters with changed outflow from behind the impeller and afflux behind the impeller are described.

1. Introduction

When designing multi-stage centrifugal pumps the method of balancing the axial thrust should be considered as an inseparable element of the pump hydraulic system significantly influencing the pump efficiency. One of the methods of balancing axial thrust consists in appropriate forming of the space between the casing walls and impeller discs aimed at increasing or decreasing the average angular velocity of the rotating liquid.

2. Axial Thrust as a Function of Specific Speed

In order to define the axial thrust, it should be assumed that:

- the pressure distribution at both sides of the impeller (Fig. 1) is axially symmetrical,
- the average angular velocity of the liquid in spaces *I* and *II* remains constant,
- there exists an equality of pressures at radius r_2 in spaces *I* and *II* with static pressure p_{r2} at the impeller outflow.

The axial thrust may be defined then by the formula:

$$\sum F = F_I - F_{II} - F_o - F_m + F_{at} \quad (1)$$

where: F_I , F_{II} denote forces resulting from pressure distributions in spaces *I* and *II* on both sides of the impeller, F_o stands for the force caused by pressure at the impeller outflow, F_m is a reaction caused by a change in liquid flow direction.

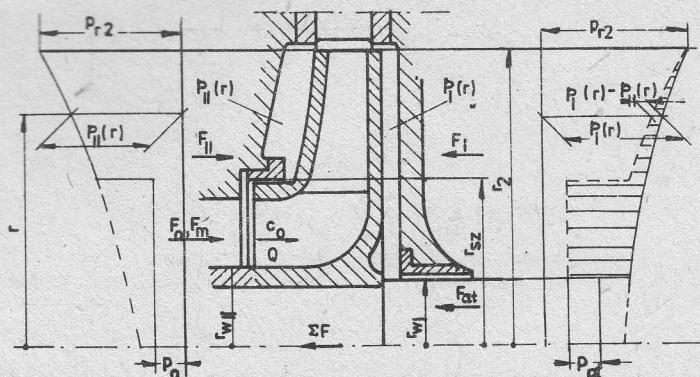


Fig. 1. Axial thrust components acting on the pump stage impeller set

The difference of forces $(F_I - F_{II}) = F_{I,II}$ results directly from the formula (1) and has the following form

$$F_{I,II} = \pi [p_{r2}(r_{sz}^2 - r_{wI}^2) + \Omega_{II}^2 \frac{\omega^2}{4} \rho (r_2^2 - r_{sz}^2)^2 - \Omega_I^2 \frac{\omega^2}{4} \rho (r_2^2 - r_{wII}^2)^2] \quad (2)$$

where $\Omega = \omega_c/\omega$ indicates the ratio of average angular velocity of liquid rotation to angular velocity of impeller, ρ stands for the liquid density.

Assuming that because of the ribbing of the casing's internal front wall $\Omega_{II} = 0$ (which has been confirmed by measurements of pressure distribution in space II) one obtains

$$(F_{I,II})_{\Omega_{II}=0} = \pi [p_{r2}(r_{sz}^2 - r_{wI}^2) - \Omega_I^2 \frac{\omega^2}{4} \rho (r_2^2 - r_{wI}^2)^2]. \quad (2a)$$

By subtracting the corresponding sides of equations (2) and (2a) the component of axial thrust due to casing wall ribbing (Fig. 2) is obtained as

$$F_{I,II} - (F_{I,II})_{\Omega_{II}=0} = \pi \Omega_{II}^2 \frac{\omega^2}{4} \rho (r_2^2 - r_{sz}^2)^2. \quad (3)$$

This component is directed opposite to the resultant axial thrust $\sum F$.

The axial thrust $\sum F$, acting on the pump stage impeller set may be defined by the following formula, which results from the dimensional analysis [2]:

$$\sum F = KY(2r_2)^2 \rho \quad (4)$$

where: Y is the unit energy head of the pump ($Y = gH$), K stands for a coefficient depending on the specific speed [3]:

$$K = 2(n_{sf} - 25) 10^{-3}. \quad (5)$$

Formula (5) was obtained by calculating from formulae (1) and (2) the axial thrusts for pumps with various specific speed, under assumption that $\Omega_I = \Omega_{II} = 0.5$ and taking into consideration the dimensions typical for single stage pumps.

Putting (5) into (4) one obtains:

$$\sum F = 2(n_{sf} - 25) Y(2r_2)^2 \rho 10^{-3}. \quad (6)$$

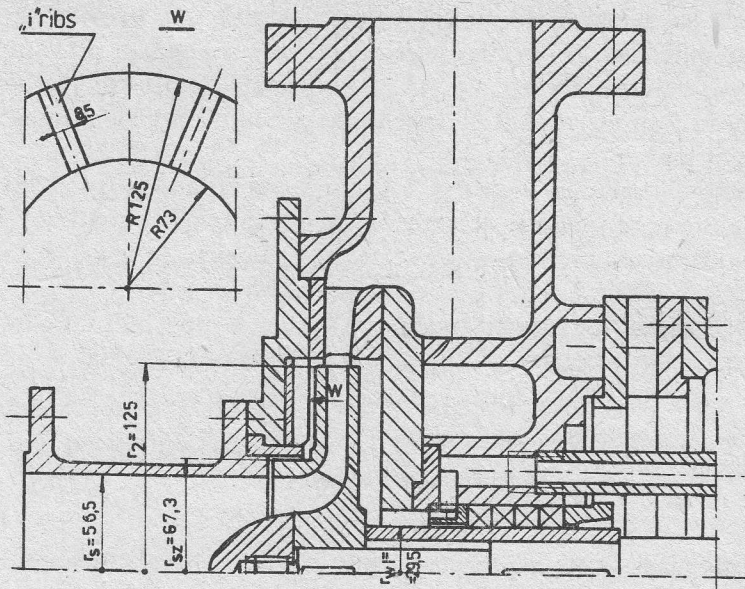


Fig. 2. Cross-section of the model pump [6]

The axial thrust will be hydraulically balanced if:

$$F_{I,II} - (F_{I,II})_{\Omega_{II}=0} = \sum F \quad (7)$$

that is when

$$\pi 0.45^2 \frac{\omega^2}{4} \rho (r_2^2 - r_{sz}^2)^2 = 2(n_{sf} - 25) Y (2r_2)^2 \rho 10^{-3} \quad (8)$$

In order to find specific speed n_{sf} of the pump stage in which the casing front wall ribbing will cause hydraulic balancing of the axial thrust, the following relations should be introduced [4]:

$$r_{sz} = r_2(0.3 + 0.0224 n_{sf}) \quad (9)$$

$$Y = \Psi \omega^2 r_2^2 = \frac{1}{2K_u^2} \omega^2 r_2^2 = \frac{\omega^2 r_2^2}{2(0.915 + 0.001125 n_{sf})^2} \quad (10)$$

After inserting (9) and (10) into (8) and several simplifications one obtains:

$$31.6[1 - (0.3 + 0.0224 n_{sf})^2]^2 - \frac{n_{sf} - 25}{(0.915 - 0.001125 n_{sf})^2} = 0. \quad (11)$$

The looked for solution of equation (11) is then:

$$n_{sf} = 47.75. \quad (12)$$

For larger values of n_{sf} , the resultant axial thrust in a pump with ribbed front wall will be directed towards the inlet, whereas for smaller values the direction it will be opposite.

Axial thrust described by formula (4) does not take into account the influence of the pressure difference at impeller inlet as well as the ambient pressure influence. This influence is described by $(p_o - p_a)\pi r_w l^2$. In practice, when calculating the axial thrust, the above formula should be taken into account only for pumps with axial inlet working at high inlet pressure.

In the presented analysis, the value Y in formulae (4) and (10) was taken for the optimum conditions of pump operation. In this situation in equation (7) a simplification is used consisting in neglecting reaction F_m , which for low-speed centrifugal pumps does not exceed a few per cent of the axial thrust value.

The analysis should be treated qualitatively since it does not take into account several other structural features of pumps which influence the axial thrust [3].

3. The Test Stand [5]

The investigations were performed using a model pump forming a stage of a multi-stage pump of OW-100A type. The model pump and its general dimensions are shown in Fig. 2.

A diagram of the measurement installation is shown in Fig. 3. The installation is a circulating pump system. The tested pump 1 operates with afflux from tank 2 and pumps water through pipeline 3 equipped with a control slide valve 4. A turbine flow-meter 5 is placed in the pipeline 3. The space behind the pump impeller is connected by tubes 6 and 7 with tank 2. Tube 6 is the drain, while pump 8 which forces

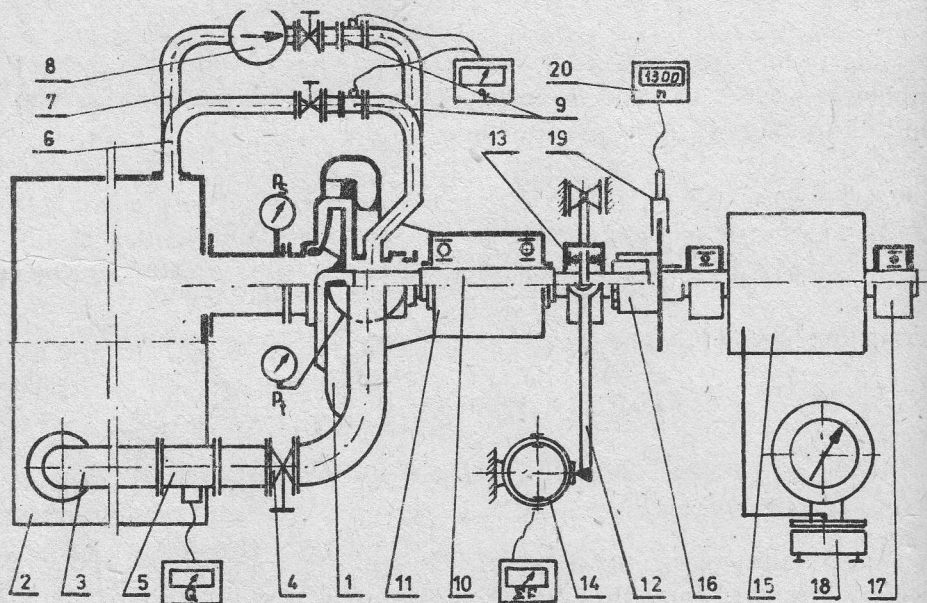


Fig. 3. Diagram of measurement system

afflux behind the impeller is placed on tube 7. Intensity of afflux and outflow from behind the impeller was measured by flow-meters 9.

The model pump shaft 10 is supported slidingly in the bearing bracket 11 and held by thrust bearing 13. The body of bearing 13 is set in lever 12 which rotates in the support bearing and the other end of the lever rests against the force converter 14. The converter 14 is a steel ring with eight strain gauges connected into an electric bridge [5] glued on. The pump shaft 10 is coupled with motor 15 by coupling 16 which does not transmit axial forces. A disc with holes is set on coupling 16 which cooperates with the photoelectric converter 19 for measuring rotational speed. This converter is connected with meter 20. The motor 15 is supported in bearings in cradle 17 and its arm rests against the balance 18. Measuring the reaction of the motor arm with the balance and measuring the rotational speed allows to determine the net torque and shaft power N . In order to find the effective unit work of the pump Y pressure was measured at the suction and discharge nozzles of the pump.

4. Results of Measurements and Conclusions

The maximum efficiency attained by the model pump was about 68% for the speed range $n = 2300 - 2600 \text{ min}^{-1}$. The basic measurements were carried out for $n = 2350 \text{ min}^{-1}$, i.e. for angular velocity $\omega = \pi n / 30 = 246.1 \text{ s}^{-1}$.

During the tests, the model pump operated at submergence head $H_{zs} = 0.3 - 0.5 \text{ m}$. Therefore its influence on the axial thrust could be neglected.

Measurement errors resulted from the class of used meters and had following values:

— for pressure	p :	0.4—0.6%
— for pump discharge	Q :	1.5%
— for outflow and afflux intensity behind the pump impeller	q :	2.5%
— for rotational speed	n :	1%
— for pump shaft power	N :	1%
— for axial thrust	$\sum F$:	2.5%

The influence of the number and height or ribs on the axial thrust and pump parameters was tested. Also the relation between these values and the intensity q of outflow and afflux behind the pump impeller was investigated. Results of the measurements were analyzed for the pump discharge Q at which the pump attains maximum efficiency, and related to the following dimensionless numbers:

— flow factors:

$$\Phi = \frac{Q}{\pi \omega r_2^3}, \quad (13)$$

$$\varphi = \frac{q}{\pi \omega r_2^3}, \quad (13a)$$

— unit pump work factor:

$$\psi = \frac{Y}{\omega^2 r_2^2}, \tag{10a}$$

— power factor (Newton number)

$$v = \frac{N}{\pi \omega^3 r_2^5 \rho}, \tag{14}$$

— pump efficiency η ,

— axial thrust number

$$X = \frac{\sum F}{4\omega^2 r_2^2 \rho}. \tag{15}$$

By comparing formulas (15) and (14) one realizes that for $\psi = 1$, $X = K$.

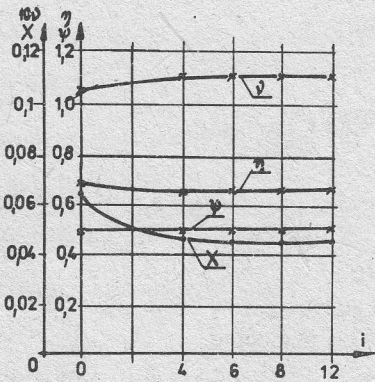


Fig. 4. Influence of the number of ribs on pump operation parameters

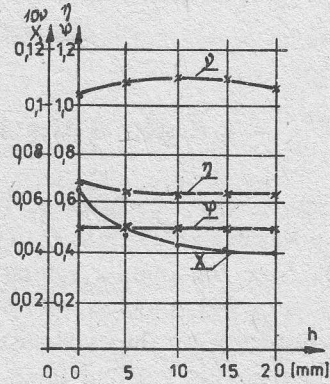


Fig. 5. Influence of the rib height on pump operation parameters

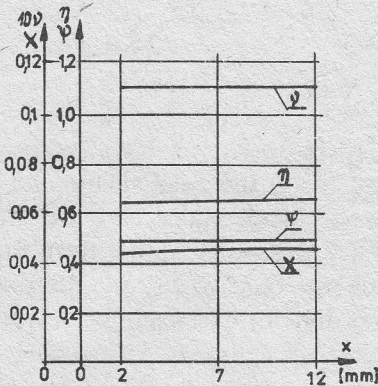


Fig. 6. Influence of the distance x of the ribs from pump impeller on pump operation parameters

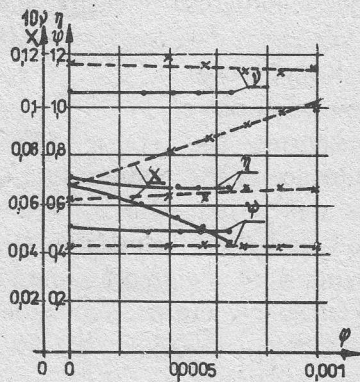


Fig. 7. Dependence of pump operation parameters on the intensity of outflow and afflux behind the impeller

— outflow from behind the impeller,
 - - - afflux behind the impeller

Fig. 4 shows the influence of the number of ribs on the remaining parameters. The rib height was $h = 10$ mm. It results from Fig. 4 that even with only four ribs the axial thrust is decreased by about 30% and while the efficiency decreases by 2.5% due to the similar increase of the power consumption. Unit pump work grows only slightly. The increase of the ribs number causes only small changes of measured parameters.

The influence of rib height on the remaining parameters is shown in Fig. 5. The number $i = 8$ ribs was assumed and their height was changed between 0 and 20 mm by a 5 mm step. The greatest influence was observed in the 0–10 mm range. Further increase of rib height had little influence.

In Fig. 6 the influence of the distance x of ribs from the impeller on axial thrust and other pump parameters is shown. 8 ribs of 10 mm height were installed. It was observed that in the range of distances $x = 2 \div 12$ mm the influence on axial thrust is very small.

The dependence of axial thrust and other pump parameters on the intensity of outflow and afflux behind the impeller is shown in Fig. 7. Outflow and afflux intensity stayed in the range of a few per cent of the rated pump capacity, i.e. in the range typical in practice [8]. It was observed that in the investigated range the outflow from behind the impeller decreases the axial thrust by several per cent, while the afflux results in an increase of axial thrust by several per cent. The remaining pump work parameters were less influenced by flow behind the impeller, affecting mainly the volumetric efficiency of the pump. The effect of outflow was investigated at $n = 2350 \text{ min}^{-1}$, and the effect of afflux was tested at $n = 1750 \text{ min}^{-1}$.

In Fig. 8, the tested pump complete characteristics at $n = 2350 \text{ min}^{-1}$ is presented without and with ribbing of the casing wall with eight ribs of $h = 10$ mm height. From the curves $v = f(\varphi)$ it results that ribs cause an increase of losses due to friction of the rotating impeller discs, which is comparable with losses resulting from other relief methods and which lowers maximum efficiency by about 3%.

In pumps with no relief, the axial thrust is close to the value calculated from formulae (1) and (2) with the assumption that $\Omega_I = 0.4$ and $\Omega_{II} = 0.5$, and is by about 30% higher than that calculated when $\Omega_I = \Omega_{II} = 0.45$ [5].

Under assumption that $\Omega_I < \Omega_{II}$ the formula (5) gives larger values of coefficient K and the solution of equations (3), (4), (5) and (7) results in a smaller value of n_{sf} .

The specific speed of the test pump calculated for parameters corresponding to its maximum efficiency was $n_{sf} = 58.8$. Ratio $r_{sz}/r_2 = 0.538$.

A stage of pump OS-80 of specific speed $n_{sf} = 50.5$ in which $r_{sz}/r_2 = 0.4$ was also investigated [7]. In Fig. 9 are presented its characteristics at $n = 1450 \text{ min}^{-1}$ for casing without and with ribbing. A relatively smaller increase of power consumption and drop of efficiency due to wall ribbing was noticed. At the rated point of the stage operation the axial thrust decrease exceeding 60% was observed.

It follows from the presented investigations results that the decrease of the specific speed increases the effectiveness of axial thrust release by casing wall ribbing and decreases its negative influence on the pump efficiency. This conclusion requires however confirmation by further investigations. It results both from calculations and

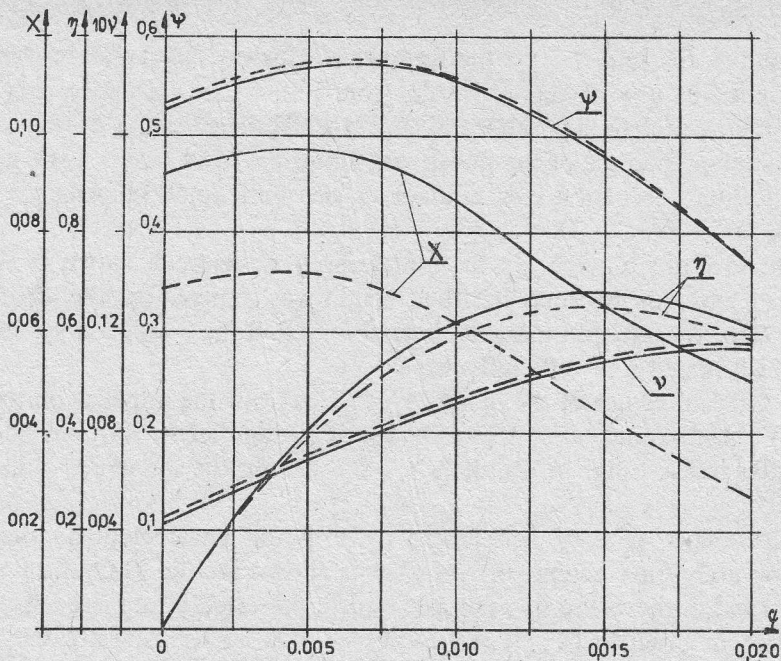


Fig. 8. Complete characteristics of the pump tested with $n_{sf} = 58.8$

— without ribs, - - - with ribs

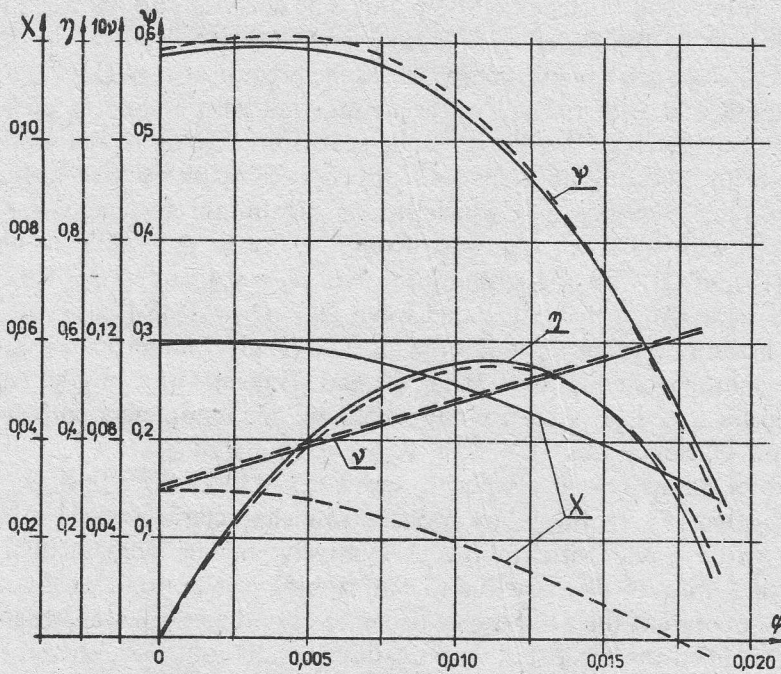


Fig. 9. Complete characteristics of the OS-8C pump stage with $n_{sf} = 50.5$

— without ribs, - - - with ribs

measurements that axial thrust decreases with the drop of r_{sz}/r_2 . Balancing axial forces by casing wall ribbing may be applied to the pumps with specific speed in the lower part of the range assumed for low speed centrifugal pumps.

Further research should be directed at developing a multistage centrifugal pump in which the casing wall ribbing will reduce axial thrust to a value which can be carried by the bearing.

References

- [1] A. Korczak, J. Rokita, *Pompy i układy pompowe* (Pumps and the pump systems). Skrypt (Textbook), Politechnika Śląska, Gliwice 1985.
- [2] C. Pfleiderer, *Die Kreiselpumpen für Flüssigkeiten und Gase*. Springer-Verlag, 1961.
- [3] J. Grychowski, A. Korczak, E. Kosek, *Wpływ cech konstrukcyjnych i parametrów pracy stopnia pompy wirowej odśrodkowej na wartość naporu osiowego* (Influence of structure and operation parameters of a centrifugal pump stage on the axial thrust). Zeszyty Naukowe Pol. Śląskiej, Energetyka 56, Gliwice 1976.
- [4] A. J. Stepanoff, *Radial- und Axialpumpen*. Springer-Verlag, Berlin 1959.
- [5] A. Korczak, J. Lamboj, *Badania wpływu uźebrowania ścianek wewnętrznych kadłuba pompy na napór osiowy oraz na charakterystykę muszlową pompy* (Investigations of the influence of internal pump casing wall ribbing on axial thrust and on shell pump characteristics (unpublished report)). Gliwice 1984.
- [6] A. Korczak, J. Lamboj, B. Hoszek, W. Kańtoch, A. Wróblewski, *Kadłub pompy wirowej odśrodkowej* Zgłoszenie patentowe, (Centrifugal pump casing — patent application). Warszawa 1983.
- [7] A. Korczak, J. Lamboj, *Wpływ uźebrowania wewnętrznej ściany kadłuba pompy wirowej odśrodkowej na napór osiowy* (Influence of ribbing of centrifugal pump internal casing wall on axial thrust). V Konferencja naukowo-techniczna „Technologia przepływowych maszyn wirnikowych”, Rzeszów 1983.
- [8] J. Grychowski, *Średnia prędkość wirwania cieczy w osłonie cylindrycznej ograniczonej wirującą tarczą z dodatkowym przepływem cieczy* (Average velocity of liquid rotation in a cylindrical shell limited by a rotating disc with additional liquid flux). Prace IMP, no 67—68, 1975.

Uźebrowanie wewnętrznej przedniej ścianki kadłuba pompy wirowej odśrodkowej jako sposób zmniejszenia naporu osiowego

Streszczenie

Na podstawie teorii podobieństwa pomp wirowych przedstawiono analizę naporu osiowego, w której uwzględniono wpływ średniej prędkości kątowej wirwania cieczy przed i za wirnikiem. Przy podanych założeniach upraszczających określono wyróżnik szybkobieżności n_{sz} pompy, w której odciążenie naporu osiowego za pomocą uźebrowania przedniej ścianki kadłuba będzie w nominalnym punkcie pracy całkowite. Następnie powyższą tezę zilustrowano wynikami badań. Opiszano pompę modelową (rys. 2) oraz stanowisko badawcze (rys. 3). Opracowane wyniki pomiarów przedstawiono w formie współzależności liczb bezwymiarowych określonych wzorami (10a, 13, 13a, 14 i 15) bądź też zależności tych liczb od cech konstrukcyjnych uźebrowania przedniej ścianki kadłuba. Wpływ liczby, wysokości i odległości żeber od wirnika na bezwymiarowe parametry pracy — w tym liczoę naporu osiowego — przedstawiono odpowiednio na rysunkach 4, 5 i 6. Na rysunku 7 uwidoczono zależność parametrów pracy

pompy od wielkości i kierunku wymuszonego przepływu za wirnikiem, natomiast na rysunkach 8 i 9 porównano charakterystyki uniwersalne oraz liczby naporu osiowego *przebadanych* pomp o wyróżnikach szybkobieżności $n_{sf} = 58,8$ i $n_{sf} = 50,5$ z żebrami i bez żeber.

Z przytoczonej analizy i opracowanych wyników badań wypływa wniosek, że uźebrowanie przedniej ścianki kadłuba może być jednym ze sposobów odciążenia naporu osiowego.

Оребрение внутренней передней стенки корпуса центробежного насоса для уменьшения осевого напора

Резюме

На основе теории подобия центробежных насосов представлен анализ осевого напора, в котором учтено влияние средней угловой скорости вращения жидкости перед и за ротором. Для данных упрощающих предположений определен дискриминант быстроходности η_{sf} насоса, в котором произойдет полная разгрузка осевого напора при помощи уребрения передней стенки корпуса в номинальном режиме работы. Затем этот тезис иллюстрируется результатами исследований.

Описаны модельный насос (рис. 2) и экспериментальный стенд (рис. 3). Обработанные результаты измерений представлены в форме зависимостей безразмерных чисел определенных формулами (10а, 13, 13а, 14 и 15). Полученные зависимости представлены графически (рис. 4—9). Рис. 4, 5 и 6 представляют влияние конструктивных особенностей уребрения передней стенки корпуса. Рис. 4 представляет влияние числа ребр на остальные величины. Рис. 5 представляет влияние высоты ребр. Рис. 6 представляет влияние расстояния ребр от ротора. Рис. 7 представляет зависимость рабочих параметров насоса от величины и направления вынужденного течения q за ротором. Рис. 8 представляет характеристику и осевые напоры исследуемого насоса характеризующегося быстроходностью $\eta_{sf} = 58,8$ с ребрами и без ребр на стенке корпуса. Рис. 9 представляет характеристику и осевые напоры другого исследуемого насоса характеризующегося быстроходностью $\eta_{sf} = 50,5$ с ребрами и без ребр.

Из приведенного анализа и обработанных результатов исследований вытекает вывод, что оребрение передней стенки корпуса может служить одним из способов разгрузки осевого напора.