P O L S K A   A K A D E M I A   N A U K
INSTYTUT MASZYN PRZEPŁYWOWYCH

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

TRANSACTIONS
OF THE INSTITUTE OF FLUID-FLOW MACHINERY

90-91

WARSZAWA—POZNAŃ 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 PRZEPŁYWOWYCH

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

REDAKCJA — EDITORIAL OFFICE

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

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.

PRACE INSTYTUTU MASZYN PRZEPŁYWOWYCH

1989 Zeszyt 90-91

ANDRZEJ ŁOJEK

Instytut Maszyn Przepływowych PAN w Gdańsku (Institute of Fluid-Flow Machinery, Polish Academy of Sciences, Gdańsk)

Influence of Prerotation on Impeller Pump Operation

Basing on a model of one-dimensional flow, the influence of prerotation on conditions of liquid flow in high-speed pump impellers is determined. The effects of prerotation influence on pump operation predicted in this way have been compared with the results of experimental tests concerning free and forced prerotation. Influence of prerotation, generated in different ways, on energy-related, cavitation and vibroacoustic pump properties is characterized. The so-called prerotational by-pass jet device is presented, which improves operating conditions of pumps controlled through forced prerotation.

Nomenclature

[illegible]	— vibration acceleration	U	— peripheral velocity
[illegible]	— flow velocity	β, δ	— angles
[illegible]	— diameter	Δh_{dyn}	— dynamic depression of pressure
[illegible]	— area	Δh_{up}	— hydraulic loss in the by-pass
[illegible]	— acceleration of gravity	ΔH_s	— pressure difference in the flow
[illegible]	— head of pump	Δp	— pressure pulsations
[illegible]	— prerotation intensity	ζ_c, ζ_w	— hydraulic resistance coefficients
NPSH	— net positive suction head	η	— efficiency
[illegible]	— power shortage coefficient	φ_s	— suction delivery number
[illegible]	— discharge	ω	— angular velocity of impeller
[illegible]	— radius		

Subscripts:

[illegible]	— before inlet		
[illegible]	— at inlet	n	— rated
[illegible]	— at impeller outlet	kr	— critical
[illegible]	— concerns streamlines a_1, a_2	r	— required
[illegible]	— concerns streamlines b_1, b_2	av	— available
[illegible]	— meridional	k	— vane
[illegible]	— circumferential	s	— suction pipe
[illegible]	— hydraulic	uk	— system
[illegible]	— theoretical	up	— by-pass

1. Conditions of Flow Upstream of the Impeller

It is assumed generally [4,8] that pump impeller supply conditions are optimum when angle β_0 of liquid streamline inflow is the same as inlet angle β_1 of the blade, i.e. $\beta_0 = \beta_1$. Such inflow conditions can be ensured for different pump operation states

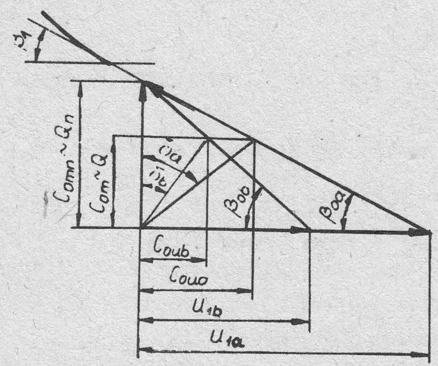

Fig. 1. Velocity triangles at impeller inlet

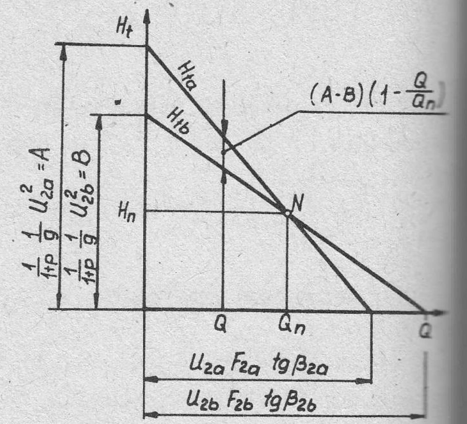

Fig. 2. Theoretical characteristics ot a high-spee pump impeller

(Fig. 1), providing that together with a change of discharge Q (meridional componen C_{0m} of flow velocity) a corresponding change of prerotation — liquid swirl befor the impeller inlet — will occur. The ratio of the circumferential component C_{0u} of flow velocity and the impeller peripheral velocity U_1 may be used as a measure of prerotation:

$$k = C_{0u}/U_1. \quad (1)$$

Constant direction of liquid inflow at the impeller ($\beta_0 = \beta_1 = \text{const}$), maintaining uniform C_{0m} distribution, is ensured by prerotation of intensity:

$$k = f(Q) = 1 - Q/Q_n \quad (2)$$

$$k = f(r) = C_{0ua}/U_{1a} = C_{0ub}/U_{1b} = \text{const}. \quad (3)$$

High-speed pump impellers with blades ot three-dimensional curvature and outle edges inclined to the axis of rotation are in most cases designed in such a way tha at the assumed axial inflow of liquid the delivery head attains a constant value alon the blade edges:

$$H_t = H/\eta_h = \text{const}. \quad (4)$$

This state concerns the design operation point ot the pump (point N in Fig. 2 With pump discharges differing from the rated value ($Q \neq Q_n$), the pump heads ar different for each streamline. Taking into account the relationship resulting fro velocity triangles, and assuming k as a measure of prerotation, the theoretical pum head can be expressed by:

$$H_t = \frac{1}{g}\frac{1}{1+p}U_2^2\left[1 - k\left(\frac{d_1}{d_2}\right)^2 - \frac{1}{U_2F_2\,\mathrm{tg}\beta_2}Q\right]. \quad (5)$$

In case of unrotated flow on the suction side ($k = 0$), the difference of heads a the pump discharge Q amounts to:

$$H_{ta}-H_{tb}=(A-B)\left(1-\frac{Q}{Q_n}\right) \tag{6}$$

Symbols A and B are explained in Fig. 2).

When flow on the suction side is prerotated ($k \neq 0$), then one can write on the basis of formula (5):

$$H_{ta}-H_{tb}=(A-B)\left(1-\frac{Q}{Q_n}\right)-\left[Ak_a\left(\frac{d_{1a}}{d_{2a}}\right)^2-Bk_b\left(\frac{d_{1b}}{d_{2b}}\right)^2\right]. \tag{7}$$

It results from (7) that the state in which condition (4) is fulfilled may be also attained when $Q \neq Q_n$, provided that the liquid has an appropriate prerotation. Therefore, if e.g. the prerotation intensity on streamline a_1a_2 is $k_a = 1-Q/Q_n$ (in accordance with formula (2)), then for streamline b_1b_2 it should be

$$k_b=\left(1-\frac{Q}{Q_n}\right)\left\{\frac{A}{B}\left[\left(\frac{d_{1a}}{d_{2a}}\right)^2-1\right]+1\right\}\left(\frac{d_{2b}}{d_{1b}}\right)^2. \tag{8}$$

Approximately, it may be assumed that in impellers with axial flow (propeller pumps) the streamlines lie on parallel surfaces. Therefore:

$$d_{1a}/d_{2a}=d_{1b}/d_{2b}\approx 1. \tag{9}$$

After substituting (9) into (8):

$$k_b=1-Q/Q_n \tag{10}$$

which means that if the propeller pump impeller generates prerotation of intensity described by formula (2) and uniform on all streamlines (formula (3)), then also in operation states differing from the rated state the impeller will attain an equal (uniform) head along the blades and the proper direction of inflow will be maintained (impact-free inflow: $\beta_0=\beta_1=\text{const}$).

For an impeller with mixed flow of liquid (diagonal pumps):

$$d_{1a}/d_{2a}\neq d_{1b}/d_{2b}<1. \tag{11}$$

In this case, as results from equation (8), prerotation necessary to fulfil condition (4) is different from the one required to ensure proper conditions of supply to the impeller (equation (2)). This means that in order to ensure proper conditions of liquid flow through a diagonal pump impeller, prerotation of unequal intensity on each streamline may be more appropriate.

The net positive suction head NPSH, required for cavitationless pump operation depends on structural parameters of the impeller and on flow parameters which influence the so-called dynamic depression of pressure, defined by the formula (5):

$$\Delta h_{dyn}=\frac{U_1^{\,2}}{2g}\left[(\zeta_c+\zeta_w)(\varphi_s^{\,2}+k^2)+\zeta_w(1-2k)\right]. \tag{12}$$

When $\varphi_s = \text{const}$, Δh_{dyn} is minimum at prerotation of intensity:

$$k_{opt}=\frac{\zeta_w}{\zeta_c+\zeta_w}. \tag{13}$$

Taking into account that the value of φ_s changes with varying prerotation inten-sity in such a way that

$$\varphi_s = C_{1m}/U_1 \approx C_{0m}/U_1 = (1-k)\,\mathrm{tg}\,\beta_1,$$

formula (12) may be written in the form:

$$\Delta h_{dyn} = \frac{U_1^{\,2}}{2g}\left\{\frac{\zeta_c+\zeta_w}{\cos^2\beta_1}k^2 + [(\zeta_c+\zeta_w)\,\mathrm{tg}^2\beta_1+\zeta_w](1-2k)\right\}. \qquad (14)$$

In this case the optimum — regarding cavitation — value of the prerotation intensity (at which Δh_{dyn} is minimum) is defined by:

$$k_{opt} = \sin^2\beta_1 + \frac{\zeta_w}{\zeta_c+\zeta_w}\cos^2\beta_1. \qquad (15)$$

It should be stressed that in both cases k_{opt} is defined assuming that $\zeta_w = \mathrm{const}$ and $\zeta_c = \mathrm{const}$. This assumption is not strictly correct, because the values of coeffi-cients ζ_w and ζ_c depend, among others, on prerotation [3,4]. Therefore in reality the value of optimum prerotation may be different from the values obtained from (13) and (15).

2. Free (natural) Prerotation

The presence of the so-called region of characteristic discontinuity is a significant feature of high-speed pump flow characteristics. This region has a definite inflexion (bend) or depression. The curve $H = f(Q)$ shows a sudden change of the character of flow through the impeller, connected with a sudden generation — at some cri-tical discharge Q_{kr} — of a strong, free prerotation induced by the impeller (Fig. 3). At $Q > Q_{kr}$ the liquid flows to the impeller with a small whirl ($k = f(r) = \mathrm{const}$). With decreasing discharge the angle of incidence β_0 becomes smaller ($k = f(Q) \neq 1-Q/Q_n$). It may be supposed that the resulting impact of liquid at the impeller inlet is the reason for flow instability and for the increased pressure pulsations and pump vibra-tions (Fig. 4). At $Q = Q_{kr} = 0.6\,Q_n$ streamlines separate from the blades. At the same value of NPSH_{av} as for $Q > Q_{kr}$ cavitation appears on the passive side of the blades. Prerotation intensity suddenly increases (Fig. 3). At $Q < Q_{kr}$, with increase of prero-tation near the suction pipe wall ($k = f(r) = \mathrm{const}$), direction of the liquid inflow changes in the centre part of the impeller in such a way that the angle of incidence β_0 becomes nearer the blade inlet angle β_1. In this state of work, pressure pulsations become slightly smaller and pump vibrations decrease (Fig. 4). However, the flow is not stable and energy losses increase with decreasing discharge.

Variability of free prerotation is different from the one described by equations (2) and (3). For partial loading of the pump, at $Q_{kr} < Q < Q_n$, its intensity is too small to ensure impact-free inflow. Prerotation increases only at $Q < Q_{kr}$. It is rather an effect than a cause of flow separation from the blades.

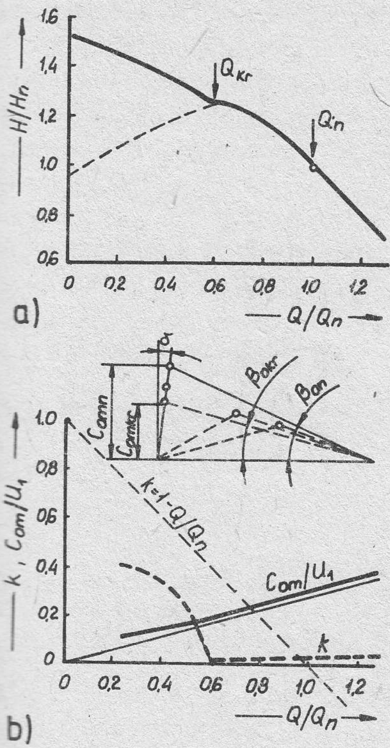

Fig. 3. Flow characteristic of the pump (a), and change of flow velocity on suction side (b)

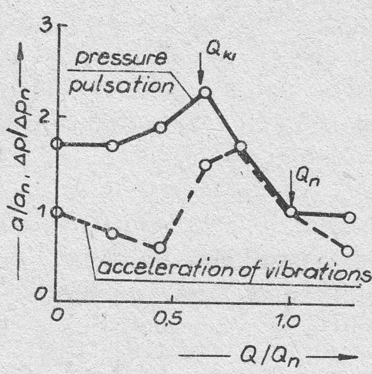

Fig. 4. Pressure pulsations in the suction pipe and vibrations of the pump impeller casing

3. Forced Prerotation

Prerotation is utilized for controlling pump operation parameters, especially of high-speed pumps. In this case, as opposed to free prerotation, the prerotation is forced by additional devices installed at the pump inlet. These are mainly the so-called prerotation guide rings, provided with adjustable blades for generating required changes of angular momentum of liquid before entering the impeller. So-called prerotational by-pass jet devices (Fig. 5) are a new solution. There angular momentum is generated before the inlet to the impeller by vanes and a by-pass jet (devices I and II) [6] or with a by-pass jet only (vaneless devices III) [7].

Prerotation guide-rings with straight guide vanes — with constant angle of inclination δ_k along the vane — generate prerotation of intensity

$$k = f(r) \neq \text{const}$$
$$k = f(Q) \neq 1 - Q/Q_n \qquad \text{(see Fig. 6a).}$$

Therefore the angle of incidence β_0 changes during prerotation control. These changes are large, especially near the impeller hub, and are larger than during throttle governing. With growing prerotation also pressure on the suction side decreases. These disadvantageous changes of flow on the suction side result in deterioration of energy and cavitation properties of the pump, and limit the permissible range of forced prerotation intensity, and therefore of prerotation governing itself.

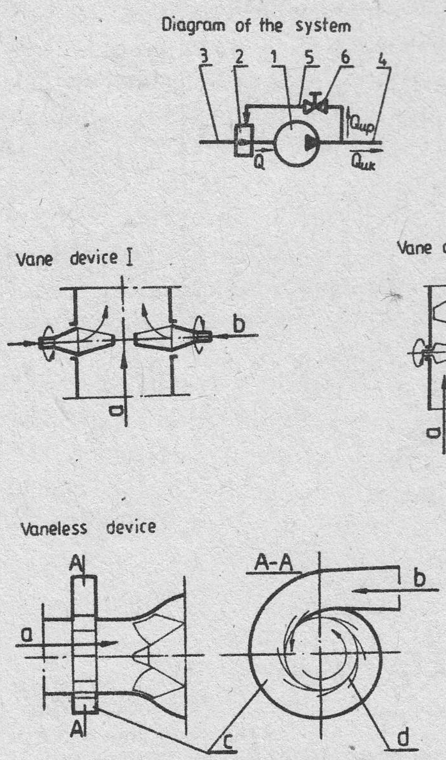

Fig. 5. Prerotational by-pass jet devices

1 — pump, *2* — prerotational by-pass jet device, *3* — suction pipe, *4* — delivery pipe, *5* — by-pass pipe, *6* — valve; *a* — main stream, *b* — by-pass jet, *c* — spiral casing, *d* — gap

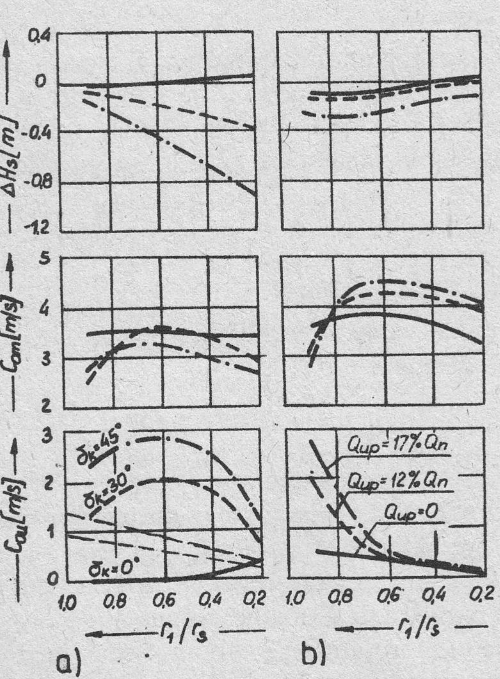

Fig. 6. Variability of flow velocity and pre- sure in the suction area caused by prerota- tion forced by means of: a) guide vane b) by-pass jet (thin dashed lines correspon to $C_{ou} = \omega r_1 k = \omega r_1\left(1 - \frac{Q}{Q_n}\right)$)

Effectiveness of the influence of prerotation on the pump properties depends not only on the method of prerotation forcing but also on the conditions of pump operation. Conditions of flow in the pump controlled with a prerotation guide ring are best when the pump operates in conjunction with a pipeline of very small hydraulic resistance, because prerotation necessary for attaining the required change of pump discharge is relatively small.

In guide ring and by-pass jet prerotation devices the prerotation is forced mainly by the guide vanes. The by-pass jet, supplied to the suction area through hollowed vanes (device I), gives only small changes of velocity and pressure distribution before the impeller. Therefore changes in the energy and cavitation curves of the pump (Fig. 7) are small.

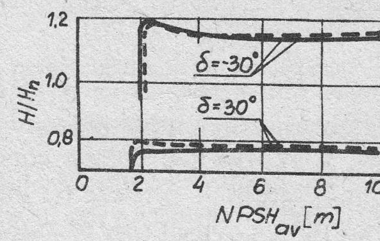

Fig. 7. Cavitation curves of a pump controlled by means of device I ($Q \approx Q_n$ = const; continuous and dashed lines correspond to $Q_{up} = 0$ and $Q_{up} = 12\%$, respectively)

The increase of the liquid angular momentum and decrease of pressure in the centre of the suction pipe are disadvantageous. This can be avoided by using short guide vanes and leading the by-pass jet mainly near the wall of the suction pipe. There is also no significant change of the pump characteristics when the by-pass jet is supplied centrally through a nozzle placed before the prerotation guide ring (device II) at a large distance from the impeller (about 4.5 d_{imp}; d_{imp} — diameter of the impeller). Large changes occur when the distance between nozzle and the impeller is small (about 1 d_{imp}). The impeller supply conditions become better, the head and efficiency of the pump improve, and therefore even the efficiency of the pump system increases* (Fig. 8) [1]. Supply to the impeller by means of an additional by-pass just results in an increase of pressure on the suction side. The obtained pressure increase is limited because of the small dynamic pressure of the by-pass jet and small intensity of the by-pass liquid flow**.

In vaneless devices (III) prerotation is forced by means of a by-pass jet supplied through a supply spiral (Fig. 5). The by-pass jet is supplied tangentially to the main

* The efficiency of a pump system is described by formula:

$$\eta_{uk} = \left(1 - \frac{Q_{up}}{Q}\frac{\Delta h_{up}}{H}\right)\eta$$

and depends on energy losses caused by the by-pass jet ($Q_{up}\Delta h_{up}$) and on the conditions of pump operation (Q, H, η) during the adjustement.

** The by-pass liquid flow rate Q_{up} is limited because of pump system efficiency. The value of dynamic pressure of the by-pass jet depends on the pump delivery head; it is small because of the relatively low delivery head of high-speed pumps.

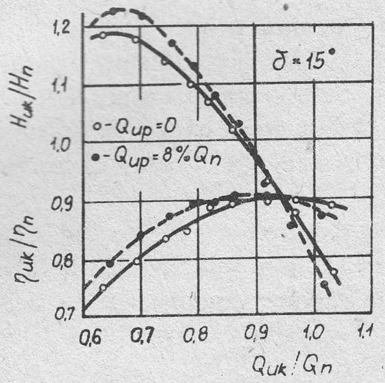

Fig. 8. Flow and efficiency curves for a pump system with a propeller pump controlled by means of device II

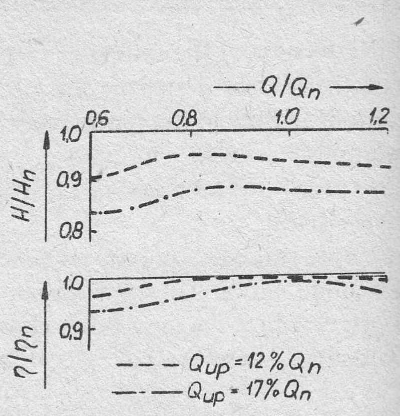

Fig. 9. Change of flow and efficiency curves of a diagonal pump under prerotation forced by a by-pass jet

stream and perpendicularly to the direction of flow, and it generates prerotation mainly near the wall of the suction pipe (Fig. 6). A small decrease of pump efficiency (Fig. 9) proves that correct conditions of flow through the pump are maintained. Small changes of pressure on the suction side do not create a danger of cavitation. If cavitation appears in the pump for other reasons e.g. because of a decrease of the inflow head, then the dynamic influence of cavitation on the pump can be reduced by leading the by-pass jet to the suction side [2]. Therefore when the required changes of the pump hydraulic parameters fall within a narrow range* (narrow range of control), this is an advantageous method of forcing prerotation.

With the increase of positive (i.e. coincident with impeller rotation) prerotation forced by the prerotation guide ring or prerotation by-pass jet devices, the level of vibroacoustic phenomena accompanying pump operation becomes lower.

4. Conclusions

Prerotation of appropriate intensity, suited to the pump loading, improves conditions of flow through the impeller and affects favourably the energy, cavitation and vibroacoustic properties of the pump. Effectiveness of the prerotation influence on pump operation depends on the method of prerotation generating.

Prerotation guide rings with straight vanes generate prerotation which, especially at high intensity, does not ensure optimum conditions of liquid flow. Therefore, the range of prerotational control is rather narrow and cannot be widened by intensifying the prerotation.

The operation of prerotation guide rings can be improved by acting on the flow on the suction side with an additional by-pass jet (vane and by-pass jet prerotation devices).

---

* In this case the intensity of forced prerotation depends exclusively on the by-pass liquid flow rate.

When the control range is narrow, the prerotation guide rings may be replaced by a vaneless by-pass jet prerotation device, which generates prerotation with a by-pass jet exclusively. The simple structure, and especially lack of moving parts, is the advantage of these devices.

References

[1] A. Łojek, *Regulacja pomp szybkobieżnych, w szczególności pomp wody chłodzącej* (Control of high-speed pumps, especially cooling water pumps). Prace IMP, no 83—84, 1983.

[2] A. Łojek, *Regulation of impeller pumps by means of prerotation induced with a by-pass jet*. Proc. Seventh Conference on Fluid Machinery, Budapest, 1983.

[3] A. Nyiri, J. Csemniczky, *Tests on a prerotation controlled cooling-water pump*. Proc. Fourth Conference on Fluid Machinery, Budapest, 1972.

[4] K. Saalfeld, *Vergleichende Darstellung der Regelung von Pumpen durch Vordrall und durch Laufschaufelverstellung*. KSB-Technische Berichte, H. 7, 1964.

[5] K. Steller, *Parametry pomp odśrodkowych ze względu na kawitację* (Parameters of centrifugal pumps with regard to cavitation). Zeszyty Naukowe IMP PAN no 18/901/77, Gdańsk 1977.

[6] K. Steller, A. Łojek, *Sposób i urządzenie do regulacji pompy wirowej* (Method and device for impeller pump control). Patent no 130690, 1984.

[7] K. Steller, A. Łojek, *Sposób regulacji pomp wirowych oraz urządzenie prerotacyjno-upustowe do regulacji zwłaszcza pomp szybkobieżnych* (Method of controlling impeller pumps and a prerotational by-pass jet device especially suitable for high-speed pump control). Zgłoszenie patentowe (Patent application), no P-243941, 1983.

[8] O. Stryček, *Možnosti zlepšovania hydraulickich vlastnosti regulacie čerpadla predrozvadzačom*. Čerpadla Potrubi Armatury, No 2—3, 1979.

Wpływ prerotacji na pracę pomp wirowych

Streszczenie

Na podstawie modelu przepływu jednowymiarowego określono wpływ prerotacji na warunki prze- ... cieczy przed wirnikiem. Stwierdzono, że prerotacja może zapewnić bezuderzeniowy napływ cieczy ... łopatki ($\beta_0 = \beta_1$) i jednakową wzdłuż krawędzi wylotowej łopatek wysokość podnoszenia (wzór 4) ... zmniejszyć dynamiczną depresję ciśnienia (wzory 12 i 14). Jej natężenie (wzór 1) powinno być do- ... do stanu obciążenia pompy (wzór 2). W pompach śmigłowych (wzór 9) powinna to być pre- ... o natężeniu jednakowym na poszczególnych liniach prądowych (wzór 3), natomiast w pompach ... nych (wzór 11) bardziej odpowiednia może być prerotacja o natężeniu niejednakowym (wzór 8). ... prerotacji optymalnej ze względu na kawitację określają wzory (13) i (15).

... opisano przebieg prerotacji swobodnej, indukowanej przez wirnik. Stwierdzono, iż jej natężenie ... wzrasta dopiero przy wydajności mniejszej od krytycznej Q_{kr} (rys. 3), kiedy dochodzi do oderwania ... cieczy od łopatek i pojawia się kawitacja. Mimo że przy $Q < Q_{kr}$ zmniejszają się pulsacje ciśnie- ... pompy (rys. 4), przepływ z silną prerotacją jest niestabilny i odbywa się z dużymi stratami ... cznymi.

... scharakteryzowano wpływ różnie wymuszanej prerotacji na warunki przepływu cieczy na ... na własności energetyczne, kawitacyjne i wibroakustyczne pomp. Wykazano, że kierownice ... prostymi wytwarzają prerotację, która nie zapewnia optymalnych warunków przepływu. ... względu zakres dopuszczalnych zmian natężenia prerotacji (zakres regulacji) jest ograniczony. ... tzw. urządzenia prerotacyjno-upustowe (rys. 5), w których prerotację wymusza się za pomocą łopatek i strumienia upustowego (urządzenia I i II) lub tylko za pomocą strumienia upustowego (urządzenie III). Stwierdzono, że działanie kierownic prerotacyjnych można polepszyć poprzez oddziaływanie na przepływ cieczy na ssaniu dodatkowo strumieniem upustowym (przykład na rys. 8). Przy niedużym zakresie regulacji korzystne jest wymuszanie prerotacji za pomocą strumienia upustowego (urządzenie III). Tak wymuszana prerotacja (rys. 6b) powoduje niewielkie obniżenie się sprawności pompy (rys. 9). Niewielkie zmiany ciśnienia na ssaniu nie stwarzają niebezpieczeństwa wystąpienia kawitacji. Zaletą urządzeń bezłopatkowych III jest prosta konstrukcja, a w szczególności brak części ruchomych.

Влияние преротации на работу лопастных насосов

Резюме

В разделе 1, на основе модели одномерного течения, определено влияние преротации на условия течения жидкости перед ротором. Обнаружено, что преротация может обеспечать безударное натекание жидкости на лопатки ($\beta_0 = \beta_1$) и одинаковый напор вдоль выходной кромки лопаток (формула 4), а также уменьшить динамическую депрессию давления (формулы 12 и 14). Её интенсивность должна быть приспособлена к состоянию нагрузки насоса (формула 2). В пропеллерных насосах (формула 9) должна это быть преротация характеризующаяся одиноковой интенсивностью на отдельных линиях тока (формула 3), а в диагональных насосах (формула 11) более соответствующей может быть преротация неодинаковой интенсивности (формула 8). Интенсивность преротации, оптимальную с учёта на кавитацию, определяют формулы (13) и (15).

В разделе 2 описан ход свободной преротации, индуцированной ротором. Обнаружено, что её интенсивность сильно возрастает лишь при расходе меньшем от критического Q_{kr} (рис. 3), когда приходит к отрывам струй жидкости от лопаток и появлению кавитации. Несмотря на то, что при $Q < Q_{kr}$ уменьшаются пульсации давления и вибрации насоса (рис. 4), течение с сильной преротацией является неустойчивым и происходит с большими энергетическими потерями.

В разделе 3 характеризуется влияние по разному вынуждаемой преротации на условия течения жидкости на всасывании и на энергетические, кавитационные и виброакустические свойства насосов. Доказывается, что направляющие аппараты с простыми лопатками производят преротацию, которая не обеспечивает оптимальные условия течения. По этому поводу предел допускаемых изменений интенсивности преротации (предел регулирования) ограничен. Представлены т.наз. преротационно-отборные устройства (рис. 5), в которых преротация вынуждется при помощи лопаток и отборного потока (устройства I и II) или только при помощи отборного потока (устройство III). Обнаружено, что действие преротационных направляющих аппаратов можно улучшить путём воздействия на течение жидкости на всасывании добавочно отборным потоком (пример на рис. 8). При небольшом пределе регулирования полезно вынуждать преротацию при помощи отборного потока (устройство III). Так вынуждённая преротация (рис. 6b) вызывает небольшое понижение к.п.д. насоса (рис. 9). Небольшие изменения давления на всасывании не способствуют опасности выступания кавитации. Достоинством безлопаточных устройств является простая конструкция, а в особенности отсутствие подвижных частей.