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RESEARCH ARTICLE

Hydraulic parameters of preotation controlled pumps

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Abstract

At the prerotation control there is an adjustable blade row before the pump impeller for producing a peripheral component of the inlet flow velocity. The effect of prerotation is changing along the $H - Q$ characteristic and the range of control depends on the geometric parameters of the pump and prerotation unit. This study is dealing with the role of geometric and hydraulic parameters on the performance characteristics. The results are based on experiments and on CFD calculations.

Keywords

controls of pump · prerotation control · inlet guide vanes for pump

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1 Introduction

At the prerotation control there is an adjustable blade row before the pump impeller for producing a peripheral component of the inlet flow velocity. The change comes forward in the head H and can be calculated by the Euler equation:

$$H_{th} = \frac{c_{2u}u_2 - c_{1u}u_1}{g}$$

namely the peripheral component of inlet velocity c_{1u} is changed during the control. When the c_{1u} direction equals to the sense of rotation, the head is decreasing, in opposite sign the head is increasing.

The basic description of this kind of control is given in the literature, e.g. [1–9]. The rate of change of head related to the prerotation-free head is increasing with the specific speed, so the main field of application can be found at the mixed and axial flow pumps. There are two basic types of construction; one for dry sump and the other for wet (open) sump application as shown in Fig. 1 and Fig. 2.

2 The inlet moment of momentum and the change of shaft power

2.1 Dry sump variant

The axes of control blades are perpendicular to the pump shaft of rotation. The blade angle α_L is constant along the blade. The velocity components:

$$c_m = \frac{4Q}{D_p^2\pi} \qquad c_u = c_m f(\alpha_L) \tag{1}$$

where Q is the volume rate of flow, D_p is the diameter of prerotation housing, c_m is the meridional, c_u is the peripheral component of flow velocity. The function $f(\alpha_L)$ depends on the prerotation unit blading. The moment of momentum M_p considering constant c_m and $f(\alpha_L)$ along the radius r is as follows:

$$M_P = \rho 2\pi \int_0^{D_p/2} c_u r c_m r dr = \rho \frac{4Q^2}{3D_p\pi} f(\alpha_L) \tag{2}$$

Due to the finite number of impeller blades the change of c_{1u} induces a small change on the outlet peripheral component c_{2u},

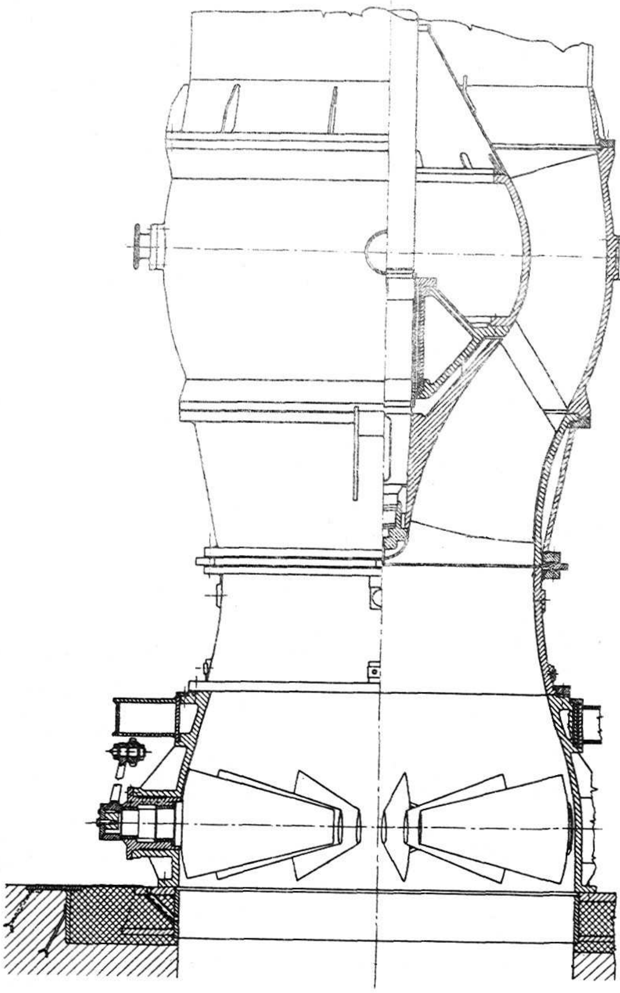

Fig. 1. Mixed flow pump with axial pretoration

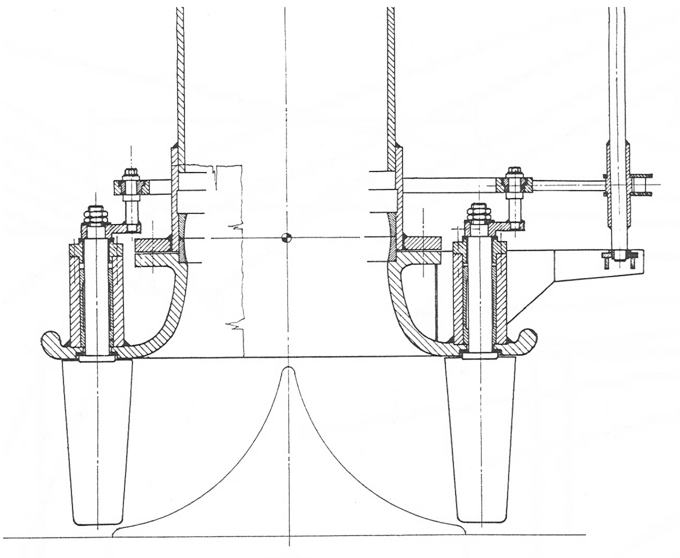

Fig. 2. Radial-flow prerotation unit,wet sump application

which lowers the effect of c_{1u}. The change of shaft power taking this into account:

$$\Delta P = \omega M_p \left[1 - k(\alpha_L, n_q)\right] = \rho \frac{4Q^2\omega}{3D_p\pi} f(\alpha_L) \left[1 - k(\alpha_L, n_q)\right] = \rho \frac{4Q^2\omega}{3D_p\pi} F(\alpha_L, n_q) \quad (3)$$

where ω is the angular velocity of pump shaft and $F(\alpha_L, n_q)$ contains the effect of finite blade numbers, the wall friction in the blade rows and the specific speed n_q.

$$F(\alpha_L, n_q) \cong \tan(K)\alpha_L \quad K < 1 \quad (4)$$

In "ideal" case $K = 1$:

2.2 Wet sump variant

The meridional velocity:

$$c_m = \frac{Q}{B_p D_p \pi} \quad (5)$$

where B_p is the height of prerotation blades and D_p is the diameter of prerotation blades pitch circle. Here the radius is constant: $r = D_p/2$

The deduction is the same as in 2.1. The change of power is as follows:

$$\Delta P = \rho \frac{Q^2\omega}{2B_p\pi} F(\alpha_L, n_q) \quad (6)$$

$F(\alpha_L, n_q) \cong \tan(K)\alpha_L)$.

The empirical factor K depends also on the prerotation unit geometry. There is a difference between the dry sump and wet sump variant, the K value is higher at the latter due to the more favourable velocity distribution behind the prerotation blades.

3 Theoretical head, hydraulic losses

In case of axial pumps and mixed flow pumps where the impeller is made without front shroud, the hydraulic losses are dominant. The volumetric loss and the disc fiction loss can be treated approximately also as hydraulic loss. In case of model measurements the mechanical loss can be eliminated. Then the theoretical head H_{th} and the hydraulic losses h' based on the approximation above are the following:

$$H_{th} = \frac{H}{\eta} \qquad h' = H_{th} - H \quad (7)$$

where η is the measured model efficiency. The hydraulic loss can be treated as the sum of the impeller-diffuser loss and the prerotation unit loss. The impeller-diffuser loss depends on the position of working point and is proportional to the square of Q:

$$h_0' = Q^2 k_0 \left(\frac{Q}{D_j^3\omega}\right) = Q^2 k_0(\Phi) \quad (8)$$

where D_j is the eye diameter of the impeller. The k_0 function means, that this loss component is the total loss including all flow friction and separation when there is no prerotation unit. The k_0 coefficient is the function of flow number Φ.

The loss of prerotation unit is the function of blade angle α_L being proportional to the square of Q and can be expressed with a loss coefficient:

$$h_\alpha' = Q^2 k_\alpha(\alpha_L) = \zeta(\alpha_L) \frac{c_m^2}{2g} \quad (9)$$

The hydraulic loss is the sum of the components:

$$h' = h_0' + h_\alpha' \tag{10}$$

The equations (Eq. 8) –(Eq. 10) are approximations since h_0' depends also on the prerotation by way of the angle of inlet relative velocity (β_1), however this effect seems to be small in the usual working range of pumps where the efficiency is high.

The change of theoretical head as a function of prerotation can be expressed with the help of power change:

$$H_{th} = H_{th,0} + \frac{\Delta P}{Qg\rho} \tag{11}$$

where $H_{th,0}$ is the theoretical head at zero prerotation. Substituting expression (Eq. 3) to the equations above:

$$H_{th} = H_{th,0} + \frac{4Q\omega}{3gD_p\pi} F(\alpha_L, n_q) \tag{12}$$

The differences in theoretical head are the linear function of flow rate. The result is similar when substituting equation (Eq. 6).

With a given impeller eye diameter, speed and discharge, the change of head is inversely proportional to the prerotation unit diameter D_p. At the same time the hydraulic loss of prerotation unit is inversely proportional to the fourth power of D_p while the hydraulic loss of impeller remains approximately constant. This guides to a kind of optimum.

4 Measurement results

The measurements presented here were made by an impeller of eye diameter D_j = 268 mm and speed 1480 rpm. The dry type prerotation control system has been modelled, the prerotation unit diameter is D_p = 400 mm and the blade cascade solidity: $l/t = 1$ (chord space ratio).

The measured performance characteristics are shown in Fig. 3. and Fig. 4. The relative curves have a common reference point with specific speed n_q = 113 and prerotation angle $\alpha_L = 0°C$.

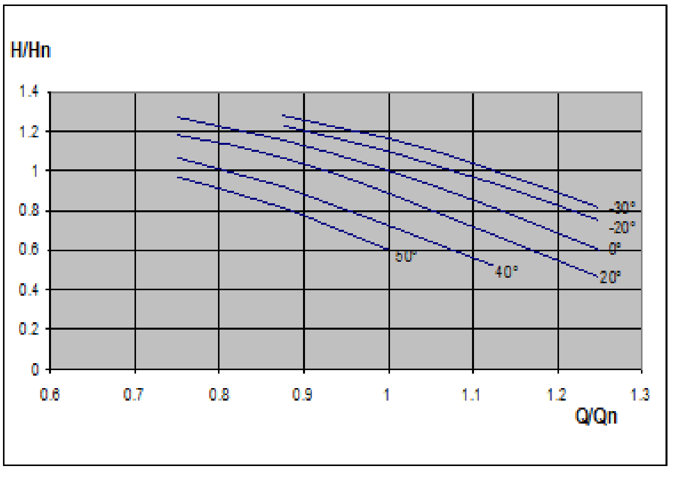

Fig. 3. H-Q curves with the prerotation angle as parameter. H_n, Q_n denote the nominal point

The theoretical head curves are given in Fig. 5.

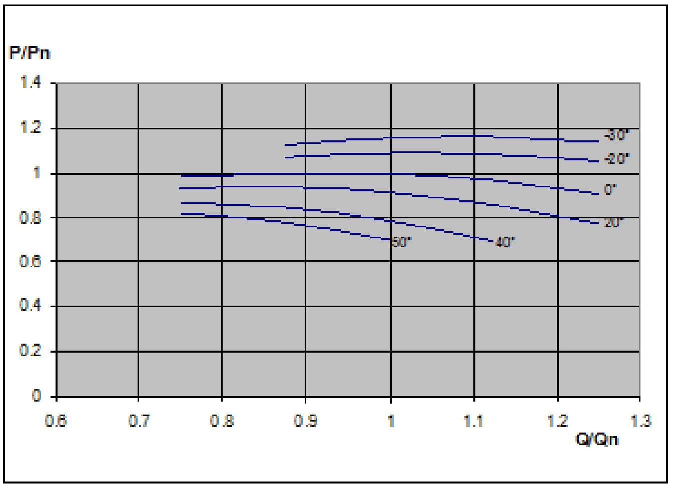

Fig. 4. P-Q curves with the prerotation angle as parameter. P_n, Q_n denote the nominal point

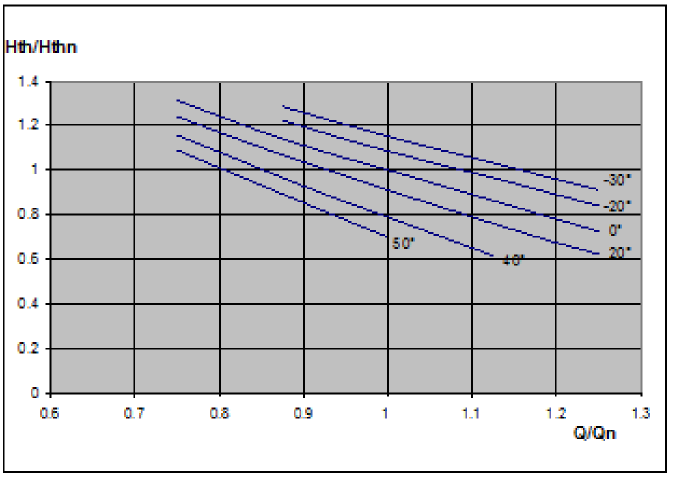

Fig. 5. Hth-Q curves with the prerotation angle as parameter

The theoretical head is a linear function of the flow rate according to eq. (Eq. 12) and the form of curves in Fig. 5. is really close to straight lines. It means that the theoretical head curves of prerotation can be drawn approximately on the basis of the theoretical head without prerotation, with the help of equation (Eq. 12). The average value of factor K in equation 4 from these curves is: K = 0.78. The next step is the determination of hydraulic losses. These losses can be separated by the following procedure:

- a.: The total hydraulic loss comes from equation (Eq. 7).
- b.: The h_0' loss can be measured when the prerotation blade angle is zero, with the small correction of prerotation unit loss at zero angle. (According to our practice, the loss coefficient is about $\varsigma = 0.2$ for this blade row position.)
- c.: The loss of prerotation unit is then the difference between the total loss and the loss without prerotation (with the correction of prerotation unit loss at zero angle).
- d.: The $k_0(\Phi)$ and the $\varsigma(\alpha_L)$ empirical functions can be calculated.

Over the measurements shown in Fig. 3 and Fig. 4, two variants with a minor modification of impeller have been also measured and evaluated. All the results are given in the form of the empirical parameters in Fig. 6 and Fig. 7. The loss coefficient is

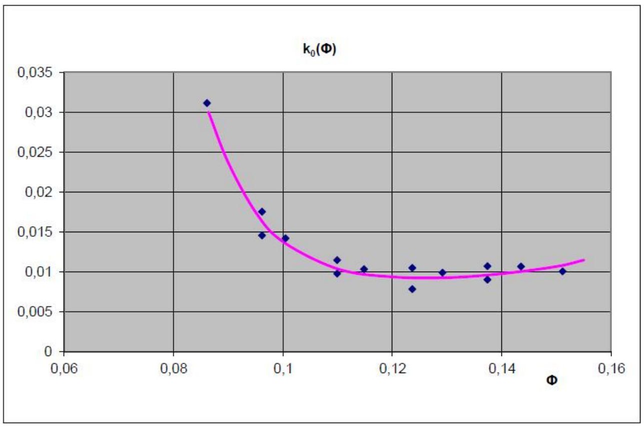

Fig. 6. Impeller-diffuser loss function without prerotation

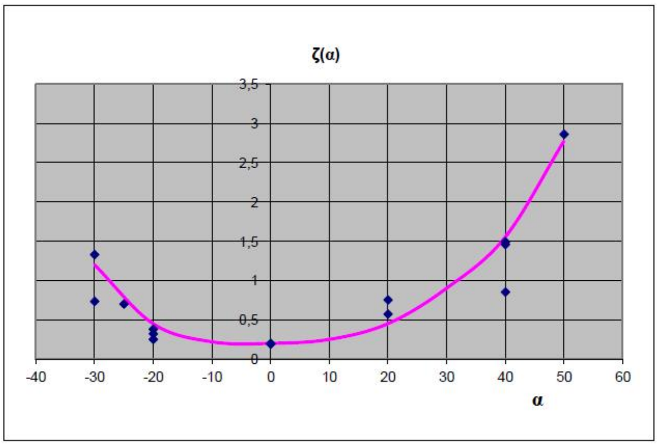

Fig. 7. Loss coefficient of prerotation unit

defined as the head loss related to the velocity head upstream of the the prerotation unit. The scattering of prerotation unit loss coefficient in Fig. 7. is high, however the loss itself is small compared to the total hydraulic loss.

5 CFD computations

The velocity distributions at the impeller inlet have been computed at different prerotation blade angles by ANSYS CFX. The moment of momentum has been determined based on the integral of eq. (Eq. 2) with changing velocity components. The hydraulic loss has been evaluated too as the difference of integral Eq. 13 calculated before the prerotation unit and at the impeller eye:

$$h = \frac{2\pi}{gQ} \int \left(\frac{p}{\rho} + \frac{c^2}{2} + gz \right) c_m r dr \tag{13}$$

For the moment of momentum the computed results are shown in Fig. 8. in form of the calculated $\arctan[f(\alpha_L)]$ in function of blade angle α_L (see eq. (Eq. 2)):

One can see in Fig. 8 that according to the computation, the deflection defined by (Eq. 2) is practically equal to the blade an-

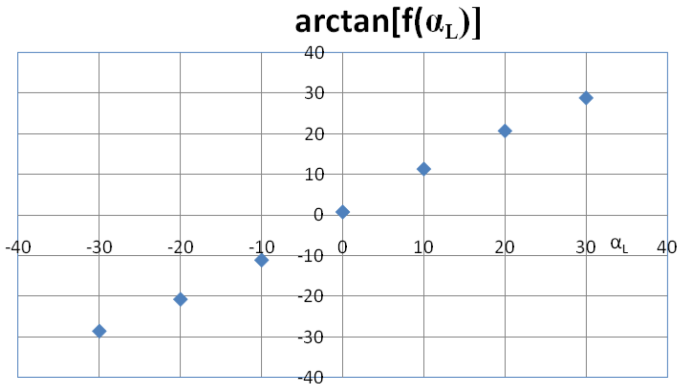

Fig. 8. Deflection of prerotation unit in function of blade angle.

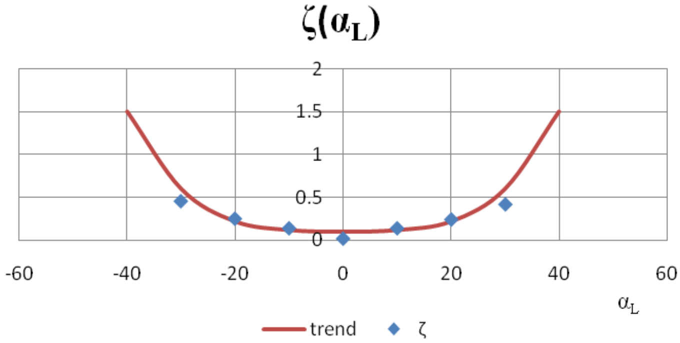

Fig. 9. Loss coefficient with trend line.

gle in the range of $-30°C \leq \alpha_L \geq +30°C$ - at the given diameter ratio and blade solidity. At the same time, the correction factor K in eq. 4 which includes the effect of finite impeller blade number is approximately 0.78 evaluated from the theoretical head curves.

Fig. 9. shows the computed loss coefficient of the prerotation unit. Compared to the results in Fig. 7, the computed loss coefficient is even smaller than that from loss separation and the results have also uncertainty e.g. at zero blade angle. The common conclusion is that the loss seems to be negligible in the range of $-20°C \leq \alpha_L \geq +20°C$. At the trend line the tendency on Fig. 7 was also considered.

Fig. 10. Mixed flow pumps with prerotation unit in the factory.

6 Conclusions

- The analysis of performance curves with the hydrodynamic model of the prerotation control leads to a deeper understanding of the role of construction parameters when realising an efficient control.
- There are some measured results available for the designers which also helps selecting the good proportions of the active elements. The cascade solidity of prerotation blades and a well-suited impeller - prerotation unit seem to be key points.
- The change of power due to prerotation can be calculated by simple expressions with the empirical constant K (see Eq. 4).
- The approximate calculation of hydraulic loss parts explained here might be very useful also in need of preliminary performance curves.
- The flow moment of momentum produced by the prerotation unit can be calculated relatively easily with good accuracy by the help of CFD computation.

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