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VOLUMETRIC LOSSES IN MULTI-SLOT SEALING OF BALANCE DISC IN HIGH-PRESSURE PUMPS

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Abstract

The paper discusses the issues of volumetric losses occurring in the axial thrust balancing system in high-pressure centrifugal pumps. The paper presents a new concept of construction that uses an additional multi-slot sealing after the pump last stage impeller. Compared the results of the calculations of volumetric losses in systems with and without multi-slot sealing.

Key words: pump, axial thrust, balance disc, multi-slot sealing, high-pressure pump

1. INTRODUCTION

The balance disc is used to balance the axial thrust in the high-pressure centrifugal pumps. The volumetric losses occurs in balancing system with balancing disk.

Occurrence of volumetric losses in balancing system with balance disc caused decreasing the efficiency of pump. Thus decreasing the volumetric losses is beneficial.

Liquid flows to a balance disc through radial gap between the sleeve mounted on pump shaft and sleeve mounted in delivery pump casing (Fig. 1).

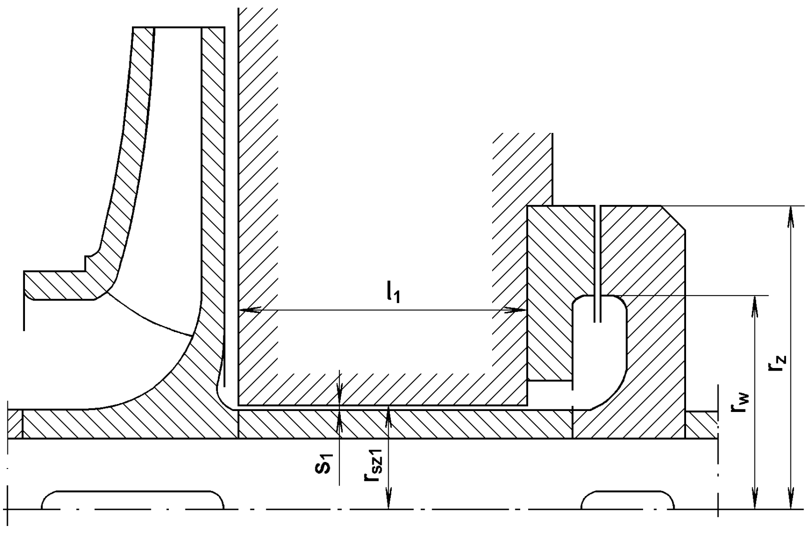


Fig. 1. Axial thrust balancing system with balance disc

From under balance disc liquid flows through axial gap between resistance disc and sliding ring of balancing disk.

The quantity of the volumetric loss depends on the pressure difference between pressure at the inlet to the radial gap and the pressure under balance disc, and the dimensions of the radial gap, i.e. its diameter, length and width (Wilk S., 1984; Korczak A., 2005).

2. CALCULATIONS OF VOLUMETRIC LOSSES IN BALANCE DISC SYSTEM WITH ONE-SLOT SEALING

2.1. Pump working without affluence

Assuming that during the flow through the axial gap pressure drop is linear and despite a small loss of pressure at the inlet to the gap, height of the pressure under balance disc is given by the formula

$$H_t = \frac{F_c}{\pi \cdot g \cdot \rho \cdot \left[\frac{1}{3}\left(r_w^2 + r_w r_z + r_z^2\right) - r_{sz1}\right]} \tag{1}$$

The total axial thrust is the sum of resultant thrusts acting on pump impellers.

Frequently, due to the improvement of the suction conditions (reduction of NPSH), first stage impeller has greater then others impellers diameter of inlet and greater diameter inlet seal, so the axial thrust acting on it is larger.

The total axial thrust transmitted through the balance disc is determined by the formula

$$F_c = F_1 + (i-1) \cdot F_2 \tag{2}$$

The rest of the work adopted one-dimensional calculating model supplemented by correction factors. On the basis of laboratory measurements (Wilk S., Wilk A., 1996; Wilk A., Wilk S.,1996; Wilk A., 1998; Wilk A., 2007; Wilk A., 2008; Wilk A., 2009) and theoretical analyzes (Wilk S., Wilk A., 1995; Wilk A., Wilk S., 2003) can be stated that such a model allows for obtain a satisfactory accuracy of calculations.

The pressure head the inlet to the radial gap defined by the formula

$$H_{sz1} = (i-1) \cdot H + H_p - \Delta H_w \tag{3}$$

The head pressure drop at the inlet to the radial gap can be calculated from the formula

$$\Delta H_{wl} = (1 + \xi_{wl}) \cdot \frac{c_1^2}{2g} \tag{4}$$

The head pressure drop in the gap can be defined as

$$\Delta H_{sz} = \lambda \frac{l_1}{2 \cdot s_1} \cdot \frac{c^2}{2g} \tag{5}$$

The coefficient of losses in the gap for turbulent flow (Re > 3000) can be calculated by Y. Yamada formula

$$\lambda = 0{,}31 \cdot \mathrm{Re}^{-0{,}24} \left[1 + \left(\frac{7}{16} \cdot \frac{\mathrm{Re}'}{\mathrm{Re}}\right)^2\right]^{0{,}38} \tag{6}$$

where Reynolds numbers are defined as

$$\mathrm{Re} = \frac{2 \cdot c_1 \cdot s_1}{\nu} \tag{7}$$

$$\mathrm{Re}' = \frac{2 \cdot u_1 \cdot s_1}{\nu} \tag{8}$$

Volumetric flow rate through the gap is defined by formula

$$Q_{sz1} = 2 \cdot \pi \cdot r_{sz1} \cdot s_1 \cdot c_1 \tag{9}$$

The relative volumetric loss in the balance disc is defined as

$$q_1 = \frac{Q_{sz1}}{Q} \tag{10}$$

Give a detailed algorithm for computing the axial pressure and volumetric losses is beyond the scope of this paper. A detailed analysis of this issue can be found for example in (Wilk A., 2003; Wilk A., 2004).

It was calculated for example the relative volumetric loss in the high-pressure pump with the following nominal parameters:
- discharge Q = 0,0875 m^3/s
- delivery head from one stage H = 80 m
- rotational speed n = 1450 rpm
- number of stages i = 10

and dimensions of balancing system:
r_w = 0,155 m, r_z = 0,19 m, r_{sz1} = 0,07 m, l_1 = 0,195 m, s_1 = 0,00035 m.

Calculated relative volumetric loss is equal to q_1 = 0,063.

2.2. Pump working with affluence

When the pump works with affluence, for example during series connection of pumps, the head of pressure at the inlet to the radial gap is increased by the head of affluence head pressure. The head of pressure at the inlet to the radial gap is then

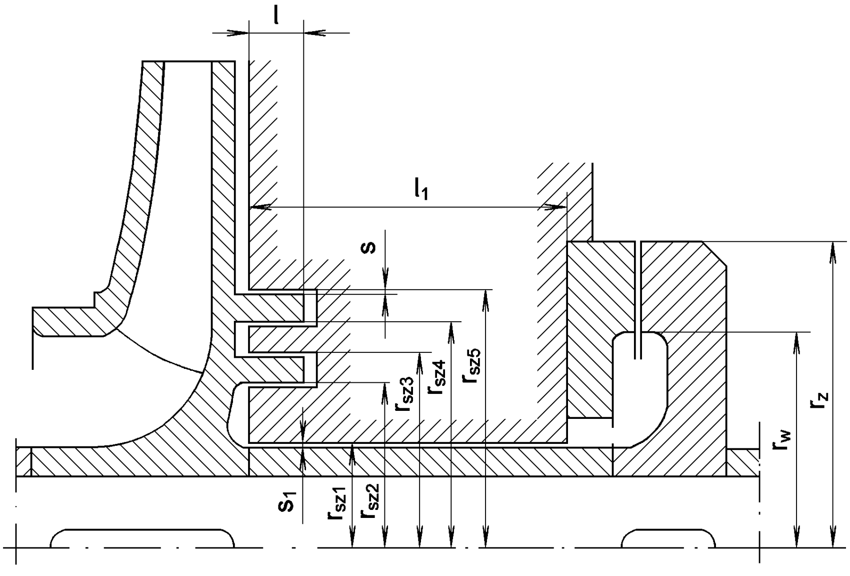
$$H_{sz1} = (i-1) \cdot H + H_p - H_w + H_n \tag{11}$$

Due to the increase of pressure at the inlet to the gap at a constant pressure under balance disc increases the relative volumetric loss in the pump working as the second in a series.

For example, during serial cooperation the 7-stage pump (first in serial connection, the head pressure of affluence H_n = 560 m) with a 10-stage pump (working as the second in a serial connection), relative volume loss increases to q_1 = 0,0921.

3. CALCULATIONS OF VOLUMETRIC LOSSES IN THE SYSTEM OF BALANCE DISC WITH MULTI-SLOT SEALING

To reduce the loss of volume in the system would increase the length of radial gap and reduce it’s width.



Rys.2. Multi-slot sealing of balance disc

Increasing the length of the gap increases the length of the shaft and the pump and thus increase the deflection of the shaft and the weight of the pump. Reducing the width of the gap would cause weakening of the sleeve.

Way to reduce the volumetric losses in the axial thrust balancing system without having to increase the dimensions lengths, can be applied multi-slot sealing after the last impeller. This design solution was developed in Zakład Mechaniki Przemysłowej ZAMEP and is protected by the Patent Office. Sample solution is shown in Fig. 2. After applying multi-slot sealing after last stage impeller increases the hydraulic resistance during flow of the liquid under balance disc and reduces the pressure acting on the rear disc of last impeller resulting in reduced axial thrust acting on the rotating assembly.

During the calculation of the relative volumetric loss in axial thrust balancing system in the pump with multi-slot sealing behind the last stage impeller is omitted the impact of possible rotation of the liquid in the spaces between the slots on the pressure distribution along the radius. It was assumed that the pressure at the inlet to the next slot is the same as the outlet from the previous slot.

For example, the relative volumetric losses was calculated for the following dimensions of the multi-slot sealing behind the impeller of last stage:

r_{sz2} = 0,08 m, r_{sz3} = 0,09 m, r_{sz4} = 0,1 m, r_{sz5} = 0,11 m, l = 0,025 m, s = 0,00035 m.

Other dimensions as before.

The calculated relative volumetric loss is q_l = 0,0478. The speed of the fluid in the gap was also significant decreases from c_l = 35,813 m/s to c_l = 27,178 m/s.

During pump operation with affluence (H_n = 560 m) the relative volumetric loss is equal q_l = 0,0691.

Of course it is possible to use a larger number of slots, but you should take into account that excessive reduction of volumetric losses reduces the axial gap width between the resistance disc and sliding ring of balancing disk causing the danger of friction (at large axial pounding of surfaces).

Sample calculations were performed assuming the width of the slots in the new pump. Over time, the gaps will be expanded, especially when pumping liquids containing solids, will increase volumetric losses, and thus will decrease volumetric efficiency and overall efficiency of the pump.

4. CONCLUSIONS

1. Application of the multi-slot sealing behind the last stage impeller in the high-pressure pump with balance disc can substantially reduce the volumetric losses in the balancing system without necessity of increasing the lengths of the pump.

2. Application of the multi-slot sealing behind the last stage impeller results in a significant reduction of liquid velocity in the gap and thus increases the of sustainability elements of the balancing system.

NOTATION SCHEDULE:

c_1 – velocity of liquid in radial gap, m/s,
g – acceleration due to gravity, m/s^2,
i – number of stages,
l – length of the gaps in multi-slot sealing, m,
l_1 – length of the radial gap, m,
n – rotational speed, rpm,
r_{sz1} – radius of radial gap, m,
r_{szi} – radiuses of gaps in multi-slot seal, m,
r_w – inner radius of balancing disc, m,
r_z – outer radius of balancing disc, m,
s – width of gap in multi-slot seal, m,
s_1 – width of radial gap, m,
u_1 – peripheral velocity, m/s,
q_1 – relative volumetric loss, –
F_1 – axial thrust acting on the impeller of 1st stage, N,
F_2 – axial thrust acting on other impellers, N,
F_c – total axial thrust acting on impellers, N,
H – delivery head from one pump stage, m,
H_n – delivery head of affluence, m,
H_p – static pressure head at the impeller outlet, m,
H_{sz1} – pressure head at the inlet to radial gap, m,
H_t – pressure head under balancing disc, m,
ΔH_w – pressure drop caused with liquid rotation on space behind the impeller, m,
ΔH_{wl} – head of pressure drop at the inlet to radial gap, m,
ΔH_{sz} – head of pressure drop in sealing gap, m,
Q – pump discharge, m^3/s,
Q_{sz} – volumetric flow rate in the gap, m^3/s,
λ – coefficient of losses in gap, –
ν – kinematics' viscosity, m^2/s,
ξ_{wl} – coefficient of losses at the inlet to radial gap, –
ρ – density of liquid, kg/m^3.

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