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Axial Balance in Centrifugal Pumps – Back Labyrinth Versus Dorsal Vanes

Lecturer PhD Sanda BUDEA[1]

[1] University Politehnica of Bucharest, Hydraulics, Hydraulic Machines and Environmental Engineering Department, 313, Spl. Independentei, 060042, Bucharest, email: s_budea@yahoo.com

Abstract: *Redesign and correct sizing of hydraulic machines, like centrifugal pumps, must take account of hydraulic thrusts. This is useful to minimize the components wear and properly size of bearings and extending the bearings durability. These are minimal conditions for pumps optimal working.*
The article analyzes two methods of balancing the axial forces - first with front and back labyrinths and wear rings of impeller, the second using dorsal vanes. The article presents the calculation procedure of axial forces customized for these two balancing methods. Calculations are exemplified for a multistage pump with 2-7 stages.
Variations of axial forces are analyzed with the number of stages of multistage pumps, with labyrinth positioning or dorsal vanes diameter, width vanes diameter and clearance between vanes and casing and with the suction pressure of the pump. It also indicates the axial thrust reversal and most convenient rotor geometry.

Keywords*: centrifugal pump, axial thrust, front and back labyrinths, dorsal vanes*

1. Introduction

For a correct design of the hydraulic machines' impellers, should be considered the axial and radial forces, their size and direction occurring in the working process. Reliability and correct rating of hydraulic machines like centrifugal pumps must take account of hydraulic thrusts. This is useful to minimize the components wear and properly size of bearings and extending the bearings durability. This article proposes a calculation procedure based on information from the literature [1- 5], [7-9], to determine the correct axial thrust in centrifugal pump impellers, single stage and multistage and comparative analyses of axial balance with wear ring and back labyrinth versus dorsal vanes on the impeller. Axial force variations are analyzed with the number of stages of multistage pumps, diameter of positioning the dorsal vanes, width of vanes and clearance between vanes and casing. Also, the pump suction pressure was varied in the case of balancing with back labyrinth on the impeller and was established when the axial forces are canceled.
All numerical examples were performed on a horizontal multistage pumps manufactured in Aversa Manufactory SRL company.
Axial thrust depends on several parameters: the impeller geometry through the suction and discharge diameters, the hub diameter, the diameter of the front and back labyrinths of the impeller, diameter of dorsal vanes, dorsal vanes width and clearance between it and the casing, the pressure discharged by the pump, also specific operating speed, pump suction pressure or the existence or not of discharge holes on the back disk. By determination of axial forces of centrifugal pumps and by finding solutions to its decrease by a correct geometry of the impeller, increases bearing durability and improves hydrodynamic behaviour of the pump in its ensemble.

2. Axial thrust – theoretical approach

In centrifugal pumps, axial forces occur due to asymmetries of the impeller - the back disk higher than the face disk generates a greater thrust in reverse fluid entry. The clearance between the casing and the impeller that turns fluid in the discharge generates swirls and hydraulic losses. Such effects create four components of axial forces: F_1 due to fluid pressure on the front disc, F_2 on the back disc, due to discharge pressure fluid that penetrates the space between the impeller and casing. F_3 is the force due to the difference of pressure on the shaft end as a result of pump suction pressure and axial impulse force F_4, as the effect of change in flow direction. These components can be seen in figures 1 and 2.

In figure 1, the axial thrust balancing by using back labyrinth, wear ring and discharge holes can be seen. To eliminate the axial thrust of a single suction impeller, the diameter of the both front and back wearing rings can be equal.
In figure 2, the axial thrust balancing using impeller with back vanes is exemplified.
The pressure axial forces, indicated in the literature, [1-5], [7-9] have the form of equation (1):

$$F_{1,2} = \int_{R1}^{R2} p_p \cdot 2\pi r \cdot dr \tag{1}$$

$$p_p = p_r - \frac{\rho \cdot \omega^2}{8}(R_2^2 - r^2) \tag{2}$$

Rotational speed is $\omega = \frac{2 \cdot \pi \cdot n}{60}$
(3)

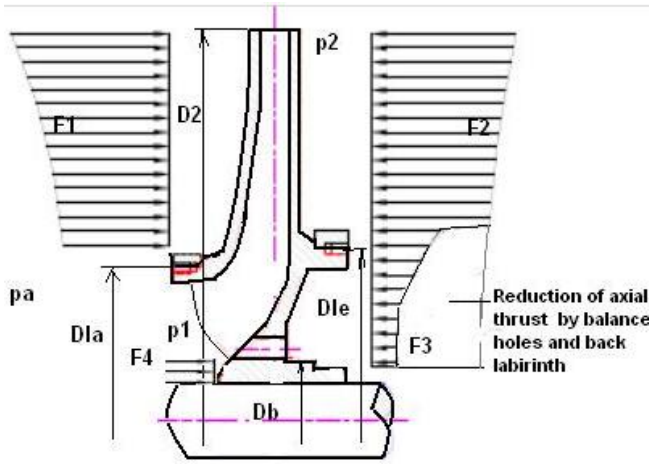

Fig.1. Back side labyrinth and discharge holes

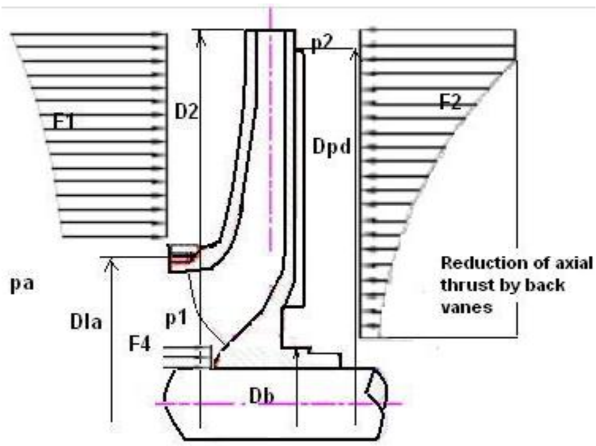

Fig.2. Radial back vanes for axial thrust balancing

Axial force generated by the pump suction pressure has the form of equation (4):

$$F_3 = \frac{\pi \cdot d_b^2}{4}(p_a - p_1) \tag{4}$$

Axial force of impulse has the form of equation (5) or (5') :

$$F_4 = \rho Q c_0 \quad \text{, for radial impeller} \tag{5}$$

$$F_4 = \rho Q(c_0 - c_{m2} cos\gamma_2) \quad \text{, for mixed flow impeller} \tag{5'}$$

The pressure at the discharge depends on the number of pump impellers in Figure 3 is an example CFD modelling to increased pressure in the case of two impellers, leading to increased axial force default. CFD approaches are found in the paper [10].

Analytical set up

Analysed pump is centrifugal multistage with 2÷7 stages, the impeller having the dimensions:

$$D_2 = 128\ mm, D_b = 33mm, D_{la} = 78mm = D_{le},\ n = 2900\ rpm,$$
$$H = (28, 43, 57, 70, 85, 100)m$$

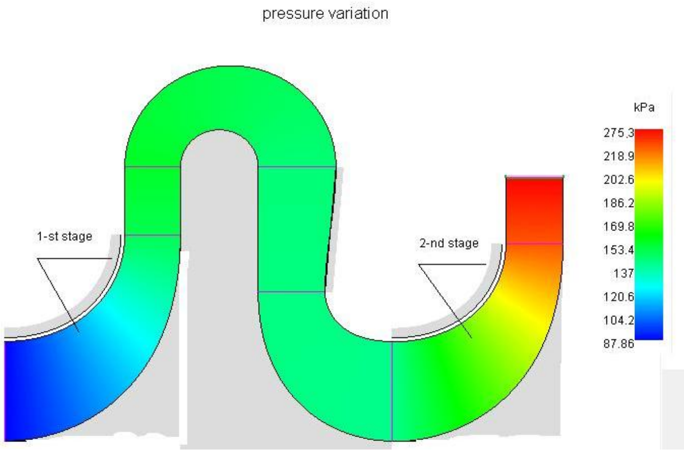

Fig. 3. Pressure variation for two stages of centrifugal pump cu H=28 m and impeller geometry are given below.

3. Axial thrust balanced by back labyrinth and wearing

Relationships above proper for balancing with back labyrinth of the impeller [6], give an axial force like in equation (6):

$$F_{ax} = \frac{\pi}{4}\left[\gamma H_p + \frac{\rho}{8}\omega^2(D_2^2 - D_{le}^2)(D_{le}^2 - D_b^2) - \frac{\rho}{16}\omega^2(2D_2^2 - D_{la}^2 - D_b^2)(D_{la}^2 - D_b^2)\right] \quad (6)$$

The diameter of the back labyrinth of impeller can be calculated so as to decrease the axial force, a good balance being provided with the same diameter for the front and the back labyrinths. The both labyrinth are equipped with wearing rings. In the space between the back labyrinth and the impeller's hub are made n_g discharge holes for a fluid flow from discharge to suction, with diameter d_g . The flow losses by discharge holes are in equation (7):

$$q = n_g \frac{\pi}{4} \cdot d_g^2 \sqrt{\frac{2\Delta p_g}{\rho}} \quad (7)$$

Variation of axial thrust with the number of stages is presented in figure 4. The chart indicates trend line equation for a quickly axial thrust calculus.

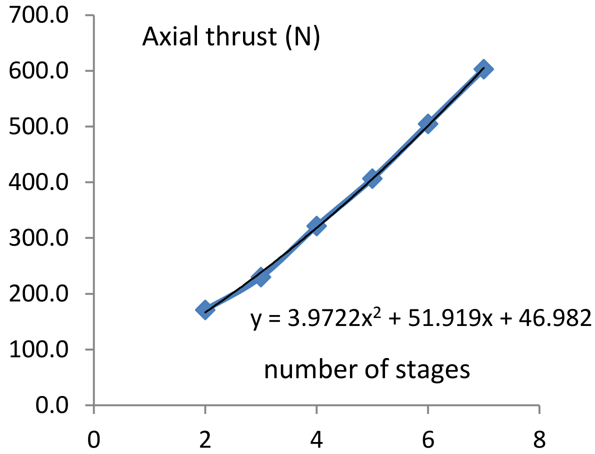

Fig. 4. Axial thrust for back labyrinth vs number of stages

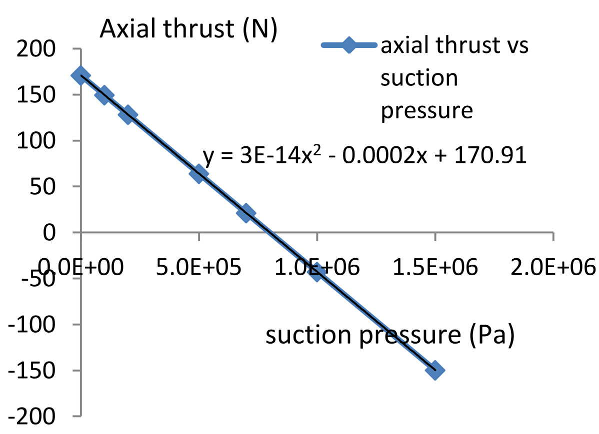

Fig. 5. Axial thrust variation vs suction pressure for a single stage

Figure 5 presents the variation of axial thrust with the suction pressure of the pump and the pressure value that balances the impeller and change the thrust direction. A convenient geometry in radial and diagonal impeller design has the front and back labyrinth diameters equal. Sign convention is: axial thrust positive in the suction direction and negative to the coupling of the pump.

4. Axial thrust balanced by back vanes

In the balancing of axial thrust with back vanes, the axial resultant force [6] has the particular equation (8):

$$F_{ax} = \frac{\pi}{4}\Big[\gamma p_1 + \frac{\rho}{8}\omega^2(D_2^2 - D_{la}^2)\Big(p_{2-}\frac{\rho}{16}\omega_1^2 \;\; (D_2^2 - D_{la}^2\Big) - (D_2^2 - D_{pd}^2)(p_{2-}\frac{\rho}{16}\omega_1^2(D_2^2 - D_{la}^2) - (D_{pd}^2 - D_b^2)\Big(D_b^2 - \frac{\rho\omega_2^2}{2}(D_2^2 - D_{la}^2\Big)\Big] \quad (8)$$

The rotational speeds of the fluid on the impeller disk, respectively on the dorsal vanes are (9):

$$\omega_1 = 0.5\frac{\pi \cdot n}{30},\ \omega_2 = \frac{\pi \cdot n}{60}(1 + \frac{B}{B+j}) \quad (9)$$

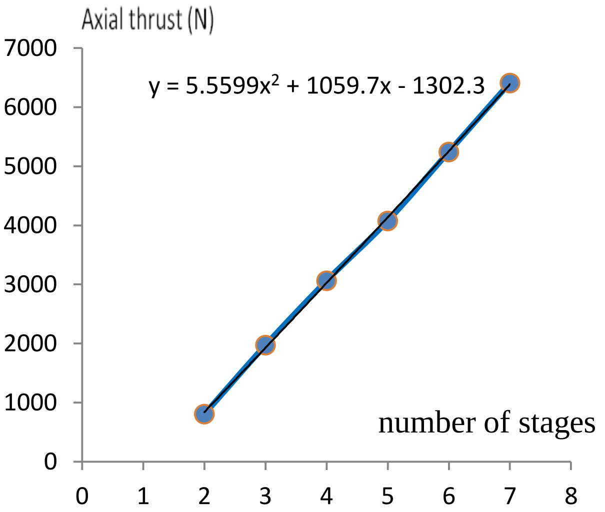

Fig.6. Axial thrust balanced by back blades vs number of stages

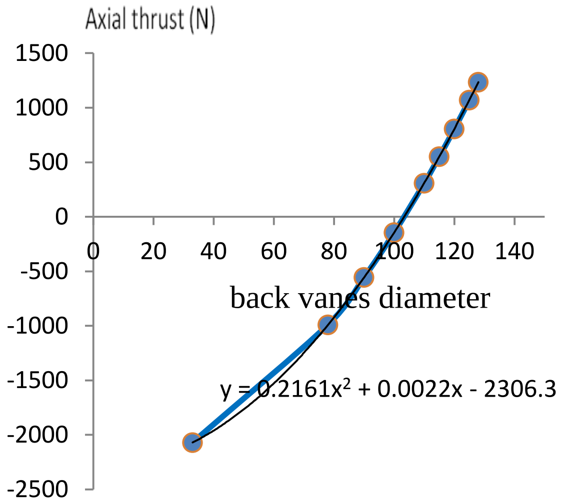

Fig.7. Axial thrust vs back vanes diameter

In figure 6 is analysed the variation of axial thrust using back vanes to balance the impeller versus number of stages of centrifugal pump. The diameter of dorsal vanes to which the axial thrust is zero – the impeller is balanced, can be established, as can see in figure 7. In both cases, trend line equation allows to determine more accurate values and extension of calculation, useful information for centrifugal pump impellers design.

Dorsal blade width *B* is important to balance the axial impeller. Same the gap *j* between dorsal vanes and casing must be properly chosen. The values of axial thrust using dorsal vanes were calculate for four widths (4, 5, 6, and 7 mm) of vanes and three gaps (1, 2, 3 mm) between impeller and casing, resumed in figure 8. Similar analyzes regarding the variation of axial thrust width and dorsal vanes diameter meet also in the paper [9].

5. Discussions and Conclusions

Reason to correct design of the impellers of centrifugal pumps must be calculated radial and axial hydraulic forces. The axial thrusts are great in the case of multistage pumps, in direct proportionality with the number of the stages. Hydraulic forces must be reduced by structural elements of the impeller and / or casing, to ensure the best possible durability of bearings.

This paper analyses two methods regarding the axial thrust balance – the first using front and back labyrinths, the second using dorsal vanes.

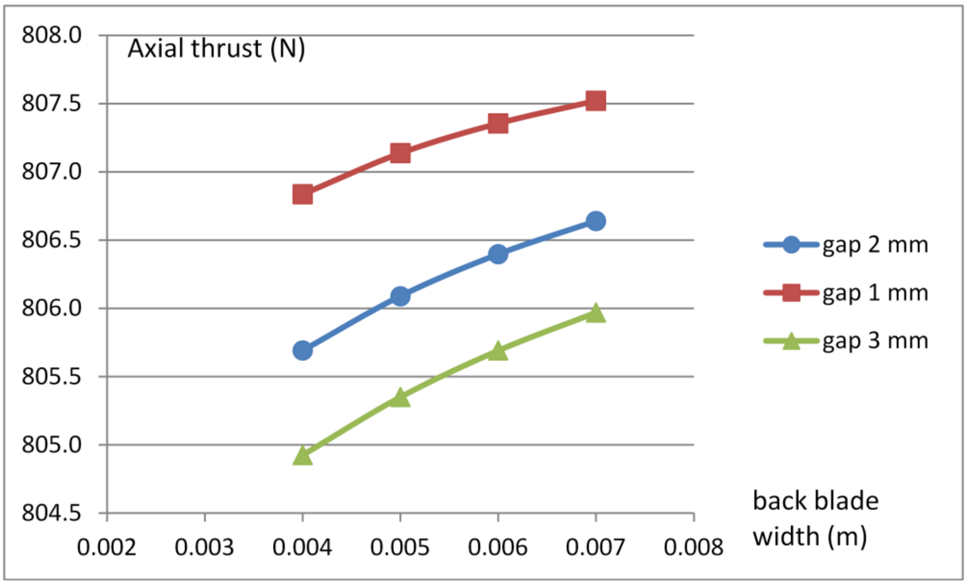

Fig.8. Axial thrust balanced with dorsal blades vs width of the blades and gap between impeller and casing

The article presents the procedures to calculate axial forces, particularized for these two methods of balancing. Calculations are exemplified for a multistage pump with 2-7 stages.
Balancing with front and back labyrinths to the same diameter is a good way and often applied, but for many stages lengthens more the shaft and the pump. It is also important the suction pressure. Article shows the pressure that cancels or changes the sense of axial thrust.
Balancing with dorsal vanes provides a more compact impeller, but the geometry is more complex, increasing the impeller’s weight. It must be taking into account the dorsal vanes diameter and the gap / clearance between impeller and casing. Article presents the variation of axial thrust with the dorsal vane diameter, indicating the sense change of axial thrust. Also, paper presents the variation of axial thrust with the gap between impeller and casing and with vanes width variation.
In conclusion the balancing method with back labyrinth and wearing is more efficient and more convenient for small number of stages. The method with dorsal vanes is the best for balancing a great number of stages, but with a more complex geometry.

Notation

F_p, F_1, F_2 – pressure forces (N)
F_{ax} – axial thrust (N)
C_0 – inlet speed (m/s)
Q –flow rate (m^3/s)
q – flow losses through the discharge holes (m^3/s)
n_g – number of discharge holes
d_g – holes diameter (m)
p_p, Δp – pressure variation (Pa)
p_1, p_2 – inlet, outlet pressure (Pa)
p_a – suction pressure (Pa)
n – rotational speed (rpm)
ω - angular rotational speed (s^{-1})
ω_1, ω_2 – idem for fluid rotation on disk, respectively back blade (s^{-1})
D_2 – outlet diameter of the impeller (m)
D_{la}, D_{le} – diameter at suction labyrinth, respectively back labirinth

D_{pd} – diameter of dorsal blades (m)
D_b – diameter of the impeller hub (m)
γ - fluid specific weight (N/m^3)
ρ - fluid density (kg/m^3)
B – back blade width
J – gap / clearance impeller casing

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