Balancing Axial Thrust in the Single – Suction one stage Centrifugal Pump by Hydraulic Balance Holes.

Abdulkareem Abdulwahab Ibrahim
Babylon University / Engineering College / Mechanical Department
abdulkareemwahab78@yahoo.com

Abstract :

One of the important performance parameter for any centrifugal pump is its bearing life. Bearing life is of more importance , it depends upon two hydraulic forces acting on the impeller, i.e. radial thrust and axial thrust . Axial thrust is dependent on the many aspects , shroud and casing clearances, peripheral shroud speeds, head developed by the pump, impeller geometry. The use of drilled holes through the impeller shroud to the balancing chamber is inferior to the arrangement using a special channel to connect the balancing size of the holes, number of holes and diameter of circle holes . In this paper , derived the equation for determine the radius of circle hole.

Key words :axial thrust , shroud , impeller, balance holes, pressure distribution

الملخص:

واحدة من معطيات الأداء المهمة لأية مضخة طرد مركزي هو المحافظة على عمر البيرنك . أن لعمر البيرنـك أهميـة كبيرة وهو يعتمد على تأثير قوتين هيدروليكيتين وهما يؤثران على البشارة وهاتين القوتين هما القوة القطرية والقوة المحوريـة . القوة المحورية تعتمد على عدة مظاهر، هيكل البشارة، الخلوص الداخلية لجسم المضخة، السرعة المحيطية للبشارة، مقدار الرفـع المنتج من قبل المضخة وأبعاد البشارة .

أن أستخدام ثقوب خلال هيكل البشارة والتي تؤدي الى غرفة التوازن يضمن ترتيب لقناة خاصة تربط بين غرفة التوازن ومدخل البشارة ، أن التوازن بهذة الطريقة يعتمد على قطر الثقب ، وعدد الثقوب وقطر دائرة الثقوب. في هذا البحث تم أشتقاق معادلة لتحديد نصف قطر دائرة الثقوب نسبتا لأبعاد البشارة .

الكلمات المفتاحية: القوة المحورية ، هيكل البشارة، البشارة ، ثقوب التوازن ، توزيع الضغط .

Nomenclatures :

F axial thrust, P_2 pressure at discharge of pump
P_1 pressure at suction of pump, F_a axial pressure force at suction side
F_d axial pressure force on the rear of the back shroud between D_y and d_b
D_y wearing ring diameter, d_b impeller hub diameter
F_D momentum force, ρ_w mass density of water
Q' volume flow rate at suction of pump , C_o velocity of water at suction
P_b pressure at hub impeller , ω angular velocity of impeller pump.
ω_T angular velocity of water between shroud of impeller and casing wall
dF_c centrifugal force , dF_p pressure force, p pressure at any point
d_m mass of element , a_n acceleration of mass element
r radius at any point , U_2 peripheral velocity of impeller at outlet
r_2 radius of impeller at outlet, D_2 Diameter of impeller at outlet
r_1 radius of impeller at inlet , D_1 Diameter of impeller at inlet

1. Introduction

Single suction centrifugal pumps are subjected to an axial thrust because the area opposite the impeller eye is under suction pressure at the front of the back shroud and under discharge pressure at the rear of the back shroud of impeller.

The axial thrust is produced by static pressure on the impeller shrouds which do not part in the generation of the head. The axial thrust depends on the pressure distribution in the space between the impeller shrouds and the stationary casing walls. The fully shrouded impeller is desirable as it reduces, the axial thrust there still remains some residual thrust because the area of the rear of the back shroud is subjected to a pressure approximating the discharge pressure, where as on the front face the axial force is less by reason of the area of the eye, which is subjected only to inlet pressure. The open impeller is subjected to greater thrust because the pressure on the inside and outside of the impeller shroud is unbalanced.[Stepanoff,1957].

Axial thrust balancing can be achieved in several ways, one from these several ways, throttling action sealing ring (wearing ring) and discharge end combined with balance holes. The pressure equalization in pump is affected simply via so called (Hydraulic balance holes).This way is simpler and cheaper from another ways for balancing axial thrust.[Shepherd,1987].

In this paper, the following parameters are discussed:

1.The location of holes (Diameter of hole circle).

2.Diameter of each hole and number of holes.

Many important conclusions predicated.

2. Hydraulic Basic Relationships

In Fig.(1) indicates variation of pressure acting on two sides of impeller. The pressure at outlet of pump (P_2) decrease toward the shaft due to centrifugal forces acting on the flow in the space between shrouds of impeller casing wall (volute casing). The pressure at suction of pump(P_1).[Shepherd,1987].

The forces acting on two sides of impeller between diameter(D_2) and diameter (D_y) are equilibrium because there are the same pressure (P_2) acting on the equal areas, as shown in Fig.(1).

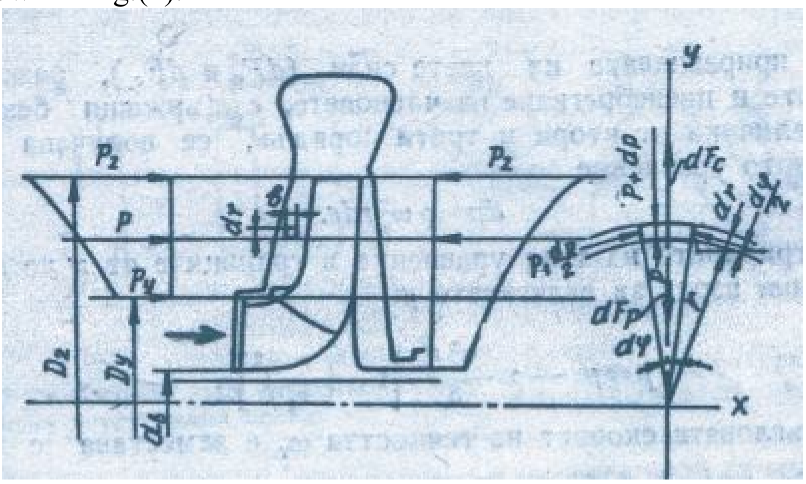

Fig.(1) Axial force acting on the impeller.

The axial force is the resultant of all forces acting on the two sides of impeller between (D_y) and (d_p).

There are three forces acting on the impeller :

1. The axial pressure force acting on front of the back shroud of impeller the suction side (F_2), the direction of this force from left to right.
2. The axial pressure force acting on rear of the back shroud of impeller (F_d)

the direction of this force from right to left .

3. The momentum force (F_o) :

$$F_o = \rho Q' C_o \qquad (1)$$

The direction of this force from left to right. The angular velocity (ω) of water between the shroud of impeller and wall casing is equal half of the angular velocity of impeller:

$$\omega_t = \omega / 2 \qquad (2)$$

The reason for this is as follows : the water in the impeller shroud clearance spaces only starts to rotate gradually during the starting process as a result of the impeller side friction, and it is entrained on one hand by the action of impeller shrouds and retarded on the other hand by the braking effect of the stationary casing surfaces.[Stevan,1987].

In addition, the inward directed gab flow in the clearance space at the suction (outer) and further excites the impeller side turbulence, as a result of coriolis accelerations. For determine the pressure at any point in the back shroud of impeller. Taking element of water between the back shroud of impeller and casing wall, as shown in Fig.(1).[Grozev,1979; Zlatarv,1982].

$$dF_c = dm\ a_n \quad (3)$$

$$dF_c = \rho\ r\ d\varphi\ dr\ b\ \omega_t^2\ r \quad (4)$$

$$dF_b = rd\varphi\ b\ (p + dp) - r\ d\varphi\ b\ p - 2\ b\ dr\ (p + \frac{dp}{2})\ \sin\frac{d\varphi}{2} \quad (5)$$

By equating the two forces dF_c and dF_b :

$$dF_c = dF_b \quad (6)$$

After neglecting some terms and cancel another terms , then ,

$$dP = \rho\ \omega_t^2\ r\ dr \quad (7)$$

Integration of equation (7) from r to r_2 :

$$P_2 - P = \rho \frac{\omega^2 r_2^2}{8} \left[1 - \left(\frac{r}{r_2}\right)^2 \right] \quad (8)$$

$$P = P_2 - \rho \frac{u_2^2}{8} \left[1 - \left(\frac{r}{r_2}\right)^2 \right] \quad (9)$$

The Equation (9) is formula for variation of pressure with radius .

Determine the resultant axial pressure force :

$$F_p = F_d - F_o \quad (10)$$

$$F_p = \int 2\pi\ r\ dr\ p - \int 2\pi\ r\ dr\ p_1$$

$$F_p = \int (2\pi\ r\ dr\ (p - p_1) \quad (11)$$

Substitution the value of (P) from Eq.(9) into Eq.(11) and integrating Eq. (11) between the limits (r_b , r_y) and take the value of (P) from Eq.(9) yields :

$$F_p = \gamma\ \pi\ (r_y^2 - r_b^2)\ [\frac{P_2 - P_1}{\gamma} - \frac{u_2^2}{8g} (1 - \frac{r_y^2 + r_b^2}{2r_2^2})] \quad (12)$$

The direction of this force from right to left , because $F_d > F_s$

$$F = F_p - F_o \quad (13)$$

The direction of this force from right to left , because F_o less than F_p

3. Balancing axial thrust by hydraulic balance holes :

Fig.(2) shows the pressure distribution between the impeller shrouds and the casing walls for an impeller(parabolic shape).For balance axial thrust, must be decrease the pressure between wearing ring R_y and impeller hub R_b which acts on rear of back shroud to value at suction pressure (P_1) which acts on front of back shroud at eye of impeller.

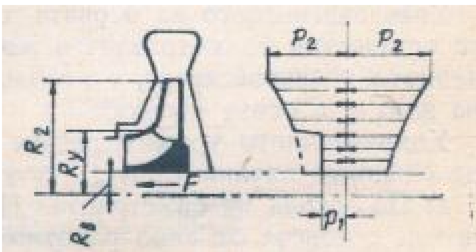

Fig.(2) pressure distribution between the impeller and casing.

The hydraulic balance hole is usually employed to reduce or eliminate axial thrust in single stage pumps. A chamber on back of the impeller is provided with a closely fitted set of wearing ring (sealing ring), and suction pressure is admitted to this chamber by drilling holes through the impeller back shroud into the eye to reduce the pressure in the space between the impeller back shroud and the pump casing (volute). [Anderson,1988] , Fig.(3) .

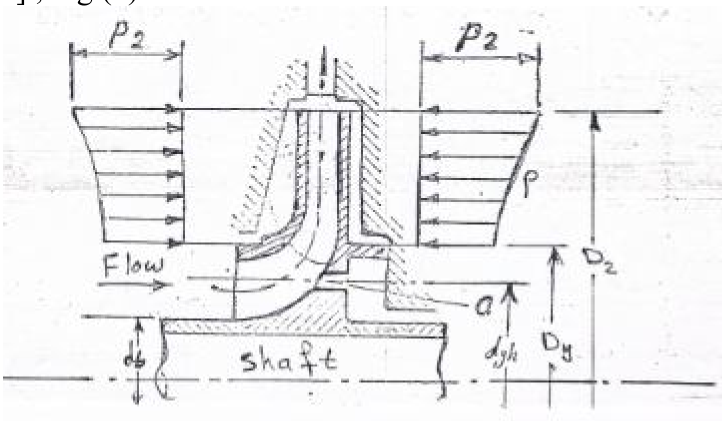

Fig.(3) Axial thrust reduction with balance holes .

Determine the radius of circle holes (d_{yh} / 2) depend on the pressure acting at this radius :

$$\Delta P = \frac{F_p}{A}$$

ΔP is the pressure difference between two sides of impeller from r_y to r_b

$$A = \pi (r_y^2 - r_b^2)$$

$$P - P_1 = \frac{F_p}{\pi \left(r_y^2 - r_b^2 \right)}$$

$$F_p = (P - P_1) \pi (r_y^2 - r_b^2) \qquad (14)$$

Equating the Eq.(14) with Eq.(12) :

$$P = P_2 - \rho \frac{u_2^2}{8} \left[1 - \frac{r_y^2 + r_b^2}{2 r_2^2} \right] \qquad (15)$$

Equating the Eq.(9) with Eq.(15) :

$$r = \sqrt{\frac{r_y^2 + r_b^z}{}} \qquad (16)$$

The Eq.(16) indicates the value of radius of circle holes . For determine the number of holes and diameter of each hole.

In this way for balancing axial thrust , causes leakage loss of the pump, which is decrease the volumetric efficiency of the pump about 2% to 4% .

Therefore , we must be carful when we choose the number of holes , diameter of each hole . The type of balancing can lead to variations in the intake condition , caused by instability in the flow at the mouth of the balance holes on the vane side of the impeller.[Karassik,1988].

Therefore the location of holes ,must be drilled between the vanes of impeller, but the radius computed by Eq.(16).

4. Application the theoretical model for the pump :

Specification of the single – suction centrifugal pump :

1. Delivery discharge of pump (Q) = 300 m^3/hr (0.083 m^3/s).
2. Head of pump (H) = 42 m .
3. RPM of pump (N) = 1450 rpm .

4. Power of motor (P) = 45 KW.
5. Outlet diameter of impeller (D_2) = 0.4 m.
6. Inlet diameter of impeller (D_1) = 0.164 m.
7. Diameter of impeller hub (d_h) = 0.061 m.
8 Number of blades of impeller (Z) = 7 .
9 Specific speed (N_s) = 92.
10 Inlet diameter of pump (D_o) = 0.166 m .
11 P_1 = 40 kpa. Abs.
12 P_2 = 183 kpa. Abs.
13 Inlet discharge (actual discharge) (Q') = 0.086 m^3/s .
14 Volumetric efficiency (η_Q) = 0.96%
15 Overall efficiency (η_t) = 0.78%

Fig.(3) shows the impeller of this pump , applying the Eq.(9) for determine the variation of pressure .

$$P = P_2 - \frac{\rho}{1000} \times \frac{u_2}{8} \left[1 - \left(\frac{r}{r_2} \right)^2 \right]$$

P_2 = 183 kpa , u_2 = 30.38 m/s , r_2 = 0.2 m , ρ_w = 1000 kg/ m^3

Then (P) is :

$p = 67.8 + 2880 r^2$, by use this equation , which the pressure varies with change of radius (parabolic shape) as shown in Fig. (4).

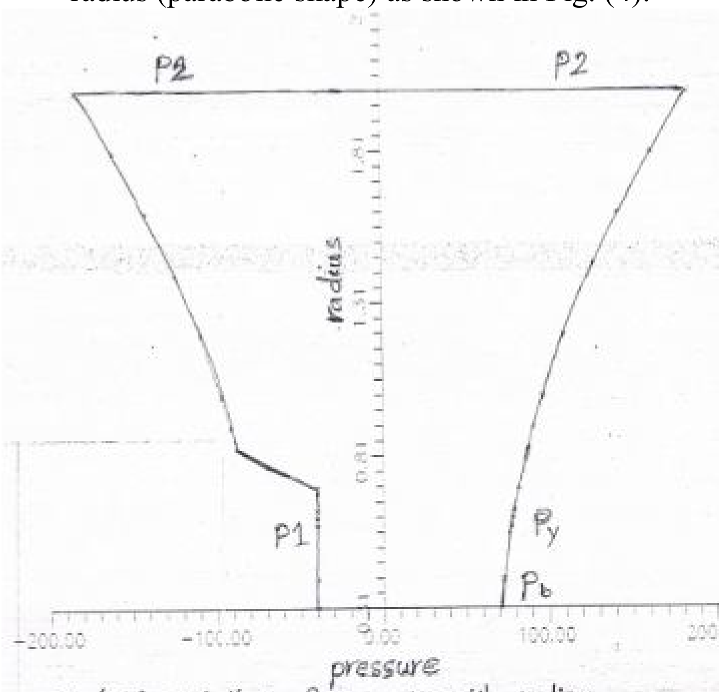

Fig.(4) Variation of pressure with radius

P_y = 87.6 kpa , at r_y = 0.083 m

P_b = 70.5 kpa , at r_b = 0.035 m

1. Applying Eq.(16) for determine the radius of circle holes (r_{yh})

r_{yb} = 0.0625 m

2.Determine the pressure at radius of circle holes :

$p = 67.8 + 2880r^2$

p = 79 kpa

3.Determine the number of holes and diameter of each hole :

$$\Delta Q_h = N_h \ A_h \ V_h \qquad (17)$$

ΔQ is leakage through holes , N_h is number of holes

A_h is area of each hole , V_h is velocity through the hole .

$\Delta Q = Q' - Q = 0.003\ m^3/s$

$N_h = 7,\ V_h = 7.89\ m/s\ ,\ A_h = \frac{\pi}{4}\ d_h^2$

From the above data and from Eq.(17) :

$d_h = 0.008\ m$

$$\eta_Q = \frac{Q}{\Delta Q + Q'} = 93\%$$

That is mean there is decrease in η_Q 3% due to balance holes .

Conclusions :

1. The balancing axial thrust by balancing holes is one way from many ways , but this way is simplest and cheapest way.
2. The balance holes method depends on the size of hole , diameter of circle holes and the number of holes.
3. The optimum location of the circle holes can be determine using the new formula developed in this work , Eq.(16).
4. For avoid the accumulation of liquid in the channel flow through the impeller and do not coincide the addition flow through the holes with the mean flow in the inlet of vane , therefore the holes drilled between vanes of impeller.
5. The leakage through the holes , caused a decrease in the volumetric efficiency , but this decrease is about 3%.
6. The effect of axial thrust on the operation of pump does not appear separately from another effects , but summation of all bad effects conclude effect of axial thrust , and this effect take much time to

appear , like damage in wearing ring , casing of pump and sleeve of shaft.

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