TECHNICAL NOTE

Parameters of Double Suction Centrifuge Pump for the Sugar Industry

Parámetros de bomba centrífuga de doble succión para la industria azucarera

http://opn.to/a/S51eL

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ABSTRACT. The Cuban sugar industry has been working for our designs of centrifugal pumps that date back to the first decades of the last century. As well as other properties and manufacturing under the principles of reverse engineering, using as a standard pumps from prestigious firms such as Ingersoll Rand, Peerless, Worthington among others. Which have presented exploitative problems, causing disruptions in the production process, which are related to the manufacturing process. One reason for this is that the work that aimed to determine the design parameters of a double-suction centrifugal pump for water and sugar sugar juice was carried out in the production facilities of the sugar sector. It describes a methodology for design parameters based on the experience of designers and manufacturers of pumps of international prestige. Within the main results, the value of the specific speed ns = 1 668 stands out, a value that indicates that our design has greater efficiency than the models manufactured and the previous use. The useful power (Nv) and absorb power (Na) with values 32,12 and 45,9 kW, respectively. Lower quantities compared to the useful and absorbent powers of the Allis Chalmers BD 150-83 centrifugal pump. Finally, the proposal became the design parameters for the manufacture of the BCP 125-420 double-suction centrifugal pump, with which it can be passed to the next manufacturing stage.

Keywords: design, efficiency, power, impellent.

RESUMEN. La industria azucarera cubana ha venido trabajando hasta nuestros días con diseños de modelos de bombas centrifugas que datan de las primeras décadas del siglo pasado. Así como otras diseñadas y fabricadas bajo los principios de la ingeniería inversa, utilizando como patrón bombas provenientes de prestigiosas firmas tales como la Ingersoll Rand, Peerless, Worthington entre otras. Las cuales han presentado problemas explotativos, ocasionando trastornos en el proceso productivo, los cuales están relacionados al proceso de manufactura. A causa de ello es que se realizó el presente trabajo que tuvo como objetivo determinar los parámetros de diseño de una bomba centrífuga de doble succión para el trasiego de líquidos (agua y jugo de caña de azúcar), en las instalaciones productivas del sector azucarero. En el mismo se describe una metodología para los cálculos de los parámetros de diseño basada en la experiencia de diseñadores y fabricantes de bombas de prestigio internacional. Dentro de los principales resultados destacan el valor de la velocidad específica n_s= 1 668, valor que nos indica que nuestro diseño tiene mayor eficiencia que los modelos fabricados y utilizados anteriormente. La potencia útil (Nv) y absorbida (Na) con valores 32,12 y 45,9 kW, respectivamente. Magnitudes inferiores en comparación con las potencias útil y absorbida de la bomba centrifuga Allis Chalmers BD 150-83. Finalmente con la metodología propuesta se obtuvieron los parámetros de diseño para la fabricación de la bomba centrífuga de doble succión BCP 125-420, con lo cual se puede pasar a la siguiente etapa de fabricación.

***Palabras clave*:** parámetros de diseño, eficiencia, potencia, impulsor.

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Received: 29/05/2017
Approved: 14/03/2018

INTRODUCCIÓN

Centrifuge pumps are machines directed to displace indeterminate fluids (Pfleiderer, 1959). They are hydraulic machines that transform a mechanical work into another with hydraulic characteristics (Cherkasski, 1986). And their applications range become wider as time passes by since the small pumps used in domestic places up to the large machines used to rise the required volumes directed to the cities' provision (Pérez, 2008).

Particularly, centrifuge pumps are machines used in the majority of industrial processes for the fluids displacement from one location to another. Consequently, the use of centrifuge pumps has undergone a meaningful rising (Reyes e Izquierdo, 2007; Pérez, 2008). As a result, the designing of the turbo- machines components and, especially, of hydraulic pumps, is experimenting a relevant evolution (Pelegrí y Font, 2011).

Diverse procedures have been carried out to evaluate and select centrifuge pumps (Pérez-Barreto, 2009), as well as researches aimed at valuing them departing from their aspiration capacity and summing up the present situation of the construction materials used in their elaboration (Pérez, 2008). In the same way, there have been researches directed to study pressure fluctuations in centrifuge pumps (González *et al.*, 2005). They have tried to find out any parameter which allows diagnosing the negative effects caused by cavitation (Weidong *et al.*, 2017). Besides, they have investigated the effects of different installed devices in the low specific velocity volute of centrifuge pump for the cavitation control (Zhao y Zhao, 2017). It is due to cavitation is a frequent phenomenon during the work with these machines, that affects considerably on their principal technical- economic indicators, though they do not only depends on the system, but also on the quality of the design and equipment's construction.

Finally, it has been also searched on the designing processes of the hydraulic pumps elements making use of 3D simulation tools, in the manufacturing of prototypes for their validation (Pelegrí y Font, 2011) and in the eccentricity influence on complex machines efficiency (Weidong *et al.*, 2017).

The ideas previously stated evidence the importance of these complex machines and of every step articulated in matters of investigation to obtain better and more efficient designs.

One of the Cuban enterprises that use centrifuge pumps, is the sugar industry, which has been working up to the present days with designs of centrifuge pumps models that date back from the first decades of the last century. Other designed and manufactured, following the principles of inverse engineering, have been also used, which have had, as a pattern, pumps derived from prestigious firms such as Ingersoll Rand, Peerless, Worthington, Allis Chalmers, among others.

These pumps have faced operation difficulties, causing disruption in the productive stage related to the manufacturing process. They were made using conventional tool machines based on a complete manual design with typical instruments, such as compass, semicircles, rulers, squares, patterns, etcetera; which did not allow obtaining the required accuracy on the drawing. It is necessary to consider that these pieces demand a great complexity to carry out their design (for example the volute), all represented in 2D. It is also relevant to point out the lack of test banks and especial machines such as the dynamic danglers used nowadays.

This situation provoked that the specialists in pump manufacturing in industries, acquire a higher level of knowledge on methodologies and techniques for designing and production of these equipment, as a determinant way to guarantee an adequate operation.

In this direction, a new centrifuge pump design is proposed, which shows, as a novelty, the inclusion of the double volute in the pump's body to eliminate the radial component that damages the rotodynamic system (particularly in the bearings) present in preceding pumps of simple volute.

In addition, the new design facilitates a better conductivity of liquid to the exterior and, at the same time, contributes to better up the velocity loss conversion and pressure increasing. It also includes disassembled supporting points to ease maintenance work instead of bearers system integrated to the body as in preceding designs.

Taking into consideration the above-mentioned aspects, it was carried out the present investigation whose objective is to determine the designing parameters of a double suction centrifuge pump for liquids displacement (water and sugar cane juice) in sugar cane industry productive plants.

METHODS

The work was developed at "Manuel Fajardo Rivero" factory from Manzanillo Municipality, Granma Province, Cuba; whose address is "Paquito Rosales" Avenue, kilometre 1 and belongs to Industrial Technical Services (ZETI) from AZCUBA firm.

CALCULATION METHODOLOGY

Calculation of the Designing Parameters for the Split Chamber Pump (SCP) 125- 420

To determine the designing parameters of the BCP 125- 420 pump, it was established a methodology based on mathematical equations and suggestions described by Pfleiderer (1959), Pfleiderer (1960); Church (1968); Díaz *et al.* (1968); Karassik y Carter (1978); Cherkassky (1986); Pacheco (1987); así como Ramos (1994).

To determine the flow weight (w) Equation 1 was used.

$$w = Q \cdot \rho \tag{1}$$

Where: Q – is the flow (l min-1) y ρ – is the water density.

Note: for the calculations development, density was used, since the designing of centrifuge pumps is based on the principle of usage of water as fluid at a temperature of 20°C and to 760 mm of Hg above sea level. If work conditions are different, the results are affected with correction factors for flow, pressure, density, number of rotations frequency number and efficiency. Consequently, the distinctive curves are modified, which constitutes an international practice. On the other hand, the sugar cane juice density according to Zhao y Zhao (2017) is ,256 g cm^{-3} and the water's 1,00 g cm^{-3}, values which show a slight difference.

Specific velocity n_s: it was determined through the mathematical expression 2.

$$n_s = \frac{n\sqrt{Q}}{H^{\frac{3}{4}}} \tag{2}$$

Where: n – is the rotations frequency number (r min^{-1}), Q – is the flow (l min^{-1}) y H – is the pump head (mca).

Useful power (Nu): was determined through equation 3.

$$Nu = \frac{w * H}{75} \tag{3}$$

Where: w – is the flow's weight in kg s^{-1} and H – is the pump head en mca.

Absorbed power (Na): its magnitude was determined according to equation 4.

$$Na = \frac{Nv}{\xi} \tag{4}$$

Where: Nv – is the useful power and ξ – is the efficiency which, according to Karassik y Carter (1978), in the case of the object of this investigation, oscillates between 70 and 73 %.

The electric motor's power is (Nm): and its value was estimated by means of equation 5.

$$Nm = 1{,}2 * Na \tag{5}$$

Where: Na – is the absorbed power and 1, 2 is the security coefficient according to Díaz *et al.* (1968).

Torque moment (Mt): was calculated through equation 6.

$$Mt = \frac{6000*75*Nu}{2\pi n} \tag{6}$$

Where: 6 000 is the constant to obtain the calculated torque moment in kg cm, Nu – is the useful power in CV, n – is the number of the electric motor's rotations frequency, π – is the constant 3,14.

The shaft's diameter in the coupling is (D) and its value was determined by means of expression 7.

$$D = \sqrt[3]{\frac{16Mt}{\pi*\delta s}} \tag{7}$$

Where: Mt - is the torque moment, δs - is the shear fatigue. 281,2 kg cm^2. The SAE steel is 1045, 16 is the constant and π - the constant 3.14.

Note: Considering only the torque's value, a calculated diameter shaft would be sufficient. However, to keep critical velocity in a higher range compared to the functioning velocity, it is necessary to increase the diameter value in the coupling. That is the reason why centrifuge pumps manufacturing firms in the international field assume coefficients as a security factor, due to the diverse causes that influence on their work. Hence, Church (1968), recommeds that the diameter value be multiplied by 1,38 security factor. Then:

$$d=1,38*D \tag{8}$$

The permissible relation among the transmitted power velocity, the shaft diameter and the torque is offered by equation 9.

$$N = \frac{S*n*d^3}{371\ 100} \tag{9}$$

Where: N - is the power, S - is the permissible torque moment for steel SAE 1045 (490 kg cm^{-2}), n - is the rotation velocity (r min^{-1}), d - is the coupling shaft diameter (cm) and 371 000- constant.

Other Parameters Object of Calculation. Geometric and Cinematic

The suction diameter of (Dsu) is the geometric parameter determined by means of equation 10, assuming the fact that through the aspiration plate 0,041 m^3 s^{-1} circulates and a velocity with a value of 3 m s^{-1}.

$$Dsu = 100\sqrt{\frac{4*Q}{\pi*v}} \tag{10}$$

Where: Q – is the flow m^3 s^{-1} and v – is the intake velocity.

Velocity in the suction pipe (Vsu) is the cinematic parameter calculated by equation 11.

$$Vsu = \frac{Q}{A} \tag{11}$$

Where: Q – is the flow m^3 s^{-1} and A – is the pipe's area in m^2.

Church (1968) states that velocity in the pump's drain should be slightly incremented. Then the velocity in the pump's drain would be:

$$V_0 = 1,018 * Vsu \tag{12}$$

Where: Vsu – is the suction pipe velocity and 1,018 – is the security factor proposed by Church (1968).

The impeller's diameter in the suction drain (D_1)is the geometric parameter calculated by equation 13, assuming that the pump is a double suction one and leaks would not exceed the 2%.

$$D_1 = \sqrt{\frac{4}{\pi} * \frac{1,02Q*10^4}{2*V_0} + DH^2} \tag{13}$$

Where: Q – is the flow m^3 s^{-1}, DH^2 – is the diameter of the impeller hub and V – is the velocity in the pump's drain.

Tangential velocity at the impeller's intake (U_1) is the cinematic parameter estimated by equation 14.

$$U_1 = \frac{\pi D_1 n}{6000} \quad (14)$$

Where: D_1 – is the impeller diameter at the suction drain (cm) y n – is the rotation velocity (r min^{-1}).

Intake radial velocity(Vr_1) is the cinematic parameter determined through equation 15.

$$Vr_1 = 1{,}09 * V_0 \quad (15)$$

Where: V_0 – is the pump's drain velocity and 1,09 – is the constant recommended by Church (1968).

Intake amplitude (b_1) is the geometric parameter which was calculated through equation 16.

$$b_1 = \frac{10^4 * Q * 1{,}02}{2\pi * D_1 * Vr_1 * \mathcal{E}_1} \quad (16)$$

Where: Q – is the flow (m^3 s^{-1}), D_1 – is the impeller's diameter in the suction drain (cm), Vr_1 – is the intake radial velocity (m s^{-1}) y $\mathcal{E}_1$ –is the contraction coefficient.

Intake angle tangent was determined by means of equation 17 (geometric parameter).

$$\tan \beta_1 = \frac{Vr1}{U1} \quad (17)$$

Where: Vr_1 – is the intake radial velocity (m s^{-1}) y U_1 – is the tangential velocity at the impeller's entrance.

The rotor's exterior diameter is the geometric parameter that was calculated through equation 19. In this direction, an adequate selection of the coefficient's total altitude was carried out ϕ. According to charts to Q=1229 l min^{-1}, H=80mca, $\phi \approx$ 0,95…0,98.

$$D_2 = \frac{8460 * \phi \sqrt{H}}{n} \quad (18)$$

Where: 8460 – is the constant recommended by Church (1968), n – is the rotation velocity (r min^{-1}) y ϕ – is the total altitude coefficient which was determined by charts, depending on the volume Q and the lifting alltitude H. Its value oscillates between 0,95 and 0,98.

The outflow radial velocity (Vr_2) was estimated by equation 19 (cinematic parameter). Referring to Church (1968) this outflow radial velocity should be slightly lesser than the intake radial velocity (Vr_1).

$$Vr_2 = 0{,}914 * Vr_1 \quad (19)$$

Where: 0,914 – is the constant recommended by Church (1968) y Vr_1 – is the intake radial velocity (m s^{-1}).

The outflow surface (A_2) geometric parameter that should be considered by the total volume of 2458 l min^{-1} plus the leaks and it was determined through equation 20.

$$A_2 = \frac{10^4 Q * 1{,}02}{V_r 2} \quad (20)$$

Where: Q - is the flow and Vr_2 - is the outflow radial velocity.

To determine the outflow total surface, as well as its amplitude b_2, (geometric parameter) it is assumed a contraction coefficient $\mathcal{E}_2$=0,85.

$$b_2 = \frac{10^4\, Q * 1{,}02}{\pi D2 * Vr2 * \mathcal{E}2} \quad (21)$$

Where: Q - is the flow, Vr_2 - is the outflow radial velocity, D_2 - is the rotor's external diameter y $\mathcal{E}_2$ - is the contraction coefficient.

The outflow's absolute velocity (V_2) to which the liquid leaves the impeller was calculated by means of equation 22.

To project the volute in the correct way, it is necessary to know the outflow's absolute velocity to which the liquid leaves the impeller V_2 (geometric parameter). Being the external diameter 419 mm and the velocity 1750 r min^{-1} , the peripheral velocity U_2 will be:

$$U_2 = \frac{\pi D_2\, n}{6000} \tag{22}$$

Where: D_2 – is the rotor's external diameter, n – is the rotation velocity (r min^{-1}).

The virtual tangential component (Vu_2) of the outflow absolute velocity was calculated by expression 23 (cinematic parameter).

$$Vu_2 = U_2 \frac{Vr_2}{\tan B_2} \tag{23}$$

Where: U_2 is the peripheral velocity, Vr_2- is the outflow radial velocity, $\tan \beta_2$- is the tangent of β_2.

The effective tangential component (Vú) was estimated by equation 24 (cinematic parameter).

$$Vú_2 = Vu_2 * n\infty \tag{24}$$

Where: $n\infty$ - is the vanes infinite number (in the case of this work is 0,5).

The outflow's effective angle tangent was determined through expression 25 (cinematic parameter).

$$\tan \alpha_2 = \frac{Vr_2}{Vú_2} \tag{25}$$

Where: Vr_2 – is the outflow radial velocity and $Vú_2$ – is the effective tangential component of the outflow absolute velocity V_2.

The outflow absolute velocity was calculated by means of equation 26 (cinematic parameter).

$$V'2 = \sqrt{Vr2^2} + Vú2^2 \tag{26}$$

Where: $Vú_2$ – is the effective tangencial component of the outflow's absolute velocity V_2 and Vr_2 – is the outflow's radial velocity.

The diametric equipment (S) is a geometric parameter determined by equation 27 according to Church (1968).

$$S = 0{,}254 + (D_3 - 152{,}4)0{,}001 \tag{27}$$

Where: D_3 - is the equipment's medium diameter = 178 mm.

The set's surface (A) is a geometric parameter that was estimated by equation 28.

$$A = \frac{1}{2}\pi D_3 * S \tag{28}$$

Where: D_3 – is the equipment's medium diameter and S – is the diametric equipment.

The pressure through the (HI) rings is a cinematic parameter that was calculated through equation 29.

$$HI = \frac{3}{4}\frac{{U_2}^2 - {U_1}^2}{2g} \tag{29}$$

Where: U_2 – is the peripheral velocity, U_1 – is the tangential velocity of the impeller's intake and g – is the gravity acceleration.

The leak's flow QL is a cinematic parameter that was estimated by means of expression 30.

$$\mathrm{QL} = \frac{\mathrm{C*A}}{10^4}\sqrt{2\mathrm{gHI}} \tag{30}$$

Where: C – is the spill coefficient, A – is the equipment's surface, g – is the gravity's acceleration and HI – is the pressure through the rings.

The vane's angle medium value is the cinematic parameter that was calculated through equation 31.

$$\beta m = \frac{\beta 1 + \beta 2}{2} \tag{31}$$

Where: β_1 –is the intake's angle β_2 - is the vane's outflow angle .

The vane's number (Z) is the geometric parameter that was 32 according to Pfleiderer (1960).

$$Z = 6{,}5\,\frac{D2+D1}{D2-D1}\sin\beta m \qquad (32)$$

Where: D_1 – is the impeller's diameter in the suction drain, D_2 - is the rotor's external diameter and βm - is the vane's medium angle.

RESULTS AND DISCUSSION

The flow's weight (*w)* was determined through equation 1, obtaining as a result a value of 40,96 kg s^{-1}. This result overcomes the value of double suction centrifuge pump flow's weight Allis Chalmers BD 150-80 (w=39,28 kg s^{-1}) operated in the sugar industry.

The specific velocity (n_s) was calculated by means of equation 2, obtaining a value of 1 668 and it is non-dimensional. Pacheco (1987), stated that the higher is this parameter, the higher is the pump's efficiency. Thus this value indicates that the proposed design has a higher efficiency compared to the manufactured models previously used and some others that are used in the sugar industry, like the Peerless (n_s= 294,27) and Allis Chalmers BD 150-80 (n_s= 284,51).

The useful power (Nu) was calculated by means of expression 3 and its value is 32,12 kW. This magnitude approaches the value of the useful power of the double suction centrifuge pump Allis Chalmer BD 150-80 which is 32,55 kW. It is necessary to point out that in spite of the fact that the power values are almost similar, the new design has double volute which provokes that the forces interacting on the impeller's circumference be almost uniform Consequently, the radial component affecting the rotodynamic system is eliminated. (and particularly the bearings) included in the preceding pumps of simple volute. In addition, it is possible to conduct in an easier way the liquid to the exterior and at the same time, to better up the velocity loss conversion and to increase pressure. Disassembled supporting points are also included to facilitate maintenance, instead of the current system of integrated platform as in the preceding designs.

The absorbed power (Na) was determined by the equation 4 obtaining a value of 45,9 kW and resulting slightly inferior to the centrifuge pump absorbed power Allis Chalmers BD 150-80 which is 46,5 kW. And, even when the values are not quite different, the new proposed design is inferior due to a series of advantages (together with the above mentioned related to the double volute) such as the modification of relevant adjustments among the impeller, the detrition ring, the axis and bearings. The preceding designs had inadequate adjustments that influenced negatively on the flow, the bearings durability and the bearings platform. The proposed design shows modifications in regards to the impeller, dimensions, cavities and radios; all that with the aim of facilitating the fluid's conduction. It also shows the turbulence chamber amplitude to ease the fluid's intake to the impeller and reduce losses caused by friction and turbulence.

The motor's power (Nm) was determined through the mathematical expression 5 and its value is 43 kW. As a result, these values are taken into consideration. And following the recommendations offered by Díaz *et al.* (1968), who states that the normalized scale for electric motors oscillates between 43 and 52 kW, an electric motor is selected with 43 kW of power and 115 r min^{-1}.

Results Related to the Shaft

The torque moment (Mt) was calculated through equation 6 and the obtained value was 1 77, 30 N m.

To determine the coupling diameter (D) expression 7 was used and its value is 3,18 cm. However, this value should be increased, since it is also required to take into account the flexion moment. In this direction, Church (1968), recommends multiplying this value by a security factor and normalized it subsequently to obtain as a result that the coupling diameter should be 45 mm.

The power (N) was obtained from equation 9 and its value is 158 kW. With this power it is possible to determine the security coefficient Ns, with the aim of testing the shaft's security. The security coefficient value Ns is 2, 87, which indicates that the shaft in the coupling zone is secure, since the load resisted is 2,87 times higher than the motor's power.

The aspiration plate's diameter (Dsu) was calculated by means of equation 10. The resulting value is 13,19 cm, which being normalized, would be 15,0 cm, that is 150 mm.

The aspiration pipe's velocity (Vsu) was calculated through expression 11 and its value valor is 2,32 m s^{-1}.

The velocity in the pump's entrance (Vo) was determined by means of equation 12 and the value 2,5 m $^{s-1}$ was obtained.

The impeller's diameter in the suction drain (D1) was determined by expression13 with a value of 12,6 cm.

The tangential velocity (U1) was estimated through equation 14 and its value is 11,54 m s^{-1}.

The radial velocity (Vr_1) was calculated by means of expression 15 and the obtained result is 2,72 m s^{-1}

The entrance amplitude b_1 was calculated by means of equation16 and its value is 3,23 cm both sides.

The entrance angle tangent ($\tan \beta_1$) was determined through expression 17 and its value is 0,235. Consequently, β_1 is equal to 13 grades.

To calculate the rotor's external diameter (D_2), equation18 was used. Its value is 41,19 cm, that is 419 mm.

The outflow radial velocity (Vr_2) was estimated by means of expression 19. Its value is 2,48 m s^{-1}.

To calculate the outflow's surface (A_2) equation 20 was used. The value of this parameter is 168,63 cm.

The width of the outflow's total surface (b_2) was obtained through equation 21. To carry out its calculation it was taken into consideration the contraction coefficient and the value obtained is 1,50 cm.

The peripheral velocity (U_2) was calculated using expression 22, obtaining a value of 38,39m s^{-1}.

The virtual tangential component of the outflow absolute velocity (Vu_2) was determined through equation 23 and obtaining 31,5 m s^{-1}.

The effective tangential component of the outflow absolute velocity V_2 is $Vú_2$ and it was estimated by equation 24. To calculate this component it was taken an infinite value of vanes and a result of 15,75 m s^{-1} is obtained.

The outflow's effective angle tangent ($\tan \alpha_2$) was calculated by means of expression 25. Its value is 8,94 grades.

The outflow's absolute velocity ($V^{\prime}2$) was determined through equation 26. Its value is 15,9 m s.

The diametric equipment (S) was obtained through equation 27. Its value is 0,28 mm.

The equipment's surface (A) was calculated by means of expression 28. Its value is 0,78 cm^2.

The pressure through the rings (HI) was estimated by equation 29. Its value is 55,6 mca.

The leak's flow (QL) was determined through equation 30 and it was obtained a value of 0,0009 $m^3 s^{-1}$ both sides of the impeller, due to the fact that the pump has double volute.

To calculate the medium value of the vane's angle (βm) equation 31 was used. The obtained result was 16,5.

The vanes' number (Z) was estimated by means of the formula 32, obtaining as a result a value of 3,5. However, for its practical use a value of 5 is used.

CONCLUSIONS

- The designing parameters for the manufacturing of the BCP125-420 centrifuge pump were obtained, which guarantees to undertake the next production stage.
- A calculation methodology to determine the designing parameters of the BCP 125-420 double suction pump is proposed based on the designing basic foundations reported by several investigators, as well as the criteria of the specialists in charge of the production.

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