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Tip clearance effects on loads and performances of semi-open impeller centrifugal pumps at different specific speeds

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Abstract. Relevant industrial standards or customer's specifications could strictly forbid any device adjusting the axial rotor/stator position, so that tip clearance between semi-open impeller and casing might become a result of the pump machining tolerances and assembling process, leading to big tip clearance variations compared to its nominal value. Consequently, large disparities of global performances (head, power, efficiency) and axial loads are observed with high risk of both specifications noncompliance and bearing damages. This work aims at quantifying these variations by taking into account tip clearance value and pump specific speed. Computational Fluid Dynamics is used to investigate this phenomenon by means of steady simulations led on a semi-open centrifugal pump numerical model including secondary flows, based on a k-omega SST turbulence model. Four different specific speed pump sizes are simulated (from 8 to 50, SI units), with three tip clearances for each size on a wide flow range (from 40% to 120% of the best efficiency point). The numerical results clearly show that head, power and efficiency increase as the tip clearance decreases for the whole flow range. This effect is more significant when the specific speed is low. Meanwhile, the resulting axial thrust on the impeller is very sensitive to the tip clearance and can even lead to direction inversion.

1. Introduction

Centrifugal pumps are commonly used in industrial facilities. Among these, semi-open impellers are well suited for specific applications (such as papermaking or sugar industry where the media being pumped contains coarse and/or long fibrous substances) because this kind of impeller is less likely to clog with solids but is also easy to clean if it does. Instead of having a front shroud, semi-open impellers are characterized with an axial tip clearance from the front wear plate, meaning they are less efficient than closed impellers of the same design. Moreover, relevant industrial standards or customer's specifications could strictly forbid any device adjusting the axial rotor/stator position, so that tip clearance between semi-open impeller and casing might become a result of the pump machining tolerances and assembling process, leading to big tip clearance variations compared to its nominal value. As a result, tip clearance effects have to be carefully considered when using centrifugal pump with semi-open impeller.

The hydraulic performance of centrifugal pumps was widely studied over the last century, leading to a thorough understanding of internal flow, losses and main design parameters [1]. Experimental works especially dedicated to the analysis of tip clearance effects on semi-open impeller performances had shown that for most of the machines, the performance deteriorates as the tip clearance increases [2-5]. More recent studies, based on CFD analysis, have led to a better evaluation of the effect of tip

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clearance on centrifugal pump efficiency and head [6-7]. They came to the conclusion that two main reasons explain the variations in pump performance: the tip clearance flow partially mixes with the core flow when it enters the blade-to-blade channel and then the resulting low stream-wise velocity area is responsible for flow blocking and decreasing pressure rise. However, few studies have focused on the effects of tip clearance variations on the hydraulic efforts. Quantifying these variations at early stage of the design is very important. In fact, mechanical design is based on hydraulic efforts calculation. Knowing accurately the range of variation of these efforts will avoid bearings over sizing and mechanical failures.

Consequently, the present work aims at quantifying the large disparities of performances (head, power, efficiency) and axial loads observed on several sizes of semi-open impeller centrifugal pump by taking into account tip clearance variations. CFD is used to investigate these effects by means of steady simulations in 3D numerical model including turbulent secondary flows.

2. Design considerations and operating conditions

This study only considers the specific case of a close-coupled centrifugal pump, featuring axial intake and radial discharge, with a semi-open single-impeller (Figure 1).



Figure 1. Visualization of single-impeller centrifugal pump.



Figure 2. Sectional drawing of semi-open impeller with front and back tip clearances.



Figure 3. Visualizations of studied semi-open impellers.

The four considered pump sizes correspond to ascending values of specific speed N_s from 8 to 50 (SI units). Each semi-open impeller contains six main blades (Figure 3) with an axial tip clearance c_1 from the front wear plate. The axial thrust balancing is achieved by back radial vanes with a tip clearance c_2 from the casing cover (Figure 2). Only the 1st size pump, related to the lowest N_s value

 $(N_s = 8)$, is fitted with a single volute casing whereas the three other sizes, corresponding to higher N_s values (from 16 to 50), are equipped with double volute casings in order to reduce radial loads. Antirotation vanes are also fixed between the suction flange and the impeller leading edge to prevent unwanted backflows at very low flow rates.

Four pump sizes at different specific speeds are simulated with three tip clearances for each size at 40%, 70% 100% and 120% of Q_{BEP} . This covers a wide range of operating conditions, with flow rates Q_{BEP} varying from 100 to 1800 m³/h while total head H_{BEP} varying in accordance from 50 to 270 m at the pumps best efficiency point (Table 1). Tip clearance variations are studied in the range of \pm 40% and \pm 30% of the rated values from the front wear plate c_1 and from the casing cover c_2 respectively, according to the detailed analysis of the pump machining tolerances and assembling process. Then, values for both tip clearances are fixed complementary for each pump size, so tip clearance variations correspond to a single axial position of the impeller against the pump casing. It means that when c_1 increases, c_2 decreases within the same value.

Table 1. Pumps theoretical characteristics and tip clearance values (front and back).

Pump	N _s [-]	$\begin{array}{c} Q_{BEP} \\ [m^3/h] \end{array}$	H _{BEP} [m]	c ₁ ^{Rated} [mm]	c ₂ ^{Rated} [mm]
1	8	105	270	1.5	2.0
2	16	720	143	2.0	2.5
3	30	1500	102	1.5	2.5
4	50	1825	50	2.0	2.5

3. Computational method

The performance characteristics of these four geometrically different centrifugal pumps with varying tip clearances have been numerically modeled using the commercial CFD package ANSYS.

3.1. Geometrical domains

The geometrical models are built with the CAD software SolidWorks (Released in 2011).

The CAD complete assembly files of each pump size for the tip clearance rated values are first simplified and cleaned in order to keep the main wall surfaces only. These four standard files are then duplicated and modified by moving the axial position of the impeller against the pump casing to obtain both other extreme tip clearance configurations $(c_1^{Min}/c_2^{Max} \text{ and } c_1^{Max}/c_2^{Min})$. Resulting from these operations, a full set of twelve geometrical domains is to be investigated (Figure 4).



Figure 4. Visualizations of used geometrical domain and generated grids

For both meshing and computational reasons, each geometrical domain is divided in several subdomains. First, the three main blocks include the semi-open impeller which is framed upstream by the anti-vortex vanes area and downstream by the volute casing. The back tip clearance from the casing cover is then taken into account by means of a single separated sub-domain including all the radial vanes, from the shaft to the outlet of the impeller and cut along the vanes tip shapes, whereas the tip clearance from the front wear plate requires six unrelated sub-domains to be well represented, from the leading edge to the trailing edge of each main blade and cut along the blades tip shapes. Lastly, the full geometrical domain is extended with an inlet pipe extra domain placed upstream the suction flange and an outlet pipe extra domain downstream the discharge flange in order to apply properly the computational boundary conditions.

3.2. Grid generation

The associated meshes are generated with the ANSYS Meshing software (Release 14.0).

Impeller and casing fluid domains are fully meshed with an unstructured approach due to 3D complexity of these regions. Tetrahedral/hybrid volume elements have been used with suitable cell sizes for the semi-open impeller, the volute casing and the anti-vortex vanes domains. Both inlet and outlet pipe extended domains are however meshed with only hexahedral volume elements by means of a sweep method built on the suction and discharge flanges grid faces with a matching condition. This sweep mesh method is also used for all the tip clearance domains (front and back) with a fixed number of ten layers of cells wherein the clearance thickness (whatever the settings of c_1 and c_2).

The 1^{st} cell thickness is fixed at $2x10^{-4}$ m on any wall surfaces (except for tip clearance domains where all the cells have the same thickness). The purpose of this value is to ensure that flow boundary layers are well taken into account with suitable cell sizes according to the turbulent model used by the CFD solver. These grid parameters involve full meshes comprising some $4.5x10^{6}$ elements.

3.3. Numerical models

Internal flow is numerically modeled based on the meshes previously built by using the CFD solver ANSYS CFX (Release 14.0).

Because of time constraints impacting this study, a numerical steady state model is used in order to reduce computing durations. Since such modeling approach is less accurate than transient model, a meticulous care is taken to ensure that the relative position of the blades trailing edges of the impeller remains strictly the same as the casing volute tongues for all computations of each pump size. Moreover, impeller angular positions are in this way chosen so these stator elements are precisely centered at equal distance between two consecutive impeller trailing edges to avoid any wake turbulence effect in case of rotor/stator interaction. With this assembling method, computations made from different sets of tip clearance values (c_1/c_2) are directly comparable for each pump size, but also for different sizes through an analysis of relative values.

All calculations are performed by defining anti-vortex vanes, volute casing, inlet and outlet pipes as stationary domains, whereas the semi-open impeller and tip clearance (front and back) domains are fixed as rotating blocks. Stationary defined domains are linked together with a standard matching connection using mesh characteristics, while both rotor/stator interfaces framing the impeller are based on the frozen rotor frame change option. The numerical interfaces created between the semi-open impeller domain and all the tip clearance sub-domains (all rotating) use the general connection option that enables to connect meshes without the same characteristics on each side.

The CFX code solves numerically the Reynolds Averaged Navier Stokes (RANS) equations using an unstructured finite-volume method where the solution of the velocity-pressure system is based on a fully coupled approach. In this study, the turbulent model k-omega Shear Stress Transport (SST) is applied with an automatically managed wall function dedicated to flow inside the boundary layers.

The modeled fluid is standard incompressible liquid water at 25 degrees with a reference density of 997 kg/m^3 . Boundary conditions are set according to instructions for an incompressible flow by a mass flow rate at the inlet section and a static pressure at the outlet section.

4. Results and analysis

Geometrical domains have been numerically modeled over a wide operating range according to usage patterns of this kind of pumps, from 40% to 120% of the theoretical best efficiency point of each size.

4.1. Computations post-processing

Flow field within the pump is first checked to validate the reliability of each calculation, especially in the both impeller front and back tip clearance areas (Figure 5).



Figure 5. Visualizations of flow field within the pump

Pump hydraulic performance and loads characteristics are then extracted from computation cases by using the ANSYS CFD-Post software (Release 14.0).

Absolute total pressure values at suction and discharge are both obtained by means of a mass flow averaged function over cross sections placed two diameters respectively upstream and downstream the flanges, as specified it tests measurements standards, leading to the total head of the pump. Torque applied on the shaft is determined by the integration of both pressure and viscous forces over all the wall rotating surfaces of the semi-open impeller, including main blades and back radial vanes, leading to the pump power consumption. Pump hydraulic efficiency is then defined using the ratio between these two values and the flow rate (1).

$$H = \frac{\Delta P_t}{\rho g} \qquad P = T\omega \qquad \eta = \frac{QH}{P} \tag{1}$$

In order to support the comparison of the hydraulic performance between different specific speeds, dimensionless quantities are defined using each time the single case of the rated tip clearances at the best efficiency point as reference value (2).

$$Q^* = \frac{Q}{Q_{BEP}} \qquad H^* = \frac{H}{H_{BEP}^{Rated}} \qquad P^* = \frac{P}{P_{BEP}^{Rated}} \qquad \eta^* = \frac{\eta}{\eta_{BEP}^{Rated}}$$
(2)

In this study, the characteristic loads of the pump only concern the resultant axial thrust applied by the flow, which is obtained by the integration of the static pressure field over all the rotating wall surfaces of the semi-open impeller, including main blades and back radial vanes.

A dimensionless form is also used for the resultant axial thrust as defined below (3).

$$F_z^* = \frac{F_z}{2\rho g H b_2 r_2} \tag{3}$$

4.2. Pump performance

The complementary variations of dimensionless pump total head, power consumption and hydraulic efficiency as a function of the dimensionless flow rate are presented below at different tip clearances and specific speeds (Table 2).

Table 2. Pumps performance and loads extreme variations $(c_1^{\text{Min}}/c_2^{\text{Max}} \text{ vs. } c_1^{\text{Max}}/c_2^{\text{Min}} \text{ at } Q^* = 1)$.

Pump	N _s [-]	ΔH* [%]	ΔP* [%]	Δη* [%]	ΔF_z^* [%]
1	8	15.9	5.2	10.7	164.1
2	16	13.8	6.7	7.1	90.6
3	30	6.1	1.7	4.5	15.9
4	50	3.5	0.9	2.6	12.0

4.2.1. Total head. The typical pump characteristic shapes with total head decreasing when flow rate increases appears whatever the specific speed and the tip clearances values (Figure 6).



Figure 6. Dimensionless head curves of semi-open impeller centrifugal pump for different specific speed and tip clearance values (numerical results obtained with steady-state calculations).

More interesting are the effects of tip clearances showing that the total head is the highest when the tip clearance c_1 is the lowest (for a given flow rate). According to the current knowledge, it is quite understandable that a small axial tip clearance from the semi-open impeller blades to the front wear plate reduces the reverse secondary flow around the tip from the pressure side to the suction side of each blade. It leads to less hydraulic losses inside the impeller involving a higher total head. Consequently it should be clear that the main part of this single effect is linked to the c_1 variation while the influence of the axial tip clearance c_2 from the back radial vanes to the casing cover (which changes in a complementary manner) remains negligible on the pump total head variation. Further, this effect of c_1 is far more sensitive for the low specific speed values than for the high values. Considering only the theoretical best efficiency point of each pump size $(Q^* = 1)$, the dimensionless

total head variation between both tested extreme values of c_1 clearly decreases from 15.9% for $N_s = 8$ to 3.5% for $N_s = 50$.

4.2.2. Power consumption. As with the total head, the typical pump characteristic shapes with power consumption growing when flow rate increases can be seen whatever the specific speed and the tip clearances values (Figure 7).



Figure 7. Dimensionless power curves of semi-open impeller centrifugal pump for different specific speed and tip clearance values (numerical results obtained with steady-state calculations).

The effects of tip clearances are quite significant for low specific speed values for which the power consumption is the highest when the tip clearance c_1 is the lowest (for a given flow rate) whereas they appear to be negligible for the larger specific speed values. Considering only the theoretical best efficiency point of each pump size, the dimensionless power consumption maximum variation reaches values of 5.2% for $N_s = 8$ and 6.7% for $N_s = 16$, which thus only represents respectively one third and one half of the corresponding dimensionless values of total head variations. By contrast, power consumption only varies of about 1.7% for $N_s = 30$ and 0.9% for $N_s = 50$ (Table 2). When front tip clearance reduces, rising power consumption is rather fully in accordance with a simultaneous total head increase. By restricting the reverse secondary flow around the tip of each blade, a low value of c₁ also involves a bigger pressure differential between pressure and suction sides and a higher shear stress in the fluid close to the front wear plate. Both contribute to an increase of the resistive torque applied on the main blades area. However, because c_1 and c_2 values vary in a complementary manner, this increase of the resistive torque linked to the main blades should be partially compensated by the decrease of the other part due to the back radial vanes. This balancing effect is one of the reasons why there is such a large difference between total head and power consumption dimensionless variations.

4.2.3. Efficiency. Except for the pump size corresponding to the specific speed $N_s = 30$, the curves first show that the theoretical best efficiency points don't match the numerical ones which are shifted to larger flow rate values (Figure 8).

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Figure 8. Dimensionless efficiency curves of semi-open impeller centrifugal pump for different specific speed and tip clearance values (numerical results obtained with steady-state calculations).

Much more interesting are the effects of tip clearances showing that the hydraulic efficiency is clearly increasing when the tip clearance c_1 is decreasing (for a given flow rate). All pump sizes are concerned but this effect of c_1 is far more sensitive for the low specific speed values than for the high values. Considering only the theoretical best efficiency point of each pump size, the dimensionless efficiency variation between both tested extreme values of c_1 thus clearly decreases from 10.7% for $N_s = 8$ to 2.6% for $N_s = 50$ (Table 2). These results bring out the high benefits for the semi-open impeller pumps to operate with low front tip clearances with a power over-consumption greatly offset by the total head increase (especially for low N_s values).

4.3. Axial thrust

The variations of dimensionless resultant axial thrust as a function of the dimensionless flow rate are presented below at different tip clearances and specific speeds (Figure 9). The axial thrust is arbitrarily defined as positive in the direction from the suction to the pump bearings.

These curves first show that axial thrust increases (in absolute terms) when the flow rate grows. This trend concerns all pump sizes for all tip clearances values but this effect clearly appears far more sensitive for the low specific speed values than for the high values. Considering only the tip clearance rated values of each pump size, the dimensionless axial thrust variation throughout the entire studied flow range (from 40% to 120% of Q_{BEP}) thus reaches large values of 97.8% for $N_s = 8$ and 135.6% for $N_s = 16$ while it only varies of 41.4% for $N_s = 30$ and 35.9% for $N_s = 50$. Furthermore, another interesting result is the direct effects of tip clearances showing the highest axial thrust values (in absolute terms) for a particular configuration: when the front tip clearance c_1 is the lowest and the back tip clearance is the highest (for a given flow rate). This trend once more concerns all pump sizes but the combined effect of c_1/c_2 is far more sensitive for low specific speed values than for high values. This time, considering the theoretical best efficiency point of each pump size (Q* = 1) only, the dimensionless axial thrust variation between both tested extreme combined values of c_1/c_2 decreases from 164.1% for $N_s = 8$ to 12.0% for $N_s = 50$ (Table 2).





Figure 9. Dimensionless axial thrust curves of semi-open impeller centrifugal pump for different specific speed and tip clearance values (numerical results obtained with steady-state calculations).

The magnitude of the combined effects of flow rate and tip clearances on axial thrust may therefore vary significantly depending on specific speed values. Curves for the large N_s values (30 and 50) both show that these two parameters only have a very limited influence on dimensionless axial thrust. Thus tip clearance variations within both extreme configurations lead to less changes on F_z* than all the typical studied flow range, which is itself reduced to no more than about 50% of a reference value. In addition, this narrow range of variation prevents the axial thrust direction from reversing throughout the entire flow range for both pump sizes, the direction remaining strictly negative under these operating conditions. However, tip clearances and flow rate have a very strong influence on dimensionless axial thrust for the small N_s values (8 and 16), not only on its magnitude but also on its direction. For both pump sizes, axial thrust might be applied to the pump suction (negative sense) at low flow rates while it could become oriented to the opposite side towards pump bearings (positive sense) at high flow rates. Combined with the very large variations of magnitude caused by uncertainties on the real tip clearances values, this makes the bearings design very complex, as it has to be able to absorb very high resultant axial thrust but also to deal with a zero level loading.

5. Conclusions

In the present work, CFD is used to quantify the tip clearance effects on performance and loads for semi-open impeller centrifugal pumps. Four pump sizes at different specific speeds are analyzed (from 8 to 50, SI units), with three tip clearances for each size (related to machining and assembling process tolerances) on a wide flow range (from 40% to 120% of the best efficiency point) by means of steady simulations done using a 3D numerical model including secondary flows and based on a k-omega SST turbulence model.

The numerical results clearly show that total head, power consumption and hydraulic efficiency increase as the tip clearance decreases within the whole flow range. These results bring out the high benefits for semi-open impeller centrifugal pumps to operate with low front tip clearances, with a power over-consumption greatly offset by the total head increase. This effect is more significant when

the specific speed is low. Meanwhile, it turned out that the resulting axial thrust on the impeller is also very sensitive to the tip clearances. This trend concerns all pump sizes but this phenomenon appears again clearly far more sensitive for low specific speed values. In these cases, tip clearances and flow rate have a very strong influence on dimensionless axial thrust, not only on its magnitude but also on its direction. Axial thrust might thus be applied towards the pump suction at low flow rates while it could become oriented to the opposite side towards pump bearings at high flow rates.

This study is part of a process undertaken by Ensival-Moret International on understanding internal flows inside semi-open impeller centrifugal pumps, compared to experimental tests and analytical correlations, to provide accurate and reliable tools for pump selection and bearings sizing. These first results related to the effects of tip clearances on both performance and loads could be complemented later by detailed and local analysis to describe the static pressure field variation on impeller and to quantify the tip clearance flow.

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Nomenclature

Quantities

b_2	Impeller blade outlet height	[mm]	Р	Shaft power	[kW]
c_1	Axial front tip clearance	[mm]	ΔP_t	Absolute total pressure rise	[Pa]
c_2	Axial back tip clearance	[mm]	r_2	Impeller outlet radius	[mm]
F_z	Resultant axial thrust	[N]	Т	Resultant torque on shaft	[Nm]
G	Gravitational acceleration	$[m/s^2]$	η	Hydraulic pump efficiency	[%]
Η	Total head	[m]	ρ	Fluid density	[kg/m ³]
Q	Flow rate	[m ³ /h]	ω	Angular velocity	[rad/s]
N_s	Specific speed	[-]			
S	ubscripts/Superscripts/Symb	ols			
*	Refer to dimensionless quantity		Max	Refer to maximal value	
Δ	Refer to variation of associated quantity		Min	Refer to minimal value	
BEP	Refer to best efficiency point		Rated	Refer to rated value	

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