Analysis the effect of the volute diffuser on unsteady pressure pulsation in a centrifugal pump at large operating conditions

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Abstract: - Impeller-volute interaction phenomenon is a major causes of high-pressure pulsation in a centrifugal pump. Such the pressure pulsation have a considerable impact on performance, vibration, stability and noise in centrifugal pumps. A numerical simulation is conducted to analyze the influence of the volute diffuser geometry on the pressure fluctuation during steady and unsteady calculation with the same impeller for several flow rate values. The main objective is to analyze the characteristics of pressure fluctuations inside the pump with radial and tangential diffuser. For a radial diffuser pump, the obtained pressure fluctuation amplitudes are lower than that in the case of a tangential diffuser. The spectral analysis of the pressure fluctuation was achieved under several flow rates for radial and tangential volute diffuser at four monitoring points set at impeller and volute. The results of the existing analysis has proven that the volute diffuser shape affected the pressure pulsation. In contrast, the tangential diffuser lead to increase of pressure amplitudes compared with radial diffuser. However, numerical results were compared with the experimental data, and an acceptable agreement was obtained. The results obtained in this paper can provide a useful reference for designing the type of volute diffuser in pump.

Key-Words: - Impeller-volute interaction; Centrifugal pump; radial diffuser; tangential diffuser; unsteady pressure pulsation; FFT analysis

1 Introduction

A Centrifugal pump is composed mainly of two important parts, the impeller and the volute. Together, these two component specify the characteristics performance of pumps. In addition, the geometry of the volute plays an important in the creation of pressure pulsations inside the machine. To adapt the various types' pipelines installation, the volute shape is designed in different types. The strong impeller-volute interaction is a major source of generating high amplitude pressure pulsations and flow-induced vibrations,[1]. In recent years, the use of CFD (computational fluid dynamics) software is highly increased for studying and analyzing the unsteady flow field behavior inside pumps,[2]. Several studies were conducted on the geometric parameters of pumps in order to control and reduce the pressure pulsation amplitude and radial forces. [3] Investigated the effects of various pump geometries (blade tip clearance, shroud-to-casing radial clearance, vane arrangement and sidewall clearance) by means of experimental and numerical ways. He found out, on the one hand, that increasing the gap between the impeller and the volute is an effective method to lower the amplitude of pressure pulsation. [4] Reported that there is considerable drop in performance up to 5% in a centrifugal blower with double volute and closed impeller, as the inlet clearance increases. He also highlighted that the impact of the gap on the impeller performance is much less compared to the volute performance. This suggests that the performance of a closed impeller with different volute configuration may be affected by inlet clearance. On the other hand, this method may have harmful effects on the efficiency pump,[5]. As for impeller-volute interaction, many studies, [6–8] demonstrated that the impeller blade geometry has an impact on unsteady pressure pulsations. However, through a numerical simulation, the dynamic and unsteady flow effects inside a centrifugal pump have been investigated taking into account to impeller–volute interaction [9]. Moreover, many researchers have studied, numerically, the unsteady pressure characteristics inside centrifugal pumps, such as in the works of

[10,11]. However, only few authors have conducted research about the effect of diffuser shapes type on unsteady pressure fluctuation. [12] Studied numerically, the effects of the volute geometry variation on radial forces. They found that the gap between the impeller and the tongue affects the radial force.

More recently, [13] numerically investigated the effect of the impeller width on the pump performance characteristics. Results show that flow rate of the pump can be regulated by variable impeller width and that efficiency for this scheme is higher than that for flow bypass.

Based on reducing rotor-stator interaction, [14,15] proposed a special slope volute to decrease pressure pulsation amplitude, and unsteady pressure pulsation characteristics were investigated and compared with the conventional spiral volute. At the blade passing frequency, they found that the pressure pulsation in the slope volute is much smaller than that in the spiral volute. However, the relevant analysis about the influence of the blade trailing edge profile on the performance and pressure pulsations in centrifugal pumps is rarely explored and discussed [16].

In this study, two different diffuser types of volute were designed to study, numerically, their effect on pressure pulsation, radial force and pressure contour features of centrifugal pump, with the same impeller geometry. Firstly, a comparison between numerical and experimental characteristic curves with the base volute configuration is presented. However, results of the use of two different diffuser types are performed and a comparison between there have been conducted. Numerical calculation is applied to compute and achieve the pump performance, pressure pulsation characteristics by mounting four monitoring points on the impeller and volute walls. Finally, a comparison between the effects of different volute shapes on the centrifugal pump are carried out and discussed.

2 Numerical simulation

2.1 Pump geometry

In this study, experimental measurement of [17] was used for CFD validation. The geometric parameters of the impeller are summarized in Table 1. The details of impeller geometry used by [17] are given in Fig. 1.

Table 1. Main geometric data of impeller.

Impeller	Value
Suction diameter, d0	0.150 m
Outlet diameter, d1	0.4084 m
Blade numbers of the impeller, Z	5
Nominal operating condition	
Rotational speed	1490 rpm
Volume flow rate	560 m3/h
Head	49 m

The quality of mesh grid has a great influence on the accuracy of numerical calculation. Near the solid walls, the mesh grids are refined and a boundary layer is created to ensure the y+ value on the surface in a reasonable range [18]. A sensitivity mesh analysis was done by comparing their effects on the pump head. After mesh sensitivity check, it is found that the performance of the model pump does not change more than 0.4% when the overall mesh element exceeds 2x10-6, as shown in Table 2. Finally, the overall mesh element used in the calculation is about 2.4x10-6. This mesh size can give correct results for the pump performance and allow details of the main unsteady flow characteristics involved to be analyzed.

Table 2. Mesh sensitivity check.

Test case	Mesh elements	Convergence precision	Head [m]
Coarse	1,144,208	1.10-4	47.5
Medium	1,784,267	1.10-4	48.7
Refined	2,435,911	1.10-4	48.9

Fig. 1 shows a 3D view of centrifugal pump with radial and tangential diffusers while the impeller geometry remains the same in both designs.

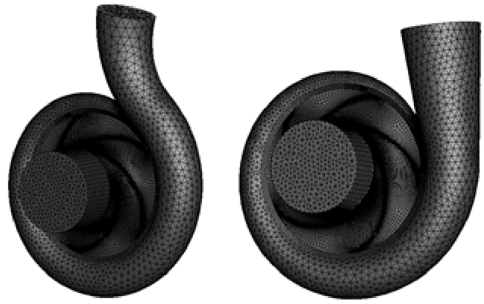

Fig. 1. Sketch of the pump mesh with radial and tangential diffuser

2.2 Turbulence model

In this paper, SST k-ω turbulence model was adopted to simulate the three dimensional flow flied in the currently investigated centrifugal pump. This turbulence model was used to analyze more accurately the wall shear flow such as separation, because its simulation results are closest to the experimental ones, and the accuracy has been validated in many reported research works [19]. The transport equations for the SST model are given:

$$\frac{\partial(\rho k)}{\partial t}+\frac{\partial(\rho k u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}\left[\Gamma_k \frac{\partial k}{\partial x_j}\right]+G_k-Y_k \quad (1)$$

$$\frac{\partial(\rho \omega)}{\partial t}+\frac{\partial(\rho \omega u_i)}{\partial x_i}=\frac{\partial}{\partial x_j}\left[\Gamma_\omega \frac{\partial \omega}{\partial x_j}\right]+G_\omega-Y_\omega+D_\omega \quad (2)$$

Where G_k is the turbulent kinetic energy source term, G_ω is the dissipation equation source term, Γ_k and Γ_ω are the effective diffusion coefficients for k and ω, and D_ω is the orthogonal divergence term.

2.3 Boundary and Calculation Conditions

Firstly, the fluid flow inside centrifugal pump is simulated under steady state and unsteady conditions using a wide range of mass flow rates. For steady state numerical calculation, a multiple frame reference (MFR) approach is applied in order to couple the rotation and stationary domains (impeller and volute). Once the steady flow convergence was achieved, the results were set as initial boundary conditions to compute the unsteady flow field. The general parameters and boundary conditions used for the numerical simulation of the pump are listed in Table 3.

Table 3. Boundary condition.

Flow simulation domain	pump
Fluid	Water 20°
Inlet	Total pressure = 101325 (Pa)
Interface impeller–volute	Steady state Flow: Frozen–rotor Unsteady flow: Rotor–Stator
Outlet	Mass flow = variable (kg/s)

The predicted pump performance curve for both steady and unsteady calculation was obtained by changing the flow rate values.

3 Experimental validations of the CFD results

3.1 Head curve

The experiments for the performance of the centrifugal pump are conducted in the Laboratory of Fluid Dynamics [17]. To approve the accuracy of the obtained numerical results, Fig. 2 shows a comparison of the experimental and the numerically predicted performance curve of the pump using unsteady computation.

The unsteady computation and experimental results under different flow rates are shown in Fig. 2 with radial and tangential diffuser. From Fig. 2, it is observed that the volute diffuser significantly affect the pump head. At partial operating points, the pump with tangential diffuser almost achieves the maximum head value.

It can be seen that, the radial diffuser has an impact on increasing the pump head at 1.2 Qn compared to the tangential ones. However, the numerical simulation with radial diffuser returns more accurate results for the head near the nominal operating condition. Based in Fig. 2, the global characteristics curve of pump from the unsteady model exhibits good agreement with the experimental results at large flow rate.

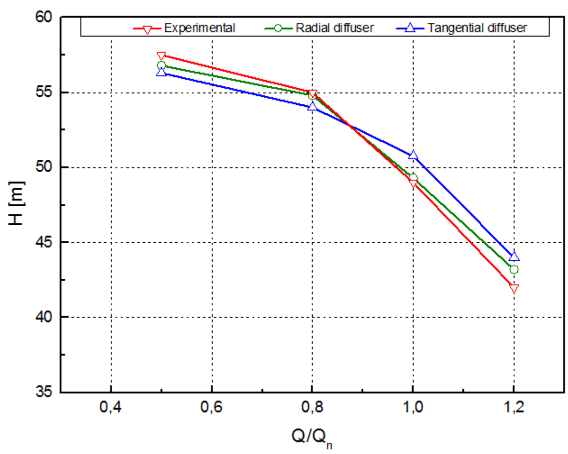

Fig. 2. Experimental and numerical pump characteristic curve.

3.2 Transient head fluctuations

Fig 3 shows the total head variations over one impeller revolution for two diffuser shapes under nominal flow rates conditions.

It can be noticed that there are different wave peaks and troughs, and the value of each peak and trough varies. The explanation for the peaks appearance in unsteady head is that the number of impeller blades is five.

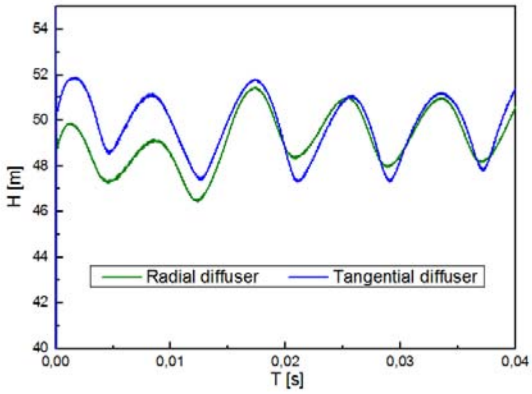

Fig. 3. Temporal evolution of the nominal transient head.

The appearance of the latter phenomenon is mainly due to the presence of unsteady and complex flow field inside the pump. For radial diffuser types, the amplitude variation of the total head curve is smaller than when a tangential diffuser is used.

3.3 Radial force

In this section, the magnitude of the radial force (steady and unsteady components) was computed by means of a full integration of the pressure distribution around the impeller's periphery. Fig. 4 gives the steady radial force curve as a function of flow rate.

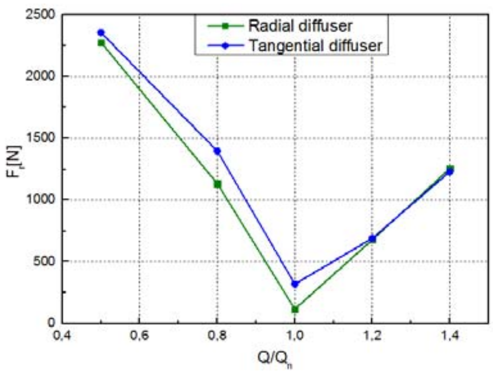

Fig. 4. Steady radial force around impeller.

According to results of Fig. 4, the magnitude of the steady component of the radial force changes with pump flow rate values. It is observed that the radial force magnitude reaches a minimum value at the nominal condition for both diffuser shapes. This force is especially high outside nominal flow rate.

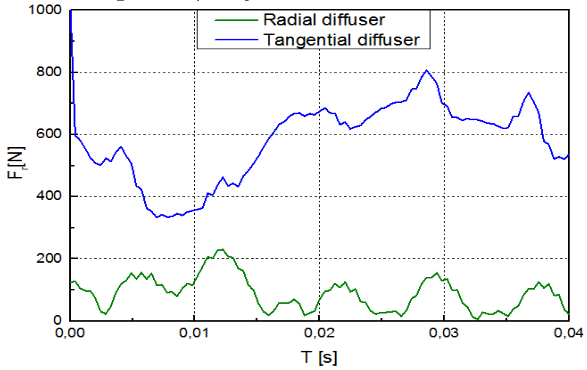

Fig. 5. Transient radial force for two-diffuser type.

The time evolution of the unsteady radial force is presented in Fig. 5 for both volute diffuser. According to the results of Fig. 5, the magnitude of the unsteady components of the radial force changes periodically with times.

The results indicated that a higher pulsation of the radial force is expected for a pump with tangential diffuser.

3.4 Pressure field

With the aid of the CFD tool, the characteristics of the flow fields through a centrifugal pump has been study in the form of a virtual image of the pressure in order to understand more and more the internal complex flow phenomenon. However, it is essential to analyze the influence of volute diffuser on flow structures inside the model pump. Fig. 6 shows a comparison of flow distributions in radial and tangential volute pumps at nominal flow rate. Under running condition, the fluid flow characteristics through centrifugal pump is well homogeneous for both diffuser shapes (radial and tangential diffuser).

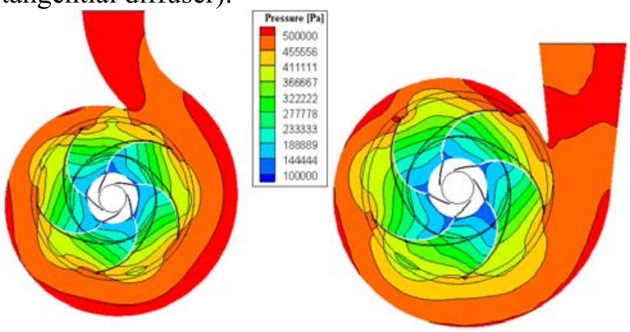

Fig. 6. Pressure contours for radial and tangential diffuser shape under nominal flow rate.

Added to that, Fig. 6 shows a low pressure zone near leading edge of impeller blades located toward tongue due to dropping the local static pressure.

This region is extended more in pump with tangential diffuser. Moreover, if the flow pressure near the volute tongue is less than the vapor pressure, there will excite cavitation at this region [20].

4 Unsteady pressure pulsations

In order to discover the pressure pulsation rule in the centrifugal pump, many measurement points are set in the impeller and volute, as shown in Fig. 7. In total, there are 4 points in the impeller and 5 points in the volute. According to pressure at these nine points, the pressure fluctuation from the impeller inlet to the volute outlet can be obtained and analyzed for two diffuser shape (radial diffuser and tangential diffuser.

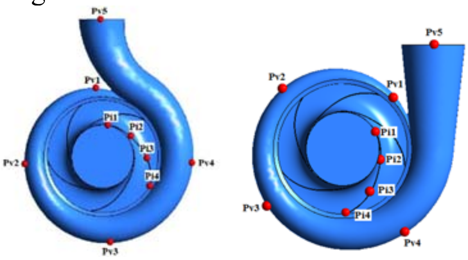

Fig. 7. Location of the monitoring points.

All simulations are conducted at three flow rate values (0.5 Q_n, Q_n, 1.2 Q_n). These points are located at impeller passage (four points) and at volute wall (five points).

Fig. 8 shows the time history of pressure fluctuations on measurements points namely Pi1, Pi2, Pi3, Pi4 and Pi5 for radial and tangential diffuser at 0.5 Q_n.

At partial load condition (0.5 Q_n), numerical comparison of the pressure fluctuation between radial and tangential diffuser was performed. According to Fig. 8, it is observed that the amplitude of pressure fluctuation increasing gradually from inlet to outlet of the impeller (from Pi1 to Pi4), for both diffuser type. Moreover, it is indicated that the variation of the pressure amplitude for radial diffuser is lower than the tangential ones for all monitoring points in the impeller.

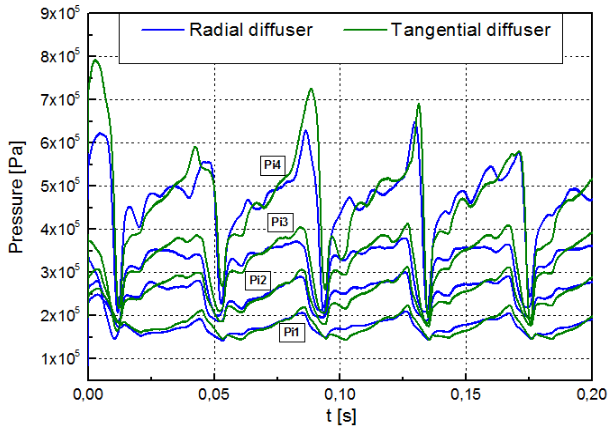

Fig.8. Pressure fluctuation of radial and tangential diffuser at 0.5 Q_n.

At nominal flow rate, Fig. 9 gives the pressure amplitude variation as a function of time for radial and tangential diffuser. For radial diffuser shape, the pressure fluctuation curves for the first three measurement points (Pi1, Pi2 and Pi3) are flat, but from monitor Pi4 peaks start appearing. Compared to tangential diffuser, these peaks appear with large amplitudes on each measurement points (Pi1 to Pi4). The difference between the pressure amplitude obtained by both volute diffuser increases gradually from monitoring point Pi 1 to Pi4.

From Fig. 9, with the same impeller, it is concluded that the pump with radial diffuser obtain a minimum pressure variation in the impeller compared to the tangential diffuser.

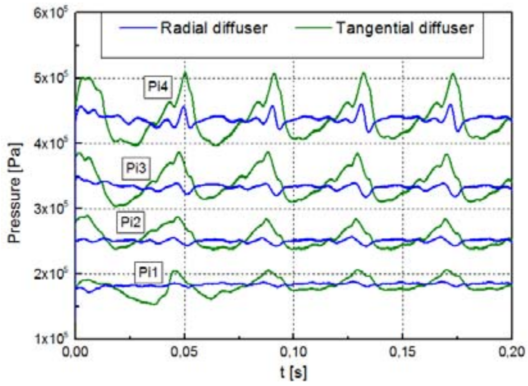

Fig.9. Pressure fluctuation of radial and tangential diffuser at nominal flow rate Q_n.

However, Fig. 10 shows the pressure fluctuation curves at higher flow rate (1.2 Q_n) for the two diffuser. It can be seen that amplitude variation of the tangential diffuser are largest than that obtained with radial diffuser on all measurement points.

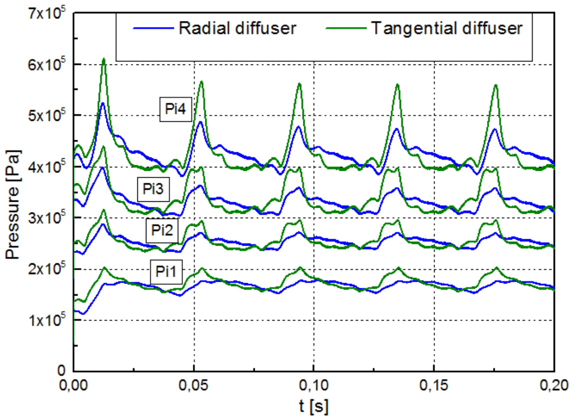

Fig.10. Pressure fluctuation of radial and tangential diffuser at nominal flow rate 1.2 Q_n.

From the above analysis, it is concluded that the energy amplitude of pressure fluctuations progressively increases from the inlet to the outlet of the impeller (Figs. 8, 9 and 10). As observed from these figures, the interactions between the impeller and volute have increasing effects as the impeller radius increases. Therefore, it is concluded that the rotor-stator interactions have more effects on the impeller for a pump with tangential volute than on a pump with a radial diffuser. In general, pressure fluctuation amplitudes, at all monitor points in an impeller, are higher and more important for a tangential diffuser than a radial diffuser. Besides, with same impeller, it is noted that pressure fluctuation characteristics show significant difference with different volute diffuser.

5 Frequency characteristics of pressure in pump

5.1. Pressure spectra in the impeller

In this part, Fast Fourier transform, (FFT processing is applied to display the time domain signals into frequency domain signals. This method is used to process the transient pressure fluctuation, and the frequency characteristics of different monitoring points under three flow rates.

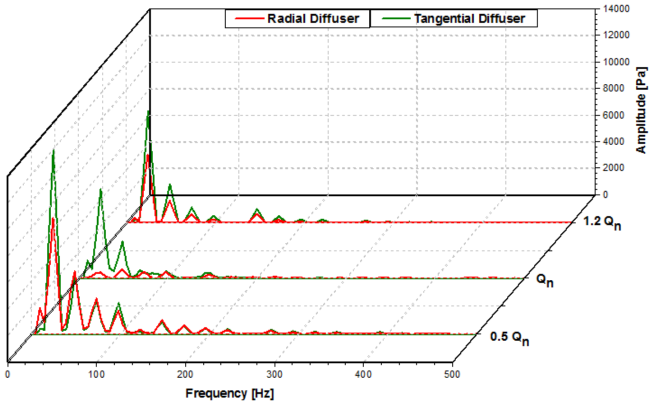

Fig. 11. Comparison of pressure frequency domain Pi1 for radial and tangential diffuser.

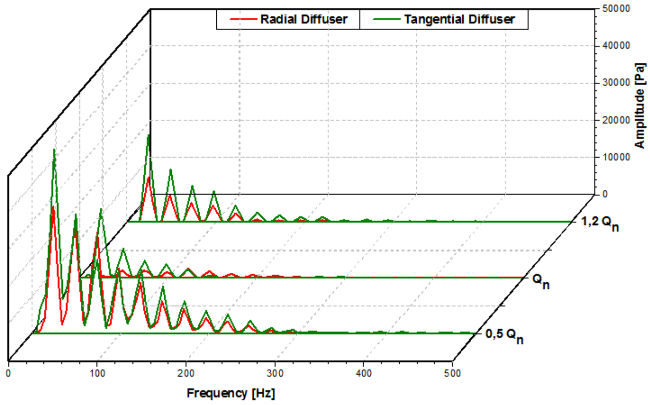

Fig. 12. Spectrum of monitor point Pi2 for radial and tangential diffuser.

Figs. 11 and 12 gives the effect of volute diffuser type and the position of measurement points on frequency spectra curves at two monitoring points (Pi1 to Pi2) for three flow rates values.

At partial flow rate (0.5 Q_n), the flow field behavior inside pump is complex: who leads to occur vortex flow as well as other unstable flow behavior in each blade passage. Consequently, the pressure amplitude under this condition (0.5 Q_n) is much larger than that other operating conditions at Pi1 and Pi2 (see figs. 11 and 12). Besides, the pressure at monitoring point Pi1 is much smaller than that for Pi2 at different operating conditions. From where, these results indicated the fluid-structure interaction phenomenon is more intense at the impeller outlet.

Under the nominal condition, the predominant pressure frequencies of Pi1 are at 49Hz ($2\times f_r$) when f_r is the rotating frequency. Concerning the measurement point Pi2, the pressure frequency is dominant at 122.48 Hz, which is equal to the blade passing frequency ($f_{BPF} = z\times f_r$). Further, it is observed that the diffuser geometry significantly affects the pressure spectra under different operating conditions. The dominant frequency increasing when the pump running at off-operating conditions, especially at 0.5 Q_n. At the nominal operating flow rate (Q_n), the frequency reaches a minimum value. From figs. 11, 12, it can be seen that the severe non-synchronous fluctuation appears under 0.5 Q_n. However, the amplitude of the dominant frequency increases from Pi1 to Pi2, and then decreases at each flow rate. Added to that, the dominant frequency appear when pump operated with tangential diffuser. Further, it is observed that the diffuser geometry significantly affects the pressure spectra under different operating conditions.

5.2. Pressure spectra in the volute

In order to investigate the effect of the diffuser type on the pressure spectra under different operating conditions, monitoring points in the volute wall Pv1 and Pv2 were defined.
Figs. 13 and 14 gives the pressure spectrums under different operating conditions for both diffuser type.

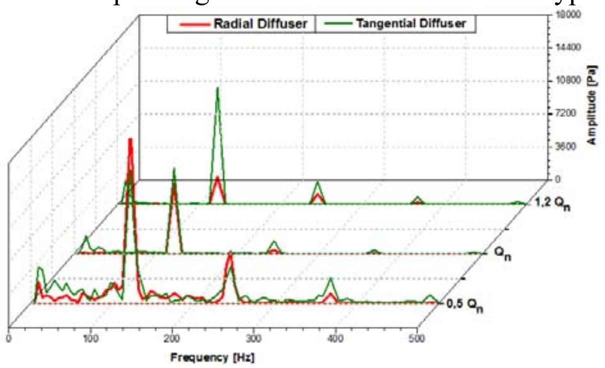

Fig. 13. Pressure Spectrum on Pv1 for pump with radial and tangential diffuser at different operating conditions.

For both diffuser types, the distributions of the pressure spectrum in the volute wall are shown in Figs. 13 and 14. Furthermore, these figures indicates that the prevailing frequency of the pressure fluctuations for all locations in the volute is set at 122.48 Hz, which is equal to the blade passing frequency (f_{BPF}).

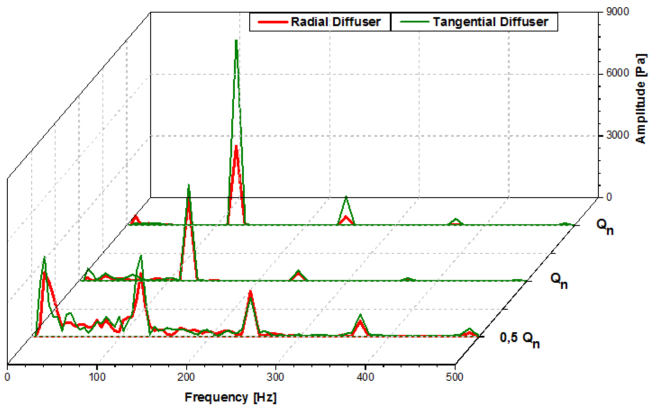

Fig. 14. Pressure Spectrum on Pv2 for pump with radial and tangential diffuser at different operating conditions.

From Figs. 13 and 14, as observed that the smallest spectra of pressure fluctuations are situated on Pv2, where the impact of the rotor-stator interaction phenomenon is trivial. Moreover, the amplitude of the pressure spectrum was found to vary as a function of the pump operating condition. At partial flow, the spectra of the pressure fluctuations are larger and the corresponding flow characteristics are more complex and unstable. With both diffuser type, under nominal flow rate, these spectra are the smallest because the flow field at nominal condition is increasingly smooth and well directed (see Figs. 13 and 14). As can be seen in Figs. 9 and 10, the pressure spectrums dominate for both diffuser types (radial and tangential) at blade passing frequency under nominal flow rate.

Conclusion

In this study, the effect of the volute diffuser shape on the pump characteristic curve, unsteady pressure fluctuation and pressure spectrum was investigated using numerical calculation. The influence of the volute diffuser shape on the unsteady pressure pulsations in a centrifugal pump is studied for several flow rate values. Furthermore, an analysis of the pressure fluctuation in time and frequency domains for both diffuser type of volute was performed. The satisfied agreement between experimental and numerical pump characteristic curves guarantees the accuracy of the numerical computation. With the same impeller and position of the monitoring points, the above analysis presents the pressure spectra at four locations in the pump with radial and tangential diffuser. The obtained results indicate that the impeller-volute interaction phenomenon have a more effects at the impeller outlet than at impeller inlet. Moreover, the pressure pulsations are more and more important when utilizing the tangential diffuser than the radial one.
Based on the above analysis, it is concluded that the volute diffuser type, pump operating points and location of measurement points are significantly affected the unsteady pressure pulsation in a centrifugal pump. The volute diffuser shape has a little affect the pump characteristic curve. At partial load condition, the obtained head with tangential diffuser is greater than radial one. Nevertheless, at higher flow rate, the pump head is lower for the tangential diffuser than the radial diffuser. With same impeller, the pressure pulsation amplitudes in the impeller are increased by utilizing the tangential diffuser. Moreover, the pressure pulsation signals are computed by mounting four measurement points

on the impeller and volute walls. The pressure spectrum characteristics are obtained by FFT analysis. Under different operating conditions, it is observed that the pressure pulsation amplitudes at the impeller are increased by utilizing the tangential diffuser. The amplitude of the pressure Spectrum at the volute is lower than it at the impeller. From that, we know that the impeller is the mainly source of the vibration. The pump operating condition mainly affect the amplitude of the pressure fluctuations, and have a little effect on the composition of the fluctuation frequency.

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