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Experimental and theoretical study of a vane pass frequency for a centrifugal pump

Centrifugal pumps are used for different applications that include pressure boosting, wastewater, water supply, heating and cooling distribution and other industrial processes. This paper presents theoretical and experimental investigations of mechanical vibrations of a centrifugal pump. The flow in this pump, which induces pressure pulsations and mechanical vibrations, have been monitored. Vibration measurements and data collection (overall vibrations levels and frequency spectrum) were extracted from the system. In addition, one of the methods used to study vibration amplitudes for this pump is forced response analysis. To study and analyze the pump system, the finite element analysis software (ANSYS) was applied. Depending on the analysis performed and investigations outcomes, the system natural frequency coincides with the vane-pass frequency (VPF) hazardously. To attenuate the system's vibration, a vibration control element was used. The vibration levels were reduced by a factor of 2 for a tuned element as obtained from a forced harmonic response analysis of the pump system with absorber. It is shown that the inserted element allows the centrifugal pump to work in a safe operating range without any interference with its operation.

1. Introduction

Centrifugal pumps are used for converting the rotational kinetic energy to hydrodynamic energy to transport fluids. Centrifugal pumps might possibly be used in wastewater treatment plants, municipal and industrial water, food processing industries, wastewater as well as noncorrosive liquids with soft solids in suspension. In case there was a mechanical unbalance or liquid stimulation, once running the centrifugal pumps close to the system natural frequency, an excessive vibration can be created. Condition-based maintenance proved to be one of the efficient

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maintenance ways for avoiding unpredicted failures as well as optimizing the maintenance expenditure. The wear due to fatigue together with rubbing in the pump components, attributed to extreme vibrations of the rotor. Each of them takes place as a result of rotor unbalance, the existence of stuck between the driver shaft centerline and the pump, extreme hydraulic force that result for instance from suction recirculation stall, and vane pass rhythms or ordinary occurrence resonance.

Pressure pulsations are attributed to the variability in the main pressure created by the pump. Sometimes, the pulsations might possibly be extremely great resulting in malfunctioning of the pumps [1, 2]. The pressure pulse, in centrifugal pumps, is produced when every rotating vane crosses the cutwater or diffuser vane attaining the highest value once the tip of the vane crosses this point. Once the tip of the vane is close to the cutwater, the peak amount of energy, in the form of fluid in motion, is directed on the way to the discharge nozzle. At every further point, the distance between the cutwater and the tips of the vane is substantial and, as a result, a part of the fluid slides by beneath the cutwater and is accordingly recirculated via the pump. The variation from the lowest to the highest transfer of energy to the discharge nozzle causes consistent alteration or differences in the discharge pressure that create pressure pulsation.

The pulsations of pressure are characterized by two features: amplitude and frequency. Frequency is considered as the number of repetitions of periodic event in unit of time. The frequency of vane is the original frequency created by a centrifugal pump. There is a pulse created whenever the vane tip crosses the cut water. In case there are 6 rotating vanes in the impeller and just one cutwater, for each impeller revolution the 6 vane tips cross the cut water, so that six pulses for each revolution will be created. Equation (1) shows how to calculate the frequency:

$$f = Z(N/60), \tag{1}$$

where: f – impeller vane passing frequency (Hz), Z – number of vanes of impeller, N – speed of the impeller (rpm).

Pressure pulse amplitude is calculated as the highest amount of the pressure by which it deviates from the typical or stable-state value. The device which is used to smooth out unexpected impulse responses, and disperse kinetic energy for a persistent velocity rotary mechanism, is referred to as the shock absorber [3]. Systems of vibration absorption, for instance adjusted mass dampers, have been extensively applied for controlling vibration in the systems of mechanical engineering. Dynamic shock absorber (DSA) theory, contemporarily, has been implemented to decrease structures' vibrations. An effective and durable essential vibration control instrument, normally connected to a vibrating major system for overturning unwanted vibrations brought through various loads, is the adjusted mass damper which comprises a mass, a spring and damping. DSA normal frequency is tuned to resonance with the necessary mode of the major structure, in an attempt that a huge amount of the basic vibrating energy is transmitted to the DSA and after that dispersed via the damping as the major structure is exposed to external turbulences.

The inspection of pump's condition is a significant task for a sustaining efficient and safe production in several industries, for example chemical and oil production. To elaborate the inspection performance, several researches have been carried out in recent years with more progressive data analysis approaches used to deal with the pump house external vibration [4–7]. Numerous researches were conducted on fluxes and features of flow [8], for instance pump features [9] and S-shaped features [10], and the resultant pump-turbines pressure instabilities. The impacts of water compressibility on pressure instabilities have been studied by Yin et al. [11]. A finding was recently achieved that the guide vanes rotation has a noticeable impact on the extent of pressure instabilities in the vaneless area [12].

An innovative technique, based on least-square support vector machine (LS-SVM) and empirical mode decomposition (EMD) has been forward by Zhou and Zhao [13] for the analysis of vibration signals of the centrifugal pump that has the non-linearity and non-stationary features of misalignment defects. Application outcomes indicated that the recommended technique is effective, which might possibly well determine the nonlinear characteristics of the imperfection and more precisely diagnose the deficiencies.

An adaptive network fuzzy inference system (ANFIS) was proposed by Farokhzad [14] to identify the pump failure type. The pump conditions, which include healthy, broken, worn, leakage and cavitation impeller were considered. The FFT technique was used to extract these features from vibration signals. The total classification accuracy, consistent with the outcomes, was 90.67%. The system, accordingly, has a great possibility to work as a smart system for failure diagnosis in practical applications. The mono-block centrifugal pump vibration-based fault diagnosis through wavelet analysis and J48 algorithm was proposed by Muralidharan et al. [15]. Five classical states on the mono-block centrifugal pump, viz., good, cavitation, bearing and impeller faults, were simulated. The CWT was used for extracting different set wavelets features which were classified by J48 algorithm.

The objective of this study was to explore mechanical vibrations causes in an actual case centrifugal pump system and to develop economical corrective actions for vibrations alleviation. With the purpose of further studying and better understanding the phenomena, and to acquire a workable solution that fulfils the objective of this study, a hypothetical and practical work was undertaken.

2. Description of the problem

Throughout the checking process, extreme ($\sim 13 \pm 1.3$ mm/s) vibration rates at the housing of bearings in a pump station comprised of three centrifugal pumps delivering water to a generation plant, were detected. The pump station specifications are listed in Table 1. The suggested maximum velocity for steady safe operation, at the operating speed of the pump, is merely 6 ± 0.6 mm/s. Excessive vibration levels on the connected discharge piping were detected. The peak signals of pump and piping vibration revealed that these maximum values had a distinctive frequency of ~ 170 ± 1 Hz, which corresponded to the VPF of the pumps. Consequently, substantial attention was given to this phenomenon of high levels of vibration, so that investigations and measurements were performed to find out the causes for these vibrations in order to reduce them.

Table 1.

Main specifications of the pump station

Type	Centrifugal pump
Manufactured by	SIEMENES
Model	CRN3-3
Number of pumps	3
Number of blades in each pump	6
Weight	1010 kg
Speed	1700 rpm
Power	3 phase
Maximum left	152 m
Maximum flow	150 m^3/h

The structure of each pump in the pump station is shown in Fig. 1. The diameter of the intake line, which is shown on the left of the figure, is 0.46 m, with a 0.46×0.30 m reducer located exactly before the 0.30 m pump intake flange. On the 0.20 m pump discharge, which is to the right of the figure, there is a 0.20 × 0.30 m reducer, tracked via a tiny spool portion fixed to a 0.30 m check valve. Precisely preceding the valve is a perpendicular elbow inside the 0.30 m line of discharge.

Fig. 1. Pump configuration

All length measurements were with ±0.005 m uncertainty. It is the first elbow in a group of elbows comprised of 4 elbows which outline the upright loop which links the discharge of the pump to a joint header. The further 2 pumps in the station are designed and constructed in the same way.

3. Measurements and analysis

One of the probable sources for failure was the configuration of the thermowell which brings about the turbulence induced by the flow. The flow preceding the thermowell results in vortices which appear at a distinctive frequency of vortex shedding. In the case when the frequency of vortex shedding equals the thermowell's natural mechanical frequency, a resonance takes place. The resonance causes extensive vibration which might possibly bring about fatigue failures. Nevertheless, two diverse geometries of thermowell were constructed and it was found it didn't make any vibration difference. The frequencies of vortex shedding were computed for the geometries of thermowell at suction and discharge settings together. Those were matched to the projected thermowells' natural mechanical frequencies (the mode of 1st cantilever). Natural mechanical frequencies were well-separated from the frequencies of vortex shedding matching American Society of Mechanical Engineers (ASME PTC 19.3). Accordingly, there was no obvious relationship between the problem of vibration and flow-induced turbulence. At that time, numerous diverse operating settings and piping configuration variations were considered. Specially, the impact of adding a damper of one quarter wave-length [16] on the shaking force was investigated. It is anticipated that a substantial (~ by an order of magnitude) drop in the shaking forces possibly will be accomplished through setting adjusted dampers in the lines of discharge. This will considerably decrease the vibrations brought by pulsation. Depending on this modeling work, a one quarter wave-length stub dampers were installed in the line of discharge of the horizontal section. The length of the dampers was about 1.52 m and was set up as demonstrated in Fig. 2. The consequences that follow after the dampers were set up were to some extent amazing. In the general piping system, the levels of pulsation were decreased from the wide range of 13÷34 ± 0.5 kPa to a one little greater than 6 ± 0.5 kPa. In the

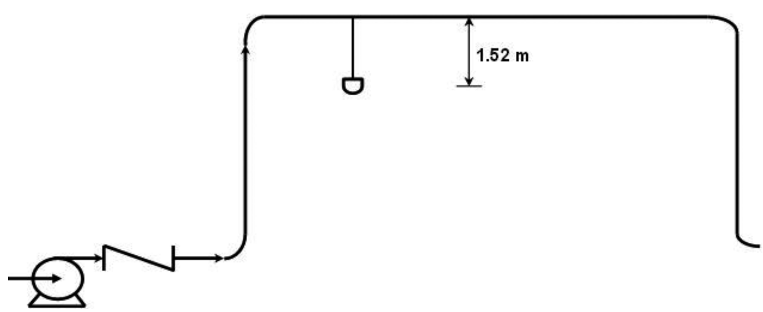

Fig. 2. Damper configuration

piping system, the vibration appeared to be remarkably decreased. Nevertheless, the level of vibrations detected on the pumps was almost persistent. Although the piping system was not still considered as a matter, the vibration levels of the pump remained extremely high. Consequently, pump testing was undertaken with the key intention to seek after the pulsation-induced matters.

With the intention to additionally explore the source of the vibrations and the impact solution, a sequence of pulsation and vibration assessments was carried out. These assessments were made for the vibration accompanying the running of the pump at around 10 positions, although the pump was running at a minimal volume of 150 ± 1 m^3/h. In the major part of these positions, assessments were carried out together in the horizontal direction, that is in line with the axis of the pump discharge track, and in the vertical direction. The assessments, on the exterior bearing housing, were carried out on the housing at a right angle with the pump shaft centerline. Accordingly, the horizontal assessments were undertaken on the lateral of the housing and the vertical assessments were documented on the uppermost of the housing, as showed in Fig. 3. On the termination of the housing, only axial (in line with the pump shaft) assessments were documented.

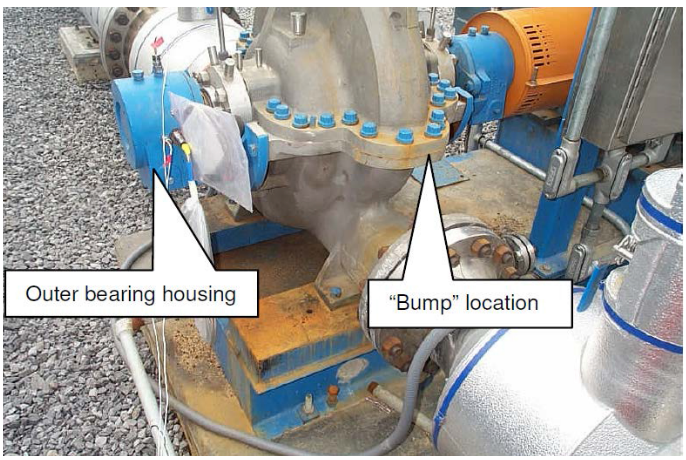

Fig. 3. Locations where the second round of measurements were taken

At the beginning, topmost hold pulsations were calculated at the pump discharge over a velocity extent of 1500 ÷ 2000 ± 0.5 revolutions per minute (rpm). As the velocity increased, the pulsation in the discharge track changed and dropped down at the peak end of the velocity extent as shown in Fig. 4. The topmost pulsation was equal to 34 ± 0.5 kPa. The maximum pulsation frequency in the discharge line was equal to 170 ± 1 Hz which coincides with the pass frequency of the pump vane. It was observed that functioning at varied flow rates (the flow rate was modified by a step at a time by means of a regulating valve in the discharge track), however, at a fixed velocity of 1700 ± 0.5 rpm, the level of pulsation dropped as the flow via the pump remained within the area of peak pump efficiency. This was almost certainly caused by the cleaner run off at the tips of the vane inside the cutwater and the pump volute.

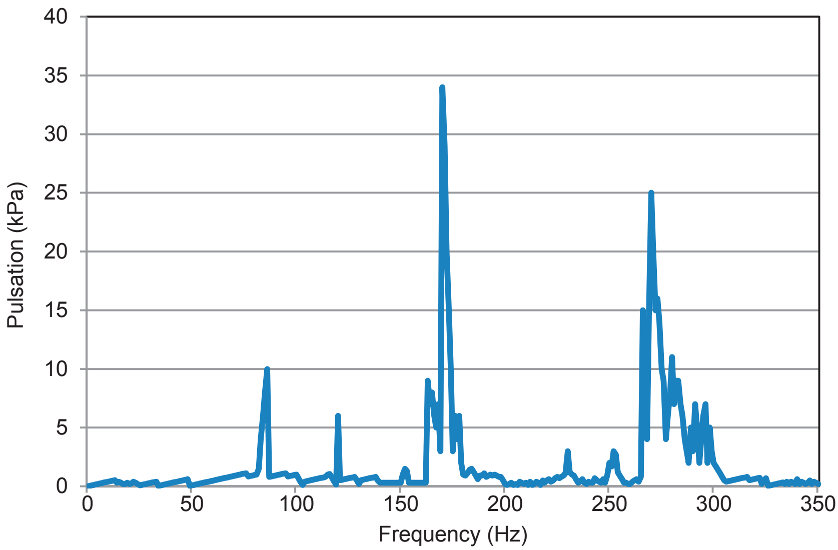

Fig. 4. Pulsations in the discharge piping over 1500–2000 rpm

It was noticed that $13 \div 34 \pm 0.5$ kPa levels of fluid pulsation in the pump lines, at the VPF of the pump, result in shaking of the connected piping. Similarly, the discharge line loop, which is set up to compensate for the typical expansion joint deficiency at the discharge line of the pump, intensifies the vibrations due to pulsation. In addition, the piping vibration has an insignificant amplitude of displacement at the VPF (of the order of ~ 0.1 mm), resulting in real difficultly in suppressing the piping.

The detected vibrations surpass 6 ± 0.6 mm/s merely on the external bearing housing as well as on the piping discharge. The detected pump discharge, also, is smaller than 6 mm/s. Generally, the pump vibrations, with the exception of those on the external bearing housing, are lower than 6 mm/s, whereas the external bearing housing vibrations surpass this value. The results of pump vertical measurements, on both sides (i.e., inlet and outlet), were out of phase with one another, which was the sign that the pump base was shaking. Pump casing and foot measurements, similarly, point at pump shaking.

Insertion of a wedge between the base plate of the pump and the ground caused a meaningful decrease in vibration of the base plate and the external housing of the bearings. The amount of vibration decrease was in the range of 20–25%. Plate vibration was noticed to increase when the wedge location was changed. A comparable vibration decrease in the base plate and the pump took place when the wedge was inserted between discharge line stanchion and piping elbow dead leg.

Vibration level revealed at the external bearing housing was high compared to the vibration level on most parts of the pump. There was an almost threefold increase in the horizontal vibration from the point of housing connection to the bearing housing center. The increase from the point of connection to the housing center, in the vertical direction, was roughly twofold.

The vertical and horizontal vibrations of the connection point between the pump and the bearing housing, as well as between other locations on the pump, were in-phase. However, there was a visible shift of 90-degree between the horizontal and vertical vibrations on the external bearing housing. This relationship was predictable at running speed if unbalances in the rotor of the pump existed. The presence of this phase relation at the VPF was deemed to be an evidence that this was the source of vibration problem.

After discussion and a more detailed analysis, it was agreed upon to exclude the hydraulic resonance as a possible cause for this excessive vibration problem and to focus, instead, on the mechanical resonance at the VPF as the cause for the pump system excessive vibration. Thus, to validate that the incidence of resonance in the system, finite elements analysis software (ANSYS) was used to predict the mode shapes and natural frequencies of the system.

Pump system parts included in the analysis were the pump shaft, the impeller, the motor, the discharge tube, the volute section, the drive shaft, the couplings and the bearings. The discharge tube was represented as a conical profile and the volute part as an envelope generated as a result of rotating adjusted half circles around the revelation axis. The rotation hub distance from the center of the circles increased in analogous order about the volute. Several assumptions were made in constructing the ANSYS model to represent the pump system. The three translations of the shaft line, at motor end, were assumed to be zero at the universal joint. The shaft line bearings' axial displacement was taken to be zero. The shaft of the pump was connected to the frame at the two locations of bearings via four spring elements. Stiffness of the springs were equivalent to the radial stiffness of the particular bearings. The base of pump was attached to the ground at spots of footing bolts. Also, damping was ignored in the simulation and the structure was considered linear-elastic. Material properties of the pump used in the simulation are given in Table 2. Prior to the simulations, a mesh was created as shown in Fig. 5. The pump

Table 2.

Material properties of the pump

Property	Value
Density	7850 kg/m^3
Young's modulus	200 GPa
Poisson's ratio	0.3
Bulk modulus	167 GPa
Shear modulus	77 GPa

was made up of 334841 tetrahedral elements of type SOLID187. This is a 10-node tetrahedral element with three displacement degrees of freedom and quadratic displacement behavior.

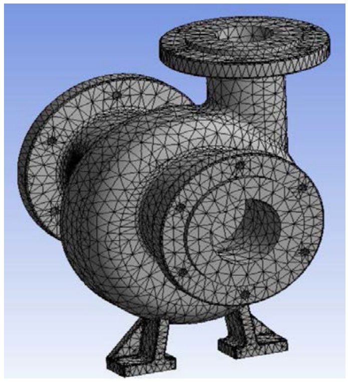

Fig. 5. The pump after meshing

Mode shapes and natural frequencies were determined following the procedure given in [17]. The finite element model of the pump was used to investigate the forced vibration response situation by solving the linear matrix equation given in [17]. The impeller rotating force vector was taken as the harmonic excitation in the system, which was represented by complex notation with a shift phase. In order to solve these equations, it is necessary to find the eigenvalue and eigenvector. Needless to say, the eigenvalue is connected to the natural frequency, while the eigenvector is related to the mode shape. From the analysis, several predicted natural frequencies for the pump system were obtained, as shown in Table 3. The corresponding mode shapes are presented in Fig. 6.

Table 3.

The natural frequencies for the pump system as predicted from ANSYS

Mode	Predicted frequencies, Hz
1	89.5
2	171
3	270
4	551
5	853

Among the several predicted natural frequencies obtained for the pump system, interestingly, one of the frequencies was almost an exact match to the measured frequency at the peak vibration amplitude (171 ± 1 Hz vs. 170 Hz) indicating mechanical resonance generation at this frequency. Therefore, it was decided to modify the characteristics of the vibrating system via fixing an expansion joint in exchange for the exiting spool piece in the discharge line.

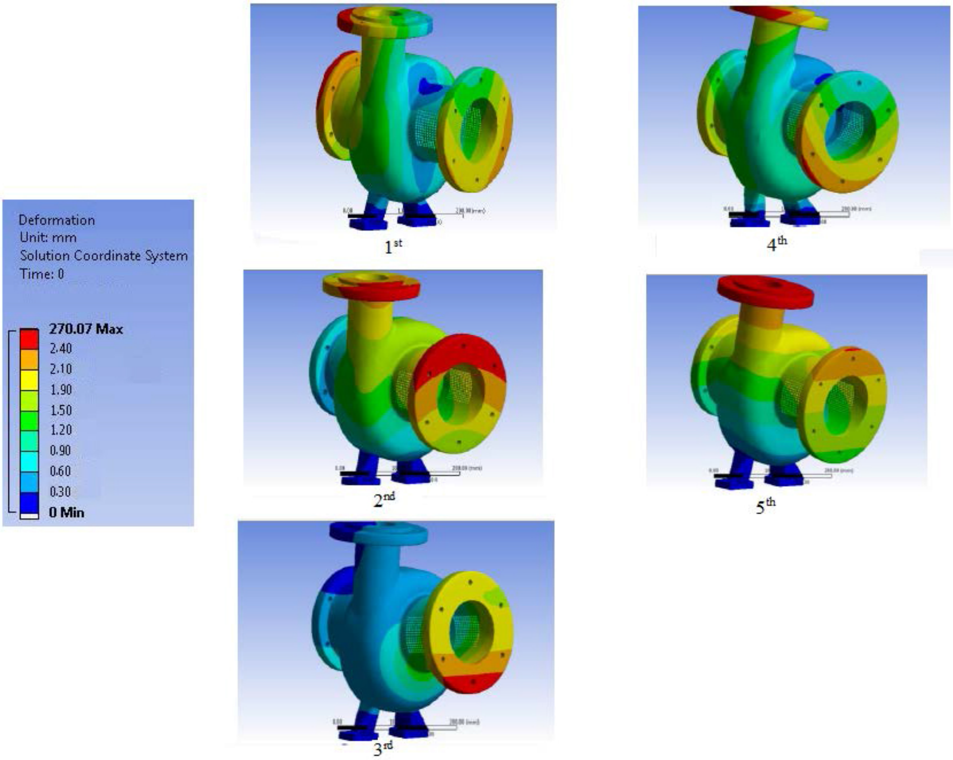

Fig. 6. The mode shapes corresponding to the natural frequencies obtained from ANSYS

In order to place the dynamic absorber at a convenient place for maximum energy dissipation, the structure mode shapes were investigated. Then, the system vibration was alleviated through application of an element to control the system vibration; specifically the spool piece was removed and substituted by a flexible coupling of bellows type, as shown in Fig. 7. Even though the coupling was still rigid, it was remarkably of less rigidity in bending than the spool piece it substituted. Forced harmonic response analysis of the pump system with an absorber installed revealed that the levels of vibration were reduced by a factor of 2. According to field testing, after placing the coupling in the pump system, there was a reduction in the structure's vibration by more than 2-fold at the VDF, as shown in Fig. 8. Thus, the modification of the rigidity of the system significantly led to the removal of the mechanical resonance away from the VPF leading to the generation of reasonable and acceptable levels of vibration.

Pulsation matters at blade pass frequencies in typical working conditions are not comprised in standards applied for screening the several stimulation mechanisms potential on centrifugal pump devices and are not generally investigated in the time of designing the station. It might be very challenging and costly to introduce changes to cope with resonance matters once a pump system has been

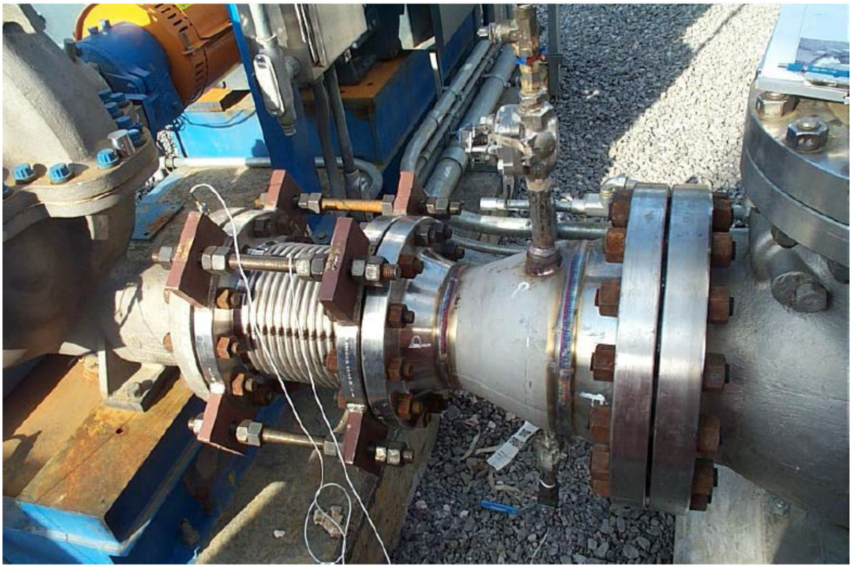

Fig. 7. Flexible coupling

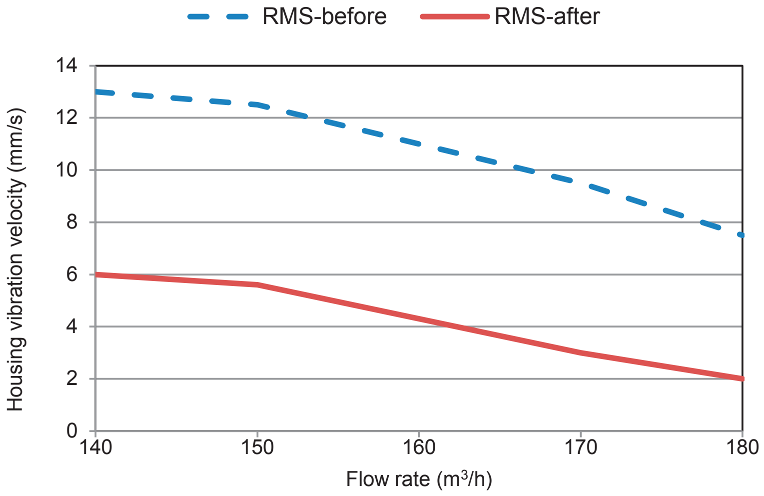

Fig. 8. Vibration before and after installing the flexible coupling

constructed. It is far simpler to adjust a system at the design phase to acquire a sufficient separation boundary concerning the natural frequencies and the source frequencies. Nevertheless, it is very doubtful that this difficulty might possibly be detected during the design phase. The interaction and complexity of several factors, including the pump mount, pump, piping system and other factors, make the disclosure of resonance at $\sim 171 \pm 1$ Hz absolutely beyond question for a regular design.

4. Summary and conclusions

The vibration problem, in the centrifugal pump system addressed in this study, required resolution. It was proposed initially that system detuning might possibly solve the problem. Unluckily, system detuning only influenced the vibration of piping rather than the dangerous and critical pump vibration. Pulsations of piping were decreased by a factor of roughly 2. Piping vibration was almost removed through the setting up of dampers' one quarter wavelength. Pump vibrations were almost unaffected. Pump system was modeled by the application of a finite element analysis. In this situation, the pump system mechanical resonance was found to be very close to the pump's vane pass frequency. The existence of the resonance allows for a comparatively small levels of pulsation, which are predictable in this sort of pump, to stimulate the system. Piping mechanical disconnection from the pump and base was necessary to deal with the extreme pump vibration. Design modifications to adjust resonances were expensive and it was challenging to undertake sufficient modifications to change resonances outside the variable operational range speed of the pump system. To foresee probable advantages of vibration-handling element and to reach an ideal design that might possibly cover the whole extent of predictable operating conditions, the finite element analysis was used. The element was added successfully to the pump system, and vibration levels were cut down to a value acceptable for secure durable operation.

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