Noise sources in centrifugal pumps

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Abstract: - The reduction of noise emission is of vital importance for pumps used in HVAC applications. Several components contribute to the noise emission of a pump, e.g. pressure pulsations originating from the hydraulics or electromagnetic forces in the electrical drive. In order to reduce the noise emission, it is necessary to identify the share of each source in the overall noise level. This can be achieved by vibration measurements and spectral analysis on the pump and appropriate models of the underlying noise-generating processes.
In this paper, the most important noise sources in centrifugal pumps and their electrical drives will be explained. The effect of these sources will be illustrated by measurement results of structure- and fluid-borne sound power. Possible remedies and their effect on noise reduction will be discussed.

Key-Words: - Noise, vibration, pulsation, structure-borne sound, fluid-borne sound

1 Introduction

The noise emission of pumps used in HVAC applications is an important quality and comfort feature. Especially the emission of structure- and fluid-borne noise has to be considered, as these can propagate along the pipes to sensitive locations far-away from the installation position. The propagation and radiation of noise depends on factors like building structure and materials, pipe connections and other components in the installation (e.g. thermostatic valves and heating radiators). As these factors are beyond the control of the pump manufacturer, it is crucial to minimize the excitation of noise at its origin.
In order to achieve a minimum noise level, the main noise sources and excitation processes must be known. These are, of course, related to the hydraulics and the mechanical assembly, but also (especially in the case of pumps with a canned motor) to the electrical drive. For each of these sub-systems, the potential noise sources will be discussed. Only noise sources inherent to the design are included while noise caused by defects is disregarded.

2 Hydraulic noise sources

The hydraulics of pumps consists of one or several rotating impellers and resting diffusers respectively a guide casing. The impellers convey energy according to the Euler equation to the fluid. In the diffuser resp. the guide casing the flow is decelerated and the fluid is guided to the next impeller or the pipe system beyond the pump. The flow inside a pump is three-dimensional, instationary and mostly very turbulent.
Several processes in the pump result in pressure pulsations and consequently excite fluid-borne sound, that propagates in connected pipes, and also excite vibration of the solid boundaries that leads to the radiation of air-borne sound [1].

2.1 Instationarity of flow

The instationarity of the flow is caused by secondary flows in the impeller due to rotation, the finite number of blades and finite blade thickness, but also by effects of turbulence. The finite number of blades leads to a secondary flow, caused by the asymmetric outgoing flow of the impeller. The finite blade thickness causes a notch in the wake flow. Both effects result in a time-dependent incident flow on the resting parts (guide vanes of the diffuser, volute tongue of the guide casing) and consequently excite vibration of these parts.

2.1.1 Pressure pulsations

Pressure pulsations are detected at discrete frequencies that are multiples of the rotating frequency and the number of blades; these frequencies are also called blade passing frequencies (BPF):

$$f_{bpf} = n_b \cdot f_{rot} \tag{1}$$

The amplitude of these pressure pulsations depends on a number of design parameters of impeller and diffuser and operating parameters. One of the most important parameters is the distance between impeller and the volute tongue. Smaller distances typically result in a much higher amplitude of the BPF. However, this

distance also effects the efficiency of the pump. Therefore, in industrial practice often a compromise has to be found between hydraulic and noise specification.
In addition to peaks at the BPF, there are sometimes peaks to be found at lower multiples of the rotating frequency. This is often the case when the number of blades is not a prime number, e.g. an impeller with 8 or 9 blades. In such a case, the impeller can be regarded as a superposition of several virtual impellers, each with a number of blades that corresponds to a prime factor of the actual number of blades, for instance an impeller with 9 blades can be regarded as a superposition of 3 virtual impellers with 3 blades each. Measurements of such impellers often reveal BPFs at the 3^{rd} and 6^{th} multiple of the rotating frequency in addition to the expected BPF at the 9^{th} multiple.
As the pressure pulsations are generated by the wake flow, their amplitude is also dependent on the operating point of the pump. In general, it is infeasible to operate a pump at partial load far away from the best efficiency point (BEP). Especially if the pump is operated at low flow rates, this also means a higher manometric head, i.e. higher differential pressure. As the overall differential pressure of the pump is the mean of the fluctuating pressure, this increase of the mean differential pressure typically results also in an increase of the amplitude of pressure pulsations, caused by an unsteady separation of the flow. Therefore, it is important during the planning of a new installation to select a pump whose BEP is close to the specified operating point. In case, that several alternatives are available, the pump with the lower hydraulics characteristics should be selected. This is especially important for installations with a varying load profile (e.g. heating systems) where the maximum load is only rarely needed. In addition, also the radial hydraulic forces acting on the impeller are minimum at the BEP. As the radial hydraulic force is transmitted via the shaft and the bearing, it will excite vibration of the pump housing and further propagate as structure-borne sound along the connecting pipes.
The amplitude of the pressure pulsations can be reduced by a design of impeller outlet and volute tongue that is optimized with respect to acoustics, as the pressure pulsations are influenced by the interaction of the wake flow and the volute tongue. If the distance between impeller outlet and volute tongue can not be increased due to the hydraulic requirements, noise generation can be reduced by improving the geometry (for instance, the shape of the trailing edge of the blades). Sharp edges and bends at the volute tongue should be avoided. In addition, it is often advantageous to design the volute tongue and the impeller blades in such a way that trailing edges of the blades and the leading edge of the volute tongue are tilted against each other. This can either be achieved with impeller blades that are twisted in axial direction or by a volute tongue that grows in axial direction so that the leading edge rises towards the diffuser outlet. By this tilt of the edges, the smallest gap between impeller and volute tongue will occur at different angular positions, with respect to the rotational axis. As a result, the pressure oscillation will be “smeared” over time, reducing the amplitude of its tonal noise.
For installations with varying load, it is recommendable to use a pump that can adapt to these conditions. For example, in heating installations with thermostatic valves, the flow resistance changes with the positions of the valves. In case of mostly closed valves, a pump without control will provide higher differential pressure that may result in unwanted whistling noise at valves and radiators. An electronically controlled pump recognizes the lower power need of the circuit and will adapt to the new state of the system.

2.1.2 Turbulence

The impact of turbulence on noise excitation is still a subject of basic research. Principally, the flow in pumps (including the sound field) can be calculated by solving an equation system formed by equations for the conservation of momentum (Navier-Stokes equations) and the conservation of mass [2]. The resulting system of partial differential equations has not been solved analytically until today. Numerical solution is only possible for a limited number of cases (e.g. relatively simple geometries, small Reynolds numbers) due to the extreme variety of different sizes of eddies that have to be taken into account. Therefore, for engineering purposes often the time-averaged equations are solved numerically. By this method, the impact of turbulence is taken into account statistically. The representation of small eddies, that may contain significant energy, is extremely simplified. The effect of this approach on the performance of acoustic calculations is a field of research nowadays. However, it may be hypothesized that even small energy-rich eddies have significant impact at least on the generation of fluid-borne sound.
Another major hydraulic source of noise is cavitation in the pump. Yet, as for most applications cavitation is a forbidden state of operation, its effects are not further discussed here.

3 Noise sources in the electrical drive

The electrical drive has to be included in the analysis of noise sources of pumps: This is obvious in the case of canned motors where the electrical drive cannot be separated from the pump, but also for glanded pumps, where the cooling fan of the motor is a dominant source of air-borne noise and the motor itself an important source of structure-borne noise. Noise in the electrical drive can be excited by mechanical forces (e.g. due to imbalance), magnetostrictive forces and radial electromagnetic forces. The noise excitation process also depends on the type of electrical drive: Though typically an asynchronous motor is used, electronically commutated motors (ECM) become more and more important due to their much lower energy consumption.

3.1 Mechanical forces

As for all rotating machinery, imbalance of rotor components can be a source of unwanted vibration. The effect of the imbalance is a revolving centrifugal force that can be detected in the vibration spectrum at the rotating frequency. This effect can be avoided by proper balancing of rotating parts. Defects in the mechanical assembly (e.g. bearing faults) may lead to additional unwanted noise and vibration, but these are not discussed here.

3.2 Magnetostrictive forces

The magnetic flux in the lamination stack causes a relative elongation (known as magnetostriction) that is proportional to the squared flux density B in the stator [1]. As the flux density is proportional to the supply current, the elongation changes periodically, causing vibration of the stator at frequencies that are multiples of twice the supply frequency. For a supply frequency of 50Hz, vibration will occur at 100Hz, 200Hz, 300Hz, etc. (Fig. 1).

Reduction of noise caused by magnetostrictive forces can be achieved by using material with low magnetostriction and uniform pressing of the lamination stack.

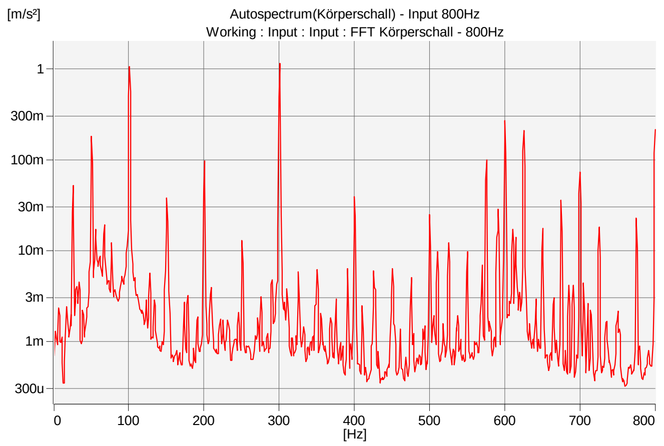

Fig. 1 Vibration spectrum showing strong peaks at multiples of 100Hz caused by magnetostriction

3.3 Radial forces

Radial forces are caused by the main magnetic flux in the air gap of an electrical machine. Due to the finite number of rotor and stator slots, the permeance in the air gap depends on the angular position. In addition, the permeance may vary due to eccentricity or dissymmetries of rotor and stator and magnetic saturation of the lamination stack [4]. By multiplying the permeance with the magnetomotive forces of rotor and stator, the magnetic flux density in the air gap is obtained. The magnetic flux in the air gap causes a radial force, that is proportional to the squared magnetic flux density. As the amplitude of this radial electromagnetic force changes periodically with the rotation, vibration of the stator is excited at particular frequencies that depend on the type of electrical drive, its design parameters and the rotational speed.

3.3.1 Induction motors

For an induction motor, it can be shown, that even if the motor is free from imperfections like eccentricity, rotor and stator dissymmetries and saturation, a radial force will occur at frequencies that depend on the supply frequency f_1, the number of rotor slots Z_{rt}, the slip s and the number of pole pairs p [4]:

$$f_i = \frac{f_1 \cdot Z_{rt} \cdot (1-s)}{p} \pm i \cdot 2 \cdot f_1 \qquad (2)$$

Additional side bands will appear at multiples of twice the supply frequency (Fig. 2). The vibration mode that is excited depends on the number of stator slots Z_{st}, Z_{rt} and p. In the presence of other adverse factors like eccentricity, the amplitude of this radial force may significantly increase, resulting in noticeable noise and vibration.

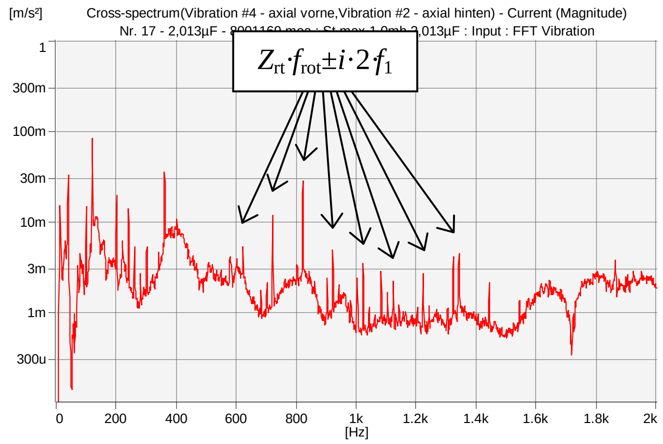

Fig. 2 Vibration spectrum caused by radial force in a single-phase induction motor

In the case of a single-phase asynchronous motor (that is often used for small circulation pumps with a canned motor), these effects can be amplified by the unbalance between the magnetic fields of main and auxiliary winding [5]. The noise caused by radial forces in induction motors can be minimised by adjusting the design parameters (number of rotor and stator slots and pole pairs) in such a way that only higher flexural modes of vibration are excited and that the excitation frequencies do not coincide with resonant frequencies of the stator core [6]. For the mechanical assembly, it is of vital importance to achieve optimum concentricity: Otherwise, the effects of the non-uniform air gap will lead to a significant increase of noise excitation [7]. Defects in electrical components like broken rotor bars or winding faults may also lead to noise and vibration, but are not discussed here.

3.3.2 Electronically commutated motors (ECM)

ECM drives are more and more often used for pumps due to their considerably lower energy consumption: As the magnetic field of the rotor is not created by inducing current in the rotor conductors, but by permanent magnets in the rotor, the internal losses of the motor are dramatically reduced. While the rotational speed of induction motors is limited by the supply frequency, the ECM drive can be designed for higher speeds. This means that size and weight of the pump can be reduced while providing comparable hydraulic power.

Similarly as for the induction motor, the excitation of vibration depends on the magnetic field in the air gap. Again, the non-uniformity of the air gap is a major source of noise excitation. For the ECM drive, eccentricity of rotor and stator will result in noise and vibration excitation at specific frequencies related to the number of rotor poles p and stator slots s [8]. Two extreme cases of eccentricity can be distinguished: static and dynamic eccentricity.

In the case of static eccentricity, the rotor is concentric with its own axis of rotation, but this axis does not coincide with the center of the stator. This results in a radial force directed to the minimum air gap that oscillates as the rotor poles pass by during a revolution. Consequently, apart from a constant component, a radial force will occur at frequencies f_i that are multiples of the rotating frequency times the number of rotor poles p:

$$f_i = i \cdot p \cdot f_{rot} \tag{3}$$

For dynamic eccentricity, the axis of rotation is concentric with the stator, but not with the center of the rotor. This means that the position of the minimum air gap is also rotating, resulting in a radial force vector that is revolving at the rotating frequency. Additionally, this force will oscillate as the minimum air gap passes by the stator slots during a revolution. Consequently, apart from the rotating frequency, a radial force will occur at frequencies f_i that are multiples of the rotating frequency times the number of stator slots s:

$$f_i = i \cdot s \cdot f_{rot} \tag{4}$$

Though the two cases of static and dynamic eccentricity provide some insight into the excitation of vibration in an ECM drive, it is not possible to generalize from these two academic cases to typical vibration phenomena found in practice. Actual vibration measurements of ECM drives often show peaks at any multiple of the rotation frequency (Fig. 3). Obviously, such a result cannot be explained by a simple superposition of the spectra caused by static and dynamic eccentricity.

Nevertheless, such a spectrum is the effect of a combination of both static and dynamic eccentricity: Depending on the angular position of the rotor, the effects of static and dynamic eccentricity will either partially compensate or amplify each other during a revolution of the rotor. This means that the radial force will become maximal when the rotor reaches a position where the air gap becomes minimal, whereas the radial force will become minimal when the center of the rotor is closest to the center of the stator: In this case, the deviations of the width of the air gap from its nominal width around the circumference reach their minimum, resulting in a minimal radial force.

In addition to the fact that the radial force vector is rotating, its amplitude is oscillating between these two extremes. The magnitude of this oscillation is typically bigger than for the cases of static or dynamic eccentricity alone, as the variation of the magnetic flux is bigger than that caused by the discrete nature of stator slots and rotor poles alone.

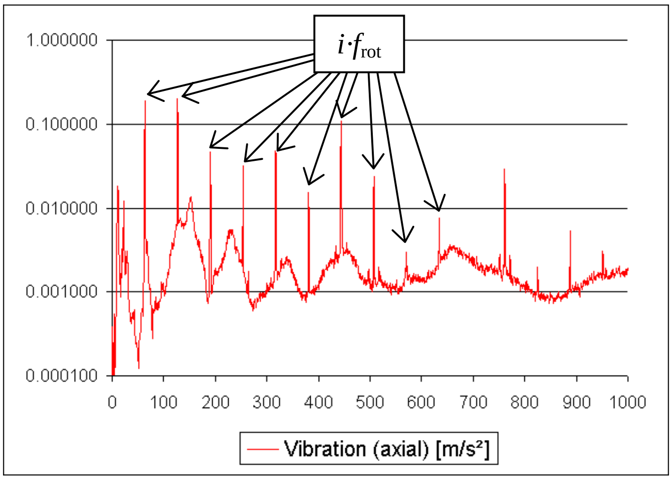

Fig. 3 Vibration spectrum of a ECM drive running at 3800RPM: Vibration is excited by radial force at multiples of the rotation frequency.

The excitation frequencies can be calculated by developing the progression of radial force over revolution angle into a Fourier series: As the fluctuation of the radial force is periodic with the revolution of the rotor, the spectrum may contain any multiple of the rotation frequency (Fig. 3). The amplitude of each frequency depends on the deviations of rotor and stator from their center and the proportion of these deviations to each other.

Furthermore, it would be an overly simplified assumption that the effect of eccentricity can be regarded as a two-dimensional case where the axes are only shifted but still parallel. In reality, the axes of rotor and stator may also be tilted against each other, meaning that the radial force will not only depend on the angular position but also will vary along rotational axis. This is especially important in the case of pumps where hydraulic forces act on the impeller, causing one-sided additional imbalance that is also load-dependent.
The noise excitation can be avoided by achieving optimum concentricity of rotor and stator. Again, defects in electrical components (e.g. winding faults, uneven magnetization of the rotor) may lead to additional noise and vibration, but are not discussed here.

4 Conclusions

The overall noise level of a centrifugal pump is determined by a limited number of tonal components that can be related to underlying periodical processes in the hydraulics and the electrical drive. It is therefore possible, to identify the contribution of each of these processes to the overall noise level by a spectral analysis of pressure pulsation and vibration measurements. Based on the known design parameters of hydraulics and electrical drive, the origin of dominant peaks in the spectrum can be deduced and appropriate counter measures can be taken. Consequently, noise and vibration measurements and according result analysis provide important clues for the continuous reduction of noise emission of pumps.

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