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Characterization of the Vibrations and Structural Noise of a Suction Unit's Cover

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In this paper we report on a study of the vibrations of a suction unit's cover and the relation with the generated structural noise. The existing cover of the suction unit was designed, taking into account only the geometrical requirements. Two different methods were used for characterization of the vibration and structural noise of the cover: numerical modal analysis made with the FEM (Finite Element Method) and experimental modal analysis made during the operation of the suction unit. The results of the experimental modal analysis of the cover's vibrations visually confirmed the results of a numerical modal analysis up to the 1st BPF (Blade Passing Frequency) and characterized the main exciting sources with the prevailing aerodynamic excitation. The vibration modal analysis tries to find a relation with the cover's structural noise using an experimental local noise modal analysis. A different method of measurement of local structural noise is introduced.

Keywords: suction unit, suction unit's cover, vibrations, structural noise, numerical modal analysis

0 INTRODUCTION

Nowadays, every household device tends to be user friendly and is designed not to have a negative effect on human health. However, the vibrations and noise caused by rotational machines are particularly problematic in this respect. The suction units produced by Domel Ltd., Železniki, Slovenia, have a variety of vibration sources that tend to produce noise. In this paper we focus on the cover of a suction unit that was only designed in terms of its fitting together with the other parts of the unit. Consequently, the cover's geometry is matched to the shape of suction unit's housing and blower. As a result there are noticeable vibrations of the cover during the suction unit's operation, and because the cover is the outermost part of the unit we can also expect an increased level of noise generated by the cover's vibrations, generally referred to as the structural noise.

Any analytical analysis of the cover's vibrations is complicated by the complex structure of the cover and the analytically unknown sources of the vibrations. In general, there is not a lot of publicly available analytical research on the structural noise associated with suction units. An experimental-based approach was made to characterize the noise of a suction unit, and it was found that aerodynamic sources prevail [1], despite evidence that the structural noise is mainly caused by the electromotor [2]. The noise caused by the vibration of the cover is probably related to the aerodynamic conditions. [3] The first part of this paper presents two different steps in the analysis of the problem: a numerical and experimental analysis of the vibrations of the suction unit's cover to characterize the main sources of the vibrations, following the basic method steps as in [4]; and, for comparison, an evaluation of the experimental modes with the numerical results.

Far more complicated than the analytically defined vibrations of the cover is the analytical computation of the structural noise generated by the vibrations of the cover. The sound-pressure level (SPL) is the sum of the different sound sources with a major or minor effect. Our aim in the second part of this paper was to compare the modes of the cover's vibrations and the sound fluctuations directly above the cover's surface to find a connection between the modes of the vibration and the modes of the structural sound. To obtain the modes of the sound fluctuations we introduce a method of measuring the fluctuations occurring directly above the surface of the cover.

Because of the complexity of the structural noise at the cover's surface, in this paper we have

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tried to introduce a simple way to understand the vibrations and the generated noise under arbitrary operating conditions.

As we mentioned above, we used two steps to characterise the vibration of the cover. Firstly, a numerical modal analysis was made with the FEM where we used some similar steps as in [5] and [6]; and secondly, an experimental modal analysis was made during the operation of the suction unit. These procedures are independent, but the results are important for comparing and evaluating the modes.

1 FEM MODAL ANALYSIS OF THE SUCTION UNIT

The aim of the FEM modal analysis was to find the modes in the frequency range of the prevailing excitation. We expected that during operation the cover's modes would follow the cover's own modal shapes. However, if this were true we could only predict the behaviour with a numerical analysis. In fact all the cover's vibrations are excited by different mechanisms during the unit's operation, which we will describe in Section 2. In a numerical simulation we cannot consider the detailed excitation mechanisms of the vibrations because we do not know their magnitude and the phase angle between them. We can only predict the places where the excited vibrations are transmitted to the cover.

To make the numerical analysis as accurate as possible we included all the unit's static parts: the housing, the electric-motor's stator, the vaned diffuser and the cover. By including all these parts we extended the time of the computation, but we ensured a better dependency of the cover on the other elements [7]. We achieved this with the correct contact conditions between the parts, because this can also contribute very importantly to obtaining the correct results [8]. Including all the parts also makes it easier to define the boundary conditions. With respect to this the suction unit was free-free supported in order to be compatible with the experimental analysis.

The results of the numerical analysis are shown in Fig. 1 for the mode at an eigen frequency of f=2284 Hz, and in Fig. 2 for the mode at an eigen frequency of f=6533 Hz. These modes are interesting, because they are the nearest modes to the frequencies where the excitations of the cover occur, which will be explained later in Section 2.2.1. Furthermore, we will compare the numerical modes with the experimental modes near these frequencies.

2 EXPERIMENTAL MODAL ANALYSIS OF THE COVER'S VIBRATIONS AND THE LOCAL NOISE

The experimental modal analysis of the cover's vibrations and the local noise had two main goals: firstly, to find the sources of the vibration excitation and the modes at the excitation frequencies

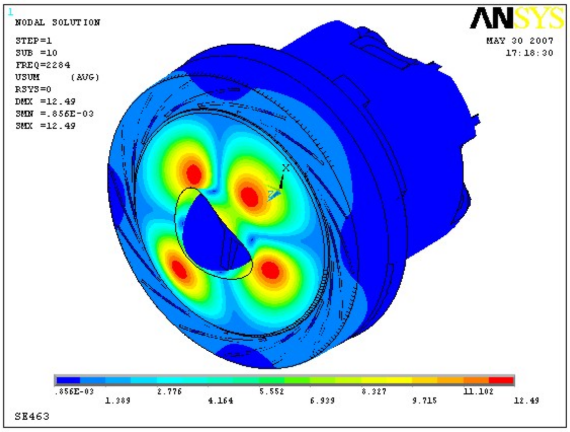

Fig. 1. *Mode at frequency f = 2284 Hz*

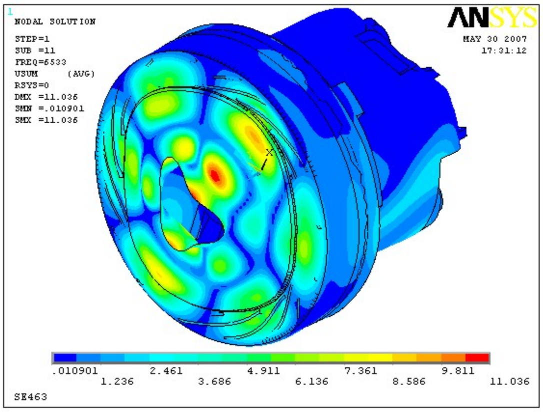

Fig. 2. *Mode at frequency f = 6533 Hz*

in order to validate the numerical results and secondly, to obtain modes of the pressure fluctuations directly above the cover's surface in order to compare them with the vibration's modes

2.1 Method for Measuring the Vibrations and the Local Noise

The measurements of the vibrations and the local noise were made in Domel's laboratory for vibrations and acoustics in an anechoic chamber using the same operating conditions for measurements of the vibrations and the local noise. The performance of the suction units with a vaned diffuser depends on the characteristics of the electric-motor, the blower, the vaned diffuser and the back-flow channels, and the working conditions, like the voltage supply, the orifice, etc. There is the best efficiency for the suction unit at the design point of operation, while at off-design operation efficiency is reduced.

Our experiments were run at a voltage supply of 220V without using the additional orifice to make inlet cross-section smaller, so these measurements differ from the usual one. The rotational speed f_R under the described conditions was 43200 min^{-1}. We must be aware that this is not the design point of operation, but we assumed the arbitrary operating conditions. This means that we are interested only in the source of the vibrations and their influence on the structural noise; we are not interested in their magnitude under different operating conditions.

We used the following equipment to measure the vibrations: a laser vibrometer (Polytec CLV 700), a laser amplifier (Polytec CLV 1000), an accelerometer (B&K 4517-002), a signal analyzer (B&K PULSE Type 3560-C) and a software environment (B&K Labshop 9.0).

Fig. 3 presents the environment in which the vibrations were measured. The suction unit lies on a massive metal block with foam to simulate the free support; this is also important during the numerical analysis in the previous section. If we want to make an experimental modal analysis of the vibrations and later compare it with the sound analysis we must get the right relations between the magnitude and the phase angles for the different measurement points. We can do this by introducing a frequency-response function (FRF). However, we must first choose a reference point on the upper surface of the cover. We can then calculate the FRF between the measured point and the reference point [9]. An accelerometer was used to measure the acceleration of the reference point, and a laser was used to measure the velocity of the measured point. With the differentiation of the laser's velocity signal to get the acceleration of the measured point, we could finally calculate FRF using Labshop software.

Another important step is the selection of the measured points. The resolvability of the cover's modes depends on the frequency range. From the FEM modal analysis we can see that the mode in Fig. 1 has a simple shape, but as the eigen frequency increases the mode is no longer simple (see Fig. 2).

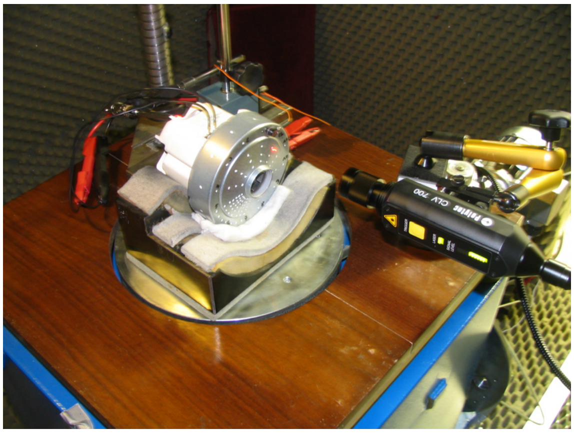

Fig. 3. *Measurement of the vibrations*

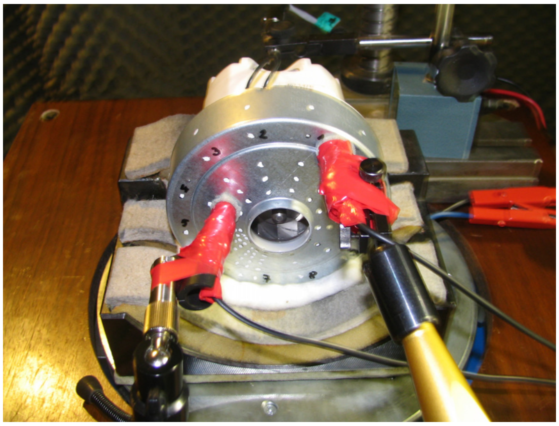

Fig. 4. *Measurement of noise directly at the cover's surface*

Regarding the numerical results in Section 1, we configured the coarse mesh (Fig. 5 and 6) with 5 measured points in each radial direction (4 were located in axial direction and 1 in radial direction of the suction unit), with an equidistant partition of 12 parts in a circular direction around the cover's circumference.

With this kind of mesh we can cover the modal lines only at first few eigen frequencies. To satisfy the modes at higher eigen frequencies we had to configure a denser mesh (Fig. 5 and 7) with 10 points in the each radial direction (7 were located in axial direction and 3 in radial direction of the suction unit) and 6 parts in the circular direction, with an equidistant angle θ_2 = 10° within the range Φ=130–180° from the starting angle (see Fig. 5).

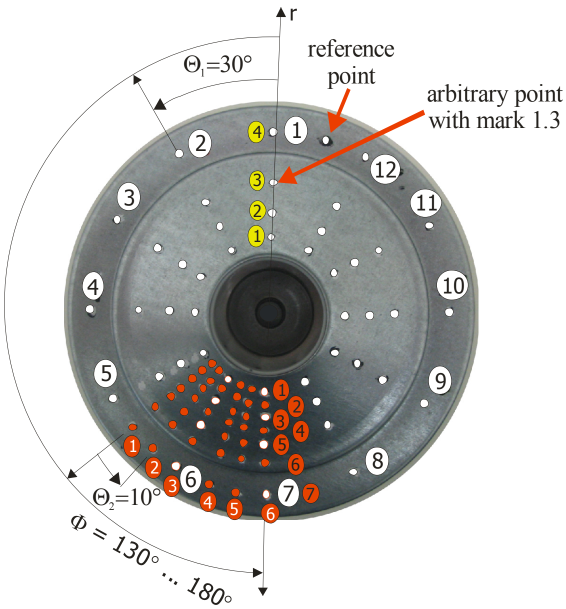

Fig. 5. *Scheme of measured points on the cover's upper surface*

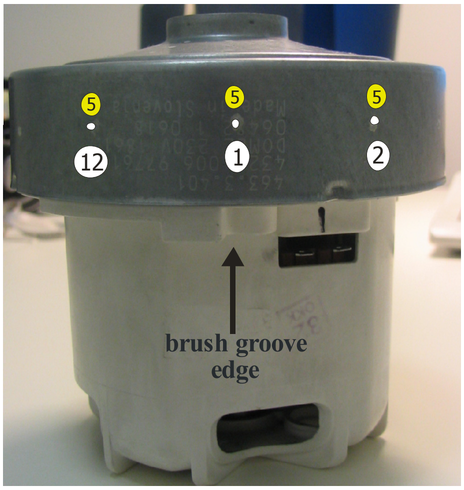

Fig. 6. *Scheme of measured points on the cover's side surface*

Fig. 7. *Scheme of measured points on the cover's side surface (denser mesh)*

With a denser mesh we were able to improve the resolution of the modes.

For the noise measurement on a local area we used two condenser microphones with an amplifier (EMKAY MD4530ASZ-1 with constant sensitivity to 10 kHz), a signal analyzer and the Labshop software.

Fig. 4 presents a noise measurement of the environment directly above the cover's surface in the distance of 2-3 mm. The small size of the microphones (Φ4.5x3 mm) means that they can be positioned above the same points that we defined during the vibration measurement; this is the reason for using our idea about comparing the vibration and the sound modes.

However, in the near sound field the measurement could be uncertain [10]. The noise directly above the cover's surface can be a combination of different sources. In general, however, we can suppose that it is a combination of the noise transmitted through the cover's thin wall and the noise generated by the thin wall's vibrations. The response of the linear dynamic model is expected to have the same frequencies as the excitation. We expect the same to happen with the sound field near the cover's surface on a small area, which the measured point certainly is. Despite the fact that we are in a near sound field we can expect pressure fluctuations. To minimize, in particular, the effect from a nearby point we isolate all around the microphone, which is situated a few millimetres above the cover's surface, with highly absorbent foam (the type used in anechoic chambers) in order to obtain the prevailing pressure fluctuation in the normal direction (see Figure 4). We use two isolated microphones at the same time: one for the reference point, which is the same as at the vibration measurements; and one for the measured point.

The suction unit was supported in the anechoic chamber in the same position as during the vibration measurement, and during both measurements we did not change any parameter of the unit's operation to ensure the same conditions for every measurement point. The signal during the noise measurement was processed in the same way as for the vibration measurement. The goal was to calculate the FRF between every measured and reference point.

Finally, we used numerous FRFs of the vibration and the sound-pressure field for presenting modes, using the *LMS* software.

2.2 Results of the Experimental Modal Analysis

2.2.1 Characterization of the Vibration

We can determine the main sources of vibrations from an amplitude auto-spectrum. We will show the representative vibration amplitude auto-spectrum (Fig. 8) on the cover surface with the arbitrary chosen point 1.3 (marked in Fig. 5). The spectrum clearly shows the discrete sources of the vibrations at characteristic frequencies. The position of the discrete frequencies in the spectrum depends on the operating conditions described and explained in the first paragraph of Subsection 2.1. On the basis of this we are interested in where and why these sources appear.

The first interesting frequency range extends up to 1 kHz, where the rotational frequency f_R of the existing suction units can be found. A distinct frequency can be found exactly at the rotational frequency f_R. The forced vibrations are caused by the unbalanced rotor, and they are transmitted firstly to the cover over the housing and the vaned diffuser through the bearings, and secondly at the blower's inlet edge at the contact with the cover's washer. The magnitude of the excitation also depends on the quality and the wearing out of the bearings, the contacts, etc. The forced excitation at the rotational frequency can be described as a mechanical excitation.

Other discrete frequencies of interest are in range above 1 kHz. We can see in the amplitude auto-spectrum in Fig. 8 that we have three significant peaks, equidistant with respect to the rotational frequency f_R up to 20 kHz. This is a consequence of the blade passing frequency (BPF) effect that happens when the rotor blade passes the diffuser vane [3]. A pass results in air-pressure fluctuations. The discrete frequencies f_i can be calculated as:

$$f_i = n \cdot z \cdot f_R \qquad n = 1, 2, ... \qquad (1)$$

In one turn of the rotor this effect repeats z times, where z represents the number of rotor blades. There is an influence of first few higher BPF harmonics; in Fig. 8, for example, we can see the BPF effect at frequencies of 6480, 12960 and 19440 Hz for a measured unit that has 9 rotor blades. The BPF effect is an aerodynamic excitation of the cover's vibration. The pressure fluctuations are transmitted directly to the thin wall of the cover, and these are the prevailing excitations.

We can also describe two excitation mechanisms that are similar to the BPF effect for a determination of the discrete frequencies.

The first excitation has an aerodynamic origin when the air flow exits the diffuser through back-flow channels in the space of the rotated rotor. The excited vibration depends on the number of vanes of the back-flow channels, which in our case is 12. Also using Eq. (1), we can explain the 12th harmonic peak at 8640 Hz.

The second excitation has its source in the contact between the electric-motor's brushes and the collector's lamellas. This typical mechanical excitation, described in [11], is transmitted to the cover over the vaned diffuser. Using Eq. (1) and considering the collector's 22 lamellas we get the 22nd harmonic peak at 15,840 Hz. Both these excitation effects have a significant first harmonic, while the higher harmonics are not significant or are above 20 kHz.

Considering the defined vibration sources on the cover's surface we can focus on the aerodynamic excitation, especially the BPF effect, which radiates a pressure fluctuation directly onto the cover's inner surface. The magnitudes of the aerodynamic sources depend on the vicinity of the design point of operation. Because the experiment was run at off-design point of operation, the BPF effect excitation was more emphatic.

2.2.2 Comparison and Findings of the FEM and Experimental Modes of the Cover's Vibrations

The final step in the vibration's signal processing was to define the FRFs between the measured points and the reference point to obtain the experimental modes at the characteristic frequencies described in Section 2.2.1 so as to compare them with those that were numerically defined in Section 1. When we make comparison, we have to assume, that the experimental vibration mode at the selected frequency will follow to the numerical vibration mode at the frequency, which is the nearest to the selected frequency.

We researched the frequency range up to 10 kHz. The reasons for this action are, in particular, the unreliability of measurements

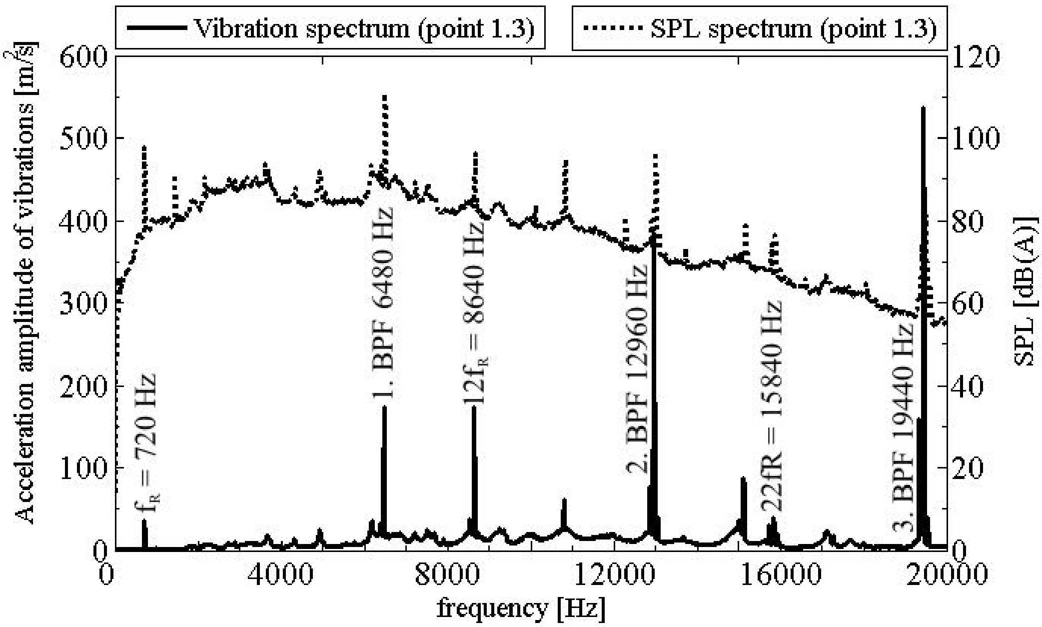

Fig. 8. *Vibration and sound-pressure level spectrum for an arbitrary chosen point 1.3*

above 10 kHz and a lot of distinctive or indistinctive modes close to each other at higher frequencies. We should, in fact, measure a very dense mesh on the cover's entire surface, but this is a very time-consuming task, and why it is better to make a comparison at the first few excitation frequencies.

The first comparison of the modes was made at the rotational frequency f_R=720 Hz. The experimental mode is presented in Fig. 9. We can compare this with the nearest numerical mode, which is surprisingly at frequency 2284 Hz, presented in Fig. 1. The result is visual accordance.

The modes become more complex with frequencies higher than the rotational frequency f_R. One part of the experimental mode is represented in Fig. 10 for the BPF (at 6480 Hz). Its orientation can be seen from Figs. 5 and 7. The experimental mode in Fig. 10 should be compared with the numerical mode at the nearest frequency 6533 Hz, shown in Fig. 2. There is accordance in the mode, especially on the cover's upper surface, where we can find one peak and one trough in the radial direction on 1/6 of the cover's circumference.

The FEM analysis was satisfactory and useful, especially in the lower frequency range at the first few eigen frequencies for determining the vibration modes, because they are sufficiently distinct. We can, however, encounter problems even when we are around BPF, which should be as low as possible. As mentioned in Section 1, we cannot consider the excitation mechanisms in the FEM analysis. In the higher frequency range the experimental modes are supposed to follow to the shapes of the nearest numerical modes. It is hard to determine experimental mode considering numerous numerical modes. On the basis of this presumption we can apply the FEM analysis in the visual predictions of new products.

2.2.3 Analysis and Findings of the Experimental Modes of the Cover's Vibrations and the Noise at the Cover's Surface

With the described measurement procedure in Subsection 2.1 we get the SPL spectrum and the FRFs for the sound-pressure field at the cover's surface.

A SPL spectrum for the arbitrary point 1.3 is presented in Fig. 8. It is obvious that the SPL spectrum has discrete tones at the same frequencies as the vibration spectrum. This is logical because the origins of the noise and the cover's vibration are related to each other. From the SPL spectrum we can see that the airborne sound prevails. The most important part of the total SPL is contributed especially by the BPF, at 6484 Hz. We cannot also ignore the contribution of the 12th harmonic of the rotational frequency f_R, because of the air outlet from the back-flow channels. The contribution of the mechanical noise is visible mostly at the rotational frequency f_R. From the SPL spectrum we cannot determine the share of the structural noise, even if there is some relation to the cover's structural noise.

The next step is the comparison between the cover's experimental vibration modes and the sound-pressure modes at the cover's surface. Modes at Fig. 9 and 10 were defined with calculation of FRF of all measured points. The research is limited to the lower frequency range, up to 10 kHz.

Fig. 9 presents a comparison of the modes around the rotational frequency f_R=720 Hz. We can see that there is no accordance between these modes. The sound-pressure mode has no expressive shape, despite the first characteristic frequency. We have two explanations for: firstly, there is no sufficiently high magnitude of vibration at this frequency; secondly, the mechanical excitation of the sound pressure by rotor at rotational frequency f_R=720 Hz, transferred to the cover through the housing, vaned diffuser and partially blower, happens under critical frequency, if we suppose theory about sound pressure fluctuation at infinite plate caused by mechanical excitation of this plate [10]. Critical frequency depends on sound velocity in air, bending wave velocity in plate and on plate's thickness.

There is a different situation in Fig. 10, which compares the modes at BPF at 6480 Hz. The modes are the most visually similar at this frequency, with only a little phase difference. We can suppose that there is an increased level of structural noise. The cover's vibrations at this frequency possibly cause intensive sound-pressure fluctuations. The explanation can be

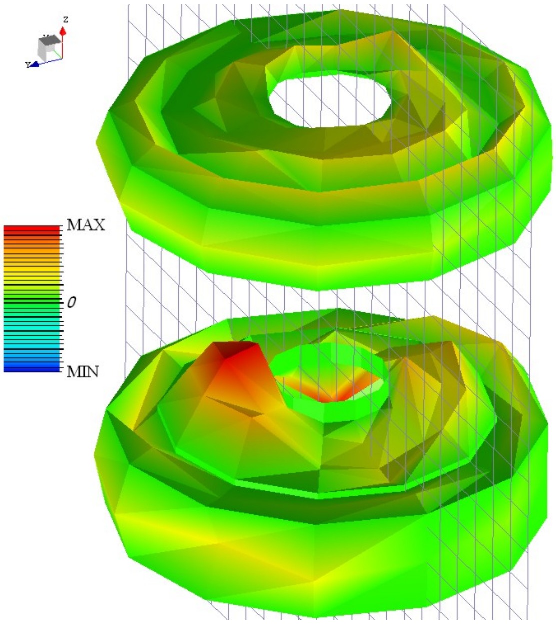

Fig. 9. *Vibration mode (below) and sound-pressure mode (above) at the rotational frequency f_R=715 Hz*

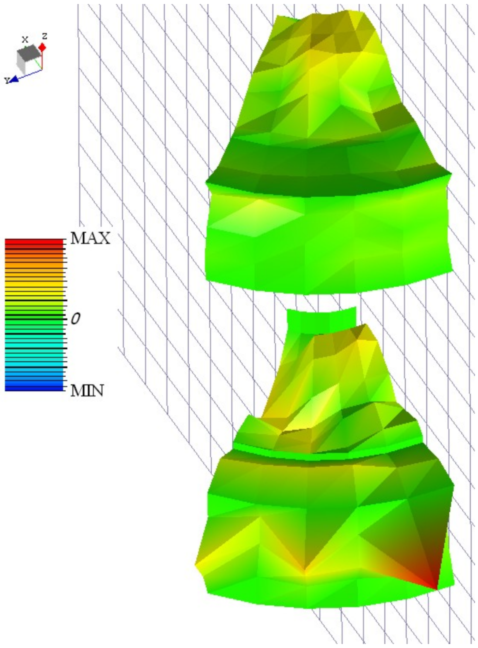

Fig. 10. *Vibration mode (below) and sound-pressure mode (above) at the BPF frequency f=6503 Hz, partial view*

found in a statement that is the opposite of that in the previous paragraph: firstly, the magnitude of the vibrations at the BPF could be high enough to excite the sound-pressure fluctuations; and, secondly, the excitation frequency in the cover's wall is above the limit frequency, which excites the sound pressure. We cannot also neglect the fact that the sound-pressure fluctuation at the upper surface of the cover could be a consequence of the transferred sound fluctuations through the cover's wall [10].

The increased influence of the BPF effect at its first frequency proved to be the most important source of the suction unit's noise, with a major contribution to the cover's vibrations and the cover's structural noise in the frequency range up to 10 kHz. In general, we have to combine it with the aerodynamic origin of the sound, whose side product is also the structural sound. We cannot find a relation between the mechanically excited vibrations and the structural noise, or this effect is untraceable using the described measurement method.

3 CONCLUSION

This paper avoids complicated and uncertain analytical expressions for determining the vibrations excited in a suction unit's cover. With an experimental modal analysis of the cover's vibrations under free delivery operating conditions we found the main sources of vibrations that have an aerodynamic or mechanical origin, where the aerodynamic BPF effect is the prevailing excitation. The BPF effect depends on the operating conditions and the geometry of the static and rotating parts of the suction unit. The smallest effect is at design point of operation. The design of the suction unit should tend to reduce the BPF effect.

A numerical FEM analysis is a convenient approach for studying the cover's vibrations. It is useful for both the cover's conception and its design. The numerical results were verified by an experimental modal analysis with a trusted frequency range up to 10 kHz. However, the vibration modes become more complex as we go higher in the frequency range. The cover responds with the sum of the nearest modes, and this sum again depends on the mode's complexity.

An exact relation between the cover's vibrations and the structural noise is hard to differentiate during the operation of the suction unit. With a comparison of the vibration's modes and the sound-pressure fluctuation's modes above the cover we found accordance between the modes at the BPF (6480 Hz). The numerical and experimental results around this frequency and its mode can help us prepare a new cover shape, according to the main vibration and the noise excitations. The cover would better resist the vibration excitation with an increased bending stiffness. The parametrical cover model is suggested for a later parametrical optimization.

4 ACKNOWLEDGEMENTS

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