The problem of mechanical and hydraulic noise in vane pumps has been presented. A separation of vanes from a cam ring turned out to be responsible for the mechanical noise. This phenomenon has been observed by means of an ultra-speed camera. Suggestions how to lower this kind of noise have been given. In turn, pressure variation both in displacing chambers and discharge side of a pump is regarded to be a main reason of hydraulic noise. The methods of pressure pulsation measurement and noise emission due to pressure pulsation have also been discussed.

1. INTRODUCTION

Apart from the performance, it is a noise level of fluid power systems that is nowadays an important criterion taken into account while accepting a fluid power system for a certain application. Thus, much of the research work done at present is focused on reducing the noise of fluid power systems to adopt them to existing regulations.

The reduction of noise level may be carried out with the help of primary means influencing the source of the noise or with the help of secondary means which restrict the noise propagation due to either sound insulation or noise attenuation.

The primary means may include both the design modifications (dimensions, material etc.) and functional changes while the noise attenuation (secondary means) may be done by means of active and non-active noise suppressors \([1, 2, 3, 4]\). In both types of suppressors the principle of superimposing of two waves in order to suppress the noise is applied.
However, to reduce effectively the noise level one must get familiar with noise sources and the noise propagation. In Fig. 1 different ways of air-borne noise generation are presented.

Fig. 1. Generation of air-borne noise

Rys. 1. Mechanizmy powstawania dźwięku powietrznego

2. MECHANICAL AND HYDRAULIC NOISE SOURCES IN VANE PUMPS

2.1. Mechanical noise sources

2.1.1. State of past studies

The phenomenon of vane separation from a cam ring was known, up to now, in pressure balanced vane pumps in which the vane separation was caused by the reverse of vane motion due to cam ring contour shape. In unbalanced pumps no separation of vane was expected since no acceleration forces towards the rotor center occurred due to continuous vane motion. In spite of this, however, the results of Widmann’s studies [5] suggest this phenomenon occurs in unbalanced pumps as well.

2.1.2. Aim of investigations

In the course of his work, carried out in the Institute of Machine Tools, University of Stuttgart, Widmann measured the level of material- and fluid-born noise with different pressure distributions. To determine the optimum transitory pressure distribution the pressure in the chamber was measured by means of a rotating pressure gauge.
Thus, it has been proved that at certain operating conditions, when the noise emitted is higher, the transitory pressure occurs too early or its value is too low, as it is shown in Fig. 2.

![Diagram showing pressure variation](image)

Fig. 2. Pressure variation in displacing chamber

Rys. 2. Przebieg ciśnienia w komorze wyporowej

Widmann regarded this phenomenon to occur due to the reverse flow between the chambers via increased clearances. It has also been stated, without more detailed proofs, that the vane is not always in contact with the cam ring contour. Accepting these results as a starting point further investigations of vane pumps have been performed.

2.1.3. Photo-optical control of vane motion

The measurement method applied by Widmann is not adequate to describe the phenomenon of vane separation in details. Thus, in further investigations the photo-optical method has been employed.

The covers of unbalanced pumps have been replaced by a specially designed elements made of acril glass and the phenomena occurring in pumps during their operation have been filmed by means of ultra-speed camera. The test stand is shown in Fig. 3. The method enabled to observe the separation of vanes in a double-vane unbalanced pump [7, 8] (see Fig. 4). As it may be seen in Fig. 4, one of two vanes remains in the slot instead of being kept tightly against the cam ring.

As far as a double-vane pump is concerned the vane separation occurring in the transitory zone between inlet and outlet ports not only has no influence on the pump noise but sometimes brings some profits, namely, the initial compression pressure is limited to a value slightly higher than operating pressure.
As far as a single-vane pump is concerned, the separation of vanes is observed in the same zone. At a pressure exceeding the operating pressure by 50% there appears the radial force towards the center of the rotor depending on the geometry of the vane. For both types of pumps such a phenomenon accompanies the cold oil and high pump delivery and causes no mechanical noise. The pressure operation beneath the vanes causes a kind of damping effect, so that no material-borne noise can be detected. Similarly, during the reverse motion the meeting of the vane with the cam ring at the end of the outlet port makes no noise.

Another phenomenon observed in both types of pumps is essential as far as mechanical noise sources are concerned. In this case the movement of the vane out of the rotor is restricted to such an extent that there exists a clearance between the vane and the cam ring contour. In the transitory zone, the vane is accelerated out of the rotor due to pressure acting beneath the vane. The measurements of distances and photographing speed (being constantly equal to 3500 pictures per second) showed that the velocity of vane amounted to 2+3 m/s. In spite of small overall vane dimensions this phenomenon has an essential effect on noise generation.
2.1.4. Reasons of vane separation

The aim of the study was to evaluate in detail reasons of vane separation in transitory and discharge zones as well as in inlet zone, where the vane separation may not be explained by overpressure as it was in the transitory and outlet zones. The first results obtained suggested further investigations to be focused on the vacuum beneath the vanes. There exists in vane pumps the so called "inner pumping effect" as a result of the fact that beneath-vane spaces operate as displacing chambers (Fig. 5). In the inlet zone the vane is moved out of the rotor due to a centrifugal force alone. If the vacuum beneath the vane exceeds the one in the inlet port the vane is subjected to an additional differential-pressure force pointing the rotor center. If that force together with friction forces exceed the centrifugal force, the vane will be stopped.
The optical measurement method may not be employed this time since the pump elements made of acril glass would be deformed differently changing thus the pump inner clearances and pressure distribution. Moreover, using the ultra-speed camera is money-and time-consuming method. Therefore the electric displacement gauges had been designed and mounted inside the rotor (Fig. 6). Rotating electronic system transmits the analog voltage of the range 1+5 V, proportional to the distance between the lower part of the vane and the vane slot bottom in the rotor, to the measuring equipment. To avoid the disturbances during the transmission the analog signal is first converted into the digital one.

2.1.5. Methods of mechanical noise reduction

As it was mentioned above, vane separation in the inlet zone causes noise emission. So, there are two possibilities of noise reduction. The first one is aimed at avoiding the difference between the beneath-vane-space pressure and the displacing chamber pressure in the inlet zone. To achieve this, cross sections of channels connecting the beneath-vane-space with the displacing chamber should be shaped in such a way as to avoid the turbulent oil flow in the entire inlet zone. The second possibility is to keep the vane in a continuous contact with the cam ring in the inlet zone by allowing the pressure to operate beneath the vane.
The results of measurements with the pressure operating beneath the vanes in the inlet zone show there is the chance of noise level reduction more than by 6 dB (Fig. 7).

![Graph showing acoustic one-third octave band spectrum](Fig.7. Acoustic one-third octave band spectrum)

2.2. Hydraulic noise sources

The main reasons of hydraulic noise are connected with the variation of pressure and flow rate caused mainly by:
- cavitation,
- pressure rise and drop in displacing chamber,
- pressure variation at the discharge side of the pump.

2.2.1. Pressure rise and drop as a noise reason

The pressure in displacing chamber ought to be adjusted to the operating conditions. Otherwise, the pressure equalizing process due to connecting a displacing chamber with a discharge port has the impact character. Different modes of pressure equalizing characteristics are shown in Fig. 8. A proper adjustment of pressure should lead to a lower pressure gradient and thus to the optimum transient process. Such a manual setting, however, works only at given operating conditions (pressure, flow rate).

2.2.2. Discharge pressure pulsation

Another source of noise results from the pump kinematics. Due to the finite number of displacing elements the pulsation of the outlet flow and outlet pressure occurs. The discharge pressure pulsation depends directly on the pressure variation in a displacing chamber (with the influence of
3. PRESSURE PULSATION MEASUREMENTS

There are three methods known to measure the pressure pulsation with the influence of the pipeline system being minimized, namely (Fig.9):
- measurement with the short discharge pipeline;
- measurement with high pipeline impedance according to BS 6335 [10];
- measurement with reflection-free end of the pipeline.

As far as the short pipeline is concerned, the distance between the pumps and the pressure valve is so short that the resonance occurs only at frequencies exceeding 3 Hz. The amplitude of the high frequency pressure pulsation is, in general, so small that no noise emission is observed.

The pulsation level measured is high and the pipeline is highly sensitive to geometry changes. As a matter of fact, the \( V_0 \) volume of oil between the pump and the closing valve is very small. Due to it the hydraulic capacitance of the system \( C_H \) which may be expressed as follows

\[
 C_H = \frac{V_0}{K_S}
\]

is also very small. Thus, the pipeline impedance is high, according to the formula

\[
 Z_H = R_H + i \left( L_H + \frac{1}{C_H} \right)
\]

and such is the pulsating pressure \( P_w \)

\[
 P_w = Q_w Z_H
\]

where
- \( Q_w \) - flow rate variation,
- \( R_H \) - hydraulic resistance,
- \( L_H \) - hydraulic inductance,
- \( K_S \) - bulk modulus (for the oil: \( K_S = 1.6 \cdot 10^4 \) bar).
To reduce the influence of the oil volume in the high pipeline impedance method [11, 12] the pipeline had been lengthened. Just behind the pump outlet connection the sudden change of impedance caused by a pipeline of a very small diameter was placed. It turned out to be necessary to exclude any resonance at the frequencies up to 3 kHz. Thus, the pipeline had a very high input impedance. It is followed, as with short pipeline, by a high noise level. So, also in this case the oil volume affects the results of measurements. This method may be regarded as a short-pipeline-with-stable-restriction method.

Employing the reflection-free pipeline end turned out to be very effective (Fig.10) [13]. It consists of a large volume low impedance pipeline with the flow restrictor. By adjusting the pipeline end impedance (by means of the flow restrictor) to the pipeline impedance the reflection at the end of the pipeline may be avoided. It means, however, that the pressure drop on the restrictor may be dependent only on the volume flow rate. Thus, the application of the method is restricted to high flow rates. In this case, however, the pressure drop through the restrictor due to pipeline end impedance setting may be higher than the desired operating pressure. To avoid such a possibility bigger pipe diameters ought to be employed. It will lead to lower pipe impedance and the pipeline end impedance will also be kept at a lower level.
The aim of the study was to determine the influence of the pump discharge pressure pulsation on the noise emission of the entire system. The pressure pulsation has been measured for several pumps of the same type (balanced vane pumps of 10 cm³/rev output volume). Some of the pumps had been previously defined subjectively as "noisy" ones, some of them - as "silent" ones. It has been proved that there are two characteristic features that differentiate the pulsation spectra of the "noisy" and "silent" pumps. Namely, either the main peaks of the "noisy" pumps spectra are much higher or there occur also some additional peaks the amplitudes of which being comparable with the main ones. The spectra of "noisy" and "silent" pumps are presented in Fig.11. The main peaks dominating in the "silent" pump spectrum are regarded to be caused by displacing kinematics or to be their harmonics. The additional peaks are regarded to be the peaks of the frequencies equal to the pump shaft rotational frequency or their harmonics. If their amplitudes are as high as the main ones are, both types of peaks may be used to determine the mode of air-borne noise emitted. The main peaks distribution is responsible for the tonal noise while the additional peaks distribution - for the stochastic one.

The additional high-amplitude peaks distributions have not been investigated yet. Probably, they are caused by the wrong operation of a displacing chamber during the discharging process, e.g. undesired sepa-
rational of vanes. This phenomenon is strongly influenced by the pump assembling process, especially torque moments applied while screwing a pump cover. Additional peaks vanish, for instance, after unscrewing and screwing again a pump cover.

Taking into account both the main and the additional peaks distribution one may distinguish between "noisy" and "silent" pumps on the basis of their spectra. For the pumps under consideration upper limits of four criteria values have been determined, namely: the first three peak value levels of the main distribution and the sum of additional peaks situated between these main peaks. If any of the criterion values for a
given pump exceeds the upper limit value, the pump is regarded to be the "noisy" one.

Such a procedure refers just to the pumps under consideration with given pipelines. For other pumps the criteria and their upper limits should be defined separately.

The research work carried out in the Institute of Machine Tools, University of Stuttgart was aimed at transferring of the above method to other hydraulic systems. Up to 15 vane pumps with the output volume 50 cm³/rev have been tested. Apart from the output pressure pulsation, the sound power levels for each pump have also been determined according to envelope surface method (DIN 45635) and compared to each other.

6. RECAPITULATION

There are two methods of reducing the sound emission in hydraulic systems, namely, either the generation of the noise in its source is restricted or the noise emission and propagation is suppressed. The noise of hydraulic systems may be caused by mechanical or hydraulic reasons. The study on the mechanical source of noise in vane pumps (vane separation) and measurement techniques enabling to observe and measure the vane movement as well as means to neutralize phenomena under consideration have been presented in the paper. The propagation of a hydraulic pressure wave in a pipeline system has also been investigated.

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MECHANICZNE I HYDRAULICZNE PRZYCZYNY HALASU
W POMPACH ŁOΠATKOWYCH

W pracy przedstawiono problem mechanicznych i hydraulicznych źródeł hałasu w pompach łopatkowych. Głównym źródłem hałasu mechanicznego jest oddzielenie się łopatki od bieźni. Zjawisko to było obserwowane przy pomocy ultrasybkiej kamery. Podano wskazówki dotyczące redukcji hałasu mechanicznego. Za główne źródło hałasu hydraulicznego w omawianych pompach uznano pulsacje ciśnienia w komorach wyporowych i po stronie tłocznjej pompy. Omówiono metody pomiaru pulsacji ciśnienia oraz emisji hałasu tym wywołaną.

MECHANICZNE I HIDRAULICZNE PRZYCZYNY HUMA W LOPATKOWYCH HACOSACH

W pracy przedstawiono problem mechanicznych i hydraulicznych źródeł hałasu w pompach łopatkowych. Głównym źródłem hałasu mechanicznego jest oddzielenie się łopatki od bieźni. Zjawisko to było obserwowane przy pomocy ultrasybkiej kamery. Podano wskazówki dotyczące redukcji hałasu mechanicznego. Za główne źródło hałasu hydraulicznego w omawianych pompach uznano pulsacje ciśnienia w komorach wyporowych i po stronie tłocznjej pompy. Omówiono metody pomiaru pulsacji ciśnienia oraz emisji hałasu tym wywołaną.

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