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A TEST BENCH ENHANCEMENT FOR PUMP VIBROACOUSTIC RESEARCH

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The pump vibroacoustic studies have strict requirements to background noise. With the exception of the pump, noise and vibration emitted from a test bench makes up the background noise. As a rule, to meet these requirements, the pump should be located in the anechoic chamber, apart from the test rig. The pump should be separated from other noise sources by the reflected plane with holes for inlet and outlet pipes. This article describes an enhancement process of the stand for pumps vibroacoustic research in the case of anechoic chamber lack or impossibility to isolate the pump from test bench by the wall. Recommendation and ISO for vibroacoustic parameters registration were analyzed. The primary sources were determined. It is motor, pipes, electronic rpm adjustment system, mounting frame. The measures for background noise reduction were designed and implemented. The improved test bench can carry out vibroacoustic measurements of the micropump in order to determine how changing the rotor material can affect emitted noise and vibration.

Keywords: test bench, pump, vibroacoustics; background noise;

1. Introduction

There are trends to miniaturization of test benches in the field of hydraulics [1,2], increase in the number of parameters registered by them [3,4,5] and provision of experimental impacts applied in a fairly wide range of parameters, often broader than those provided with certified expensive test benches.

A test bench for studying vibro-acoustic performances of a gear micropump has to meet the following requirements: sufficient precision of its units and parts, compact size, possibility of conducting vibro-acoustic measurements with the help of different devices and sensors.

Most test benches are applicable for standard size pumps [6]. The aim of this work is to improve the test bench which can provide vibroacoustic measurements of micropumps.

An attempt to create the test bench for a compact pumping units is described in the article [7] (Fig. 1). This study is aimed at reducing background noise of the test bench developed earlier [8,9] and to obtain more accurate vibroacoustic parameters of the micropump when the rotor material is changed.

Figure 1: Physical configuration of the micropump made of different materials.

2. The purpose of the test bench

The complex test bench for vibroacoustic tests of compact pumps (up to 1 l/min) is designed to control noise characteristics [10], noise intensity [11,12], vibration of the micropump [13] and its resource [14].

3. Components of the test bench

Design scheme of the test bench is similar to the one developed earlier [7]. Hydraulic system layout for the proposed test bench is shown in Figure 2.

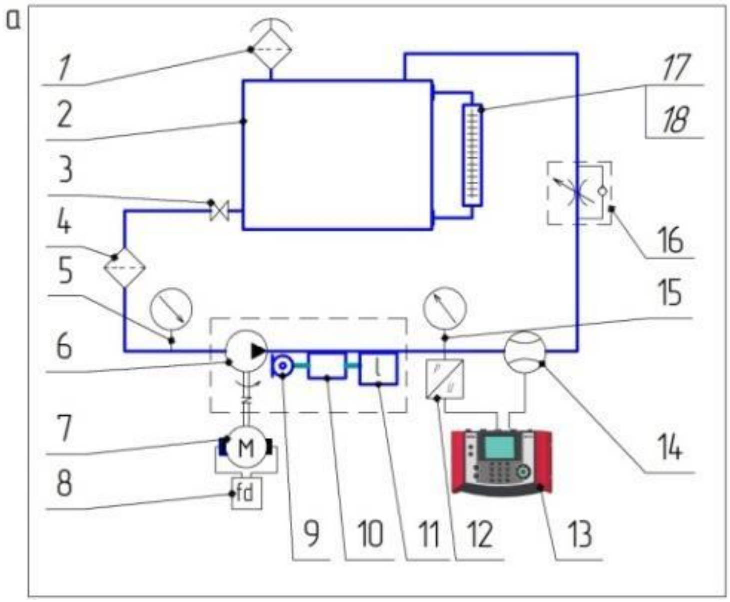

Figure 2: Scheme of the test bench: 1 – air filter, 2 – tank, 3 – valve, 4 – preliminary filter, 5 – vacuum gauge, 6 – gear micropump, 7 – motor, 8 – frequency drive, 9 – microphone, 10 – measuring system, 11 – laptop, 12 – pressure transducer, 13 – portable registration module, 14 – flowmeter, 15 – pressure gauge,16 – throttle, 17 – fluid level gauge, 18 – temperature gauge.

The test bench consists of a base plate and mounting frames, a pump, a motor, a frequency drive, a sound-proof housing, a tank, piping, a portable registration module, a load device (throttle), a flowmeter and filters.

4. Measures for background noise reduction

The accuracy of the measured vibroacoustic parameters of developed compact pumps is associated with noise and vibration from additional units, which create additional interference.

To improve the accuracy of measurement results, it is necessary to apply complex methods of noise, vibration isolation and noise, vibration absorption to the test bench.

It should be noted that the pump noise is the noise of the pump without the motor, coupling and piping.

Methods for noise and vibration reduction during their propagation from the source to the protected object (pump) were applied in this research (Fig. 3) according to Russian State Standard 26568-85 [15] and 12.1.029-80 [16].

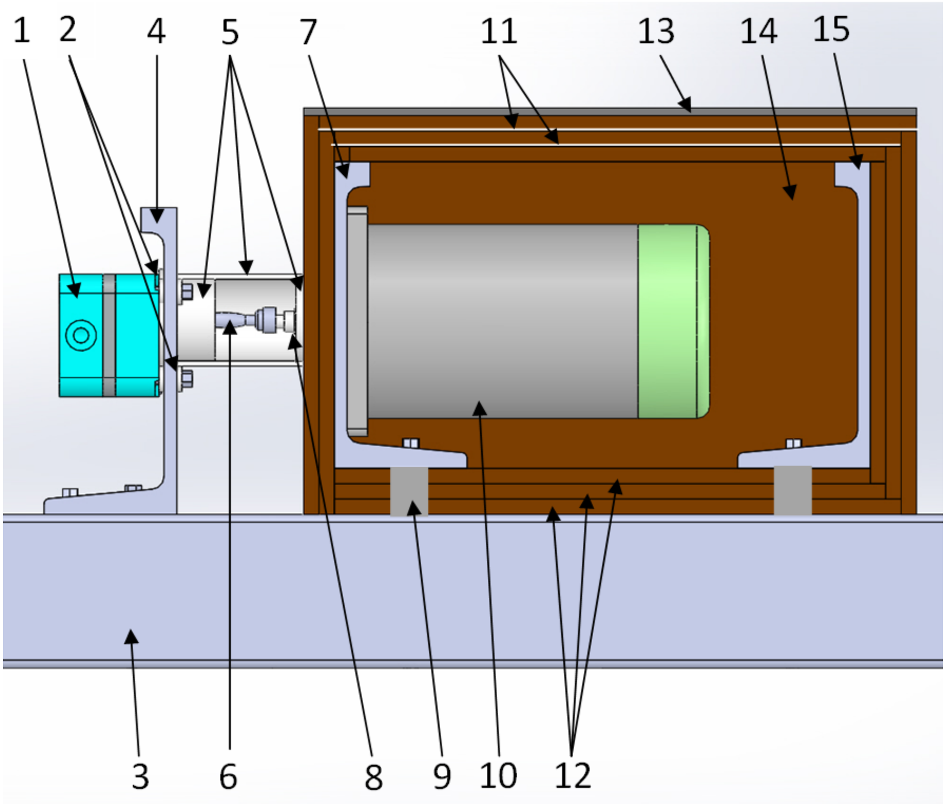

Figure 3: The sketch of improved test bench: 1 – micropump, 2 – fluoroplastic washers, 3 – plate, 4 – frame, 5 – sound-proof shell, 6 – elastic coupling, 7 – front frame, 8 – adapter, 9 – steel sleeve, 10 – motor, 11 – duralumin plate, 12 – felt floor, 13 – housing covering, 14 – felt side wall, 15 – free frame.

The applicable methods to this test bench were divided into the following groups:

- Sound insulation (reference numbers 3,5,7,11,13,15 on Fig. 3).
- Sound absorption (reference numbers 12,14 on Fig. 3).
- Vibration isolation (reference numbers 2,8 on Fig. 3).
- Vibration absorption (reference numbers 2,3,6,8 on Fig. 3).

The test bench plate is a measure for vibration dampening and is chosen because of following criteria:

- The plate material must have a large energy absorption coefficient (Ψ=0.23).
- The mass of the plate should exceed the mass of the examined unit by about 10 times [17].

The most vibrating and noisiest element of the test bench is the motor. The weight of the plate was selected in accordance with the total mass of the motor and frames (2.6 kg), so the mass of the vibration absorbing plate is 33 kg. The material used for the plate is cast iron due to its high energy absorption coefficient.

According to [18], a rigid vibration damper is effective at low frequencies. Therefore, the expected efficiency of the plate vibration absorption is also expected in the low-frequency range.

The path of sound emission through the housing is shown in Figure 4: through the enclosing structure (A), through the openings (B1, B2), the emission from the housing of sound penetrating into its walls with a rigid support on the base frame, and also in contact with the machine parts that stick out (C1, C2), through the transmission of structural noise at the places of contact of the insulated machine with the foundation or floor of the room (D1, D2).

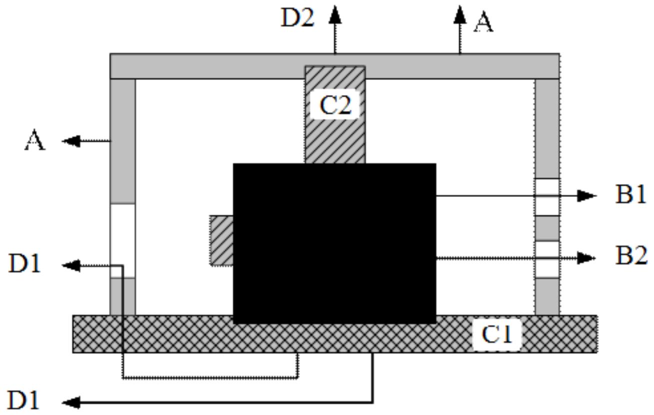

Figure 4: The possible ways of sound emission through the covering.

The noise reduction along path A is achieved due to an effective multilayer housing design using [19,20].

The propagation of air noise along path B should be primary taken into account [21]. Considering the fact that the test bench is designed to be used at short time intervals sufficient enough to bring the pump to a stable operating mode and then record its parameters, the noise reduction along B paths is reached by the absence of technological holes for ventilation and air ducts.

Reduction of noise along paths C in designed test bench is ambiguous: on the one hand the housing actually rely on the foundation and comes in contact with the front frame of the test bench, and on the other hand it is not rigidly connected to the front frame (through the felt layer).

The noise reduction along paths D is achieved by using the plate made of material with the highest energy absorption coefficient.

The walls of the sound-proof housing are designed according to the type of a multi-layer structure [22,23]: layers of sound absorbing material (technical felt, h=6mm) alternate with soundproofing (duralumin, h=1mm) and create a single-layer structure. The side walls and covering are designed as a five-layer structure. The wall on the motor side is double-layered, the floor of the casing is three-layered. Two layers of duralumin, alternating with layers of felt serve as an isolation for the side walls and covering. The plate, front and free frame (reference numbers 3, 7 и 15 on Fig.3) serve as the isolation for the floor, front and rear walls respectively. The layers of felt and duralumin are glued together with glue KL-88 and connected by the principle of labyrinth sealing in order to reduce leakage of noise from the housing [24]. The sound insulation of the wall from the motor side is the lowest due to the technological holes for the motor output shaft.

A plain bearing was inserted under the shaft of the front frame in order to increase sound insulation in the hole. The bearing prevents the propagation of noise at the exit of the shaft due to a good sound insulation capacity.

According to [21], sound insulation of single-walled structures, for panels with a surface weight of up to 15 kg/m^2 and up to 2000 Hz depends on the mass. The preliminary attenuation of noise by the [25] casing is determined by the formula:

$$\Delta L = 20 \cdot \lg \cdot m \cdot f + 10 \cdot \lg \cdot a - 60 \qquad (1)$$

where *m*- mass of 1 m^2 structure;

f – sound energy frequency (Hz);

α– coefficient of material sound absorption.

The structure mass was calculated [25] taking into account two materials - felt and duralumin. The estimated mass of 1 m^2 of the used structure is 12.16 kg. The sound absorption coefficient (SAC) for felt was chosen from the reference data for h=22 [26]

The result of reducing the noise using the housing is shown in Figure 5.

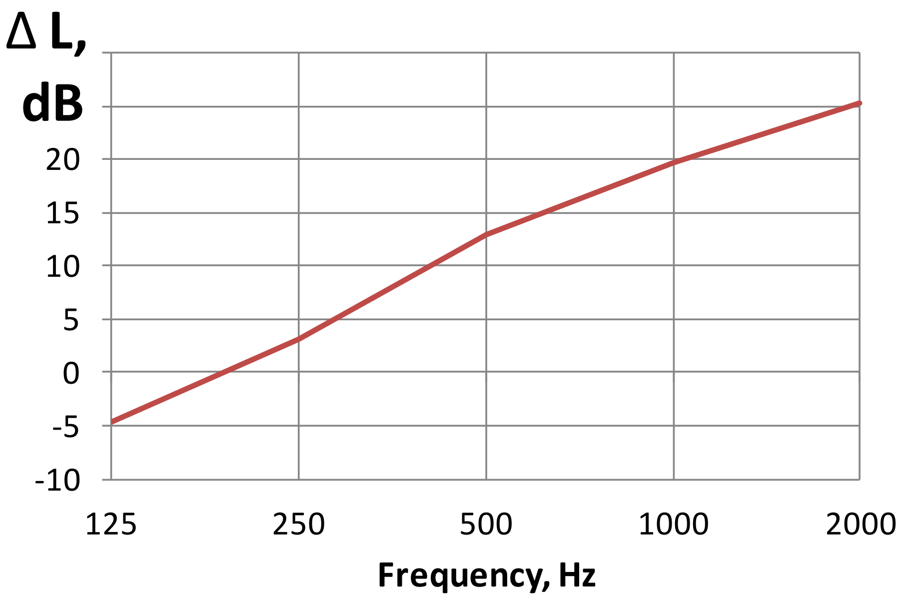

Figure 5: An acoustic efficiency of housing application.

Thus, the expected efficiency of the housing is calculated from 300 to 2000 Hz, which correlates with [21]. Determination of the actual housing acoustic efficiency should be carried out experimentally due to the existence of connections between the casing and the unit [27].

Considering availability and good values of the sound absorption coefficient for the walls of the housing, felt with a diameter 6 mm [28] is chosen. The dependence of the SAC on frequency for felt made of polyvinylchloride fibers (PVC) is shown in Figure6.

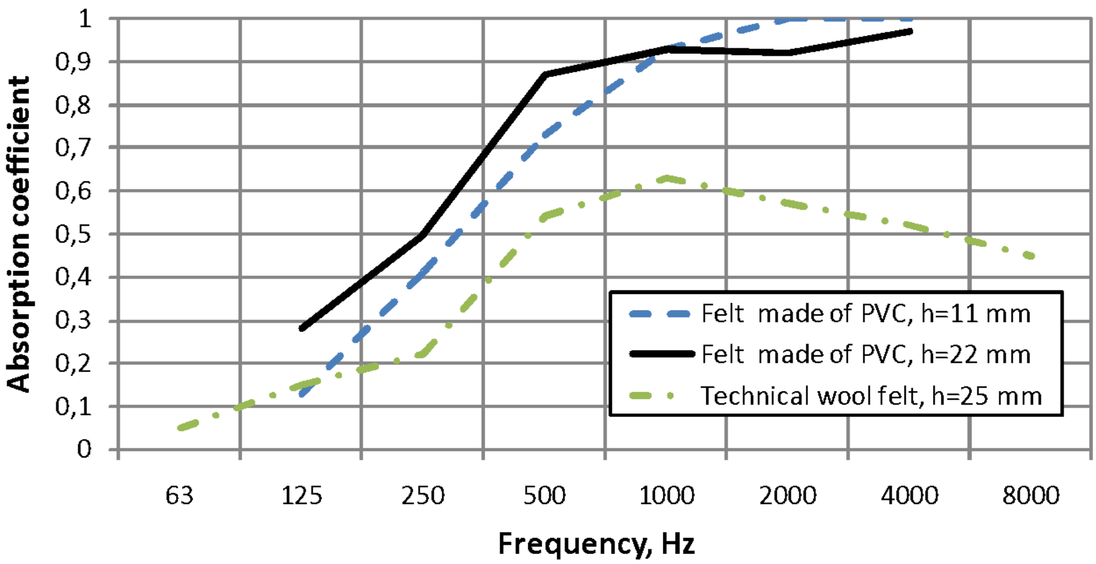

Figure 6: The felt sound absorption coefficient.

The coupling for connecting the motor and the pump is soundproofed by a fluoroplastic tube, which also reduces the emission of noise from the motor through the coupling to the environment.

According to [29], the coupling is the acoustic bridge, directly transmitting vibration from the motor to the pump. Its reduction is a vital task.

According to the Carletti [30], the noise emission from the pump housing due to the propagation of vibration from the motor and the mechanical interaction of the teeth can significantly exceed the noise from other processes. The following measures have been used to reduce the noise and vibration emission from the pump: selection of a low noise motor, vibration isolation of the pump and motor from the frame (through fluoroplastic spacers), using of a vibration damper (Fig. 7).

Figure 7: The elastic coupling.

Thus, a number of measures have been applied to reduce noise and vibration emitted from the additional elements of the test bench. Noise measurements at the test bench are planned to be carried out using an acoustic probe to measure sound intensity, which excludes the effect of reflection from walls and outside objects. The methods for vibroacoustic loads reduction leads to the background noise decreasing and obtaining more accurate acoustic parameters of the micropump when the rotor is changed to polymeric.

5. Conclusions & Discussion

The designed test bench allows conducting studies of vibroacoustic characteristics of pump units and its resource.

Further research will be focused on proving the efficiency of measures taken to reduce background noise. Future research will be continuously related to development tests, vibro-acoustic and other tests on the test bench. The test bench is also necessary to be regularly upgraded and improved.

Ways of improving the designed test bench:

- Evaluation of the acoustic isolation efficiency of the housing using calculation and experimental method;
- Increasing of the housing sound insulation at low frequencies;
- Implementation of a laser centring system of motor-pump shafts assembly;
- Pump temperature control;
- Pump housing tightness control;
- Control of parameters from one device/PC;
- Modal analysis of the pump unit [31];
- Automatic generation of test reports.

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