

Acoustic tests of type KPF1 high-pressure external gear pumps

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Abstract

This article presents the results of testing the sound pressure level and sound power level of the experimental 3PW-KPF1-24-40-2-776 high-pressure gear pump. Acoustic tests were conducted in an reverberation chamber. The results of the acoustic power tests indicate good acoustic parameters of the tested high-pressure unit.

Keywords: gear pump, high-pressure gear pump, noisiness of gear pump

1. Introduction

Among the displacement pumps used in hydrostatic drive systems, external gear pumps are the most commonly used as energy generators. Globally, they are estimated to comprise over 50% of all pumps produced (Osiński, Różycki, Rutański, 2015; Osiński, Palczak, Rutański, 2013). These pumps are widely used due to their simple and compact design, operational reliability, resistance to contamination of the working fluid and their relatively low production cost (Osiński, Palczak, Rutański, 2013). During operation, however, they are characterised by a relatively high operating noise level of ~80–85 dB(A). The noisy operation of the gear pump results from hydraulic factors related to phenomena occurring during the flow of the working fluid and from mechanical factors associated with the interaction between individual structural elements of the unit (Parise, Miccoli, Carletti, 2015; Miccoli, Pedrielli, Parise, 2016; Pavić, Chevillotte, 2010; Mucchi, Rivola, Dalpiaz, 2014). Mechanical factors are mainly related to errors in the manufacture and assembly of gears and bearings, excessive play in moving joints and errors in execution and assembly. These problems are relatively well recognised and can be effectively eliminated by accurate and advanced processing technology (Osiński, Palczak, Rutański, 2013). However, the basic reasons for the considerable noise of the gear pump are hydraulic phenomena which consist of:

- ▶ a sudden increase in pressure between the suction and discharge area of the pump,
- ▶ flow ripple and the resulting pressure pulsation,
- ▶ the phenomenon of liquid capping in the notches of gears,
- ▶ cavitation,
- ▶ variable loads resulting from the liquid impact on the gears.

The development of modern gear units is currently taking the following directions: increase of working pressures (Osiński, 2013), improvement of total efficiency (Zardin, Natali, Borghi, 2019), reduction of the occurrence of flow ripple and dynamic loads (Svishchev, Aistov, 2015), cavitation (Battarra, Mucchi, 2018; Stryczek et al., 2015), the minimisation of mass and the level of noise emitted by the pump (Rodionov, Rekadze, 2017; Zhao, Vacca, 2018).

The maximum discharge pressures achieved by conventional gear pumps reach 32 MPa. Obtaining higher operating pressures requires modification of the conventional pump. The pumping of pressurised liquids of up to 40 MPa is enabled by high-pressure gear units with modified structure. Changes that allow achieving such high pressures concern:

- ▶ compensation of the circumferential clearance,
- ▶ gears with a wider tooth tip and chamfering of the edge between the tip/ the top land and the face of the tooth on the pressure side.

A circumferential gap occurs between the tooth tips and the gear pump housing. The area where the gap occurs starts in the plane defined by the axes of rotation of the gears, and ends with the start of the discharge opening.

The above-mentioned criteria were defined for the first time in the application of the research project entitled 'Gear pump with compensation of circumferential clearance'. The concept of compensation of circumferential clearance with the aforementioned criteria was created for the implementation of research project N N502 147938. The research project was funded by the Ministry of Science and Higher Education at the request of Wrocław University of Science and Technology (application No. 73222 filed in July 2009).

Increasing the operating pressure of the gear pump leads to an increase the power-to-weight ratio. The possibility of achieving a high density of transmitted power is a significant advantage of a hydrostatic drive system (Kollek, Kudźma, Rutański, 2005). An operating pressure of 40 MPa provides approximately 24 kW of power from a stream of liquid with a flow rate of 0.6 dm³/s (36 dm³/min). Thus, increasing the operating pressure of the gear pump leads, on the one hand, to an increase of the generated power flux, and on the other hand, to an improvement of the power-to-weight ratio of the pump, which shows the compact design of this element.

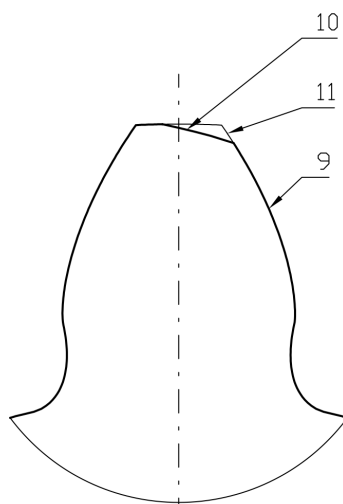
The power-to-weight ratio for pumps and hydrostatic motors can be as high as 10 kW/kg, while the power-to-weight ratio of standard electric motors is between 0.1–0.25 kW/kg (Kollek, 2011).

The elements of the hydrostatic drive system are characterised by a compact construction and low mass. Both the density of the transmitted power flux and power-to-weight ratio of the unit increase as the operating pressure increases. An important limitation in increasing these coefficients is the increase in the level of noise emitted that accompanies the increase of the power generated and transferred by the positive displacement unit and the hydraulic system. In modern machines and devices with hydrostatic drive, one can observe an aspiration to increase the discharge pressures of positive displacement units at the expense of decreasing the flow rate of the working medium (Kollek, Osiński, Rutański, 2007).

An attempt to develop a high-pressure gear pump with low flow pulsation was the basis for conducting acoustic measurements on the 3PW-KPF1-24-40-2-776 SN: 1experimental unit.

2. Object under study

The tested pump is a construction developed at Wrocław University of Science and Technology marked with model number 3PW-KPF1-24-40-2-776. It is an experimental, high-pressure pump with a displacement of $q = 24 \text{ cm}^3/\text{rev}$ based on the construction of a PZ4 type gear pump. In the experimental unit, the number of teeth and the width of the tooth heads were increased, improving the tightness between the wheels and the casing. The teeth also have chamfered edges between the head and the tip of each tooth on the pressure side (Fig. 1).



This solution is covered by patent number PL230846. The key feature of the pump is the profiling of the tip of each tooth in such a manner as to form a convergent gap, which will allow achieving a hydrodynamic lubrication effect. Such a profile in the region of the tooth tip creates more favourable conditions for creating an oil film between the teeth and the casing whilst preventing the occurrence of cuts to the body during pump operation. Modernised axial compensation results in higher axial tightness. Due to the much higher load resulting from the high pressure, the gears are made from steel with parameters much higher than those of the base pump. In addition, the gear bearings have been significantly upgraded.

Fig. 1. Outline of gears with widened tooth tip and chamfering of the edge between the head and the tip of the tooth on the pressure side, where: 9 – pressure side flank; 10 – chamfered edge; 11 – outline before chamfering (original numbering) (source: Osiński, Bury, 2018)

3. Measuring apparatus

The reverberation acoustic chamber in which the high-pressure pump was tested has a volume of 102 m^3 . The chamber is made of two similar irregular polyhedrons placed one inside the other. Each of these two parts of the chamber is mounted on an independent foundation. The floor in the room is a concrete block which is not connected to the walls and fulfils the function of the foundation of the tested elements mounted inside it. Each of the two walls of the chamber has different dihedral angles, which eliminates the possibility of an acoustic standing wave. The uniformity of the acoustic field distribution in the chamber is within acceptable limit. The reverberation time in the chamber is 4.13 s, which provides a diffuse field for the interior volume of 102 m^3 (Kollek,

Osiński, Rutański, 2001). A network of eight condenser microphones is placed in the acoustic chamber measurement field at a height of 1.3 m from the floor. According to the PN-EN ISO 3743-2 standard, the chamber can be used to determine the sound power levels of noise sources by a technical method based on the measurement of sound pressure.

The tested pump (Fig. 2) placed in the acoustic reverberation room was driven by a DC electric motor (2) with a power of 100 kW cooperating with a thyristor control system. The Pxob-94a DC motor and thyristor control system type DSI-0360 / MN-503 enabled a smooth change of the rotational speed of the pump in the range from 0 to 2400 rpm. The thyristor control system allows the precise selection of rotational speed and ensures its constant value during measurement. There was a torque gauge (19) on the shaft between the pump and motor. The torque gauge has a speed measurement function. The hydraulic system shown in Fig. 2 ensured a continuous pressure level on both the suction side and the discharge side. Control of the pressure level on the suction side was made possible by a system consisting of a feed pump (3) and adjustable needle valves (9 and 11). The load of the tested pump was obtained through a needle valve (10), and the pump was secured with a safety valve (7). Pressure gauges on the suction side of the pump were operated by vacuum/pressure gauges (13, 14), while the manometer (15) was located on the pressure side. The flow rate was set using sequentially

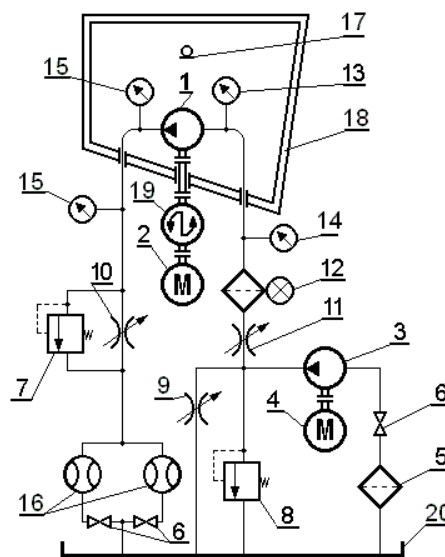


Fig. 2. Scheme of the measurement stand: 1 – tested gear pump; 2 – DC drive motor; 3 – feed pump; 4 – AC motor; 5 – suction filter; 6 – shut-off valve; 7, 8 – safety valves; 9, 10, 11 – stop valves; 12 – flood filter; 13, 14 – vacuum/pressure gauges; 15 – manometer; 16 – flow meter; 17 – measuring microphone; 18 – acoustic chamber; 19 – torque gauge; 20 – tank (source: Osiński, 2013)

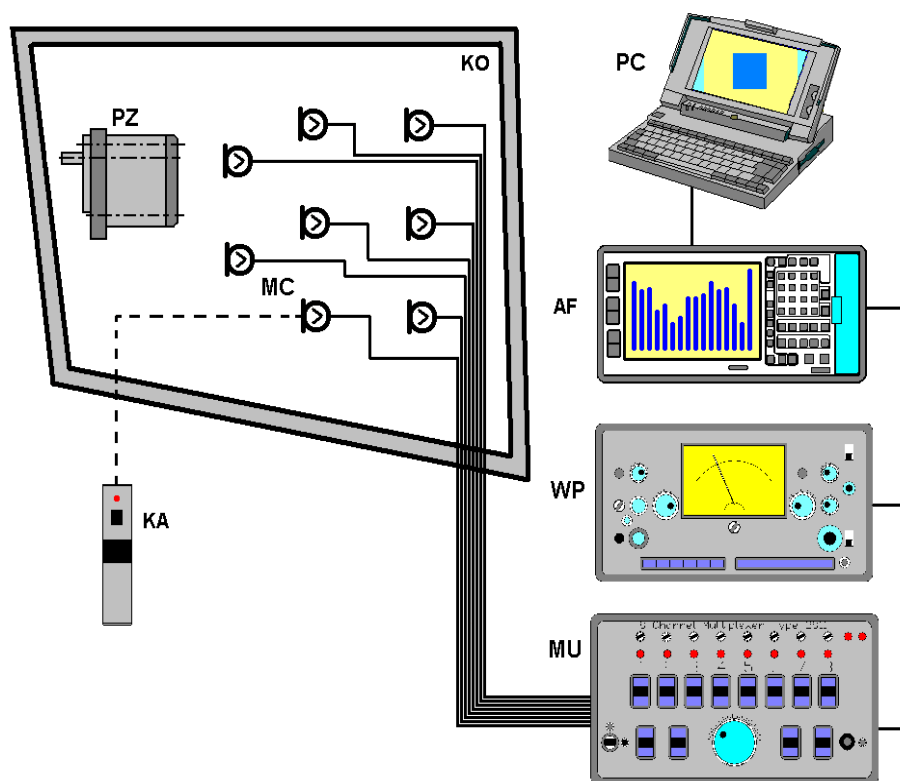


Fig. 3. Block diagram of the apparatus for measuring noise generated by the gear pump KO – reverberation chamber; PZ – hydraulic object, gear pump; MC – capacitive measurement micro-phones with preamplifiers type 4165 + 2639, B & K; MU – 8-channel multiplexer, type 2811, B & K; WP – universal measuring amplifier, type 2607, B & K; AF – two-channel frequency analyser type 2144, B & K; KA – acoustic calibrator (pistonphone), type 4220 for B & K; PC – computer (source: Osiński, 2013)

switched flow meters (16). The temperature of the working fluid was controlled via a thermistor sensor placed in the tank (20).

4. Apparatus for acoustic measurements

Figure 3 presents a block diagram for determining the octave band of sound pressure level in the band 125–8000 Hz and the resultant effects: the sound pressure level L_m and the sound level L_A in the frequency range 0–16000 Hz. The processing of the observed signal from the network of eight microphones arranged in the chamber was made according to the pre-set measurement function – a two-channel B&K frequency analyser of type 21144. The computer system was used to acquire and process data obtained from measurements and in the process of editing test results (Kollek, Osiński, Rutański, 2001). Registered octave spectra of the sound pressure level allowed, in accordance with the procedure contained in the PN-EN ISO3743-2 standard, determination of the sound power level. The entire apparatus for measuring noise was qualitatively checked, before and after the measurement, using a reference sound source (pistonphone).

5. Results

Acoustic tests of the experimental high-pressure gear pump type 3PW-KPF1-24-40-2-776 were conducted at rotational speeds of $n = 800; 1000; 1500; 2000$ rpm. As part of the research, the following measurements were performed: the sound pressure level L_p , the A-weighted sound pressure level L_A , the sound power level L_W and the A-weighted sound power level L_{WA} . The average value of the sound pressure level L_p was calculated according to the following formula:

$$L_p = 10 \lg \left(\frac{1}{n} \sum_{i=1}^n 10^{0.1 L_{pi}} \right) \quad (1)$$

where:

L_{pi} – sound pressure level at the i -th measuring point,
 n – total number of measuring points.

The mean value of the A-weighted sound level was determined in the same manner:

$$L_A = 10 \lg \left(\frac{1}{n} \sum_{i=1}^n 10^{0.1 L_{Ai}} \right) \quad (2)$$

where:

L_{Ai} – sound pressure level at the i -th measuring point,
 n – total number of measuring points.

The value of the A-weighted sound level was determined on the basis of the measured sound pressure level L_p^j in the j -th band and after the correction K_A^j resulting from the characteristics of the weighted curve using the following formula:

$$L_A^j = L_p^j + K_A^j \quad (3)$$

where:

L_p^j – sound pressure level in the j -th frequency band,
 L_A^j – A-weighted sound level in the j -th frequency band,
 K_A^j – correction according to the A characteristic for the j -th frequency band.

The sound power level L_W and the A-weighted sound power level L_{WA} were determined according to the exact method:

$$L_W^j = L_p^j + 10 \lg \frac{A_j}{A_0} + 10 \lg \frac{1 + \frac{S_V \lambda}{8V}}{1 - \frac{A_j}{S_V}} - 6 + C \quad (4)$$

where:

- L_p^j – the average value of the sound pressure level in the j -th frequency band,
- L_W^j – the average value of the sound power in the j -th frequency band,
- A_j – sound absorption in m^2 calculated in the j -th frequency band,
- $A_0 = 1 m^2$,
- S_V – chamber surface with floor,
- V – chamber volume in m^3 , $V_0 = 1 m^3$,
- λ – wavelength,
- C – correction depending on climatic conditions (for an atmospheric pressure of 100 kPa and a temperature of 20°C, $C = 0$).

The A-weighted sound power level L_{WA}^j in the j -th frequency band was determined according to the formula (5):

$$L_{WA}^j = L_W^j + K_A^j \quad (5)$$

The results of the L_W power level measurement and the A-weighted sound power level L_{WA} as a function of the pumping pressure p_t and the rotational speed of the pump shaft n for the experimental version of the high-pressure gear pump are shown graphically in the form of diagrams (Figs. 6 & 7).

Figures 4 and 5 present the changes in the sound pressure level L_p and the A-weighted sound level L_A as a function of the discharge pressure p_t set for the assumed rotational speeds. The resultant values of the sound pressure level and sound level which occurred at nominal operating parameters $n = 1500$ rpm and the delivery pressure $p_t = 40$ MPa are $L_p = 85.4$ dB and $L_A = 83.0$ dB (A), respectively.

Figures 6 and 7 show the course of the level of unweighted sound power L_W and the A-weighted sound power L_{WA} as function of the pumping pressure p_t

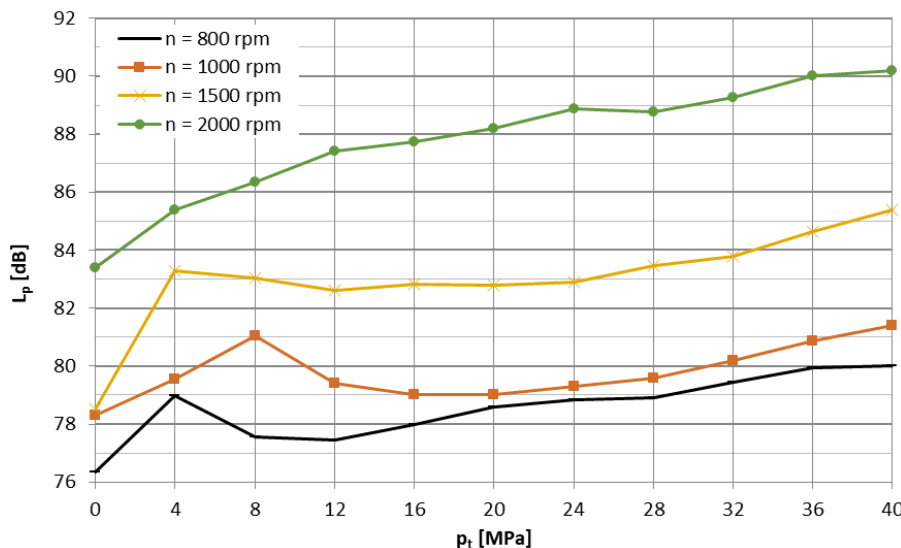


Fig. 4. Sound pressure level L_p as a function of discharge pressure p_t of experimental high-pressure gear pump 3PW-KPF1-24-40-2-776 (source: Autor)

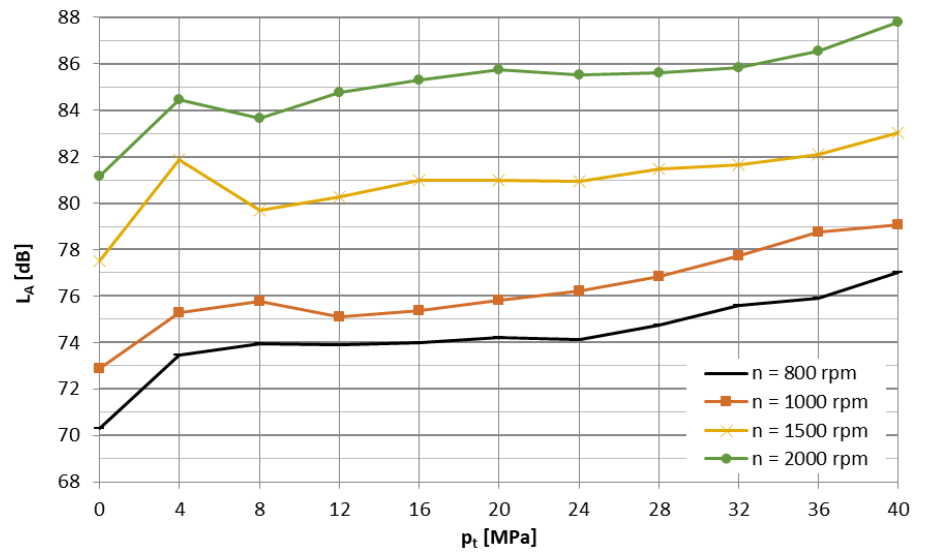


Fig. 5. A-weighted sound level L_A as a function of discharge pressure p_t of experimental high-pressure gear pump 3PW-KPF1-24-40-2-776 (source: own compilation)

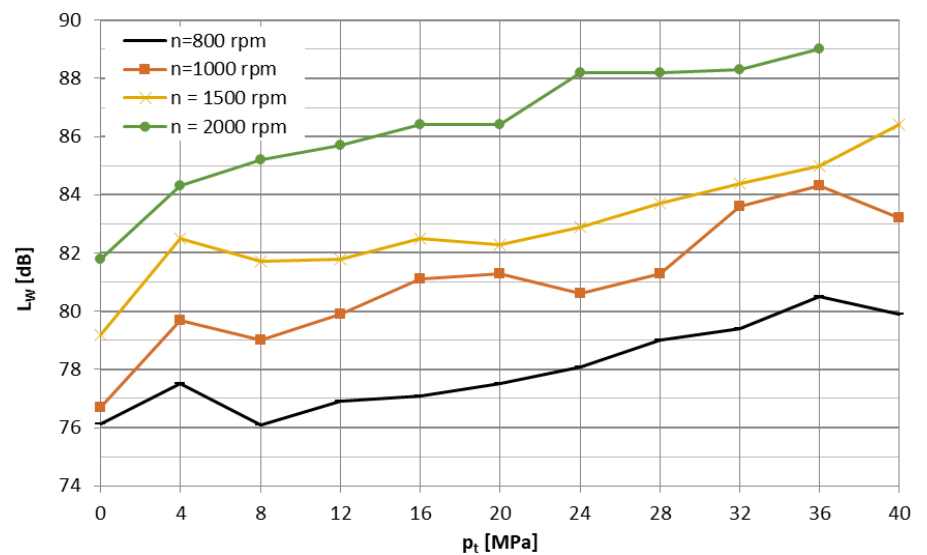


Fig. 6. Unweighted sound power level L_W as a function of the discharge pressure p_t of the experimental high-pressure gear pump 3PW-KPF1-24-40-2-776 (source: own compilation)

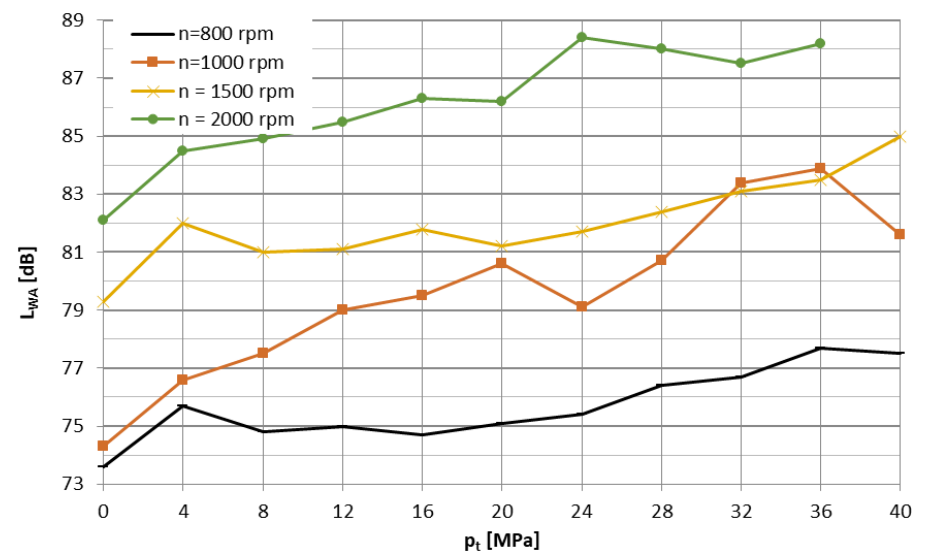


Fig. 7. A-weighted sound power level L_{WA} as a function of discharge pressure p_t of experimental high-pressure gear pump 3PW-KPF1-24-40-2-776 (source: own compilation)

set for four rotational speeds $n = 800; 1000; 1500$ and 2000 rpm. The values of the acoustic level of the unweighted L_W and the L_{WA} A-weighted sound power determined at the nominal operating parameters $n = 1500$ rpm and the discharge pressure $p_t = 40$ MPa, in the standardly accepted frequency range $f = 125\text{--}8000$ Hz, are $L_W = 86.4$ dB and $L_{WA} = 85.0$ dB (A), respectively.

Figures 8 and 9 show the characteristics of the uncorrected sound power level and the corrected sound power level of the tested unit with a conventional unit. For comparison, a conventional pump with a similar nominal output power was selected.

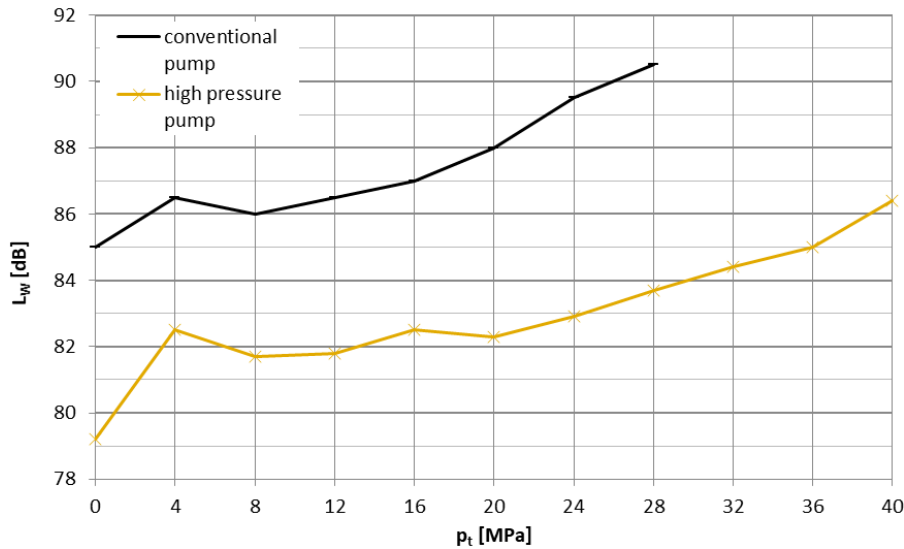


Fig. 8. Comparison of unweighted sound power level L_W as a function of discharge pressure p_t of conventional and high-pressure pump (source: own compilation)

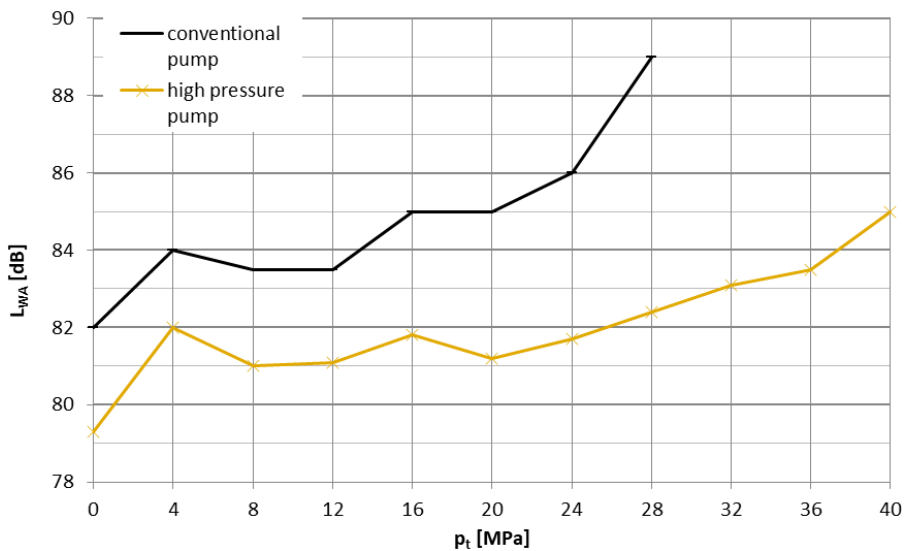


Fig. 9. Comparison of A-weighted sound power level L_{WA} as a function of discharge pressure p_t of conventional and high-pressure pump (source: own compilation)

6. Conclusion

The determined resultant sound level of $L_A = 83.0$ dB(A) obtained at high operational parameters resulting from the load $p_t = 40$ MPa indicates favourable acoustic properties of the tested pump. The currently valid normative recommendations specify the permissible value of the A-weighted sound pressure level for hydraulic pumps of $L_A = 85$ dB(A).

The performed acoustic measurements of the KPF1 experimental high-pressure gear pump are promising. It should be noted that the developed design is at the stage of conducting applied tests. The implementation of the new unit requires dimensional optimisation and optimisation of manufacturing tolerances to achieve high efficiency and durability.

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Testy akustyczne wysokociśnieniowych pomp zębatych o zazębieniu zewnętrznym typu KPF1

Streszczenie

W artykule przedstawiono wyniki badań poziomu ciśnienia akustycznego i poziomu mocy akustycznej dla eksperymentalnej wysokociśnieniowej pompy zębatej 3PW-KPF1-24-40-2-776. Badania akustyczne przeprowadzono w akustycznej komorze pogłosowej. Uzyskane wyniki mocy akustycznej świadczą o dobrych parametrach akustycznych badanej jednostki wysokociśnieniowej.

Słowa kluczowe: pompa zębata, wysokociśnieniowa pompa zębata, hałaśliwość pomp zębatych