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3D printed turbine-like inserts as an effective pressure pulsation dampers in positive-displacement compressors manifolds

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ABSTRACT

Pressure pulsations in positive-displacement compressors manifolds are well known phenomenon to industry and science. However, in the field of developing methods for damping pressure pulsations, there is still no universal solution that would be simple, cheap and did not significantly affect the compressed gas system. One of the proposed solution, which has been developed for several years, is the installation of shaped inserts in the compressor discharge pipeline for pressure pulsation damping. It has already been shown that such inserts suppress pressure pulsations at wide range of frequencies. It is very important especially in refrigeration installations with continuous compressor operation. The significant disadvantage of the narrowing of the pipeline is pressure drop, which increases the demand for compression unit power. Thanks to the 3D printing techniques, it is possible to design very complex shapes that will allow for the damping of pressure pulsations to a sufficient extent and at the same time will not have a significant impact on the compression power. The conducted experimental studies for two different turbine-like shapes show that it is possible to influence pressure pulsations and at the same time not to generate significant losses in the flow. As part of the work, the influence of two different shapes was compared. Fixed nozzles and nozzles with 1 degree of freedom of movement (rotation) were compared. The tests were carried out on two test stands. A reciprocating compressor was used for lower frequencies investigations and a screw compressor stand was used for higher pressure pulsation frequencies.

1. INTRODUCTION

Pressure pulsations which occur in installations of the volumetric compressor are the main noise and vibration source in gas manifolds. According to the American standards API 618, the elements used to dampen pressure pulsations should keep their peak-to-peak values below strictly defined percentages in relation to the average pressure in the system (API STANDARD 618, 2007). The most popular method of damping pressure pulsations is mounting Helmholtz resonators (Ma and Min, 2001) in the compressor manifold. However, this method is effective only for constant value of pulsation frequency. For contemporary applications of variable revolution speed compressors, other pressure pulsation attenuation methods are needed. Cyklis & Mlynarczyk proved in papers (2017)(2018)(2020) the effectiveness of pulsation damping, in a wide frequency range, by shaped nozzles. The results presented in these investigations shows the influence of standard, axially symmetrical shapes on the damping of pressure pulsations when they are rigidly fixed in a pipeline or fixed on an elastic element, creating a mass-spring system. Therefore, the differences between a fixed nozzle and nozzle with linear motion were determined. No evaluation, of the influence of the rotational movement of the shaped element on the damping of pressure pulsation, was found in the literature. In order to investigate the effect of the rotational motion of the elements inserted into the installation, the element can not be axisymmetric. In order to make the element rotate, the forces generated by the flowing compressed gas must generate asymmetrical radial forces on the walls/blades. The manufacturing of these types of elements using subtraction methods is expensive and complicated. Additive manufacturing methods, such as 3D printing, are therefore a suitable, cheap and effective way to make these types of elements. Currently, this type of approach seems to be more common in this type of research. In several papers different approach of 3D printing and usage of that elements was described. In paper (Costa-Baptista et al., 2019) an unexpected acoustic absorption phenomenon in 3D printed Helmholtz resonators is described. As the compression increase the gas temperature the possibility of using 3D printed materials in high temperatures must be considered. Such an approach was described in paper (Park et al., 2021). In the presented investigations nozzles were printed from different types of materials and in both cases the nozzles were not degraded under the influence of temperature and oil contained in the compressed air.

2. INVESTIGATED NOZZLES

During the research 6 different nozzles, which represent two approaches of the turbine-like shape design, made with 3D printing techniques were tested. First approach (I) is a nozzle with blades fitted around the perimeter of the nozzle and the second (II) is based on the two axially attached intersecting twisted blades. The nozzles had to be adapted to the dimensions of the installation. For the SAF-23 reciprocating compressor installation, the nozzles have a diameter of 17 mm and a length of 30 mm, while in the case of the DEMAG.DS-40 screw compressor installation, the diameter of the nozzles is 35 mm and the length is 120 mm. In the figure 1. designed CAD models of the nozzles for both test setups are presented.

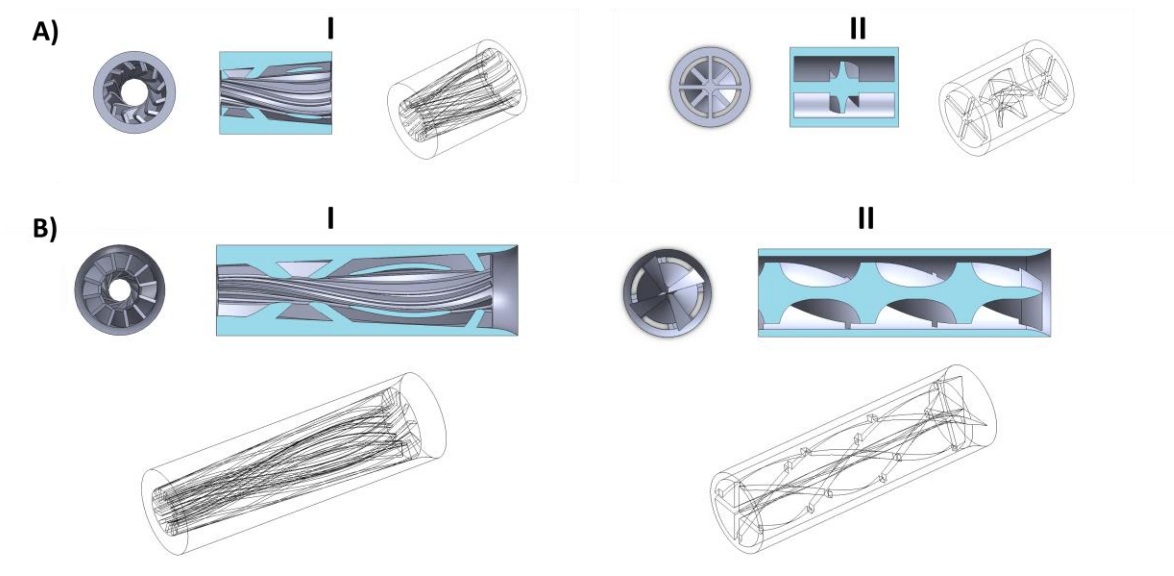

Figure 1: Turbine-like nozzles designs: I – nozzle with perimeter blades, II – nozzle with two intersecting twisted blades. A) Design for the SAF-23 reciprocating compressor test installation, B) Design for the DEMAG.DS-40 screw compressor test installation.

All investigated shapes were printed with the use of a Formlabs FORM 3 printer in SLA/LFS technology. For the four, rigid nozzles, presented in the fig. 1., the main design assumption was to obtain 30% clearance from the face of the nozzle. Two nozzles, presented in the fig. 2. were designed with elements that were rotating under the influence of gas flow in the installation. Due to the construction and assembly reasons, each nozzle consisted of 3 elements.. The body consisted of two permanently glued elements inside which a freely moving rotor was placed. About 70% of the cross-sectional area of the pipeline was filled in the space where the rotor was located. At this stage, the use of bearings was not planned, therefore, in order to ensure movement and reduce the friction surface, tabs with a circular profile were made. The nozzle components were printed from transparent resin. In order to confirm that the nozzles were rotating due to the gas flow in the installation, black lines were painted on the rotors and bodies. During assembly, the rotor was positioned in such a way that the lines coincide. After the test, the nozzles were gently disassembled. It was checked whether the lines were displaced, which indicated rotation. Rotation was observed in both nozzles in all performed tests.

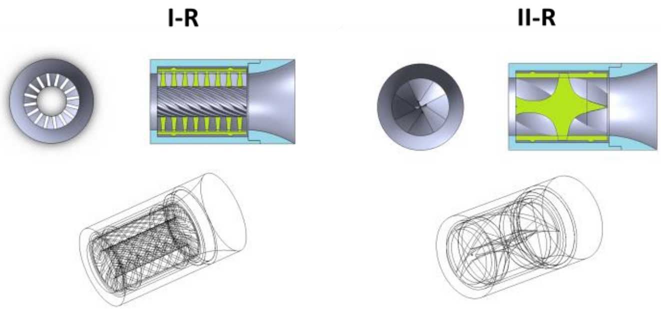

Figure 2: Rotary turbine-like nozzles designs for screw compressor installation: I-R – rotating nozzle with perimeter blades, II-R – rotating nozzle with two intersecting twisted blades.

3. EXPERIMENTAL INVESTIGATIONS SETUP

Measurements of the pressure pulsations were performed on two test stands. First stand is equipped with the reciprocating compressor SAF-23 while the DEMAG-DS 40 screw compressor is the basis of the second stand. Results of the experimental investigations, for both test stands, are shown for the pressure pulsation sensor placed behind the nozzle.

3.1 Reciprocating compressor test stand

The two-cylinder reciprocating air compressor SAF-23 was used to perform the experimental investigations for low frequency of pressure pulsations. The reciprocating compressor is driven by a variable-speed electric motor and an inverter was used to control the rotational speed of the drive shaft. For all measurements, the discharge pressure was kept constant at 1,5 bar. For each case the mass flow rate was calculated with the use of values from thermoanemometer and the compression power was calculated from the indicator chart. The test stand scheme is shown in Fig. 3. Compressor outlet pipe has an inner diameter of 12 mm, installation pipe diameter is 22 mm and the diameter of nozzle mounting pipe is 17 mm. The total length of the installation from the compressor to the tank is 4 meters.

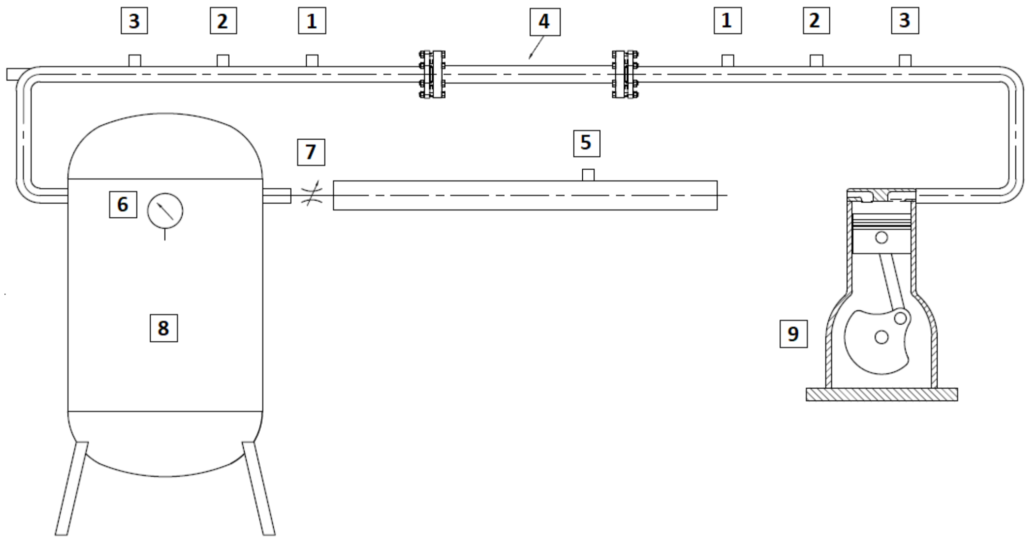

Figure 3:. The scheme of the SAF-23 test stand. 1 - Differential pressure measuring points, 2 – Dynamic pressure sensors, 3 -Thermocouples, 4 – Nozzle mounting pipe, 5 – Thermoanemometer, 6 – Manometer, 7 - Throttle valve, 8 - Compressed air tank, 9 – Compressor

The two-cylinder reciprocating air compressor SAF-23 was used to perform the experimental investigations for low frequencies between 16 and 36 Hz. Nozzles I and II for this test stand are shown in the figure 4.

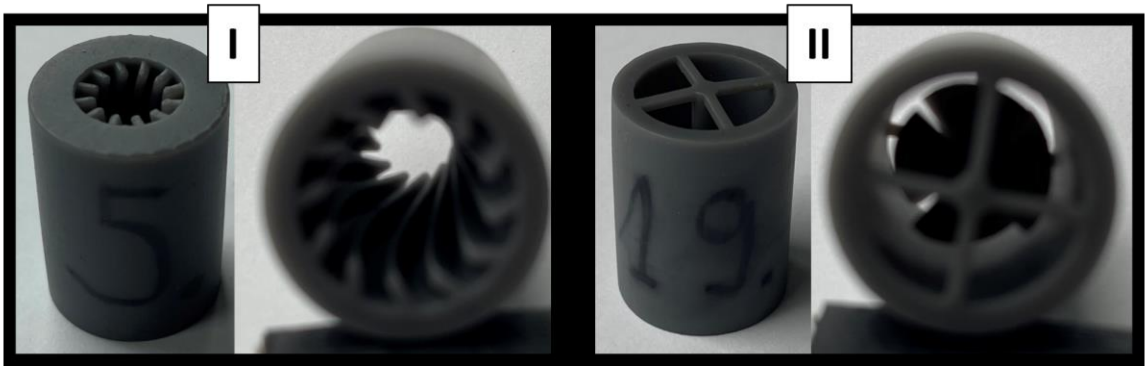

Figure 4: 3D printed nozzles for reciprocating compressor manifold

3.1 Screw compressor test stand

The main element of the stand is the DEMAG.DS-40 CompAir compressor unit. A three-cylinder combustion engine with a capacity of 2.2 liters and a power of 35,600 watts serves as the drive. This engine drives a screw compressor with a nominal capacity of 4.2 cubic meters per minute at an operating pressure of 7 bar. Test stand scheme is shown in Fig. 5.

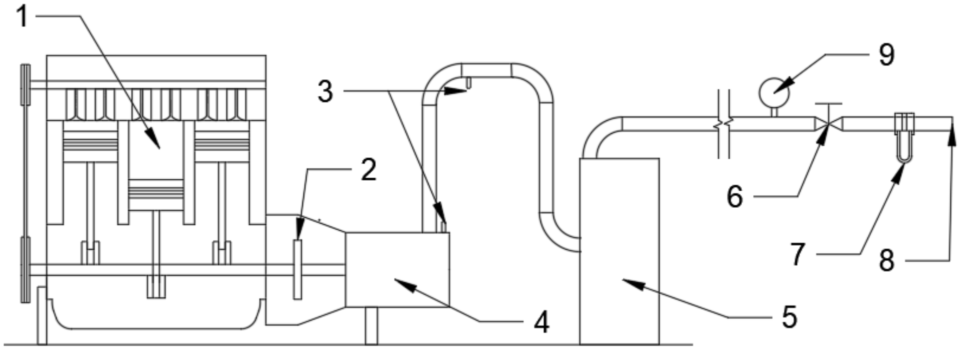

Figure 5: The scheme of the DEMAG.DS-40 test stand. 1- Internal combustion engine; 2 - Torque meter on the propeller shaft; 3 - Pressure pulsations sensors; 4 - Screw compressor; 5 - Oil separator; 6 - Throttle valve; 7 - Metering orifice; 8- Outflow; 9 - Manometer

For all measurements, the discharge pressure was kept constant at 3,1 bar. For each case, the mass flow rate was calculated with the use of a metering orifice and the compression power was read from a Magtrol torque meter display. Compressor outlet pipe, installation pipe and nozzle mounting pipe have an inner diameter of 36 mm. The total length of the installation from the compressor to the oil separator tank is about 1 meter long. Nozzles I and II, for this test stand, are shown in the figure 6.

Figure 6: 3D printed nozzles for screw compressor manifold. Left – rigid nozzles, right – rotating nozzles

The measurements of the pressure pulsation value at both test stands were carried out with the use of the same measurement circuit. The signal from dynamic pressure sensors goes through the 4-channels ICP Sensor Line Power unit to the NI USB-6251 data acquisition module and was recorded by LabView/SignalExpress software.

4. RESULTS

The main purpose of the research was to assess the effect on the compressed air flow parameters by the turbine-like shapes in the discharge collectors of piston and screw compressors. As an additional element of the research, the assessment of the impact of the rotation of these elements on the damping of pressure pulsations and the demand for compression power in the screw compressor installation was specified.

4.1 Rigid nozzles

The first stage of the presented investigations concerned the assessment of the influence, of the rigid shapes described in chapter 2, on the pressure pulsation damping in the installations described in chapter 3. The tests were carried out for various pulsation frequencies, with the use of nozzles of various dimensions adapted to the installation pipes. On the reciprocating compressor test stand, the pressure pulsation frequencies between 16 and 36 Hz were examined. Frequencies ranging from 230 to 314 Hz were investigated on the screw compressor test stand. The measured signals are shown in Fig.7, in order to show the difference in the nature of pressure pulsation damping for both stands at different frequency ranges.

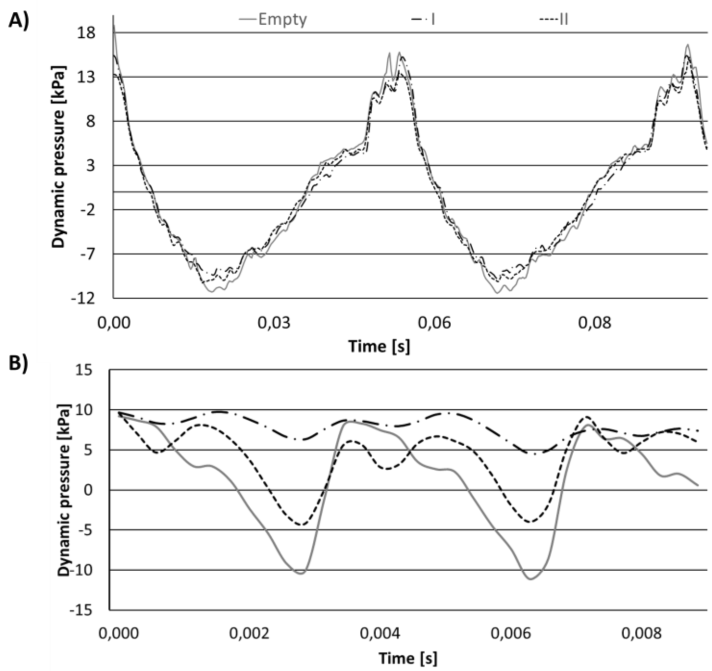

Figure 7: Pressure pulsation signals measured for empty installations and for installations with rigid nozzles. A) Reciprocating compressor manifold – forcing frequency 20 Hz (600 $^{rev}/_{min}$), B) Screw compressor manifold – forcing frequency 286 Hz (2000 $^{rev}/_{min}$)

As it can be seen in the figure 7. influence of inserting damping element into the compressor manifold results are very different for both installations. Pressure pulsation damping in the reciprocating compressor manifold is barely noticeable in the signal waveform (A), while for a screw compressor installation, significant input damping can be seen (B). The difference in attenuation for the tested cases may result from two phenomena:

- low efficiency of suppression of very low pulsation frequencies (16-36Hz) in reciprocating compressor installation;

- the ratio, between nozzle length and the length of the test installation between the compressor and the tank, which is much bigger for the reciprocating compressor manifold.

In the figure 8. the pressure pulsation peak-to-peak values for different revolution speeds on both stands are presented. As can be seen the pressure pulsation values are rather similar for all shaft revolution speeds on the screw compressor stand. Nature of the pressure pulsations in the reciprocating compressor manifold are highly dependent on the forcing frequency. Apart from the obvious influence of the nature of such low frequencies, it can also be caused by the long run of the discharge manifold.

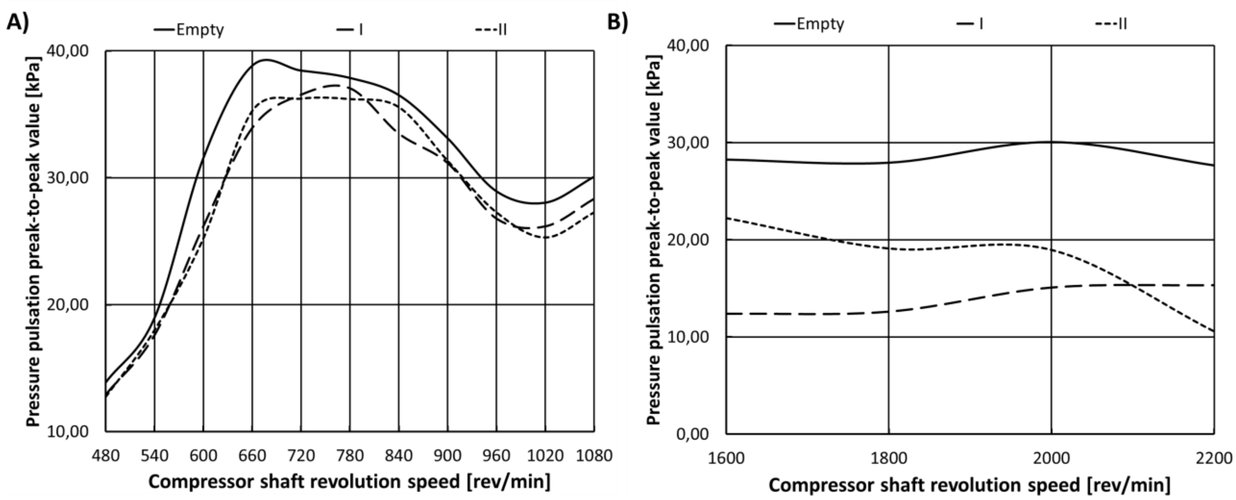

Figure 8: Pressure pulsations peak-to-peak values measured for empty manifold and with nozzles inserted for different forcing frequencies, A) Reciprocating compressor manifold, B) Screw compressor manifold.

Comparing graphs 7 and 8, it can be seen that, for a screw compressor and a rotational speed of 2000 rpm, the differences between the delta p value for nozzle II and the empty pipe in graph 7B are higher than it appears from the values shown in graph 8B. In the waveforms of the nozzles mounted on the screw compressor stand, a large influence of a very low frequency - independent of the compressor's forcing frequency - on the pressure pulsation peak-to-peak value was observed. This phenomenon appeared for each printed nozzle installed in the screw compressor test installation and perhaps it was related to the small ratio of the length of the measuring installation to the length of the installed nozzle. This frequency did not occur in empty installation measurements. Therefore, the values of the first three harmonic frequencies resulting from the forcing frequency of the compressor are also compared and shown in the fig. 9.

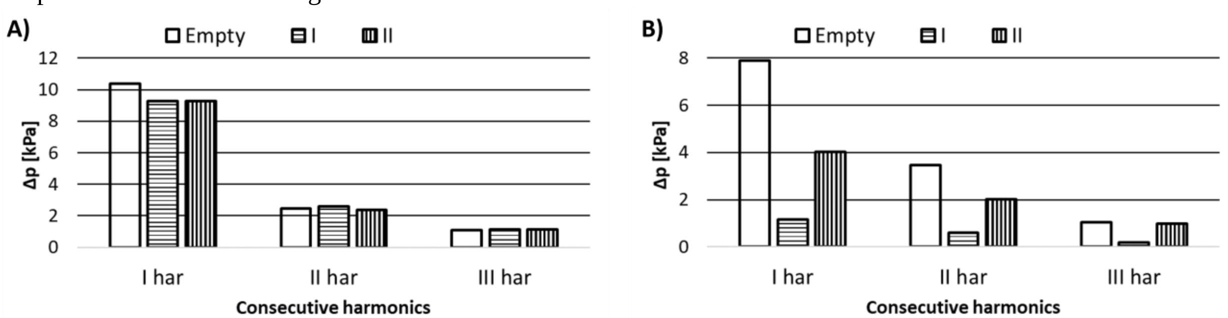

Figure 9: The value of the pressure pulsations for the consecutive harmonics resulting from the forcing frequency of the compressor. A) Reciprocating compressor manifold, B) Screw compressor manifold

The damping of the main harmonic frequencies has the greatest effect on the overall damping of peak-to-peak pulsation values (Fig.9).

Shaped nozzles affect the compression power and the stream of compressed gas. In order to evaluate influence of different shapes on these values, an additional new parameter in a form of the specific compression power (SCP) was introduced and described by equation 1:

$$SCP = \frac{\dot{V}_{SI}}{N} \tag{1}$$

To compare influence of nozzles on the pressure pulsations damping, and on the specific compression power gain, the average value for all measurements on both test stand were calculated. In the figure 10. the comparison between average percentage pressure pulsation damping and average percentage specific compression power gain for all rigid nozzles investigations are presented.

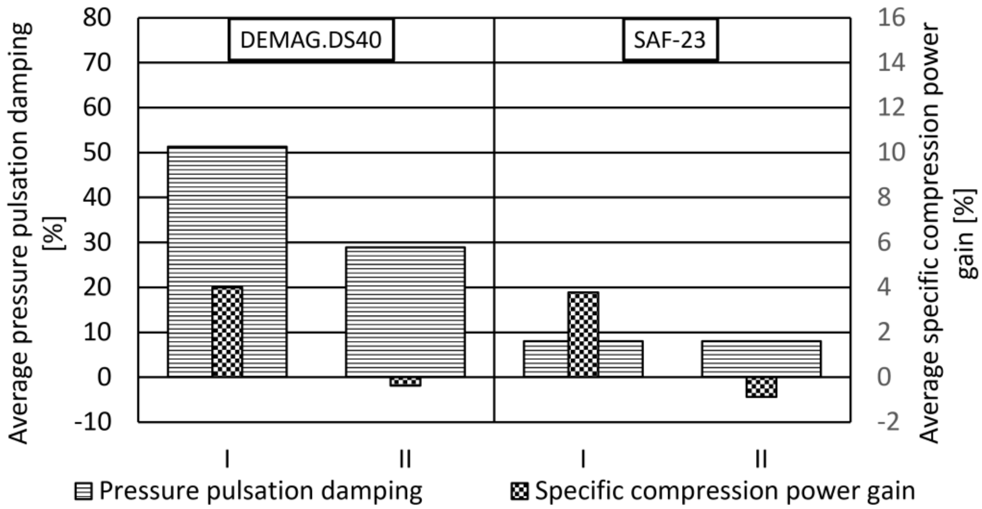

Figure 10: Average pressure pulsation damping and average specific compression power gain of testes nozzles for two installations.

As it can be seen in the figure 10. both nozzles have similar average pressure pulsation damping values on the SAF-23 installation. On the Demag.DS-40 installation nozzle no. I reaches almost two times higher pressure pulsation damping than nozzle no. II. In the fig. 10. also can be seen that the both nozzles influence the SCP in similar way on both installations. Nozzle II is particularly distinguished by the fact that the SCP parameter for measurements with its use is lower than for an empty installation. This could mean that the nozzle has a positive effect on the dynamics of the gas flow downstream of the compressor discharge valve. However, the percentage difference in both cases is below 1%, which is within the measuring error range.

4.2 Rotating nozzles

The second part of the presented investigations was related to the assessment of the impact on pulsations and compression power by rotating nozzles. Rotating nozzles shape philosophy was based on the rigid nozzles shapes and are presented in the fig. 2. Rotary nozzles, however, were made in a shorter version and were tested only on a screw compressor installation. The resultant waveforms obtained for the 2000 rev/min compressor shaft speed compared to the waveforms of rigid nozzles are presented in the figure 11.

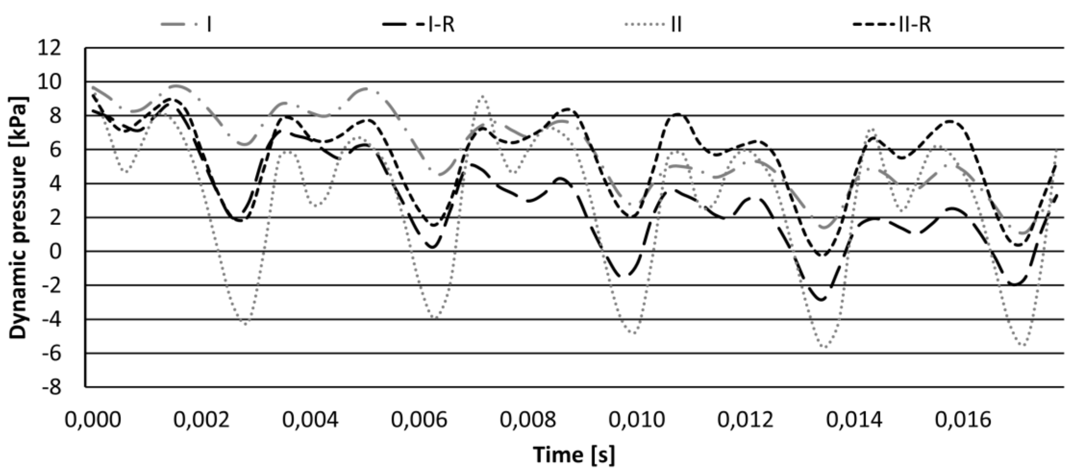

Figure 11: Pressure pulsations waveforms for rigid and rotating nozzles in screw compressor manifold for compressor shaft revolution speed equal to 2000 $^{rev}/_{min}$

It is visible in the figure 11 that rotating elements change the waveform behavior in different way. In particular, it may seem that the amplitude of the main excitation for the rotating shape I-R increases relative to the stationary nozzle, while it is quite different in the case of shape II-R. In Figure 12. it can be seen that indeed type II better suppresses the main excitation in the rotary form, while type I in the stationary form.

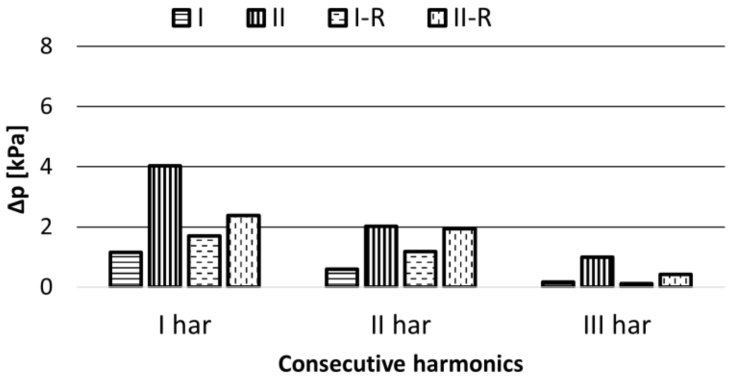

Figure 12: Comparison of consecutive harmonics values for rigid and rotating nozzles in DEMAG.DS-40 compressor manifold.

In table 1. damping efficiency coefficient, introduced in (Mlynarczyk & Cyklis, 2018), defined in equation (2) is presented as an indicator to compare different nozzles and its usefulness in pressure pulsation damping.

$$C_{D\eta} = \frac{B}{\sqrt{\Psi}} \tag{2}$$

The smaller the value of the coefficient, the more justifiable the use of a nozzle for pressure pulsation damping. When the value of the coefficient reaches a negative value, it means that apart from damping pressure pulsations, an increase in the efficiency of the gas compression process is obtained.

Table 1: Measured and calculated parameters for all investigated cases on the screw compressor test stand

Nozzle	Average pressure pulsation peak-to-peak value [kPa]	Pressure pulsation damping [%]	Specific compression power gain [%]	**Damping efficiency coefficient [-]**
Empty	28,49	0	0	**-**
I	13,86	51,3	3,99	**0,56**
II	20,22	28,9	-0,38	**-0,07**
I-R	21,82	23,2	5,19	**1,08**
II-R	16,04	43,6	3,89	**0,59**

According to the API standard 618 in the compressor manifold, where the discharge pressure is below 3,5 bar, the maximum allowable peak-to-peak pulsation level must be below the percent of the average mean absolute pressure calculated from equation (3) :

$$P_1 = \frac{4,1}{\sqrt[3]{(P_L)}}\% \tag{3}$$

where P_L – average mean absolute line pressure, in bar.
In presented investigation, for average mean absolute line pressure equal to 3,1 bar the P_1 is equal to 6,86% which corresponds to 21,26 kPa allowable pressure pulsation peak-to-peak value. In the presented investigations. empty installation and rotating nozzle I-R do not meet this criterion. As it can be seen in the table 1. the stationary nozzle II meets this criterion and has the best value of the damping efficiency coefficient. On the other hand, the best value of pressure pulsation damping was noticed for the I nozzle. However, due to the significant economic and ecological significance related to the compressor energy consumption, considerable attention should be paid to this aspect of operation.

4. CONCLUSIONS

- 3D printed turbine-like nozzles can effectively dampen pressure pulsations in a screw and reciprocating compressor manifolds in wide range of their frequencies,
- Rotary turbine-like nozzles have negative effect in the terms of specific compression power gain,
- Further optimization of the shape of rotating nozzles can allow for effective damping of pressure pulsations and minimize the impact of such nozzles on the demand for compression power.

NOMENCLATURE

SCP	specific compression power	(kW/m_{SI}3/s)
V_{SI}	volumetric flow in normal SI conditions	(m_{SI}^3/s)
N	compressor power	(kW)
$C_{D\eta}$	damping efficiency coefficient	(–)
B	relative dimensionless volumetric power consumption	(%)
ψ	dimensionless relative pressure pulsation damping parameter	(%)
P_1	percent of the average mean absolute pressure	(%)
P_L	average mean absolute line pressure	(bar)

REFERENCES

API STANDARD 618 FIFTH EDITION. (2007). *Reciprocating Compressors for Petroleum, Chemical, and Gas Industry Services.* American Petroleum Institute.

Ma, Y.-C. , Min, O.-K. (2001). Pressure calculation in a compressor cylinder by a modified new Helmholtz modelling. *Journal of Sound and Vibration.* 243, 775–796 .

Cyklis, P., Młynarczyk, P. (2017). An innovative simulation method for the estimation of the nozzle pressure pulsation attenuation. *Journal of Vibration and Control*, 23(16), 2690-2703.

Młynarczyk, P., Cyklis, P. (2018). The application of nozzles for the attenuation of volumetric compressor pressure pulsation. *International Journal of Refrigeration*, 90, 108-118.

Młynarczyk, P., Cyklis, P. (2020). The estimation of the pressure pulsation damping coefficient of a nozzle. *Journal of Sound and Vibration*, 464, 115002.

Costa-Baptista, J., Fosting, E.R., Therriault, D., Ross, A. (2019), Impact of 3D printed thin wall on Helmholtz resonator acoustic response: experimental investigations. Editor: ICSV26 Local Committee in Montreal, Proceedings of the 26th International Congress on Sound and Vibration : Montreal bridges. Montreal: Canadian Acoustical Association

Park, S.J., Lee, J.F., Park, J., Lee, N.-K., Son, Y., Park, S.-H. (2021). High-temperature 3D printing of polyetheretherketone products: Perspective on industrial manufacturing applications of super engineering plastics. *Materials & Design*, 211, 110163

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