Experimental investigations of pressure pulsations damping in screw compressor system

MSc eng. Przemysław Młynarczyk

Supervisor: Prof. dr hab. inż. Piotr Cyklis

Abstract

Pressure pulsations are one of the major problem in compressors systems. Most of the phenomena caused by pressure pulsations had bad influence on the whole installation. Wide range of machines which induce pressure pulsations cause that there is no universal solution to this problem. Nowadays the refrigerating compressors with variable rotational speed are becoming increasingly important. In this article author presents results of a series of measurements of passive pressure pulsations attenuation for selected rotational speed which, according to the author's thesis, depends mostly on the geometry of the damping element. Author also conducted a series of measurements with variable rotational speed for selected muffling elements. The results are shown in the paper, as well as discussion of the results and some conclusions.

Keywords

Compressors, Pressure pulsations

1. Introduction

Periodic work flow of the volumetric compressors causes the pressure and mass flow pulsations. Most of the phenomena triggered by pressure pulsation, like vibrations, acoustic noise, energy loss, valves wear are harmful to the installation and its components. Wide range of compressors, from large, low-speed natural gas compressors to small, compact refrigeration compressors with variable speed makes it difficult to find a universal solution. Especially in small refrigeration compressors it is important to mute the work process of these devices. Since in the modern refrigeration compressors variable speed work are frequently introduced, the simple pressure pulsation damper designed for basic pressure pulsation frequency may not work over a wide range of revolution speed. While there are many studies of pressure pulsation damping using classical methods there is no practical solution to suppress different values of pressure pulsation with variable mass flow rate over time using one compact damping element.

2. Investigated choking elements

The passive pressure pulsations damping were analysed with using specially shaped nozzle, which is placed in the gas duct flow, directly after the compressor outlet chamber place of the straight tube. Author used three different shapes in different configurations to check which element, or which configuration can be improved for further consideration. Every shape was investigated in three different dimensions of the inner flow diameter which was: φ10, φ15 and φ20. Some chosen shapes are used in measurements in both directions and in configuration of two elements joined together.

2.1 Venturi orifice

This kind of orifice, presented in the fig. 1., is geometrically the simplest one. This element suppress pressure pulsations well and at the same time does not much loss in the fluid flow rate. The biggest disadvantage of this element is relatively large geometrical dimensions according to the others.

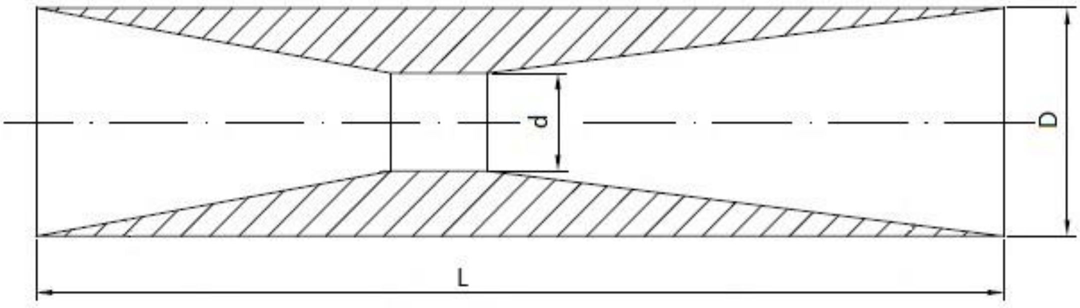

Fig. 1. *Shape and main dimensions of the Venturi orifice*

2.2 Venturi nozzle

Shape of this nozzle is still not difficult to perform and reduces significantly the geometrical dimensions of the damping element. It is important in the small compressor systems. This nozzle is characterized by very similar parameter of pressure pulsations attenuation and mass flow rate like Venturi orifice. Shape of this element is shown in the figure 2.

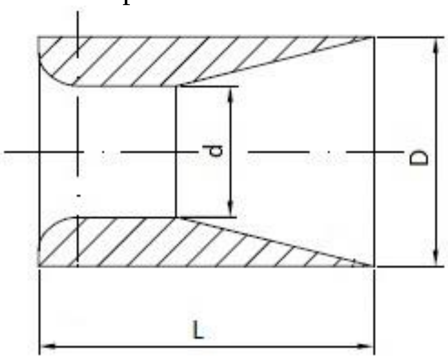

Fig. 2. *Shape and main dimensions of the Venturi nozzle*

2.2 Hyperboloid nozzle

Hyperboloid nozzle has a most complicated geometry of all investigated elements. This nozzle is characterized by a high attenuation of each harmonic at the same time without too much influence in the power demand. Hyperboloid nozzle is shown in the figure 3.

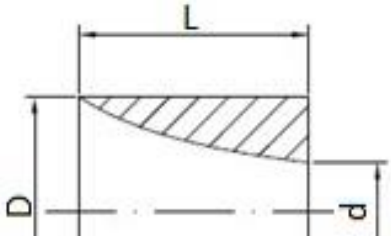

Fig. 3. *Shape and main dimensions of the hyperboloid nozzle*

The investigated choking elements dimensions are shown in table 1.

Table 1. *– Dimensions of investigated choking elements*

Element	***Outer diameter D [mm]***	***Inner diameter d [mm]***	***Length L [mm]***
Venturi nozzle	φ35	φ 20	51
Venturi nozzle	φ35	φ 15	55
Venturi nozzle	φ35	φ 10	60
Venturi orifice	φ35	φ 20	120
Venturi orifice	φ35	φ 15	145
Venturi orifice	φ35	φ 10	160
Hyperboloid nozzle	φ35	φ 20	35
Hyperboloid nozzle	φ35	φ 15	35
Hyperboloid nozzle	φ35	φ 10	35

3. Installation and laboratory test stand

All measurements of the pressure pulsations have been performed in the DEMAG screw compressor outflow pipe. The place of assembly of the investigated elements is located 17mm above the compressor discharge chamber. The specially prepared installation with dimensions is shown in the figure 4.

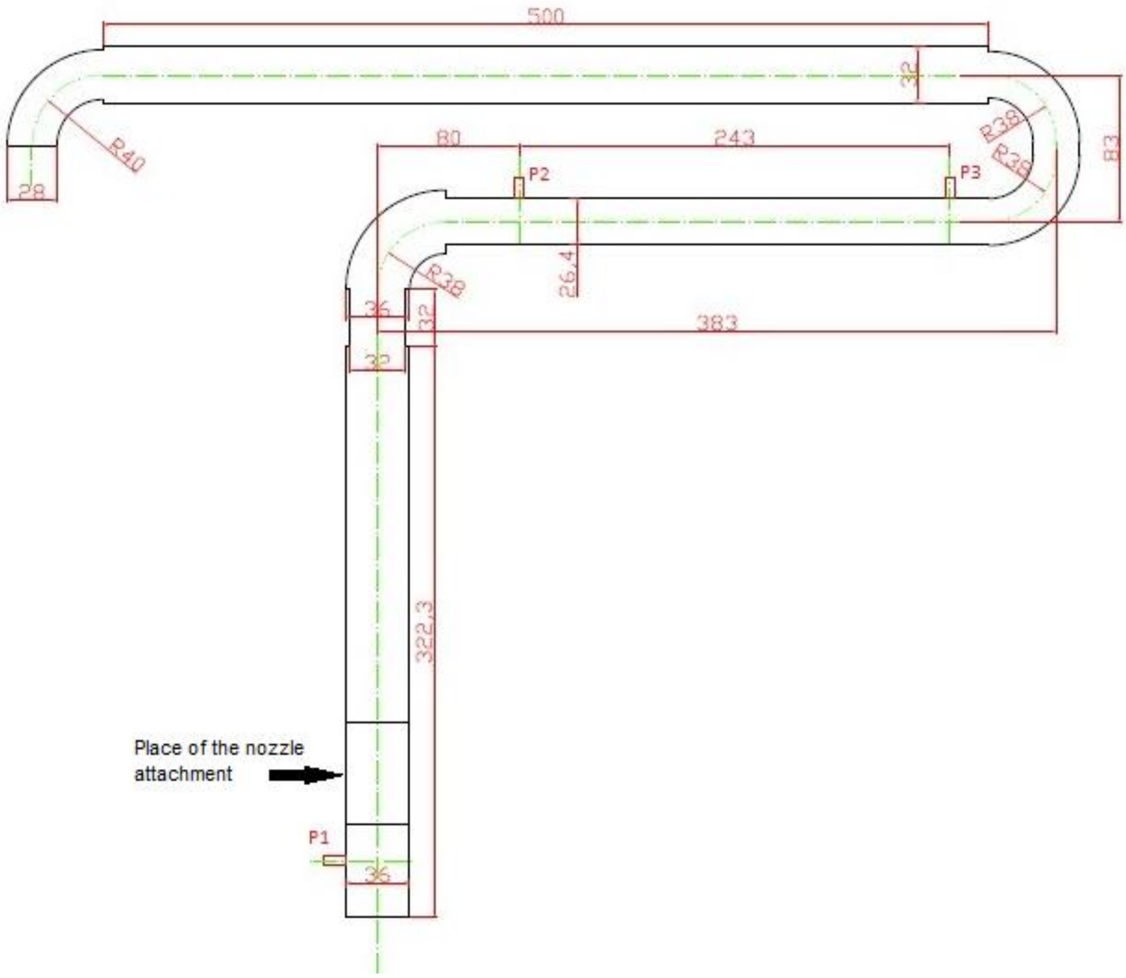

Fig. 4. *Dimensions of the compressor outflow installation where p1,p2 and p3 are dynamic pressure sensors*

All ICP Dynamic Pressure Sensors are made by PCB Piezotronics. The signal from sensors goes through the 4-channel ICP Sensor Line Power unit and to the NI USB-6251 data acquisition module as well as to the computer with LabView data acquisition system as it is shown in the figure 5.

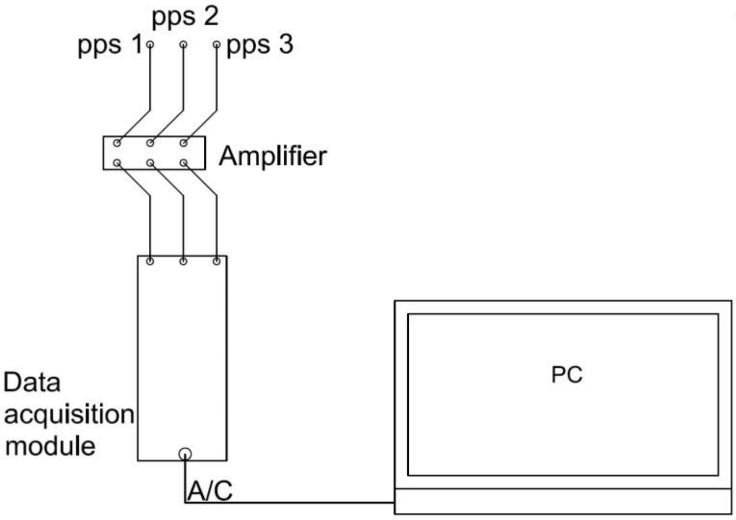

Fig. 5. *Measuring circuit where pps1, pps2 and pps3 are dynamic pressure sensors*

Measurements have been conducted after the equilibrium state of the compressor have been reached. This means that both: temperature and static pressures, have reached constant values. Then parameters: pressure and temperature for various components of the system, power, torque and revolution speed, and ambient conditions (pressure, temperature and humidity) have been recorded. The volume flow rate have been determined using the accurate metering orifice. Most of the measuring equipment is 0.2 accuracy class. Accuracy of the temperature recorders (Czaki EMT50 digital thermometer) is 0.4°C. Accuracy of the Magtrol 3410 torque display is 0.01% for speed measurements and 0.01% of range (+/- 5V) for the torque value. For the static pressure in the pipeline the digital manometer: DigibarII PE350 has been used. The pressure difference on the choking element has been measured by Comarc C9557 class 0.5. Data acquisition module for analogue input with the accuracy class of 0.025 has been used. Detailed measurement system is shown in the figure 6.

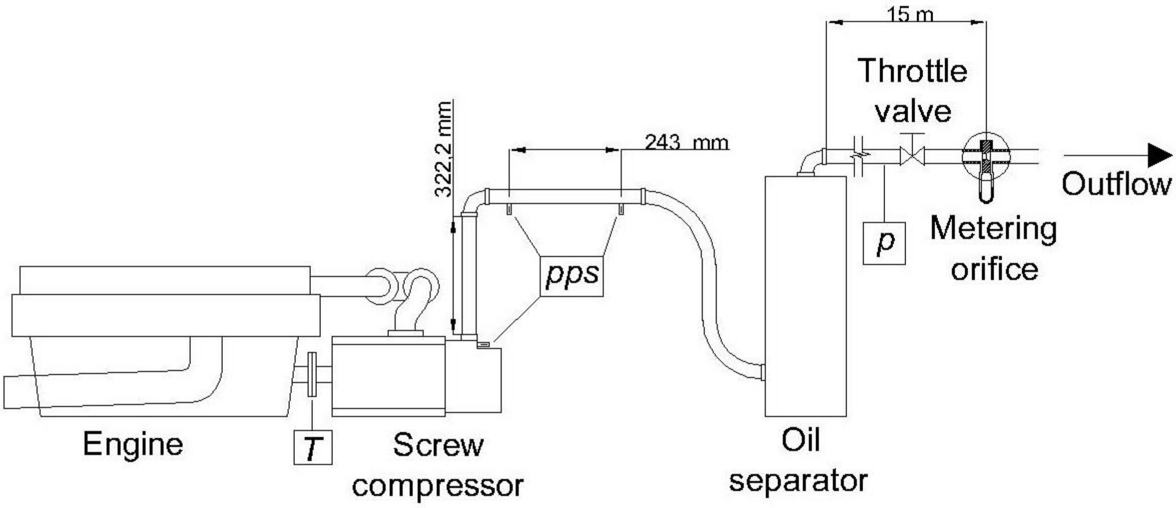

Fig. 6. *Measuring system, where: T- Torque meter on the propeller shaft,*
pps – Pressure pulsations sensors, p- static pressure transducer

4. The results of the experiments

The basic experimental results are shown for two different revolution speed: 1615 and 2100 [rpm]. Comparison graphs for peak to peak characteristics and power to mass flow ratio for different types of damping elements are presented in the next part of the article. These are the most relevant parameters obtained when carrying out research on these elements.

4.1 Experiment results for revolution speed 1615 [rpm]

The author, referring to these rotational speed, was saving measured dynamic pressure signal with sampling rate 27563 Hz. Such frequent sampling was needed to get the 1024 signal samples per one revolution. In the fig. 7. the damping influence on the peak-to-peak pressure pulsations characteristic is shown.

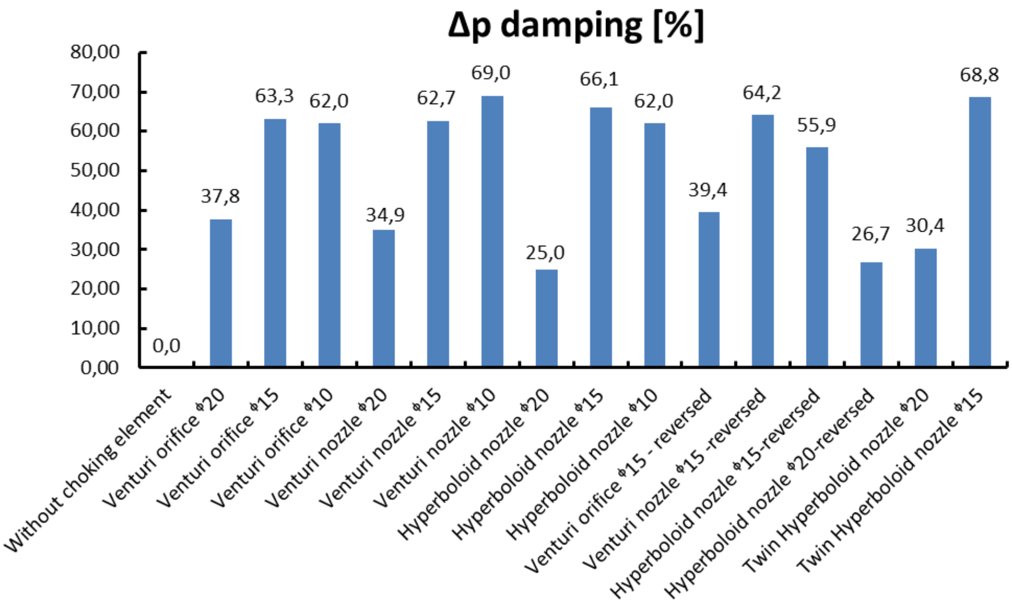

Fig. 7. *Peak-to-peak damping for 1615[rpm] referring to empty pipe*

The influence of the suppressor on the power and mass flow values is also very important on the compressor installation. Such a relation is shown in the fig.8.

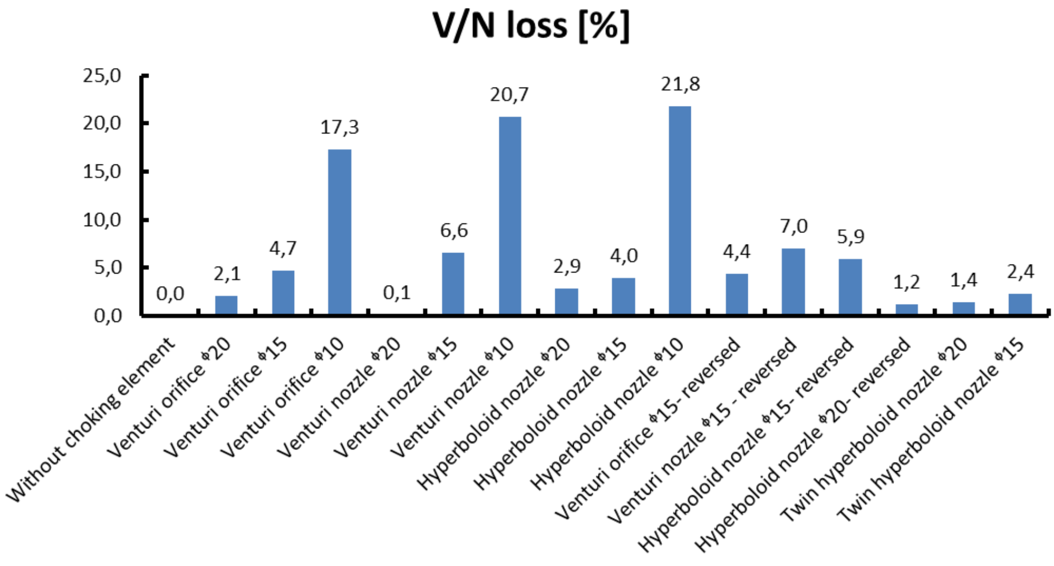

Fig. 8. *Mass flow rate to compression power loss for 1615 [rpm] referring to empty pipe*

The results show that the nozzles with diameter φ10 mm are not acceptable due to a serious loss of volumetric flow rate with the respect to the compressor shaft power (V/N loss). The results are shown in [%] ratio to the non-distributed flow without choking element. After preliminary investigations the author chose four elements with best attenuation coefficient and located it in the reversed direction to the flow. Then the author selected two hyperboloid nozzles with different inner diameter to check the impact on the pressure pulsations by combination of two components.

4.2 Experiment results for revolution speed 2100 [rpm]

To gain desirable 1024 samples per revolution the recording frequency for this case was 35840 Hz. As can be seen in the fig. 9 and 10 for most of the nozzles revolution speed does not have much influence on the behaviour of pressure pulsations damping.

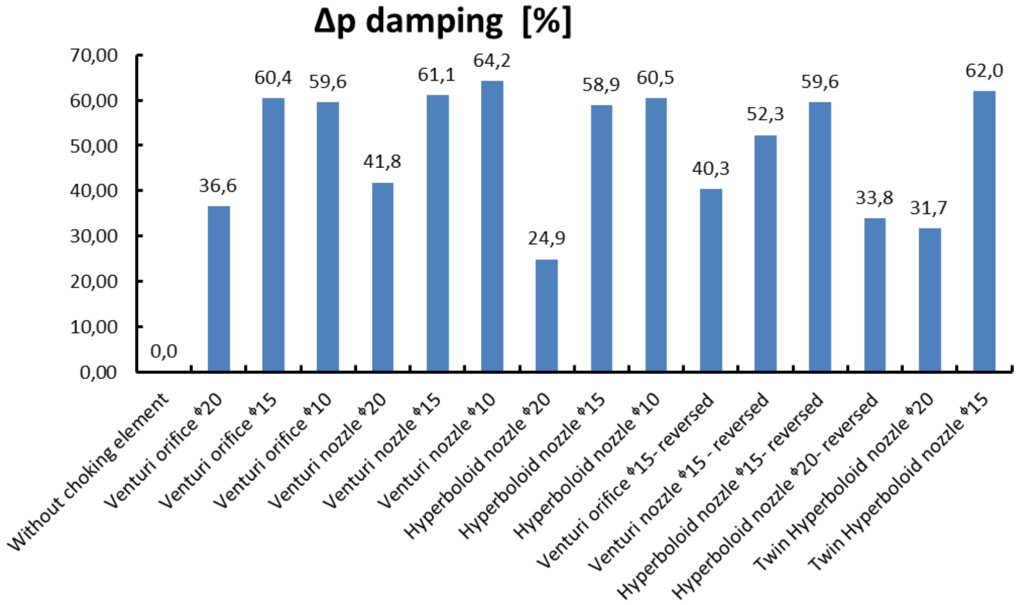

Fig. 9. *Peak-to-peak damping for 2100[rpm] referring to empty pipe*

Analysing fig. 7 and 9, one can see particularly drastic decrease of pressure pulsation damping by reversed Venturi nozzle φ15 from more than 64% to less than 53%. It is also very interesting that damping increases with the reversed hyperboloid nozzle φ20. Such exceptional behaviour in these two cases may suggest that some of the nozzles work better for different rotational speed. The similar relation for V/N characteristic can be seen in the fig 10.

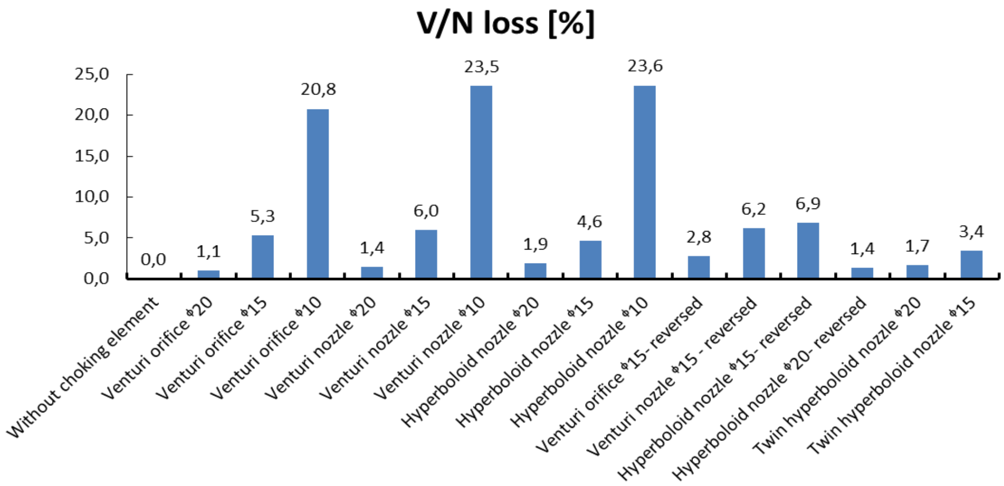

Fig. 10. *Mass flow rate to compression power loss for 2100 [rpm] referring to empty pipe*

4.3 Experiment results for variable revolution speed

Referring to the results of measurements for hyperboloid nozzle φ20 and Ventouri nozzle φ15, which supress pressure pulsations in different ways depending on the revolution speed, the author decided to further investigate four, in his opinion most promising elements. It was two Venturi and two hyperboloid nozzles. It was hard to set on the power unit specific rotational speed, so all of the measurements have been made with the accuracy of 50 revolutions per minute relative to a certain, pre-determined value. This may be considered sufficient for preliminary study like this. The author investigated the entire rotational speed range between 1250 and 2300 [rpm] in order to detect a specific behaviour of the system, however, data was saved only for selected values of rotational speed. The four chosen rotational speed values were 1350, 1800, 1940 and 2270 [rpm]. Pressure pulsations values for different rpm’s are shown in the fig. 11.

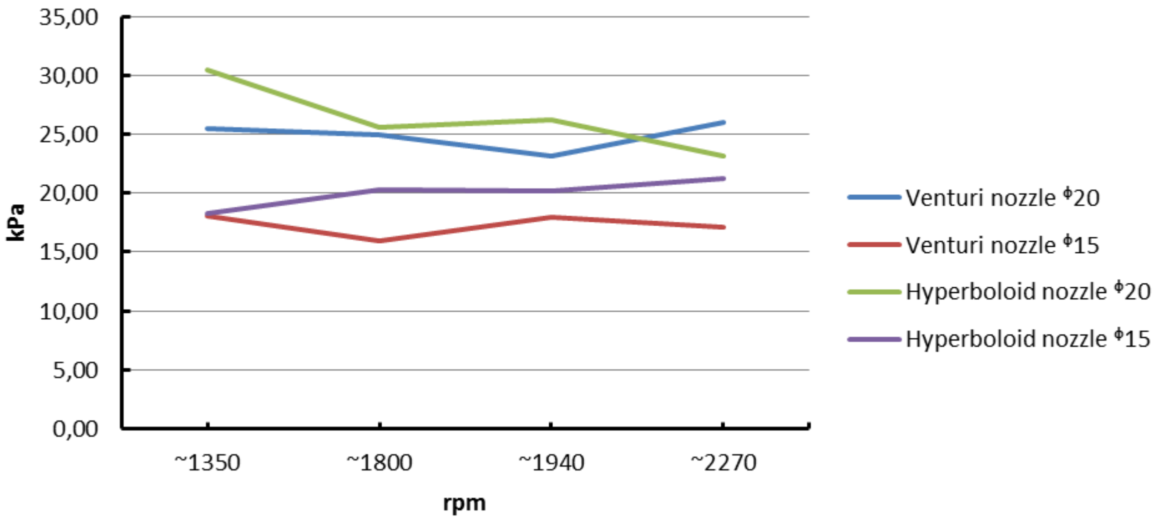

Fig. 11. *Pressure pulsations peak-to-peak values for variable rotational speed*

As can be seen on the graph, the pressure peak-to-peak values are very similar both for minimum and maximum revolution speed in the case of the Venturi nozzles. Obviously this is not a constant value but the differences are very slight. In this case only the results for the nozzle with 20 mm diameter can by quite interesting because of a big difference between the pressure value for the two highest rotational speeds. Experiment results for the hyperboloid nozzles are quite different. There might be seen a clear trend for both hyperboloid nozzles. What is interesting here, is that - the maximum revolution speed pressure pulsation values for both diameters are very similar. Another important phenomenon is the increase of the pressure pulsation attenuation for higher mass flow values (which depends on rotational speed) through the nozzle with 20 mm diameter. The difference between pressure values for the lowest and highest rotational speed is significant. On the other side for the hyperboloid nozzle φ15 mm, the pressure pulsations damping slightly decrease while increasing rotational speed. Certainly this two cases will require more detailed measurements.

5. Conclusions

In the paper the results of the investigation of the influence of different types of nozzle geometry on the pressure pulsation damping have been shown. Generally, the best solutions characterized by a low impact on the compression power and high damping factor are the nozzles with φ15 inner diameter. Selection of the appropriate nozzle shape has to be done with optimisation analysis. However it can be seen that the element composed of two hyperboloid nozzles φ15 stands out from the others. The paper also presents the results of the preliminary investigation on the effects of pulsation attenuation through some elements for variable rotational speed. For such preliminary studies it is hard to draw any specific conclusions and it requires further investigations. A large difference of the pressure pulsations suppression in the hyperboloid nozzle φ20 for 2270 and 1350 [rpm] may suggests that there could be an optimal geometry for each frequency.

References

1. Cyklis P., Symulacja procesów termodynamicznych w sprężarkach wyporowych i ich instalacjach, Kraków 2009.Monografia ISBN 978-83-7242-496-9
2. Cyklis P., Identyfikacja Termodynamiczna Elementów instalacji sprężarek wyporowych. Uogólnienie modelu Helmholtza., Kraków 1999. Monografia PL ISSN 0860-097X.
3. Advanced Reciprocating Compression Technology (Arct) Final Report SwRI® Project No. 18.11052 DOE Award No. DE-FC26-04NT42269 Reporting Period: 1.10.2004 – 30.09.2005.
4. Cyklis P., Młynarczyk P., The influence of different nozzle types on the pressure pulsations in volumetric compressors, Compressors 2013 – 8th international conference on compressors and coolants, Casta Papiernicka, 2.09.2013-04.09.2014.