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ON THE NOISE AND VIBRATION CHARACTERISTICS OF A RECIPROCATING COMPRESSOR WITH TWO DIFFERENT TYPE OF CYLINDER HEADS: CONVENTIONAL HEAD AND A NEW DESIGN HEAD

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Abstract.
Reciprocating compressors commonly used in brake systems of trucks and buses are one of the major sources of the noise and the vibration. In this study, the noise and the vibration characteristics of a single cylinder, reciprocating piston compressor, mainly used in brake systems of heavy-vehicles are investigated and the results are compared for two different constructions of the compressor heads which are referred to the new and the traditional heads in this paper. The main advantage of the new construction is that it is more efficient hence consumes less power during its operation. This is mainly due to the fact that a compressor with a traditional head supplies compressed air continuously, either to an air reservoir or to the athmosphere via a pressure regulator, as long as the vehicle's engine is running. With the new design head, the outlet valve is opened and the same volume of air is compressed and released in every cycle without discharging the compressed air in the cyclinder when there is no demand, i.e. when the air reservoir is full. This feature avoided the unnecessary compressed air circulation and the valve motion, resulting in less noise and vibration during the operation of the compressor. A number of noise and vibration measurements are made at several operating conditions using both types of compressor heads and the results presented in this paper have demonstrated the superior performance of the new desing in terms of noise and vibration levels as well as the power consumption.

INTRODUCTION

Air compressors are the source of energy for brake systems using compressed air. The compressed air is not only used by the brake system but also by some other vehicle components such as pneumatic suspension, door control etc.

A compressor supplies required compressed air to one or more reservoirs. In traditional brake systems, the compressor is directly connected to the vehicle engine and, the air is constantly fed by the compressor as long as the vehicle engine is running. However, all the compressed air cannot be consumed by the system. For instance, in a bus which may have many pneumatic components, compressor needs to work approximately 50 percent of all driving time. For other vehicles, such a truck this rate is about only 20-30 percent. So the most of compressor lifetime passes at free run. It is evident that the energy consumption by the compressor can be reduced significantly if the energy consumption is reduced while the compressed air is not needed. Schematic diagram of the reciprocating compressor used in this study is shown in Fig.1.

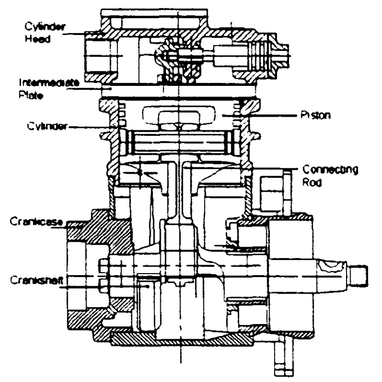


Fig. 1. Schematic diagram of reciprocating compressor.

Several kinds of systems can regulate the supply pressure of the compressed-air brake system which remains within the operating range, i.e. between the cut-in and the cut-off pressures of the pressure regulator. Although there is no need for more compressed air, the compressor have to keep running. Different kinds of systems are designed for decreasing the amount of energy used by compressors when there is no need for compressed air. At first, this was done by some kind of check-valve system on reservoir. These check-valves prevent to exceed the maximum allowable pressure in the reservoir. But at this running condition compressor still consumes energy as under normal operating condition. Another system uses a pressure regulator. The task of the pressure regulator system is to direct the compressed air either to the air reservoir or to the atmosphere. At this running condition air is taken at atmospheric pressure from inlet and is discharged at atmosphere pressure at the outlet. When the reservoir does not require compressed air, this system consumes less energy than the previous one. Also there are other systems which do not use pressure regulators.

The function of a reciprocating compressor cylinder head is to contain pressurised air that builds up inside the cylinder. The head is bolted to the compressor cylinder and is sealed with gasket. Schematic diagram of the new design compressor cylinder head used in this study is shown in Fig.2.

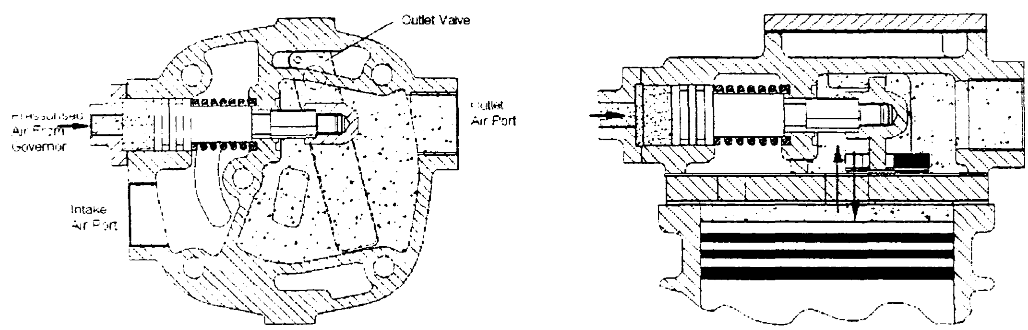


Fig. 2. Schematic diagrams of the new design compressor cylinder head.

Operating principles of the new design can be described as: before the air reservoir there is a non-return valve which lets air flow only one direction to the reservoir. Also there is a component which is called govarnor. When the reservoir reaches the cut-off pressure, the govarnor sends pressurised air from the reservoir to the compressor head. Aid of a mechanism in the head, the outlet valve is opened by pressurised air. While the outlet valve is opened, air between piston and the check valve escapes to inlet section of the head. After the outlet valve is opened, the volumetric efficiency decreases dramatically. Therefore, the compressor compresses the same volume of air in every cycle until the pressure reaches the cut-in pressure in the reservoir. When the pressure decreases to cut-in pressure, the govarnor is stopped to send pressurised air to the head and the outlet valve returns to its original position on the plate. In this working condition compressor demands less energy from the power supplying engine than when working with a pressure regulator. This system is designed especially for reducing energy consumption. The compresor with the new design head in economic operating conditions consumes less energy than the previous one, aproximatelly 73% in the same conditions. However, it also has beneficial effects on reducing the noise and the vibration levels of this type of compressors. The diameter of the piston is about 92 mm and the stroke is 37.6 mm. The sweep volume of the compressor is 252.5 cm^3.

In this study, effects of the new design cylinder head on the noise and the vibration characteristics of the compressor were investigated by performing several measurements. The measurements were carried out on two different cylinder heads: the traditional type head which is used with a pressure regulator and the new design head. In order to compare the effects of two different heads, the sound power and the vibration levels of a compressor at several operating conditions were measured.

EXPERIMENTAL SET-UP

Measurements are made on the compressor itself in an anechoic room which is designed to provide the same operating conditions as brake systems of trucks and buses. The anechoic room is divided into two sections. In the first section there are all the required components for the compressor. These are an AC motor for driving the compressor, a frequency controller in order to control the speed of the compressor, an oil pump, an air reservoir and a coolant water pump and a water reservoir. The compressor is located in another part of the room and it is connected to the AC motor in a way similar to the connections to the trucks and buses.

Acoustic intensity methods offered a number of advantages over traditional measurement methods, and therefore acoustic intensity method is opted in order to determine sound power levels of the compressor. The sound power analysis is based on sound intensity measurements in conjuction with Brüel&Kjær Noise Source Location software type 7681 [1]. The measurement surface on which the measuring points are located is a cubic grid enveloping the compressor in order to measure sound power according to ISO 9614-1 standard [2]. There are 84 points on the surface for measuring sound intensity. Schematic diagram of the experimental set-up and the anechoic room used in this study is shown in Fig.3.

Both vibration and noise measurements are made while the compressor is operated at different pressures of air reservoir and different rotational speeds of crank shaft. Brüel&Kjær dual channel real time frequency analyzer type 2144 is used for these measurements. The measurement of sound intensity and vibration levels are carried out under the steady state operating conditions. Sound power measurements are performed with Brüel&Kjær sound intensity probe type 3548 with a 12mm spacer. The microphones are calibrated by a Brüel&Kjær sound intensity calibrator type 3541.

Vibration measurements are performed with Brüel&Kjær accelerometer type 4371. The accelerometer is located on the cylinder head in the direction along with the piston movement. A computer is also used for the permanent digital data storage.

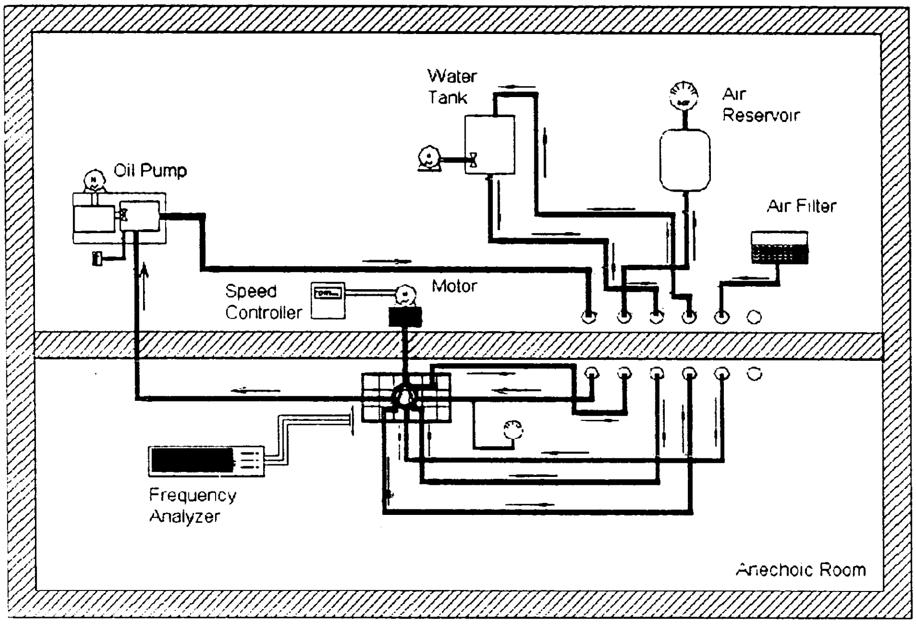


Fig. 3. Schematic diagram of the experimental set-up and the anechoic room.

RESULTS AND DISCUSSION

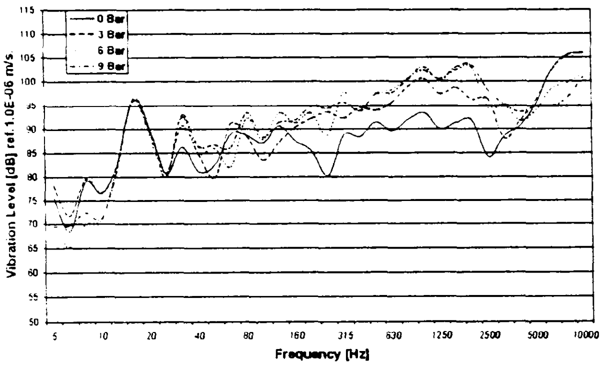
Fig.4. presents comparisons of vibration levels of the compressor with the conventional head at different frequencies. The effects of the pressure of air reservoir on the vibration levels of the compressor, as one of the changing parameter, are shown in this figure. The pressures of air reservoir are selected as 0 bar, 3 bar, 6 bar and 9 bar respectively. And also the effects of the crank shaft rotational speed on the vibration levels of the compressor are shown in this figure separately. The crank shaft rotational speeds are selected as 1000 rpm, 1500 rpm and 2500 rpm respectively.

Measurements presented in Fig.4. are used to establish the main components of the vibration spectrum at two frequecies. These are the fundamental frequency of the vibration related to the unbalance of the slider-crank mechanism at the rotational speed and the second harmonic of the fundamental frequency.

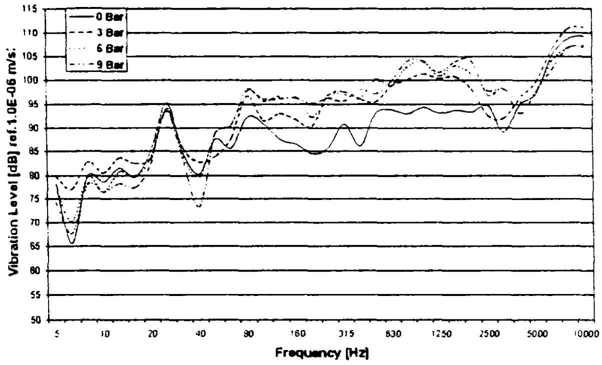
Overall vibration levels are compared for the conventional cylinder head at different pressures of the air reservoir as shown in Fig.6a. to give an idea on the performance of the head. The effects of the crank shaft rotational speed on the overall vibration levels of the compressor are shown in this figure.

Table 1 shows overall vibration levels for both the conventional cylinder head and the new design cylinder head. This gives an idea on the performance of the two different heads in terms of the vibration levels. Results of the vibration measurements made at several operating conditions using both the conventional cylinder head and the new design cylinder head have demonstrated the superior performance of the new design cylinder head in economic operating conditions in terms of overall vibration levels.

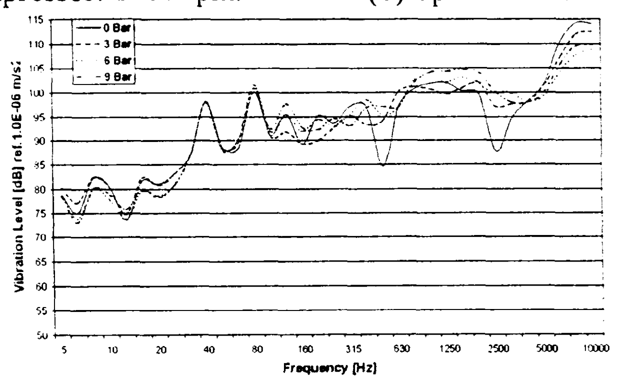
Fig.5. presents comparisons of sound power levels of the compressor with the conventional head at different frequencies.



(a) Speed of the compressor: 1000 rpm.



(b) Speed of the compressor: 1500 rpm.



(c) Speed of the compressor: 2500 rpm.

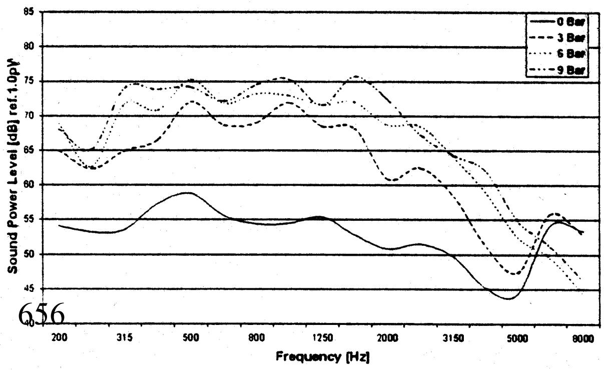
Fig. 4. Vibration levels of the compressor with the conventional head at different frequencies.

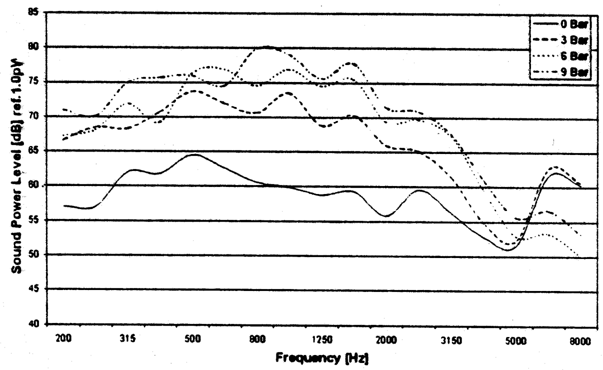
Overall sound power levels are compared for the conventional cylinder head at different pressures of the air reservoir as shown in Fig.6b. illustrate the performance of the head. The effects of the crank shaft rotational speed on the overall sound power levels of the compressor are shown in this figure.

Table 1 shows overall sound power levels for both the conventional cylinder head and the new design cylinder head. This gives an idea on the performance of the two different heads in terms of the noise.

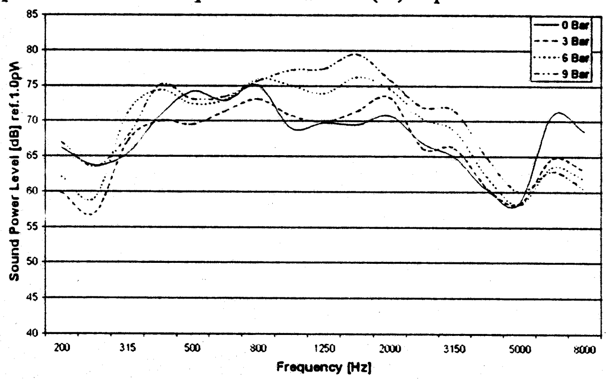
Fig.7. presents the vibration levels of the compressor with the new design cylinder head at different frequencies. This figure shows that the main components of the vibration spectrum are at two frequecies. These are the fundamental frequency of the vibration related to the unbalance of the slider-crank mechanism at the rotational speed and the second harmonic of the fundamental frequency.

Results of the vibration measurements made at several operating conditions using the new design cylinder head have demonstrated the better performance of the new design cylinder head in economic operating conditions than the others especially for the mid- and high-frequencies as shown in Fig.7.





(a) Speed of the compressor: 1000 rpm. (b) Speed of the compressor: 1500 rpm.

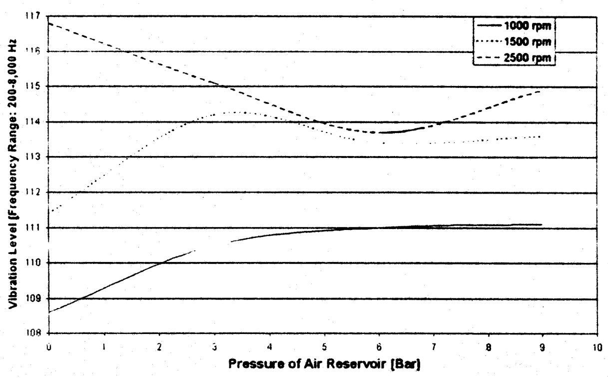


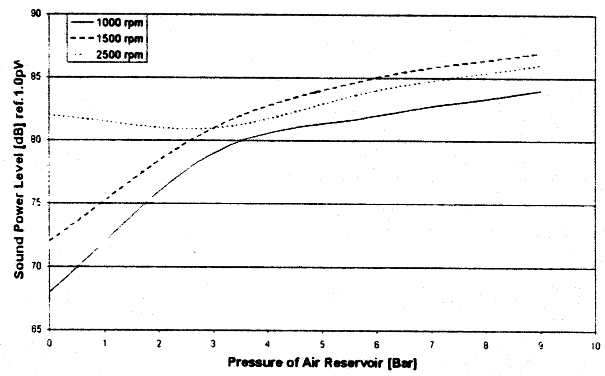
(c) Speed of the compressor: 2500 rpm.

Fig. 5. Noise levels of the compressor with the conventional head at different frequencies.

Fig.8. shows the sound power levels of the compressor with the new design cylinder head at different frequencies.

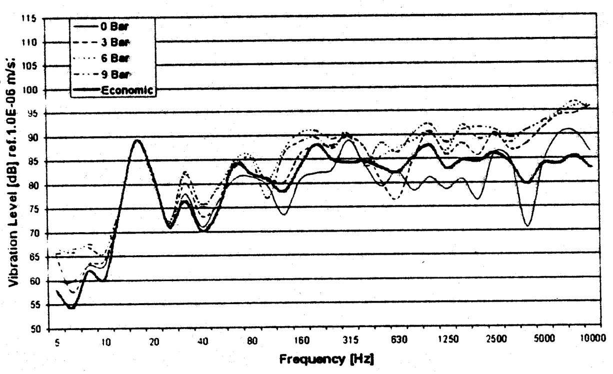
From the previous measurements, there are no major differences between the conventional cylinder head and the new design cylinder head in the normal operating conditions in terms of the vibration and the sound power levels. However, the new design cylinder head in economic operating conditions has good vibration reduction effects on overall levels.



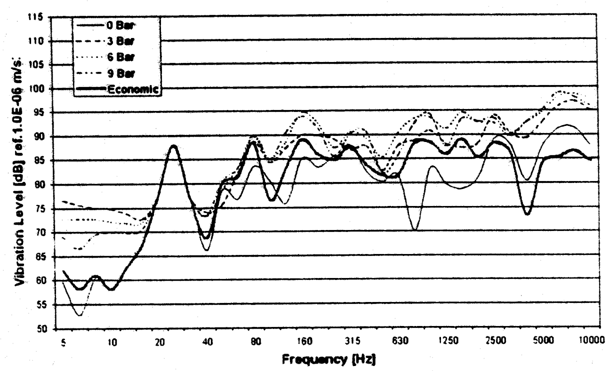


(a) Overall vibration levels. (b) Overall noise levels.

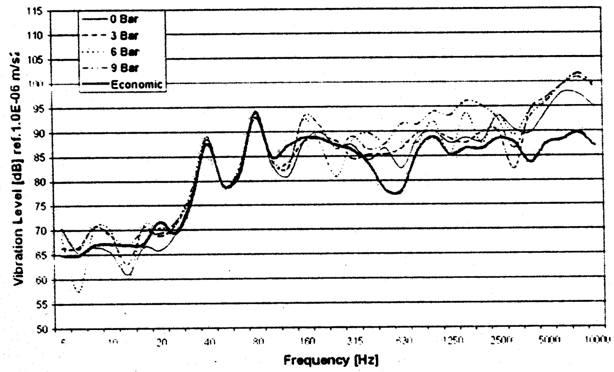
Fig. 6. Comparison of total vibration and noise levels of the compressor with the conventional head at different pressures of air reservoir.



(a) Speed of the compressor: 1000 rpm.



(b) Speed of the compressor: 1500 rpm.



(c) Speed of the compressor: 2500 rpm.

Fig. 7. Vibration levels of the compressor with the new design head at different frequencies.

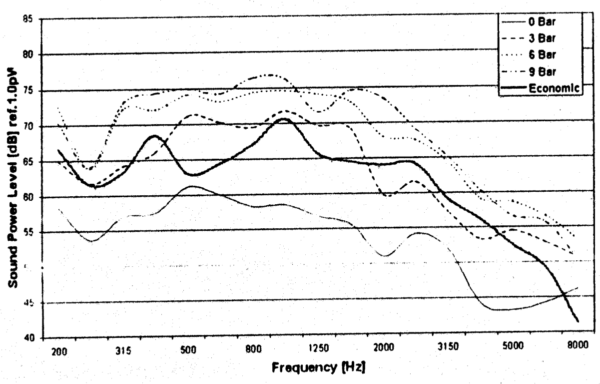
Table 1. Overall vibration and sound power levels for both the conventional type cylinder head and the new design cylinder head.

	Overall vibration Levels [dB] ref.1.0E-06 m/s2			Overall noise Levels [dB] ref.1.0pW		
The pressure of air reservoir	Crank shaft rotational speed			Crank shaft rotational speed [rpm]		
	1000rpm	1500rpm	2500rpm	1000rpm	1500rpm	2500rpm
	For the compressor with the conventional cylinder head					
0 bar	111	114	119	66	70	82
3 bar	112	116	117	79	81	81
6 bar	112	115	115	82	85	84
9 bar	112	115	116	84	87	86
	For the compressor with the new design cylinder head					
0 bar	98	99	105	66	72	82
3 bar	103	104	106	79	81	81
6 bar	104	106	107	82	85	84
9 bar	104	106	108	84	87	86
Economic	97	98	99	76	80	81

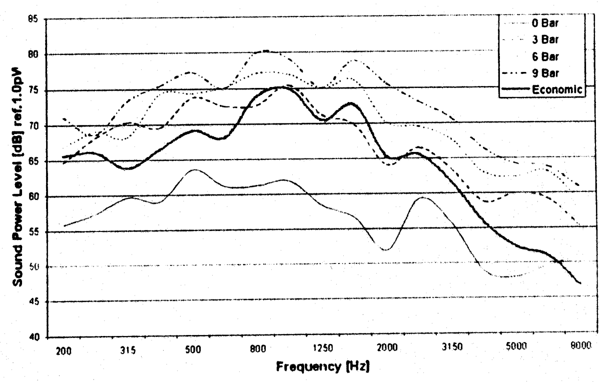
CONCLUSIONS

In this study, effects of the new design cylinder head on the noise and the vibration characteristics of a compressor are investigated by conducting various measurements. The measurements were carried out using two different cylinder heads: the traditional head which is used to with pressure regulator and the new design head. In order to compare the effects of two different types of heads, the sound power and the vibration levels of the compressor at several operating conditions were measured and the results presented in this paper have demonstrated the superior performance of the new desing in terms of the noise and the vibration levels as well as the power consumption. The compresor with the new design head in economic operating conditions consumes less energy than the previous one, aproximatelly 73% in the same conditions. However, the comparison of the measured flow rates of the compressor with different cylinder heads at different discharge pressures reveal that the new design cylinder head has no effect on the volumetric efficiency in normal operating conditions.

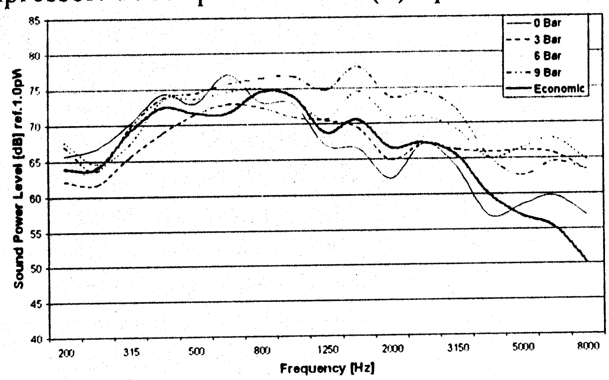
Results obtained in this study will be usefull data for the design of the noise and the vibration reduced compressor for the brake systems.



(a) Speed of the compressor: 1000 rpm.



(b) Speed of the compressor: 1500 rpm.



(c) Speed of the compressor: 2500 rpm.

Fig. 8. Noise levels of the compressor with the new design head at different frequencies.

REFERENCES

[1] Brüel&Kjær Noise Source Location software type 7681.

[2] ISO 9614-1: 1993 Acoustics – Determination of sound power levels of noise sources using sound intensity – Part 1: Measurement at discrete points.