THE NOISE AND VIBRATION CHARACTERISTICS OF A RECIPROCATING COMPRESSOR: EFFECTS OF SIZE AND PROFILE OF DISCHARGE PORT

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Abstract.
Reciprocating compressors commonly used in break systems of trucks and buses are one of the major sources of noise and vibration. At design stage it is very important to find the optimum cross sectional area and the profile for discharge hole on the intermediate plate. In this study, the noise and the vibration characteristics of a single cylinder, reciprocating piston compressor used in break systems of heavy-vehicles are investigated and the results are compared for four different constructions of a compressor discharge port. A number of noise and vibration measurements are made at several operating conditions using four different intermediate plates for the same compressor. The results presented in this paper have demonstrated the effects of the cross sectional area and the profile of discharge hole in terms of the noise and the vibration levels of compressor.

INTRODUCTION

A reciprocating compressor operates on a repeated step cycle process. The piston moves down on its suction stroke causing a flow of gas to enter the cylinder until the piston reaches the end of that stroke and then proceeds on its compression stroke. At this point the flow into that cylinder is stopped and remains so until the next suction stroke commences. On the discharge side, as soon as the internal pressure reaches the level in the discharge port plus the pressure required to open the valve, the valve opens and gas starts to flow immediately from the discarge port. As soon as the piston has reached the end of the compression stroke the valve closes and the gas flow stops. This on-off flow process causes noise and vibration in discharge port and its connections.

Air compressors are the source of energy for break systems using compressed air. The compressed air is not only used by the break system but also by some other vehicle components such as pneumatic suspension, door control etc. The crankshaft of the compressor is directly connected to the vehicle engine through a v-belt or a toothed gear in traditional break systems.

Schematic diagram of the reciprocating compressor used in this study is shown in Fig.1. The speed of compressors used by the break system is dictated by the speed of the vehicle engine. The compressor speeds vary 1000 rpm to 3000 rpm in normal operating conditions. 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.

At design stage it is very important to find the optimum cross sectional area and the profile for discharge hole on the intermediate plate. Schematic diagrams of the conventional intermediate plate and the new design intermediate plates used in this study is shown in Fig.2.

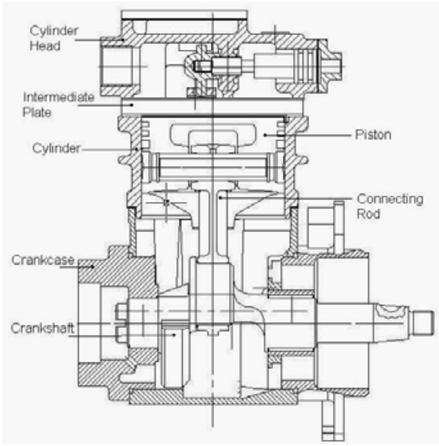


Fig.1. Schematic diagram of the reciprocating compressor.

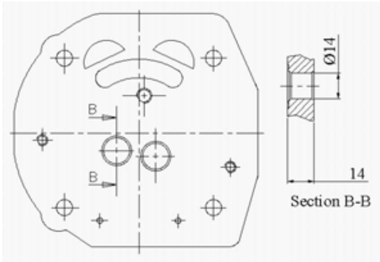
In this study, effects of the form and the size of discharge port on the noise and the vibration characteristics of the compressor were investigated by performing several measurements. The measurements were carried out on four different intermadiate plates on the same compressor. In order to compare the effects of four different intermadiate plates, the sound power and the vibration levels of the compressor at several operating conditions were measured.

EXPERIMENTAL SET UP

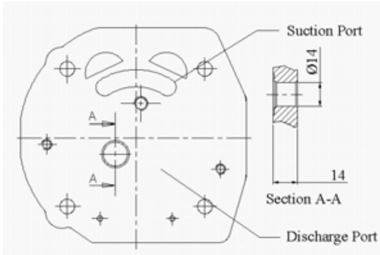
Measurements are made on the compressor itself in an anechoic room which is designed to provide the same operating conditions as break systems of trucks and buses. The anechoic room is divided into two sections. In the first section there are all the required components for the compressor. These are 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.

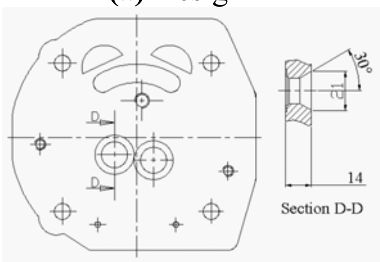
Schematic diagram of the experimental set-up and the anechoic room used in this study is shown in Fig.3. Both the vibration and the noise measurements are made while the compressor is operated at different pressures of air reservoir and different rotational speeds of crank shaft. Brüel&Kjær dual channel real time frequency analyzer type 2144 is used for these measurements.



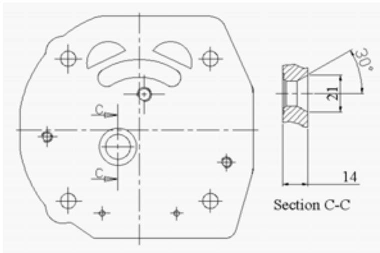
(a) Design A



(b) Design B



(c) Design C



(d) Design D

Fig.2. Schematic diagrams of the conventional and the new design intermediate plates.

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 12 mm spacer. The microphones are calibrated by a Brüel&Kjær sound intensity calibrator type 3541. Vibration measurements are performed with Brüel&Kjær accelerometer type 4371. A computer is also used for the permanent digital data storage. The accelerometer is located on the cylinder head in the direction along with the piston movement.

RESULTS AND DISCUSSION

Fig.4. presents comparisons of the vibration levels using four different intermediate plates for the same compressor at different frequencies. The effects of the pressure of the 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 four different intermediate plates for the same compressor at different pressures of the air reservoir as shown in Table 1. This gives an idea on the performance of four different intermediate plates according to the different pressures of the air reservoir and the different crank shaft rotational speeds.

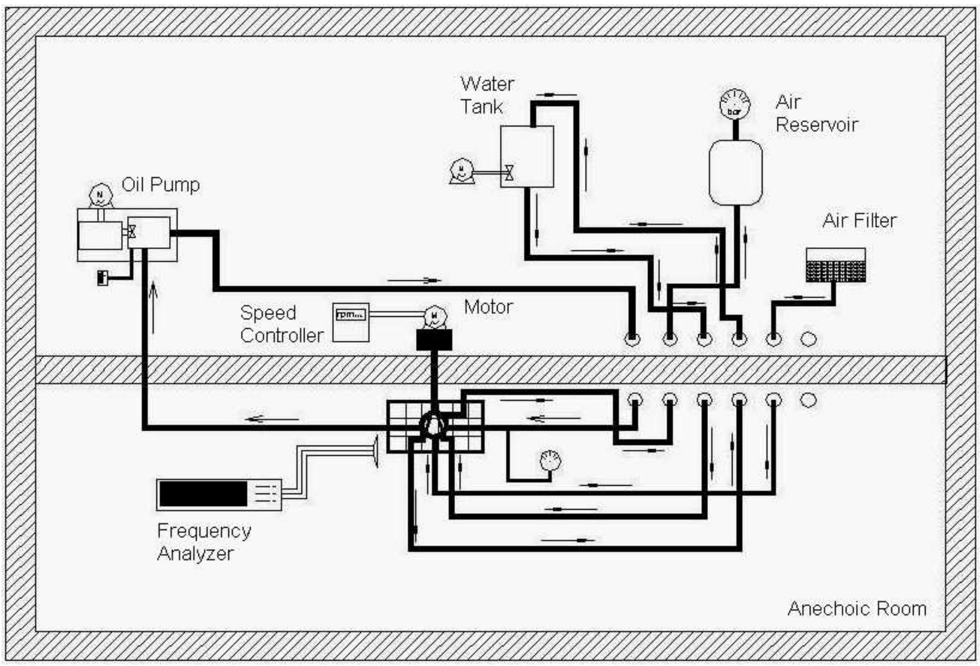


Fig.3. Schematic diagram of the experimental set-up and the anechoic room.

Results of the vibration measurements made at several operating conditions using four different intermediate plates have demonstrated the superior performance of design-D in terms of overall vibration levels.

Fig.5. presents comparisons of the sound power levels using four different intermediate plates for the same compressor at different frequencies. The effects of the pressure of air reservoir on the sound power levels of the compressor, as one of the varying parameter, are shown in this figure. The pressures of the 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 sound power 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.

Overall sound power levels are compared for four different intermediate plates for the same compressor at different pressures of the air reservoir as shown in Table 1. This gives an idea on the performance of four different intermediate plates according to the different pressures of the air reservoir and the different crank shaft rotational speeds.

Results of the sound power measurements made at several operating conditions using four different intermediate plates have demonstrated small differences in terms of overall sound power levels. However, design-D shows better performance than the others in terms of overall sound power levels.

CONCLUSIONS

In this study, effects of the form and the size of discharge port on the noise and the vibration characteristics of the compressor were investigated by performing several measurements. The measurements were carried out on four different intermadiate plates on the same compressor. In order to compare the effects of four different intermadiate plates, 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 desings in terms of the noise and the vibration levels.

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

Table 1. Overall vibration and sound power levels for four different intermediate plates.

	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**
	Intermediate Plate Design A						
0 bar	111	114	119		66	72	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
	Intermediate Plate Design B						
0 bar	96	98	103		66	70	83
3 bar	97	101	103		77	80	81
6 bar	101	104	104		82	85	82
9 bar	102	104	104		83	86	84
	Intermediate Plate Design C						
0 bar	98	102	106		70	73	83
3 bar	102	103	105		79	82	81
6 bar	103	104	106		83	86	86
9 bar	103	106	106		85	88	88
	Intermediate Plate Design D						
0 bar	95	99	102		65	69	81
3 bar	100	101	103		77	79	79
6 bar	101	103	103		81	83	81
9 bar	103	103	103		82	84	84

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.

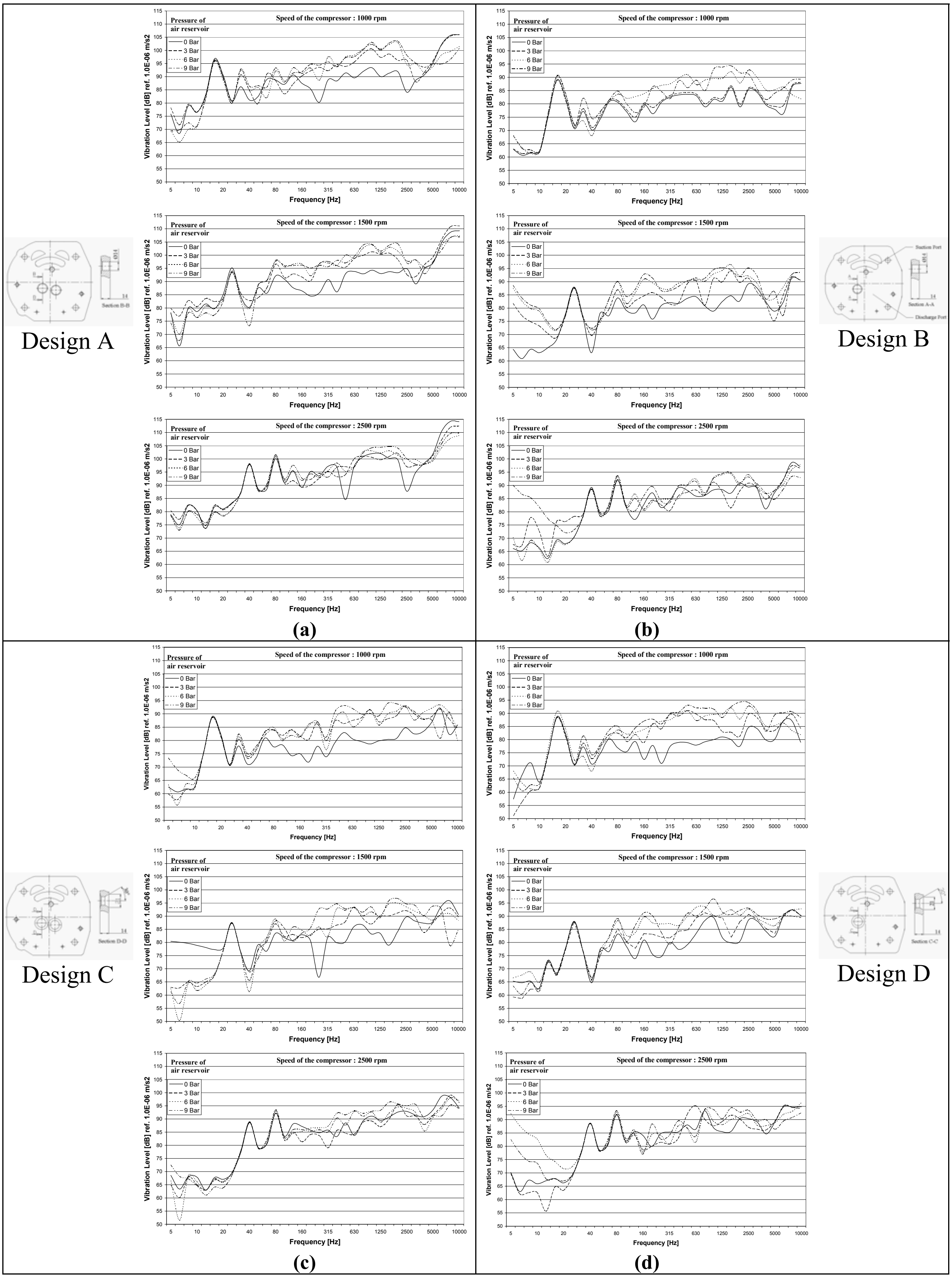


Fig.4. The vibration levels using four different intermediate plates for the same compressor.

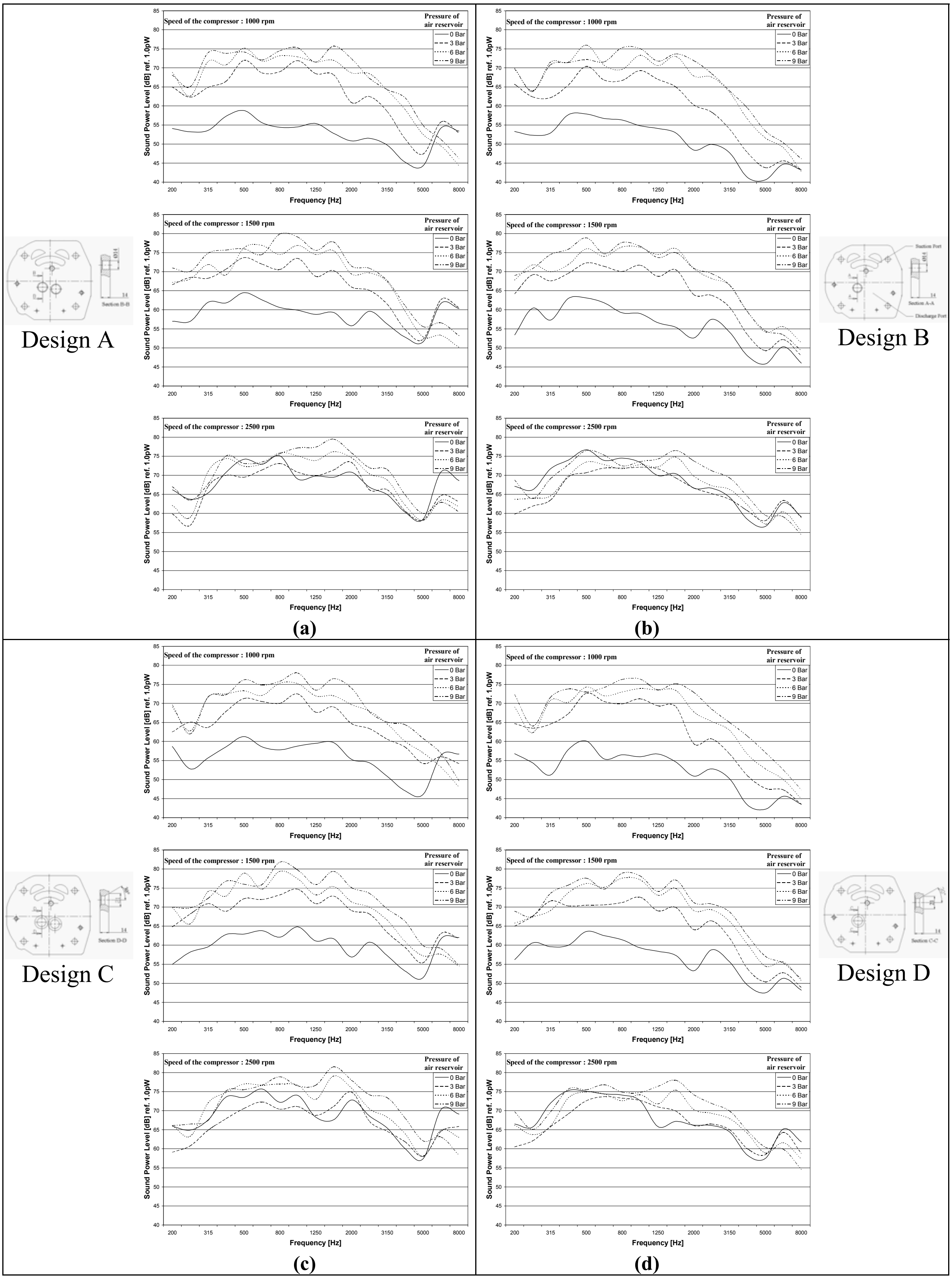


Fig.5. The noise levels using four different intermediate plates for the same compressor.