2008

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ABSTRACT

Screw compressors for bus air conditioning (A/C) have many advantages: low noise, reduced vibration eliminating the need for isolation mounts, a large operating speed range, and quiet unloading. However, sometimes interesting resonance situations can occur. The opportunity for this to happen is largely increased for roof-mounted A/C systems, where the discharge tubing can become quite long, given that the compressor is situated in the engine compartment. This paper discusses different features, such as unloading, volume ratio, oil viscosity, oil flow and bracket design, etc. from sound and vibration perspectives.

1. INTRODUCTION

This particular bus A/C screw compressor consists of a step-up gear (68/17 gear ratio) that drives a male rotor with 5 lobes, which drives a female rotor with 7 lobes. The input shaft has an electric clutch, and the pulley on the shaft is belt driven by the main bus engine. The speed of the compressor varies within a range of about 700 to 3,000 rpm. This compressor has an oil separator and oil sump located on the discharge side as shown in Figure 1. The rotors, the bearings, the gears and the shaft seal are lubricated with oil supplied with pressure differential between the discharge pressure and the lubrication points.

One dominant noise source of the screw compressor is the pressure pulsations coming from discharging the gas from the closed flutes under compression as they open into the discharge cavity. This discharge pulsation is described in the following equation:

$$p(t)=\rho * c\,(Q(t)-\text{Avg. }Q)/S \qquad (1)$$

Where: $p(t)$ = pressure pulsation amplitude in the discharge cavity
ρ = density of the gas,
c = the speed of sound in the gas,
S = the cross section area of the discharge cavity,
$Q(t)$ = the volume flow in the discharge cavity,
Avg. Q = the average volume flow in the discharge cavity,

Besides the pressure pulsation amplitude, the corresponding frequency is of high interest when identifying noise sources.

$$Fp = nm * ZM / 60 \quad (2)$$

Where: Fp = the basic frequency or first harmonics of the pulsations, Hz
nm = the male rotor speed, rpm,
ZM = the number of lobes on the male rotor,

When the compressor has a step-up gear, following equation applies:

$$nm = G * n \quad (3)$$

Where: nm = the male rotor speed, rpm,
G = the gear ratio,
n = the compressor input shaft speed, rpm,

Combining equation (2) and (3) gives:

$$Fp = G*n*ZM / 60 \quad (4)$$

Example: The gear ratio G = 68/17 and the male rotor lobes ZM = 5 give a frequency Fp of 333 Hz when the input speed is 1000 rpm (Fp = (68/17)*1000*5 / 60 = 333 Hz).

The discharge pulsations are affected by the contact between the two rotors. The rotor contact noise is affected by transferred torque between the rotors, rotor contact band defined by the rotor profile, rotor lead, and rotor divide. It is also indirectly affected by oil flow, oil viscosity and proper oil drainage out of the discharge port shown in Figure 2.

The drive gears (see Figure 3) are another noise source.

$$Fg = n * ZG / 60 \quad (5)$$

Where: Fg = the basic frequency or first harmonics of the gear meshing, Hz
n = the compressor input shaft speed, rpm,
ZG = the number of teeth on the drive gear,

Example: The number of teeth on the drive gear ZG = 68 gives a frequency Fg of 1133 Hz when the compressor input shaft speed is 1000 rpm (Fg = 1000*68/ 60 = 1133 Hz).

Noise coming from gas pulsations and gear meshing can be structural transmitted, in this case over the bearings into the housings and into a mounting bracket. From the mounting bracket it can continue into the frame of the bus. The noise can also be transmitted over the discharge fluid (refrigerant gas and oil) from the discharge port and into the oil separator. From there it can continue out into the discharge lines of the A/C system. Capacity control affecting the displacement can also give different sound characteristics. One reason for this is that usually the optimum volume ratio is not obtained during partial load. This particular compressor is arranged with axial capacity lift valves (Sjoholm, 1986).

For the noise evaluation of compressors, numerous research methods have been applied including Experimental Statistical Energy Analysis (Ma and Bolton, 2002). To quickly resolve this noise issue, most of the testing was done in the lab with follow-through on the bus to confirm effectiveness of design changes.

2. TESTS OF THE COMPRESSOR IN THE LABORATORY

The low-floor articulated bus equipped with a screw compressor is shown in Figure 4, and the sketch of the compressor and its installed view in the bus is also displayed in Figure 5. In the beginning, we focused on the correlation that appeared in the spectrum data between the vibration frequency and the noise frequency coming from compressor body. Figure 6 shows very close correlation of the vibration and the noise, in particular from the mounting bracket described in Figure 5. Therefore the first approach was to modify this bracket to avoid the resonance frequency around 1 and 2 kHz. Figure 7 shows the noise improvement from this new modified design at 2000 rpm at a partial loading condition. The compressor noise level was reduced about 8 dBA overall due to a 9 dBA reduction at the 1.25 kHz 1/3 octave band. Other design parameters were also tested for their effects on noise level: oil mass flow rate, oil viscosity, internal check valve design and so on. These various alternative designs are introduced in Figure 8, and the results are displayed in relative noise levels compared to the compressor that was returned from the field due to noise issues.

3. FEEDBACK FROM TESTING IN THE BUS

Based on those different approaches shown in Figure 8, noise tests were performed in a bus at the customer's site. The results are plotted in Figure 9. Although we proposed a couple of new alternative designs to reduce noise level and also confirmed their validity through lab testing, the field test result from the bus operation was not satisfactory. During the comparison between the lab testing and the field test, some different circumstances were noted. For the lab testing, the compressor was run at 1000, 2000, and 3000 rpm, and the sound pressure was also measured at one foot away from the surface of the compressor. For the bus operation, however, the bus engine was run from 600 through 1200 rpm, which is equivalent to compressor speeds of 885 - 1770 rpm, and the sound was measured at the rear passenger seats. We also recognized there was a lack of correlation between noise behaviors on the electrically-driven compressor in the lab and the engine-driven compressor in the bus. Lab tests could not account for torsional fluctuations of the engine and its effects on the discharge pulsations. Changes that showed an average of 4 dBA improvements at any speeds in the lab were not seen in the bus. Recognizing this, other design factors had to be evaluated, especially those that are affected by torque variations into the compressor.

4. CONCLUSIONS

To resolve the sound issue of the bus in the field, a couple of new design parameters were studied:

- A very close correlation between the vibration and the noise from the compressor body was found.
- The improvements that were confirmed in the lab testing were not seen in the bus.
- There was a lack of correlation between noise behavior on the electrically driven compressor in the lab and the engine-driven compressor in the bus.

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ACKNOWLEDGEMENT

The authors wish to thank the management of Ingersoll Rand Climate Control for permission to publish this paper. The authors also acknowledge the assistance to this work by Mr. Peter Lawrence and Mr. Jon DeTuncq, both of Ingersoll Rand Climate Control.

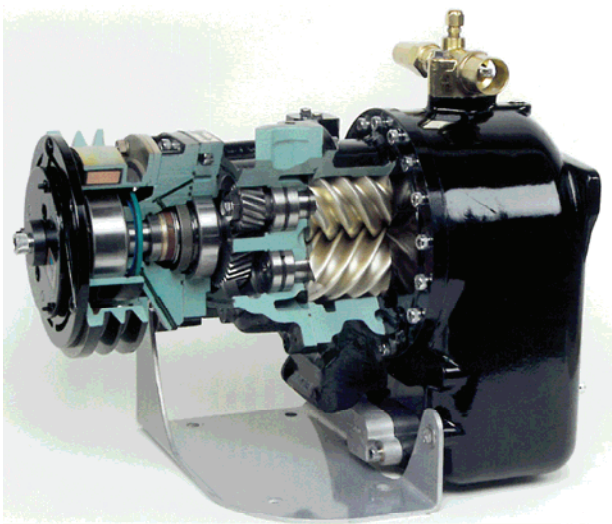

Figure 1: Screw compressor for bus A/C showing (from left to right) clutch, gears, rotors, oil separator and oil sump

Figure 2: The discharge port of a bus A/C screw compressor, the red arrow indicates important area for drainage of oil

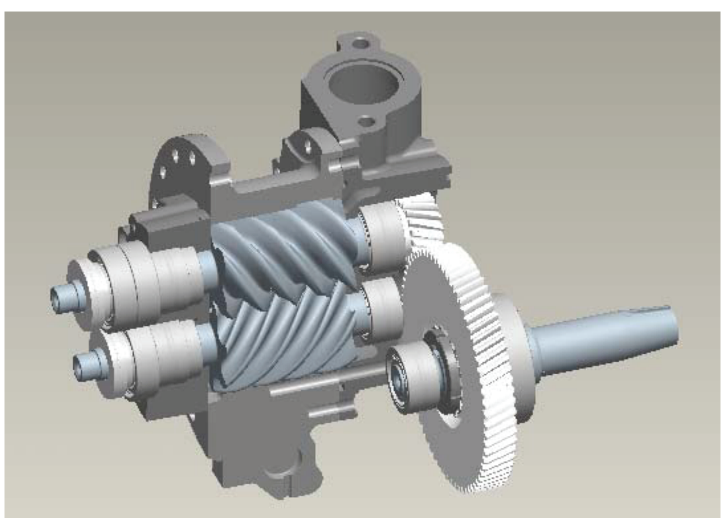

Figure 3: The drive gears are shown on a screw compressor for bus A/C

Figure 4: Low-floor articulated Bus

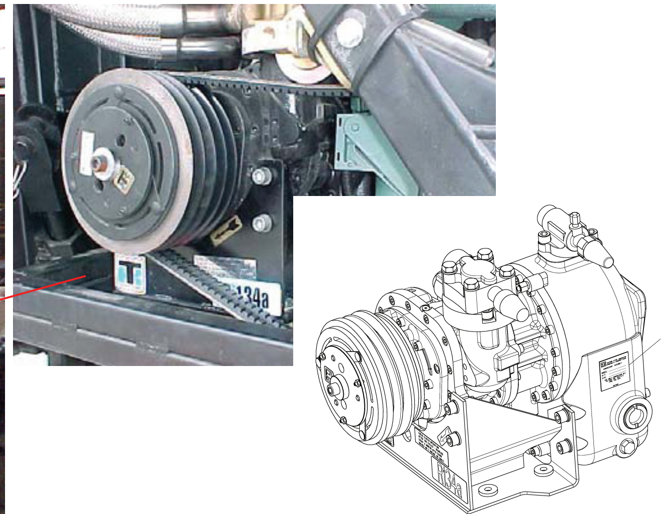

Figure 5: Screw compressor installed with a bracket

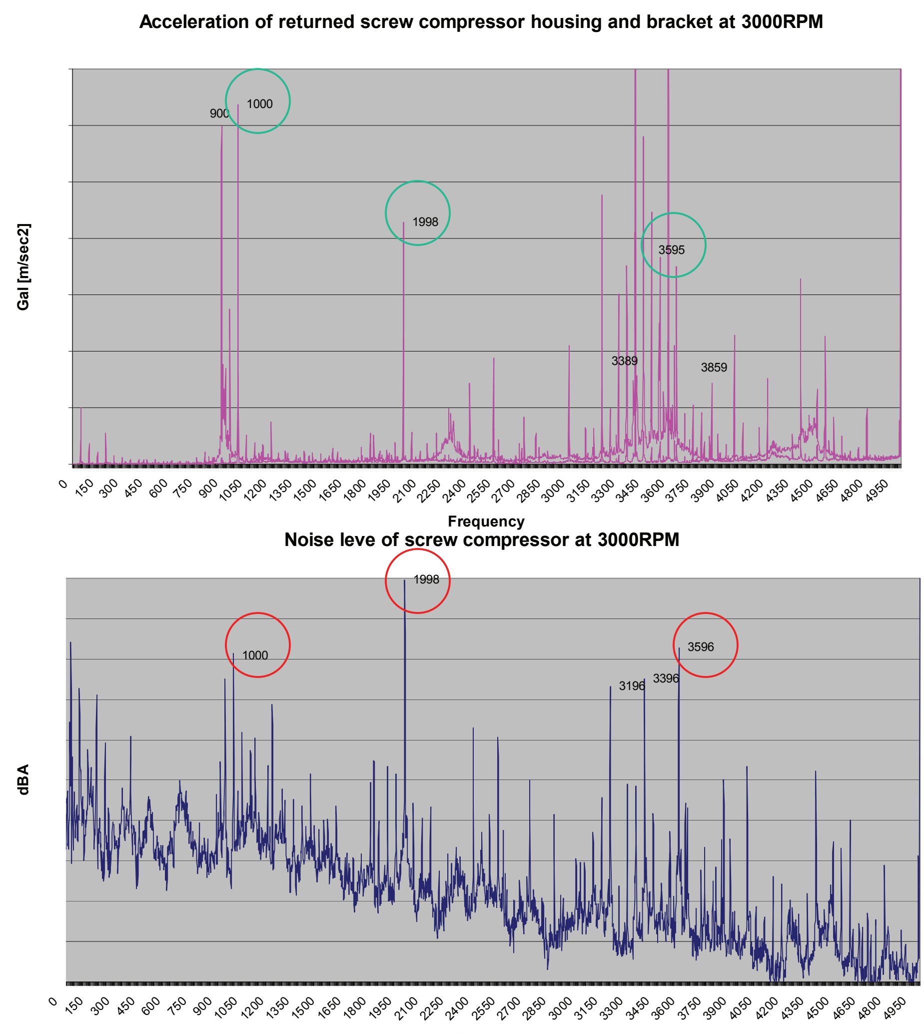

Figure 6: Correlation of the vibration and the noise of returned screw compressor

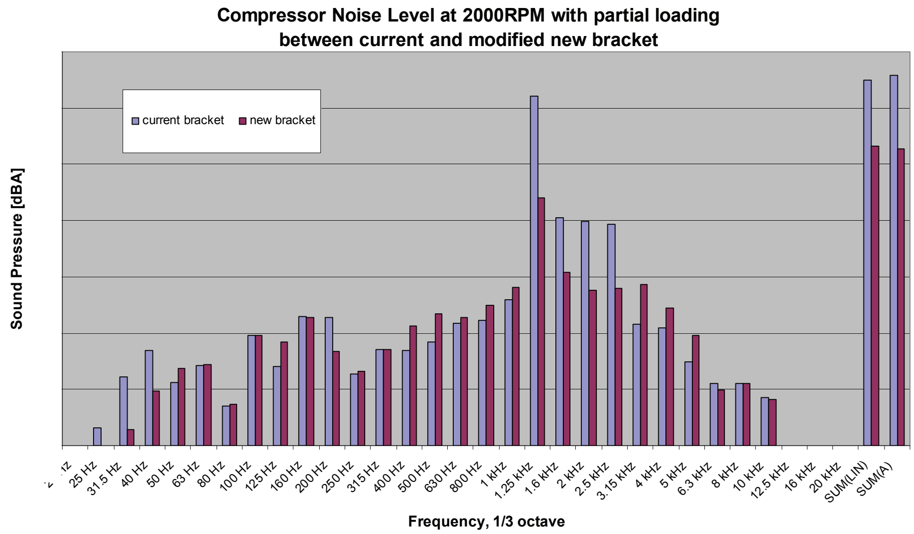

Figure 7: Noise improvement with a modified new bracket at 2000 rpm at a partial loading in the lab testing (delta Y: 5dBA)

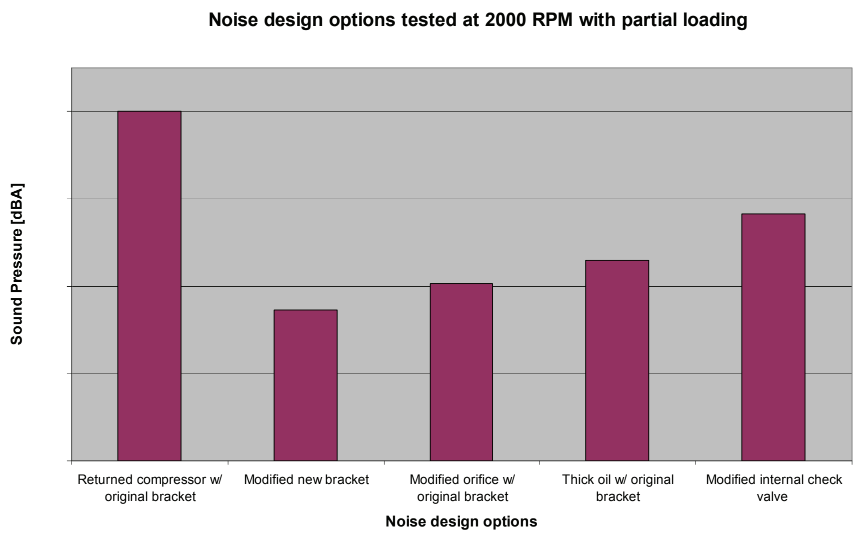

Figure 8: Alternative design changes tested in the lab for the noise reduction (delta Y: 4dBA)

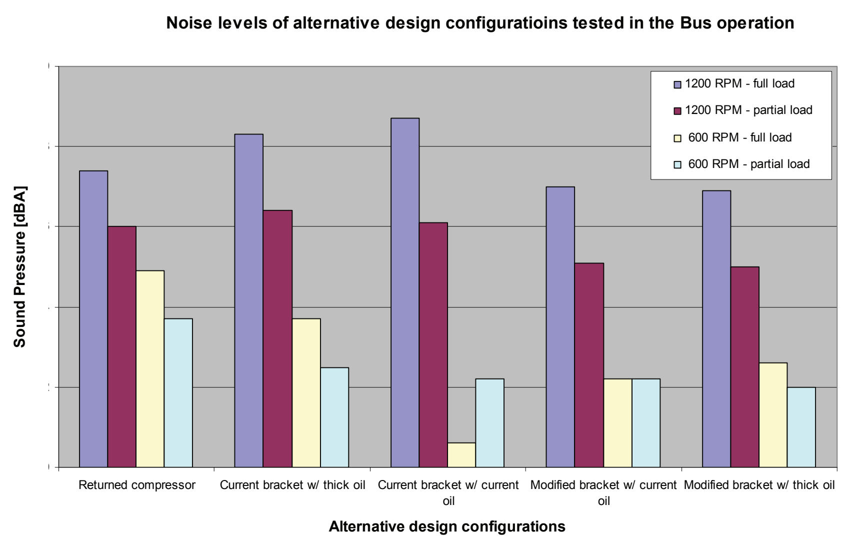

Figure 9: Noise level of various design changes tested in the Bus operation (delta Y: 2dBA)