2008

Noise Reduction in Bus A/C Systems with Screw Compressors Part II

Lars I. Sjoholm
Thermo King

Steve Gleason
Thermo King

Youngchan Ma
Thermo King

Follow this and additional works at: https://docs.lib.purdue.edu/icec

https://docs.lib.purdue.edu/icec/1859

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html
Noise Reduction in Bus A/C Systems with Screw Compressors
Part II: Compressor With System Interaction

Lars Sjoholm1, Steve Gleason2, YoungChan Ma3

1Ingersoll Rand Climate Control Technologies, Minneapolis, Minnesota, USA
Lars_Sjoholm@irco.com, 952-887-3430

2Ingersoll Rand Climate Control Technologies, Minneapolis, Minnesota, USA
Steve_Gleason@irco.com, 952-887-2485

3Ingersoll Rand Climate Control Technologies, Minneapolis, Minnesota, USA
YoungChan_Ma@irco.com, 952-887-2526

ABSTRACT

This paper evaluates screw compressors with different features and the response of bus air conditioning (A/C) systems to those features from a sound and vibration perspective. Due to the relatively high discharge pulsation frequency of the screw compressor, the resonance situation in bus A/C systems can be challenging. Evaluations are for complete bus A/C systems in which the compressor is belt-driven from the bus engine. Results are presented for both steady state operation and the transient conditions encountered with start/stop operations and gear changes during road tests. Data from a common coach bus with one rooftop unit and an articulated transit bus with two rooftop units are presented for systems with noise problems resolved using screw compressors with modified features.

1. INTRODUCTION

Whether we are taking the bus from the airport terminal to the car rental facility, riding the bus across town on our daily commute, or enjoying the scenery from a chartered luxury coach, we all appreciate the comfort that comes from an air conditioned bus. It’s a great change from the days when an opened window offered the only respite from the heat of the day. In most applications, a screw compressor quietly provides excellent capacity to cool a bus on the hottest of days, and few people give a second thought as to what mechanical system has relegated operable windows on buses to school kids.

While all buses of a similar type – i.e. coach, transit, articulated, etc. – tend to look very much alike, after close observation it quickly becomes apparent that they all differ in numerous ways. The underlying bus structure tends to be unique, and there are numerous engine choices and mounting configurations used; these two factors affect where the air conditioning compressor is located and how it is driven, and then the style of the bus helps determine what type of A/C system is installed and where it is positioned on the bus. Rooftop systems requiring long runs of refrigerant tubing from the back of the bus tend to be the applications where noise complaints associated with screw compressors have occasionally occurred.

It was becoming clear that one particular screw compressor application had a noise problem that was not going to be resolved by testing alternate designs in the laboratory. Changes that led to noise improvements in isolated compressor tests typically had no effect in the bus. A modified compressor bracket that showed notable improvement in the lab was not a solution in the field. (Ma, et al., 2008) Very capable field service personnel had already tried rubber isolation under the compressor to further eliminate structure-born noise as the likely culprit. The compressor was not audible over the engine at the back of the bus, which points away from an air-born noise...
problem, so it was becoming apparent that it was a fluid-born noise problem. The noise was transient and was affected by both compressor speed and the loading condition of the compressor. This compressor has axial unloading that activates according to ambient conditions, and the noise was notably different at partial load compared to full load. Figure 1 depicts an overhead view of the rear seating area of the bus with the location of the compressor highlighted in green. The noise was most intense in the seats closest to the compressor, so noise measurements were made in this area as shown by the red dots in Figure 1. Sound pressure maps were generated to help compare the effects as changes were made in the compressor to mitigate the noise. Figure 2 shows four of these sound maps at one particular speed for three different loading conditions as well as the engine-only case when the compressor was not operating. These particular maps show the overall A-weighted sound level, and there is a notable change as the compressor transitions from the fully loaded case. The resulting noise increase was very tonal near 1 kHz, so the level of disturbance is greater than might be assumed from the 2 – 4 dBA increases shown on the maps.

![Figure 1: The rear seating area in the bus with the compressor indicated in green and the microphone locations in red](image1.png)

![Figure 2: Baseline sound maps in the bus for different loading conditions at the compressor](image2.png)
2. NOISE MITIGATION ATTEMPTS ON THE BUS

Previous experience in a somewhat similar application had shown there was a potential for the screw compressor’s internal check valve to affect the noise from the compressor, so one of the first modifications was to install a modified check valve. The particular modification that was tried had the unfortunate effect of exciting the discharge line not only in the rear corner of the bus but all the way to the front unit on the long, articulated bus as well. While that was certainly not the desired result, it did show the nature of the problem was sensitive to changes we could make in the compressor. The challenge was to sort out the most effective changes that could be made and quickly implemented to resolve the noise problem.

Reducing the discharge pulse from the compressor would minimize the fluid-born noise problem. Stabilizing the rotors was identified as one way to accomplish this within the design parameters. A set of reversed-thrust bearings shown in Figure 3 was installed in the compressor after the baseline check-valve was reinstalled. Tests of that setup showed a much more favorable result as seen in Figure 4; the tonal noise from the screw compressor was no longer apparent at any steady-state condition that we could test. The bus was taken for a test drive in this condition, which proved to be quiet until the bus reached high speeds on the expressway or the engine sped up quickly during gear changes. It turned out that an electronic governor had limited the idle speed of the bus when it was in stationary, so the reversed-thrust bearings were only a solution up to a certain speed range.

Figure 3: Arrows show the forces on the rotors from the reversed-thrust bearings

Figure 4: Sound maps in the bus after the reversed-thrust bearings were installed

International Compressor Engineering Conference at Purdue, July 14-17, 2008
Additional changes were necessary to account for the greater dynamic forces at high speeds and conditions with rapid speed transitions, and modifying the drive gears in the compressor proved to be a good way to do that. From our trials with reversed-thrust bearings, we know that a stabilizing force on the rotors had a positive effect. The blue arrows in Figure 5 show the axial gas force on the rotors. The red arrows show the axial forces from the original gears on the male rotor and the drive shaft. The axial force from the gear on the male rotor is acting in the opposite direction of the gas force on the male rotor. This is typically done to balance the axial gas force on the male rotor and give the thrust bearings longer life. The green arrows show the axial forces from the reversed helix gears acting on the male rotor and the drive shaft. Notice that the blue and green arrows in Figure 5 are pointing in the same direction on the male rotor, making it very stable; however, this can reduce the life of the thrust bearings. These particular bearings were designed to handle this additional load. The reversed-thrust bearing on the male rotor was no longer needed because of the force from the reversed helix gears; however, the reversed-thrust bearing on the female rotor remained in the compressor.

Figure 5: Compressor with reversed helix gears. Blue arrows show axial gas forces on the rotors, green arrows show axial forces from the reversed helix gears and red arrows show axial forces from the original gears.

Figure 6 shows a contour plot of noise measured in the rear seat of the bus before and after the reversed helix gears were implemented along with the reversed-thrust bearing. The bus in this condition no longer had a compressor noise issue. Figure 7 shows a contour plot of a speed sweep from low idle to high engine speeds and back to low idle both before and after these noise mitigation steps were implemented on a coach bus. Note how the noise tends to intensify near 1 kHz and 2 kHz as the compressor went through a particular speed range in the baseline condition, but this noise was eliminated once the reversed-thrust bearing and reversed helix drive gears were implemented.
Figure 6: Steady-state noise contour plot for the baseline compressor in this application (top) and the same test after the compressor with the reversed-thrust bearing and reversed helix gears was installed.
3. CONCLUSIONS

- With the modifications to the screw compressor to stabilize the rotors, noise from the compressor is no longer audible in the bus through the operating speed range.
- Compressor noise after the modifications is insensitive to the loading of the compressor.

REFERENCES

Ma, Y., Sjoholm, L., Gleason, S., 2008, Noise Reduction in Bus A/C Systems with Screw Compressors, Part I: Compressor Evaluation, *International Compressor Engineering Conference*, Purdue University, IN, USA

ACKNOWLEDGEMENT

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