Source identification and noise reduction of a compressor at Grasso

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Introduction

During the development of the Grasso V compressor a prototype was created consisting of bended plates welded together, as shown in Figure 1. The new assembly is straight forward, allowing a more efficient production as compared to other compressors. Besides, separated cylinder heads have advantages in terms of thermo-dynamic behavior. However, it turned out that the steel plated compressor produced an unwanted increase in noise compared to its predecessor, the Grasso 412 compressor, which used a pipe welded cylinder head.

The aim of the work described in this paper was an acoustic analysis and redesign to restore former noise levels. Design directions were based on in-situ measurements and laboratory predictive modeling.

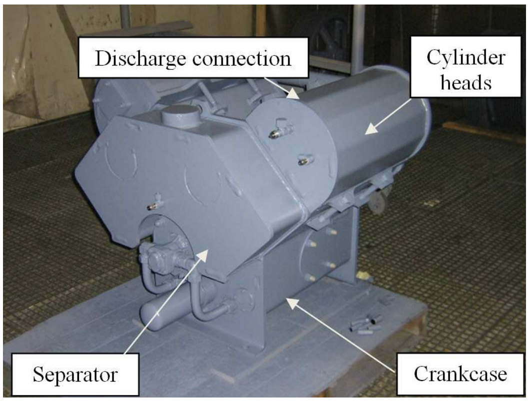

Figure 1: Prototype Grasso V700.

Noise Analysis

Compressor harmonics

The compressor noise was dominated by the rotational frequency and its harmonics (see Figure 2). Shifting operational speed in order to avoid structural modes is obviously not an option.

For a compressor operating at 1200 rpm, the fundamental frequency of the noise spectrum corresponds to 1200/60=20 Hz. Naturally harmonics of this fundamental frequency will be present, partly due to the fact that the compressor consists of a number of cylinders (four in this case) but also because of the fact that the compression in the cylinder is far from smooth, causing an abrupt increase of the pressure inside the cylinder. Moreover, valve noise also attributes to the noise spectrum, which contains many harmonics because of its impulse repetitive character. Excitation due to discrete harmonics is typical in many structural acoustic

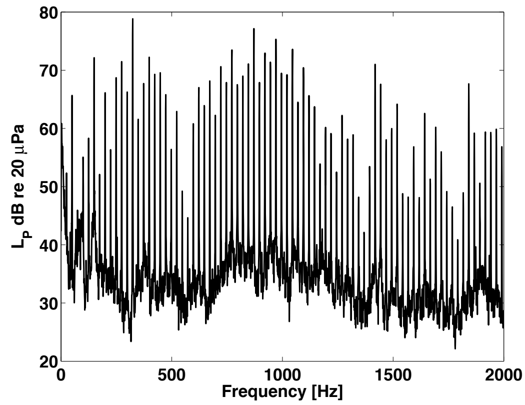

Figure 2: Typical sound pressure level, compressor operating at 1200 rpm.

cases.

Though the acoustic noise consists only of discrete frequencies, the dynamics of the structural components of the compressor play an important role. Resonant behavior of structural components of the compressor may increase the structural response. However, the presence of structural resonances is now more obscured compared to cases in which a mechanical system is excited by a more broadband spectrum.

The objective is to identify the strongest noise contributors, which most probably will be found in the resonant parts of the machine. Obviously the new cylinder plating forms a likely candidate. But before ranking structural components we take a look at the overall noise spectrum and rank the various A-weighted frequency bands. The radiated sound power is presented in Figure 3, comparing the Grasso V700 prototype with its predecessor, the Grasso 412 compressor, showing the unwanted increase of noise.

Driving Point Measurements

To obtain some information on possible resonant behavior of the sound radiating cylinder head, a driving point measurement was performed. By means of a hammer the structure is excited, and the structural response is measured by means of an accelerometer, as illustrated in Figure 4. This time the machine is not in operation and the cylinder heads have therefore an ambient temperature. The ratio of the structural velocity v and the force F exerted by the hammer is calculated and presented as the structural mobility $Y = F/v$. A high mobility denotes a "flexible" structure, whereas a low mobility denotes a "stiff" structure. A high mobility Y may indicate contributing radiators. Additionally the

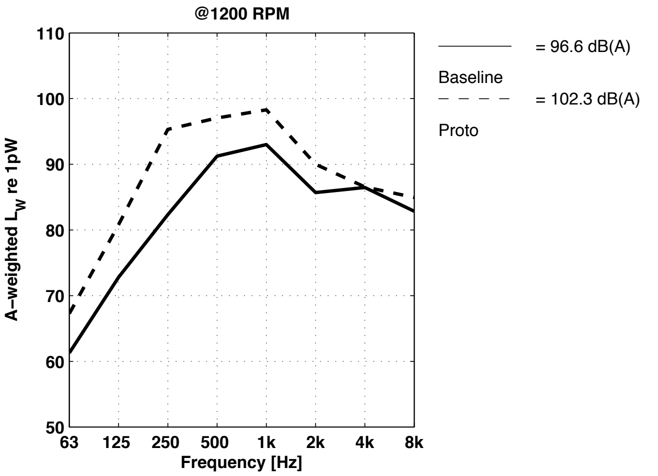

Figure 3: A-weighted Soundpower Spectrum of Grasso 412 (baseline) and prototype Grasso V700 (proto).

modal behavior can be analysed.

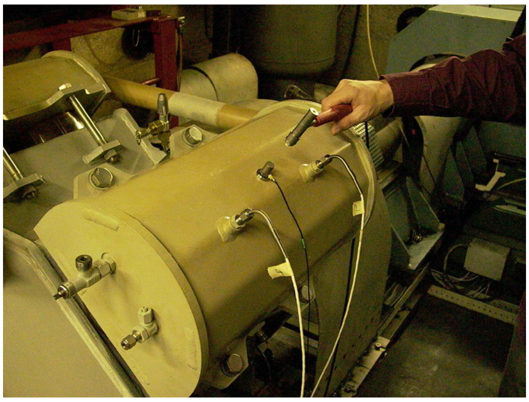

Figure 4: Cylinder head driving point measurement set-up.

The driving point measurements were performed using a hammer, which was instrumented with a Aptech force transducer, type AU-01. A Bruel&Kjaer accelerometer type 4383 was placed nearby the point of excitation to measure the structural response due to the hammer excitation.

Figure 5 shows the measured structural mobility of a number of plate-like parts of the compressor, including the cylinder heads, the crankcase and the separator (see Figure 1). Knowing that the compressor is acoustically most dominant in the 250 up to 1000 Hz 1/3rd octave frequency bands, as shown in Figure 3, the cylinder heads seem to be the most likely contributors, as theirs mobility is highest of all in the frequency range from 200-1200 Hz.

Modal Analysis Simulations

The structural dynamics of the cylinder heads were simulated with the finite element method. The model was able to predict most of the experimentally obtained normal modes (not shown), of which one mode is shown in Figure 6.

Noise Reduction

To reduce the noise of the steel plated cylinder head it was decided to redesign the head, using ductile cast

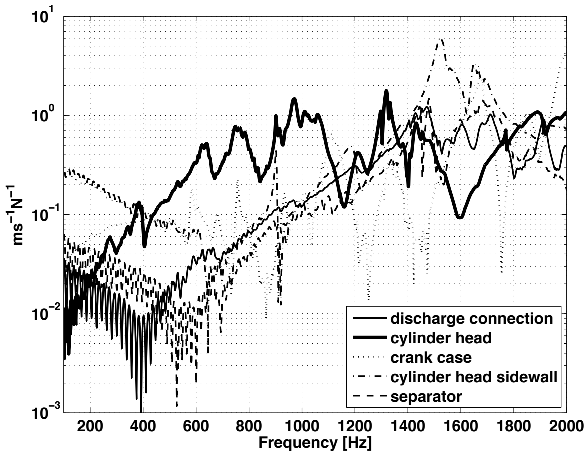

Figure 5: Structural mobility measurements taken on a number of plate-like structural parts of the compressor.

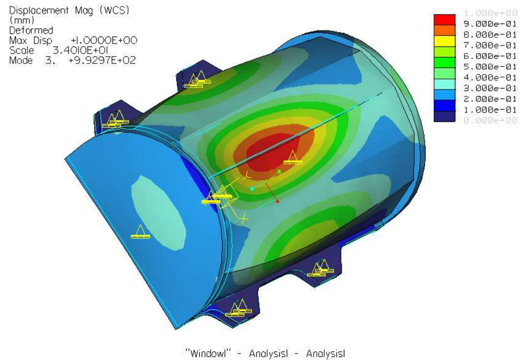

Figure 6: Predicted natural mode at approximately 1000 Hz, Grasso V700 prototype

iron. Ductile cast iron has inherently much more material damping, which should dampen the resonant behavior, which was one of the main problems in the prototype. Furthermore, it was decided to construct the cylinder head more compact, thus decreasing the mobility of the head structure. Supported by finite element simulations the cylinder heads were redesigned.

Finite Element Analysis Redesigned Cylinder Heads

Due to the more compact design of the cylinder heads, as shown in Figure 7 the resonance frequencies were shifted upwards. The finite element model predicted the mode as shown in Figure 6 to be shifted up from 1000 Hz to 1320 Hz, as shown in Figure 8. As the excitation spectrum lowers with increasing frequency, this will reduce the acoustic noise production.

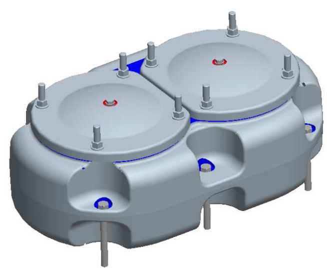

Figure 7: Solid model of the ductile cast iron cylinder head

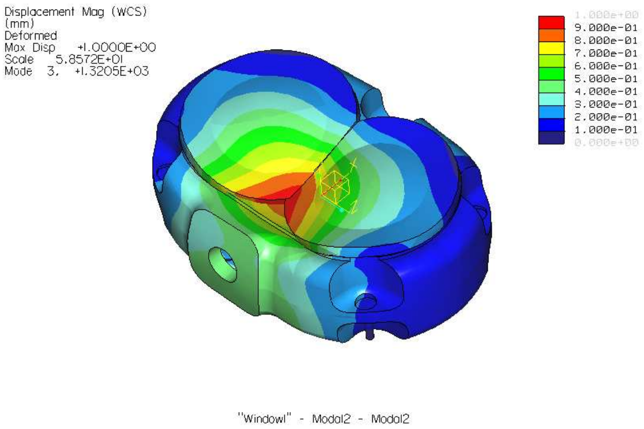

Figure 8: Ductile cast iron cylinder head predicted normal mode at 1320 Hz.

Verification Driving Point Measurements On Improved Prototype

Measurements were performed on the compressor with a redesigned cylinder head, Grasso V700 pre-production. Driving point measurements were conducted which showed that the point mobility was decreased significantly, as expected. Figure 9 shows the test set-up of the mobility measurements on the cast iron cylinder head. The results of these measurements are shown in Figure 10. As compared to the prototype cylinder head, shown with a dotted line, the mobility is reduced almost a factor 10 (20 dB) at frequencies around 1000 Hz. Moreover, for frequencies above 800 Hz resonant behavior is completely absent. However, between 400 and 700 Hz there seems to be some resonant behavior of the redesigned cylinder head, which is probably the cause of the somewhat more low-frequency dominance of the spectrum as will be seen in the next section.

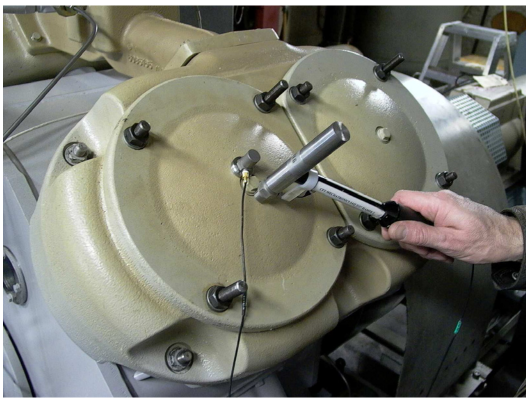

Figure 9: Ductile cast iron cylinder head driving point measurement set-up.

Validation Intensity Measurements

Sound intensity measurements according to ISO 9614-1 were again performed on the new prototype, determining the sound power of the source by means of a surface integral across the measurement surface of the normal component of the sound intensity vector. To improve the quality of the sound intensity measurements (i.e. to reduce the PI index) the reverberation of the test cell was reduced by means of 10 cm thick foam walls during the

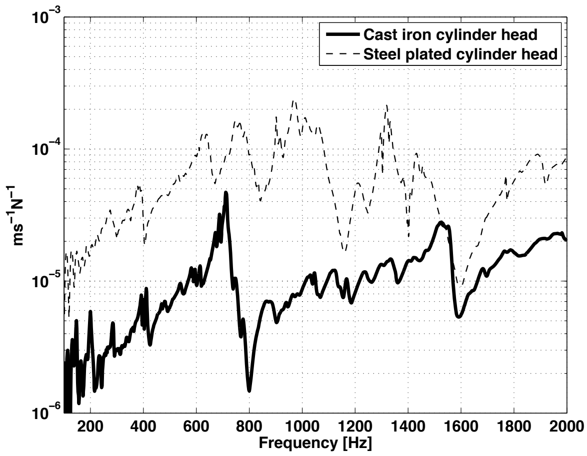

Figure 10: Driving point mobility prototype and redesigned cylinder head.

verification measurements.

The measured sound power radiated by the cast iron pre-production as well as the prototype and its predecessor, the Grasso 12 compressor, are presented in Figure 11. The 1000 Hz 1/1 octave band sound power levels are reduced by 7 dB, resulting in an overall reduction of the A-weighted sound power level in the order of 5 dB(A) as compared to the steel plated cylinder head V700 prototype. As compared to the predecessor, the Grasso 12 compressor, the cast iron cylinder head V700 pre-production emits an insignificant 1 dB(A) more sound power.

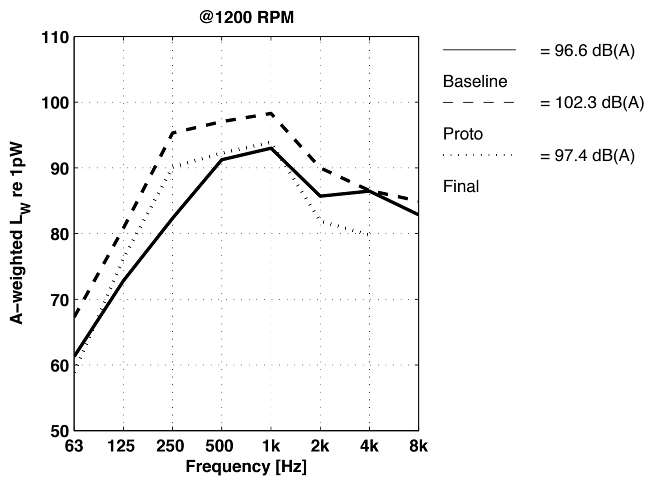

Figure 11: Compressor sound power level of Grasso 412 (baseline), prototype Grasso V700 (proto) and pre-production Grasso V700 (final).

Conclusions

1. The unwanted noise increase of 7dB due to a new plated cylinder head design was successfully counteracted by means of a proper use of noise control engineering insights and analysis tools.
2. From a perceptual point of view, probably because of shifting frequencies, customers characterize the sound itself as "solid".