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A Large Modern High Speed Reciprocating Compressor

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ABSTRACT

The Grasso RC12, introduced early 1996, belongs to the largest reciprocating compressors in industrial refrigeration. The high-speed variant, the RC12E, has been introduced in Europe by the end of 1997: the ammonia variant runs at 1500 rpm, whilst the halocarbon variant has a maximum speed of 1200 rpm. The design philosophy will be illustrated focusing on the valve design. The compressor housing and the piston assembly design methodology will be presented briefly.

INTRODUCTION

The RC12 compressor is the successor of the well-known RC11 range and is the latest development of Grasso's large recips family with a bore of 160mm (6,3 inches) and stroke of 110 mm (4,3 inches).
The first phase of the development programme was oriented towards a new crankcase design and the second phase towards a performance increase realised by a significant speed increase.
The target of the first phase is to design a compressor crankcase to improve noise and vibration levels. Further, to have a design optimised for the manufacturing with lower cost. This implies a compressor fit to be produced on modern welding robots and CNC machining centres, thus reducing the number of labour hours per compressor. The serviceability had to be set at state of the art levels; future regulations had to be met.

The target of the second phase was to significantly improve the price performance ratio by increasing the speed of the compressor up to 1500 rpm for ammonia applications and 1200 rpm for halocarbons.

The paper starts with the design of the rigid and stiff compressor housing to meet the stringent requirements on noise and vibration levels. In addition, the development of the piston assembly is reviewed. Next, the paper focuses on the development of the system that is of major importance in speeding up the compressor: the valve assembly.

COMPRESSOR HOUSE DESIGN

The welded crankcase was kept as design philosophy to keep unique selling points in comparison with casting designs. The external positioning of the cylinder heads gives large heat dissipation. The external discharge lines do not heat up the housing and therefore do not give a heat input to the suction gases. The external suction line guarantees a low superheat, therefore lower compression end temperatures. A large field of application results without additional cooling. The welded design opens lots of possibilities for modular designs, reducing manufacturing costs and giving flexibility in modelling: the RC12 series consists of 6 single and 9 two-stage variants.

The vibration level of the compressor is tackled via an improved stability of the crankcase, through large stiffness and extended footprints, and reduction of the free forces and moments that result from the compression process. This reduction is achieved through an optimum cylinder placement and reduced cylinder angle, the position of one- and two-stage cylinders and the sequencing of loaded and unloaded cylinders. The new extremely stiff crankcase and new cylinder position have been modelled in a Finite Element Model, see Fig. 1.

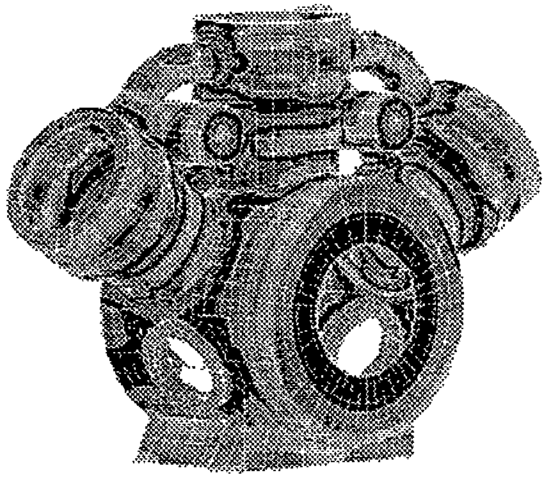

Figure 1 – Finite Element Model of a 3-cylinder RC12 for static calculations

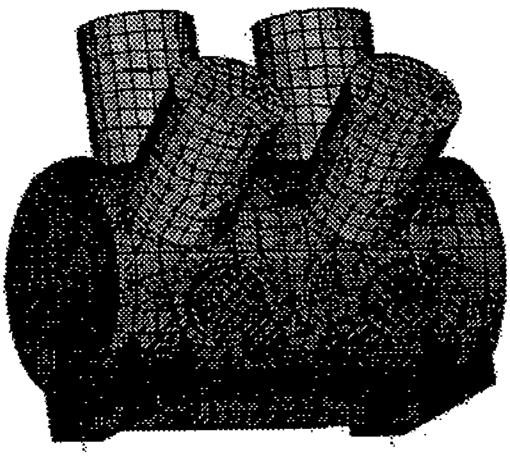

Figure 2 – Dynamic 6-cylinder FEM Model

Design details were calculated and reviewed. The FEM calculations have been verified through strain gauge measurements, showing a very good correlation. An overall deformation test of the complete crankcase has been performed on a 6-cylinder compressor. Three test runs were done up to the maximum working pressure. Lower deformations were found than those resulting from the FEM calculations. Further, all deformations remained in the elastic domain. The final crankcase design has been modelled as a dynamic FEM, see Fig. 2. The footprint position proved to be OK; stiffness of the feet has been increased to meet the stringent design criteria for vibration, and thus stiffness. The measurements have been verified in the testing laboratory, showing again a good correlation between calculation and measurements.

A lower tangential moment was realised due to the cylinder angle change from 62.5 degrees with the RC11, to 60 degrees with the RC12. The variation of the tangential moment around the nominal value is a measure for the vibrations of the compressor.

Final vibration level verifications have been carried out under real conditions. A nearby skating rink offered the possibility to check the compressor against the vibration criteria. Moreover, both a 6- cylinder RC11 and RC12 have been installed at the ice rink installation. The conditions under which was measured were very similar.

The vibration levels have been measured at the centre line of the compressor, the suction and discharge connections and the compressor feet. The tests carried out showed that the vibration level of the RC612, the 6-cylinder, at centre line and the connections are about half of the RC611 levels. Further, the RC12 has much less vibrations at high frequencies. The comparison between RC11 and RC12, 958 rpm, is given in Fig. 3. The RC12 falls easily in the category 'good' of VDI 2056, group T.

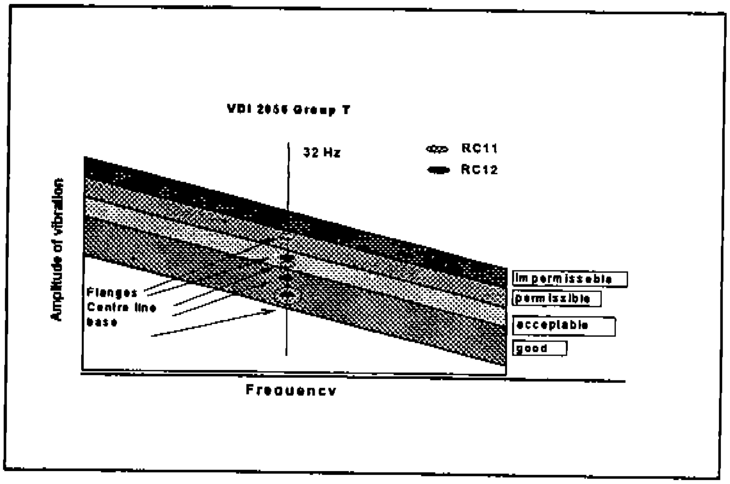

Figure 3 – Vibration level comparison of RC11 and RC12, 958 rpm

The RC12E ammonia versions run at 1500 rpm, i.e. a 50% speed increase in comparison with the 1000 rpm RC12. Consequently, measures have to be taken not to increase the vibration level. One of the key achievements in this is a piston assembly mass reduction of 20%; this will be highlighted in the next chapter. The measurements at the ice rink have been repeated to prove that the vibration level is within the design requirements. Moreover, a 3-cylinder RC12E was taken; it is well known that a 3-cylinder variant has a higher vibration level than e.g. a 6-cylinder variant.

The vibration measurements show, both for the 1000-rpm RC12 and the 1500-rpm ammonia variant of the RC12E, that the compressor has significantly improved characteristics over the RC12's predecessor, the RC11. Moreover, the speed increase has been compensated by the piston mass reduction in combination with the stiff crankcase.

PISTON ASSEMBLY DEVELOPMENT

The piston and piston ring development for the high speed RC12E was focused on the reduction of translating mass, durability, robustness and low oil carry over. Reduction of piston mass, three in stead of five piston rings and a smaller diameter of the piston pin achieved vibration level reduction. Fig. 4 shows the cylinder head and the piston top configuration. The piston assembly design change from RC12, 1000 rpm, to the RC12E, 1500 rpm, has been done in close collaboration with the piston assembly supplier. Most energy took the optimisation towards durability of the piston rings and a low level of oil carry over. The final design showed a translating mass reduction of 20%. The design has been validated in a large laboratory and field-testing programme.

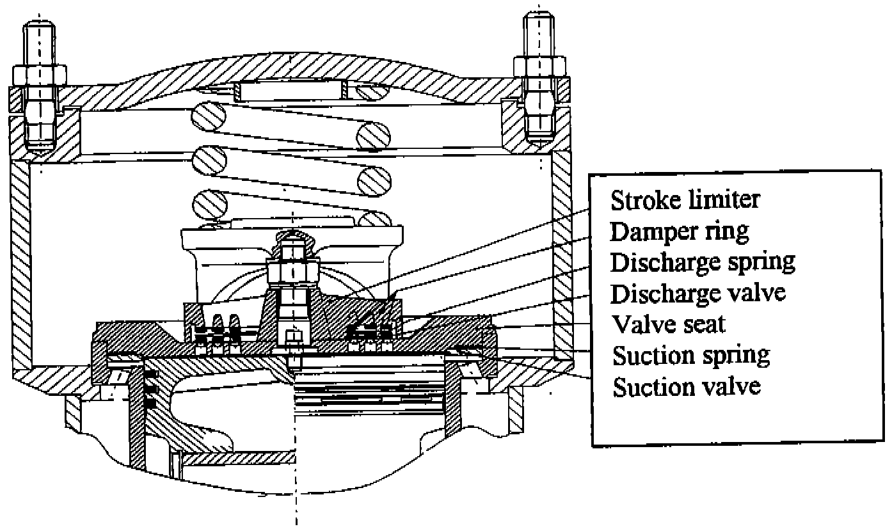

Figure 4 – RC12E cylinder head and piston top configuration

VALVE ASSEMBLY DEVELOPMENT

The objective to improve the price performance ratio of the RC12E by at least 20% has been achieved not only by a more cost effective crankcase design, but to a large extend by an increase in the compressor speed. Early investigations showed that this requirement could be met by the differentiation between ammonia, a relatively low mass refrigerant, and the halocarbons. This was done by defining different maximum compressor speeds: 1500 rpm and 1200 rpm respectively. Higher speeds, in principle, mean a significantly higher mass flow rate. Another objective was to keep the same isentropic efficiencies as with the 1000 rpm variants. This led to the requirement that the suction and discharge ports had to be increased to reach a good flow resistance. Further, the valve lift had to be increased both on the suction and the discharge side. The latter was confirmed by calculations using the Grasso compressor simulation model, based on the well-known Touber formulation. Following effects were taken into account: flow characteristics, flow forces, spring forces and damping, valve sticking forces and impact characteristics. Fig. 5 shows a typical example of a simulation result. The simulation programme has been verified extensively to prove the reliability of the predictions. Higher lift results in larger impact speeds. Calculations showed that these speeds increased significantly; this could result in an 'unwanted' life reduction eventually.

Durability and robustness are key items for the RC12. Therefore, a risk for life reduction has been considered as unacceptable. The decision was made to change the valve material technology to escape from this problem. RC11 and RC12 utilise steel valve rings and steel sinusoidal springs, both for suction and discharge valves. The RC12E valve design is based on PEEK material for the suction- and discharge rings. PEEK is a polymer, reinforced, for this application, with short glass fibres. Material testing has shown that the PEEK material has a

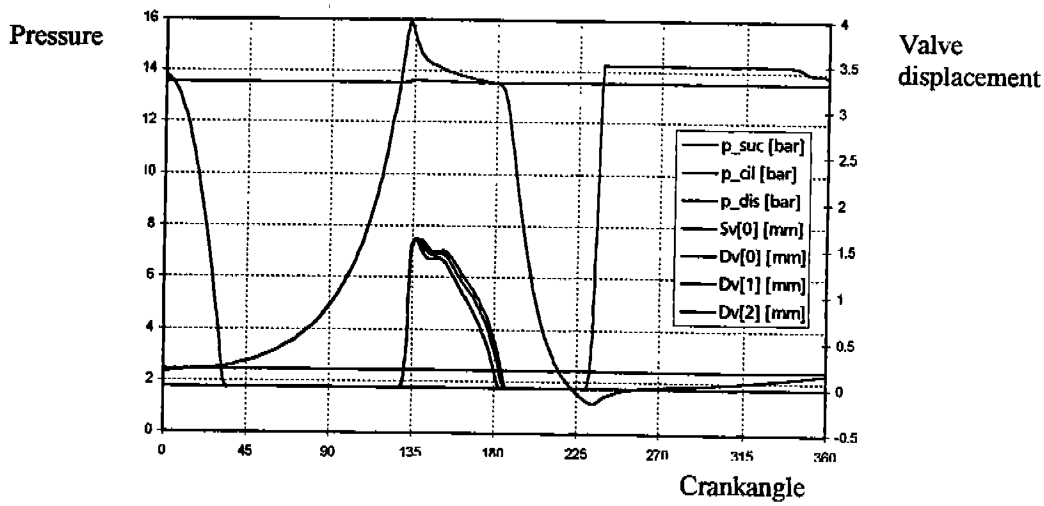

Figure 5 – Compressor simulation result for NH3

much higher allowable impact speed than the corresponding steel variants. Further, extensive testing showed that the PEEK material has an excellent chemical resistance, e.g. versus ammonia in combinations with the temperatures that occur in the compressor, at suction and discharge. The steel springs were kept because the PEEK valve rings soften the impact. Calculations were done on spring stiffness and spring strength, see for example Fig. 6.

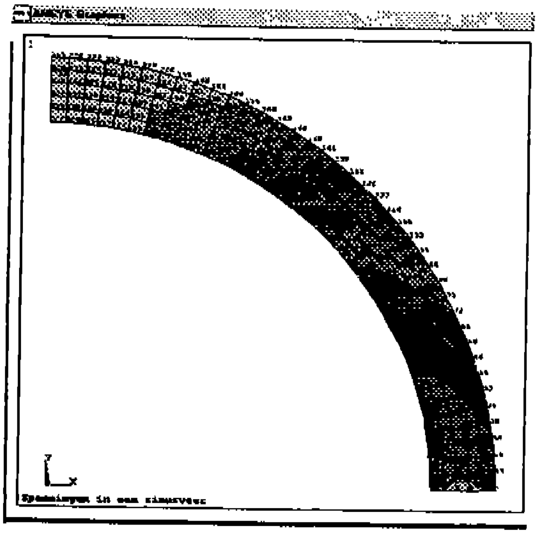

Figure 6 – spring strength FEM calculation result

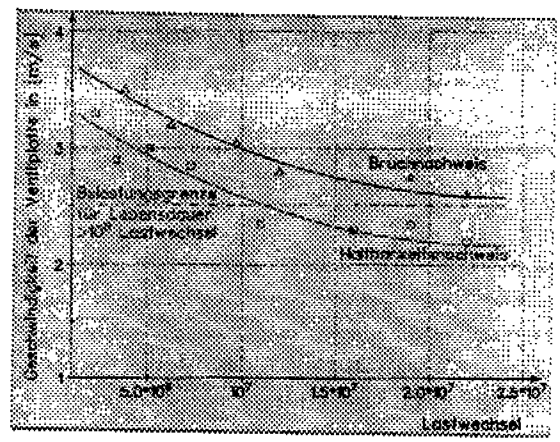

Figure 7 – Fatigue test results

Next to the static strength, the dynamic loads are of importance, i.e. the impact loads. The valve velocities have been measured on both the suction and discharge valves. Results were compared to the material design limits, showing considerable margins of safety for all valves and springs, except for the discharge valve spring. Therefore PEEK damper rings were added to the discharge valve. The single discharge spring is then 'sandwiched' between the PEEK valve- and damperrings. Fatigue tests has been carried out for various impact velocity levels, comparing the design without and with damper rings. Fig. 7 shows the results in terms of 'impact velocity' – 'number of load cycles' curves. The springs in the 'damper' ring design showed a considerably longer life at the various impact velocities, indicating that the allowable impact speed is substantially higher.

The efficiencies of the compressor are determined strongly by the performance of the valves. Extensive testing, with ammonia and other common refrigerants, generated general data on impact speeds, p-V diagrams and isentropic- and volumetric efficiencies. A typical p-V diagram is given in Fig. 8 showing discharge- and suction valve losses to be small.

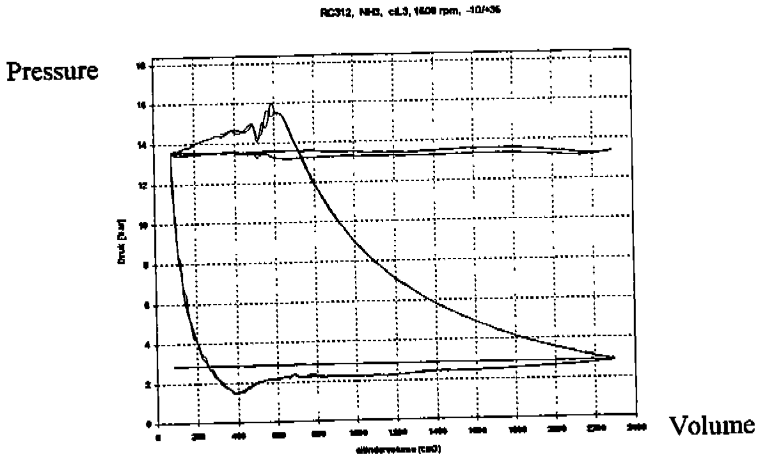

Figure 8 – Typical p-V diagram, ammonia as refrigerant

The isentropic- and volumetric efficiencies for the 1500 and 1200-rpm variant proved to be at the same level as these values for the 1000-rpm variant. This, as such, is a large achievement. In ammonia, a compressor speed increase of 50% has been achieved without a negative effect on the isentropic efficiency. Moreover, the volumetric efficiency has been maintained at about the same level.

CONCLUSIONS

This paper highlights the development of the RC12E, the high speed version of the RC12. The RC12E runs at 1500 rpm for ammonia and 1200 rpm for the halocarbons. The objective was to develop a reliable and efficient reciprocating compressor. The systemetic development of e.g. the new crankcase, piston assembly and fibre reinforced composite valves have been completed succesfully, as an extensive testing and durability testing programme showed. Final proof has been generated in a field testing programme; e.g. three ammonia variants have been followed closely for over 14,000 hours. Grasso's low mass piston assembly gives low vibration levels. The valves, applying new PEEK fibre reinforced polymer rings, show an excellent behaviour on both isentropic and volumetric efficiencies, durability and reliability.