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HIGH PRESSURE INVESTIGATION OF REFRIGERANTS HFC245fa, R134a and R123.

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ABSTRACT

To increase the power efficiency of refrigeration and air conditioning machinery it would be advantageous to be able to run the compressors without any lubricant added to the refrigerant. Without any added lubricant in the refrigerant it would also be possible to simplify the whole refrigeration machinery by excluding all oil handling systems, and there would not any longer be any risk for oil accumulation in the heat exchangers.

To find out if the refrigerants themselves can be used for lubrication purposes in rolling contacts, HFC245fa, R134a and R123 were investigated in a high pressure chamber. There they were compressed to more than one gigapascal (1 GPa) and their compressibility and solidification pressure were measured as well as their shear strength at pressures above the solidification pressure.

Key Words: High pressure, Refrigerants, HFC245fa, R134a, R123, Compressibility, Solidification, Shear strength, Friction.

1. INTRODUCTION

The use of rolling bearings in early refrigeration compressors did initially give some unexpected failures of two types. Failures which now are understood and easy to avoid.
One is that some refrigerants, e.g. ammonia, do chemically attack the bearing surfaces under certain conditions and increase the bearing wear rate dramatically. This happens if roller bearings with inherent high kinematic sliding speeds are used. The high sliding speed at the roller-flange contact combined with high contact pressure in tapered roller bearings makes the local energy levels high enough to transform the bearing oxide layers together with the ammonia into free water and bearing surfaces without any protective oxide layer. This can give material removal rates of up to a few millimetres per day, and the bearings fail dramatically. This is solved using other rolling bearing types.
The other type of failure was caused by the fact that many refrigerants, when they are dissolved into lubricants, decrease the lubricant viscosity very rapidly. Even as little as 9% R134a in an ester oil decreases the oil film

thickness to half its value. At high concentrations of R134a (35 % by weight) also the pressure viscosity coefficient decreases to half its value for neat oil (Wardle *et al.*, 1992, Jacobson, 1994). This problem has been solved by separating off refrigerant from the oil to increase the lubricant viscosity at the bearing position.

When lubricants are dissolved in refrigerants and circulated in refrigeration and air conditioning systems, the flow velocities are normally so high that the solution is not in equilibrium. This means that the refrigerant either is being absorbed by or boiling off from the lubricant at the bearing to be lubricated. Depending on if the bearing surface is warmer or colder than the local refrigerant-oil mixture boiling point at the local pressure, refrigerant will start to condense on the bearing surface or boil off from the lubricant on the surface. This unstable behaviour has a strong influence on the possibility to build up a thick enough lubricant film.
To avoid all these problems, it is thus interesting to investigate if it is possible to lubricate rolling bearings using only the refrigerant as lubricant.

2. REFRIGERANTS IN ROLLING BEARINGS.

When pure refrigerants are used as lubricants, the operating conditions of the rolling contact bearings are extremely difficult. In such a case, the cooling conditions for the bearing have to be maintained adopting a design that assures a continuous flow of refrigerant through the bearing at all times. An increase of bearing temperature can lead to reduced film and thus to a further quick increase of friction and temperature for the bearing and an increase of the risk of failure.
Furthermore, to make hybrid bearings able to withstand pure refrigerant lubrication for very long time periods, an effective separation of the mating surfaces has to be achieved. This in turn requires the optimisation of the raceway topography to ensure maximum elastic deformation of the asperities in the Hertzian contact. Tests until now have shown that hybrid bearings have a significantly improved ability to withstand the severe conditions developed with pure refrigerant lubrication compared to bearings made of steel only. Therefore there is good indication to show that hybrid bearings can succeed in pure refrigerant lubrication applications.

In marginal lubrication situations where the lubricant film is thin, either because the viscosity is too low or because the rotational speed is too low, metallic contact through the lubricant film takes place. Such metallic contact is very detrimental for sliding contacts like in journal bearings. It is much less of a problem for rolling contacts like in rolling bearings, especially if the lubricant has some boundary lubrication properties.
Some earlier types of refrigerants,e.g. R22, could be used together with naphthenic mineral oils, and the chlorine in the refrigerant worked like an EP-additive making it possible to use rather thin lubricant films in rolling bearings. The lubricant films were very thin due to the low viscosity of the refrigerant oil mixture, but the pressure viscosity coefficient α was quite high. For ester oils α is typically 10 per GPa and for naphthenic oils α can be as much as 35 per GPa. Refrigerant R22 has an α-value of about 30 per GPa, which gives a high viscosity also at rather low pressures.

If the viscosity at the inlet of a rolling contact increases fast enough with the pressure, a thick lubricant film can be built up also if the the viscosity at atmospheric pressure is very low. It is only necessary that the viscosity becomes high enough for good lubrication before the flat Hertzian contact zone is reached by the lubricant.

Tests in a WAM-machine at SKF Engineering & Research Centre have shown that it is possible to build up a separating film of pure refrigerant R123 in a ball bearing already at rolling speeds of 3 to 4 m/s. To maintain the smoothness of the bearing surfaces under those conditions it is necessary to use ceramic (silicon nitride) rolling elements, though. The extremely smooth surfaces of ceramic balls and rollers combined with high quality steel rings have been shown to need only very thin lubricant films to work very well and to give long endurance lives. The traction coefficient in the WAM-machine actually was 25 % lower for pure R123 lubrication than for a very low viscosity oil having the viscosity 2.4 mm^2/s at 40 °C.

The elastohydrodynamic lubricant film thickness is roughly proportional to the speed times the viscosity raised to the power 0.7, multiplied with the pressure-viscosity coefficient α raised to 0.5. A low viscosity fluid with a high α-value might thus be able to lubricate smooth rolling bearing surfaces. To investigate which type of refrigerant could be possible to use as a rolling bearing lubricant without any oil, three refrigerants were tested in a high pressure chamber. The tested refrigerants were HFC245fa, R134a and R123.

3. THE HIGH PRESSURE CHAMBER

The detailed design of the high pressure chamber has been described elsewhere (Jacobson, 2000, Ståhl and Jacobson, 2003). The high pressure chamber had to be modified to make it possible to feed in the refrigerant into the cemented carbide container, see figures 1 to 4. Figure 1 shows an overview of the central part of the high pressure chamber with the lower hydrostatic bearing at the bottom. Through the top part of the hydrostatic bearing is a horizontal channel drilled to make it possible to feed in the refrigerant gas to be tested, see figure 2. The horizontal channel connects to a vertical channel which leads to the lower plunger. The three types of plungers used are shown in figure 3. The plunger type shown in the middle has a central hole, which makes it possible to feed in gas from the channel through the hydrostatic bearing to the volume to be compressed. To retain the gas in the high pressure chamber, a seal with a built-in non-return valve is used, see figure 4. The seal has a larger outer diameter than the inner diameter of the high pressure chamber, so a strong press fit holds the seal in place while the bottom plunger is pressed against it. This makes it possible for the refrigerant to be fed through the hydrostatic bearing, the plunger, and the non-return valve into the compression chamber. The seal at the top plunger is first put in place so high up, that it leaks slightly and the air can leak out and be replaced by refrigerant from below before the top plunger is moved down and the the volume is sealed off. The feed pressure is then increased until the compression volume is filled with liquid gas. The top plunger is then pressed down against the refrigerant, so the high pressure makes the non-return valve in the lower seal close and prevent leakage down through the lower plunger.
When the refrigerant is contained between the seals, the compression and shear stress measurements can be performed just like the measurements for liquids and solids.

4. COMPRESSIBILITY AND SHEAR STRESS MEASUREMENTS

A number of compression and shear tests were carried out for each material. The data collections for the compression tests were automatic, and the computer sampled 3 000 – 17 000 data points for each compression. Each sample consisted of two measured values, which corresponded to the pressure and the volume of the compressed medium at this pressure. A compression test was executed in a few seconds when the pressure was increased from zero to the maximum pressure.

The shear stress measurements were much more work intensive. The shear stress measurements were made at different pressures and different shear velocities, and the mean shear stress for a sliding distance of 4.2 mm was calculated by the computer. The computer also calculated the contact surface area between the compressed medium and the cemented carbide cylinder. It also registered the total shear force, the total compression force and the height of the compressed medium. All these data were manually collected into tables, and for each shearing velocity typically ten different pressures were analysed. A shear investigation of one material at four different shearing velocities took typically four hours.

The diameter of the seals were marginally larger than the diameter of the high pressure cylinder. Thus the force needed to overcome the friction between the seal and the cylinder wall had to be measured. The seal friction force was measured by decreasing the pressure until the gaseous materials were in the liquid state, which was shown by a constant shear force independent of pressure. Compression tests without any material between the seals indicated that the seal friction was quite independent of the pressure in the high pressure chamber. The seal friction varied with the sliding speed and the type of compressed material though. When the shear stresses in the compressed medium were measured, the total shear force was measured for decreasing pressure for each shear velocity until only the seal friction was remaining and could be subtracted from the measurements.

5. RESULTS HFC245fa

The measured compression curve and shear strength curves for HFC245fa is shown in figures 5 through 8. All the refrigerants tested have slightly higher compressibility in the liquid and solid state compared to mineral oils. In figure 5 is shown the compression measurements for refrigerant HFC245fa. As can be seen from figure 6, it converts to a solid already at 130 MPa at room temperature when the shear stress at the shear rate 0.89 s^{-1} departs visibly from zero. At the same pressure the compression curve in figure 5 changes its curvature and the parameter η in the Vinet (1987) equation for compression decreases towards zero.

$$p=(3B_0/x^2)(1-x)e^{\eta(1-x)}$$

where p= pressure, Pa
B_0= bulk modulus at zero pressure, Pa
1-x= linear compression such that $1\text{-}x^3 = \Delta V/V_0$, dimensionless
V_0= initial volume, m^3
ΔV= volume decrease, m^3
η= curvature parameter, dimensionless.

The compressibility also decreases from $(\Delta V/\Delta P)/V_0=4.3*10^{-10}\ Pa^{-1}$ at zero pressure to $1.4*10^{-10}\ Pa^{-1}$ at 1.1 GPa.
Figures 6 through 8 show the shear stress as a function of pressure for the mean shear strain rates 0.89, 21.37 and 142.86 s^{-1}. The mean shear strain rate was calculated using the sliding speed divided with the gap height 20 μm. The measurement results indicate, though, that the shear took place in a plane rather than in a volume, as all the measurements show a straight line increase of the shear stress with the pressure increase for all speeds. This is contrary to low shear stress viscometry measurements, where the viscosity increases exponentially with pressure.

The refrigerant HFC245fa seems to be a rather good lubricant as can be seen from figure 8. It converts to a solid at a very low pressure, 130 MPa at 20°C. That indicates a viscosity pressure coefficient of about 100 GPa^{-1}, which is extremely high. The refrigerant shear stress as a function of pressure is less sensitive for shear rate increase than the other tested refrigerants. HFC245fa decreases its shear stress only 30% when the shear rate is increased from 0.89 to 142.86 s^{-1} at 1.5 GPa pressure. This can be compared with the results for R123, when it is solidified at 2.25 GPa and the shear rate 0.89 s^{-1} gives the shear stress 75 MPa, but when the shear rate is increased to 142.86 s^{-1} the shear stress falls to 5 MPa, a shear strength decrease of 93%.

The compression measurements for HFC245fa gave the density as a function of pressure at room temperature of

$$\rho(p)/\rho(0) = 1.1482 + 0.125*p - 0.1482*e^{(-1.856*p)}$$

where p is in GPa and $\rho(0)$ =1320 kg/m^3 at 23 °C.

6. RESULTS R134a

The measured compression curve is shown in figure 9. It is constructed by subtracting the compression of a heat insulating Teflon cup surrounding the R134a from the total compression of R134a plus the cup. The insulating cup was used to keep the R134a cold and thereby more easily inserted into the compression volume.

The shear stress measurements for R134a at the different shear rates are shown in figures 10 to 12. The shear stress increase is again proportional to the pressure increase starting already at 100 MPa pressure for the different strain rates. There is a big difference between R134a and HFC245fa though. While the shear stress decreases only 30% for HFC245fa when the shear rate increases from 0.89 to 142.86 s^{-1} it falls much more dramatically for R134a. At 1150 MPa pressure the shear stress falls from 160 MPa to 0.76 MPa, a factor of 210, when the shear rate increases from 0.89 to 142.86 s^{-1}. The large differences between the shear stresses at different shear rates are shown in figures 10 and 12, where stresses for the speeds 0.89 s^{-1} and 142.86 s^{-1} are shown.

By subtracting the influence of the Teflon cup compression it was possible to get an approximative equation for the density of R134a.

$$\rho(p)/\rho(0) = 1.15 + 0.256*p - 0.15*e^{(-21.15*p)}$$

where p is in GPa and $\rho(0)$ =1232 kg/m^3 at 20 °C.

7. RESULTS R123

The measured compression curve is shown in figure 13. The compressibility is similar to R134a and larger than the compressibility of HFC245fa. As R123 did not solidify at 1.2 GPa and room temperature, the high pressure chamber plunger with a non-return valve was exchanged for a normal plunger, see the left plunger in figure 3. The boiling point for R123 at room pressure is 27.8 °C, so it could be filled into the high pressure chamber like a normal liquid lubricant.

With the normal plungers it was easy to reach the solidification pressure for R123 at room temperature, and it was between 1.5 GPa and 1.6 GPa depending on the shear rate for the experiment. Figures 14 and 15 show the shear stress as a function of pressure for R123 at different shear rates. Just like the behaviour of R134a also R123 is very thixotropic. It decreases its shear strength at 2250 MPa pressure more than an order of magnitude from 65 MPa when the shear rate is 0.89 s^{-1} to 5 MPa when the shear rate is 142.86 s^{-1}.
Another phenomenon became apparent during the experiments with R123. Despite the very stiff design of the high pressure chamber, an extremely strong stick-slip vibration was created during the shear strength measurements. This can partly be seen in figure 15 for the speed 75.75 s^{-1}, where the shear stress jumps from 13.5 to 20.5 MPa due to the stick-slip, indicating a jump in seal friction of more than 500 N.

The compression measurements for R123 gave the density at room temperature as a function of pressure to be

$$\rho(p)/\rho(0) = 1.018 + 0.32*p - 0.018*e^{(-7.778*p)}$$

where p is in GPa.

8. CONCLUSIONS

The compression and shear stress measurements have shown that the three investigated refrigerants HFC245fa, R134a and R123 behave quite differently.
HFC245fa increases its viscosity more than any mineral oil when the pressure is increased. R134a has a pressure viscosity coefficient similar to a poly alpha olefin oil and R123 has a viscosity pressure coefficient lower than a typical ester oil.

The shear stress measurements have shown that all three refrigerants are non-Newtonian at high shear stresses, and that the shear strength increase at a given shear rate is directly proportional to the pressure increase at pressures above the solidification pressure. At high shear rates all three refrigerants decrease their shear strength, but at different rates. HFC245fa looses about 30% of its shear strength when the shear rate is increased from 0.89 s^{-1} to 142.86 s^{-1} while R134a and R123 decrease their shear strength much more, 99.5 and 93 % respectively.

The small decrease of shear strength with shear rate for HFC245fa compared to R134a and R123 at the same time as the pressure viscosity coefficient is much higher for HFC245fa indicates that it from a mechanical point of view is a good candidate for refrigeration compressor bearing lubrication without any addition of oil. There are naturally many other considerations to take into account when a refrigerant is chosen for a specific application.

NOMENCLATURE

B_0 =	bulk modulus at zero pressure, Pa
p =	pressure, Pa
V_0 =	initial volume, m^3
1-x =	linear compression such that $1-x^3 = \Delta V/V_0$, dimensionless
α =	pressure – viscosity coefficient, Pa^{-1}
ΔV=	volume decrease, m^3
η =	curvature parameter, dimensionless.

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Figure 1. High pressure chamber and hydrostatic bearing.

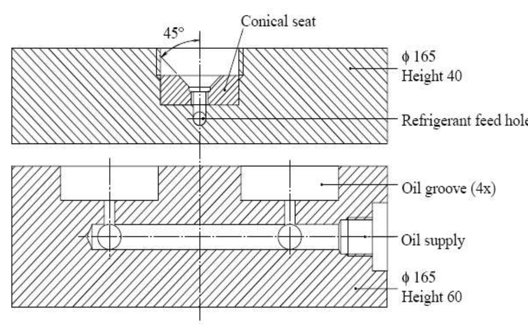


Figure 2. Hydrostatic bearing.

Figure 3. Plungers.

Figure 4. Lower seal vith built-in non-return valve.

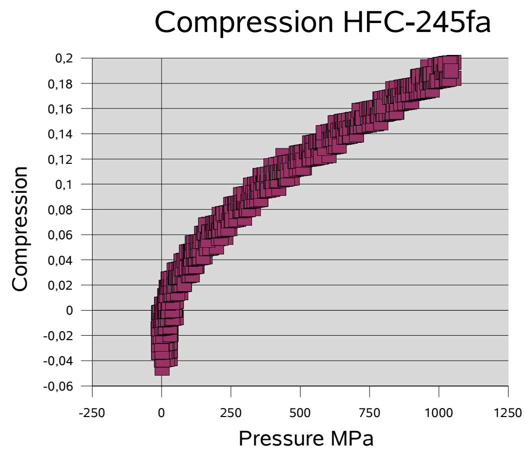


Figure 5. Compression of HFC245fa.

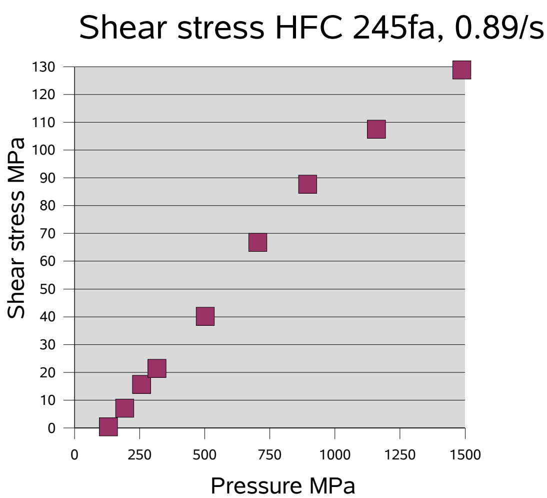


Figure 6. Shear stress for HFC245fa at 0.89 s^{-1}.

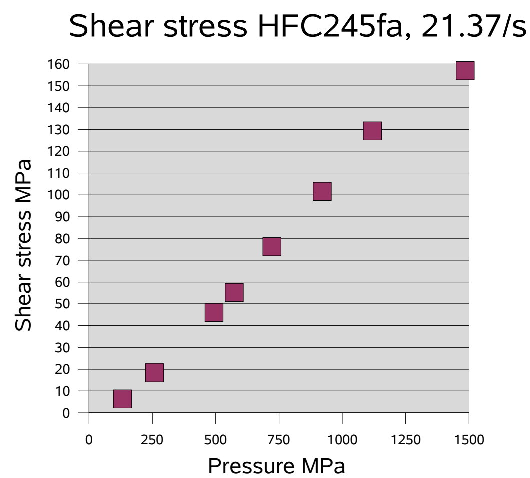


Figure 7. Shear stress for HFC245fa at 21.37 s^{-1}.

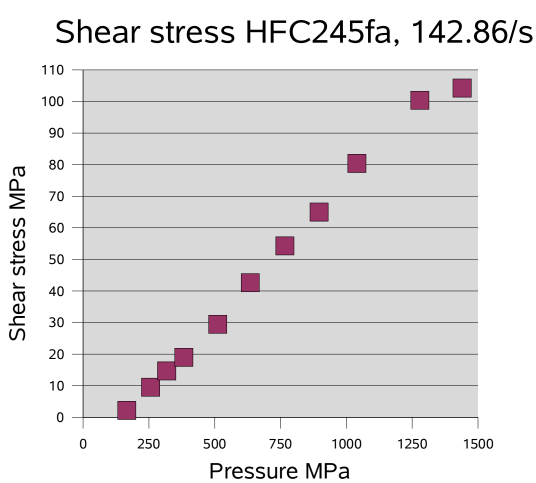


Figure 8. Shear stress for HFC245fa at 142.86 s^{-1}.

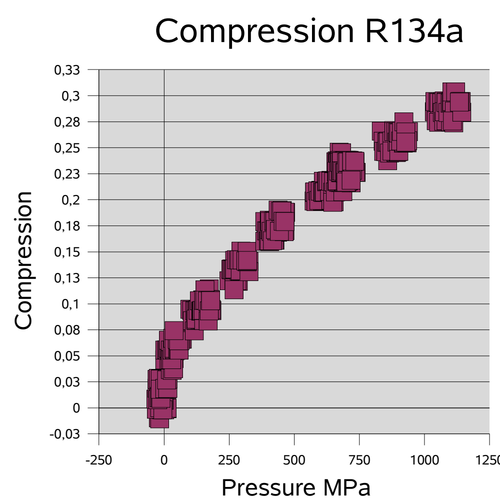


Figure 9. Compressionof R134a.

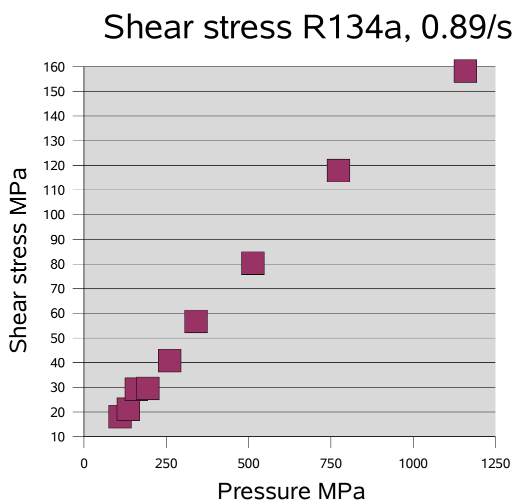


Figure 10. Shear stress for R134a at 0.89 s^{-1}.

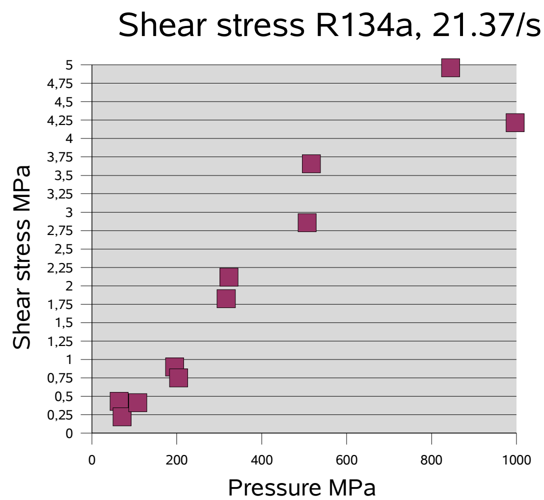


Figure 11. Shear stress for R134a at 21.37 s^{-1}.

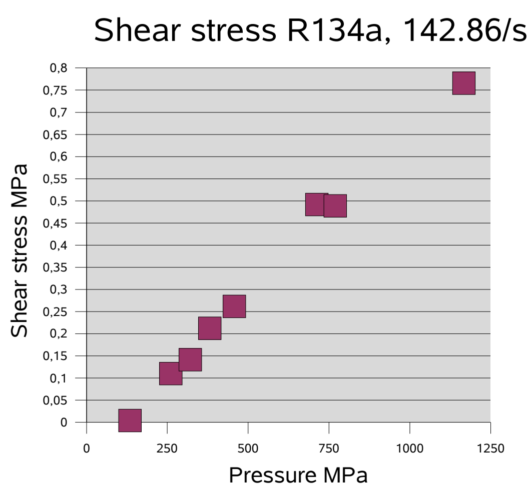


Figure 12. Shear stress for R134a at 142.86 s^{-1}.

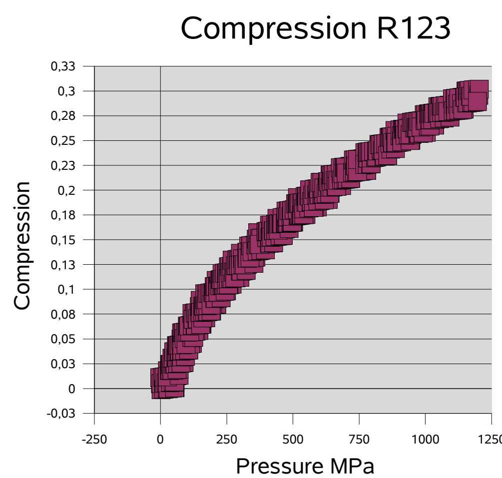
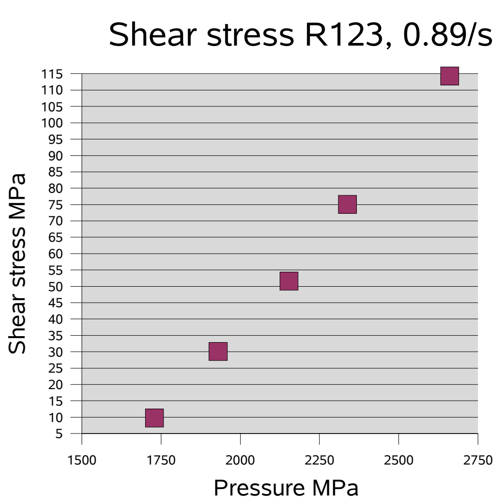


Figure 13. Compression of R123.



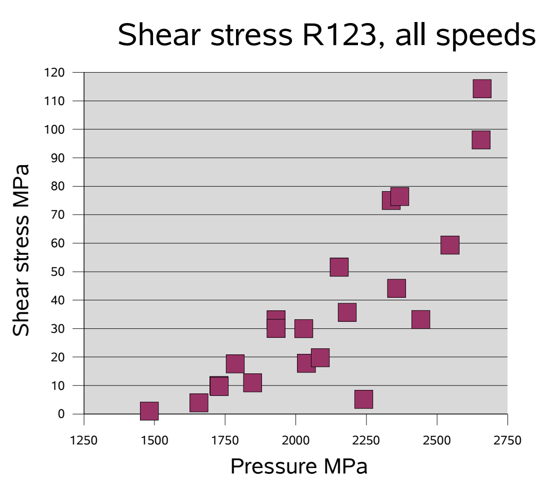


Figure 14. Shear stress for R123 at 0.89 s^{-1}. Figure 15. Shear stress R123 at all speeds between 0.89 and 142.86 s^{-1}.