Oil Atomisation in Oil-Injected Screw Compressors

Dr. ir. **M. de Paepe**, ir. **W. Bogaert**, ir. **D. Mertens**, Dr. ir. **M. Vandevoorde**, ir. **J. Agache**, Gent/B

Summary

This paper addresses the effect of oil atomisation in an oil-injected screw compressor. A test rig was build to assess the performance of different types of atomisers. Experiments on the test rig show that lowering the oil droplet diameter results in a considerably higher heat transfer. Growing oil flow rate, also gives a better cooling effectiveness.

In parallel with the experiments, a thermodynamic model is developed by which the compression process can be calculated for every degree of revolution of the male-rotor. An oil-cooling effectiveness is defined. This way the influence of cooling oil temperature, cooling oil mass flow rate and injection point can be analysed. Having a better heat transfer effectiveness does not give a considerable gain in specific work. Lowering oil temperature gives better results, while changing the oil flow rate and the point of injection only gives small gains. Furthermore it is shown that the cooling oil coming from the bearings has a negative influence on the performance.

This paper shows that trying to reach isothermal compression through oil atomisation is not possible. The importance of the cooling effectiveness in the thermodynamic process is too small to have a significant influence.

NOMENCLATURE

$\dot{Q}$	:	transferred heat [W]
T	:	temperature [K]
V	:	volume [litre]
c_p	:	heat capacity [J/kg K]
$\dot{m}$	:	mass flow rate [kg/s]
n	:	polytropic exponent [-]
n_{rot}	:	number of revolutions [rpm]
p	:	pressure [Pa]
α	:	angle of revolution of the male rotor [°]
ε	:	cooling effectiveness [-]

Subscript

air	:	of air
i	:	step *i* of the calculation process
in	:	inlet conditions
oil	:	of oil
out	:	outlet conditions

Introduction

Screw compressors can be divided in mainly two types: oil-free and oil-injected screw compressors. Injecting oil in the compressor air has several purposes.

First of all the temperature of the air leaving the compressor is reduced and as a consequence the compression work is lowered. As a result, the pressure ratio can get higher and the compressor casing does not have to be cooled externally.

Oil does not only cool, it is also responsible for the sealing of the compressor elements. This causes the volumetric efficiency to be raised and the pressure difference across the rotor to be higher.

The oil also serves as lubricant. Contact between the male and female rotor is avoided this way and no gear box is needed.

This paper focuses on the cooling function of the oil. Gneipel [1] claims that isothermal compression is reached with oil injection. However, most compressors in operation are working with isothermal efficiencies of only 65%. This means that the ratio of the specific compression work needed for isothermal compression to the actually needed specific compression work is 0.65.

Oil is injected in the compressor through large holes in the compressor casing. This causes the heat transfer from the air to the oil to be relatively bad inside the compressor. The large holes result in low inlet velocities of the oil causing the oil droplets that form to be big. Furthermore, the contact time between the air and the oil is small. Rinder [2] has calculated that with high revolution speeds the contact time amounts only to 1 ms.

Rinder and Hammerl [3] showed that in theory the contact time between oil and gas can be augmented to several seconds if the oil is atomised. This can be done by inserting atomisers in the casing and using high pressure oil injection. Small droplets are created and the residence time of the droplets gets higher. Experiments by Persson [4] and Rinder [3] on an actual compressor with atomisers showed that no big change is detected in the air outlet temperature and thus in the specific work. Both authors analyse with great care the heat transfer mechanism inside the compressor. However no clear explanation is given why the atomisation has so little effect.

In this paper a thermodynamic model of an oil injected screw compressor is developed. It is validated with measurement data of a screw compressor. In the model special attention is paid to the description of the cooling process. The concept of cooling effectiveness is introduced. The cooling effectiveness of several atomisers is experimentally determined on a test rig, which mimics the flow configuration of an actual compressor. Using these data the effect of oil-atomisation on compressor cooling is analysed.

Experimental verification of the cooling performance

1. Introduction

Oil atomisation should result in a better cooling performance of the oil. In order to be able to evaluate the cooling effect a test rig was built. The test rig is constructed in such a way that the flow conditions for the test rig are similar to the conditions inside the compressor. This way the performance of different injection nozzles can be compared in conditions as close as possible to reality. It was not possible, nor interesting to install nozzles in a real compressor. Therefore a strong simplification was introduced. The nozzles are mounted into a tube of plexiglass. The section of the tube was chosen to represent one slot between the rotor elements.

2. *Test Rig*

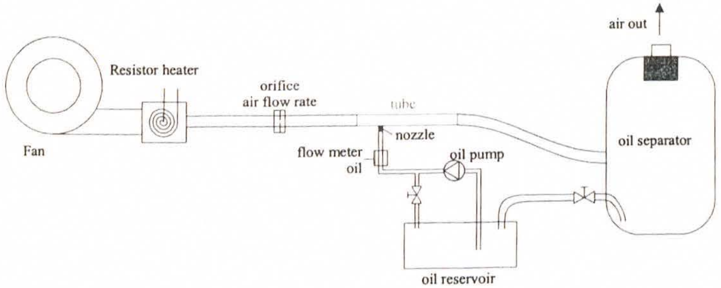

Figure 1 : Test rig

Figure 1 shows a diagram of the test rig. A fan with variable speed delivers the air. The air is heated with a resistor heater. It is then introduced into the plexiglass tube. In the tube different types of nozzles can be mounted and before and after the nozzle several thermocouple sensors are placed at different positions. The oil is supplied through a pump. The mixture of oil and air leaving the tube is sent through an oil separator. The oil is recovered in an oil reservoir. The air flow rate is measured with an orifice plate, the oil flow rate with a rotha-meter.

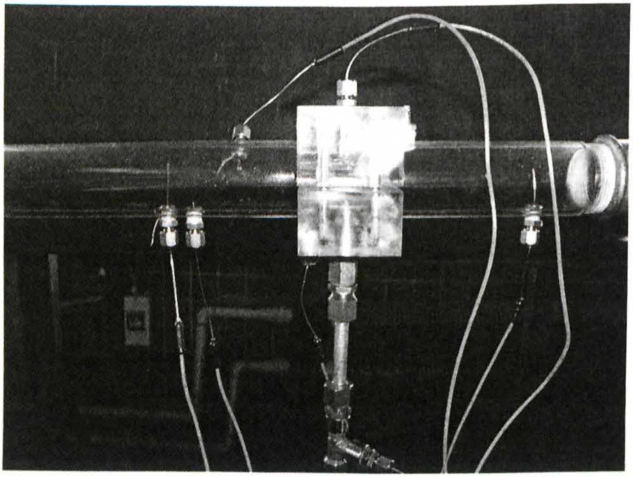

Figure 2 : Close-up of the measurement section

Figure 2 shows a close-up of the measurement section. In the middle the injection nozzle mounted on the tube can be seen, showing also the position of the thermocouples. Six thermocouples are inserted in the tube. One is situated upstream of the nozzle, one is placed above it and the four others are inserted downstream. The thermocouples are positioned in such a way that the air temperature or the oil temperature is measured. As the oil will flow at the bottom of the tube, the shorter thermocouples are in the oil flow.

3. *Measurements – cooling effectiveness*

By measuring the inlet temperature of air $T_{air,in}$, the outlet temperature of the air $T_{air,out}$ and the oil inlet temperature $T_{oil,in}$, the effectiveness of cooling can be defined as [1]:

$$\varepsilon_T = \frac{T_{air,in} - T_{air,out}}{T_{air,in} - T_{oil,in}} \qquad (1)$$

Five different nozzle types were tested : a hole of 3 mm diameter, 2.5 mm and 2 mm and 2 atomisers of type TG-10 and TG-6.5 [6]. The 3 mm hole gives an injection similar to the situation in the real compressor. According to [5] the Sauter Mean Diameter (SMD) of the droplets resulting from the five nozzles is respectively : 3.53 mm, 1.97 mm, 0.96 mm, 0.438 mm and 0.278 mm for a mass flow rate of 6.25 kg/min, which is the oil mass flow rate necessary to obtain similarity.

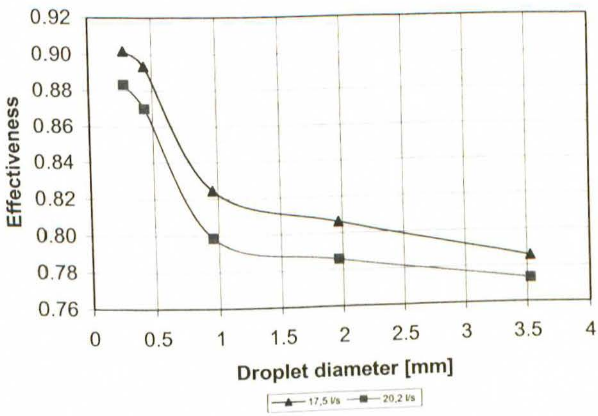

Figure 3 : Influence of droplet size on effectiveness

[1]This definition is different from the definition taken in the thermodynamic model. A relation between the two definitions can be derived, if the mass flow rates are taken into account. More measurement accuracy was obtained using this definition. The definition of the model was better to describe the cooling process.

4. *Results*

1. Droplet diameter

In Figure 3 the influence of droplet diameter on effectiveness is shown for two different oil flows. It is clearly shown that the effectiveness goes up for diminishing droplet diameter. Reducing the droplet diameter by a factor 7, raises effectiveness by 11 points. For the cases similar to the compressor a fairly high effectiveness can be reached.

2. Ratio of oil to air flow rate

Changing the ratio of oil to air, will result in a different effectiveness. The oil flow rate was kept constant, as to keep the droplet diameter constant for a certain nozzle. The air flow rate was varied. In Figure 4 the results are presented for the two nozzles giving the smallest droplets.

The effectiveness grows as more oil is injected. This was expected as more oil is available to take heat out of the air. The effectiveness for the nozzle with the smaller diameter is again higher at all measurement points.

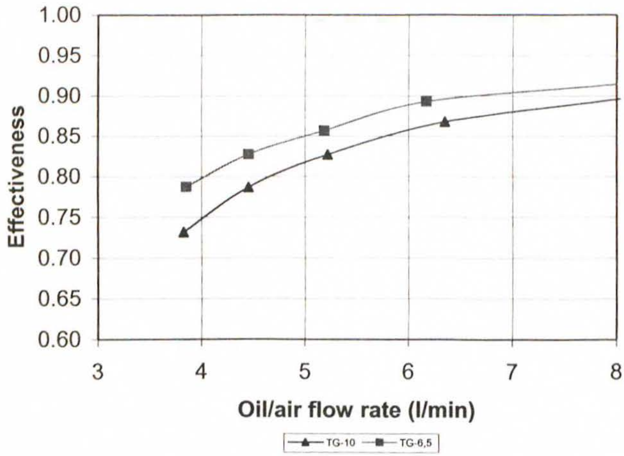

Figure 4 : Influence of oil flow rate

Thermodynamic model

1. Implementation

In order to be able to study the effect of oil injection on the compression, the compression process is divided in small steps, for which a polytropic relation between volume and pressure describes the transformation. The volume of the working space depends on the angle of revolution of the male rotor. From the "herring-bone" diagram (Figure 5) , ie the plan view of the unwrapped compressor envelope, it is clearly seen how the V-shaped groves will be squeezed against the outlet end wall and decreased in volume, until the outlet port is reached. The inlet ports are located at the maximum volume, and the outlet port is located for a specified built-in volume ratio.

On Figure 5 the oil injection ports are shown. Oil injection commences at different points in the male and female rotor. This is not taken into account, so everything is referred to the male rotor.

In the calculation model a step of 0.5° revolution of the male rotor is taken as step size.

Four different phases can be discerned: oil-free compression, compression with injection of oil, oil-containing compression and expulsion. For each phase different equations have to be solved.

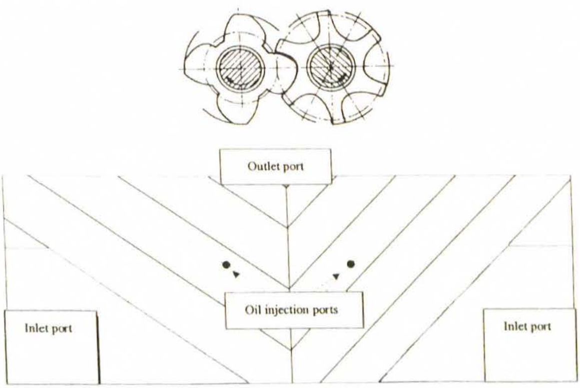

Figure 5 : Herring-bone diagram

2. *Oil-free compression*

The moment the inlet port closes the compression starts. At this moment The working space has no contact with the oil injection ports. The compression process is given by :

$$p \cdot V^n = cte \qquad (2)$$

or for temperature as function of volume :

$$T_{air,i+1} = T_{air,i} \cdot \left(\frac{V_i}{V_{i+1}} \right)^{n-1} \qquad (3)$$

3. *Compression with oil-injection*

The moment the working space reaches the oil-injection ports (at α_{oil}), oil is injected into the working space. As a result the volume changes and the air is cooled.

The compression process can now be split up in two separate steps (Figure 6). First there is the pressure change due to volume change caused by the rotation of the compressor elements and due to the volume taken by the oil :

$$V_{i+1}' = V_{i+1} - \frac{\dot{m}_{oil} \cdot (\alpha_{i+1} - \alpha_{oil})}{n_{rot} \cdot 360° \cdot \rho_{oil}} \qquad (4)$$

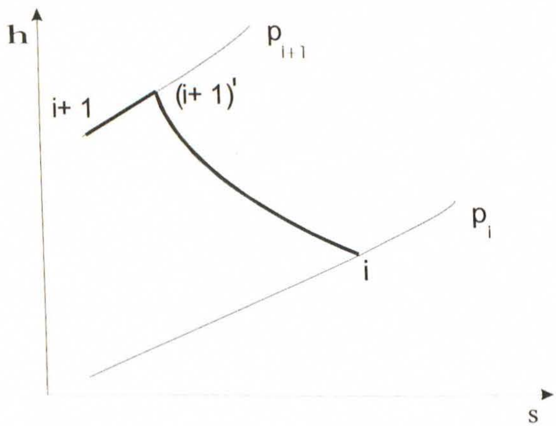

Figure 6 : The cooled compression process

The temperature changes accordingly (i -> (i+1)') and can be calculated with Eq. (3) , using V_{i+1}'.

The cooling ((i+1)'-> (i+1)) is not perfect because it has to happen in a short time span. As a result the oil and air do not get time to obtain the same temperature as would be reached in equilibrium. To take this effect into account a cooling effectiveness is defined:

$$\varepsilon_C = \frac{\dot{Q}}{\dot{Q}_{MAX}} \tag{5}$$

The maximum heat absorbed by the oil $\dot{Q}_{MAX}$ is the heat absorbed when reaching equilibrium :

$$\dot{Q}_{MAX} = \dot{m}_{air}\, c_{p,air} \left(T_{air}' - T\right) \tag{6}$$

$$\dot{Q}_{MAX} = \dot{m}_{oil}\, c_{p,oil} \left(T - T_{oil}'\right) \tag{7}$$

T is the equilibrium temperature given by :

$$T = \frac{\dot{m}_{air}\, c_{p,air}\, T_{air}' + \dot{m}_{oil}\, c_{p,oil}\, T_{oil}'}{\dot{m}_{air}\, c_{p,air} + \dot{m}_{oil}\, c_{p,oil}} \tag{8}$$

The real temperatures for air T_{air} and oil T_{oil} are given by :

$$T_{air} = T_{air}' - \varepsilon_C \cdot \frac{\dot{m}_{oil}\, c_{p,oil} \left(T_{air}' - T_{oil}'\right)}{\dot{m}_{air}\, c_{p,air} + \dot{m}_{oil}\, c_{p,oil}} \tag{9}$$

$$T_{oil} = T_{oil}' + \varepsilon_C \cdot \frac{\dot{m}_{air}\, c_{p,air} \left(T_{air}' - T_{oil}'\right)}{\dot{m}_{air}\, c_{p,air} + \dot{m}_{oil}\, c_{p,oil}} \tag{10}$$

The moment oil is present the mixing of old and new oil, changes the oil temperature inside the working space as given by :

$$T_{oil,i+1}' = \frac{\dot{m}_{oil,i} \cdot T_{oil,i} + \dfrac{\dot{m}_{oil} \cdot \left(\alpha_{i+1} - \alpha_i\right)}{n_{rot} \cdot 360°} \cdot T_{injection}}{\dot{m}_{oil,i+1}} \tag{11}$$

After calculation of the compression temperature by Eq. (3) and (4), Eq. (11) is applied to obtain the oil temperature. Using Eq. (9) and (10) the air and oil temperatures after each step are determined.

4. *Oil-containing compression*

The oil being present in the working space is standing at a temperature lower than the compression air. As a result cooling of the air as described by Eq. (9) and (10) is still applicable. Only the mass of oil no longer changes.

5. *Expulsion*

The moment the outlet port opens the air is driven out of the working space. If the pressure given by the volume ratio is different from the pressure of the system where the compressor is connected to, the air will suffer from an isochoric compression or expansion. The isochoric process is given by :

$$p^{-1} \cdot T = cte. \quad (12)$$

The heat exchanged with the oil can again be calculated with Eq. (9) and (10).

6. *Polytropic exponent*

The polytropic exponent *n* of the compression process, strongly depends on the volume ratio of the compressor and the losses during the compression process. The exponent will thus change with changing revolution speed. If the volume occupied by the oil and the volume change as given by the volume curve are taken into account the change of the exponent was determined for the studied compressor. The losses caused by leakage and friction depend on the revolution speed of the elements. Following [3] the leakage losses diminish with growing speed. The frictional losses on the contrary grow with it. For the studied compressor the correction coefficient for the exponent was calculated.

7. *Oil heating in the bearings*

Part of the oil injected in the compressor is also used for lubrication of the bearings. This causes the oil to heat. As a result an extra amount of heat is introduced into the cooling system. About 7% of the total oil flow rate is used for lubricating the bearings. This is taken into account by calculating the temperature rise of the oil in the bearings, according to the heat dissipated as given in Table 1. The oil injection temperature is then calculated as the weighted average of the heated and fresh oil flow. The nominal oil temperature entering the compressor is 50 °C.

8. *Model validation*

In Table 2 the calculated and measured results for a specific compressor type are given. The inlet temperature is taken to be 20 °C, the inlet pressure 101325 Pa. The outlet pressure is 7 bar. If the polytropic exponent *n* is known, then the only parameter to be taken into account is the cooling effectiveness, which is set constant to 0.5.

The difference of the measured and calculated air temperatures is always lower than 0.25 °C. The model is thus sufficiently accurate for the following study.

	male	female	male	female
	inlet bearing	inlet bearing	outlet bearing	outlet bearing
[rpm]	*[W]*	*[W]*	*[W]*	*[W]*
900	17,4	12,2	56,3	52,9
1100	23,6	16,2	76,7	72,5
1500	37,7	25,5	124	119
2000	58,7	39	196	188
3000	111	72,1	374	362
3615	148	96	504	490
4000	174	112	593	578
4260	193	124	657	641
4800	233	149	796	778

Table 1 : Heat dissipated to the lubrication oil

n_{rot} *[rpm]*	T measured *[°C]*	T calculated *[°C]*
900	53,96	53,93
1100	55,04	55,07
1500	57,20	57,23
2000	59,90	60,09
3000	65,30	65,48
3615	68,62	68,71
4000	70,70	70,71
4260	72,10	72,05
4800	75,02	74,79

Table 2 : Validation of the model

Simulation Results

1. Air temperature change during compression

In Figure 7 the temperature change of the air and oil during compression is shown as function of the revolution angle of the male rotor, for 2000 rpm. Oil temperature is shown from the moment injection starts. Before injection the air temperature rises steeply. At the start of injection, the air temperature is still lower than the oil temperature. This causes the air temperature to raise even more steeply. This is a phenomenon of short duration.
Due to the compression the air is heated and the air temperature rises above the oil temperature. From now on the two curves lie close together. The moment the oil injection stops, the oil and air temperature start to rise more steeply. The final temperature jump is caused by the isochoric compression.

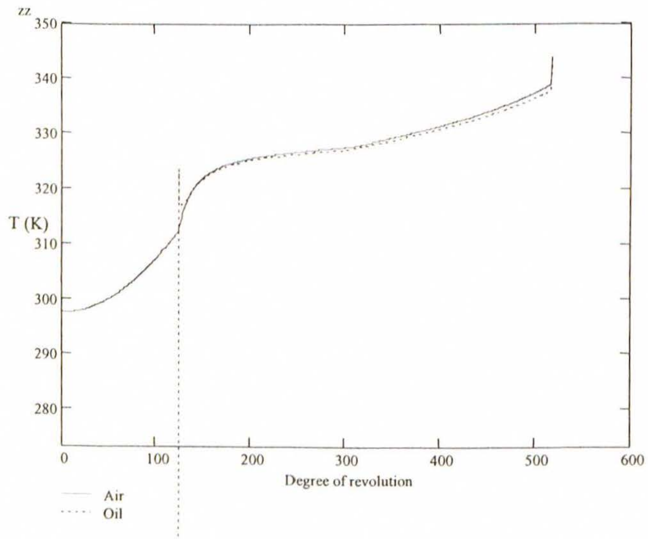

Figure 7 : Oil and air temperature during compression

2. *Oil flow rate*

Raising the oil flow rate also raises the volume ratio. Secondly the air will be better cooled. In Figure 8 the specific work of the compressor is shown as function of the injected mass flow rate of oil, at a constant revolution speed of 2000 rpm. Raising the oil flow rate with 10%, lowers the specific work with 0.2%. For other revolution speeds the same effect is observed. The outlet temperature of the air though is clearly diminished.

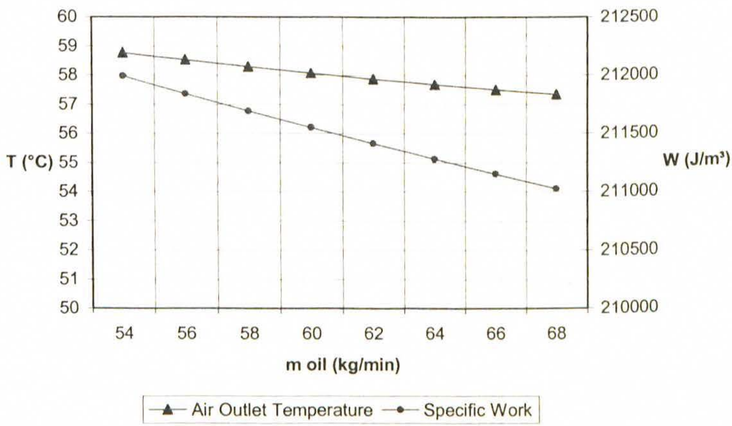

Figure 8 : Influence of oil flow rate

3. *Oil injection temperature*

Lowering the oil injection temperature (not taking into account the atomisation) will result in better cooling of the compressor air. As shown in Figure 9, for 2000 rpm and 4000 rpm, the specific work goes down as the injection temperature is lowered. Lowering the oil temperature by 5 °C, lowers the specific work by 0.5%. Lowering the oil temperature however will result in bigger heat exchangers for cooling of the oil.

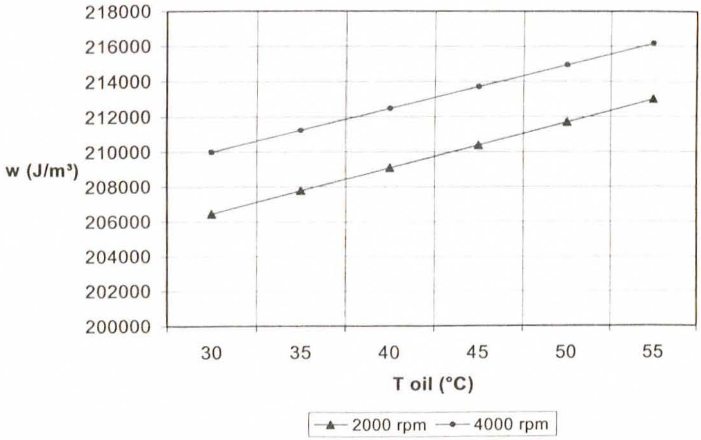

Figure 9 : Influence of oil injection temperature

4. Oil atomisation

The effect of oil atomisation can be modelled by raising the cooling effectiveness in the calculations. In Figure 10 the specific work for 2000 rpm, with changing effectiveness is given, as well as the air exit temperature. If the effectiveness is raised with 10 points, the specific work only lowers with 0.05%. The atomisation of oil has thus very little influence on the performance of the compressor. The reason for this is that the maximum cooling which can be obtained, is reached at equilibrium. As the real oil and air temperatures only slightly differ, the raising of the cooling effectiveness will have little effect.

5. Point of injection

As previously shown the oil temperature at injection can be higher than the air temperature. This can be avoided by moving the injection ports to a higher pressure. In Figure 11 the air exit temperature and the specific work is shown for changing point of the injection, being the position of the male rotor. There is very little influence of injection point on performance. The optimum specific work is reached at the current injection point, so no amelioration can be found by changing the injection point.

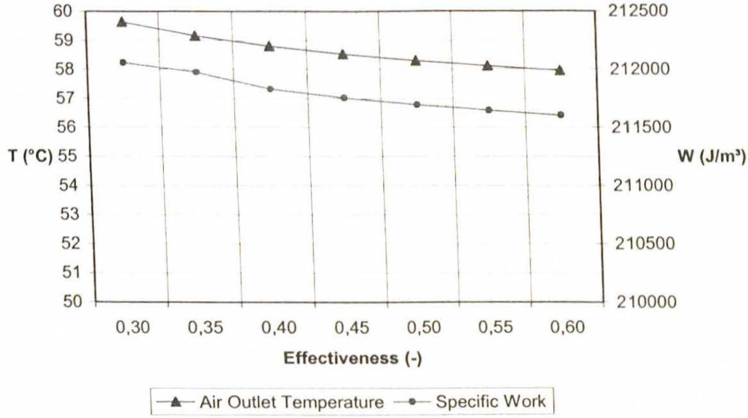

Figure 10 : Influence of oil atomisation

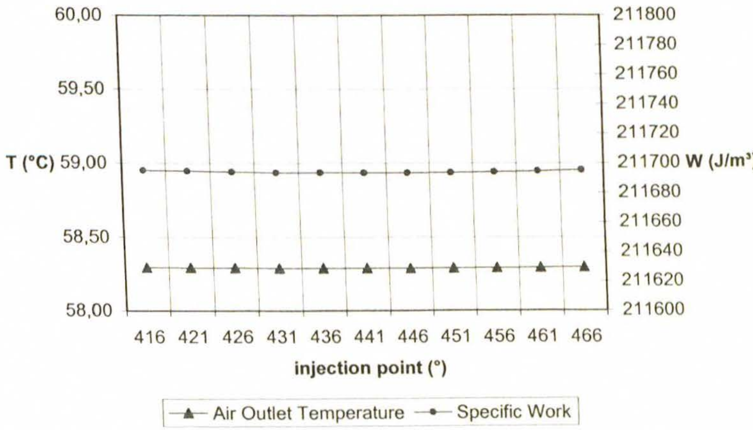

Figure 11: Influence of injection point

6. *Bearing lubrication*

In the modeled compressor, the oil heated by the bearing lubrication is mixed with the oil coming directly from the oil-cooler, before injection. If this oil is not send through the compressor, but directly to the cooler, the heat released by the bearings could be taken out of the compression process. In Figure 12 the two cases, with introduction and without introduction of lubrication oil are shown as function of revolution speed.
The difference in compressor performance is not high. Only for larger speeds the effect can be detected. About 0.12% less energy is needed.

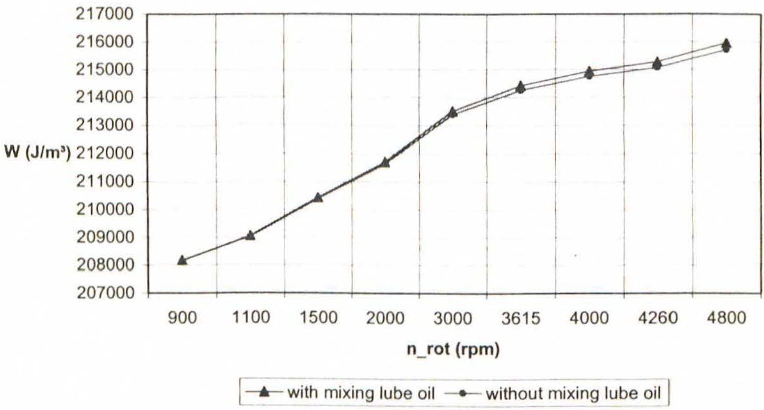

Figure 12 : Influence of bearing lubrication oil

Conclusion

In this paper the performance of the oil cooling in a oil injected screw compressor is analysed.

Experiments on a test rig, show that oil atomisation can raise the cooling effectiveness of oil by 10 point. The most important reason is that atomisation produces smaller droplet diameters and thus a better heat transfer.

A thermodynamic model was developed of the compression process in the oil injected screw compressor. With this model different parameters influencing the cooling inside the compressor are studied.

Oil atomisation has a very small effect on compressor performance. As the cooling process is spread out along the compression process, the importance of the injection performance is rather limited.

Injecting oil at a lower temperature or, especially for higher revolution speeds, separating lubrication oil from injected oil, gives a small gain.

Isothermal compression is consequently not easily reached by using oil injection.

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