Design, Testing and Feasibility Analysis of an Oil-Free Twin Screw Compressor with In-Chamber Flash Cooling

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Abstract. With the current aim of reducing CO2 emissions worldwide compressor efficiency is more in focus than ever before. An estimated 20% of all electricity generated worldwide is used for compressing air. Oil injected twin screw compressors form a large percentage of the total. The injected oil is used for cooling, as well as for sealing and lubrication of the rotors. Typical pressure ratios lie between 8:1 and 16:1. There are many investigations being conducted into oil-free screw technology for specific applications requiring no oil carry over. In this presentation we will take a close look into the possibility of producing a single stage oil-free twin-screw compressor operating at a pressure ratio of 8:1 and using evaporated water for cooling. The main objective in this case is to improve compressor efficiency. Additionally, we will compare efficiency results with a state-of-the-art single stage oil injected twin screw compressor. Previous results from an earlier paper by the author and team from City University of London on CFD analysis of a similar system are taken as the basis. Further simulated calculations using SCORG for the actual case in question allow preliminary performance analysis using the multi-domain chamber model. From simulation we can see that the oil drag force acting on the rotors forms a significant part of the total power requirement. It is therefore suggested that an oil-free solution will achieve improved specific power. To confirm this concept, verify the simulated calculations and understand the heat transfer process in the compression chamber more fully, experimental analysis was undertaken.

1. Introduction

The object of this work is to design a compressor which will produce better efficiency over the specified operating range than is currently achievable from a typical oil injected compressor. This is a market driven requirement and is considered to be an important development step for the screw compressor by Jaecklin GmbH. During this stage of the project we would like to produce a stable running prototype providing data to be used for further development. This is a big challenge and the design will be a compromise between high efficiency, low manufacturing cost and high reliability. Additional targets are low noise, low rpm, high start/stop capability, low operational temperature and good air quality.

Certain design decisions were made in advance taken from the above list of requirements. A single stage solution is the only commercially acceptable possibility. The size of the compressor will be in the range of 22 kW up to 45 kW. Direct drive is required with variable speed up to 6000 rpm. The designed pressure ratio will initially be 8:1 which may be extended up to 11:1 during future development.
Since cooling during compression will be a major challenge in the oil free screw compressor, various alternatives were considered. The possibility of using impinging jet rotor cooling, commonly referred to as flash cooling, has an advantage of reducing compressor temperature at the same time reducing fluid injection quantity. It was suggested that a reduction in injected fluid induced rotor drag might lead to reduced power and possibly to higher compressor efficiency.

The challenge of removing oil from an oil-injected compressor is expected to provide the following effects. Possible increase in the leakage between rotors and housing may result. Reduced lubrication properties between the rotors may cause reliability issues. The suggested advantage of reduced drag generated between the rotors and housing was numerically simulated using SCORG multi-domain chamber model and the results indicate a power saving of 17% at the target operating condition. The increase in leakage will be measured by comparing the volumetric efficiency of both the oil injected and oil free versions and the results used for designing optimised rotors for future evaluation. For the comparative tests both oil injected and oil free versions use exactly the same rotors and the clearances in both compressors are identical. At this stage it was not clear how effective the flash cooling would be. The SCORG simulation model predicted a gas temperature at the outlet of 156°C.

A two stage dry running solution was also considered and rejected on the basis of increased complexity and manufacturing cost. A dry running single stage twin screw compressor is normally limited to a pressure ratio of 3:1 and for this reason was not considered to be a suitable solution for the specified 8:1 pressure ratio. The high gas temperature produced during the compression process deforming the rotors and housing, and at a pressure ratio of 8:1 leading to damage or eventual seizure. This was confirmed during a CFD analysis commissioned by Jaecklin GmbH at an earlier stage of the project. Nevertheless, an interesting development taken from dry running compressors was considered as a possible solution for cooling the oil free rotors in our case. Referring to a previous paper, Stosic et al 2011, Numerical and experimental research in heat transfer to screw compressor rotors, we can see that the pressure ratio of a dry running single stage compressor could be extended considerably by injecting minute quantities of a volatile liquid such as water into the housing at the high-pressure port end in any circumferential position. Experimental results from the paper confirm that the theory of flash cooling works effectively up to a pressure ratio of nearly 6:1. The conclusion from this previous work tells us that the liquid would then impinge on the rotors at a rate which would be instantly evaporated on contact with them. This concept was taken over and analysed more fully as a suggested solution for a single stage oil-free compressor with an operating pressure ratio of 8:1.

![Figure 1](image_url): Shows the CFD thermal conduction calculation at a pressure ratio of 12:1. The impinging jet cooling is directed at both rotors as close to the discharge end as possible.
The results of a more recent paper, *Stosic et al 2018 numerical investigation of water evaporation in twin screw compressors* provide a numerical study into the feasibility of using flash cooling to operate a single stage compressor up to a pressure ratio of 12:1 by using a very small amount of water injection, just enough to provide cooling by evaporation. A numerical procedure was used to estimate the mass of water required to produce saturated air at delivery pressure. A multiphase CFD model was set up to solve air and water-liquid flow along with a simplified evaporation model to account for the latent heat of evaporation. The main physical effect is rotor cooling from heat exchange produced by the impinging water jet. The conclusion tells us that flash cooling operates more efficiently at 4500 rpm than at 6000 rpm due to the longer time that the injected water mass has to provide heat transfer. Nevertheless, the numerical results from this paper show that when the water mass is selected to provide full evaporation at a pressure ratio of 12:1 the exit temperature exceeds 300°C therefore thermal expansion in the machine would lead to rotor damage unless the injected water amount was higher than that required for evaporation only. It should be emphasised that the main difference between impinging jet rotor cooling and fluid flooding is the quantity of fluid used and the exact injection position. The minute quantity of water used evaporates on contact with the rotors. The rotors are therefore not subjected to a fluid induced drag force during operation. This effect was confirmed in CFD calculations and during comparison of numerical models in SCORG.

The compressor chosen for the comparative tests was an oil injected machine designed by Jäcklin GmbH. This particular machine had been previously measured in the CUL test laboratory in London.

| Table 1: The compressor unit used for this experiment was 4/5 male driven configuration with the following dimensions: |
|-------------------------------------------------------------|
| **Male rotor diameter** | 130 mm |
| **Length/diameter ratio** | 1.36 |
| **Female rotor diameter** | 102 mm |
| **Centre distance** | 90 mm |

By using known test results as input data for SCORG an accurate case for the oil injected compressor was modelled. After completing and verifying the oil injected SCORG model, the oil injection was removed and replaced with flash water cooling in the multi chamber model. No other changes were made in the numerical model at this time. In particular the simulated influence of water drag on the rotors compared to oil drag, the calculated leakage rates and the predicted volumetric efficiency without oil would be investigated. The measured results from the oil free compressor would be compared to the results taken from the oil injected compressor of identical design. To eliminate any other design influences from the comparison both reference machines use the same rotors, the same clearances, and the same basic housing design. Modifications were made to the oil free compressor to provide flash cooling. Sealing of the compression chamber from the bearing chamber at the high-pressure end using proprietary shaft seals was considered necessary to prevent the ingress of water vapour into the bearings.
2. Oil injected compressor validation of numerical simulation model.

Table 2: Shows the measured results taken over a speed range from 1500 rpm up to 6000 rpm. The measurements were made in accordance with ISO 1217. Inlet conditions were 1000 mbar and 20°C. The pressure ratio was 8:1.

| Shaft Speed (RPM) | Air Flow (m³/min) | Power (kW) | Specific Power (kW/m³/min) |
|------------------|-------------------|------------|-----------------------------|
| 1500             | 1.72              | 10.18      | 5.90                        |
| 2000             | 2.31              | 13.59      | 5.89                        |
| 2500             | 2.95              | 17.42      | 5.91                        |
| 3000             | 3.60              | 20.93      | 5.81                        |
| 3500             | 4.19              | 24.53      | 5.86                        |
| 4000             | 4.80              | 28.40      | 5.91                        |
| 4500             | 5.39              | 32.34      | 6.00                        |
| 5000             | 6.05              | 37.04      | 6.12                        |
| 5500             | 6.63              | 41.51      | 6.26                        |
| 6000             | 7.29              | 46.60      | 6.39                        |

The measurement was made in the CUL laboratory in London. The results were used for verification of the SCORG multi-chamber model during the modelling and simulation process. All input data for the simulation model was taken from the actual compressor and measured data. The numerical results from the SCORG simulation case show an extremely good correlation with the measured results taken from the oil injected compressor over the entire performance range of this machine (Figures 2&3).

Table 3: Simulated performance calculation for the identical operating conditions used during testing.

| Speed rpm | Vol m³/m | M kg/min | Vol Eff. | Power kW | Pspec | Ad Eff. | Psuc bar | Tsuc degC | Pdis bar | Tdis degC | Moil kg/sec |
|-----------|----------|----------|----------|----------|-------|---------|----------|-----------|----------|-----------|-------------|
| 1500      | 1.66     | 1.98     | 0.81     | 10.10    | 6.05  | 0.78    | 1        | 20        | 8        | 63.41     | 1.12        |
| 2000      | 2.28     | 2.72     | 0.83     | 13.60    | 5.94  | 0.79    | 1        | 20        | 8        | 64.26     | 1.14        |
| 2500      | 2.91     | 3.45     | 0.85     | 17.23    | 5.92  | 0.79    | 1        | 20        | 8        | 65.12     | 1.15        |
| 3000      | 3.53     | 4.19     | 0.86     | 20.99    | 5.94  | 0.79    | 1        | 20        | 8        | 65.98     | 1.15        |
| 3500      | 4.15     | 4.94     | 0.86     | 24.86    | 5.98  | 0.79    | 1        | 20        | 8        | 66.83     | 1.16        |
| 4000      | 4.78     | 5.68     | 0.87     | 28.83    | 6.02  | 0.78    | 1        | 20        | 8        | 67.68     | 1.16        |
| 4500      | 5.40     | 6.24     | 0.87     | 32.89    | 6.08  | 0.77    | 1        | 20        | 8        | 68.53     | 1.16        |
| 5000      | 6.03     | 7.17     | 0.88     | 37.04    | 6.13  | 0.77    | 1        | 20        | 8        | 69.37     | 1.16        |
| 5500      | 6.66     | 7.92     | 0.88     | 41.26    | 6.19  | 0.76    | 1        | 20        | 8        | 70.20     | 1.16        |
| 6000      | 7.29     | 8.67     | 0.88     | 45.54    | 6.24  | 0.75    | 1        | 20        | 8        | 71.03     | 1.16        |
In the right-hand column, the quantity of injected oil in kg/sec can be seen. This corresponds to 66 l/min at 4500 rpm. The measured amount was 69.8 l/min at 4500 rpm. This is a reasonable comparison.

**Figure 2:** The air flow delivery in m³/min.

**Figure 3:** The power comparison in kW.

**Figure 4:** The simulated volumetric efficiency.

**Figure 5:** The simulated power loss (%) due to oil drag.

In figure 4 the volumetric efficiency taken from the simulated model of the oil injected compressor is illustrated. At 4500 rpm the measured value was 83.6 % the simulated model shows 87.9 % at the same point. The difference will be investigated.

The total shaft power in kW calculated in SCORG is the sum of the indicated power, shaft seal power, bearing power and oil drag power. In the oil injected compressor, the oil drag power (kW) forms a considerable part of the total sum. By calculating the oil drag value in kW for each rpm step from 1500 rpm up to 6000 rpm and comparing it to total shaft power (kW) the percentage difference can be determined and plotted in figure 5.

The result illustrates that the oil drag expressed as a percentage of the total shaft power increases from 10% at 2000 rpm up to 20% at 6000 rpm. It can be seen that the drag losses increase with speed so the power saving between 4500 rpm and 6000 rpm is particularly interesting. From the previous CFD numerical study *Stosic et al 2018 numerical investigation of water evaporation in twin screw*
Compressors we know that flash cooling is likely to be more effective at 4500 rpm than at 6000 rpm due to the longer time that the injected water mass has to provide heat transfer. In consideration of these two factors it was decided to concentrate on 4500 rpm for the initial comparison of prototypes on test. This decision was also influenced by a restriction on test resources. The oil free test rig has a maximum motor rpm limit of just over 4500.

3. The oil free compressor numerical simulation case.

The procedure followed for simulation of the oil free compressor was to copy the oil injected model and remove the oil injection completely replacing it with flash cooling. All other input parameters remain unchanged and a reasonable result was achieved. Due to uncertainty of actual clearances required it was decided to leave interlobe clearance, radial clearance and axial clearance at 0.05 mm, the same values as in the oil injected machine. At this time no form of optimising of the model was undertaken. The comparative results after testing would be used for gathering data to be included in the final design. A fully optimised oil free version will be designed at the next stage of this project.

It was anticipated that oil free cooling of the rotors would be a challenge and it would be required to keep the compression temperature low to avoid thermal deformation and contact between rotors and housing. The flash water injection in the model uses 0.340 l/min at the test conditions. The modelled result produces 160°C outlet port temperature and a volumetric efficiency of 91% which compares favourably with the 88% modelled and verified in the oil injected version. Why the oil free model shows less leakage is unclear, but this will be examined during later verification.

In the right-hand column, the quantity of injected water in kg/sec can be seen. At 4500 rpm this corresponds to 0.34 l/min and the specific power (Pspec) is 5.5 kW/m³/min. By comparison the validated oil injected model indicates a specific power of 6.1 kW/m³/min under the same conditions.

| Speed rpm | Vol m³/m | M kg/min | Vol Eff. | Power kW | Pspec | Add Eff. | Psuc bar | Tsuc degC | Pdis bar | Tdis degC | Mwat kg/sec |
|-----------|----------|----------|----------|----------|--------|----------|----------|-----------|----------|-----------|-------------|
| 1500      | 1.59     | 1.89     | 0.77     | 10.60    | 6.64   | 0.71     | 1        | 20        | 8        | 156.07    | 0.0054      |
| 2000      | 2.27     | 2.69     | 0.82     | 14.04    | 6.19   | 0.76     | 1        | 20        | 8        | 159.59    | 0.0054      |
| 2500      | 2.94     | 3.49     | 0.86     | 17.45    | 5.93   | 0.79     | 1        | 20        | 8        | 157.63    | 0.0055      |
| 3000      | 3.60     | 4.27     | 0.87     | 20.81    | 5.78   | 0.81     | 1        | 20        | 8        | 161.38    | 0.0056      |
| 3500      | 4.27     | 5.07     | 0.89     | 24.22    | 5.67   | 0.83     | 1        | 20        | 8        | 157.69    | 0.0057      |
| 4000      | 4.93     | 5.86     | 0.90     | 27.64    | 5.59   | 0.84     | 1        | 20        | 8        | 155.79    | 0.0057      |
| 4500      | 5.59     | 6.65     | 0.90     | 31.08    | 5.55   | 0.85     | 1        | 20        | 8        | 156.46    | 0.0057      |
| 5000      | 6.25     | 7.43     | 0.91     | 34.54    | 5.52   | 0.85     | 1        | 20        | 8        | 158.91    | 0.0057      |
| 5500      | 6.91     | 8.22     | 0.91     | 38.02    | 5.49   | 0.86     | 1        | 20        | 8        | 162.45    | 0.0058      |
| 6000      | 7.56     | 8.99     | 0.92     | 41.50    | 5.48   | 0.86     | 1        | 20        | 8        | 168.86    | 0.0058      |
4. Experimental investigation.

| Table 5: Measured results comparison, the difference between the specific power (kW/m³/min) measured at the block outlet in comparison to the compressor outlet is due to the oil separation pressure loss. |
|--------------------------------------------------|------------------|------------------|
| Input speed | 4500 rpm | 4500 rpm |
| Volume delivery | 5.39 m³/min. | 5.0 m³/min. |
| @20°C /1000 mbar | @20°C /1000 mbar |
| Shaft power | 32.34 kW | 29.2 kW |
| Working pressure | 8 bar(a) | 8 bar(a) |
| Suction pressure | 1 bar(a) | 1 bar(a) |
| Pressure ratio | 8:1 | 8:1 |
| Outlet temperature | 76°C | 100°C |
| Specific power block only | 6.0 kW/m³/min. | 5.8 kW/m³/min. |
| Specific power at compressor outlet | 6.75 kW/m³/min. | 6.0 kW/m³/min. |

Comparison of results and discussion.

Volumetric efficiency: Comparing the results at 4500 rpm which is 30 m/sec tip speed the air delivery of the oil free version is 7% less than the oil injected version. The reason for this could be increased leakage between rotors and housing due to removing the oil. The simulation results show a different situation so this point will need to be examined further. In the oil free version, the clearances are more critical in the sense that dimensional accuracy influences leakage fluctuation. Reducing clearances without compromising reliability will be investigated in the next prototype.
Cooling: In this experiment 0.0057 kg/sec water was injected into the compression chamber. In comparison the oil injected version uses 1.1645 kg/sec oil. The flash cooling effect was stronger than expected. The exit gas measured 100°C whereas the simulation model predicted 165°C. The difference between simulated and test results will be investigated and used to verify SCORG for these conditions. A possible explanation is that in the simulation not all of the injected water actually evaporates. At 7 bar evaporation occurs at 170°C and the test compressor is running cooler than this. During operation all compressor parts record a temperature of less than 100°C so there is no risk of damage due to inadequate clearance. The maximum temperature occurs at the exit port close to the temperature sensor. This is as close as possible to the point of maximum compression. It is assumed that the rotor temperature is less than the gas temperature due to the impinging jet water cooling.

The hot gas/water mixture exiting through the compressor outlet pipe goes straight into an aftercooler and the water content condenses. Because the injected water mass is extremely low compared to a fluid flooded compressor the separation process is much easier and requires no special separation equipment. The external housing temperature was between 45°C and 65°C and the bearing cover reaches a maximum of 85°C in the area directly adjacent to the exit gas. Further away from the exit port the temperature reduces to 65°C.

Power saving: A reduction of 10% at 30 m/s tip speed was measured. The simulation shows the power saving is greater as the tip speed increases. It was not possible to test this effect at higher rpm due to the test rig limitation. The numerical simulation indicates that power loss due to water drag in the flash cooled compressor is close to zero over the entire speed range. This has not been confirmed in the provisional test results to date.

Specific power: The specific power improvement predicted from simulation was - 6.08 reducing to 5.55 kW/m³/min. an improvement of 8% for the bare block only. In fact, the measured value was 6.0 reducing to 5.8 kW/m³/min an improvement of 3%. Comparison of results measured at the compressor outlet show 6.75 reducing to 6.0 kW/m³/min an improvement of 11%. Despite the measured reduction in power the volumetric efficiency also reduced in the experiment so the predicted advantage was not fully achieved. The net result is a significant gain in overall efficiency. More testing time for further optimisation is required. Considerable potential remains to reduce the clearances and increase the volumetric efficiency which in turn will improve the adiabatic efficiency. These experiments will continue during 2021. A new prototype is being produced with the intention of correcting these issues. Comparison over the entire rpm range and verification of results will then be possible.

Uncertainty of measurement: The oil injected compressor was measured in the City University Laboratory and the results reported in accordance with ISO 1217 for bare screw compressors. The simulation case was built around these results. The correlation between measured results and numerical simulation was good and provides an accurate basis for this work.
5. Conclusions
The result confirms that flash cooling can be used most efficiently in a screw compressor up to a pressure ratio of 8:1. A reduction in power has also been confirmed. A measured efficiency improvement of 3% has been achieved. The numerically predicted 8.7% specific performance improvement has not been achieved at this time. The likely reason for this is higher leakage rate in the experimental case than numerically predicted. The numerical study indicates increased oil drag with increasing speed. The leakage reduces with increasing speed so higher rpm may improve efficiency. Testing up to 6000 rpm is planned for confirmation.

There are two advantages which have been confirmed by testing. An additional 10% improvement in efficiency for the whole compressor unit is the result of having no oil separation equipment. The second advantage is that any condensation produced after cooling is reused by the flash cooling system. There is no need to separate out and collect the condensation. The separation system in an oil injected compressor is normally required to prevent the oil from becoming contaminated by condensation produced during the compression and cooling process. The condensation produced in an oil injected compressor normally contains a small amount of oil carry over which makes disposal in line with environmental restrictions costly. In the flash cooled compressor these issues do not exist and there is no negative environmental impact. Certain applications for Jäcklin GmbH customers may be interested in this advantage.

6. Acknowledgements
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