Thermal expansion in liquid-injected screw compressors

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Abstract. In screw compressors gaps between rotors and casing have to be small, in the range of 30 to 300 µm. These gaps are influenced by thermal expansion when the screw compressor is in operation. The amount of thermal expansion is about the same size as the gaps, even in oil-injected screw compressors. Especially when starting a cold compressor, there are temperature differences. These temperature differences were measured in a 22 kW oil-injected screw compressor for industrial compressed air as an easily manageable example. Measuring the temperatures in the rotor teeth was done with Pt1000-sensors and transmission with Wireless-LAN. The results showed lower temperature differences than expected. The influence of the oil-injection-temperature is dominant. In bigger screw compressors the temperature differences tend to be larger because of dimensional reasons: A simple numerical model for the temperature differences in a screw compressor with fluid injection allows an extrapolation for larger rotors.

1. Introduction
The subject of this investigation is liquid-injected screw compressors. These include the oil-injected screws for industrial compressed air, the widest spread air compressor in industry. And there are also the process gas screw compressors in the chemical industry, often very large machines. They often have an injection with water, oil or a process liquid.

Figure 1. View on an oil-injected screw-compressor block.
Figure 1 shows such a screw compressor block. There are no special seals for the working chambers, only the littleness of the gaps gives the sealing effect. The gap size is about 30\(\mu\)m in small compressors and up to 300\(\mu\)m in big compressors. Gaps too big cause increased leakage between the working chambers and lower efficiency, gaps too small cause jamming and often total damage of the screw compressor block. The amount of thermal expansion is about the same size as the gaps. As long as rotors and the casing have the same coefficient of thermal expansion and their temperatures are rising and falling simultaneously, the gap widths remain practically unchanged. But that is not secured. In order to clarify that, this investigation was made. As we can see in figure 1, the teeth of the gate rotors are slimmer than those of the main rotor. As thin parts are more prone to fast heating and cooling, the investigation concentrated on these teeth and on the casing.

2. Type of screw compressor

The machine for the tests was a typical oil-injected screw compressor for industrial air with the type name "S29", with 29 HP shaft power, fixed-speed (about 4200 r.p.m. for 10 bar (g)), and cooled with ambient air, figure 2. The screw compressor block casing is made of grey-cast iron; the rotors are of spheroidal cast iron (practically the same coefficient of thermal expansion for both).

It was chosen because of being typical and manageable for a test: not too big, what had caused higher costs, not too small, allowing standard temperature probes. Figure 3 shows the scheme of air and oil flow. The oil is separated from the compressed air, cooled and recirculated. The machine starts on compressed-air demand and shuts down, when air demand is satisfied. Starting and stopping can be repeated many times per hour. The heating of the machine comes from the heat of compression and from losses.

![Figure 2. View of opened compressor.](image-url)
3. Way of measurement

The thermal behaviour of the casing was measured in the standard way, with small thermocouples, embedded in tiny holes drilled into the casing, Figure 4, and fixed with adhesive.

The measurement of the temperature in the gate rotor teeth needs small temperature probes, in order not to influence the heat flow. So small Pt1000-probes in small drilled holes, Figure 5, fixed with thermally conductive (0.1 W/m*K) adhesive are suitable. The thermo-signal voltage is small and needs transformation into a rugged signal before being sent from the rotating rotor to the recording device. After comparing different methods the use of an electronic device, rotating with the rotor and sending the temperature signal in digital form by WLAN to the recording device, was chosen. This resulted in low costs (almost independent of the number of temperatures) and reliable transmission for
several temperatures (If the transmission is disturbed, the results are not falsified; either they are transmitted correctly, or there is no signal). The supply voltage is transferred with the single contact pin and a carbon brush, Figure 6.

![Figure 5: position of the temperature probes in the gate rotor.](image)

![Figure 6. Rotating WLAN.](image)

The rotating parts are enclosed with a plastic part (manufactured as a turning workpiece), keeping the pressure and the oil within and allowing the WLAN signals to be transmitted. The WLAN-signals were received and recorded with a standard personal computer. The signal transmission rate was 1 record of temperature data per second. A higher transmission rate is possible, but wasn’t necessary in this case. The supply of electric energy through the contact pin (with closing the electric circuit through the rotor ball bearings and the casing) is not suitable for long-time measurements because of wear; in that case a supply through induction would be better. The monitoring of the temperature at the electronic devices, rated for a maximum 85°C, showed a temperature more than 10 K lower. The number of temperature measuring points in the rotor is limited only by the number of holes, which can be drilled without weakening the rotor, so 6-20 would be possible.
4. Results
The maximum temperature differences between rotor tooth and casing were recorded when starting
the cold compressor. The oil-injection temperature shows a spike at 13:23 (figure 7), caused by a delay
in the thermostat. When the oil temperature reaches 60°C, the thermostatic valve does not open
immediately; it takes about half a minute, until the wax is molten and the valve open. Before the
thermostatic valve opens, the temperatures increase quickly, after that, they are rising slower, and after
a quarter of an hour, they almost reach a steady state. Only the casing temperature at the low pressure
end of the rotors and the gate rotor tooth temperature at the low pressure end show a much slower
reaction. Because the oil is injected in an already closed working chamber, oil reaches these points
only indirectly and in low quantity. After stopping the compressor, the temperatures are converging
and then returning to ambient temperature slowly.

![Figure 7. Temperatures recorded during star-stop-cycle (here and in Figure 9 and 10: horizontal axis – time hh:mm:ss, vertical axis – temperature in °C).](image)

The temperature spike at 13:23 is in the outlet temperature and the gate rotor temperature at the high
pressure end, too, what shows a very strong influence of oil inlet temperature on the whole screw
compressor block. The casing temperatures show a bigger inertness (slower reaction / more delay)
compared with the corresponding rotor tooth temperature, but the difference is lower than between
high-pressure-end and low-pressure-end.
The test runs were repeated several times, and the repeatability was < 2 K.
The temperature differences between gate rotor tooth and casing reach their maximum at about 1 to
2 minutes after starting, during the temperature spike, with some 12 K. At steady state the difference is
only 2 K (high-pressure end) or 4 K (low-pressure end).
When computing the diameter change and the gap change resulting from temperature differences (gate rotor - casing) at the high-pressure-end and at the low-pressure-end, the maximum gap change is only approximately 8 µm.

Conclusion: The temperature difference between rotor tooth and casing is lower than we expected for medium-size oil-injected screw compressors, and the main influence on temperatures is from the oil thermostat.

5. Extrapolation:
Temperature differences are important for bigger compressors, too. Here we find a dependence on length L. When the lengths L of a screw compressor are redoubled, the heat flux through a surface is 4-times higher, being proportional to L², the area. The heat flux through a volume, like for example a cube-like block is 2-times higher (because the surface is 4-times higher, and the gradient of temperature, ∆T/L, is halved), proportional to L. The thermal capacity is 8 times higher, being proportional to the mass, and therefore proportional to L³. As a result, the temperature influence on gaps might be more critical with bigger machines. So the test should be repeated with a big screw – but the costs for an experiment with a 200 kW-machine are prohibitive.

An extrapolation method for temperatures could be helpful, even without an experimental proof. Using words like "thermal capacity" and "heat conductivity" points to an analogy with electric circuits with capacitors and resistors, figure 8. The temperature (corresponding to the voltage U in the model) at a rotor surface spot varies during rotation, as it is exposed to different states of the working chamber: hot when the content of the working chamber is near to full pressure, cold when the working chamber is drawing in fresh air. So an effective medium temperature Teff has to be found which can be modelled as, for example, x % of intake temperature, y % of oil injection temperature and (100-x-y) % of outlet temperature.

![Figure 8. Model for extrapolation of rotor tooth temperature.](image)
Factors for effective temperature and heat flow resistance were found by an iterative process: comparing the results of the model with measured results in a diagram, changing factors until the model's results are getting better. The factors can be seen in [1].

Figure 9 shows the results of the model with those factors compared with the measurements. The temperature of the high-pressure gate-rotor tooth, when running, is almost the same in the model and in the measurement. Other temperatures show a deviation up to 3 K. These deviations can be explained by the fact that heat transfer is not constant; for example caused by changes in oil flow through gaps, when oil viscosity is lowered by higher temperature.

This model can be used for an extrapolation. When increasing all lengths with a factor 3, the heat capacity is 27-times higher. Heat-flow-resistance within the rotor and most probably at the high-pressure-end tooth is dominated by heat-flow through a volume, so these resistances are reduced by a factor 3. The heat-flow-resistance at the low-pressure side of the rotor is probably dominated by the heat transfer through the surface and so is reduced with a factor of 9. The power of a compressor of that size would be some 200 kW.

Figure 10 shows the differences in gate-rotor tooth temperatures for that case. The spike at the high-pressure-end rotor-tooth-temperature, when starting, practically disappears, the low-pressure-end rotor-tooth- temperature has a much bigger delay.

This model needs more calibration in different machines before being used as a standard, but it gives a first hint for thermal expansion of rotor teeth under the conditions:
- liquid-injected screw compressor, with "thermal capacity of injected liquid flow" > 2 x "thermal capacity of compressed gas flow"
- the liquid is injected into a closed working chamber, at a place that is not near to the suction-side rotor-end

The question might arise: Why not extrapolating the casing temperature as well ? In order to evaluate the gap changes when starting, both temperatures, of rotor tooth and of casing, are needed.
One answer is: The shape of a casing – wall thickness, stiffening ribs, oil channels - can differ very much, so an extrapolation for a casing is more uncertain than for a rotor. The second answer is: You can measure the heating-up of a casing rather easily, with a handheld thermo-element for example. That gives casing temperature curves. And this extrapolation (not forgetting the influence of the oil-thermostatic-valve, so the oil injection temperature curve should be measured, too) helps you with the rotor tooth temperature curves.

![Figure 10. Result of extrapolation to triple size](image)

6. Outlook
This consideration is not exhaustive - there exists not just a thermal expansion of diameters, but also some distortion of screws and casing, which may be evaluated with the help of a more complex model.

![Figure 11. Infrared image of screw block from Figure 2; 4 minutes after starting](image)
These measurements (including a series of IR-photographs at different instants of time, like in figure 11) could be used for the calibration of detailed computational models of liquid-injected screw compressors including heat transfer, thermal modelling, and thermal deformation for gap calculation. Boge does not intend that at the moment. Giving results of the measurements to other researchers is not planned, but possible.

This work was done in cooperation with the Ostwestfalen-Lippe-University-of-Applied-Sciences.

References
[1] Thermal expansion in liquid-injected screw compressors; pages 291-01 in: Proceedings Compressor Users International Forum 2016 (International Rotating Equipment Conference 2016 Düsseldorf), VDMA-Verlag, Frankfurt

Appendix

Heat capacity: \[ Q = V \Delta T \rho c = L^3 \Delta T \rho c \]

heat flow through volume: \[ \frac{Q}{t} = A \frac{\Delta T}{L} \lambda = L^2 \frac{\Delta T}{L} \lambda = L \Delta T \lambda \]

resistance to heat flow: \[ \frac{\Delta T}{\frac{Q}{t}} = \Delta T/(L \Delta T \lambda) = 1/(L \lambda) \]

(when dominated by resistance in the volume)

heat flow through surface: \[ \frac{Q}{t} = A \Delta T \alpha = L^2 \Delta T \alpha \]

resistance to heat flow: \[ \frac{\Delta T}{\frac{Q}{t}} = \Delta T/(L^2 \Delta T \alpha) = 1/(L^2 \alpha) \]

(when dominated by resistance of the heat transfer at the surface)

V volume; Q heat; t time; \(\Delta T\) temperature difference (temperature at same place and different moments for heat capacity; temperature at different sides of the volume for heat flow); \(\rho\) density; \(c\) specific heat capacity; \(L\) edge length of the cube (as an example for a geometric body); \(\lambda\) heat conductivity (of the cast iron in this case); \(A\) area of the surface at one side of the cube.