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Improvement of the volumetric and isentropic efficiency due to modifications of the suction chamber

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ABSTRACT

The flow of gas in the suction chamber of an air brake compressor was analyzed, and losses due to pressure drop and heat transfer were identified by CFD simulations. Based on the results of the calculations the geometry of the suction chamber was modified to investigate possible improvements. Due to simple design modifications the flow direction of the gas was changed, which resulted in a reduction of heat transfer and probably also in pressure drop. Compared to the original construction the modified design brought a measured improvement of the volumetric and isentropic efficiency by about 1 to 5 percent dependent on the speed of the compressor.

1. INTRODUCTION

Compression of a gas leads to a temperature increase of the gas. Because the compressed gas is thus hotter than the compressor walls, there is a heat flow from the compressed gas to the housing. From the housing this heat flows either to the surroundings, to the low pressure suction gas or to the cooling water, which flows through some sections of the compressor housing.

The heat flow into the low pressure suction gas and the associated increase of the effective suction temperature has two negative effects: It leads to a decrease of the volumetric efficiency and an increase of the specific power requirement.

Roughly one can state that the volumetric efficiency is proportional to the inverse of the effective absolute suction temperature

$$\lambda = \frac{\dot{m}_{eff}}{\dot{m}_{ideal}} \sim \frac{T_s}{T_{s,eff}} \quad (1)$$

and the specific power is proportional to the effective absolute temperature

$$w = \frac{\Delta h_s}{\eta} = \frac{1}{\eta} \cdot c_p \cdot T_{s,eff} \cdot \left(\Pi^{\frac{\kappa-1}{\kappa}} - 1 \right) \quad (2)$$

So a temperature increase of the suction gas from 300K to 320K (26.85 to 46.85°C) will lead to a decrease of volumetric efficiency and increase of the specific power by about 7%.

So one should try to minimise the head flow into the suction gas by two methods: On one hand the cooling water system should be arranged in a way that the heat flow from the cylinder and the exhaust chamber is intercepted, and secondly the heat transfer process on the inner wall of the suction chamber should be kept to a minimum.

In a similar way also the pressure loss of the suction gas in the suction chamber leads to negative effect on the volumetric efficiency and the power requirement.

In this paper it is described, how pressure drop and heat transfer within the suction chamber can be influenced.

2. INFLUENCES WHEREON THE SUCTION FLOW

A sketch of a small reciprocating compressor can be seen in figure 1. The cold suction chamber and the hot discharge chamber are usually right side by side just separated by the hot partition wall. The chamber bottom consists of the hot valve plate. The arrows in figure 1 show examples how to connect the suction line.

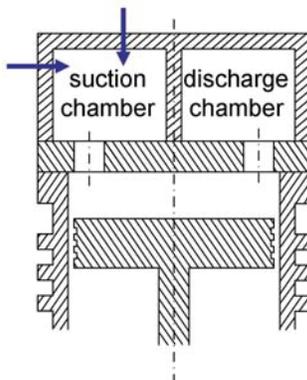


Figure 1: Sketch of a small reciprocating compressor head

In any case the connection of the suction line to the suction chamber has the form of a sudden extension. This leads to an intensive flow impact on the opposite wall or the valve plate respectively. In the area the turbulences are in contact with the surface the local heat transfer is increased by the accelerated flow and the suction gas is stronger warmed up at hot surfaces.

Another effect is a jet stream stall at the inlet and associated vortices. This leads to pressure loss by stall and vortices because these turbulences dissipate impulse from the flow. Furthermore is a vortex possibly not part of the sucked in fluid because it is a standing vortex for the reason that the flow is a pulsating flow.

3. THEORETICAL INVESTIGATIONS

To get a better knowledge of the fluid flow and the heat transfer in the suction chamber of these compressor numerical calculations were carried out. The commercial CFD package Ansys CFX was used to investigate and in progress of the investigation to optimize the flow and heat transfer situation in the suction chamber.

In detail, the flow in the suction chamber can be described as highly pulsating and turbulent.

The used numerical grids of the test geometries were generated with a directly resolved boundary layer. Hence, it is possible to simulate the boundary layer directly without the approximation of any implemented wall functions.

The boundary conditions were defined as follows: The mass flow through the suction chamber was implemented by prescribed mass flow – time – functions to Ansys CFX. These functions depend on the working frequency of the compressor and some boundary conditions. To calculate the heat transfer coefficient the well known equation for the area averaged heat flux is used.

$$\dot{q} = \alpha \cdot (T - T_{ref}) \quad (3)$$

The reference temperature T_{ref} is set to 293K and defines the temperature of the flow through the inlet. The temperature T on each wall of the suction chamber is assumed to be constant, too. To realise a heat transfer this temperature difference between flow and wall was usually set to 150K.

The length of the transient simulation was defined as two turns of the compressor, and the Courant Number was regulated to a value lower than five to find a compromise between calculation duration and precision. Test calculations have shown that two turns are necessary to simulate steady state initial conditions. The first turn is used to generate these initial conditions for the investigated second turn. To simulate more than two turns has shown no more significant changes in the numerical results.

The simulations of the unsteady flow field have shown the following basic results with the original geometry configuration. Due to high forced convection effects the temperature difference between wall temperature and inflow temperature has no significant effect on the heat transfer coefficient. The main influence on the heat transfer coefficient is caused by frequency changes of the compressor. In figure 2 the streamlines in a cross section area of the original geometry configuration are plotted. The current simulation time represents a specific time step when inlet and outlet are both open.

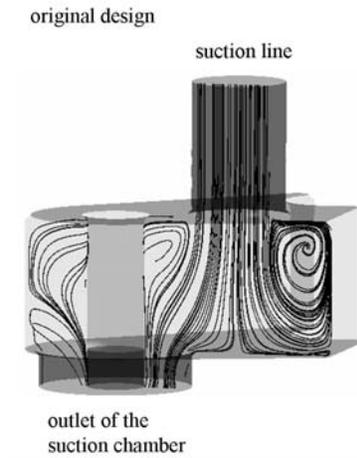


Figure 2: Original design of the suction chamber with simulated flow field

The analysis of that first step of the numerical calculation reveals a powerful impinging inflow jet straight to the lower wall. Obviously, this behaviour could be expected. Nevertheless, the result of that jet was not only a massive increase of the local heat transfer between the gas and the hot wall but also a pulsating flow field in the whole suction chamber. As another result the heat transfer at all walls was increased. A plot of all heat transfer coefficients can be found at the end of this section. Due to these effects it was suggested to optimize the original geometry, especially the inlet of the suction chamber.

As a first geometry variation the inlet was placed directly over the outlet. Therefore, the inflow and outflow direction is on the same axis. In figure 3 the streamlines of that first geometry variation are plotted in the same cross section area and time step as in figure 2. The boundary conditions were not changed and the simulation conditions can be seen as equivalent.

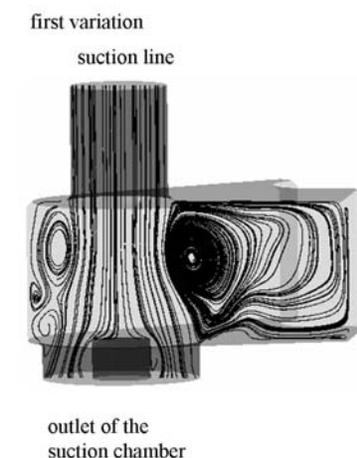


Figure 3: First design variation of the suction chamber with simulated flow field

Now, the streamlines of that inlet optimization are conducted straight from inlet to outlet as foreshown. During this procedure the jet is surrounded by a vortex. This vortex occurs in the whole suction chamber and causes a further rotation of the flow field beside the main jet. Nevertheless, this first optimization of the geometry leads to a significant decrease of the local and global heat transfer in the suction chamber. Especially, the impinging jet straight to the lower wall is nearly avoided.

The first variation has offered an attractive way to minimize the heat transfer with simple geometry modifications. To minimize the supply of the jet surrounding eddy another geometry suggestion was made. The intention is to reduce the contact between the jet and the surrounding flow field but not to loose suction chamber volume as compensation of gas oscillation. The second variation of geometry is a truncated cone shaped slotted wall. This is to conduct the jet with particular solid wall conditions through the suction chamber. The streamlines resulting from the numerical calculation of that second variation can be seen in figure 4.

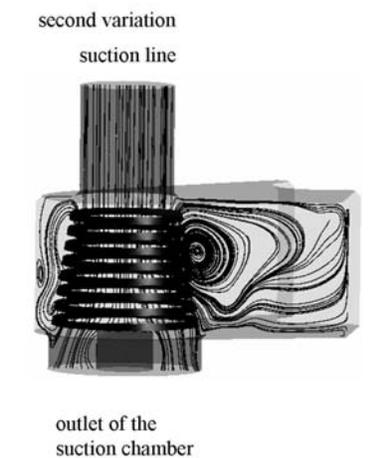


Figure 4: Second design variation of the suction chamber with simulated flow

This second optimization causes further deterioration of the heat transfer between the walls and the gas. The magnitude of the heat transfer decrease is not as significant as in the first variation but it is obviously another opportunity for heat transfer minimization.

In general the following aspects could be observed in all three variation steps: The velocity in the flow field massively depends on the frequency of the compressor. Due to wall friction effects the heat flux coefficient depends directly from the near wall velocity and the induced shear stress through the boundary layer.

The calculated heat transfer coefficients of all three geometry configurations are plotted in figure 5. The general shape of the α -time-function is in variant frequency cases similar. These diagrams are developed by using a rotational speed of $n = 2430 \text{min}^{-1}$. All values are scaled with the global maximum of the three geometry variants. The heat transfer coefficient was averaged over each wall section (upper, side and lower wall section). Additionally, the average was calculated over the whole suction chamber wall area.

As a general result of the numerical investigation the following aspects should be mentioned:

With a conducted jet straight from the inlet to the outlet it is possible to decrease the averaged heat transfer between wall and incoming air flow significantly. The perforated wall conduction of the investigated jet generates a further decrease of the heat transfer.

Another effect of this optimized flow situation is a reduced pressure loss through the suction chamber. This result does not come from the numerical calculation but can be derived easily by comparing the streamlines in the geometry variants.

Due to the decrease of the local and global heat transfer in the suction chamber a reduction of the gas temperature at the outlet to the cylinder should be possible and seems to be logically. An evidence for this assumed effect is not possible by the carried out numerical calculations. Indeed, by using a theoretical model with the calculated heat

transfer coefficients it is possible to obtain a heat balance of the steady state system. Nevertheless, the duration of the carried out unsteady calculations were only two turns. Due to the time changing mass flow and the unknown exchange of warm and cold gas between the jet and the turbulent eddies in the suction chamber it is not possible to predict the outflow temperature behaviour with only two turns of the compressor. The effort to perform a long duration unsteady calculation that shows a constant turn – averaged outflow temperature was too high for the given computational and time resources.

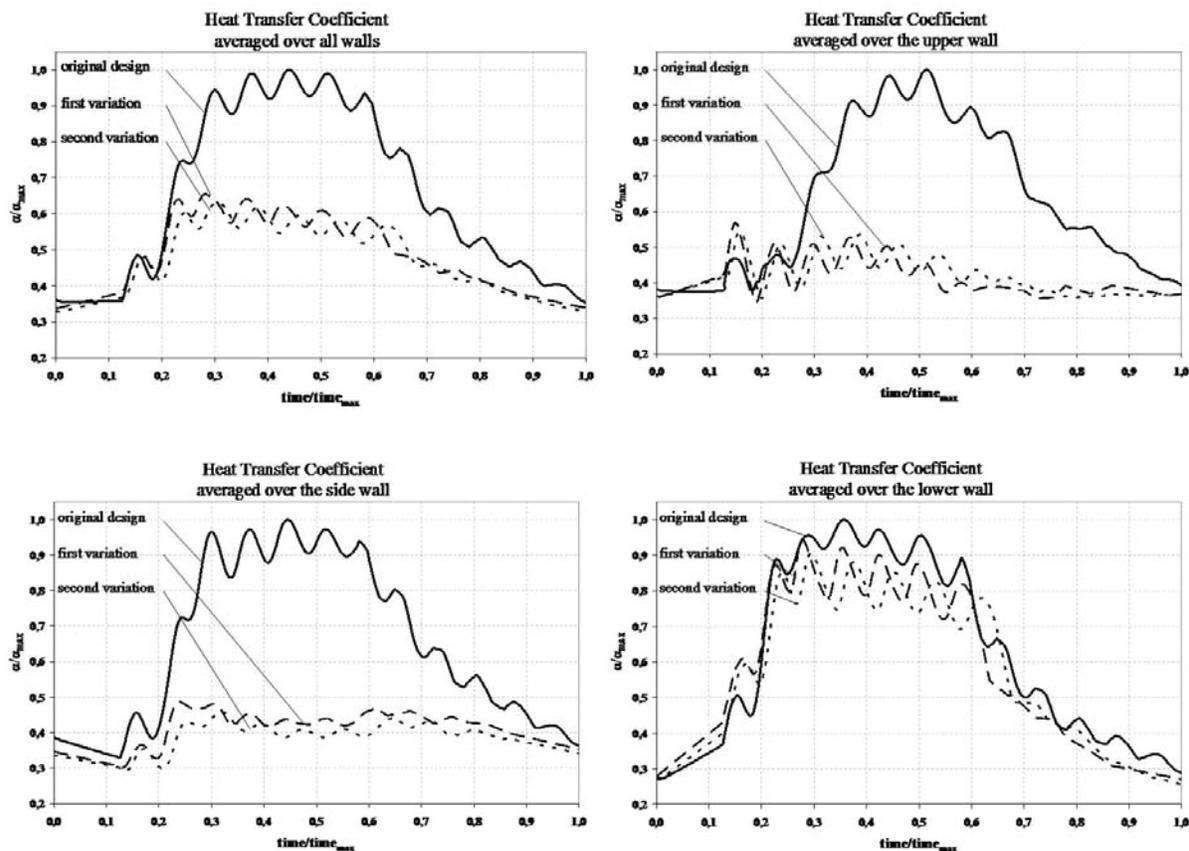


Figure 5: Heat transfer coefficient; Example for rotational speed of $n = 2430 \text{ min}^{-1}$

4. EXPERIMENTAL INVESTIGATIONS

Having simulated the original design and the two variations as well an experimental investigation was following. The geometry data and the parameters of the investigated and simulated air brake reciprocating compressor can be seen in table 1.

Table 1: Geometry and parameter data of the investigated reciprocating compressor

geometry	cylinder diameter	piston stroke	cylinder clearance	suction chamber volume
	d (mm)	s (mm)	ε_0 (-)	V_{sc} (ml)
	100	48	0,0244	≈ 130
parameter	suction gas temperature	coolant temperature	pressure ratio	number of revolutions
	t_{sh} (°C)	t_{cw} (°C)	π (-)	n (min^{-1})
	60	80	13,5	750, 1600, 2430

Figure 6 shows a sketch to provide an insight into the suction chamber with the original design left, the displaced suction inlet in the middle and the displaced inlet additional with the slotted truncated cone (PTFE) at the right. It should be pointed out that the change of the position of the inlet pipe between original and variation was only 35 mm.

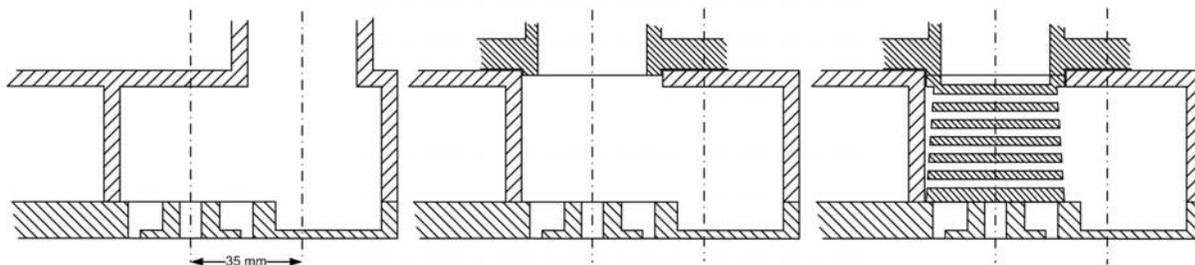


Figure 6: Sketch of the experimental investigated suction chamber design (original, first variation, second variation)

The valve plate is equipped with cooling channels beneath the discharge chamber (not shown) and a cut-out on the suction side to provide more volume.

The truncated cone is placed, as shown, directly on the valve plate. With this it is possible to avoid the flow over the hot edge of the suction valve inlet. The pitch of the experimental investigated cone is 3mm for manufacturing reasons. Its aperture angle is 4.7° to obtain an optimal diffuser (but of course slotted). The open area with the mentioned connection to the suction volume capacity is about 42%.

The following picture shows the volumetric efficiency λ of the investigated variations for the investigated speed range. The volume flow is calculated with $p = 1\text{bar}$ and $t = 60^\circ\text{C}$, the suction conditions of the measurement.

As described in the simulation we can see the effect of the decrease of the heat transfer coefficient (and of less pressure loss). It leads obviously to a smaller increase of the suction gas temperature. This results in more mass per rotation in the cylinder of the compressor.

The highest positive effect can be seen at low speed where between first variation and original design a relative improvement of about 4.2% was obtained. The improvement becomes smaller with higher rotational speed. As main reason it can be named the smaller process velocity at low speed. For this reason at low speed the gas remains longer time in the suction chamber where the turbulence is even with closed suction valve quite high in the original design. With the displaced suction line connection of the first variation, and therefore less turbulence into the suction chamber, the heat transfer becomes smaller and within the same retention time the suction gas warming up becomes smaller. At high speed the process time is more than three times higher than at low speed ($2430/750 \approx 3.2$) but the average heat transfer coefficient is just two times higher (simulation, original design $\alpha_{2430}/\alpha_{750} = 115.6/53.2 \approx 2.$). So the heat transfer increase with higher speed but the time for the gas to remain into the suction chamber, also necessary for heat transfer, is more reduced. With this the effect of displacing the suction line wanes with higher compressor speed.

The second variation with the guided flow improves the advancement of the volumetric efficiency. The reason is on the one hand further decrease of the heat transfer coefficient, e.g. the gas does not longer flow over the sharp hot edge at the suction valve inlet (mentioned above) and with the truncated cone it succeeds completely to avoid the flare of the flow.

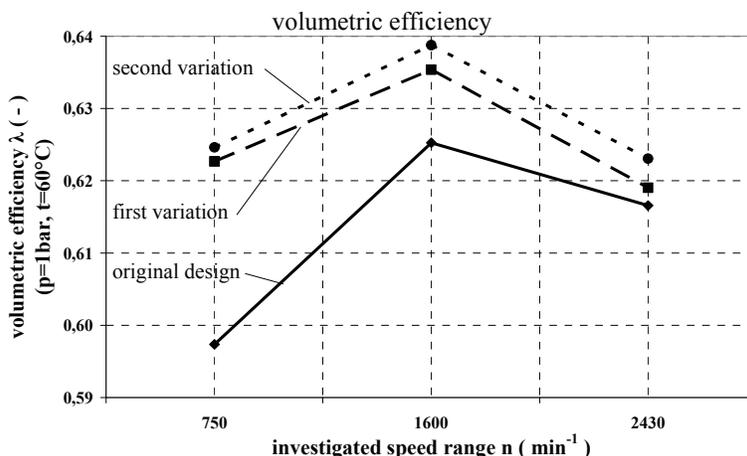


Figure 7: Volumetric efficiency and its improvement by geometric variation of the suction chamber

If it is possible to increase the mass flow (by reducing the effective suction temperature and pressure loss in our case) the next question is for the energy effort. The left side in figure 8 shows the specific work over the investigated speed range for the original design, the first and second variation. It can be seen that at low speed (750min^{-1}) the first and second variation with higher volumetric efficiency and therefore more mass flow also shows less specific power requirement. So it is possible to update the complete compressor. But just at low speed. With higher speed (1600min^{-1}) the specific work is not longer clear below the original design. And at high speed (2430min^{-1}) the specific work crosses the original design.

The indicator diagram at the right side of figure 8 can give us an idea for the reason. An ideal compressor is working without valve loss (at discharge valve and also the suction valve). It can be seen that of course in a real compressors there are losses and they increase with increased speed. One can compare at low speed the original design with the first and second variation in one diagram and it is obviously that the pressure loss into the discharge valves is nearly negligible so the power effort with higher mass flow is negligible. With increase the mass flow it leads to a reduction of specific work simultaneous ($> 3\%$, relative).

This does not apply to middle and high speed. With higher speed the mass flow is still increased but less then with low speed (statement figure 7 as well). It can be seen that the valve losses at 2430min^{-1} are no longer negligible. The power effort is visible increased for the first and second variation. This leads to an increase of specific work also with an increase of mass flow. The valve losses outweigh the increased mass flow but it is uncertain because of the error margin of the measurement (figure 8, first and second variation crossed for 2430min^{-1}). So in the case of higher speed it is obviously not certain possible to improve the complete compressor with volumetric efficiency and specific work as well.

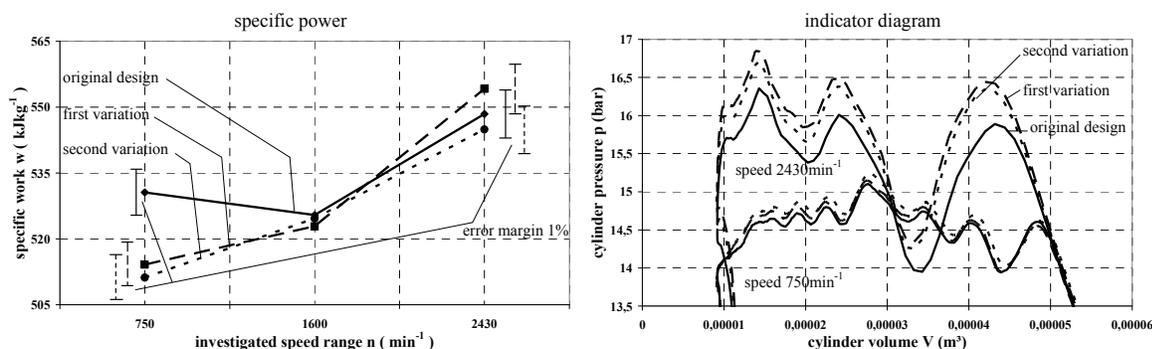


Figure 8: Specific work and indicator diagram (high pressure part) with comparison of low and high speed

Last but not least it should be mentioned that with displacing the suction inlet directly above the suction valve the sound level to the environment did not increase, also not with the investigated rotational speed of the variations.

5. CONCLUSIONS

A small air brake compressor was investigated theoretically and experimentally with the original design and two kinds of modifications. Different cognitions were gained:

1. Simulation

- For simulation just two runs are mandatory.
- Heat transfer coefficient can be reduced by avoiding a jet flow toward walls into the suction chamber (first variation).
- Guidance of the flow leads to further decrease of the heat transfer coefficient (second variation).

2. Experiment

- The effect of decrease the heat transfer coefficient is to increase the volumetric efficiency in either case. Here the best effect is at low compressor speed.
- The change (decrease) of flow temperature is not measurable. Decreased pressure loss was not measurable as well. Here it is necessary to think about effects of the environment to the sensor (location)!
- The modification of the suction chamber did not lead to reduce the discharge temperature.
- It is just possible to decrease the specific work clearly at low speed.
- An increase of the sound level because of the direct connection in the first and second variation is not to determine at all compressor speed!

NOMENCLATURE

Δh	enthalpy difference	(kJkg ⁻¹)
c_p	specific heat capacity	(kJkg ⁻¹ K ⁻¹)
n	rotational speed	(min ⁻¹)
\dot{q}	heat flux	(Wm ⁻²)
T	absolute temperature of a wall	(K)
T_{ref}	temperature of the fluid flow	(K)
T_s	gas temperature at suction valve without warming up, ideal	(K)
$T_{s,eff}$	gas temperature at suction valve with influence of heat transfer	(K)
\dot{V}	volume flow	(m ³)
w	specific work	(kJkg ⁻¹)
α	heat transfer coefficient	(Wm ⁻² K ⁻¹)
η	efficiency caused by temperature influence	(-)
λ	volumetric efficiency	(-)

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