TREATMENT ON HEAT TRANSFER IN SMALL PISTON COMPRESSORS

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Abstract

This paper presents an investigation about heat transfer in small, fast running piston compressors. A introductory chapter outlines the motivation of this investigation and gives some insight into the necessity to deal with heat transfer.

Starting with some basics of heat transfer in general focus is laid on the heat transfer from the compression chamber into adjacent structures. The basic modelling of piston and valve motion is described as well as the dynamic discretization of the fluid volume. Time dependent heat fluxes and heat transfer coefficients are calculated for all involved physical interfaces. It is shown how the variation of pure geometrical terms influences the thermal behaviour of the whole system.

Experimental data were gathered to evaluate the degree of accuracy of the numerical model.

A short outlook of how these results could help to optimize the valve system from the thermal point of view concludes the paper.

1 Introduction

An unpleasing feature of each compressor in operation is that it dissipates energy in form of heat. Especially in small, fast running machines with high pressure ratios the amount of generated heat becomes more and more a problem. E.g. in a one-stage 380 cm^3 air-compressor running at 3600 rpm with pressure ratio of 13 compressed gas temperature easily exceeds 300 °C when no extra cooling is put on the system.

Gas of that temperature level is extremely hazardous for the compressor itself, i.e. degradation of material properties, and succeeding components (e.g. sealings).

Much work has been invested in the past to reduce the discharge gas temperature from an empirical point of view and only little investigations with a more theoretical approach have been made so far (Abidin et al., 2005, Aigner et al., 2007).

With a deeper understanding of the thermal balance during a complete compression cycle conclusions can be drawn and proper actions can be taken to get the compressed gas as cool as possible.

2 Synopsis

Although piston machines and compressors are known for more than 100 years there is still some space for recent research. Besides the well investigated mechanical and flow issues in a piston engine, heat transfer and cooling had been studied many decades ago (Wintergerst, 1940, 1 p.; Pflaum et. al., 1970). Not so for piston compressors. It seems that heat management in open and semi-hermetic piston compressors of small size has been only of little interest so far.

From the viewpoint of the gas there are several transfer paths to consider in a compressor (see table 1 & fig. 1).

Unwanted heat flow inevitably leads to shortened performance (e.g. “suction gas heating” by heat transmission from discharge plenum to suction plenum), material degradation and unease of operation.

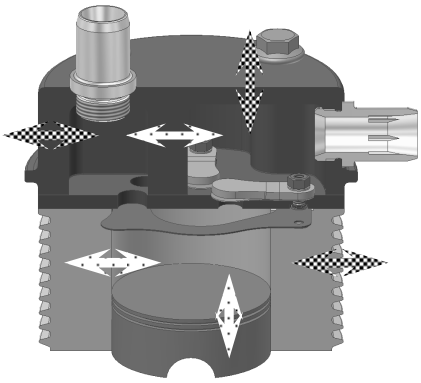


fig. 1: heat transfer paths (selection)

table 1: possible heat flow paths	
heat transfer	*heat transmission*
gas → cylinder liner	gas → cylinder liner → ambient atmosphere
gas → valve plate	gas → valve plate → discharge/suction plenum
gas → piston	gas → piston → oil sump etc.
gas → discharge port walls	gas → discharge port walls → cooling system
gas discharge plenum → cylinder cover walls	gas discharge plenum → cylinder cover walls → suction plenum
	gas discharge plenum → cylinder cover walls → ambient atmosphere

A full analysis of all possible heat transfer paths is a very complex and laborious task and some simplifications are necessary to keep the model clear and comprehensible. Nonetheless a careful balancing is of utmost importance.

3 Theoretical Background

Starting point is an ideal p-V-diagram of a compressor with clearance, performing a polytropic process (fig 2a). Assuming a perfect gas as working fluid the total compressor work input is given by (Küttner, 1991)

$$W = V_s p_s \frac{n}{n-1}\left(\gamma^{\frac{n-1}{n}} - 1\right) \qquad (1)$$

The total amount of heat transferred from the gas into the machine is then (Frenkel, 1969)

$$Q = W \frac{n-\kappa}{n(\kappa-1)} \quad , n \neq \kappa \qquad (2)$$

In a real compressor there are additional amounts of heat *Q'* to dissipate which arise from valve and other kind of losses (shaded areas, fig. 2b).

Although equation (2) is suitable to calculate the dissipated heat in total it can neither reveal its kinetics nor its spatial distribution. This information is obtained in an numerical approach only.

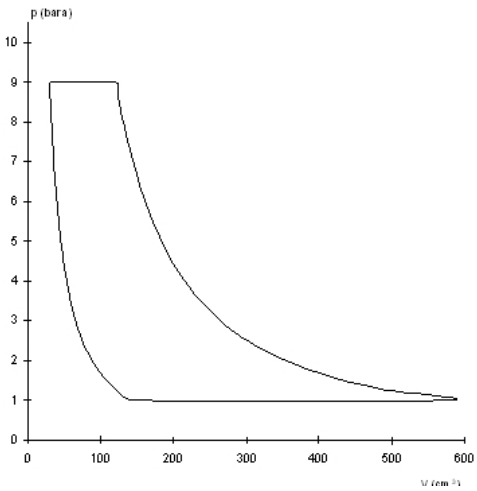
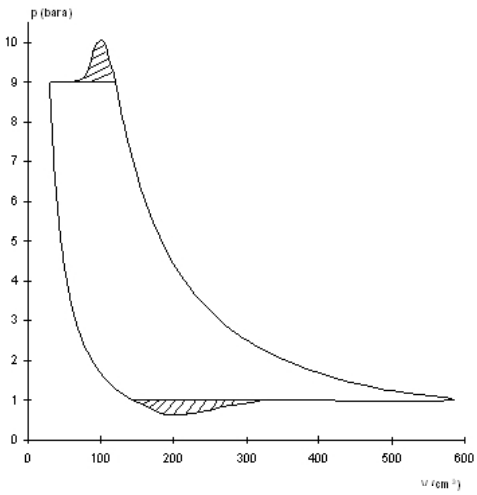


fig. 2a: ideal compressor cycle



2b: real cycle

Entropy is a powerful means to judge the thermal conditions in a compressor. In this work we assume the working gas to be a perfect gas. The equation for specific entropy is then given by:

$$s - s_0 = \int_{T_0}^{T} \frac{c_p}{T'} dT' - R \ln\left(\frac{p}{p_0}\right) \qquad (3)$$

In a fast running compressor with high compression a big temperature difference is expected. In this case c_p has to be considered as temperature-dependent.

The heat flux through a wall of given area is proportional to the driving temperature difference and the transfer properties of the gas and solid body. An analytical solution of equation (4), with α and ΔT both varying in time and space, can solels be found for some special cases. A compressor with its unsteady gas flow due to valve operation is far beyond this potential.

$$\frac{d\dot{Q}}{dA} = \alpha \Delta T \qquad (4)$$

Finally, only the numerical approach remains.

4 Concept of the study

Pursuing the goal of determining the heat flow in a piston compressor we have chosen a quite straight forward model to keep calculation time low.

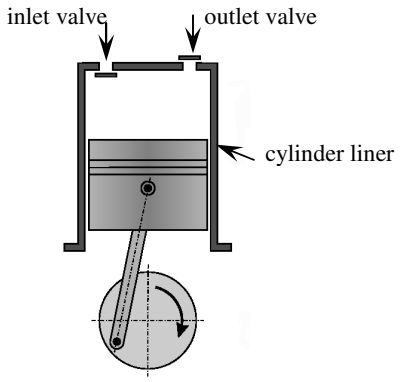


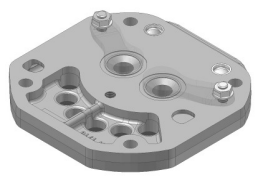
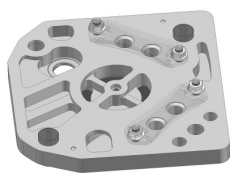
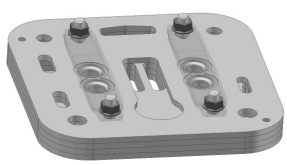
fig. 3: schematics of compressor model

The model comprises a cylinder liner, a revolving piston and inlet and outlet valve (see fig. 3). The liner and every other surrounding wall is held at constant temperature (120 °C); thus no heat transfer into the solid was calculated. Inlet and outlet valve are modelled as a mass-spring-system including lift-dependent flow areas.
With these inputs some characteristics of the compression cycle can be obtained:

- pV-diagram
- T-s-diagram
- heat flow
- heat transfer coefficients

In order to see if there is some influence of size, number and position of inlet and outlet valves on the thermal characteristics, 4 variants were examined (see table 2).
These valves are a typical choice of water cooled units used in mobile compressors for vehicle air supply.

table 2: valve configurations

			
type 1	*type 2*	*type 3*	*type 4*

5 MODELLING

The working model for the CFD-calculations consists of different components:

- valve modelling, i.e. dynamics and flow features
- heat transfer model
- meshing

Some simplifications regarding valve behavior and boundary conditions had to be made, but they don't affect the conclusions in general to be drawn from the results.

5.1 Valve model

The valves are completely self-acting and their kinematics is driven only by pressure differential and gas force. For the purpose of describing opening and closing performance of the valve the equation of motion can be written as:

$$m\ddot{h} = F - k(h + L) \qquad (5)$$

Discretization of (4) leads to a coupled set of equations for h and $\dot{h}$. With known spring rate k and driving force F (i.e. pressure differential times valve area) the system can be solved by iteration.
Main characteristic of every valve is its effective flow area Φ dependent from the actual lift h. With given values for maximum flow area Φ_{max} and effective lift h_{eff} a good representation is given by:

$$\Phi(h) = \Phi_{max} \frac{h}{\sqrt{h^2 + h_{eff}^2}} \qquad (6)$$

Mass flow through the valves is described by the equation from St. Venant / Wantzel (e.g. Arsenjev et al., 2003).

5.2 Flow and heat transfer model

As temperature varies in a wide range during a compressor cycle, temperature dependent coefficients for specific heat c_p, dynamic viscosity μ and thermal conductivity λ were introduced:

$$\begin{aligned} c_p(T) &= c_{p,1} + c_{p,2}T \\ \mu(T) &= \mu_1 + \mu_2 T \\ \lambda(T) &= \lambda_1 + \lambda_2 T \end{aligned} \qquad (7)$$

In this work the well known RANS-equations for unsteady turbulent flow are used in combination with the k-ε-model for a tube. The finite volume method was used to solve the equations. The heat flow through the walls was modelled using special wall-functions that couple the heat flux density at a certain point of the wall with the gas temperature in this point using Reynolds' analogy between momentum and energy transport.
A well established model to describe heat transfer in a piston machine is that of turbulent, highly compressible flow. That yields for the heat transfer coefficient a superposition of terms linear in molecular and turbulent Prandtl-number Pr and Pr_t respectively (see FLUENT, 2006).

$$\alpha \propto f(\mathrm{Pr}, \mathrm{Pr}_t) \qquad (8)$$

However the weak point of this approach is that the mean flow velocity v_m, one of the 'drivers' of heat flow is not known a priori. As usual in simulating such complex systems an iterative process must be applied on.

5.3 Meshing

When trying to realize the piston kinetics of a compressor in FEA or CFD software one has to deal especially with the problem arising from the top dead center (TDC): to achieve a satisfying degree of accuracy the number of segments of the generated lattice mustn't be too small. On the other hand when the gap between piston top and compressor valve is nearly zero all segments are squeezed to almost evanescent height causing big problems in the numerical processing.
A good compromise between accuracy and numerical stability is the concept of "Dynamic Layering". When the volume increases new segments are added and vice versa. Consequently only few cells are left when the piston has reached TDC (number of cells TDC:BDC ≈ 1:15).

6 Numerical Results

All calculations started at TDC. Cylinder volume, pressure and temperature are well defined in this point: $V_{TDC} = V_c$, $T_{TDC} = T_d$ & $p_{TDC} = p_d$. Cylinder liner, piston and valve are held at constant temperature. For best possible accuracy at reasonable calculation time angular increment was set to 0.01 degree.

In our compressor model four regions were identified to compare the thermal properties of the variants of interest (see fig. 4):

- head: bottom of valve plate
- piston: piston top
- outwalls: surrounding wall of outlet flow channel
- side: shell of cylinder liner

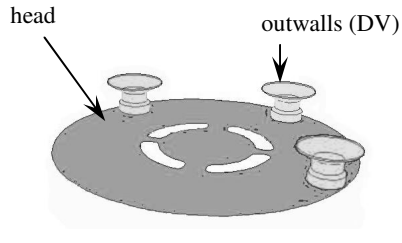
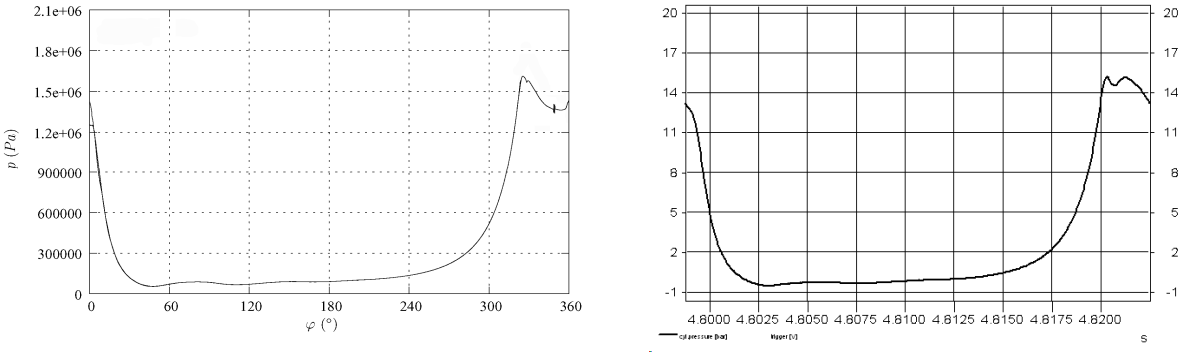


fig. 4: regions of interest for thermal analysis; type 2 valve

Thus the areas of heat transfer are well defined and only the characteristic length *D* (eq. 8) was left to fix for each calculation.

6.1 Pressure Curve

To see how good the modelling matches the real compressor in the first step we have determined the pressure vs. crank angle curve and compared it with curves taken from measurements (fig. 5).



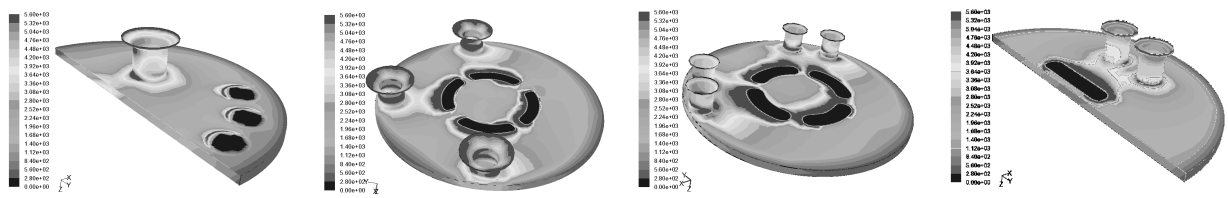
(a) calculated (unit: Pascal, absolute pressure) (b) measured (unit: barg)

fig. 5: comparison of simulated (a) and measured (b) pressure curve

Bigger differences in the march of pressure are only seen in the phase while the outlet valve is open. In the simulation the pressure drops markedly after opening of the valve; the measurements however show only a slight indent. This different behaviour is most likely due to a too small flow resistance in the model when the valve is partly open. But as the height of the calculated pressure peak comply very good with the measured one we rely on having a matching model.

6.2 Local Heat Transfer Coefficient (htc)

On basis of our transfer model (see section 5.2) it is possible to calculate the topology of heat transfer coefficients. The snapshots in fig. 6 are taken at 335° crank angle where mass flow through the discharge valve is at maximum.



(a) type 1 (b) type 2 (c) type 3 (d) type 4

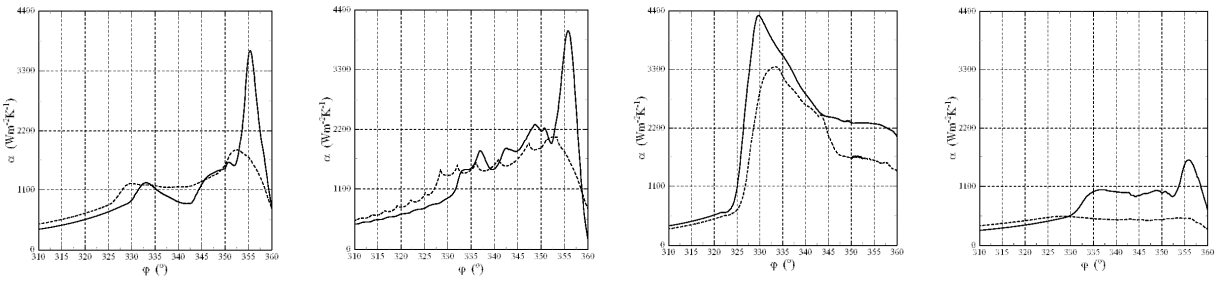
fig. 6: local heat transfer coefficient at 335° crank angle (the area of inlet ports is not considered → black shading)

In that piston position the gas in the cylinder is under high pressure and the outlet valves are fully open. Due to the high flow velocity in the discharge ports the heat transfer coefficient is also at maximum in those regions.
Focussing on the gas velocity as leading parameter of *htc (*see eq. 8), it is obvious that there must be a difference in magnitude of *htc* when changing the effective flow area. However it has proven that more the smallest port diameter *D* is of importance than the total flow area.

Transforming equation (8) and assuming that temperature is nearly constant the quotient $\frac{(vD)^a}{\alpha}$ should be an invariant. The mean deviation from this invariant[i] was found to be only 2.4% for all type of valves.

6.3 Average Heat Transfer Coefficient

A more comprehensible representation than the local *htc* is the average *htc* (fig. 7). This value is determined when dividing the heat flow through a zone by the difference of the wall temperature and an average gas temperature.



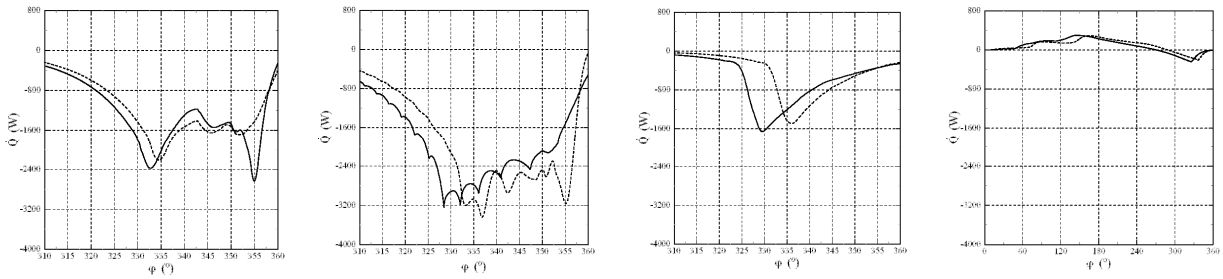
(a) head (b) piston (c) outwalls (d) side

fig. 7: mean *htc* for different walls and different valve geometries

Appreciable magnitude is seen only for the re-expansion, compression and discharge phase. Highest values appear while the outlet valve is fully open.
In comparison to the other boundaries the *htc* for the cylinder liner (fig. 7d) is nearly 4 times smaller. This drop of *htc* can be explained by considering that all gas is forced to stream through the outlet ports. Thus the vector of velocity has big components in direction of ports and only small ones towards the liner wall.
When the discharge ports lie close to the liner wall (e.g. type 2 & 3) the *htc* is greater than in the other cases (7d: dotted line → type 4, solid line → type 3).

6.4 Heat Flow

For given surfaces and with the calculated *htc*s heat flow can be obtained by applying equation (4). The following diagrams (fig. 8) show exemplarily how heat flow through the individual walls changes when the gas stream forms a time varying flow field (i.e. gas velocity differs in a wide range with crank angle).



(a) head (b) piston (c) outwalls (d) side

fig. 8: heat flow at different walls (dotted and solid lines indicate two different types, not necessarily the same in each case). Numbers greater than zero mean the gas takes heat from the adjacent wall and vice versa.

A closer look at the heat transfer model gives a good explanation for this observation: valve motion determines the effective flow area at every time step. Size of flow area directly influences flow velocity of the working gas which in combination with geometrical distribution is an essential quantity of the whole modelling.
Again the heat flow through the cylinder liner wall is much different from the others. The strikingly small numbers are a product of small *htc* in combination with a small heat transfer area when a markedly amount of heat is present.
Besides the time-dependent behaviour of heat flow the total distribution of heat going through the walls is of interest. Nearly 50% of total heat is dissipated into the piston, while only 4% is transferred into the cylinder liner. Table 3 gives a complete overview:

table 3: distribution of heat flow[ii]			
piston: 47% head: 35% outw.: 14% side: 4% *type 1*	piston: 49% head: 34% outw.: 13% side: 3% *type 2*	piston: 51% head: 34% outw.: 12% side: 4% *type 3*	piston: 47% head: 35% outw.: 14% side: 4% *type 4*

6.5 *T-s-diagram*

For a more global look at the heat balance of the compression cycle it is suitable to make a T-s-plot. The state of reference s_0 is commonly accepted to be $p_0=10^5$ Pa and $T_0=273.15$ K.

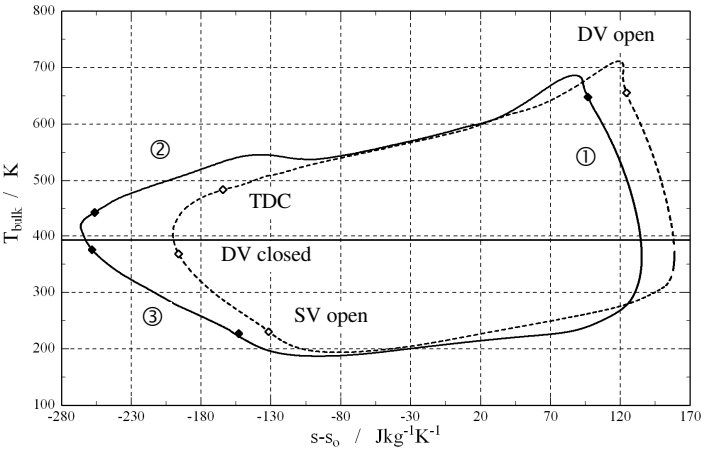


fig. 9: heat transfer properties of different valves

Some general features can be seen from the diagram (cp. Fig. 9):

- even in a virtual compressor with ideal boundary conditions, neglected losses and perfect gas behaviour the cycle does not follow the widely used law of adiabatic change of state

$$pV^{\kappa} = const. \qquad (9)$$

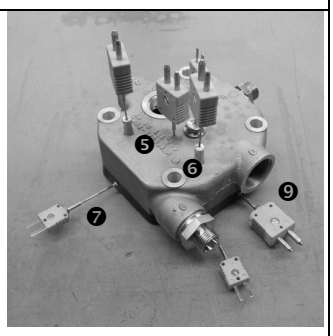
during compression and re-expansion. In a totally adiabatic process the points of state would lie on a vertical line. Each deviation expresses the extent of heat transfer taking place. For the compression phase (section ①, fig. 9) a constant polytropic exponent holds much better than for the re-expansion phase (section ③).

- while the outlet valve is open and piston is moving towards TDC the hot working gas dissipates energy to the adjacent walls (section ②). After TDC is reached and the valve is closed re-expansion begins and the compressed air cools down even below wall temperature (solid line, fig. 9) and takes heat from its surroundings (section ③). The total energy consumed or dissipated during these phases is represented by the area enclosed. between the curve and the bottom line.

7 Experimental Results

In order to check the quality of the described model an extensive bench experiment has been set up[iii]. In total 20 temperature probes were installed to keep control of the dissipated heat (see table 4).

table 4: installation of temperature probes

1	ambient air	12 - 17	cylinder liner	
2	coolant IN	18	oil supply IN	
3	coolant OUT	19	oil supply OUT	
4	suction neck	20	cylinder plenum	
5, 6	suction plenum			
7	suction valve			
8	separation wall (suc./dis.)			
9	discharge valve			
10	discharge plenum			
11	valve plate			probes 5, 6, 7, 8, 9, 10, 11

Unfortunately it turned out that it was not possible to determine heat transfer coefficients with acceptable accuracy from the gathered data. Calculated coefficients were 3 times(!) greater than measured ones.
For more proper results it would have been necessary to acquire temperature at much more spots than it is possible when using standard Type K probes.
After the disappointing results in trying to determine heat transfer coefficients the most exciting questions was, if the general thermal performance of the compressor, represented by the T-s-diagram, could be verified.
Following conditions, for which experimental data were available, were chosen for this check:

- mean temperature and pressure in discharge plenum at TDC
- inlet pressure and suction pipe temperature at BDC

As can be seen in fig. 10 experimental (dots) and calculated values (lines) match quite well. The relative deviation of 16% / 9% (type 1 / type2) is excellent for thermal measurements.

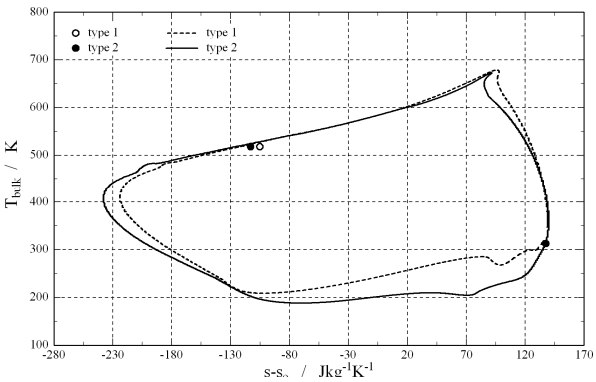


fig. 10: comparison of experimental and numerical results

The second check aims at the total compressor work and its flow through our pre-defined walls (cp. Fig. 4). Calculated compressor work is about 15% smaller than that from experiment. Most likely the assumption of a constant wall temperature over the whole system is too simplifying (fig. 11).

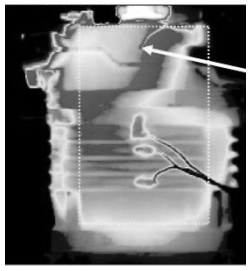


fig. 11: non-uniform temperature

Also the calculated march of pressure during discharge phase is different from the observed one (see fig. 5).

Both items contribute to the lack in power consumption. With an improved valve model and by calculating the solid temperatures instead of giving them as boundary condition a better alignment between simulation and experiment should be achieved.

8 Conclusions

- With even a quite elementary model a CFD-simulation yields thermodynamic and flow data which agree very well with experimental data. Moreover some numerical results can be determined where experimental options are still lacking.
- Heat generated during compression is dissipated mainly into regions where the gas has a high flow velocity and ample transfer area is at hand. Thus piston and valve plate take more than 90% of the heat while only 4% is dissipated into the liner walls.
- Magnitude of heat transfer not only depends on size of surrounding wall area but is also strongly influenced by size and geometrical arrangement of inlet and outlet valves.

9 Acknowledgements

We would like to thank the Hoerbiger Company for funding this project and we acknowledge the support of Dipl. Ing. Stephan Lehr from TU Dresden who contributed all measured data and who has spent lots of hours to gain this level of accuracy.

10 Nomenclature

A	area of heat transfer	(cm^2)	α	heat transfer coefficient	(W/m^{2}K)
c_p	specific heat for p=const	(J/kgK)	γ	pressure ratio	()
F	gas force	(N)	κ	specific heat ratio	()
h	valve displacement	(mm)	λ	thermal conductivity	(W/mK)
k	spring rate	(N/mm)	μ	dynamic viscosity	(Ns/m^2)
L	prestress of spring	(mm)	ρ	density of gas	(kg/m^3)
n	polytropic exponent	()	Φ	effective flow area	(cm^2)
p	gas pressure	(Pa)			
Pr	Prandtl-number	()	**Subscripts**		
R	specific gas constant	(J/kgK)	s	suction	
S	specific entropy	(J/kgK)	d	discharge	
ΔT	temperature difference	(K)	p	pressure	
v_m	mean flow velocity	(m/s)	eff.	effective	
V_c	clearance volume	(cm^3)	TDC	top dead center	
V_s	suction volume	(cm^3)	BDC	bottom dead center	

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[i] invariant calculated only for the case "outwalls"

[ii] excluding the heat carried away by the gas

[iii] built up and performed at the TU Dresden; Institute of Power Engineering; Prof. Quack, Dipl.-Ing. Lehr