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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. Focus of interest is the heat transfer from the compression chamber, into adjacent structures.
In a second step the influence of geometrical issues in terms of size, spatial distribution and shape of inlet and outlet ports has been studied.
An – idealized – compressor system has been modeled with a commercially available CFD program. The kinetics of the crank gear as well as the dynamics of inlet and outlet valve was implemented in the model. Constant temperature at boundaries has been described to keep calculation time low.
The first section of the paper describes the basic modeling of piston and valve motion and the dynamic discretization of the fluid volume. In the second section 4 different valves are presented with differing geometrical shape, size and arrangement of ports.
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 behavior" of the whole system.
A short outlook of how these results could be used 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 a problem. E.g. in a one-stage 380 cm^3 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. THEORETICAL BACKGROUND

Starting point is an ideal p-V-diagram of a compressor with clearance, performing a polytropic process (fig. 1). 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)

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

Although this equation is suitable to calculate the dissipated heat in total it can neither reveal neither its kinetics nor its spatial distribution.
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:

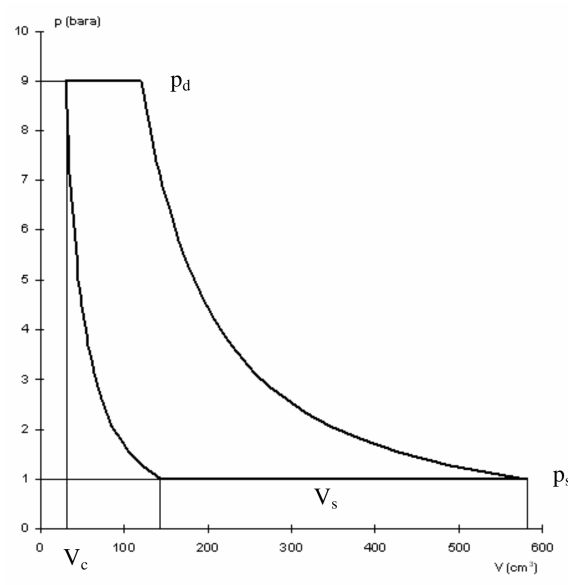


fig. 1: ideal compressor cycle

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

As a fast running compressor with high compression ratio is subject of the project a big temperature difference is expected. In this case c_p has to be considered as temperature-dependent.
To learn more about the evolution of heat during one compression cycle the following equation has to be solved:

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

Using the well known RANS-equations for unsteady turbulent flow in combination with the k-ε-model and appropriate wall functions the temporal and spatial distribution of the heat transfer coefficient α can be determined. Considering the complex geometrical boundary conditions of a real compressor this task could be performed by numerical means only.

3. CONCEPT

Pursuing the goal of determining the heat flow in a piston compressor we have chosen a straight forward model to keep calculation time low. The model comprises a cylinder liner, a revolving piston and inlet and outlet valve (see fig. 2). The liner is held at constant temperature, inlet and outlet valve are modeled as a mass-spring-system including lift-dependent flow areas.
With these inputs some characteristics of the compression cycle could be obtained:

- pV-diagram
- T-s-diagram

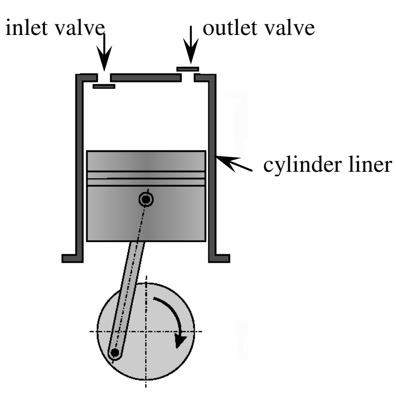
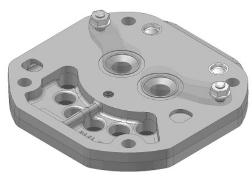
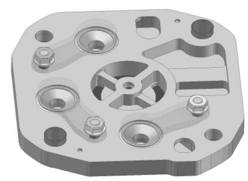
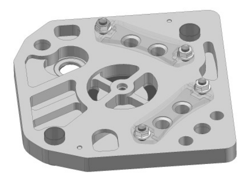
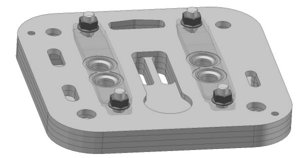


fig. 2: schematics of compressor model

- 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 1).

table 1: valve configurations			
			
type 1	*type 2*	*type 3*	*type 4*

4. MODELING

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

- valve modeling, 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.

4.1. 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).

4.2. 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:

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

The heat flow through the walls was modeled 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 flow in a tube using the k-ε-model (see Woschni 1970).

$$\alpha = \frac{\lambda C}{D}\left(\frac{v_c D\rho}{\mu}\right)^a \mathrm{Pr}^b \qquad (8)$$

However the weak point of this approach is that the characteristic velocity v_c causing the heat flow is not known a priori. Primary intention in the present work is to determine the characteristic flow velocity from the heat transfer coefficients α obtained from the CFD model.

4.3. 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 $\approx$ 1:15).

5. 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 4 regions were identified to compare the thermal properties of the variants of interest (see chap. 3):

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

Thus the areas of heat transfer are well defined and only the characteristic length *D* (eq. 6) was left to fix for each calculation.

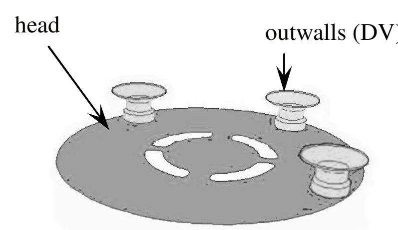
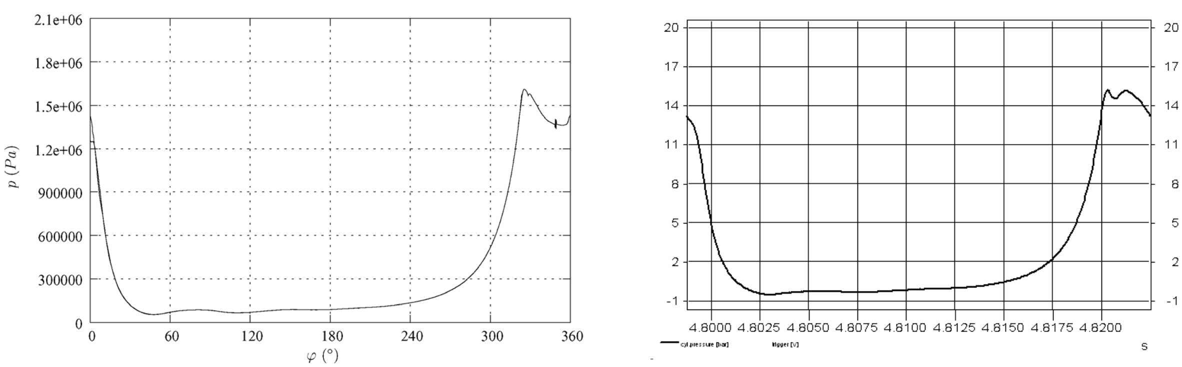


fig. 3: regions of interest for thermal analysis, type 2 valve

5.1. Pressure Curve

To see how good the modeling 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. 4).



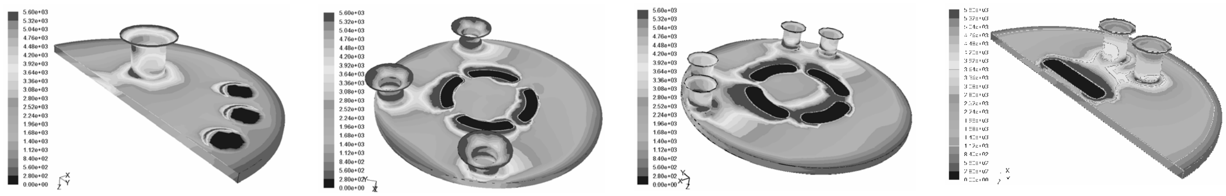
(a) calculated (unit: Pascal, absolute pressure) (b) measured (unit: barg)

fig. 4: 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 behavior 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.

5.2. Local Heat Transfer Coefficient

On basis of our transfer model (see section 4.2) it is possible to calculate the topology of heat transfer coefficients. The snapshots in fig. 4 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

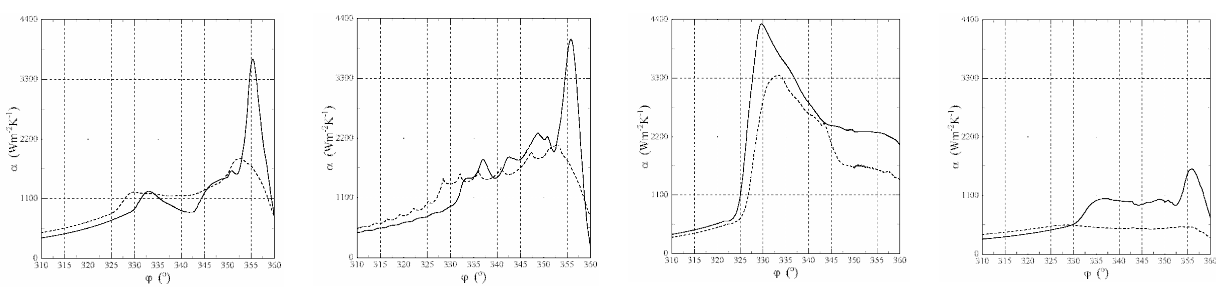
fig. 5: local heat transfer coefficient at 335° crank angle (the area of inlet ports is not considered → black shading)

In that position of the piston 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 heat transfer coefficient *(htc)*, compare eq. 7, 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.

5.3. Average Heat Transfer Coefficient

A more comprehensive representation than the local *htc* is the average *htc* (fig. 6). This value is determined when dividing the heat flow through a zone by the difference of the wall temperature and an average gas temperature.



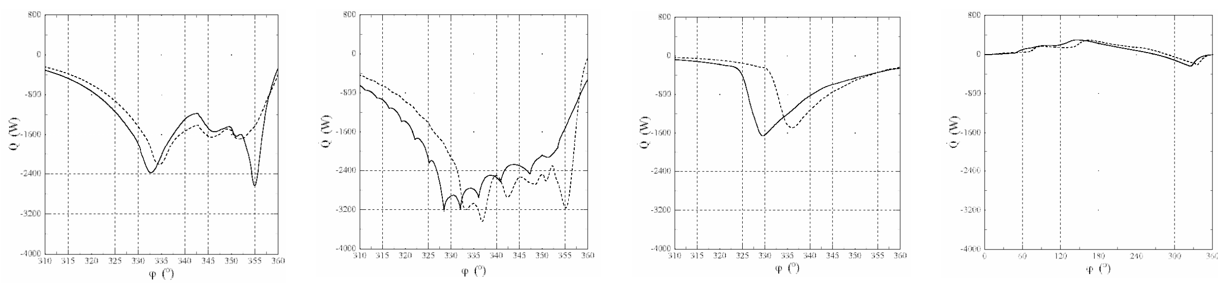
(a) head (b) piston (c) outwalls (d) side

fig. 6: 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. 6, d) 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 (fig. 6d: dotted line → type 4, solid line → type 3).

5.4. Heat Flow

For given surfaces and with the calculated *htc*s heat flow can be obtained by applying equation (3). The following diagrams (fig. 7) show exemplarily how heat flow through the individual walls changes when the gas stream forms a time varying flow field.



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

fig. 7: 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 modeling.
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 behavior 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 2 gives a complete overview:

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

5.5. 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.

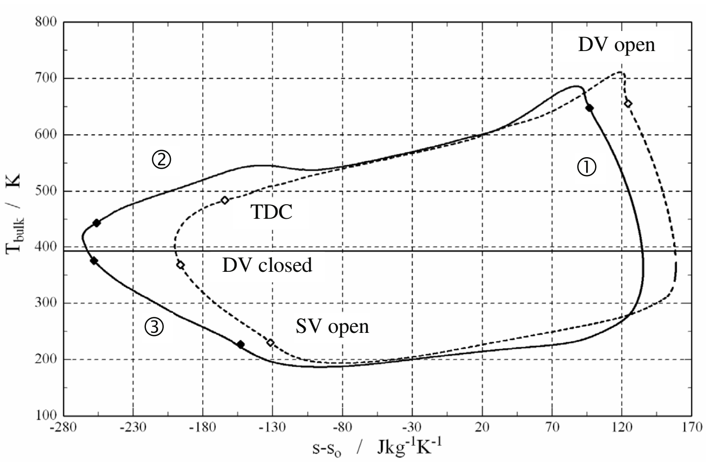


fig. 8: heat transfer properties of different valves

Some general features can be seen from the diagram (cp. fig. 8):

- even in a virtual compressor with ideal boundary conditions, neglected losses and perfect gas behavior 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. 8) 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. 8) 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.

6. EXPERIMENTAL RESULTS

In order to check the quality of the described model an extensive bench experiment has been set up[iii]. Much work has been spent on the issue how to keep the dissipation of heat under control to establish a meaningful heat balance. The cylinder liner was insulated with PUR foam and thermocouples were distributed all over the assembly (i.e. pressure chamber, valve plate, liner etc.).

All measurements were made in steady-state condition, which means results presented below are all mean values without any time dependency.

Despite the great efforts it was not possible to determine *htc*s from the gathered data. For this a more subtle distribution of temperature pick-ups would have been necessary to reach every possible path of heat flow. However the limited mounting space available enforced a compromise.

Also for this reason only type 1 and type 2 valves were taken into account.

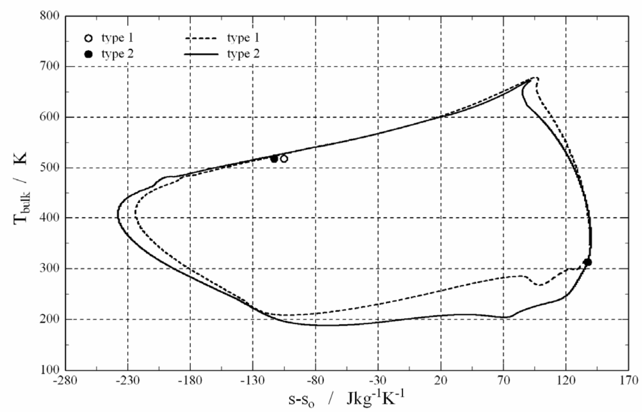


fig. 9: comparison of experimental and numerical results

One of the most exciting questions was, if the general thermal performance of the compressor, represented by the T-s-diagram, could be verified. Two outstanding conditions were chosen: pressure and discharge chamber temperature at TDC, inlet pressure and suction pipe temperature. As seen in fig. 9 experimental and calculated values match quite well. The relative deviation of 16% / 9% (type 1 / type2) is excellent for thermal measurements.

The second check aims at the total compressor work and its flow through pre-defined walls. Calculated compressor work is about 15% smaller than that from experiment. Most likely the assumption of a constant temperature all over the system walls is too simplifying. But looking at the heat flow into the liner walls we again see a deviation of approximately 12% in absolute figures, when considering the relative flow we have excellent match (also cp. chap. 5.4).

7. 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.

NOMENCLATURE

a, b	exponents k-ε-model	()	α	heat transfer coefficient	(W/m^{2}K)
A	area of heat transfer	(cm^2)	γ	pressure ratio	()
c_p	specific heat for p=const	(J/kgK)	κ	specific heat ratio	()
D	characteristic length	(m)	λ	thermal conductivity	(W/mK)
F	gas force	(N)	μ	dynamic viscosity	(Ns/m^2)
h	displacement	(mm)	ρ	density of gas	(kg/m^3)
k	spring rate	(N/mm)	Φ	effective flow area	(cm^2)
L	prestress of spring	(mm)			
n	polytropic exponent	()			
p_s	suction pressure	(Pa)			
Pr	Prandtl-number	()			
R	specific gas constant	(J/kgK)			
S	specific entropy	(J/kgK)			
ΔT	temperature difference	(K)			
v	gap velocity	(m/s)			
V_c	clearance volume	(cm^3)			
V_s	suction volume	(cm^3)			

Subscripts

s	suction
d	discharge
c	clearance
p	pressure
eff.	effective
TDC	top dead center
BDC	bottom dead center

REFERENCES

Abidin, Z., Almbauer, R., Burgstaller, A., *Simulation of the heat transfer between cylinder, piston and gas in a hermetic reciprocating compressor*, 2005, 4th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics Cairo, Egypt

Aigner, R., Steinrück H., *Modeling Fluid Dynamics, Heat Transfer and Valve Dynamics in a Reciprocating Compressor*, 2007, 5th EFRC Conference, Prague, p. 171 - 180.

Anderson, J. D., 1995, *Computational Fluid Dynamics*, McGraw-Hill, New York

Arsenjev, S.L., et al., 2003, *The flowing system gasdynamics*, http://arXiv.org/abs/physics/0302038

Berger, T.P.M., 2005, *Entwicklung eines numerischen Berechnungsverfahrens für Rührwerksschwingungen*, Ph.D. thesis, TU Munich

Chlumsky, V., 1965, *Reciprocating and Rotary Compressors*, SNTL – Publishers of technical literature, Prague

Frenkel, M.I., 1969, *Kolbenverdichter*, VEB Verlag Technik, Berlin

Hoff, K. H., 2002, *6. Workshop Kolbenverdichter*, Rheine, p. 169-184

Küttner, K.-H., 1991, *Kolbenverdichter*, Springer, Heidelberg

Pflaum, W., Mollenhauer, K., 1977, *Wärmeübergang in der Verbrennungskraftmaschine*; Springer, Wien

Pohlmann, H., 1988, *Strömung und Wärmeübergang im Zylinder eines geschleppten Kolbenmotors*, Ph.D. thesis, RWTH Aachen

Woschni, G., 1970, *Die Berechnung der Wandverluste und der thermischen Belastung der Bauteile von Dieselmotoren*, MTZ, vol. 31, no. 12: p. 491-499

[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