Modelling of fluid flow and heat transfer in a reciprocating compressor

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Abstract. Efficiency of reciprocating compressor is strongly dependent on several parameters. The most important are valve behaviour and heat transfer. Valves affect the flow through the suction and discharge line. Heat flow from the walls to working fluid increases the work of the cycle. Understanding of these phenomena inside the compressor is a necessary step in the development process. Commercial CFD tools offer wide range of opportunities how to simulate the flow inside the reciprocating compressor nowadays, however they are too demanding in terms of computational time and mesh creation. Several approaches using various correlation equation exist to describe the heat transfer inside the cylinder, however none of them was validated by measurements due to the complicated settings. The goal of this paper is to show a comparison between these correlations using in-house code based on energy balance through the cycle.

1. Introduction

Reciprocating compressors with high efficiency and low production costs are employed in many technical fields, ranged from household appliances to large-scale compressors used in refineries and gas engineering. Reduction in compressor energy consumption by improving the efficiency is a relevant step in decreasing the global energy consumption. Analysis according to [1] shows that contribution of household compressors to overall electric energy consumption in US reaches up to 8 %. Statistics from industry sector are not known, however it is expected to be even higher. Overall efficiency of compressor consists of three main sub-efficiencies: electrical, mechanical and thermodynamic. Thermodynamic efficiency is the lowest (80 – 83 %), therefore there is a significant effort to improve compressor parts that cause decrease in efficiency: cylinder leakage, pressure losses in suction and discharge line and the superheating of the gas. High temperature causes material degradation (sealing, valves…), increases the risk of oil ignition inside the compressor and also increases the compression work. Therefore keeping the temperature of the gas as low as possible decrease the work input to a compressor and improve durability. The lowest compression work could be achieved with isothermal process, but refrigeration requires also high temperature on the compressor discharge to release heat in condensation unit. In this case the evaluation of thermodynamic efficiency is done by comparing isentropic and actual compressor work, see equation (1)

$$\eta_{isen} = \frac{w_{isentropic}}{w_{actual}} \tag{1}$$

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$$w_{isentropic} = \frac{\kappa R T_1}{\kappa - 1}\left[\left(\frac{p_2}{p_1}\right)^{(\kappa-1)/\kappa} - 1\right] \tag{2}$$

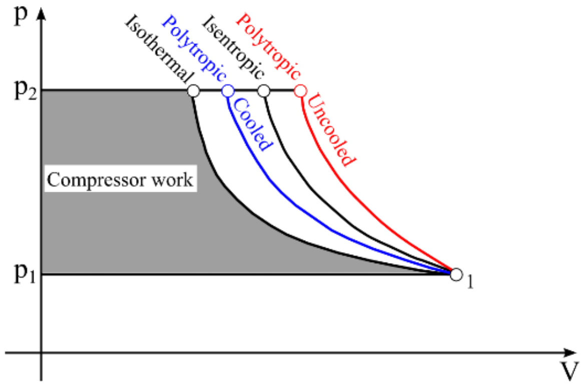

Figure 1. Processes inside the compressor cylinder.

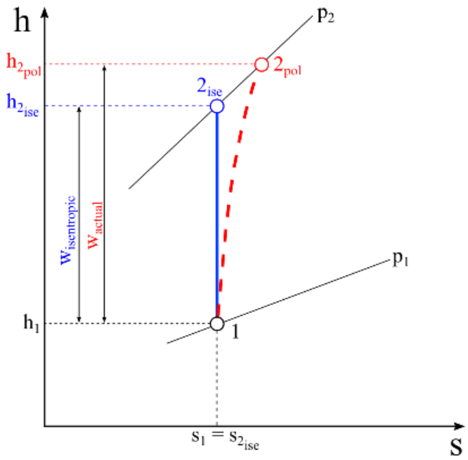

Figure 2. Comparison of isentropic and actual proces in the compressor.

Actual work of compression in a not cooled compressor is visible in the figure 1 and 2. In this paper actual work differs from isentropic (equation (2)) in heat transfer through the walls. Understanding heat and fluid flow inside the compressor during a working cycle is a fundamental step of development process aiming for better efficiency. As the compression of gas is the biggest source of heat, it is necessary to examine heat flow in the cylinder. It is possible to use numerical CFD tools, although integral models offer much faster results.

This paper presents a critical review of integral heat transfer models used for thermal analysis of compressor and their effect on the compressor efficiency.

2. Compressor modelling

Fluid flow modelling was done with 0D simulation tool using the energy balance over a control volume [2]. It has a benefit of fast calculation and could be easily combined with heat transfer model. Several approaches were used for heat transfer and they are presented in detail further in the paper. The 0D code was written in Matlab.

2.1. Fluid flow

The base of 0D model is an energy balance over a control volume using the 1st Law of Thermodynamics, equation (3). In this case the control volume is the whole volume of a cylinder. Flow inside the cylinder is neglected and both pressure and temperature are considered uniform in the cylinder.

$$dW + dQ + \sum_i dm_i \cdot h_i = dU \tag{3}$$

where Q stands for heat transfer from/to the cylinder, $dm_i h_i$ is inflow or outflow through the valves, dU is the change of inner energy and dW is the piston work. The piston work is done by the change of a volume via piston motion

Mathematical description of valve movement was introduced by [3] as a mass-spring system, examined more deeply in the work of [4]. Description uses single degree of freedom equation (4) solved by using 4th order Runge-Kutta method.

$$m \cdot \ddot{x} + d \cdot \dot{x} + k \cdot x = \sum_i F_i \tag{4}$$

Properties of the spring system are formed by the mass of the valve *m*, stiffness of the spring *k* and damping constant *d*, which must be usually determined experimentally, as it is dependent on several parameters. The damping constant is usually very low, therefore it is often neglected in equation (4) [5]. Main forces acting on the valve are pressure difference over the valve, stiffness of the spring and adhesive forces due to oil between valve surface and seating. Mass flow through valves is calculated by Fliegner's equation (5):

$$\dot{m} = \varphi_{eff} \cdot \rho_1 \cdot \left(\frac{p_2}{p_1}\right)^{\frac{1}{\kappa}} \sqrt{\frac{2\kappa}{\kappa-1}\frac{p_1}{\rho_1}\left(1-\left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}}\right)} \tag{5}$$

where φ_{eff} is the effective flow area. It is determined from valve position and the local pressure loss coefficient ζ. p is the pressure and ρ is the density.

2.2. Heat transfer correlations

Improvement of the compressor efficiency could be done by optimization of thermal flows inside the compressor. The main source of the heat is the compression process inside the cylinder, nevertheless it is also important to analyze the heat transfer during the suction process. Hot walls of cylinder and suction manifold heat up the working fluid before the compression process and every increase of temperature by 1 K causes reduction of COP by 0.32 % [6]. Increase in temperature decreases the volumetric efficiency and increase in compression work, since the specific volume of a gas is proportional to temperature. This is particularly important during the suction process when the walls are superheating gas in suction line.

There are several approaches to quantify heat transfer coefficient, however the correct one has not been stated yet. Differences are significant especially during the suction or discharge process. Pereira [1] analyzed most important correlations for heat transfer and found the considerable disagreement with numerical simulation. All of these correlations are based on representation of Nusselt number (equation (6)) with different constants *a*, *b* and *c*, except for the approach used by Aigner [5], which uses Stanton number instead.

$$Nu = a\,\mathrm{Re}^b\,\mathrm{Pr}^c\,. \tag{6}$$

Correlations of Woschni (7) and Annand (8) were developed for internal combustion engines, in which a different flow could occur due to the different operation of the valves [7]. Equation (9) and (10) are used to calculate velocity in Woschni correlation.

$$Nu = 0.35\,\mathrm{Re}^{0.7} \tag{7}$$

$$Nu = 0.26\,\mathrm{Re}^{0.75} \tag{8}$$

$$u = 6.618 \cdot u_p \tag{9}$$

$$u = 2.28 \cdot u_p \tag{10}$$

Adair [7] proposed a new correlation (equation (11)) based on the investigation of reciprocating compressor. He used variable equivalent diameter *De*, equation (12) and (13) instead of cylinder diameter and the approximation of swirl velocity by angular speed of a crankshaft. Previous authors used a mean piston velocity in Reynolds number, which is much smaller compared to angular speed of crankshaft. According to Adair the compressor efficiency could be affected by heat transfer correlation by just 3 percent, however volumetric efficiency could decrease by 10 – 20 percent [7].

$$Nu = 0.053\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.6} \tag{11}$$

$$D_e = \frac{6 \cdot Cylinder\ volume}{Cylinder\ area} \tag{12}$$

$$\mathrm{Re} = \frac{\rho D_e u}{\mu} \tag{13}$$

The latest contribution to heat transfer modelling was done by Disconzi [8]. He divided the compressor cycle in 4 main processes: suction, compression, discharge and expansion. For each of them he developed new correlation. The definition of Nusselt number is similar to equation (6), however there are different constants for each process. Reynolds number is also different. Mean piston velocity is used during compression and expansion. The main innovation is in suction and discharge process, where the flow through the valve is taken into account, see table 1. The prediction of Disconzi's model was in good agreement with numerical simulation. He noted that the highest amount of heat is transferred during the suction because of the high temperature difference between sucking gas and cylinder walls. The heat transfer area is significantly larger compared to discharge, which increases the importance of a correct heat transfer prediction in cylinder.

Table 1. Reynolds number and constants for processes inside the cylinder according to Disconzi.

Process	**Reynolds number**	**Constants** a	b	c
Compression	$\mathrm{Re} = \frac{\rho_{(t)} D u_p}{\mu_{(t)}}$	0.08	0.8	0.6
Discharge	$\mathrm{Re} = \frac{\rho_{(t)} D \left(u_p + u_p^{0.8} u_{c(t)}^{0.2} \right)}{\mu_{(t)}}$	0.08	0.8	0.6
Expansion	$\mathrm{Re} = \frac{\rho_{(t)} D u_p}{\mu_{(t)}}$	0.12	0.8	0.6
Suction	$\mathrm{Re} = \frac{\rho_{(t)} D \left(u_p + u_p^{-0.4} u_{c(t)}^{1.4} \right)}{\mu_{(t)}}$	0.08	0.9	0.6

$$u_c = \frac{\left| \dot{m}_{(t)} \right|}{\rho_{(t)} \cdot A_c} \tag{14}$$

where $\rho_{(t)}$ is the density of gas in the cylinder, A_c is the cross-sectional area of the cylinder and u_p stands for the piston velocity.

Aigner [5] also uses dimensionless numbers, but the correlation is built on Stanton number, which is the ratio of heat flux to the wall and energy flux in flow relative to wall, equation (15).

$$St = \frac{h}{\rho c_p u} = \frac{Nu}{\mathrm{Re}\,\mathrm{Pr}} \tag{15}$$

Stanton number could be determined as a function of skin friction coefficient C_f, which could be estimated from Reynolds number. In the work of Aigner, heat fluxes were reconstructed with the full three-dimensional calculation and for sufficient number of cases Aigner proposed more suitable Stanton numbers.

$$St = f\left(C_f\right) \tag{16}$$

The compressor cycle was also divided in main processes: inflow, outflow and compression with expansion together. Moreover he analyzed heat fluxes for individual surfaces in the compressor (piston, cylinder wall and cylinder head). It is worth noting that there is a difference between the cooling compressor and the compressor in Aigner's research. The latter is considerably bigger.

2.3. Compressor
Comparison of different heat transfer approaches was carried out with parameters of a real compressor. Dimension of the compressor and the properties of valves are shown in table 2. The working fluid differs in reality according to the application of the compressor. For the purposes of presented analysis an ideal air was used in order to simplify the comparison. Boundary conditions are shown in table 3.

Table 2. Compressor specification.

Parameter	Value	Unit
Cylinder bore	220	mm
Crank height	45	mm
Rod length	250	mm
Crankshaft speed	980	rpm
Clearance length	1.5	mm
Valve weight	0.08	kg
Spring stiffness	50000	N

Table 3. Boundary conditions used in the simulation tool.

Boundary condition	Value	Unit
Suction pressure	0.97	bar
Discharge pressure	5.0	bar
Inlet temperature	310	K
Wall temperature	350/450	K

3. Results and discussion
Numerical tools are used in the pre-development phase of the compressor to evaluate changes in design or working conditions. In comparison with experimental testing it is usually faster and less costly approach, nevertheless numerical results must be in reasonable agreement with experimental data. It has been proven that quasi-static simulation tool could be used for small reciprocating compressor with satisfying accuracy when examining the pressure behavior during the cycle. Evaluation of the heat transfer is however complicated task to do. Heat transfer rates change their direction and magnitude rapidly in fast running compressor and there are significant time lags in thermal processes. Mathematical description of the heat transfer is also not straightforward task as there are various approaches used in literature with no clear conclusion which one is most accurate. Literature shows the significant differences, but the effect of using different approaches on compressor efficiency was not discussed so far. Five correlations were examined in this paper and compared focusing on efficiency of the compressor. Evaluation of the efficiency of refrigerating compressor is convenient using isentropic efficiency, which is defined by equation (1) and (2).

The analysis of heat transfer models was carried out for different wall temperatures. In first case the wall temperature was set to 350 K. Results were compared with cylinder wall temperature set to 450 K. In the last case a balanced cycle temperature was used as cylinder wall temperature. This is the temperature, which is achieved when there is no heat transfer between cylinder and surrounded components or outside gas.

3.1. Heat transfer
Heat transferred during the crankshaft cycle is presented in the figure 3, where the negative values represent the heat flux from the cylinder to the walls. Positive values stand for heat going from walls to the gas. Each process inside the cylinder is demarcated by line. Interval A limits the compression, B is the discharge process, C is the expansion and D limits the suction process. The temperature rises as the gas is compressed and when it overcomes the wall temperature the heat changes the direction. This can be seen in the figure 3. Few discontinuities occur during the discharge process in the model of Woschni

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and Disconzi. The reason is the change of the characteristic velocity in Reynolds number. Woschni uses equation (9) during the discharge or suction and equation (10) for compression and expansion. Equations used by Disconzi are shown in table 1. The valve may bounce during the suction or discharge process closing up the cylinder. The discontinuities can be also seen in figure 3. Gas in the cylinder is heated up by surrounding walls during a part of expansion and the whole suction process, which is limited by discontinuities of Woschni's and Disconzi's model.

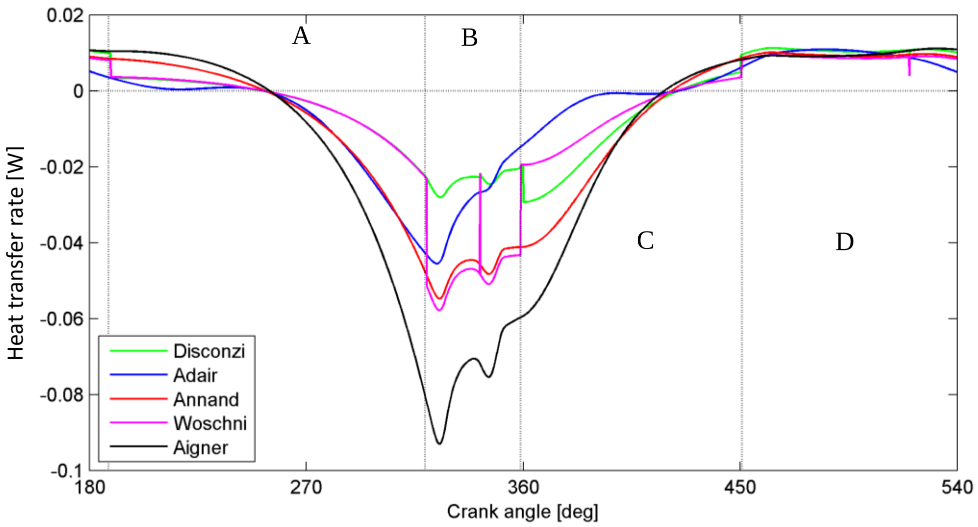

Figure 3. Heat transfer during the crankshaft cycle (T_{wall} = 350 K).

Heat transfer during the suction shows very similar behavior in all models. The total amount of transferred heat is very similar in table 4, except for the model of Disconzi and Aigner, which are slightly above all. Increase in the temperature causes reversed heat flow. The differences between models are rather significant during discharge. The models of Woschni and Annand are comparable and the values of total heat transfer rates are alike during the discharge. The reason could be in analogous expression of Nusselt number. The model of Aigner predicts the highest net heat flow during the cycle, however it must be pointed out that the Aigner adjusted Stanton numbers according to sufficient number of CFD analysis. In this work we used approach from [9] to calculate Stanton number.

The model used by Adair has different behavior during the cycle, especially after the suction and also the amount of heat transferred during discharge and re-expansion is very small. The reason could be also found in the definition of velocity and the characteristic dimension in Reynolds number. Whereas Woschni, Annand and Disconzi use mean piston velocity, Adair [7] uses the angular crankshaft speed as characteristic gas velocity.

Expression of characteristic gas velocity in all the models is simpler during the compression and expansion process, because authors usually use just piston mean speed. Nevertheless there are still significant differences in the amount of transferred heat caused by various expressions of Nusselt number or the velocity (Adair). However all models have analogous tendencies.

Heat transfer from cylinder outer surface is not known for the simulation purposes, therefore it was set as adiabatic to find balanced wall temperature. The balanced wall temperature will not be similar for all models as the net heat transfer is different. For this settings the highest amount of heat is transferred during the suction and discharge phase, see table 5 and figure 4. The total amount of heat transferred during the suction phase is very similar for all models in both cases (steady temperature and adiabatic wall). Other phases are significantly different. The differences between heat transfer models are expected since the authors use particular correlations. The effect of using various models on compressor efficiency was not examined in detail so far and this information could be beneficial for simulation tools to improve their accuracy.

Table 4. Heat transfer rate for particular models and wall temperature 350 K.

	Heat transfer rate [W]				
	Q_{cycle}	Q_{comp}	Q_{exp}	Q_{suc}	Q_{dis}
Isentropic	0	0	0	0	0
Disconzi	-13.92	-4.90	-9.31	9.73	-9.44
Adair	-17.46	-11.98	-1.77	8.09	-11.80
Annand	-30.17	-9.01	-11.93	9.10	-18.33
Woschni	-21.79	-4.66	-5.99	8.28	-19.41
Aigner	-47.26	-13.92	-14.44	9.82	-28.73

Table 5. Heat transfer rate for particular models and balanced wall temperature.

	Heat transfer rate [W]					
	T_{wall}	Q_{cycle}	Q_{comp}	Q_{exp}	Q_{suc}	Q_{dis}
Isentropic	-	0	0	0	0	0
Disconzi	383.33	0.00	-1.40	-6.09	15.51	-8.02
Adair	394.88	0.00	-5.34	-0.06	14.77	-9.37
Annand	404.69	0.00	2.78	-4.93	16.30	-14.15
Woschni	403.21	0.00	1.16	-2.40	15.79	-14.55
Aigner	421.27	0.00	7.20	-4.86	18.00	-20.33

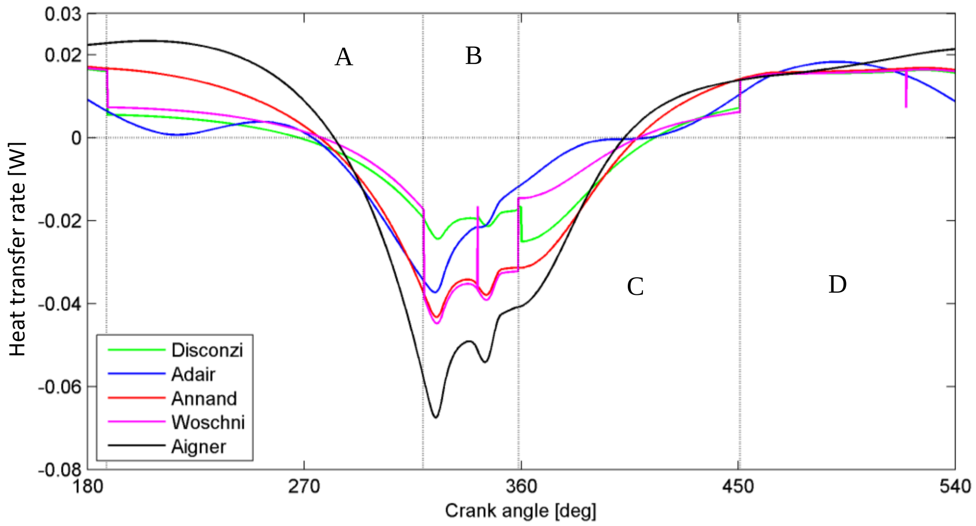

Figure 4. Heat transfer rate for particular models and balanced wall temperature.

3.2. Efficiency

The efficiency of compressor used in refrigeration is usually described with the isentropic efficiency, equation (1). Evaluation of the full crankshaft cycle shows that there is not necessarily analogy between isentropic efficiency and the amount of transferred heat. This is valid for all heat transfer models. Compression and expansion work corresponds slightly better with the bulk transferred heat. Overall the isentropic efficiency is not distinctive using different models and the differences between models are around 1 % (table 6). This is also hardly visible in indicated p-V diagram.

The initial temperature has a significant importance at the beginning of compression when determining the compressor efficiency. Higher wall temperature will cause increase in the initial gas temperature and decrease of the mass flowing into the cylinder. When the inlet temperature of the gas was set to 310 K and wall temperature was 350 K, almost all models predicted higher gas temperature at the beginning of compression by approximately 6 K, however all models were under the isentropic temperature. The model of Aigner predicted even lower temperature than it is in suction line, which is the consequence of gas re-expansion from a clearance volume. Gas overheating presented in table 6 is caused only by heat transfer from cylinder walls, the suction line is not taken into account. Decrease in the temperature at the suction helps to suck more mass inside the cylinder, which enhances the volumetric efficiency.

When the wall temperature was changed to 450 K, the mass inside the cylinder decreased by app. 6.5 %, see table 6. Hot wall heats up the gas and the specific volume raises, therefore the free volume of cylinder is filled up faster with smaller amount of mass. Change in the wall temperature by 100 K resulted in higher temperature by nearly 21 K at the beginning of compression.

Comparison of the cases with different wall temperature in table 6 showed also that the decrease in isentropic efficiency due to the raise of gas temperature at the beginning of compression is not significant, however the decrease in volumetric efficiency is more important. The difference of 21 K caused the lower mass load of the gas in the cylinder by approximately 0.31 % per 1 K. Important conclusion is the agreement of the models on this fact.

There are results of balanced cycle in the table 7. Balanced temperature is different for each model, nevertheless the isentropic efficiency remains very similar, ranging from 99.6 % to 96.9 %. Both the temperature and the mass in the cylinder after closing of suction valve are almost identical for all the models.

Table 6. Comparison of compressor with different wall temperatures.

	T_{wall} = 350			**T_{wall} = 450**		
	m_{comp_0}	T_{comp_0}	η_{isen}	m_{comp_0}	T_{comp_0}	η_{isen}
Isentropic	0.00446	318.59	100.00	0.00446	318.59	100.00
Disconzi	0.00448	317.11	98.91	0.00422	336.71	98.75
Adair	0.00449	316.60	99.99	0.00426	333.54	99.31
Annand	0.00455	312.37	98.41	0.00424	335.43	97.52
Woschni	0.00452	314.62	99.18	0.00428	332.53	98.78
Aigner	0.00462	307.80	98.03	0.00426	333.59	96.61

Table 7. Work and isentropic efficiency of compressor with balanced temperature.

	Work [J] and isentropic efficiency [%]			
	T_{wall}	m_{comp_0}	T_{comp_0}	η_{isen}
Isentropic		0.00446	318.59	100.0
Disconzi	384.80	0.00439	323.96	98.8
Adair	394.93	0.00438	324.24	99.6
Annand	404.71	0.00437	325.08	97.8
Woschni	403.21	0.00439	324.19	98.9
Aigner	421.28	0.00436	326.39	96.9

4. Conclusion

Overheating of the gas inside the compressor is the problem causing the decrease in the efficiency. In the presented paper several heat transfer models were examined to see their effect on the efficiency. It can be concluded from results that various heat transfer models do not affect compressor efficiency significantly, however the amount of transferred heat alters greatly. The results also show that overheating in the cylinder do not cause important drop in isentropic efficiency, but the difference in volumetric efficiency is rather more decisive. Further tests are still required to verify the statements for different compressors and conditions.

5. Nomenclature

A_c	[m^2]	cross section of the cylinder
c_p	[J/kg/K]	thermal capacity for const. pres.
C_f	[-]	friction coefficient
D	[m]	characteristic dimension
d	[-]	damping constant
F	[N]	force acting on valve
h	[J/kg]	specific enthalpy
k	[N/m]	spring stiffness
m	[kg]	mass
p	[Pa]	pressure
Q	[J]	rate of heat transfer
R	[J/kg/K]	gas constant
t	[s]	time
T	[K]	temperature
T_{wall}	[K]	wall temperature
u	[m/s]	velocity
u_p	[m/s]	mean piston velocity
u_c	[m/s]	flow velocity through the valve
U	[J]	internal energy
x	[m]	valve position
w	[J/kg]	specific work
W	[J]	work
κ	[-]	adiabatic exponent
η_{isen}	[%]	isentropic efficiency
ρ	[kg/m^3]	density
φ_{eff}	[m^2]	effective flow area
μ	[m^2/s]	kinematic viscosity

Index	**Meaning**	**Index**	**Meaning**
cycle	full cycle	exp	expansion
comp	compression	suc	suction
comp_0	beginning of compression	dis	discharge

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