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SIMPLE BUT EFFICIENT METHODS FOR ESTIMATION OF VALVE LOSS, CAPACITY LOSS DUE TO SUCTION VALVE THROTTLING AND HEAT TRANSFER IN CYLINDER

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ABSTRACT

The paper addresses to engineers who do design and optimisation work in the field of reciprocating compressors. Formulae based on theoretical and empirical considerations are given which allow to predict the valve losses. Suction valve losses also result in a reduction of capacity. A formula for the prediction of this capacity loss is presented. The heat transfer process from cylinder walls to gas during intake is discussed. A new approach based on heat conduction is proposed which results in a simple formula for heat transfer and the according capacity loss.

1 FORMULAE FOR THE ESTIMATION OF VALVE LOSSES

In case of small compressors the design engineer has to find the optimum apportion of the available area in cylinder head among suction and discharge valve (SV, DV). The main point in this optimization is to minimize the sum of valve losses (SV+DV), Fig.1. The authors have derived formulae for valve loss power based on general theoretical results and on computer results considering typical valve designs with reasonable valve dynamics and poor valve flutter [1]; Table 1.

The first term in equ(1) and (4) gives the main contribution, which is derived from theoretical considerations. These equations give reasonable estimations; sophisticated effects are of course not included.Normally the results are within a $\pm 15\%$ span as compared with the losses calculated with a sophisticated valve dynamics simulation [1].

The so-called intake heating factor λ_A results from gas heating during intake but in addition also from gas leakage (mainly gas leaking back through the closed discharge valve during the intake process). λ_A is the factor which brings the theoretical mass flow rate to the actual value as can be seen from equ(2) in Table 1. In the absence of specific experience equ(6) may be used:

$$\lambda_A = 1.023 - 0.023\Psi \qquad (6)$$

The smaller the value λ_A the less gas enters the cylinder and the less is the valve loss power. For optimisations one has to bear in mind also the specific power loss (e.g. in Nm/kg or in kWh/kg) related to the mass flow rate.

Additional losses occur in discharge valves of big process gas compressors due to the restriction of the flow in the pocket passage by the (discharging) piston, Fig.2. In many designs the pocket passage is already restricted when the DV opens! The correction factor f_{pi} usually is in the range from 1 to about 1.5, see [1].

Even a new compressor valve is not 100%-tight. In the compressor some portion of the gas leaks back from the cylinder into the suction plenum during the compression and discharge process. Another portion of (hot) gas —already discharged— leaks back into the cylinder during the intake process. So a certain percentage of the gas passes the valves twice. This results in a decrease of mass flow rate (hence of λ_A). The precise leakage in a working compressor is hardly to measure. Therefore there is not much reliable experimental data on leakage published, though the matter of valve tightness is an essential problem for the operation of reciprocating compressors. In equ(1) and (4) the influence of leakage in some way is included in λ_A.

Table 1 FORMULAE FOR ESTIMATION OF VALVE LOSSES

SI - units; (compressor speed n in 1/sec!)

SUCTION VALVE

P_l: Valve loss power in Watts. This quantity helps to find out the necessary power of the driving motor

$$P_l = 3.41 \cdot \frac{(V_H \cdot n)^3 \rho_s \cdot \lambda_A^2}{A_{eff}^2} \cdot (1-\varepsilon) \cdot [1+0,85(PF-1)] \cdot [1-(0.7\lambda_A Ma_m)^2] \qquad (1)$$

$\dot{m}$: mass flow rate in kg/s

$$\dot{m} = V_H \cdot n \cdot \lambda_A \cdot \rho_s \cdot [1 - \varepsilon(\psi^{\frac{1}{m}} - 1)] \qquad (2)$$

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w_l: Specific loss power or loss work per kg gas. A more convenient unit is kWh/kg gas. This quantity allows an easy estimation of the operating costs caused by the valve.

$$w_l = P_l/\dot{m} \ (Nm/kg) \quad (3) \qquad\qquad w_l(kWh/kg) = w_l(Nm/kg)/3.6 \cdot 10^6 \qquad (3a)$$

DISCHARGE VALVE

$$P_l = 3.41 \cdot \frac{(V_H \cdot n)^3 \cdot \rho_s \cdot \psi^{\frac{1}{m}} \cdot \lambda_A}{A_{eff}^2} \cdot (\frac{1}{\psi} - 0.065) \cdot (1 - 1.76 \cdot \varepsilon) \cdot [1+1.75(m-1)] \cdot PF \cdot f_{pi}$$
$$\qquad (4)$$

$$w_l = P_l/\dot{m} \ (Nm/kg) \qquad (5)$$

V_H .... swept volume	PF .... so-called "pocket factor" according to F. Bauer [4]. PF considers the additional flow losses, if the valve is situated in a pocket at the side of the cylinder. This is typical for process gas compressors. PF is defined as ratio (loss valve + pocket): (loss valve only). Typically $PF = 1.5 \div 2.5$. Diagrams for estimating PF are given in [4]. For designs where the valve ports have a direct access to the cylinder working space: $PF = 1$. Usually this is he case for hermetic refrigerant compressors.
ρ_s ..... density of gas in suction plenum	
λ_A ..... intake heating factor; in absence of specific experience use equ(6)	
A_{eff} .... effective valve flow area (valve fully open)	
Ψ ..... pressure ratio	
m ..... polytropic index	
k ...... isentropic index	
ε ...... relative clearence volume V_c/V_H	
s ...... stroke	
Ma_{m} .... mean valve Mach number S.V. $\qquad Ma_m = (A_{piston}/A_{eff})2sn/\sqrt{kP_s/\rho_s}$	f_{pi} ..... Coefficient considering the fact that PF is defined for gas flow through the passage into the discharge pocket unrestricted by the piston. $f_{pi}>1$ considers the additional losses by this restriction; see Fig. 2.
P_s ..... abs. pressure in suction plenum	

The formulae in Table 1 give a first estimation. For a more precise and detailed design work experimental values of effective valve flow area should be used. To get those, steady state flow tests with geometrical valve designs in consideration should be done. With these results and a tailored valve dynamics simulation program (see e.g. [2]) one can get excellent loss data for optimisations.

2 CAPACITY LOSS DUE TO SUCTION VALVE THROTTLING

When doing the apportion of the area between suction and discharge valve, one has to keep in mind that a consequence of the throttling process in the suction valve is not only the power loss but also a reduction in capacity, Fig.1. Throttling brings along a temperature increase ΔT_{sv} and this in turn results in a volume increase of the gas taken in. Hence the mass flow rate (capacity) of the compressor is reduced.

A short remark to avoid misunderstandings: It is well known from basic thermodynamics that a steady throttling process is isenthalpic and hence (at least for an ideal gas) isothermic, i.e. $\Delta T = 0$. But one has to keep in mind that the compressor intake process is a non steady process: contrary to steady state throttling with pressure decrease, in the intake process the pressure in suction plenum is achieved more or less also in cylinder towards the end of the intake. This results in a temperature increase ΔT_{sv}. The detailed derivations are given in the book [1].

The formula for capacity loss by suction valve throttling, expressed in reduction of volumetric efficiency $\Delta \eta_{vol}$ becomes:

$$\Delta \eta_{vol} = -\left[1 - \varepsilon\left(\Psi^{\frac{1}{m}} - 1\right)\right]\frac{w_l}{c_{cp} \cdot T_{s,pl}}$$

(7)

c_{cp}........specific heat of intake gas at const. pressure $T_{s,pl}$........Kelvin temperature in suction plenum

While throttling in the suction valve results (besides power losses) in a capacity reduction, throttling in discharge valve results in a temperature increase of the gas calling for an additional heat exchanger area. Now let us show how to use the fomulae of Table 1 in an example.

EXAMPLE

Propane gas Compressor: $s/D = 0.175/0.300$m $\rightarrow$ $V_H = 0.0124$m³; $n = 700/60 = 11.67$/sec; density of gas in suction plenum ($T_s = 310$K); $\rho_s = 8.5$kg/m³; $P_s = 5$bar; $P_d = 20$bar; $\rightarrow$ $\Psi = 20/5 = 4$; isentropic index $k = 1.15$; polytropic index $m = k = 1.15$; $\varepsilon = 0.20$ (20%); $PF = 1.7$(SV) $PF = 2$(DV); 2SV+2DV, each $A_{eff} = 17.5$cm², $f_{pi} = 1$.

Estimate: Valve losses, mass flow rate, capacity loss by SV-throttling!

In the absence of special experience we calculate the intake heating factor λ_A from equ(6): $\lambda_A = 0.931$
The mass flow rate, equ(2), becomes: $\dot{m} = 0.0124 \cdot 11.67 \cdot 0.931 \cdot 8.5 \ [1-0.2(4^{1/1.15}-1)] = \mathbf{0.61}$**kg/s**

SV loss power, equ(1)

$Ma_m = (A_{piston}/A_{eff})2sn/\sqrt{kP_s/\rho_s} = 82.5/260 = 0.317$

$P_1 = 3.41 \ \frac{(0.0124 \cdot 11.67)^3 \cdot 8.5 \cdot 0.931^2}{(2 \cdot 0.00175)^2} \cdot 0.8 \cdot (1 - 0.85 \cdot 0.7) \cdot [1-(0.7 \cdot 0.931 \cdot 0.317)^2] = 7590 \ W$

$w_l = P_l/\dot{m} = 7590/0.61 = 12400 \ Nm/kg = \mathbf{3.46 \cdot 10^{-3}} \ kWh/kg$

DV loss power, equ(4)
A similiar procedure results in: $P_l = 6745 \ W$ $w_l = 11060 \ Nm/kg = \mathbf{3.07 \cdot 10^{-3}} \ kWh/kg$

The total kWh consumption for a running time of 6000 hr a year becomes:

$W_l = (w_{l,sv} + w_{l,dv}) \cdot \dot{m} \cdot t = (3.46 + 3.07) \cdot 10^{-3} \cdot 0.61 \cdot 6000 \cdot 3600 = \mathbf{86.000} \ kWh$ a year
     This quantity can be related to the price of the valves !

Capacity loss by SV-throttling, equ(7)

$\Delta\eta_{vol} = - [1 - 0.2(4^{1/1.15}-1)] \cdot 12400/1700 \cdot 310 = - 0.0126 \hat{=} - 1.26\%$ a relatively small value!

In the same way one can use the formulae for hermetics and do optimisations as indicated in Fig.1.

3 CAPACITY LOSS DUE TO GAS HEATING DURING INTAKE

To calculate this capacity loss one has to find at first the quantity of heat Q transferred from cylinder walls to the gas during the intake period. The traditional procedure for this is marked by the following equations:

$$\dot{Q} = h(t).A(t).\Delta T \qquad\qquad Q = \int_{o}^{\Delta t_i} \dot{Q}.dt \qquad \Delta t_i......\text{suction time interval} \qquad\qquad (8)$$

with: h.......heat transfer coefficient A......(variable) surface of cylinder working space
          ΔT....temperature difference cylinder wall - gas

This heat transfer is an extremely complex and highly **non steady** phenomenon. Equ.(8) merely denotes all difficulties by the symbol "h". In the 1980 - Purdue Compressor Conference Prof. *Touber* from the Techn. Univ. of Delft presented a paper [3], entitled "Modelling of cylinder heat transfer - Large effort, little effect?". His summing up of the existing literature –applied to a specific compressor– resulted in h-values varying between a 1 to 10-fold value depending on the literature used!

The authors want to call in question the application of equ(8) to the cylinder heat transfer problem in general. The following reasons are crucial for this:

- Newtons law of heat transfer assumes implicitly the existence of a temperature boundary layer at the walls. In our problem the piston sets free the cylinder wall and "cold" gas touches this wall. A boundary layer is not existing, but has to be formed in a heat conduction process. Heat convection starts after a certain minimum formation of a temperature boundary layer.

- The duration of the whole process (one intake cycle) is in the range of 0.01 to 0.1 second. In such small periods the heat conduction process is predominant as compared to convection.

- The above is valid for the cylinder surface swept by the piston. For the rest of the surface of the cylinder working space a similar effect takes place resulting from the reexpansion of the gas in cylinder prior to intake: the temperature in (the existing) temperature boundary layers decreases rapidly more or less adiabatically in proportion to the pressure decrease.

Having in mind the physical basis of the heat transfer in cylinder a new approach based on a heat conduction model has been worked out. Heat conduction in the short period in question exceeds heat convection by far. Fig 3 gives an idea of the new approach. This approach is discussed in detail in the book [1]. The results are presented below.

Heat Q transferred from cylinder walls to gas during intake time period Δt_i:

$$Q = \frac{2}{\sqrt{\pi}} (T_{wall} - T_{gas}) \cdot A \sqrt{\lambda \rho_s c_{cp}} \cdot \sqrt{\Delta t_i} \cdot f_{Tu} \qquad\qquad (9)$$

A......total surface of cyl. working space in b.d.c. λ.......thermal conductivity of the gas at intake state
c_{cp}.....specific heat of gas at intake state f_{Tu}......coefficient bringing the theoretical heat conduction result for Q to reality; typically $f_{Tu} = 1.5 \div 2$ considering turbulence effects in addition to conduction.

For the temperature increase ΔT_H of gas during intake one gets, estimating $\Delta t_i \approx 0.3/n$ (intake time = 30% of cycle time)

$$\Delta T_H = \frac{Q}{\varrho_s V_H \eta_{vol} c_{cp}}$$

$$\Delta T_H = 0.62 \cdot \frac{(T_{wall} - T_{gas}) A f_{Tu}}{V_H \eta_{vol} \cdot \sqrt{n}} \cdot \sqrt{a} \qquad (10)$$

with $a = \lambda/\rho_s c_{cp}$ .....thermal diffusivity of gas (m^2/s)

The influence of the gas on ΔT_H is given by $\sqrt{a}$. As λ and c_{cp} are fairly independant of pressure, there is:

$$\Delta T_H \sim \frac{1}{\sqrt{\varrho_s}} \sim \frac{1}{\sqrt{p_s}} \qquad (11)$$

Looking at gas property tables it comes out that (for 1bar) the thermal diffusivity "a" is in the range of $20 \cdot 10^{-6}$ m^2/s for a great variety of gases, but about $200 \cdot 10^{-6} m^2/s(!)$ for hydrogen. Hence a hydrogen compressor will experience about a $\sqrt{10} \approx 3$-fold intake gas heating effect as compared to other gases! According to equ(11) intake gas heating is more and more reduced with increasing suction pressure. Small compressors (A/V_H is big!) and compressors with small speed n will experience a relatively higher intake gas heating.

The capacity loss is given by

$$\Delta \eta_{vol,H} \approx -\frac{\Delta T_H}{T_{s,pl}} \cdot [1 - \varepsilon \cdot (\Psi^{1/m} - 1)] \qquad (12)$$

To demonstrate the procedure we calculate the capacity loss of a small refrigerant compressor using equ(10) and (12):

Data: s/D = 25/18mm $\quad T_{wall} - T_{gas} = 60K$; $A = 20 \cdot 10^{-4} m^2$; $\quad V_H = 6.25 \cdot 10^{-6} m^3$; $\eta_{vol} = 0.55$; $\quad$ n = 48/sec;

$f_{Tu} = 2$; $a = 6 \cdot 10^{-6} m^2/s$; $\quad T_{s,pl} = 340K$; $\quad [1 - \varepsilon(\psi^{\frac{1}{x}} - 1)] = 0.9$.

from equ(10) results: $\Delta T_H = 15.3K$; equ(12)$\rightarrow \Delta \eta_{vol,H} = -0.04 \triangleq -4\%$

Hence the actual capacity reduction (0.9 - 0.55) $\triangleq$ 35% must be mainly due to leakage! The basic laws of heat transfer do not support capacity loss predictions which are essentially bigger than -4% (in this case). For big process gas compressors the capacity losses by intake heating are much smaller (exception: low pressure hydrogen compressors).

4 CONCLUSIONS

- Valve losses can be estimated easily by approximation formulae

- Intake gas heating can be estimated by a simple formula based on pure heat conduction. It turnes out that the intake heating effect is of minor importance except low pressure hydrogen compressors and very small compressors. **Valve leakage plays the important role in capacity reduction.**

5 REFERENCES

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valve loss power P_1

specific loss $P_1/\dot{m}$

mass flow rate $\dot{m}$

$A_{eff,dv}$

$A_{eff,sv}$

pocket-passage

SV

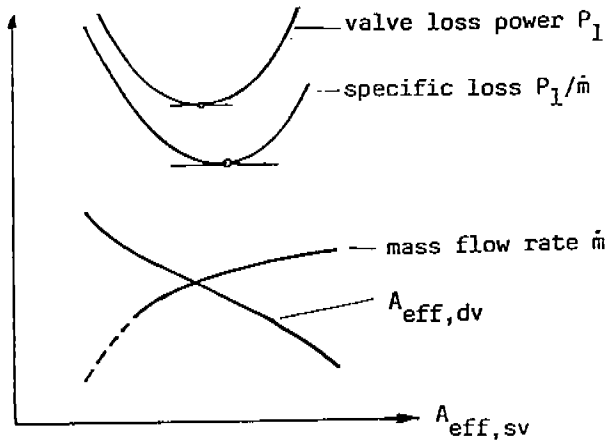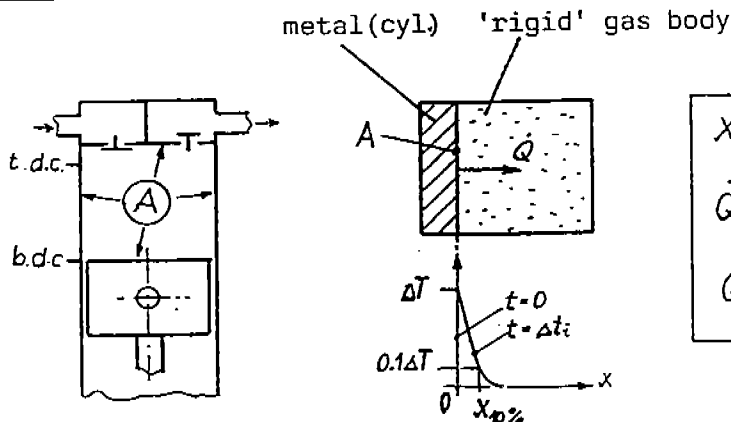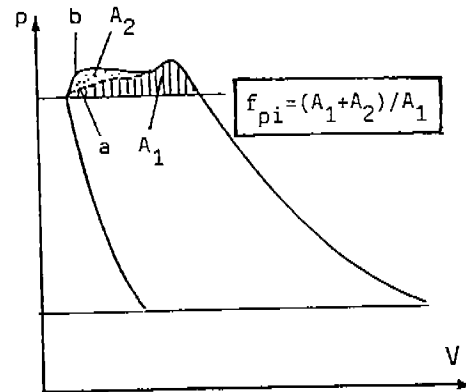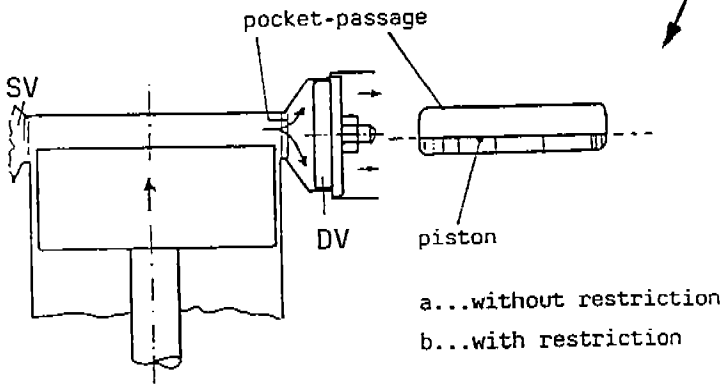DV   piston

a...without restriction
b...with restriction

Fig.1 Optimisation of valve losses for a small compressor by apportioning the available space among SV and DV. The decrease in mass flow rate $\dot{m}$ at the left side is due to suction valve throttling and due to increased clearance volume in a bigger DV.

Fig.2 Factor f_{pi} considering the additional losses in discharge valves due to the restriction of the gas flow in the pocket passage by the discharging piston (process gas compressors only)

p   b A_2

a A_1

$$f_{pi} = (A_1 + A_2)/A_1$$

V

metal(cyl.)   'rigid' gas body

A $\dot{Q}$

t.d.c.

b.d.c

ΔT $t \cdot 0$
$t \cdot \Delta t_i$

$0.1\Delta T$

0   $X_{10\%}$ x

$$X_{10\%} = 2.3 \cdot \sqrt{\alpha \cdot \Delta t_i}$$

$$\dot{Q} = \frac{1}{\sqrt{\pi}} \cdot \Delta T \cdot A \cdot \sqrt{\lambda \rho_s C_p} \cdot \frac{1}{\sqrt{t}}$$

$$Q \Big|_0^{\Delta t_i} = \frac{2}{\sqrt{\pi}} \cdot \Delta T \cdot A \cdot \sqrt{\lambda \rho_s C_p} \cdot \sqrt{\Delta t_i}$$

Fig.3 Conduction model for calculating heat transfer from cylinder walls to gas during intake.