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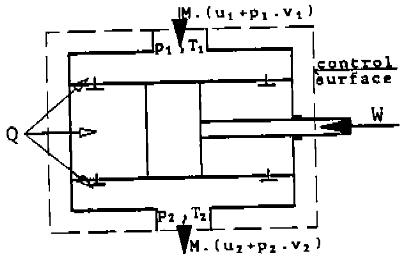
ABSTRACT

This paper discusses the importance of tightness of sealing elements. It will show that in some instances the tightness of valves in closed position has a greater effect of efficiency (volumetric and isentropic) than good flow areas when the valve is open.

1. CONSERVATION OF ENERGY EQUATION

Let us consider a compressor with one double acting cylinder. When the outside conditions are stationary then, for every complete revolution of the crankshaft, it takes in a mass of gas M (kg) at normal suction conditions p_1, T_1 from a suction pipe, and delivers M to the discharge pipe with higher pressure p_2.

Fig. 1: Schematic drawing of a double acting cylinder with a control surface around it.



If we draw a control surface around such a cylinder as shown in Fig. 1, then, for a mass of gas M (kg) entering the suction flange or leaving the discharge flange, the following amounts of energy pass through this control surface (changes in kinetic and potential energy of the gas are neglected): As **energy input**, we have +W (J) for the work from the piston rod, +Q (J) for the heat transferred from the cylinder block to the gas (in the working chambers, suction and discharge plenums), $+M \cdot u_1$ (J) for the internal energy carried by the inflowing gas, and $+M \cdot p_1 \cdot v_1$ (J) for the flow work due to the inflow of gas. In the same way, we have as **energy output** $-M \cdot u_2$ (J) for the internal energy carried by the outflowing gas, and $-M \cdot p_2 \cdot v_2$ (J) for the flow work due to the outflow of gas.

By definition the sum of internal energy and flow work is equal to enthalpy, i.e.

$$M\,(u + p \cdot v) = M \cdot h \qquad \text{(J) ... (1)}$$

According to the 1st law of thermodynamics, when conditions inside the control surface are stationary, the energy input has to be equal to the energy output, and we get

$$W + Q + M \cdot h_1 = M \cdot h_2 \qquad \text{(J) ... (2)}$$

or

$$W + Q = M \cdot (h_2 - h_1) \qquad \text{(J) ... (3)}$$

If heat exchange is neglected, a hypothesis usually not too far from reality (see [1]), i.e. if Q = 0, equation (3) becomes

$$W = M \cdot (h_2 - h_1) \qquad \text{(J) ... (4)}$$

2. THE IDEAL AND THE REAL COMPRESSOR

In the **ideal compressor** the only change of state is an isentropic pressure rise from p_1, T_1 to p_2, $T_{2,is}$. The required compression work is given with

$$W_{ideal} = W_{is} = M \cdot (h_{2,is} - h_1) \qquad (J) \ldots (5)$$

In the **real compressor** however, heat exchange may already take place between the suction plenum and the gas. It is also assumed the suction valves leak, thereby allowing hot gas to mix with the gas at T_1, thus raising the temperature of the gas in the suction plenum from T_1 to the normally higher $T_{1,spl}$. The intake process through the suction valve causes a pressure drop necessitating extra work ΔW_s, defined by the area beneath the suction line in the indicator diagram. This extra work, delivered by the driver, results in an additional increase of specific enthalpy and temperature rise to $T_{1,s}$ according to equation (6), which is equation (4) applied to the suction process:

$$W_s = M_s \cdot (h_{1,s} - h_{1,spl}) \qquad (J) \ldots (6)$$

In the same way, an extra amount of work ΔW_d is required for the discharge process, raising the enthalpy of the gas from $h_{2,c}$ (at p_2, $T_{2,c}$ at the end of the compression) to $h_{2,dpl}$ (at p_2, $T_{2,dpl}$ in the discharge plenum):

$$W_d = M_d \cdot (h_{2,c} - h_{2,dpl}) \qquad (J) \ldots (7)$$

With leaking valves, the mass M_s of the gas flowing through the suction valve during the suction phase in the normal direction is bigger than the mass of gas M passing the whole cycle from the suction to the discharge flange, since part of M_s leaks back when the suction valve is closed, and the same is true of the mass M_d passing through the discharge valve during the discharge phase.

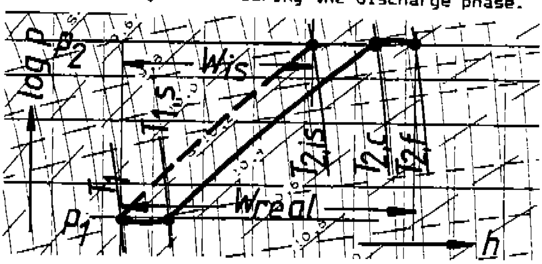


Fig. 2:

Changes of state of the gas flowing from the suction flange to the discharge flange of a compressor cylinder.

Each of these streams of gas leaking back represents a loss of compression work already supplied by the driver. They also carry enthalpy from the chambers with the higher pressure and heat to chambers with lower pressure and temperature thus raising there entropy, enthalpy and temperature in a most undesirable way. In fact, reciprocating compressors are volumetric machines, such that the amount of work required for intake, compression and delivery of a given mass of gas M will be the smaller, the smaller its volume, i.e. the lower its temperature and viceversa. Let Δh_{irr} be the increase of enthalpy due to all irreversible dissipations of energy until the gas finally leaves the compressor through the discharge flange, i.e.

$$h_{2,f} = h_{2,is} + \Delta h_{irr} \qquad (J/kg) \ldots (8)$$

then equation (4) for the real compressor becomes

$$W_{real} = M \cdot (h_{2,is} + \Delta h_{irr} - h_1) = M \cdot (h_{2,f} - h_1) \qquad (J) \ldots (9)$$

Dividing equation (5) by equation (9) we get the ratio of work required to compress a mass of gas M from p_1, T_1 to p_2, isentropically in an ideal machine to the work in the real one. This ratio is called the **isentropic efficiency**.

$$\eta_{is} = W_{is}/W_{real} = (h_{2,is} - h_1)/(h_{2,f} - h_1) \approx$$

$$\approx (T_{2,is} - T_1)/(T_{2,f} - T_1) \qquad (\%/100) \ldots (10)$$

In the last line of equation (10), we replaced the differences of enthalpy with the differences of temperature, which is allowable under the assumption of ideal gas behaviour and constant specific heat between T_1 and $T_{2,f}$. This means the isentropic efficiency of a compressor can be determined knowing T_1, $T_{2,f}$ and $T_{2,is}$.

It should also be noted that the term "**volumetric efficiency**", defined as the length of the suction stroke, has to be used to calculate indicated power, but is not appropriate to find M, the mass of gas delivered per revolution, because it contains no information about temperatures or leakage losses. To correlate M with the nominal suction density ρ_1 and the stroke volumes of both compression chambers $V_{stroke,HE}$ and $V_{stroke,CE}$, it seems better to use the term "**effiency of delivery**" denoted by the greek letter λ,

$$\lambda = 100 \cdot M/(\rho_1 \cdot (V_{stroke,HE+CE})) \qquad (\%) \quad ...(11)$$

It can be determined empirically by measuring the mass of gas M delivered by a compressor cylinder during one revolution of the crankshaft, or theoretically by integrating gas flow through all suction valves or all the discharge valves over one revolution of the crankshaft.

3. ESTABLISHING THE MODEL

The following indices shall be used:

k=1 for the working chamber head end
k=2 for the working chamber crank end
k=3 for the suction plenum
k=4 for the discharge plenum
k=5 for a clearance pocket on the head end

By differentiating the equation of conservation of energy (3) with respect to crank angle X, the rotative speed of the crankshaft assumed to be constant, i.e. dX/dt is constant, the energy balance in every chamber "k" is

$$dW_k/dX + dQ_k/dX + dE_k/dX = dU_k/dX \qquad (J/rad) \; ... \; (12)$$

The term dU_k/dX on the right hand side signifies that the conditions are **no more stationary.** Any external unbalance of the energy passing through the control surface remains in the system "k" to change its internal energy U_k. The differential quotients of (12) have the following significances:

For the work input from the piston rod:

$$dW_k/dX = - p_k \cdot dV_k/dX \qquad (J/rad) \; ... \; (13)$$

The expression is negative since if work is delivered to the gas from outside its volume must decrease. It should be noted that no work can be done in suction and discharge plenums or in the clearance pockets, i.e. $dW_3/dX = 0$, $dW_4/dX = 0$ and $dW_5/dX = 0$.

For the heat transferred from the walls, we have

$$dQ_k/dX = f(A_k, Re_k, Nu_k, Pr_k) \qquad (J/rad) \; ... \; (14)$$

We shall leave the term **dQ/dX** as it is and refer to the bibliography [1] and [2] for determining the instantaneous rate of heat exchange between the walls of the chamber and the gas therein as a function of the instantaneous wall area A_k of the chamber "k", the temperature difference between wall and gas and the coefficient of heat transfer α, which in turn is a function of the Reynolds number Re_k, the Nusselt number Nu_k and the Prandtl number Pr_k of the gas whirls therein.

dE_k/dX represents the total change of enthalpy due to streams of gas flowing into (positive sign) and out of (negative sign) the chamber "k" through different orifices "j" with gas state functions p_i, T_i, h_i upstream of each orifice. The orifices represent open or leaking valves, etc.

$$dE_k/dX = \sum (h_i - h_o) \cdot dM_j/dX \qquad \text{(J/rad) ... (15)}$$

We wrote dE_k/dX and not dH_k/dX since enthalpy H or specific enthalpy h are functions of state (see [3], vol. 1, chap. 2) and must have a complete differential, which is not the case for the right hand side of equation (15).

The gas stream dM_j/dX through an an orifice "j" from a neighbouring chamber with the condition p_i, T_i and specific enthalpy h_i carries the energy $h_i \cdot dM_j/dX$. To obtain the complete energy change due to the in- and outflowing gas ee have to total all $h_i \cdot dM_j/dX$ for all orifices "j", see equations (33) through (36).

For the specific enthalpy h_i of a real gas at p_i, T_i, we have the relationship

$$h_i - h_o = \int_o^i c_{v,o} \cdot dT + \int_o^i \left[T \cdot \left(\frac{\partial p}{\partial T}\right)_v - p\right] \cdot dv + R \cdot T_i \cdot Z_i =$$
$$= \int_o^i c_{v,o} \cdot dT + R \cdot T_i \cdot (Z_i + \chi_i) - R \cdot T_o \qquad \text{(J/kg)...(16)}$$

where

- h_o is the specific enthalpy at the ideal gas state p_o, T_o where p_o is a pressure tending to zero. T_o can be chosen arbitrarily, since we are looking only at monophase systems and are interested in differences of (specific) enthalpy only. In this case the specific heat c_v in the 1st term of the right hand side of (16) is the $c_{v,o}$ of the ideal gas. Their values for different temperatures can be taken from published tables [2], [3]. The first integral of (16) can therefore be evaluated.

- $R \cdot T_i \cdot Z_i$ stands for $p_i \cdot v_i$ in equation (1), at p_o, T_o we have of course $Z_o = 1$ and $\chi_o \triangleq 0$.

- $R \cdot T_i \cdot \chi_i$ is the evaluation of the second integral in the first line of equation (16). It is the real gas deviation of the specific internal energy u_i at p_i, T_i. The formulas needed for evaluating the second integral on the basis of the Redlich-Kwong or the Redlich-Kwong-Soave equation of state can be taken from any good textbook, for example [3] or [4]. For dU_k/dX we get

$$U_k - U_o = M_k \cdot (u_k - u_o) \qquad \text{(J) ... (17)}$$

$$u_k - u_o = \int_o^k c_{v,o} \cdot dT + R \cdot T_k \cdot \chi_k \qquad \text{(J) ... (18)}$$

$$dU_k/dX = (u_k - u_o) \cdot dM_k/dX + M_k \cdot du_k/dX =$$
$$= \left(\int_o^k c_{v,o} \cdot dT + R \cdot T_k \cdot \chi_k\right) \cdot dM_k/dX \; +$$
$$+ R \cdot M_k \cdot T_k \left\{\left[c_{v,o}/(R \cdot T_k) + \chi_k/T_k + \partial\chi_k/\partial T\right] \cdot dT_k/dX \; +\right.$$
$$\left. + (\partial\chi_k/\partial p) \cdot dp_k/dX\right\} \qquad \text{(J/rad) ... (19)}$$

Finally, the change of mass in a chamber "k" is the sum of mass flows through all orifices "j" as in equation (15), i.e.

$$dM_k/dX = \sum dM_j/dX \qquad \text{(kg/rad) ... (20)}$$

From equations (12) through (20) we can by rearranging establish a linear differential equation for each chamber, still contained both differential quotients of pressure and temperature with respect to crank angle. To separate them we need another equation, like

$$p_k \cdot V_k = Z_k \cdot M_k \cdot R \cdot T_k \qquad \text{(J) ... (21)}$$

which after differentiating with respect to crank angle becomes

$$(1/p_k)\cdot dp_k/dX + (1/V_k)\cdot dV_k/dX = (1/Z_k)\cdot dZ_k/dX +$$

$$+ (1/M_k)\cdot dM_k/dX + (1/T_k)\cdot dT_k/dX \qquad (1/rad)\ ...\ (22)$$

The derivatives dp_k/dX and dT_k/dX can be found by solving equations (12) and (22). They are also contained in terms like dZ_k/dX and $d\chi_k/dX$, since

$$dZ_k/dX = (\partial Z_k/\partial p)\cdot(dp_k/dX) + (\partial Z_k/\partial T)\cdot(dT_k/dX) \qquad (1/rad)...(23)$$

The solution for $d\chi_k/dX$ is similar. The partial differential ***quotients*** (23) can be solved as long as an analytical expression for Z_k and χ_k are available, for example from the Redlich-Kwong or the Redlich-Kwong-Soave equation of state. Finally, after separating derivatives and rearranging we get one ordinary, linear differential equation for the variation of pressure p_k in each chamber "k" with respect to crank angle X

$$\frac{dp_k}{dX} = \frac{1}{B\cdot C + A\cdot D}\cdot\left[-\frac{C+A}{V_k}\cdot\frac{dV}{dX} + \frac{C}{p_k\cdot V_k}\cdot\frac{dQ}{dX} + \sum\frac{dM_j}{dX}\cdot\frac{C\cdot E + A}{M_k}\right] \qquad (Pa/rad)\ ...\ (24)$$

Solving for T_k with respect to X, we have:

$$\frac{dT_k}{dX} = \frac{1}{B\cdot C + A\cdot D}\cdot\left[-\frac{D-B}{V_k}\cdot\frac{dV}{dX} + \frac{D}{P_k\cdot V_k}\cdot\frac{dQ}{dX} + \sum\frac{dM_j}{dX}\cdot\frac{D\cdot E - B}{M_k}\right] \qquad (K/rad)\ ...\ (25)$$

where

$$A = (1/Z_k)\cdot\left[c_{v,o}/(R\cdot T_k) + \chi_k/T_k + \partial\chi_k/\partial T\right. \qquad (1/K)\ ...\ (26)$$

$$B = (1/Z_k)\cdot\partial\chi_k/\partial p \qquad (1/Pa)\ ...\ (27)$$

$$C = (1/T_k) + (1/Z_k)\cdot\partial Z_k/\partial T \qquad (1/K)\ ...\ (28)$$

$$D = (1/p_k) - (1/Z_k)\cdot\partial Z_k/\partial p \qquad (1/Pa)\ ...\ (29)$$

$$E = \left[\left(\int_k^i c_{v,o}\cdot dT + R\cdot(T_i\cdot\chi_i - T_k\cdot\chi_k + T_i\cdot Z_i)\right]/(R\cdot Z_k\cdot T_k) \qquad (-)\ ...\ (30)$$

For the ideal gas $Z_o = 1$ and $\chi_o = 0$, therefore all derivatives of Z and being then equal to zero and we get

$$A_o = c_{v,o}/(R\cdot T_k) = 1/(T_k\cdot(\kappa-1)) \qquad (1/K)\ ...\ (26.a)$$

$$B_o = 0 \qquad (1/Pa)\ ...\ (27\cdot a)$$

$$C_o = (1/T_k) \qquad (1/K)...(28.a)$$

$$D_o = (1/p_k) \qquad (1/Pa)...(29\cdot a)$$

$$E_o = (\kappa\cdot T_i/T_k - 1)/(\kappa-1) \qquad (-)\ ...\ (30.a)$$

The actual volume of the head end compression chamber is

$$V_{HE} = (1/2)\cdot\Omega_{HE}\cdot V_{stroke,HE} \qquad (m^3)\ ...\ (31)$$

where the volume function Ω_k for any chamber "k" is defined as

$$\Omega_k = 2\cdot V_k/V_{HE} \qquad (-)\ ...\ (32)$$

With crank radius r and connecting rod length l, we get for the head end, the crank end working chamber and a plenum

$$\Omega_{HE} = 1 + 2\cdot s_{HE} - \cos X + (r/2\cdot l)\cdot\sin^2 X \qquad (-)\ ...\ (32.a)$$

$$\Omega_{CE} = \frac{A_{piston,CE}}{A_{piston,HE}} \cdot (2 + 2\cdot s_{HE} + 2\cdot s_{CE} - \Omega_{HE}) \qquad (32.b)$$

$$\Omega_{plenum} = 2 \cdot V_{plenum}/V_{stroke,HE} \qquad (-) \ldots (32.b)$$

Let $\omega = dX/dt$ (rad/s) be the rotational speed of the crankshaft. Then we get for the following expression for the term $(dM_j/dX)/M_k$

$$\frac{dM_j}{dX} = \frac{dt}{dX} \cdot \frac{dM_j}{dt} = \frac{1}{\omega} \cdot \phi_j \cdot \rho_i \cdot \sqrt{2\cdot R\cdot T_i \cdot Z_i} \cdot \Psi_j \qquad (kg/rad) \ldots (33)$$

Inserting this in equations (24) and (25) we get the actual gas flow rate in a valve "j", with gas state p_i, T_i upstream of the valve, divided by the mass of gas M contained in chamber "k". Expansion ratio ε_j in valve "j" is given by

$$\varepsilon_j = p_{downstream}/p_{upstream} \qquad (-) \ldots (34)$$

where of course ε_j must be within the limits

$$(2/(m+1))^{m/(m-1)} = \varepsilon_{cr} =< \varepsilon_j =< 1 \qquad (-) \ldots (35)$$

Ψ_j being the expansion function in the valve "j". For isenthalpic throttling, with negligible upstream and downstream velocities we get:

$$\Psi_j = \sqrt{\frac{m}{m-1} \cdot \left[\varepsilon^{2/m} - \varepsilon^{(m+1)/m}\right]} \approx \sqrt{(1-\varepsilon) \cdot \varepsilon^{1.526/m}} \qquad (-) \ldots (36)$$

where m is the exponent of isentropic volume change. For M_k the mass of gas in the chamber "k", whose volume is given by $V_{stroke,HE} \cdot \Omega_k/2$, we get

$$M_k = (\Omega_k/2) \cdot V_{stroke,HE} \cdot \rho_1 \cdot \frac{p_k \cdot T_1 \cdot Z_1}{p_1 \cdot T_k \cdot Z_k} \qquad (kg) \ldots (37)$$

With the mean gas velocity $v_{m,j}$ according to API618(1986), not in the geometric but in the equivalent valve area ϕ_j, however always calculated for the head end cylinder side

$$v_{m,j,HE} = (V_{stroke,HE}/\phi_j) \cdot \omega/\pi \qquad (m/s) \ldots (38)$$

we obtain, by substituting the expressions according to equations (31) through (37) in (33)

$$\frac{dM_j}{dX} \cdot \frac{1}{M_k} = \frac{2\cdot\sqrt{2}}{\pi \cdot v_{m,j}} \cdot \sqrt{R \cdot T_1 \cdot Z_1} \cdot \frac{p_i \cdot T_k \cdot Z_k}{p_k \cdot T_i \cdot Z_i} \cdot \sqrt{\frac{T_i \cdot Z_i}{T_1 \cdot Z_1}} \cdot \frac{\Psi_j}{\Omega_k} =$$

$$= \frac{1}{\sqrt{q_{j,HE}}} \cdot \frac{p_i \cdot T_k \cdot Z_k}{p_k \cdot T_i \cdot Z_i} \cdot \sqrt{\frac{T_i \cdot Z_i}{T_1 \cdot Z_1}} \cdot \frac{\Psi_j}{\Omega_k} \qquad (1/rad) \ldots (39)$$

In the valve throttling constant $q_{j,HE}$, we have lumped together all constants depending on the geometry of valve ϕ_j, the rotational speed ω, the stroke volume of the head end working chamber $V_{stroke,HE}$, as well as gas properties and gas duty $R.T_1.Z_1 = p_1.v_1 = p_1/\rho_1$, i.e.

$$q_{j,HE} = (\pi^2/8) \cdot (\rho_1/p_1) \cdot v_{m,j}^2 \qquad (-) \ldots (40)$$

This throttling constant $q_{SV,HE}$ for the head end suction valve can be interpreted as the maximum pressure drop in the suction valve, divided by p_1. We proceed as follows: By assuming ideal gas behaviour, i.e. using equations (26.a) through (30.a), a long connecting rod (r/l=0), and small maximum pressure drops in the suction valve $\Delta p_{max,SV,HE}$, such that $T_i/T_k = T_{1,spl}/T_{HE} \approx 1$, where

$$\Delta p_{max,SV,HE} = p_1 - p_{min,HE} = p_1 \cdot (1-\mathcal{E}_{min,HE}) \qquad (Pa) \; ... \; (41)$$

and $\Psi_{SV} \approx \sqrt{(1-\mathcal{E})}$, then we have for this maximum $dp_{HE}/dX=0$, $X=\pi/2$ and $d\Omega_{HE}/dX=1$, and it follows from equation (24) that

$$q_{SV,HE} \approx \Delta p_{max,SV,HE}/p_1 \qquad (-) \; ... \; (42)$$

When solving equations (24) and (25) simultaneously, the mass of gas M delivered per revolution is found by integrating gas flow in both directions (normal and leaking) through suction or discharge valves, indicated power from integrating equation (13), and final temperature $T_{2,f}$ as the mean temperature in the discharge plenum.

4. Results

The model was run with hydrogen H_2 and methane CH_4 at the following pressures and temperatures

p_1 = 16 bara = 232 psia, p_2 = 50 bara = 725 psia,
T_1 = 30 °C = 86 °F, such that
$T_{2,is}$ = 146.87 °C = 296.37 °F when compressing H_2, and
$T_{2,is}$ = 119.83 °C = 247.69 °F when compressing CH_4

with clearance volume ratios 13.2% on HE, and 12.82% on CE, the piston area on the CE side was 94.2% of the one on the HE side due to the piston rod.

The size of the equivalent passage areas $\Phi(v,n)$ of the opened valves was respectively equal to 80 cm², 40 cm², and $80/\sqrt{10}$ =25.30 cm², to have normal, 4-fold and 10-fold nominal valve throttling for intake and discharge. The nominal leakage areas $\Phi(v,1)$ of the closed valves were set equal to 0.04 cm². This value could be multiplied by a valve leakage factor, such that valve leakage factor zero means tight valves, and valve leakage factor 10 means leakage area 0.4 cm². Similarly, for piston rings, leakage factor zero means that the latter were assumed to be tight, and dQ -factor zero means calculation without heat exchange between the gas and the walls of working chambers and plenums.

For each case, four diagrams were plotted:

- p/p_1 versus crank angle,
- p/p_1 versus crank piston travel,
- T/T_1 versus crank angle,
- T/T_1 versus crank piston travel.

Those diagrams that differ only by the leakage factors were plotted one over the other. The tables beneath these diagrams give, for each set of leakage factors, the following results:

VE = volumetric efficiency (= efficiency of delivery λ) acc. to (11),
$T_{2,f}$ = final temperature in the discharge plenum,
P = indicated power acc. (13) neglecting the negative sign,
P/VE = indicated power P divided by volumetric efficiency VE to give an idea about the increase of specific power,

For the two extreme cases, namely

- H_2-service with large valves (Φ=80cm², q=0.68%), and
- CH_4-duty with the smallest valves (Φ=25.3cm², q=56.13%),

the computed diagrams of pressure and temperature versus piston travel and crank angle are shown in Figures 4 and 5.

Figures 3a through 3c give plots versus leakage factors of

- final temperature $T_{2,f}$ (°C),
- volumetric efficiency ("efficiency of delivery") VE (%),

- indicated power/vol. efficiency, which, in a way, shows the variation of specific power consumption.

5. COMMENTING THE RESULTS:

5.1. Influence of valve leakage on pressure and temperature variations.

It can be seen from figures 4 and 5, that as valve leakage increases,

- there is little variation in cylinder pressure, such that leakages of valves can hardly be detected by taking indicator diagrams,
- there is a considerable impact on the variations of temperature, resulting in a rise of final temperatures. According to equation (10), these variations are a direct measure of isentropic effiencies.

5.2. Influence of valve leakage on efficiency of delivery (volumetric efficiency) and indicated power.

It may be trivial to state that valve leakage must have a direct influence on the efficiency of delivery, but it is surprising to see to what extent the latter can be reduced when leakage factors are increased from 0 to 10: In the case of H_2, there is a fall from 83% to 54%, with CH_4 from 80% to 70% approximately, for valve throttling coefficients q smaller than 25%.

Indicated power rises as valve leakage increases: In fact, when suction and discharge valves have identical leaking areas that are not too large, the curves of compression and expansion are deflected towards the outside of the indicator diagram (see figures 4 and 5, compression line), with the surface increasing as a consequence of this, always on condition that suction and discharge pressures remain unchanged.

From the changes in gas flow rate and indicated power, the change in specific indicated power "power/VE" can be computed.

5.3. Influence of valve throttling on the performance figures of the tight compressor.

From equations (6) and (7) it can be seen that the increment ΔW of indicated power during intake and delivery required to overcome valve throttling results in an increase in enthalpy of the gas. The corresponding temperature rise decreases the mass of gas contained in a working chamber at the end of the intake stroke, and thus decreases the efficiency of delivery.

For **hydrogen** H_2, as valve throttling is increased by the factor 10, from **q=0.68%** to **q=6.8%**, $T_{2,f}$ rises from **150.9°C** to **159.9°C**, and **VE** falls from **83.78%** to **82.8%** only.

For **methane** CH_4, a reduction of equivalent valve area from 80 cm^2 to 25.30 cm^2 results in a rise of 5.61% to 56.1% for q. This produces an increase in final temperature $T_{2,f}$ of **127.7°C** to **163.9°C**. Simultaneously, **VE** falls from **81.1%** to **67.4%**!. Nevertheless, for q-values bigger than 16%, part of the loss in efficiency of delivery is due the cylinder pressure reaching suction pressure after the piston has passed the outer dead center position (in this case a real loss in volumetric efficiency, to be considered when calculating indicated power), and part due to heating of the gas.

5.4. Relative importance of valve throttling and valve leakage.

Let us compare two compressors by looking at curves **n=1** and **n=3**, or **n=4** and **n=6** in Fig. 3. One compressor has small valves with equivalent valve areas for normal flow $\Phi(v,n)$ = **56.57** cm^2, the other has $\Phi(v,n)$ = **80.00** cm^2. If the small valves are tight, the large ones leaking, performance data will equalize, when working with methane (CH_4), curves n=4 and n=6,

- as to $T_{2,f}$ with leakage factor 5.7,
- as to specific power (power/VE) with leakage factor 4,
- as to VE with leakage factor 3.7.

When both compressors work on hydrogen (H_2), then valve size, whether $\Phi(v,n)$ = 80, 40 cm^2 or even 25.3 cm^2 has almost no importance: The advantage of the bigger but leaking valves is completely offset

- as to $T_{2,f}$ with leakage factor 2.6,
- as to specific power (power/VE) with leakage factor 1.7,
- as to VE with leakage factor 0.8.

The differences between calculated results for $T_{2,f}$ and specific power - which, according to equation (10), should vanish or almost - come mainly from the fact that, in the calculation, pressures and hence power figures converge faster than temperatures, and calculated final temperatures are perhaps somewhat optimistic, depending on the termination of an iteration process.

5.5. Conclusion

The above comments clearly show the importance of tightness of compressor valves. When compressing light gases, good valve tightness becomes considerably more important than good flow passages. An ever increasing percentage of newly built reciprocating compressor are for the compression of hydrogen! Overall valve performance, i.e. the cumulated effect of normal flow passages and tightness of the closed valve, can easily be measured on site by measuring the discharge temperature as close as possible to the valves, in addition to the commonly taken readings.

6. NOMENCLATURE

Symbols:

A	m^2	area
p	Pa	absolute pressure
T	K	absolute temperature
V	m^3	volume
VE	%/100	volumetric efficiency
λ	%/100	efficiency of delivery
v	m^3/kg	specific volume
v	m/s	velocity
ρ	kg/m^3	gas density = 1/v
t	s	time
M	kg	mass of gas
R	J/kg.K	specific gas constant = 8314.51/mol.weight
Z	-	real gas compressibility factor
Q	J	quantity of heat exchanged with walls
W	J	mechanical work
X	rad	crank angle
Φ	cm^2	equivalent area of a valve
Ψ	-	expansion function, equation (36)
Ω	-	volume function, equation (32)
ω	rad/s	rotational speed
H	J	enthalpy

h	J/kg	specific enthalpy
V	m^3	volume
s	%/100	clearance volume ratio
U	J	internal energy
u	J/kg	specific internal energy
χ	-	real gas deviation of spec. internal energy, divided by R.T
E	J	energy
c_v	J/kg.K	specific heat capacity at constant volume
c_p	J/kg.K	specific heat capacity at constant pressure
	-	Cp/Cv, for the ideal gas only
v_m	m/s	mean gas velocity in
q	-	valve throttling coefficient
m	-	volume change exponent of an isentropic

Suffixes:

1	nominal suction state p_1, T_1
2	real discharge state, at p_2, T_2
is	isentropic discharge state, at p_2, $T_{2,is}$
f	final state, normally at p_2, $T_{2,f}$
o	ideal gas state
spl	in the suction plenum, at p_1, T_{spl}
dpl	in the discharge plenum, at p_2, T_{dpl}
s	in a working chamber, at the end of the suction stroke, at $p=p_1$, T_s
c	in a working chamber, at the end of the compr. stroke, at $p=p_2$: T_c, h_c
k	index of chamber "k", e.g. p_k, T_k
j	index of a valve (or throttling orifice) "j", e.g. to describe the gas flow rate in it: dM_j
i	index of chamber upstream of a throttling orifice
HE	head end side of double acting cylinder
CE	crank end side of a double acting cylinder

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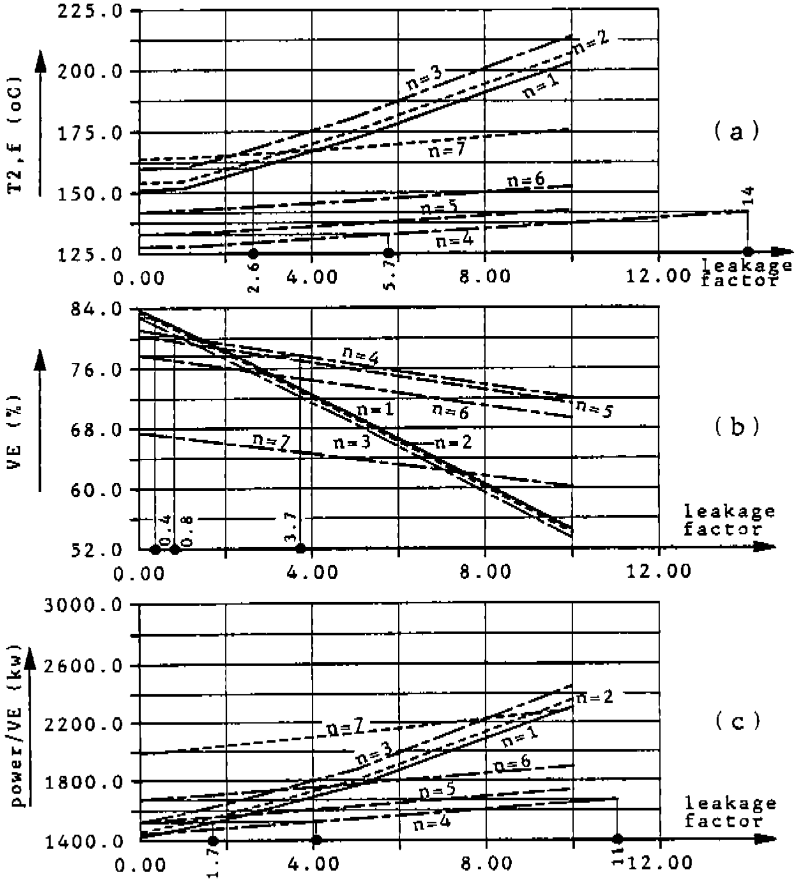
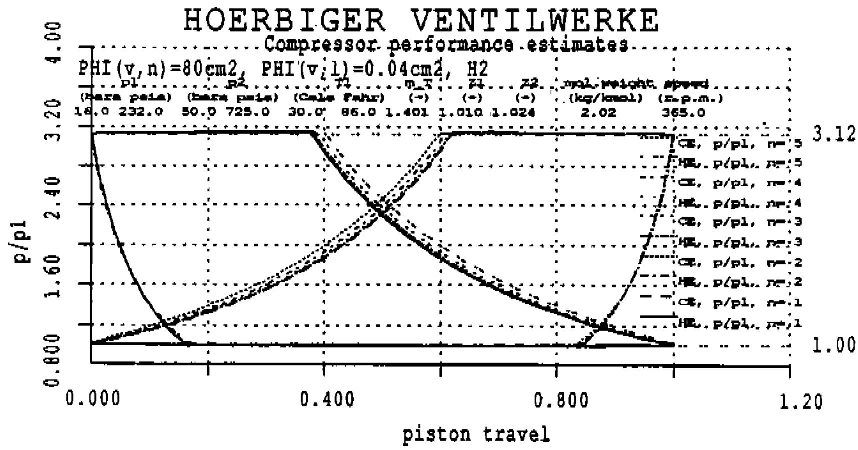
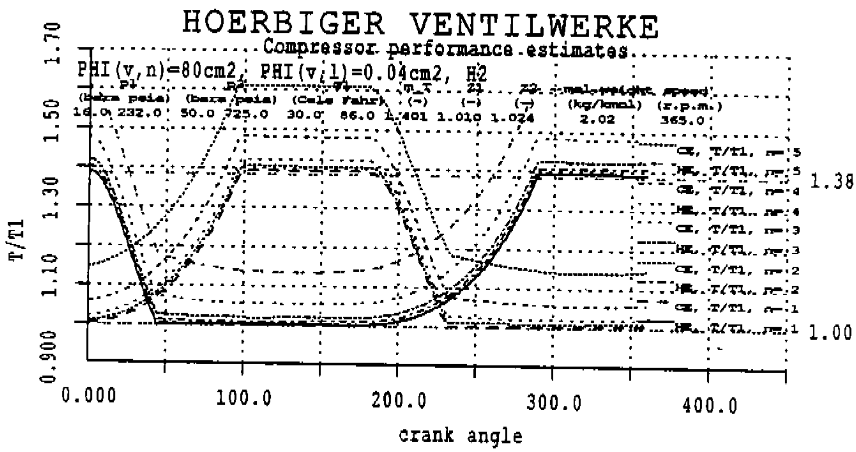


	gas	PHI(v,n) (cm2)	PHI(v,l) (cm2)	q(v,n) (%)	q(v,leak) (%)
n=1:	H2	80.00	0.04	0.680	2.72E+06
n=2:	H2	40.00	0.04	2.719	2.72E+06
n=3:	H2	25.30	0.04	6.797	2.72E+06
n=4:	CH4	80.00	0.04	5.614	2.45E+07
n=5:	CH4	56.57	0.04	11.227	2.45E+07
n=6:	CH4	40.00	0.04	22.455	2.45E+07
n=7:	CH4	25.30	0.04	56.131	2.45E+07

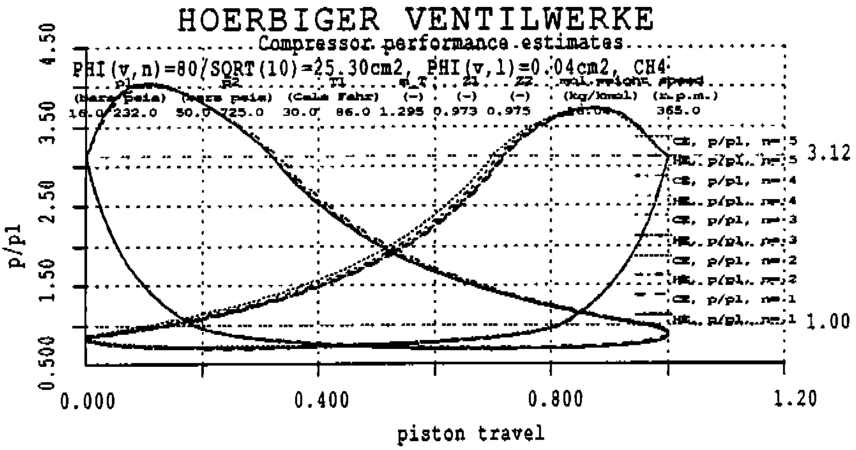
Fig.3-Variation of final temperature, volumetric efficiency and specific power with leakage factor.

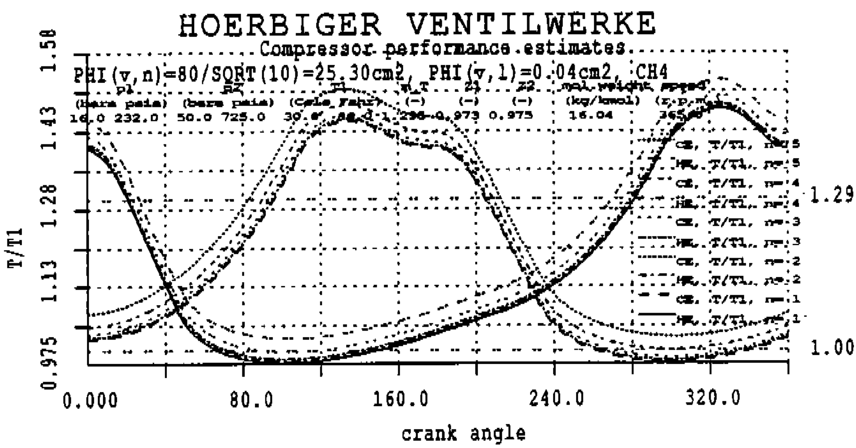




n	SV	DV	pist.	dQ	speed	T2,f	VE	dM/dt	ind.pw.	sp.pw.
	leakage	factors		-	r.p.m.	oC	%	t/h	kW	kW/VE
1	0.0	0.0	0.0	0.0	365.00	150.93	83.78	2.48	1193.81	1425.00
2	1.0	1.0	0.0	0.0	365.00	151.67	81.11	2.40	1200.43	1480.02
3	2.0	2.0	0.0	0.0	365.00	156.59	78.36	2.32	1207.15	1540.51
4	5.0	5.0	0.0	0.0	365.00	172.25	69.60	2.06	1226.52	1762.14
5	10.0	10.0	0.0	0.0	365.00	203.38	54.47	1.61	1257.13	2307.74

Fig. 4 - Hydrogen (H2), q(v,n)=0.68%
above: p/p1 = f(piston travel)
below: T/T1 = f(crank angle)





n	SV	DV	pist.	dQ	speed	T2,f	VE	dM/dt	ind.pw.	sp.pw.
	leakage	factors		-	r.p.m.	oC	%	t/h	kW	kW/VE
1	0.0	0.0	0.0	0.0	365.00	163.93	67.40	16.42	1338.33	1985.76
2	1.0	1.0	0.0	0.0	365.00	164.36	66.70	16.25	1342.50	2012.68
3	2.0	2.0	0.0	0.0	365.00	165.22	65.96	16.07	1346.23	2041.13
4	5.0	5.0	0.0	0.0	365.00	168.48	63.86	15.56	1358.22	2126.74
5	10.0	10.0	0.0	0.0	365.00	175.91	60.15	14.66	1377.08	2289.35

Fig. 5 - Methane (CH_4), q(v,n)=56.13%
above: p/p1 = f(piston travel)
below: T/T1 = f(crank angle)