1992

Valve Throttling, Its Influence on Compressor Efficiency and Gas Temperatures, Part II

E. H. Machu
Hoerbiger Ventilwerke A.G.; Austria

VALVE THROTTLING, ITS INFLUENCE ON COMPRESSOR EFFICIENCY AND GAS TEMPERATURES

Part II. Zero load and half load operation

by Erich H. Machu
HOERBIGER VENTILWERKE A.G.
Vienna, Austria

Abstract

When completely unloaded, gas temperatures in the suction or discharge plenum of a reciprocating compressor cylinder depend only on the equilibrium between work losses in the valves, and heat exchange between the gas and different coolants as cooling water or ambient air. At half load, part of the heat generated in the unloaded end is still removed from the cylinder by heat exchange, the other part by the gas passing through the loaded end, the corresponding temperature changes are estimated.

7 COMPLETE UNLOADING, ESTIMATION OF STATIONARY PLENUM TEMPERATURES AT ZERO LOAD OPERATION

In part I, we have shown that the enthalpy of the compressed gas is increased due to work losses in the valves, due to heat exchange and due to hot gases leaking back. With a loaded cylinder end, and as long as the efficiency of delivery is bigger than zero, most of this extra heat is carried by the gas to the aftercooler where it is removed. At zero load there is no compression and no isentropic enthalpy increase, the heat input resulting from the work ΔW required to overcome the pressure drops in the unloading mechanism and dissipated in the gas has to be removed by heat exchange between the gas and the different parts of the cylinder block, from where it is passed on to cooling water and ambient air. As long as there is more work put into the system from the driver to overcome the pressure drops in the valves and unloaders than heat is removed by heat exchange, the internal energy and hence the temperature inside the control surface will rise. In doing so, the gas will become thinner, thus reducing the work lost in the valves and hence the work input. On the other hand, a rising gas temperature will intensify the heat transfer from the gas towards the outside. Stationary zero load (suction or discharge) plenum temperature will be reached when the work input from the driver per crankshaft revolution exactly equals the heat removed by heat transfer, and this algorithm is used to get estimates for zero load gas temperatures in suction or discharge plenum, as shown in Figure 2. Friction between liner and piston rings, or rod and packings is neglected.

7.1 Zero load obtained by suction valve unloading or by opening a plug to the suction plenum

To find the steady state gas temperature $T_{spl,0\%}$ in the suction plenum when fully unloaded, we use again the principle of conservation of energy, equation 6.1, the different terms having the following values:

1. *No gas flow* through the cylinder, i.e. $M = 0$.
2. *Work dissipated* in the unloader: Suction ("intake-") stroke η_v and efficiency of delivery λ are both equal to 1 or 100% and equal to the volumetric efficiency for discharging ("outflow") back into the suction plenum. For temperature T_1 and gas density ϱ_1 in the

suction plenum (i.e. for the very first revolution after switching suddenly from full load operation to zero load), we can find $\Delta p_{m,in}$ using Φ_{in} instead of Φ_{SV} and $\Delta p_{m,out}$ using Φ_{out} instead of Φ_{SV}, both times with equation 6.2. Φ_{in} is of course the equivalent area of valve and unloader available at zero load for the flow from the suction plenum into the working chamber and Φ_{out} for the flow in the reverse direction, out of the working chamber back to the suction plenum. The corresponding work loss ΔW_1 is found from

$$\Delta W_1 = V_{st}\left(\Delta p_{m,in} + \Delta p_{m,out}\right) \tag{7.1}$$

Assuming that $\Delta p_{m,in}$ and $\Delta p_{m,out}$ are inversely proportional to the actual suction plenum temperature $T_{spl,0\%}$, we get for other suction plenum temperatures $T_{spl,0\%} \neq T_1$

$$\Delta W = \frac{T_1}{T_{spl,0\%}} \Delta W_1 \tag{7.2}$$

3. *Work input due to friction* between liner and piston rings, or rod and packings is neglected.
4. *Energy removed by heat exchange* = ΔQ.

To find ΔQ, we start by deriving a simple expression for the heat exchanged between the gas and the surrounding walls. Let α_g be the coefficient of heat transfer between a gas and a surrounding wall, A_g the area of this wall on the gas side, t its thickness, λ its thermal conductivity and A_m its mean area. Further, let α_c be the coefficient of heat transfer and A_c the wall area on the side of the coolant. The coefficient of overall heat transfer K from the fluid to the wall, through the wall and from there to a coolant is given by

$$K_i = \frac{1}{\frac{1}{A_g \alpha_g} + \frac{t}{A_m \lambda} + \frac{1}{A_c \alpha_c}} \tag{7.3}$$

Given the heat transfer coefficient $\alpha_{n,g}$ for normal conditions, α_g for other conditions can be found using one of the standard heat transfer conversion formulas. From [13] we have (for $Re > 10^5$ and $Pr > 0.6$, for gases $Pr \approx 0.7$) $Nu = \frac{\alpha_g l}{\lambda} \approx 0.05\, Re^{0.8}\, Pr^{0.4}$. Nusselt number Nu is directly proportional to α_g and a length l, Reynolds number Re to the gas density ϱ_g and the same length l. If the usually small changes of the thermal conductivity of the gas λ (contained in Nusselt number) and dynamic viscosity η (contained in Reynolds number and proportional to ϱ, for details, see [13]) of the gas with pressure and temperature as well as changes in gas velocity are neglected, a gas density change from ϱ_n at p_n, T_n, to ϱ_{spl} at p_1, T_{spl} will cause α_g to change from $\alpha_{n,g}$ to

$$\alpha_g \approx \alpha_{n,g} \left(\frac{\varrho_{spl}}{\varrho_n}\right)^{0.8} \tag{7.4}$$

Since heat transfer usually takes place along different paths towards different coolants, numbered for example

i=1 from the suction plenum towards the cooling water jacket,

i=2 from the suction plenum towards the ambient air,

i=3 from the working chamber via the cylinder liner towards the cooling water jacket,

i=4 from the working chamber via the cylinder cover towards the cooling water jacket,

we have, for the heat transferred per revolution of the crank shaft from the gas in the suction plenum with temperature $T_{spl,0\%}$ to the different coolants with temperatures $T_{c,i}$

$$\Delta Q = \frac{60}{n} \sum_{i=1}^{4} K_i \left(T_{c,i} - T_{spl,0\%}\right) \tag{7.5}$$

To treat the unknown $T_{spl,0\%}$ as an explicit variable, we rearrange as follows:

$$\Delta Q = \frac{60}{n}\left(\underbrace{\sum_{i=1}^{4} K_i T_{c,i}}_{=C1} - \frac{T_{spl,0\%}}{T_{ref}}\underbrace{\sum_{i=1}^{4} K_i T_{ref}}_{=C2}\right) = -\frac{60}{n}\left(\frac{T_{spl,0\%}}{T_{ref}}C_2 - C_1\right) \tag{7.6}$$

For abbreviation, we have introduced the two constants C_1 and C_2, depending on cylinder geometry, coolant temperatures and heat transfer coefficients, T_{ref} being an arbitrary reference temperature, for example we can choose $T_{ref} = T_1$. From 7.6 we have

$$C_1 = \sum_{i=1}^{4} K_i T_{c,i} \qquad \text{and} \qquad C_2 = \sum_{i=1}^{4} K_i T_{ref} \tag{7.7}$$

In order to enable simple algebraic solutions, we neglect the variation of α_g with ϱ_g as suggested by equation 7.4 within the temperature range from T_1 to $T_{spl,0\%}$ and consider K_i, C_1 and C_2 to be constants. If $T_{spl,0\%}$ (and hence α_g) is found to be very different from T_1, then the $T_{spl,0\%}$ of the first calculation can be used to find a better α_g from equation 7.4, thus improving K_i from equation 7.3 and finally $T_{spl,0\%}$ in a second iteration. Substituting $M = 0$ and for ΔW and ΔQ from equations 7.2 and 7.6 into equation 6.1, we get

$$\Delta W_1 \frac{T_1}{T_{spl,0\%}} = \frac{60}{n}\left(\frac{T_{spl,0\%}}{T_{ref}}C_2 - C_1\right) \tag{7.8}$$

which is a quadratic equation in $T_{spl,0\%}$. Solving for the unknown $T_{spl,0\%}$ we obtain

$$T_{spl,0\%} = T_1 \frac{C_1}{2C_2}\left(1 \pm \sqrt{1 + \frac{n}{15}\frac{\Delta W_1 C_2}{C_1^2}\frac{T_1}{T_{ref}}}\right) \tag{7.9}$$

For physically meaningful solutions, the square root must have a positive sign. Equation 7.9 can be used to find the zero load temperature in the suction plenum $T_{spl,0\%}$ from given compressor and heat exchange data, i.e. knowing ΔW_1, C_1 and C_2.

If ΔW_1 is determined for one cylinder end only, the same has to apply for the heat exchange areas used for calculating the different K_i in C_1 and C_2.

7.2 Problem inversion: How to calculate heat transfer constants from measured temperatures

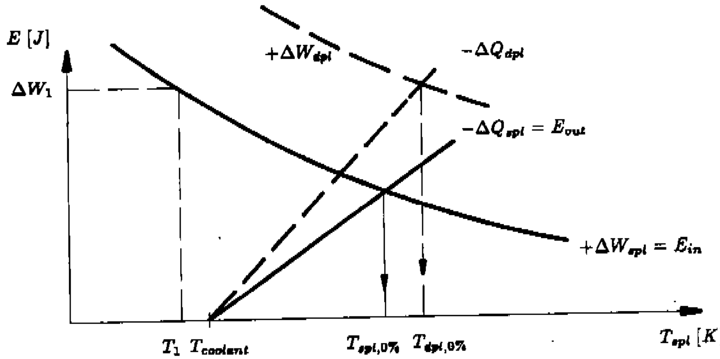
Sometimes it is tedious and not convenient to determine the overall heat transfer coefficients K_1, K_2 etc. for a given cylinder block, but instead we are given the steady state zero load temperature $T^*_{spl,0\%}$ measured on site in the suction plenum with the same cylinder block, i.e. identical heat transfer areas, wall thicknesses and materials, but fitted with another, given valve equipment and working at another, given operating condition, i.e. $T^*_{spl,0\%}$ is associated with another, given zero load work loss ΔW_1^*. Assuming all coolant temperatures T_c equal, i.e. with $T_c \approx T_{coolingwater} \approx T_{ambientair}$ equations 7.7 can be solved to find C_1

$$C_1 = \frac{T_c}{T_{ref}} C_2 \tag{7.10}$$

and, on the trivial condition that $T^*_{spl,0\%} > T_c$, i.e. that the heat is really transmitted from the gas under $T^*_{spl,0\%}$ to the coolants under T_c we obtain for C_2

$$C_2 = \Delta W_1^* \frac{n}{60} \frac{(T_1^*)^2}{T^*_{spl,0\%}\left(T^*_{spl,0\%}\frac{T_1}{T_{ref}} - T_c\right)} \tag{7.11}$$

Having found C_1 and C_2 for the cylinder block and assuming both to be constant for other operating conditions – a good assumption when only valves change – we can of course calculate



Full line curves are for the suction plenum, dotted line curves for the discharge plenum.
For identical operating conditions and cylinder geometry: $T_{spl,0\%} < T_{dpl,0\%}$

Figure 2: Energy balance in the suction plenum: Find T_{spl} for $\Delta W + \Delta Q = 0$.

ΔW_1 from equation 7.1 for any other valve equipment and operating condition and obtain an estimate for the corresponding $T_{spl,0\%}$ from equation 7.9.

7.3 Zero load by opening a plug to the discharge plenum

The method described under section 7.1 can of course be used to find the stationary zero load temperature in the discharge plenum as well. $\Delta p_{m,in}$ and $\Delta p_{m,out}$ have to be found from equation 6.2 using the equivalent flow areas Φ_{out} and Φ_{in} of the discharge valves and unloaders. Gas density ϱ_{ref} has to be taken at pressure p_2 and arbitrary reference temperature T_{ref}. It is however recommended to use $T_{2,is}$ and not T_1 since the corrections for α_g according to equation 7.4 as described in the paragraph following equation 7.7 will be smaller. The method is covered by the equations of section 9.

8 OPERATION AT HALF LOAD

8.1 Operation with an unloaded suction side

Half load of a double acting cylinder can be obtained by having one cylinder side operating at full load, the other one being unloaded. In a first approximation, each cylinder side can be looked at separately, signifying that the side operating with full load gets gas with nominal suction temperature T_1 up to its suction valves and that the unloaded side has a steady state suction plenum temperature as given by equation 7.9 above. In reality we have to assume that some of the hotter gas from the unloaded side mixes with the fresh gas from the suction line under T_1. This means that the mass of gas M taken in by the loaded end per cycle is composed of a portion XM coming from the unloaded end and having a temperature $T_{spl,0\%}$ and another portion $(1 - X)M$ coming directly from the suction line with temperature T_1, see Figure 3. Although the degree of mixing X cannot be calculated from a thermodynamic approach, $X \geq 0.4$ is a good assumption: Less is improbable (where to indeed should a volume of gas pushed out of and taken back again into the unloaded end and equal to the latter's stroke volume disappear and reappear, if not mix with and separate from the neighbouring gas stream passing through the same plenum, according to the above assumptions?), more does not or almost not change the results. In addition, it is found that the higher the value assumed for the degree of mixing X, the smaller the impact of heat exchange data on compressor performance. Since it is not easy to determine these heat exchange data with the same precision as the data of compressor, gas, valves and unloaders, we can assume that the estimates of power requirement and gas temperatures for half load operation and based on the assumption of $X \geq 0.4$ can even be considered to be more reliable than those for completely unloading a cylinder, in spite of the uncertainty in determining X.

8.2 Operation with an unloaded discharge side

The gas flows in the discharge plenum are similar to those shown in Figure 3. In this case, no extra work is put into the suction plenum whose temperatures remain unchanged, i.e. $T_{spl,50\%} = T_1$, such that *the isentropic efficiency of the loaded end is not reduced*, a considerable advantage of having the plug not on the suction but on the discharge side. However, this heating in the discharge plenum takes place to a higher degree than under identical conditions in the suction plenum, as becomes evident from the following: Reference work ΔW_2 (in analogy to ΔW_1 in equation 7.1) increases proportionally with ϱ_2/ϱ_1, while, according to equation 7.4, the heat transfer coefficient α_g increases with $(\varrho_2/\varrho_1)^{0.8}$ only. Moreover, this slower increasing α_g is a factor of only one of several terms in K_i in equation 7.3, with the other terms remaining unchanged. As a result, ΔW_1 increases faster than C_1 and C_2, and steady state zero load temperatures will be higher than what they would be in the suction plenum, with all other conditions unchanged.

9 COMBINED EQUATION FOR ZERO OR HALF LOAD OPERATION OF A DOUBLE ACTING CYLINDER UNLOADED ON ITS SUCTION OR DISCHARGE SIDE

In order to obtain general equations applicable either to suction or discharge side unloading, we are going to use the notation given in Table 1. In the case of half load operation and as already stated, heat from the unloaded end is not only removed by heat transfer, but also by the enthalpy increase of a mass of gas XM whose temperature has risen from T_{in} to $T_{x,pl,0\%}$. The nominal work loss per cycle in the unloader mechanism of the unloaded end due to gas flow and associated pressure drop at nominal gas state p_x, T_x in the plenum is given by

$$\Delta W_x = V_{st}\left(\Delta p_{m,in} + \Delta p_{m,out}\right) \tag{9.1}$$

When operating at zero load $\lambda_{0\%} \approx \eta_{v,0\%} \approx 100\% = 1$. With $\varrho_x = \frac{p_x}{RT_xZ_x}$ and $\dot{V}_1 = \frac{n}{60}V_{st}$ we have from equation 6.2

$$\Delta p_{m,in} + \Delta p_{m,out} = 3.75\,\varrho_x \frac{n^2 V_{st}^2}{3600}\left(\frac{1}{\Phi_{in}^2} + \frac{1}{\Phi_{out}^2}\right) \tag{9.2}$$

At the same plenum pressure $p_x = p_{x,pl}$ but another plenum temperature $T_{x,pl,0\%}$, the work loss becomes

$$\Delta W = \Delta W_x \frac{\varrho_x}{\varrho_{x,pl,0\%}} = \Delta W_x \frac{T_x Z_x}{T_{x,pl,0\%} Z_{x,pl,0\%}} \tag{9.3}$$

where Z_x and $Z_{x,pl,0\%}$ are usually assumed to be equal. The heat transferred to the gas per crankshaft revolution by heat exchange with the walls of the cylinder liner, cover and walls of the plenum of the unloaded end is given by

Table 1: Values to be used for p_x, T_x, etc. in equations 9.1 through 9.13

	suction side unloading	discharge side unloading	comment
p_x	p_1	p_2	nominal pressure
T_x	T_1	$T_{2,is}$	nominal temperature
$T_{x,pl,0\%}$	$T_{spl,0\%}$	$T_{dpl,0\%}$	unknown zero load plenum temperature
T_{ref}	T_1	$T_{2,is}$	arbitrary reference temperature in C_2
T_{in}	T_1	$T_{2,d}$	temperature of the incoming gas

$$\Delta Q = \frac{60}{n}\left(C_1 - \frac{T_{x,pl,0\%}}{T_{ref}} C_2\right) \tag{9.4}$$

with C_1 and C_2 according to equations 7.7.

The enthalpy removed from the unloaded cylinder end by a stream of gas XM whose temperature is thereby increased from T_{in} to $T_{x,pl,0\%}$ is given by

$$\Delta H = X\,M\,(h_{x,pl,0\%} - h_{in}) \tag{9.5}$$

Assuming ideal gas behaviour, ΔH is given by

$$\Delta H = \underbrace{XMRT_x\frac{\kappa}{\kappa-1}}_{=C_x}\left(\frac{T_{x,pl,0\%}}{T_x} - \frac{T_{in}}{T_x}\right) = C_x\left(\frac{T_{x,pl,0\%}}{T_x} - \frac{T_{in}}{T_x}\right) \tag{9.6}$$

where C_x is a constant, introduced for abbreviation and M is the mass of gas delivered per crank shaft revolution of the loaded end, according to equation 2.5, assuming T_1 in the suction plenum.

$$C_x = XMRT_x\frac{\kappa}{\kappa-1} \tag{9.7}$$

The energy balance equation 6.1 for the unloaded cylinder end takes the form

$$\Delta W + \Delta Q = \Delta H \tag{9.8}$$

Substituting for ΔW, ΔQ and ΔH from equations 9.3, 9.4 and 9.6 we get

$$\Delta W_x \frac{T_x}{T_{x,pl,0\%}} + \frac{60}{n}\left(C_1 - \frac{T_{x,pl,0\%}}{T_{ref}}\cdot\frac{T_x}{T_x}C_2\right) - C_x\left(\frac{T_{x,pl,0\%}}{T_x} - \frac{T_{in}}{T_x}\right) = 0 \tag{9.9}$$

Multiplying equation 9.9 by $\frac{T_{x,pl,0\%}}{T_x}$ and rearranging we get

$$\Delta W_x - \left(\frac{T_{x,pl,0\%}}{T_x}\right)^2\left(\underbrace{\frac{60}{n}\cdot\frac{T_x}{T_{ref}}C_2 + C_x}_{=C_{x2}}\right) + \frac{T_{x,pl,0\%}}{T_x}\left(\underbrace{\frac{60}{n}C_1 + \frac{T_{in}}{T_x}C_x}_{=C_{x1}}\right) = 0 \tag{9.10}$$

Putting

$$C_{x1} = \frac{60}{n}C_1 + \frac{T_{in}}{T_x}C_x \qquad \text{and} \qquad C_{x2} = \frac{60}{n}\cdot\frac{T_x}{T_{ref}}C_2 + C_x \tag{9.11}$$

we get from equation 9.10

$$\left(\frac{T_{x,pl,0\%}}{T_x}\right)^2 - \frac{T_{x,pl,0\%}}{T_x}\frac{C_{x1}}{C_{x2}} - \frac{\Delta W_x}{C_{x2}} = 0 \tag{9.12}$$

This is a quadratic equation in $\frac{T_{x,pl,0\%}}{T_x}$ having the solutions

$$T_{x,pl,0\%} = T_x \frac{C_{x1}}{2C_{x2}} \left(1 \pm \sqrt{1 + 4\frac{\Delta W_x C_{x2}}{C_{x1}^2}}\right) \tag{9.13}$$

Physically, only the positive sign of the square root is meaningful. For zero load without any gas stream for heat removal, i.e. when putting $X = 0$ and hence also $C_s = 0$, equations 9.13 and 7.9 become identical. Equation 9.13 can be used for half load or zero load with suction or discharge side unloading indifferently, according to the notations given in Table 1.

9.1 Application to suction side unloading

The gas flows, input and output of energy are shown in Figure 3. In this case, $T_{x,pl,0\%} = T_{spl,0\%}$ is the steady state gas temperature in the suction plenum of the unloaded end. The degree of mixing X has to be assumed, good assumptions are $X \geq 0.4$ as stated earlier. Equation 9.13 can now be evaluated. The gas temperature $T_{spl,50\%}$ upstream of the suction valves of the loaded end is found from the mixing rule

$$T_{spl,50\%} = (1 - X)\, T_1 + X\, T_{spl,0\%} \tag{9.14}$$

Thus, the real compression cycle of the loaded cylinder end does not start from temperature T_1 in the suction plenum but from the higher $T_{spl,50\%}$ and we can assume that final temperature T_{dpl} in the discharge plenum also rises almost proportionally with $T_{spl,50\%}$. If T_{dpl} is the final temperature at full load (with the cycle starting from T_1), we have, in analogy to equation 2.6 for the isentropic efficiency of the *loaded cylinder end only* at half load

$$\eta_{is,50\%} = \frac{T_{2,is} - T_1}{T_{dpl}\frac{T_{spl,50\%}}{T_1} - T_1} \tag{9.15}$$

Since $T_{spl,50\%} > T_1$, the denominator increases and $\eta_{is,50\%}$ decreases as compared to full load operation. The *total work per cycle* required by a double acting compressor cylinder operating at half load is of course the sum of

- the work requirement ΔW of the unloaded cylinder side according to equation 7.2, plus
- the work necessary for the full load compression cycle in the loaded cylinder end, which in analogy to equation 2.6 is given by

$$W_{real} = \frac{W_{is}}{\eta_{is,50\%}} \tag{9.16}$$

We finally get for the overall isentropic efficiency of a double acting cylinder operating at half load

$$\eta_{is,50\%,overall} = \frac{W_{is}}{\Delta W + W_{real}} \tag{9.17}$$

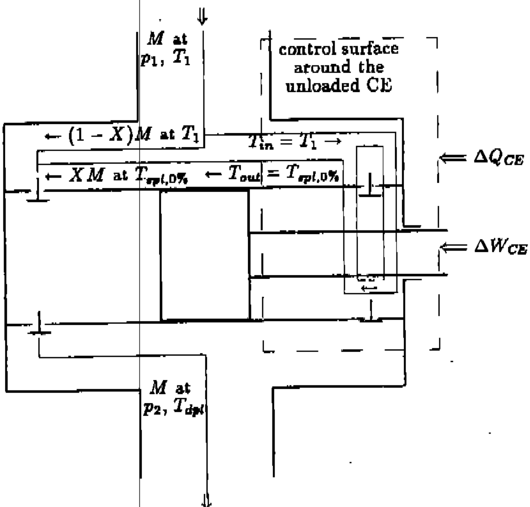
Note that, with $T_{spl,50\%} > T_1$, the mass M delivered per cycle of the loaded end will be smaller at half load in the proportion $T_1/T_{spl,50\%}$ than at full load, such that calculations shall be run a second time with an improved value for M and C_e according to equation 9.7. To find the suction plenum temperature *for complete unloading* $T_{spl,0\%}$, simply put $X = 0$.

9.2 Application to discharge side unloading

In that case, suction plenum temperature $T_{spl} = T_1$ and discharge temperature $T_{2,d}$ *of the loaded end* are those of full load operation, isentropic and volumetric efficiencies *of the loaded end remain unchanged*. To find the work loss in the unloaded end, we can determine $\Delta p_{m,in}$ and $\Delta p_{m,out}$ from equation 9.2 for isentropic discharge conditions, i.e. put $\varrho_x = \varrho_{2,is}$. The heat transfer coefficients on the gas side α_g should also be adjusted to gas density $\varrho_{2,is}$ using equation 7.4. Having found the unknown $T_{dpl,0\%} = T_{s,pl,0\%}$ from equation 9.13, the additional work $\Delta W_{0\%}$ required by the unloaded end can also be determined, which reduces overall isentropic efficiency according to equation 9.17. Half load final temperature $T_{dpl,50\%}$ follows from the mixing rule

$$T_{dpl,50\%} = (1 - X)T_{2,d} + XT_{dpl,0\%} \tag{9.18}$$

To find the discharge plenum temperature *for complete unloading* $T_{dpl,0\%}$ put $X = 0$.



Thin lines symbolize gas flows, either directly from the suction pipe to the loaded head end working chamber for the part $(1 - X)M$, or via the unloaded crank end working chamber for the remaining part XM.

Figure 3: Half load with head end loaded, crank end unloaded on its suction side.

References

[1] 1986 International Compressor Engineering Conference at Purdue, proceedings pages 210 - 227, *Energetic and volumetric characteristics for the uniform evaluation of gas and refrigeration compressors* by H.J. Kleinert and H. Najork, University of Dresden, Germany.

[2] 1988 International Compressor Engineering Conference at Purdue, proceedings pages 375 - 384, *Comprehensive model of a reciprocating compressor applicable to component desgign issues* by T. Morel and R. Keribar, Integral Technologies Incorporated, Wetsmont, Illinois.

[3] 1988 International Compressor Engineering Conference at Purdue, proceedings pages 454 - 463, *Heat transfer and component temperature prediction in reciprocating compressors* by T. Morel and R. Keribar, Integral Technologies Incorporated, Wetsmont, Illinois.

[4] 1990 International Compressor Engineering Conference at Purdue, proceedings pages 254 - 263, *Utilization of volumetric displacement in reciprocating compressors* by J.A. McGovern, Department of Mechanical and Manufacturing Engineering, Trinity College, Dublin, Ireland.

[5] 1990 International Compressor Engineering Conference at Purdue, proceedings, and HOERBIGER Engineering Report 54 (HOERBIGER publication A5V019E01, 1991), *How leakages in valves can influence the volumetric and isentropic efficiencies of reciprocating compressors* by Erich H. Machu, Vienna, Austria.

[6] HOERBIGER Engineering Report 56 (HOERBIGER publication A5V022E01, 1991) *A simplified theory on the influence of valve throttling on compressor efficiency and gas temperatures* by Erich H. Machu, Vienna, Austria.

[7] Sixth Annual Reciprocating Machinery Conference, Sept. 1991, Salt Lake City, Utah, *Compressor Cylinder Performance Analysis: The Next Generation of Software Capabilities* by Ralph E. Harris, CAMPt, San Antonio, Texas.

[8] Sixth Annual Reciprocating Machinery Conference, Sept. 1991, Salt Lake City, Utah, *Compressor Cylinder Performance Prediction - Some Recent Developments* by C. Richard Gerlach, CAMPt, San Antonio, Texas.

[9] *EGINEERING DATA BOOK* 1980 (in SI-units), Gas Processors Suppliers Association, 1812 First Place, Tulsa, Oklahoma 74103.

[10] API Standard 618, *Reciprocating Compressors for General Refinery Services*, 3rd edition, February 1986, American Petroleum Institute, 1220 L Street, Northwest, Washington, D.C. 20005.

[11] *HOERBIGER information V009E*, Hoerbiger Ventilwerke A.G., P.O. Box 91, A 1110 - Vienna, Austria, 1987.

[12] Proceedings of the PCRC-SwRI Seminar on Pulsation Control, Vienna, Austria, Oct. 1991, HOERBIGER part.

[13] VDI-Wärmeatlas (Data book on heat transfer), in German language, VDI-Verlag Düsseldorf, Germany, 1984.

List of Symbols

symbol	unit	comment
A	m^2	area
C	$J/kg\,K$	specific heat
C_s	-	suction throttling constant
C_d	-	discharge throttling constant
C_1	W	heat transfer constant
C_2	W	heat transfer constant
C_x	–	mixing constant
h	mm	valve lift
h	J/kg	specific enthalpy
H	J	enthalpy
M	kg	mass of gas
$\tilde{M}$	$kg/kmol$	molar mass of gas
n	$r.p.m.$	rotational speed of crankshaft
Nu	–	Nusselt number
p	Pa	absolute pressure
P	W	power
q	–	valve ampleness coefficient
Q	J	heat transferred
R	$J/kg\,K$	gas constant
Re	–	Reynolds number
s	%/100	clearance volume ratio
T	K	absolute temperature
u	m/s	velocity
u	$J/kg\,K$	specific internal energy
V	m^3	(stroke) volume
W	J	mechanical work
X	%/100	degree of mixing
z	%/100	relative piston displacement = *piston travel / piston stroke*
Z	–	real gas compressibility factor
α	$W/m^2\,K$	heat transfer coefficient
Δ	–	small increment
ε	–	expansion ratio = $p_{downstream}/p_{upstream}$
η	%/100	(volumetric) efficiency = length of suction stroke
η	$Pa\,s$	dynamic viscosity
θ	rad	crank angle
κ	–	ideal gas isentropic exponent = C_p/C_v
λ	%/100	efficiency of delivery
λ	$W/m\,K$	thermal conductivity
ν	%/100	length of effective discharge stroke
ν	m^2/s	kinematic viscosity = η/ϱ
Φ	m^2	equivalent flow area
ϱ	kg/m^3	density
φ	–	correction factor
Ψ	–	expansion function
ω	rad/s	angular crankshaft velocity

List of Subscripts

subscript	comment
1	suction, or just a counter
2	discharge, or just a counter
0%	zero load
50%	half load
100%	full load
*	not yet accounting for temperature changes
c	compression
c	coolant
cyl	inside a working chamber ("cylinder")
d	discharge
dpl	discharge plenum
DV	discharge valve
ex	expansion
g	gas side
gap	referring to the valve gap area
i	indicated, referring to the full stroke volume
is	isentropic
l	leakage
m	mean, referring to stroke volume × η_v
n	normal condition, at $p_n = 10^5$ [Pa], $T_n = 273.15$ [K]
p	at constant pressure
ref	state of reference
s	suction
spl	suction plenum
st	stroke (volume)
SV	suction valve
v	volumetric, or at constant volume