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Energetic and Volumetric Characteristics for the Uniform Valuation of Gas and Refrigeration Compressors

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ENERGETIC AND VOLUMETRIC CHARACTERISTICS FOR THE UNIFORM VALUATION OF GAS AND REFRIGERATION COMPRESSORS

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ABSTRACT

The uniform valuation of refrigeration and gas compressors is a basic supposition for the purposeful further development. The historically originated energetic and volumetric characteristics are not uniform despite of many efforts and in parts not clear in the signification. An analysis has been accomplished at the basis of the international literature as well as of rules and standards. A proposition has been explained under the consideration of the constructions enclosed to the motor with the object to unify the characteristics.

1. INTRODUCTION

Studying special literature and in cooperation with design and test engineers,in the teaching process as well as working as authors and editors of relevant special books at the same time we found out that is necessary to consider such fundamental questions of this subject which are assumed to have been solved for a long time. The uniform valuation of machines is a basic supposition for our engineers work,but it is not given in general for gas and refrigeration compressors.

First of all we want to consider the design and the development work and their valuation. For that reason great importance is due to the ideal machine as an orientation to the object aspired to. Often quite general principles are lost out of the field of view and that isn't well. The isothermic power input P_T and the isentropic power input P_S are the comparative power inputs in the compressor engineering. Furthermore the actual displacement volume rate $\dot{V}_D$ which can be defined from the theoretical displacement volume rate V_H or the recirculation volume rate V_U in rotary compressors respectively with the volumetric efficiency λ_h is decisive for that quantities.

The analysis of losses opposite the ideal machine must deal with the

- pressure losses,
- wall losses (heating),
- leakages,
- effects of the clearance volume.

Many of the represented here isn't new and however shall serve the information and orientation of the research work.

2. HISTORICAL DEVELOPMENT OF THE CHARACTERISTICS FOR THE VALUATION OF THE COMPRESSORS

After the beginning of the refrigeration compressors with ethyl ether (John Perkins 1834) ammonia had been used succesfuelly in absorption refrigeration machines (Ferdinand Carré 1860) especially by Carl von Linde from 1870 until 1900,and with the development of powerful NH_3 piston compressors it has been become the predominant refrigerant also in refrigeration compressors. As results from /1/ the calculation of required compressor piston displacement took place on the basis of the indicator diagram already considering the required increase of piston displacement because of reexpansion of the vapour in the clearance volume, pressure losses,heating and leakage. In /2/ a volumetric efficiency has been defined as the ratio of the effective to the total piston displacement,but it has been dismissed for a calculation because the definition is difficult by reason of liquid particles evaporating only in the piston displacement. The volumetric efficiency has been stated to be useful for the calculation as a ratio of the weight really sucked to the weight theoretically possible at dry-saturated suction.

$$\lambda = \frac{\dot{m}_S}{n \cdot V_H / v_o''} \qquad (1)$$

The piston displacement could be calculated from the required refrigerating capacity $\dot{Q}_0$ with fixed specific refrigerating capacity q_0 corresponding the cycle selected

$$V_H = \frac{\dot{Q}_0}{\lambda \cdot n \cdot q_0 / v_o''} \qquad (2)$$

By it the volumetric refrigerating capacity

$$q_{ov} = q_c / v_o'' \tag{3}$$

has been calculated for dry-saturated suction at different process conditions and different refrigerants and later given in tables /3/.

For the predominant refrigerant ammonia the volumetric efficiency had been experimentally investigated by Fischer /4/. Linge /5/ gave a new definition of the volumetric efficiency as the volume ratio of the really sucked volume flow rate referred to dry-saturated suction to the theoretical displacement volume rate

$$\lambda = \dot{V}_S / \dot{V}_H \tag{4}$$

how it has also been received by R.Plank in Handbuch der Kältetechnik /6/ and in Kältemaschinenregeln /3/. Using the experimental results and the statement

$$\lambda = \eta_v - (1 - \eta_w) \tag{5}$$

of Fischer /4/ the DKV-Arbeitsblatt 3-01 /8/ has been developed, whereby in $(1 - \eta_w)$ - the wall losses - the losses by heating and leakage are comprehended and

$$\eta_v = (\varepsilon_0 + 1)\left(1 - \frac{\Delta p_o}{p_o}\right)^{1/m} - \left(\frac{p_K + \Delta p_K}{p_o}\right)^{1/n} \tag{6}$$

is calculated. This worksheet has been used until today for NH_3 compressors.

For refrigeration compressors with halogen refrigerants until now no exist data of general application for calculation of the volumetric efficiency. Only for open plunger compressors with R22 new diagrams for the definition of λ have been proposed by Heimbach /9/ also for NH_3. The effect of the clearance volume was calculated in the most simple way neglecting the pressure losses in the self-acting valves and assuming isentropic reexpansion as a quantity denoted "volumetric efficiency"

$$\eta_v = 1 - \varepsilon_o(\pi^{1/\kappa} - 1) \tag{7}$$

The other influencing factors was summarized in the characteristic K_1 so that follows for the volumetric efficiency

$$\lambda = K_1 \cdot \eta_v \tag{8}$$

The characteristic K_1 was found out experimentally and has been shown in diagrams depending on the geometrical swept volume rate of a working cylinder $\dot{V}_Z$ with the parameter compression ratio. The volumetric efficiencies defined by /8/ for NH_3 are 3% lower then the calculated by /9/. Calculating the compressor power input is gone out from the indicator diagram of the ideal machine /1/,/2/ and the indicated efficiencies defined experimentally by Fischer /4/ are used. In this case the indicated efficiency of the compressor is defined as the ratio of the refrigerating capacity achived with the indicated power input to the refrigerating capacity att inable with the isentropic compression:

$$\eta_i = \frac{(\dot{Q}_0/P_i)_{exp}}{(\dot{Q}_0/P_s)_{th}} \tag{9}$$

The ratio $\dot{Q}_0/P_s$ is denoted by Hirsch /2/ as economic C.O.P. and shown in diagrams for different refrigerants at different subcoolings. Later Linge /6/ denoted this ratio as specific refrigerating capacity

$$K_{th} = \frac{\dot{Q}_0}{P_s} = 860\ \frac{h_1 - h_3}{h_2 - h_1} \quad in\ \frac{kcal}{kWh} \tag{10}$$

what has been undertaken in the Kältemaschinenregeln /3/. The process with dry-saturated suction and subcooling has been underlain as the theoretical process of comparing, see figure 1. After the introduction of SI-units the specific refrigerating capacity was received as a non-dimensional C.O.P.

$$\varepsilon_{th} = \frac{\dot{Q}_0}{P_s} \tag{11}$$

Linge /5/ defines the indicated efficiency of the refrigeration compressor of reciprocating type as the ratio of the isentropic compression work to the indicated compression work or as the ratio of the adequate average indicated pressures respectively

$$\eta_i = \frac{W_s}{W_i} = \frac{p_{ms}}{p_{mi}} \tag{12}$$

and calculates it for NH_3 using simplifying assumptions for the pressure losses in the self-acting valves ($\Delta p = p/20$) and average polytropic exponents $n_c = 1.25$ and $n_r = 1.15$.

In the DKV-Arbeitsblatt 3-01 /8/ at this basis in addition to the volumetric efficiency also the indicated efficiency can be read out. For refrigeration compressors with halogen refrigerants also data of general application for calculation of the indicated efficiency not exist. New diagrams for the determination of the power input have been published by Heimbach only for open plunger compressors with R22 and NH_3 on the basis of experimental investigations. Supposing that in this case the same relationship exists for the actual power input as for the isentropic power input of the compressor calculated with the equation

$$P_S = \dot{V}_S \cdot p_S \frac{\kappa}{\kappa - 1}\left[\left(\frac{p_D}{p_S}\right)^{\frac{\kappa-1}{\kappa}} - 1\right] \quad (13)$$

the characteristic

$$K_Z = \frac{P_K}{p_S \cdot \dot{V}_Z} \quad (14)$$

is determined according to that from experiments. The characteristic K is shown in diagrams for R22 and NH_3 depending on the compression ratio p_D/p_S and the geometrical swept volume rate of a working cylinder.

Valuating air and gas compressors the real flow rate of the discharge is considered,and the development turned out differently as to denomination and definition /20 to 25/. A partition of the volumetric characteristics according to the particular influences is widely common used,where also different definitions are existing yet. For the utilization of the piston displacement of gas compressors in German the designation "Ausnutzungsgrad" (utilization efficiency) is preferred instead of "Liefergrad" (volumetric efficiency) and all separate influences effected by clearance volume,underpressure at the beginning of the compression,heating and leakage are discribed by these factors:

$$\lambda_h = \lambda_V \cdot \lambda_p \cdot \lambda_T \cdot \lambda_d \quad (15)$$

Only in the sovietic literature /10/ refrigeration compressors are delt with in this way,and the unique separation of the compressor characteristics from that of the cycle is taken. In Kältemaschinenregeln /7/ and other recent publications /11/,/18/ furthermore the C.O.P. is used for the energetic valuation of refrigeration compressors.

In parallel to these characteristics which carry out a comparision with the ideal machine,in the last 20 years especially for refrigeration compressors characteristics have been developed allowing a global valuation concerning the value in use with the object to compare different products of different producers and also to compare different compressor designs for the same range of application,as for instance reciprocating and screw compressors. Whereas up to the 60th main attention was paid to reduce the weight and the volume /13/ the aspect of energy consumption was even more underlined from the 70th /14/,/15/,/16/.

3. CHARACTERISTICS FOR COMPARISION WITH THE IDEAL MACHINE

For design and development work and the valuation of that a high importance is due to the comparision with the ideal machine as an information for the object aspired to. The operation in piston compressors for technical gases and for refrigerants is the same and the losses accuring are from the same type, consequently the problems of the research and development in this field too. In the mind of the mutual understanding and the transferability of research results it is useful to employ uniform characteristics for gas and refrigeration compressors.
It is necessary to regard the refrigeration compressors as an open thermodynamic system /17/ as happens at the calculation of refrigeration compressors by means of computer programs /18/.
A valuation of the refrigeration compressor by means of C.O.P.

$$\varepsilon = \frac{\dot{Q}_0}{P} = \frac{\dot{m} \cdot q_0}{\dot{m}\,(e_{q_0} + \Delta e_{12} + e_{23} + \Delta e_{34} + \Delta e_{41})} \qquad (16)$$

enclosing the entire refrigeration cycle consequently containing in addition to energetic losses in the compressor Δe_{12} also the exergetic losses of the other parts of the equipment as results from figure 2 /19/ is not very appropriate for the refrigeration compressor. The calculation of the compressor power input from the specific work of the refrigeration machine cycle depictured in manuels of the refrigeration engineering many times is only correct supposing the isentropic or adiabatic-irreversible compression what comes not true for the real compression process. At coupled with electromotors hermetical or semihermetical refrigeration compressors the heating of the refrigerant vapour by heat losses of the electromotor adds to this.

In the following uniform characteristics for the valuation of compressors in comparision to the ideal machine are proposed.

3.1. Utilization of the Displacement

The volume flow rate $\dot{V}_D$ pressed into the discharge line is lower than the theoretic displacement flow rate $\dot{V}_H$. The reasons for that decrease result from the work cycle ahead the indicator diagram after figure 3 and the mass balance after figure 4. For the determination of the piston displacement or recirculation volume the degrease of displacement flow rate must be predeterminated sufficiently precise.

The utilization efficiency also designated as real volumetric efficiency is the ratio of the volume flow rates or mass flow rates and considers the total decrease of displacement flow rate of the measured at discharge flow rate $\dot{V}_D$ in comparision to the theoretical displacement flow rate $\dot{V}_H$:

$$\lambda_h = \frac{\dot{m}_D}{\varrho_S \cdot V_H \cdot n} = \frac{\dot{V}_D}{\dot{V}_H} \qquad (17)$$

A predetermination is only possible over the analysis of the separate influences,so that an ingenious partition is suitable.

Separate Influences to the Degrease of the Displacement Flow Rate

The different reasons are appropriatly taken in consideration by a multicativ statement:

$$\lambda_h = \lambda_V \cdot \lambda_p \cdot \lambda_T \cdot \lambda_d \qquad (15)$$

The clearance volumetric efficiency λ_V takes in consideration the decrease of the volume used for the suction V_i in comparision to the piston displacement V_H by reexpansion of the residual mass remained in the clearance volume (figure 3). It receives with the relative clearance volume ε_0 and the compression ratio π the equation (see figure 5):

$$\lambda_V = 1 - \varepsilon_0 \left(\pi^{1/n_r} - 1 \right) \qquad (18)$$

For diatomic gases is the average polytropic exponent of the reexpansion $n_r = 1.2 \ldots 1.4$. For the compensation of uncertainties of the predetermination the internal compression ratio π_i of the stage may be used instead the real compression ratio π.
The pressure volumetric efficiency λ_p takes in consideration the decrease of the sucked mass by underpressure in comparision to the suction pressure p at the end of suction. It follows from equation (19) as

$$\lambda_p = 1 - \frac{1 + \varepsilon_0}{n_e \cdot \lambda_V} \cdot \frac{p_S - p_1}{p_S} \qquad (19)$$

Assuming the compression from p_1 to p_S to be isothermic the approximation $\lambda_p \approx p_1/p_S$ takes place.
The pressure p_1 is determinated by the pressure variations in the suction line or in the intermediate stage system on the suction side respectively and therefore difficultly predeterminable. At the design is assumed $\lambda_p = 0.95 \ldots 0.98$. In convenient cases $\lambda_p > 1$ can occur. Due to the factors λ_V and λ_p can be defined from the indicator diagram they are sumarized under the term indicated volumetric efficiency

$$\lambda_i = \lambda_V \cdot \lambda_p = V_i / V_h \qquad (20)$$

The thermometric efficiency λ_T expresses the de rease of the sucked mass by heating the suction medium at suction. It submits from the ratio of temperatures:

$$\lambda_T = T_S / T_1 \qquad (21)$$

The heating decreases with increasing compression ratio and isentropic exponent. Its direct determination from the ratio ot temperature causes metrologic difficulties. At the indirect determination from the displacement flow rate it can be only difficultly separated from the denseness efficiency. Figure 6 shows the great range of variance of the task where different definitions takes a further part. The thermometric efficiency becomes especially low at suction gas cooled compressors if the heating for cooling purposes isn't separatly shown.

The denseness efficiency λ_d is the ratio of the displaced mass flow rate $\dot{m}_D$ to the internal mass flow rate $\dot{m}_i$ which must be through-feeded in the operating space. This definition doesn't become commonplace in the special literature and therefore leads to complications of the separate influences.
It can be divided in an internal and an external denseness efficiency which compare the mass flow rate displaced with the sucked or the sucked with the internal respectively

$$\lambda_d = \frac{\dot{m}_D}{\dot{m}_i} = \frac{\dot{m}_i - \dot{m}_L}{\dot{m}_i} = \frac{\dot{m}_D}{\dot{m}_S} \cdot \frac{\dot{m}_S}{\dot{m}_i} = \lambda_{da} \cdot \lambda_{di} \qquad (22)$$

In figure 4 the partial fluxes are shown in the mass balance and also the influence of moisture is contained. Especially at multistage compressors but also generally it is worth to obtain a clear idea about the movement of the leakage mass fluxes.

The calculation of the leakage mass fluxes is difficult due to the changing by time boundary conditions and inaccurate due to the uncertainty of the dimension of the clearance. That's why the using of round empirical data is ingenious.
The external denseness efficiency for reciprocating compressors can assumed to be $\lambda_{da} = 1 \ldots 0.995 \ldots \ldots 0.95$ while the internal denseness efficiency is $\lambda_{di} = 0.98 \ldots 0.85$. The lowest values for λ_{da} occur at open plunger machines and for λ_{di} at dry piston compressors. The denseness efficiency increases if the rotational speed increases because of remaining nearly constant of the leakage mass flow rate and decreases if the working time increases in consequence

of the wear of the sealing elements.
With special designs without relatively moved external sealings (hermetic or membrane design) for high-quality or dangerous suction mediums and at refrigeration compressors external leakage can be avoided. The mass losses that have to be appreciated as external leakage flow rates, for instance at plunger machines then must be appreciated as internal leakage flow rates.

Especially high are the internal leakages at recirculating piston compressors with contact-free sealings (oil free screw compressors, Roots-blower) assuming thorough seal toward outside which is true for refrigeration compressors ($\dot{V}_S = \dot{V}_D$). With $\lambda_{dd} = 1$ consequently results the identity of the utilization efficiency according to equation (16) and the volumetric efficiency according to equation (4) introduced into the refrigeration and traditionally used there. With that an uniform definition of the utilization efficiency and the volumetric efficiency for gas and refrigeration compressors is given which enables also an uniform using of the separate factors according to equations (16) to (22).

The gas sucked by the first stage of the gas compressor is often moist, that means, it contains water vapour. At compression the partial pressure of the vapour increases and at following recooling the dew-point temperature is fallen below and the excess moisture is condensated out.
Adding water vapour the volume of a dry gas $V_{S,tr}$ increases at constant suction condition p_S, T_S to

$$V_{S,f} = V_{S,tr} \frac{p_S}{p_S - \varphi \cdot p_t} \qquad (23)$$

where: φ = relative humidity,
p_t = saturation pressure of water vapour at T_S.

The humidity efficiency is defined by

$$\lambda_f = \frac{V_{S,tr}}{V_{S,f}} = \frac{p_S - \varphi \cdot p_t}{p_S} \qquad (24)$$

For the real volume flow rate after the deposition of the condensat it submits

$$\dot{V}_{D,tr} = \dot{V}_H \cdot \lambda_h \cdot \lambda_f \qquad (25)$$

Compressing gas mixtures with easily condensable components (for instance hydrocarbons) must be delt with a balance of quantity as at humidity.
Compressing halogen refrigerants a part of it can solve in the oil with increasing pressure despite increasing temperature depending on the kind of oil.

A part of the oil is pressed into the discharge line and if it isn't separated at the oil separator it runs throw condensor and evaporator and returns into the compressor. The other part of the oil remains at the cylinder walls where an evaporation of the refrigerant can happen when the pressure falls. These processes are widely vague but of high importance at screw and rolling piston compressors with big amounts of oil.

3.2. Energetic Valuation

The uniform basis for the valuation of compressors of differnt constructions and numbers of stages is the isothermic compressor power input of the ideal machine

$$P_T = w_{tT} \cdot \dot{m} = [h_2 - h_1 - T(s_2 - s_1)]\dot{m} = n \cdot p_1 \cdot V_1 \cdot \ln(p_2/p_1) \qquad (26)$$

For the valuation of one stage compressors the isentropic compressor power input is also suitable.

$$P_S = w_{tS} \cdot \dot{m} = (h_2 - h_1)\dot{m} = n \frac{\varkappa}{\varkappa - 1} p_1 \cdot V_1 \left[(p_2/p_1)^{\frac{\varkappa-1}{\varkappa}} - 1\right] \qquad (26a)$$

Using the ideal machine as standard of comparison the inputs must be calculated with the real displacement flow rate and the nominal compression ratio at suction conditions. The calculatio of the isentropic comparison power input over the compression ratio with isentropic exponent is strongly taken only valid for one-atomic gases and must be carried out considering the behaviour of real gas with an average real isentropic exponent. That's why for refrigeration compressors power input is calculated with the enthalpy difference taken from the diagrams.

The shaft power input P_K of the compressor submits as the sum of the internal power input of all stages P_i and of the mechanical power losses of the compressor P_m

$$P_K = P_i + P_m \qquad (27)$$

The internal power input P_i as the work transmitted by the piston to the suction medium in a time unit is for the stage of the reciprocating compressor

$$P_i = n \cdot V_H \cdot p_{mi} \qquad (28)$$

where the average piston pressure p_{mi} must be taken from the measured or predeterminated indicator diagram. The predetermination of the p,V-diagram is approximately possible using values of experience of the single losses (also subroutines, for instance restrictional losses).

Supposing a constant polytropic exponent the average piston pressure becomes

$$p_{mi} = \frac{n}{n-1} p_S' \left[\pi_i^{\frac{n-1}{n}} - 1\right] \qquad (29)$$

whereby in the internal compression ratio π_i the pressure losses in the suction valve and in the discharge valve (π_S, π_D) and in the system of intermediate stages are also sized π_Z.

$$\pi_i = \pi \cdot \pi_S \cdot \pi_D \cdot \pi_Z = \frac{p_{Sj+1}}{p_S} \cdot \frac{p_S}{p_S'} \cdot \frac{p_D'}{p_D} \cdot \frac{p_D}{p_{Dj+1}} \qquad (30)$$

For the valuation of the compressors in comparision with the ideal machine the indicated efficiency η_i or the effective efficiency η_K respectively are used.

isothermic indicated efficiency $\eta_{iT} = P_T / P_i$ (31)

isentropic indicated efficiency $\eta_{iS} = P_S / P_i$ (32)

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isothermic effective efficiency $\eta_{KT} = P_T / P_K$ (33)

isentropic effective efficiency $\eta_{KS} = P_S / P_K$ (34)

The attainable efficiencies depend on the gas type, the range of pressure, the rotational speed and the construction of the compressors.

For similar compressors and by help of values of experience a determination of the input power is possible with η_i or η_K respectively.
Only the isentropic compressor capacity of the ideal machine is principially used for comparision at refrigeration compressors because the temperature at the compressor input in the refrigeration cycle without regeneration by an internal heat exchanger lies lower then the condensation temperature and therefore in addition to the increase of pressure also an increase of temperature to the condensation temperature is necessary, see figure 7. Only with ideal regeneration that means applying an internal heat exchanger a heating of the refrigerant vapour outside the compressor to the condensation temperature is possible and with it also a comparision with the isothermic compression, figure 8, what however isn't usual for vapour compression systems but only for air compression systems as for instance the Philips-refrigeration machine.

3.3. Specialities at Hermetic and Semihermetic Refrigeration Compressors

In addition to the so called open type (with rotary seal) the capsulated compressors have been developed at which the compressor and the electromotor are arranged in a common pressure-tight casing with the object to save losses of refrigerant outward by leakages. By this it is distinguished according to the kind of the encapsulation between the hermetic (pressure-tightly weldet capsule) and the semihermetic (pressure-tightly bolted casing) construction and according to the kind of motor cooling between suction gas cooled, discharge gas cooled, particularly spray oil cooled and external cooled types. From that the variants summarized in table 1 follow.

table 1

kind of the encapsulation	kind of the motor cooling suction gas cooled	discharge gas cooled	particularly spray oil cooled	external cooled
hermetic	x	x	x	-
semihermetic	x	x	-	x

Because the rotor of the electromotor is arranged direct on the stub shaft of the crankshaft of the compressor at the encapsulated construction the power input transmitted by the shaft from the motor to the compressor is difficult to determine. That's why the motor output is measured as the electric power input and the electric isentropic efficiency is determined

$$\eta_{es} = P_s / P_e \qquad (35)$$

The effective efficiency becomes with the motor efficiency η_M

$$\eta_{KS} = \eta_{es} / \eta_M \qquad (36)$$

The volumetric efficiency of the encapsulated refrigeration compressor must be completed with the motor thermometric efficiency by the loss heat of the motor λ_{TM} at suction gas and particularly spray oil cooled constructions.

$$\lambda_h = \lambda_V \cdot \lambda_p \cdot \lambda_T \cdot \lambda_d \cdot \lambda_{TM} \qquad (37)$$

Often also at open refrigeration compressors only the electric input of the motor is determined and in the same way is delt with .

4. CHARACTERISTICS FOR THE TECHNICAL COMPARISION OF DIFFERENT CONSTRUCTIONS AND TYPES

For the producer of compact compressor installations as well as for the project engineer it is necessary to choose the most appropriate construction

an the best suitable typ of a compressor from the offer of compressors to provide an installation as optimal as possible for the purpose of use. For that characteristics are needed as simple,uniform and global as possible.
Characteristics characterizing the power consumption, the volume or the floor space requirement respectively, the sound emission,the reliability and the consumption of auxilaries are of fundamental importance.

From the producers are shown in the prospectuses and the offers of refrigeration compressors the following data:

- main dimensions: stroke,bore,number of cylinders, rotational speed,external dimensions,weight
- nominal refrigerating capacity referred to a determined theoretical rating refrigeration cycle which is often not uniformly defined
- diagram or tables of the refrigerating capacity respectively with the figure of the refrigerating capacity depending on the evaporation and condensation temperature for determined refrigerants and subcooling and superheating.

As already said in 3. about the C.O.P. it is regarded to be unsuitable to state a refrigerating capacity as a characteristic of a compressor because the refrigeration machine cycle is decisive for that and not only the compressor. Refrigerating capacity and C.O.P. are certainly of great interest for the user choosing the size of the compressor or compressors consequently aren't suitable for the technical comparision of different types and constructions.

Characterizing the size is proposed to take the suction volume flow rate $\dot{V}_S$ measured at the suction side as a basis. Defining the suction state and the compression ratio is proposed to give in parentheses in the succession refrigerant,evaporation temperature, condensation temperature and suction superheat temperature:

$$\dot{V}_S\,(R;\, t_0;\, t_K;\, t_{0h}) \qquad (38)$$

The energetic valuation of refrigeration compressors for purposes of comparision should appropriatly take place with the efficiencies directly referred to the isentropic compression,at hermetic and semihermetic refrigeration compressors with the electric isentropic efficiency referred to the electric input

$$\eta_{es} = P_s / P_e \tag{39}$$

and at open refrigeration compressors with the isentropic effective efficiency

$$\eta_{Ks} = P_s / P_K \tag{39a}$$

For the definition of the reference conditions is proposed to give in parentheses in the same way as at $\dot{V}_s$ in the succession refrigerant, evaporation temperature, condensation temperature and suction superheat temperature:

$$\eta_{es} (R; t_o; t_K; t_{oh}) = \frac{P_s}{P_e} \tag{40}$$

$$\eta_{Ks} (R; t_o; t_K; t_{oh}) = \frac{P_s}{P_K} \tag{41}$$

For t_o, t_K and t_{oh} should be used standarized quantities if possible, as for instance for the GDR determined in TGL 13695.

Further

the volume $V_B = L \cdot B \cdot H$ (42)

the floor space requirement $A_B = L \cdot B$ (43)

are decisive for the technical comparision for the purpose of sorting the compressor into a compact installation, or the specific characteristics referred to the displacement flow rate give a statement to the compactness:

the specific volume $v_B = \frac{L \cdot B \cdot H}{\dot{V}_s}$ (44)

the specific floor space requirement $a_B = \frac{L \cdot B}{\dot{V}_s}$ (45)

The weight of construction M_B is of essential importance especially for transport refrigeration installations and characterizes the material expenditure as a specific characteristic referred to the suction volume flow rate and the production costs under the supposition of the production expenditure proportional to the weight and should be stated as a specific weight of construction:

$$m_B = \frac{M_B}{\dot{V}_s} \tag{46}$$

With regard to the demands of environment protection the sound intensity level in dB(A) is proposed to be stated analogous to the conditions of comparision mentioned above.

Further the reliability and the durability are essential for the quality of a product. However it would be difficult to define these with regard to their reproducible measurability because besides the influences of the operating conditions also oil renewal,attendance and so on are of importance. It is recommended to state an average operating time until the general overhaul or until the interchange of the unit respectively.

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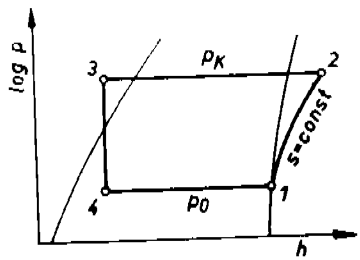


Fig. 1
Theoretical cycle with suction of dry-saturated vapour and subcooled liquid

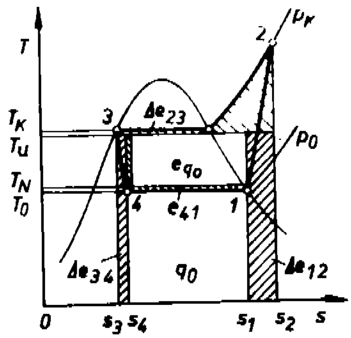
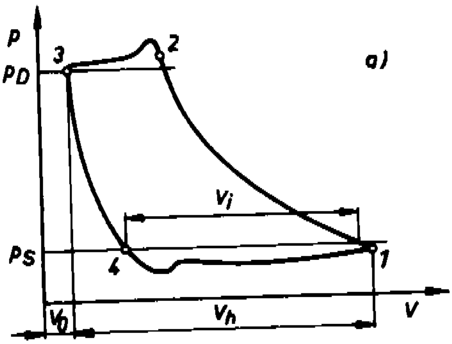


Fig. 2
Exergetic losses of the compression refrigeration machine with adiabatic-irreversible compression



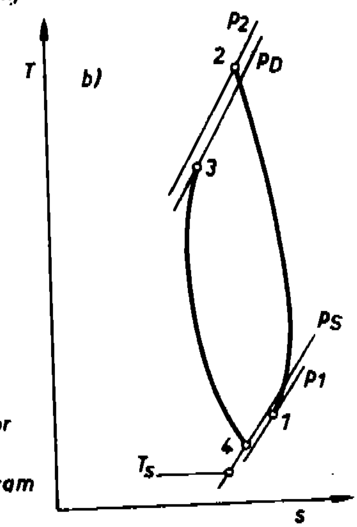


Fig. 3 Cycle of reciprocating compressor
a) Indicatordiagram
b) Compressor cycle in T,s-diagram

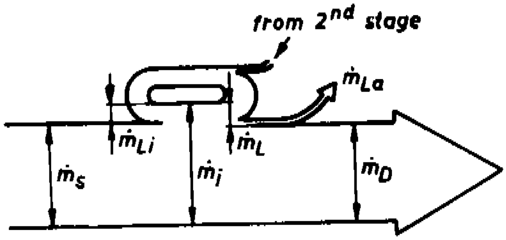


Fig. 4 Mass flow rate of compressor cycle

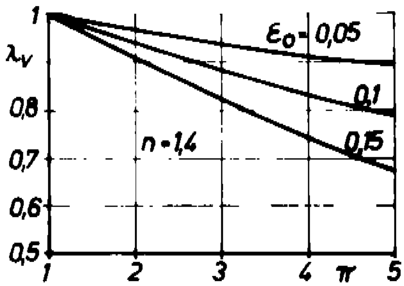


Fig. 5 Clearance volumetric efficiency λ_V

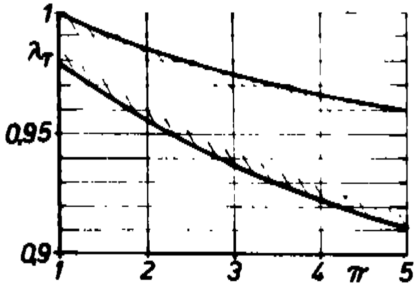


Fig. 6 Thermometric efficiency λ_T

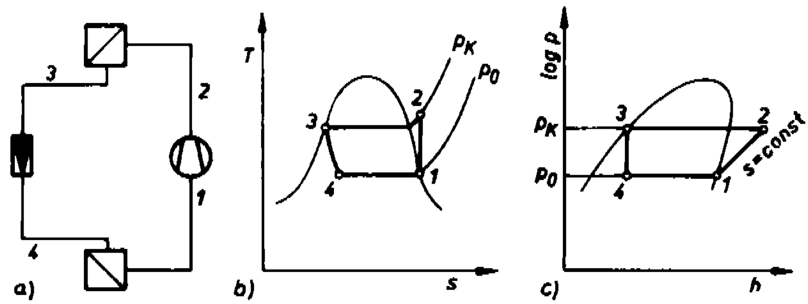


Fig. 7 Compression refrigeration machine with isentropic compression
a) System b) Cycle in T,s - diagram
c) Cycle in log p,h - diagram

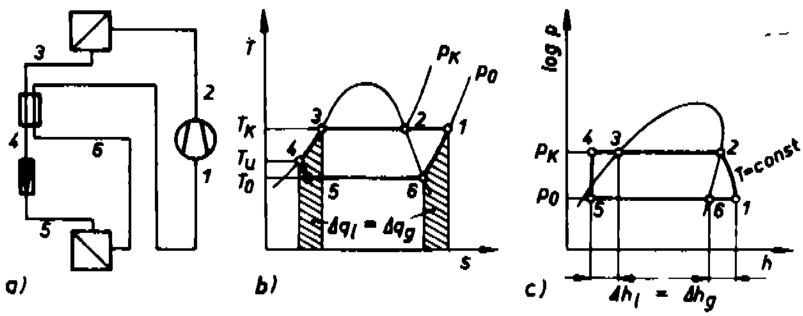


Fig. 8 Compression refrigeration machine with regeneration and isothermal compression a) System
b) Cycle in T,s - diagram c) Cycle in log p,h - diagram

PERFORMANCE EVALUATION OF HERMETIC REFRIGERATION COMPRESSOR USING TORQUE MEASUREMENT METHOD

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ABSTRACT

The measurements were performed concerning the shaft torque in the rolling piston-type rotary compressor. In order to measure the shaft torque, the torque sensor technique was proposed. In the present technique, the electromagnetic torque sensor was used and was connected between the motor and the compressor. The validity of the present technique was verified in comparison with the load torque calculated based on the measured data and that predicted using the previous study. It was found that the calculated load torque was good agreement with the predicted one. In addition, the friction torque was evaluated with the torsional two degree of freedom model.

INTRODUCTION

In response to the need for the variation of heating or cooling capacity with indoor and outdoor air temperature conditions, a heat pump air conditioner having variable speed compressor and fans has been developed for consumer use in Japan. Presently, major concerns with such a heat pump air conditioner are turned to the improvement in seasonal performance to satisfy the energy savings and also compactness of the air conditioner.

Concerning related problems on the rolling piston-type rotary compressor used in the heat pump air conditioner, a few researches on the evaluations of indicated gas power and friction losses operated at various rotating speeds have been undertaken. Sakurai, et al. (1),(2) examined mechanical and indicated efficiencies for various rotating