Heat-exchange in liquid-injected screw-compressors

J.-G. Persson, M. Sc., Stockholm (S)

Zusammenfassung

Bei öl- oder wassereingespritzten Luft-Schraubenkompressoren wurde immer wieder gefordert, einen höheren Grad an isothermer Kompression zu erreichen und dadurch die spezifische Leistungsaufnahme beträchtlich zu senken. Darum wurde ein vereinfachtes thermomechanisches Modell zur Untersuchung der Einspritzkühlung entwickelt. Dabei wurde vorausgesetzt, die eingespritzte Flüssigkeit werde zu einem einheitlichen Sprühnebel zerstäubt. Bei der Öleinspritzung wurde der zwischen Luft und Öl stattfindene Wärmeaustausch untersucht. Im Falle der Wasserinjektion konzentrierte sich die Untersuchung auf den Transport von Wärme und Masse, wie er bei der Kühlung durch Verdunstung stattfindet. Dabei wurde jeweils die aufgrund der Kühlung mögliche Minderung der spezifischen Leistungsaufnahme mit dem für die Zerstäubungsarbeit nötigen Energieverbrauch verglichen. Diese Untersuchungen zeigten: Einerseits erfolgt die Verdichtung immer nur über sehr kurze Zeit. Andererseits ist die Kontaktfläche zwischen eingespritzter Flüssigkeit und Luft begrenzt, und der zur Erhöhung dieser Kontaktfläche erforderliche Energiebedarf für eine Feinzerstäubung in äusserst kleine Tröpfchen gleicht das Einsparungsziel derart wieder aus, dass in der Praxis nur wenige Verminderung der spezifischen Leistungsaufnahme erreicht werden kann.

Abstract

A significant potential for reduction of specific power consumption in oil- or water-flooded air compressors of screw type, due to virtually isothermal compression, has often been claimed. A simplified thermodynamic model for injection cooling has been developed. The injected liquid has been considered atomized into a uniform spray. For oil

injection, transient heat transfer air - oil has been studied. For water injection, transient heat and mass transfer, i.e. evaporation cooling, has been studied. Specific power reduction due to cooling has been compared to power consumption for liquid atomization. Due to the very short time of compression, as well as the balance between a limited contact area liquid - air, or an increased power consumption for atomization into very small droplets, only a minor reduction in specific power can be obtained in practice.

Symbols / Symbole

A	m^2	liquid surface area; Flüssigkeitfläche
c	kg/m^3	vapour concentration; Dampfkonzentration
c_p, c_v, c_n	$J/(kg \cdot K)$	specific heat capacity; spezifische Wärmekapazität
D	m^2/s	diffusion coefficient; Diffusionskoeffizient
d	m	droplet diameter; Tropfendiameter
k	m/s	coefficient of mass transfer; Massentransportkoeffizient
L	m	length of compression chamber; Länge der Verdichterraum
M	$kg/kmol$	molar mass; Molare Masse
m	kg	mass; Masse
n	-	polytropic exponent; Polytropenexponent
P	Pa	pressure; Druck
P''	Pa	vapour saturation pressure; Sättigungsdruck
q	J/kg	specific heat; Spezifische Wärme
q_m	kg/s	rate of mass flow; Massendurchfluss
q_v	m^3/s	rate of volume flow; Volumendurchfluss
R	$J/(kg \cdot K)$	gas constant; Gaskonstante
r	J/kg	specific latent heat of vaporization; spezifische Verdampfungsenthalpie
T	K, s	thermodynamic temperature, or time constant; thermodynamische Temperatur, oder Zeitkonstante
t	^oC	Celsius temperature; Celsius-Temperatur
v	m^3/kg	specific volume; spezifisches Volumen
W	J/m^3	specific compression work; spezifische Verdichterarbeit
w	m/s	velocity; Geschwindigkeit
x	-	vapour/air mass ratio; Massenverhältnis Dampf/Luft

α	$W/(m^2 \cdot K)$	coefficient of heat transfer; Wärmeübergangskoeffizient
β	-	rate of heat capacity ratio; Wärmekapazitätverhältnis
Δw	m/s	slip velocity; Relativgeschwindigkeit
Θ, ϑ	K	temperature difference; Temperaturdifferenz
$\varkappa$	-	isentropic exponent; Isentropenexponent
λ	$W/(m \cdot K)$	thermal conductivity; Wärmeleitfähigkeit
ν	m^2/s	kinematic viscosity; kinematische Viskosität
π	-	pressure ratio; Druckverhältnis
ρ	kg/m^3	density; Dichte
τ	s	time; Zeit
Φ	W	rate of heat flow; Wärmestrom

$Re = w \cdot d/\nu$	Reynolds number
$Bi = \alpha \cdot d/(2 \cdot \lambda_L)$	Biot number
$Nu = \alpha \cdot d/\lambda$	Nusselt number
$Sh = k \cdot d/D$	Shannon (Sherwood) number
$Pr = \rho \cdot \nu \cdot c_p/\lambda$	Prandtl number
$Sc = \nu/D$	Schmidt number

Subscripts / Indizes

ax	axial; axial
i	initial; Anfang
L	liquid; Flüssigkeit
t	tangential; tangential
v	vapour; Dampf
w	droplet slip; Relativgeschwindigkeit
1	inlet, or bulk gas; Einlauf, oder Gasmasse
2	outlet, or droplet surface; Ablass, oder Tropfenfläche

1. Idealization of the liquid injected screw compressor

For compression of atmospheric air to 8 bar(a), the reduction in power consumption should theoretically be 27 % for isothermal compression, as compared to single-stage isentropic compression. This study /1/ was performed to investigate, if the potential for power reduction could be utilized in practice.

The tip speed for an oil-injected screw compressor for air compression, should be within the range 15 - 40 m/s (somewhat higher with water injection). For a medium size compressor, the compression time will then be roughly 3 - 15 ms. Thermal equilibrium air-liquid can then not be obtained during the compression.

1.1 General approximations

- All injected liquid assumed to be in the form of a uniform spray of spherical droplets.
- Droplets assumed to flow through the compressor with the axial air velocity.
- Accumulated liquid, due to internal leakage, assumed to be in the form of a liquid film, i.e. contribution to heat transfer could be neglected.
- Heat transfer to rotors and casing neglected.
- Uniform droplet temperature ($Bi < 1$).
- Losses due to air and liquid leakage not considered.
- Air flow throttling losses not considered.
- Polytropic compression of air, with polytropic exponent $1 \lesssim n < \varkappa$.

1.2 Droplet slip velocity

The slip velocity components Δw between air and liquid droplets, have been defined in fig. 1:

Fig. 1. Droplet slip velocities

Bild 1. Tropfen-Relativgeschwindigkeit

Tiny droplets injected into the screw compressor will rapidly be accelerated in axial and tangential direction, by the aerodynamic drag. For very low Re, the drag coefficient should be calculated according to Stoke, and a first order differential equation will then be obtained for the droplet motion. The time constant T_w is independent of the absolute air velocity component w, and will be the same for axial and tangential motion:

$$T_w = \frac{1}{18} \cdot \frac{\rho_L}{\rho} \cdot \frac{d^2}{\nu} \qquad (1)$$

The droplet response time T_w has been calculated vs' droplet diameter d, in table 1. Due to the rotation of the screw compressor rotor cavities, liquid droplets will be exposed to a centrifugal force, giving a radial droplet velocity. The radial steady state slip velocity will be determined by the balance between centrifugal force, due to rotation, and aerodynamic drag. The radial slip velocity Δw has also been calculated vs' droplet diameter, in table 1, with the drag coefficient according to Stoke, or Prandl (for Re > 1). If injected at low radial velocity, droplets should rapidly accumulate on the compressor casing, due to centrifugal force. However, for this simplified study, all droplets have been assumed to survive in droplet form during the compression, representing a theoretical upper limit for the heat transmission air/liquid.

DROPLET DIAMETER	AXIAL & TANGENTIAL SLIP		RADIAL SLIP		
d µm	Re_{max}	T_w ms	Δw m/s	Re	radial (lobe) traversing time ms
1	1	0.002	0.02	0.002	2500
10	13	0.2	1.5	1.7	33
20	25	0.9	4	9.1	12
50	60	6	12	68	4
100	130	23	23	260	2

Table 1. Droplet slip calculation

Tabelle 1. Tropfen-Gleitberechnung

2. Oil injection - transient heat transfer

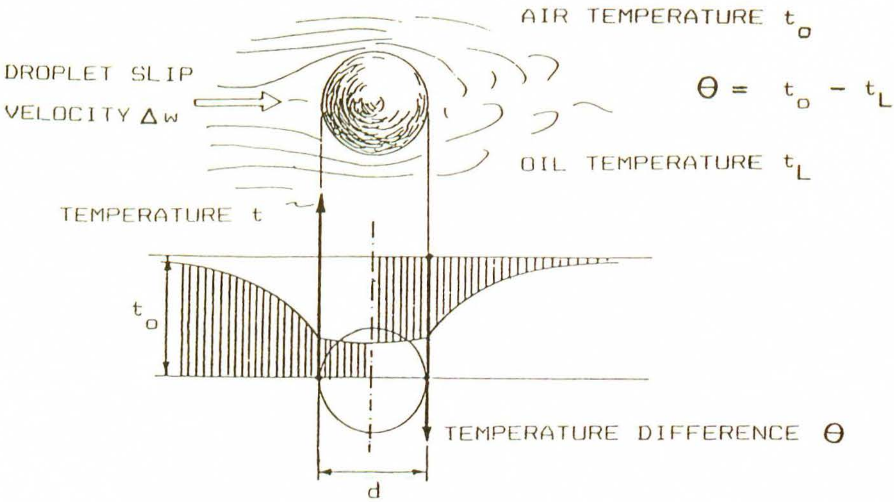

Fig. 2. Heat transfer and temperature distribution for an oil droplet

Bild 2. Wärmetransport und Temperaturverteilung eines Öltropfens

For convective heat transfer between spheres and air (fig. 2), without phase change, the Nusselt number will be, according to /2/:

$$Nu = \begin{cases} 0.37 \cdot Re^{0.6} \; ; & 17 < Re < 70\,000 \\ 2 \; ; & Re \leq 17 \end{cases} \qquad (2)$$

With droplet size and Reynolds number according to table 1, Nu will be within the range 2 - 10, and $Bi < 1$. The thermal resistance on the air side will then be dominating and the droplet temperature should be approximately uniform /3/. Equations for the heat exchanger model in fig. 3:

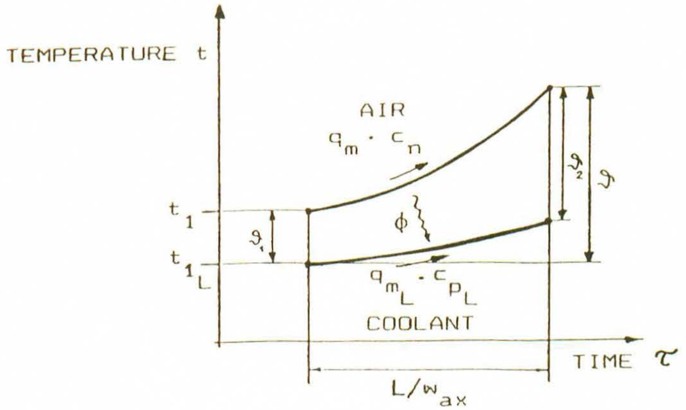

Fig. 3. Heat exchanger model for the oil injected compressor

Bild 3. Modell des Wärmeaustauschs in einem öleingespritzten Schraubenkompressor

Liquid surface area in the compression zone:

$$A = \frac{6 \cdot q_{m_L}}{d \cdot \rho_L} \cdot \frac{L}{w_{ax}} \tag{3}$$

Polytropic specific heat capacity of air:

$$c_n = \frac{R \cdot (\varkappa - n)}{(n-1) \cdot (\varkappa - 1)} \tag{4}$$

Rate of heat flow air $\longrightarrow$ coolant:

$$\phi = \alpha \cdot A \cdot \vartheta_m = q_m \cdot c_n \cdot (\vartheta - \vartheta_1) = q_{m_L} \cdot c_{p_L} \cdot (\vartheta - \vartheta_2) \tag{5}$$

Logarithmic mean temperature difference:

$$\vartheta_m = \frac{\vartheta_2 - \vartheta_1}{\ln(\vartheta_2/\vartheta_1)} \quad ; \quad (\vartheta_1 > 0) \tag{6}$$

Air/coolant rate of heat capacity ratio:

$$\beta = \frac{q_m \cdot c_n}{q_{m_L} \cdot c_{p_L}} \tag{7}$$

Combining equations (3) - (7) yields:

$$\frac{6\cdot Nu(Re(d))\cdot\lambda\cdot L}{d^2\cdot\rho_L\cdot c_{p_L}\cdot w_{ax}}\cdot\frac{1}{\beta} = \begin{cases} \dfrac{\ln\left[(\vartheta/\vartheta_1)\cdot(1-\beta)+\beta\right]}{1-\beta}; & \beta\neq 1\\ \vartheta/\vartheta_1 - 1 & ; \beta = 1\end{cases} \tag{8}$$

ϑ_1 is given by the difference in inlet temperature between air and oil injected. The air temperature increase during compression, will be calculated from the polytropic relation:

$$\vartheta - \vartheta_1 = T_1\cdot(\Pi^{\frac{n-1}{n}} - 1) \tag{9}$$

Lower limits for the mass flow ratio are given by $\vartheta_2 = 0$, corresponding to thermal equilibrium:

$$(q_{m_L}/q_m)_{\alpha\cdot A\to\infty} = \left[1 - \vartheta_1/\vartheta(n)\right]\cdot c_n\,(n)/c_{p_L} \tag{10}$$

From equations (8) - (10), the specific work of compression and the polytropic exponent have been plotted vs' mass ratio oil/air, with the droplet diameter as parameter. One representative example is presented in fig. 4. As seen from the diagram, very small droplets will be required to provide surface enough for any significant heat transfer and power saving.

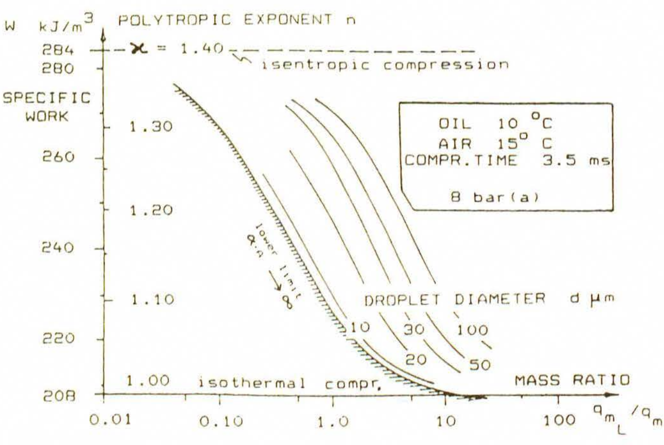

Fig. 4. Specific work of compression and polytropic exponent vs' oil/air mass ratio and droplet size

Bild 4. Spezifische Kompressionsarbeit und Polytropenexponent im Vergleich zum Öl/Luft-Verhältnis und zur Tropfengrösse

3. Water injection/evaporative cooling - transient heat and mass transfer

For volatile coolants, calculations based on the assumption of complete thermal equilibrium have shown a dominant influence from the vaporization /4/. With this assumption, only a small amount of water should be required for efficient cooling, due to the high heat of vaporization, as compared to the sensible heat capacity of water. For a high-speed screw compressor, however, a model for non-stationary heat and mass transfer will be required (fig. 5).

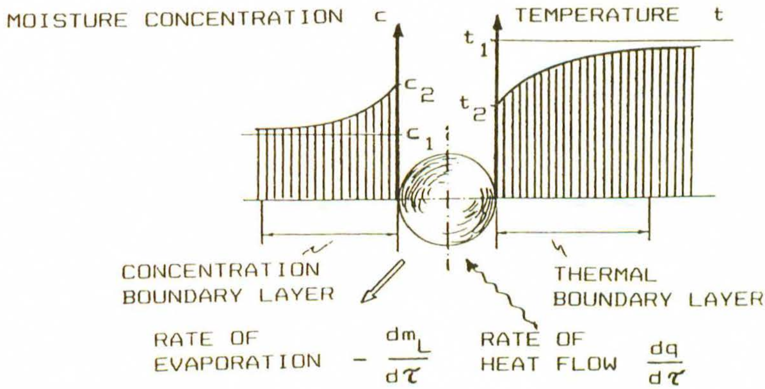

Fig. 5. Heat and mass transfer, and temperature and vapour concentration distributions for a water droplet

Bild 5. Wärme- und Massetransport sowie Temperatur/Wasserdampf-Konzentrationsverteilung eines Wassertropfens

Correlations for forced convection heat and mass transfer /5/:

$$Nu = 2 + 0.6 \cdot Re^{1/2} \cdot Pr^{1/3} \tag{11}$$

$$Sh = 2 + 0.6 \cdot Re^{1/2} \cdot Sc^{1/3} \tag{12}$$

Rate of heat flow from air to water droplets, per unit mass of air:

$$\frac{dq}{d\tau} = \frac{\alpha \cdot A}{m} \cdot (T_1 - T_2) \tag{13}$$

Rate of evaporation:

$$\frac{dm_L}{d\tau} = - k \cdot A \cdot (c_2 - c_1) \tag{14}$$

Vapour pressure in bulk air:

$$p_{v_1} = \frac{p_1}{\left(1 + \frac{M_L}{x \cdot M}\right)} \leq p''(T_1) \tag{15}$$

Vapour concentration in bulk air:

$$c_1 = \frac{p_{v_1}}{R_L \cdot T_1} \tag{16}$$

Vapour concentration at droplet surface:

$$c_2 = \frac{p''(T_2)}{R_L \cdot T_2} \tag{17}$$

Change of specific volume of air with time:

$$\frac{dv}{d\tau} \approx - \frac{w_{ax}}{L} \cdot v_i \cdot (1 - \Pi^{-1/n}) \tag{18}$$

Changes of state during time element $d\tau$:
Air temperature change:

$$dT_1 = - \frac{1}{c_v} \cdot (p_1 \cdot dv + dq) \tag{19}$$

Droplet temperature change:

$$dT_2 = \frac{1}{c_{p_L}} \cdot \left[\frac{dq}{m_L/m} + r(T_2) \cdot \frac{dm_L}{m_L} \right] \tag{20}$$

Change of air humidity:

$$dx = - \frac{m_L}{m} \cdot \frac{dm_L}{m_L} \tag{21}$$

Compression work per unit volume of air:

$$dW = \rho_i \cdot (c_p \cdot dT_l + dq) \tag{22}$$

Equations (13) - (22) have been solved numerically. Graphs for the specific work of compression vs' relative liquid amount, with droplet size as parameter, are given in fig. 6, for one representative case of air compression to 8 bar(a) with water injection. Lower limit for the specific work has been calculated based on the assumption of thermal equilibrium. The very small amount of water required theoretically, is clearly seen. For realistic values of the droplet diameter, however, considering the time required for evaporation, as well as the coupling between droplet temperature and rate of evaporation, the contribution to cooling from evaporation will be negligible.

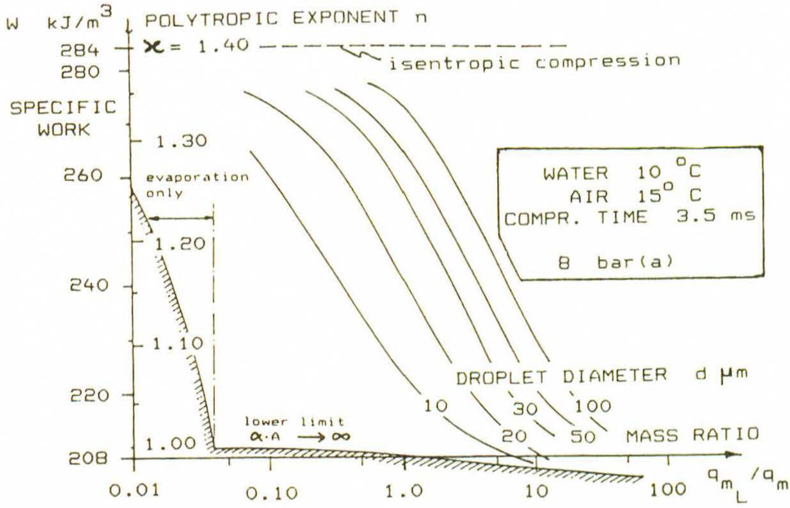

Fig. 6. Specific work of compression and polytropic exponent, vs' water/air mass ratio and droplet size

Bild 6. Spezifische Kompressionsarbeit und Polytropenexponent im Vergleich zum Wasser/Luft-Masseverhältnis und Tropfengrösse

4. Injection nozzles

Though theories for liquid jet and film breakup have been established /6/, no general method for prediction of drop-size distributions is available. A variety of empirical correlations, valid for specific nozzle types, have been proposed, however /6/. The theoretical energy requirement for atomization of a liquid is defined by the product of surface tension and the surface area produced. For commercial atomizers, the efficiency is very low, usually 0.1 - 0.5 % only. As $A \propto 1/d$, a considerable amount of energy will be required for atomization to droplets of small diameter d.

4.1 Hydraulic nozzles of centrifugal (swirl) type

Pressure-driven hydraulic nozzles, and the full cone type in particular, produce a rather coarse spray. In hollow cone nozzles, the liquid enters the nozzle chamber tangentially, and a free vortex will be formed. Due to the pressure drop in the centre of a free vortex, the liquid cavitates and an air core will be formed. In full cone nozzles, a forced vortex will be created by means of internal helical vanes. In a forced vortex, the liquid rotates like a solid body and no air core will be formed.

4.2 Two-fluid (pneumatic) nozzles

Compressed air and liquid will be mixed in the nozzle. These nozzles, especially the sonic type, produce a very fine spray, at the expense of a high power requirement. In sonic nozzles, the air is expanded through a supersonic nozzle, and the liquid is passed through a shock wave.

4.3 Spinning disc atomizers

For this type of atomizer, work is imparted to the fluid by means of an impeller. The tangential velocity is therefore independent of the flow rate and the feed pressure. The power requirement will be of the same order of magnitude as for hydraulic nozzles.

4.4. Nozzle performance

Empirical data on droplet size vs' flow rate for atomization of water have been summarized in fig. 7. In fig. 8, rough estimations of nozzle power consumption vs' water flow rate have been plotted.

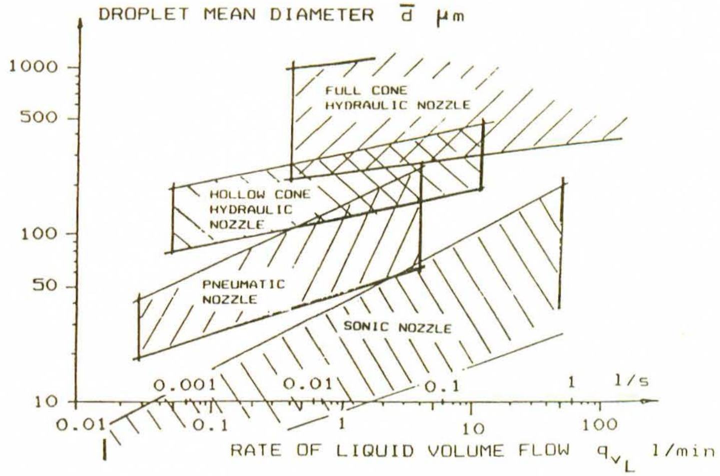

Fig. 7. Droplet mean diameter vs' flow rate, for commercial atomizer nozzles

Bild 7. Mittlerer Tropfendurchmesser im Vergleich zur Durchflussrate marktüblicher Zerstäuberdüsen

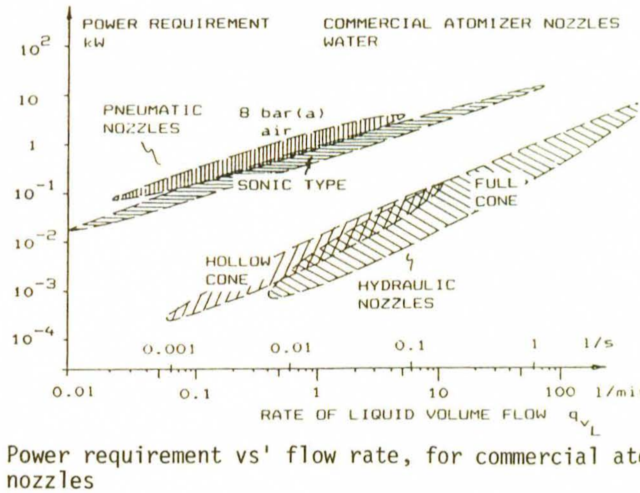

Fig. 8. Power requirement vs' flow rate, for commercial atomizer nozzles

Bild 8. Leistungsbedarf im Vergleich zur Durchflussrate marktüblicher Zerstäuberdüsen

5. Comparison of compression and atomization work

In fig. 9, the specific work of atomization for a water injected screw compressor has been plotted together with the specific work of compression. For the high degree of atomization required to reduce the compression work, a pneumatic nozzle with a very high power consumption should be required. The reduction in compression work due to improved cooling will then be counterbalanced by the considerable energy requirement for atomization.

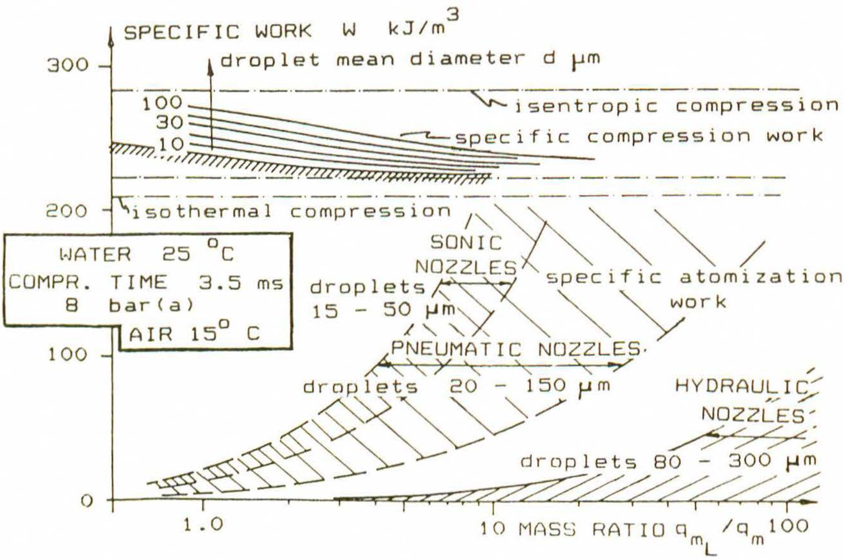

Fig. 9. Specific work of atomization compared to specific work of compression, for a water injected screw compressor

Bild 9. Spezifische Zerstäubungsarbeit verglichen mit der spezifischen Kompressionsarbeit eines wassereingespritzten Schraubenkompressors

Experience from tests of oil and water injected screw compressors has also shown that, in high-speed screw compressors, any significant reduction of the specific work for compression cannot even be obtained by injection of a very fine spray. This is an indication that, due to internal turbulence and centrifugal action, the droplets will rapidly form a liquid film, with a much lower surface/volume ratio, on the compressor casing.

The main advantage of liquid injection will therefore be the improved sealing of the clearances, the improved cooling of rotors and casing, and the lubricating film in the rotor mesh. By improved cooling of the metal parts, thermal distorsions will be reduced and a higher air temperature and hence a much higher stage pressure ratio can then be accepted, than for dry screw compressors.

6. References

/1/ Persson, J-G: Heat Exchange in Liquid Injected Compressors. Licentiate thesis; Stockholm: The Royal Institute of Technology, department of Fluid Technology, Jan. 1986.

/2/ McAdams, W H: Heat Transmission; 3. ed. New York, London: McGraw-Hill 1954.

/3/ Jakob, M: Heat Transfer. Vol I; 7. printing. New York, London: Wiley 1959.

/4/ Gneipel, G: Innenkühlung von Hubkolbenverdichtern - Eine Möglichkeit zur Wirkungsgradsteigerung. Maschinenbautechnik Vol. 27.6 (1978).

/5/ Ranz, W; Marshall, W R Jr: Evaporation from drops. Chemical Engineering Progress Vol. 48.3 - 4 (1952).

/6/ Marshall, W R Jr: Atomization and Spray Drying. New York: American Institute of Chemical Engineers 1954.