Calculus of Indicated Power by Mathematical Modeling Method of Compression Process and Study of Exergetic Efficiency of the Helical Screw Compressor with Oil Injection

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Abstract: The first part of the article studies the problem of establishing a mathematical model that ensures the calculus of the output power for the helical screw compressor with oil injection. In the second part the problem of establishing the equation of exergetic efficiency for helical screw compressors with oil injection is further explored. The obtained results are checked by carrying out case studies on real compressors.

Key words: indicated power, exergy, helical screw compressor, mathematical modeling method, exergetic efficiency, modeling of working conditions

1 Determination of mechanical power of the helical screw compressor with oil injection using mathematical modeling

1.1 Introduction

Helical screw compressors are used in the pneumatic equipment to lift the air pressure to the desired level meeting the demand of the technological process that uses compressed air.

The working conditions of a compressor are imposed by two unconnected terms: the compression ratio π and the velocity of rotation n.

The basic characteristics of these equipments are: volumetric efficiency η_v; the indicated power P_i; outlet pressure p_2,; the main rotor shaft speed n.

The energy consumption of a pneumatic equipment using helical screw compressors changes accordingly to the intake air parameters and the variation of network pressure.

1.2 Calculus method

The inlet power of the helical screw compressor can be obtained using the analytical modeling method in relation to their specific parameters. For other compressor models these parameters are not that significant[10].

The connection between power and the related parameters can be illustrated using an equation of form:

$$P = f(\pi, n, M, k, c_p, \mu, \lambda, p_1, T_1) \quad (1)$$

where: π is the compression ratio;
n – main rotor shaft speed;
M – molecular weight of air;
c_p - constant pressure mass heat capacity;
μ - dynamic viscosity of air;
λ - thermal conductivity;
p_1, T_1 – inlet air parameters.

Using the dimensionless form of the equations and reducing the number of unconnected terms the analysis of characteristics of equipments using helical screw compressors can be simplified.

Accordingly, the adiabatic efficiency can be expressed related to the key parameters (π and k) and the dimensionless terms, using an equation like:

$$\eta_{ad} = f(\pi, k, Ma, Re, Pr) \quad (2)$$

where: Ma - Mach number;
Re - Reynolds number;
Pr - Prandtl number.

The Mach, Reynolds and Prandtl numbers can be calculated using the well-known relations:

$$Ma = \frac{w_1}{w_s} \tag{3}$$

where: w_1 - air speed in the inlet pipe;
w_s – speed of sound at inlet conditions.

$$Re = \frac{w_1 \cdot \varphi_1 \cdot D}{\mu} \tag{4}$$

where D equivalent (hydraulic) diameter.

$$Pr = \frac{c_p \cdot \mu}{\lambda} \tag{5}$$

The Mach and Reynolds number can be calculated using specific values:
- axial velocity of the so-called "contact line" of the rotors:

$$w_a = \frac{u_{e1}}{tg\,\beta_{e1}} \tag{6}$$

where u_{e1} – rotational velocity on the external diameter of the main rotor;
- local speed of sound in the gas at inlet conditions;
- the average hydraulic diameter D is the ratio between 4 times the frontal surface of rotor cavities of the main rotor S_1 and the secondary rotor S_2, which together shape an intermeshed cavity and the entire humidified perimeter of frontal surface Pr_1 and Pr_2 (where Pr_1 and Pr_2 are the perimeters of the frontal surface of main and secondary rotor cavities and their inner facing) of the body:

$$D = \frac{4 \cdot (S_1 + S_2)}{Pr_1 + Pr_2} \tag{7}$$

The influence of viscosity and thermal conductivity of gas on the dimensionless characteristics of the helical screw compressor (the adiabatic efficiency η_{ad} and the volumetric efficiency η_v) can be calculated only using the values of Reynolds and Prandtl numbers.

The leakage coefficient ξ is used when the hydraulic method for the study of processes in the air distribution machinery of helical screw compressors is applied.

The use of method is appropriate when Re < Re_{pr} (Re_{pr} – preliminary Reynolds number).

Re > $Re_{pr} = (1 \div 2) \cdot 10^5$, therefore ξ is not a function of Re:

$$\xi = \frac{2 \cdot l_p}{c^{-2}} \tag{8}$$

where: l_p mechanical energy loss divided by the mass unit of air flow;
c - the flow velocity on the normal surface.

In the working zone of the compressor equipment the Re number is above Re_{pr}, and Pr number for the greater part of gases fulfilling the condition $14 \le M \le 90$ and $1.2 \le k \le 1.5$ is 0.7 ± 0.1, the conclusion is that the similarity is not a function of Re and Pr numbers.

Therefore, the equation (2) became:

$$\eta_{ad} = f(\pi, Ma, k) \tag{9}$$

From equation (2) results that keeping Ma and π constant, the characteristics of compressor equipments are not functions of molecular weight M.

The connection of the dimensionless characteristics of the compressor to the adiabatic index k, can be established using formula [1]:

$$\begin{aligned}\eta_v &= (0.97 - 0.99) - f(S_1 + S_2)^{-1} \beta \cdot Ma^{-1} = \\ &= \left[2(k-1)^{-1}\right]^{\frac{k+1}{k-1}} \cdot \left[\left(\beta^{\frac{k-1}{k}} - 1\right) \cdot \eta_v^{-1} \cdot \eta_{iT}^{-1} \cdot \eta_h^{-1} + 1\right]^{0.5}\end{aligned} \tag{10}$$

and

$$\eta_{ad} = \eta_v \cdot \eta_h \cdot \eta_{iT} \tag{11}$$

which shows the connection between: η_{ad} , η_v, π, Ma, k.

The relative surfaces of the cavities of rotors were marked with S_1 and S_2, the theoretical indicated efficiency with η_{iT} and η_h the hydraulic efficiency.

$$\eta_{iT} = \left(\pi^{\frac{k-1}{k}} - 1\right) \cdot \left[\varepsilon^{k-1} - (k-1) \cdot k^{-1} \cdot \left(\varepsilon^{k-1} - \pi\varepsilon^{-1}\right) - 1\right]^{-1} \tag{12}$$

where: $\varepsilon = \varepsilon_r$ is the geometric compression ratio which represents the ratio between the volume of double cavities of rotors at the time they disconnect from the inlet chamber and the volume of the same cavity at the time of overlapping the outlet chamber.

The hydraulic efficiency η_h is the expression of the pressure loss as a result of flow resistance, linked to the roughness of surface. This can be as much as 10% from the sum of losses in the compressor.

Calculus can be performed using relation [1]:

$$\eta_h = \left\{0.5 \cdot \xi (k-1) \varepsilon_r \cdot Ma^2 \cdot \eta_v \cdot \left[\left(\varepsilon_r^k - \pi\right) + \pi - \varepsilon_r\right]^{-1}\right\}^{-1} \quad (13)$$

Experimental data [2] points that the surface of clearances f, which is function of the amount of losses, is in relation with the temperature:

In figure 1 the surface variation of hot clearances function of the outlet temperature T_2 and the inlet temperature T_1 is presented, calculated with the empiric equation [1]:

$$f = f_1 \cdot \left(1.45 - 3.85 \cdot \lg \frac{T_2}{T_1}\right) \quad (14)$$

where: f_1 is the ratio between the surface of clearances calculated for $T_2 \cdot T_1^{-1} = 1.3$ and the surface of the normal section of main and secondary rotor cavities.

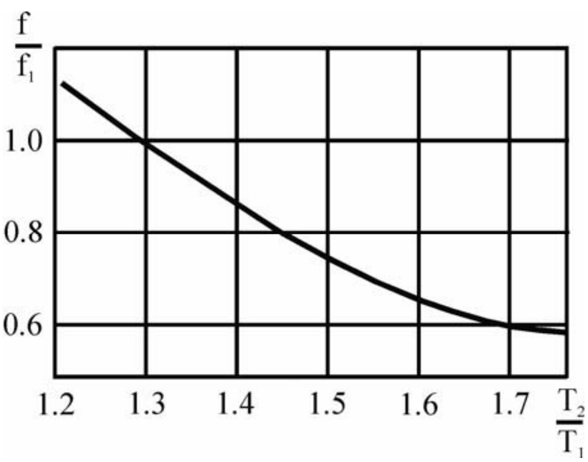

Fig 1 Variation of the calculus surface of clearances relative to temperatures T_1 and T_2

As the temperature ratio rises, the relative surface of rotor cavities drops.

Considering that in the compressor stage the highest temperature raise for the inlet air (to the upper limit) is taking place, the minimum adiabatic efficiency, can be calculated using relation [1]:

$$\eta_{ad-min} = T_1 \cdot \frac{1}{\Delta T} \cdot \left(\varepsilon_r^{\frac{k-1}{k}} - 1\right) \quad (15)$$

If η_{ad-min} is less than $0.8 \div 0.83$ calculus must be continued.

For $\eta_{ad-min} > 0.8 \div 0.83$ a multi stage compression must be considered with air cooling between stages.

Compressors with oil injection cooling in the compression stage have distinct structure and as a result some limitations:

- minimum inlet air temperature must be $T_1 \geq 258$ K (-15^0C);
- maximum inlet air temperature $T_{1\ max} < 308$ K ($+35^0$C);
- maximum temperature increase per stage $\Delta T_{max} = 210\ ^0$C, using oil injection temperature drops to $\cong 45\ ^0$C;
- maximum pressure increase per stage $\Delta p_{max} \leq 7$ bar;
- maximum peripheral speed on the diameter of initial rotor perimeter $u_{max} = 110\ m \cdot s^{-1}$;
- maximum volumetric efficiency: $\eta_{v\ max} \leq (0.9 \div 0.96)$;
- theoretical indicated efficiency for $\pi > \varepsilon_r^k$; $\eta_{iT\ min} \geq 0.95$;
- maximum theoretical compression ratio $\pi = 6$.

Usually a value for η_{iT} can be obtained for two different values of π, larger values on the right branch of the characteristic, smaller ones on the left figure 2:

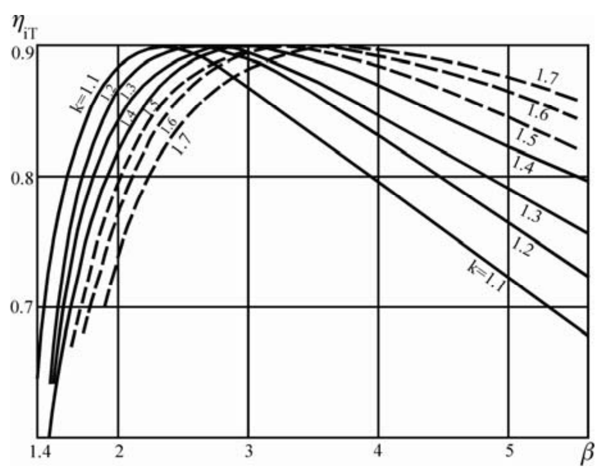

Fig 2 Function $\eta_{iT} = f(\pi)$ for k = ct.[1]

1.3 Case study

If η_{iT} obtained using formula (12) is on the right branch of the characteristic, $\pi > \varepsilon_r^k$ (fig 2), then calculus may continue.

From equation: $6 = \varepsilon_r^{1,4} \Rightarrow \varepsilon_r = 3.6$.

Results:

$$\eta_{iT} = \left(\pi^{\frac{k-1}{k}} - 1\right) \cdot \left[\varepsilon_r^{k-1} - \left(\frac{k-1}{k}\right) \cdot \left(\varepsilon_r^{k-1} - \frac{\pi}{\varepsilon_r}\right) - 1\right]^{-1} \quad (16)$$

$$\eta_{iT} = \left(6^{0.286} - 1\right) \cdot \left[3.6^{0.4} - 0.286\left(3.6^{0.4} - \frac{6}{3.6}\right) - 1\right]^{-1} = = 0.99$$

Condition $\eta_{iT} > 0.95$ is satisfied.

Adiabatic efficiency relative to temperature ratio can be determined from diagram presented in figure 3 [1].

$$\frac{T_2}{T_1} = \frac{273 + 210}{283} = 1.7\,;\ \eta_{ad} = 0.59$$

Condition $\eta_{ad\ max} \leq (0.8 \div 0.83)$ is satisfied since $0.59 < (0.8 \div 0.83)$.

As a result, the equation suggested for the calculus of the indicated power of helical screw compressors is:

$$P_i = \frac{k}{k-1} \cdot (\eta_v \cdot Ma) \cdot S_a \cdot w_s \cdot p_1 \cdot \left[\pi^{\frac{k-1}{k}} - 1\right] \cdot \frac{10^{-3}}{\eta_{ad}} [kW] \quad (17)$$

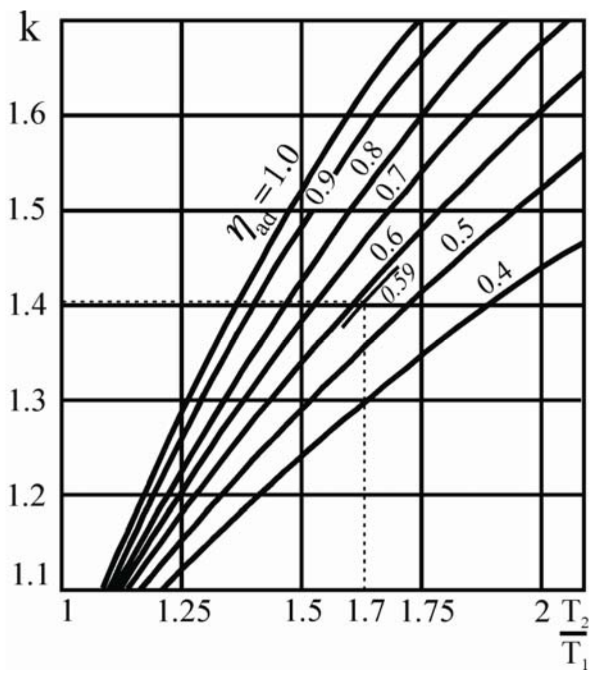

Fig 3 Adiabatic efficiency relative to temperature ratio

where: - $(\eta_v \cdot Ma)$ is the product between volumetric efficiency η_v and Mach number with the significance of an dimensionless complex. In other words $(\eta_v \cdot Ma)$ is a volumetric ratio between the equipment and the inlet pipe at a flow velocity equal with the sound velocity;

- S_a, conventional surface of the inlet;
- w_s, sound velocity of air at the inlet parameters;
- p_1, inlet pressure of air;
- π, compression ratio of air;
- η_{ad}, adiabatic efficiency of the helical screw compressor.

Equation (17) results from mathematical process modeling, so that the dimensionless complex $(\eta_v \cdot Ma)$ may be determined using mathematical modeling of dimensionless numbers:

$$\eta_v = f(\pi, Ma, Re, Pr, k) \quad (18)$$

This according to the relation of η_{ad} became:

$$\eta_v = f(\pi, Ma, k) \quad (19)$$

In order to establish the equation of indicated power some of the terms that occur must be explained:

- inlet sound velocity:

$$w_s = (k \cdot R \cdot T_1)^{0.5} = (1.4 \cdot 287 \cdot 283)^{0.5} = 337.2\ \frac{m}{s}$$

- inlet air density:

$$\rho_1 = \frac{p_1}{R \cdot T_1} = \frac{1 \cdot 10^5}{287 \cdot 283} = 1.23\quad \frac{kg}{m^3}$$

- inlet air mass flow:

$$\dot{m}_1 = \dot{V}_1 \cdot \rho_1 = \frac{43.5}{60} \cdot 1.23 = 0.89\quad \frac{kg}{s}$$

- conventional area of the inlet:

$$S_a = 9 \cdot \frac{\dot{V}_1}{w_s} = 9 \cdot \frac{0.725}{337.2} = 0.0193\quad m^2 \quad (20)$$

- the $(\eta_v \cdot M_a)$ dimensionless complex for S_a = 0,0193 m^2 results from equation (20) (see fig 4 also):

$$(\eta_v \cdot M_a) = \frac{\dot{V}_1}{w_s \cdot S_a} = \frac{0.725}{337.2 \cdot 0.0193} = 0.1114 \quad (21)$$

Hydraulic efficiency is calculated using equation (13).

The leakage coefficient ξ from equation (13) it's hard to be considered because is in relation with a large number of parameters as: roughness of surfaces in contact with the air flow; distribution of the velocity field in the intermeshed cavity of the compressor; changes in the air flow direction; when oil injection occurs during compression the medium became biphasic etc.

Recommended values in literature are $\xi = 0.075 \div 0.1$, using analogy with other rotary machines.

Assuming $\xi = 0.09$. Results:

$$\eta_h = \left\{0.5 \cdot 0.09 \cdot (1.4-1) \cdot 3.6 \cdot 0.32^2 \cdot 0.76 \cdot \left[\frac{3.6^{1.4}-6}{1.6} + 6 - 3.6\right]^{-1}\right\}^{-1} = 0.95$$

Volumetric efficiency $\eta_v = 0.76$, is obtained from figure 4[2].

Using this data, the adiabatic efficiency of helical screw compressor is:

$$\eta_{ad} = \eta_v \cdot \eta_{iT} \cdot \eta_h = 0.76 \cdot 0.99 \cdot 0.95 = 0.73 \quad (22)$$

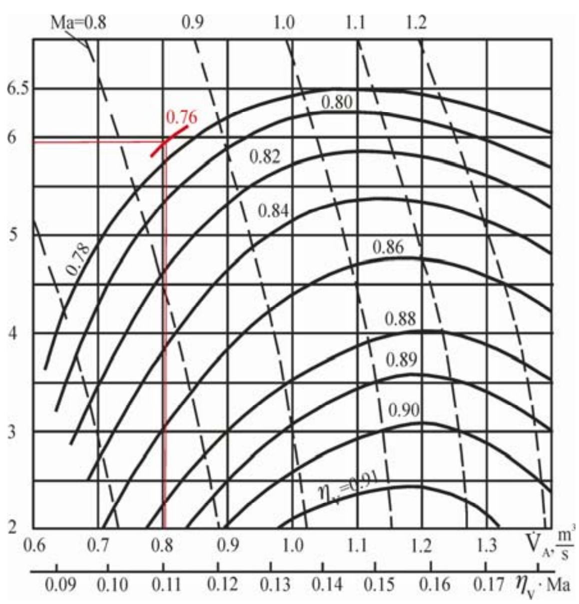

Fig 4 Volumetric efficiency evaluation

As all the parameters that occur in equation (17) where estimated, results:

$$P_i = \frac{1.4}{0.4} \cdot 0.1114 \cdot 0.0193 \cdot 337.21 \cdot 10^5 \cdot \left[6^{\frac{0.4}{1.4}} - 1\right] \cdot \frac{10^{-3}}{0.73} \cong 230 \quad \text{kW}$$

Using values given in the literature for the thermodynamic parameters involved with equation (17), and giving values for $T_1 = 258 \div 298$ K, and accordingly for velocity $w_1 = 321.97 \div 351.79$ m/s and for the inlet pressure $p_1 = 8.589 \cdot 10^4 \div 1.025 \cdot 10^5$ $N \cdot m^{-2}$, the matrix of values for the indicated power of helical screw compressor results:

$$P_i = \begin{pmatrix} 186.941 & 194.187 & 201.432 & 208.678 & 215.924 & 223.17 \\ 190.529 & 197.914 & 205.299 & 212.684 & 220.069 & 227.454 \\ 194.051 & 201.573 & 209.094 & 216.616 & 224.137 & 231.658 \\ 197.511 & 205.166 & 212.822 & 220.477 & 228.133 & 235.788 \\ 200.911 & 208.698 & 216.485 & 224.272 & 232.059 & 239.847 \\ 204.254 & 212.17 & 220.087 & 228.004 & 235.921 & 243.838 \end{pmatrix}$$

Indicated power P_i function of $T_1(w_1)$ and p_1 is presented in figure 5:

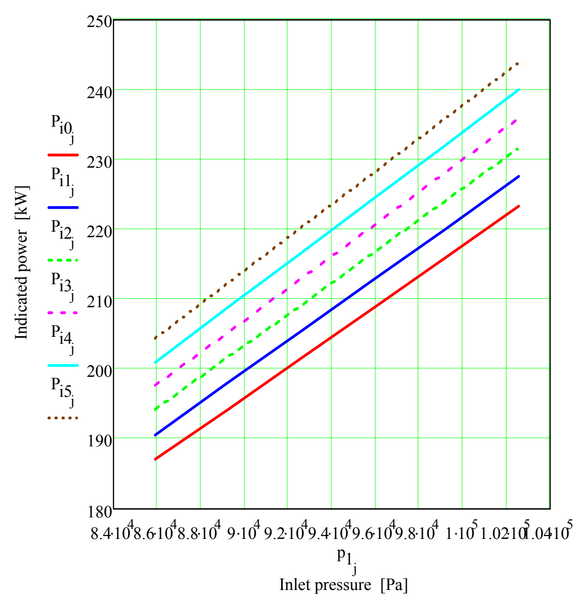

Fig 5 Indicated power P_i function of inlet parameters p_1, T_1 and gas velocity w_1

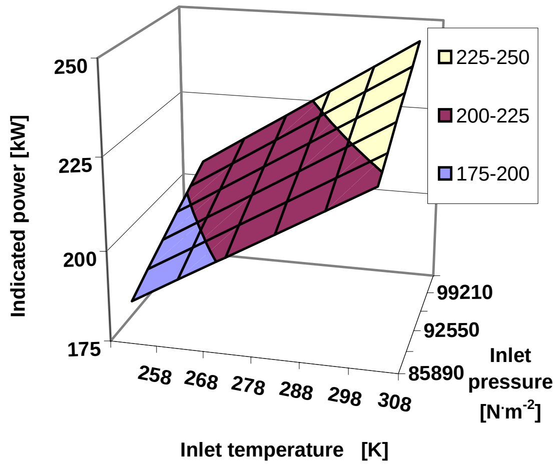

Fig 6 Power of helical screw compressor function of inlet pressure and temperature

2 Study of Exergetic Efficiency of the Helical Screw Compressor with Oil Injection

2.1 Introduction

The exergetic efficiency represents the ratio between the exergy of a process (as work) and the inlet exergy of the process.

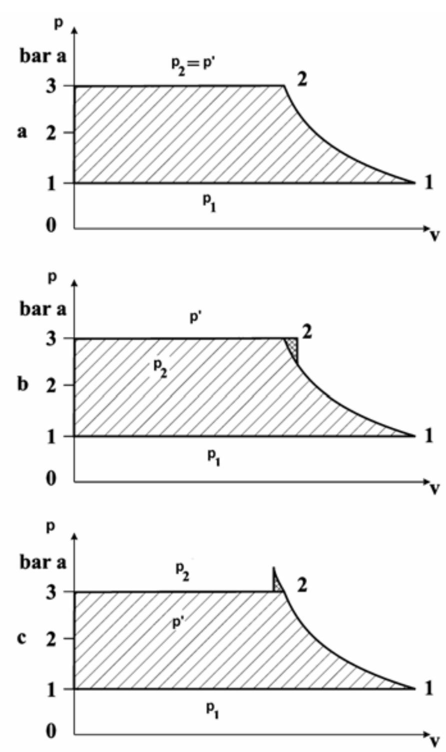

Fig 7 Different possible cycles for the helical screw compressor

The method of exergetic analysis is taking in account the influence of the environment over the conversion capacity of heat to work.

Exergetic analysis method provides a thorough investigation in analyzing thermal machinery that has energy losses due to the irreversibility of processes, like no other method can do.

Irreversible processes are part of models describing the behavior of modern machinery, so that the thermodynamic cycle analysis of thermal equipment must point to the possibilities of thermodynamic optimization relying on the losses produced by irreversibility.

Since the literature in the field of helical screw compressors presents no data regarding the exergetic efficiency, the authors considered that is appropriate to develop this problem.

The expression of exergetic efficiency of the helical screw compressor with oil injection will be established as follows.

2.2 Problem formulation

2.2.1 The equation of exergetic efficiency of helical screw compressor with oil injection

The cycle of the helical screw compressor is represented in figure 7 in all possible instances: optimum compression (a) under compression (b) over compression (c).

The analysis was carried out with regard to the parameters of environment p_0 and T_0, so that the suction takes place at the pressure of the environment $p_0 = p_1$, and the temperature $T_1 \neq T_0$.

The discharge occurs at the pressure $p_2 = p'$ (where p' is the pressure from the discharge pipe) required in the equipment that uses the compressed air.

$$\pi_{ir} = T_0 \cdot \Delta s_{ir} \tag{23}$$

The entropy raise is given by:

$$\Delta s_{ir} = c_p \cdot \ln\frac{T_2}{T_1} - R \cdot \ln\frac{p_2}{p_0} \tag{24}$$

where c_p is air specific heat; R – air gas constant.

Using $\pi = \frac{p_2}{p_0}$ as the pressure ratio in the compression process.

The temperature ratio is given by:

$$\frac{T_2}{T_1} = 1 + \frac{\pi^{\frac{k-1}{k}} - 1}{\eta_{ad}} > 1 \tag{25}$$

where η_{ad} is the adiabatic efficiency of the compressor, and k is the adiabatic index for air.

In figure 8 presents the ideal or isentropic compression (1-2s) and the non-isentropic compression (1-2).

The hatch area gives the exergy losses due to irreversibility of the process.

Taking in account of the inlet and outlet energy, the following relation can be developed:

$$e_1 + |l_t| = e_2 + \pi_{ir} \text{ or } |l_t| = e_2 - e_1 + \pi_{ir} \tag{26}$$

where l_t is the specific work.

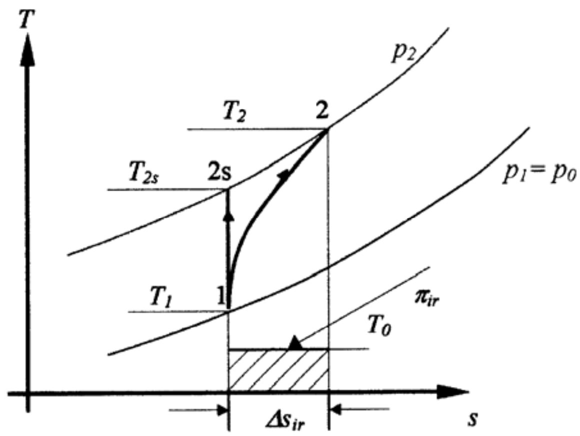

Fig 8 Entropy raised in real compression process

The entropy rise in account of irreversibility gives [5]:

$$\Delta s_{ir} = c_p \cdot \ln\frac{T_2}{T_1} - R \cdot \ln\frac{p_2}{p_0} = c_p \cdot \ln\left[1 + \frac{\pi^{\frac{k-1}{k}} - 1}{\eta_{ad}}\right] \cdot \pi^{\frac{k-1}{k}} \tag{27}$$

and

$$\pi_{ir} = T_0 \cdot \Delta s_{ir} = T_0 \cdot c_p \cdot \pi^{\frac{k-1}{k}} \cdot \ln\left(1 + \frac{\pi^{\frac{k-1}{k}} - 1}{\eta_{ad}}\right) \tag{28}$$

Calculating the degree of thermodynamic excellence of the compressor, the concept of exergetic efficiency is used, determining the ratio between the exergy obtained and the consumed work, for the compressor:

$$|l_t| = e_2 - e_1 + \pi_{ir} = c_p \cdot T_1 \cdot \left(\frac{\pi^{\frac{k-1}{k}} - 1}{\eta_{ad}}\right) \tag{29}$$

Subsequently, using relation (28), (29) and $\Delta e = e_2 - e_1$ results:

$$\eta_{Ex} = \frac{\Delta e}{|l_t|} = 1 - \frac{\pi_{ir}}{|l_t|} = 1 - \frac{T_0 \cdot c_p \cdot \pi^{\frac{1-k}{k}} \cdot \ln\left(1 + \frac{\pi^{\frac{k-1}{k}} - 1}{\eta_{ad}}\right)}{c_p \cdot T_1 \cdot \left(\frac{\pi^{\frac{k-1}{k}} - 1}{\eta_{ad}}\right)} \tag{30}$$

$$\eta_{Ex} = 1 - \frac{T_0}{T_1} \cdot \frac{\pi^{\frac{1-k}{k}} \cdot \ln\left(1 + \frac{\pi^{\frac{k-1}{k}} - 1}{\eta_{ad}}\right)}{1 + \frac{\pi^{\frac{k-1}{k}} - 1}{\eta_{ad}}}$$

The expression of the exergetic efficiency includes the sum of the energy losses caused by the irreversibility of the process.

The relationship of exergetic efficiency reveals that this relation depends not only on the pressure ratio, and the adiabatic efficiency, but also on the suction temperature T_1 and the environment temperature T_0.

The literature points that the natures of losses, due to irreversibility are both mechanical and thermal:

$$\pi_{ir} = \pi_{ir\,T} + \pi_{ir\,p} \tag{31}$$

The thermal component is the absolute value of the exergy of the heat exchange with the environment, as the temperature varies from T_0 toT_1.

Assuming that the compression process is adiabatic, then $\pi_{irT} = 0$.

The mechanical part leads to a drop of exergy due to lamination of air [10]:

$$\Delta e_p = R \cdot T_0 \cdot \ln\frac{p_2'}{p_0} < 0 \tag{32}$$

where p_2' is the pressure after lamination.

Note $\Delta p = p_2 - p_2'$, the pressure drop in the lamination process and $\Psi_p = \frac{\Delta p}{p_0}$, the relative pressure drop in the same process, and developing in series the logarithm, gives:

$$\pi_{ir\ p} = \pi_{ir\ l} = -\Delta e_p = -R \cdot T_0 \cdot \left(1 - \frac{\Delta p}{p_0}\right) \cong \cong R \cdot T_0 \cdot \Psi_p \tag{33}$$

If the pressure drops during lamination affects negatively the discharge pressure.

With this statement, the expression of exergetic efficiency is given by:

$$\eta_{Ex} = 1 - \frac{\pi_{ir} + \pi_{ir\,l}}{|l_t|} = 1 - \frac{T_0}{T_1} \cdot \frac{\pi^{\frac{1-k}{k}} \cdot \ln\left(1 + \frac{\pi^{\frac{1-k}{k}} - 1}{\eta_{ad}}\right) + 0.286 \cdot \Psi_p}{\frac{\pi^{\frac{1-k}{k}} - 1}{\eta_{ad}}} \tag{34}$$

Consequently, during compression the used work can be found in the increase of air exergy and in the losses due to the irreversibility of lamination process.

Equation (34) is an original and is the contribution of the authors to the study of the exergetic efficiency of the helical screw compressors.

Regarding the coefficient Ψp the literature doesn't provide any experimental data. As result of laboratory measurements, was established that the value of Ψp between 6÷10% from the pressure raise.

In order to validate the relation (34) a program was developed, that takes account of the parameters defined in the expression of the exergetic efficiency, using specific values.

2.3 PROBLEM SOLVING

2.3.1 Validating the expression of exergetic efficiency of the helical screw compressor with oil injection

Oil injection provides a way to fully control the discharge temperature regardless of compression ratio, and therefore the operating limits of the compressor can be established on efficiency basis, and not by discharge temperature limitations.

Discharge temperature is usually kept under 100 ºC, but sometimes can get up to 130 ºC using adequate lubricant. [4].

The volume of injected oil is under 1% of the relative volume of suction air.

The suction gas is air therefore value for the adiabatic index is k = 1.4, and the gas constant for air $R = 0.287 \frac{kJ}{kg \cdot K}$.

Analytic study of exergetic efficiency is inconvenient, providing results difficult to use, that's why a graphical analysis method will be applied.

In order to use the graphical method, different values that cover all practical working conditions will be stated for the parameters, so that by combining them a comprehensive set of different situations can be covered.

Therefore the limits for the adiabatic efficiency where stated between 0.6÷0.85 with iterations of 0.05, the compression ratio between 4÷8 with iterations of 0.5. Using the original equation (34) calculus was carried out using different data sets in order to determine the appropriateness of the equation that gives the exergetic efficiency of the helical screw compressor with oil injection.

η_{Ex} =

	0	1	2	3	4	5
0	0.49	0.478	0.467	0.457	0.447	0.438
1	0.521	0.509	0.498	0.488	0.479	0.47
2	0.546	0.535	0.524	0.514	0.505	0.496
3	0.568	0.557	0.546	0.536	0.527	0.519
4	0.587	0.576	0.565	0.556	0.547	0.538
5	0.603	0.592	0.582	0.573	0.564	0.556
6	0.618	0.607	0.597	0.588	0.579	0.571
7	0.636	0.625	0.615	0.606	0.598	0.59
8	0.643	0.632	0.623	0.614	0.605	0.597
9	0.773	0.765	0.758	0.75	0.744	0.737

Fig 9 The matrix of values for the exergetic efficiency

In the matrix of values (figure 9) [1], can be found the limits of values for the exergetic efficiency of the helical screw compressor obtained using the parameters defined before, and for the following temperature values: the environment temperature (different from suction temperature) T_0 = 273 K; the medium suction temperature for the last six months T_1= 283K; the medium discharge temperature T_2 = 313 K.

Using data given in the matrix above, graphics were plotted to determine the correctness of equation (34).

The limits in which the compression ratio varies where $\pi = 1,5 - 8$ (fig. 10), and $\pi = 1.5 - 20$ for (fig. 11) in order to highlight the trend of the variation.

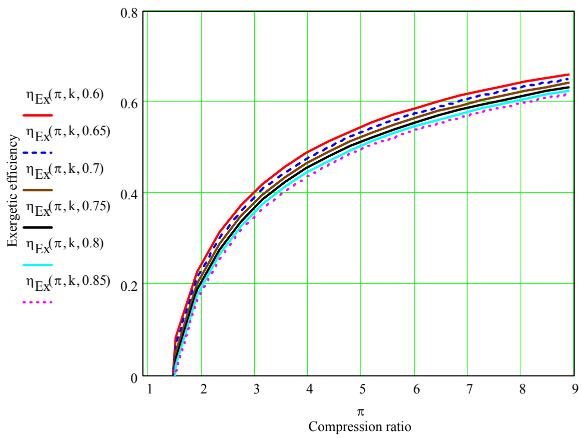

Fig 10 Exergetic efficiency [1]

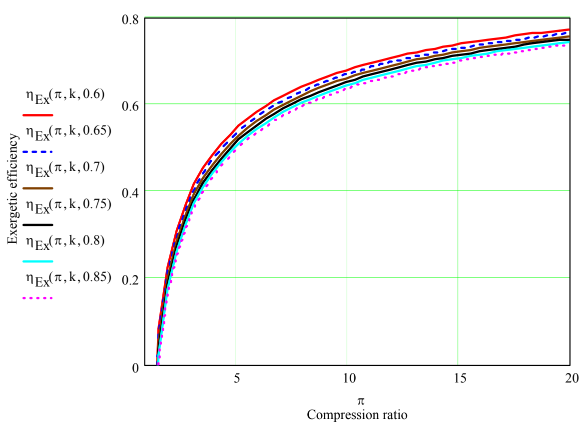

Fig 11 Exergetic efficiency for high pressure ratio values [1]

More impressive plot is the three dimensional presentation of the variation of exergetic efficiency, plotted in figure 12.

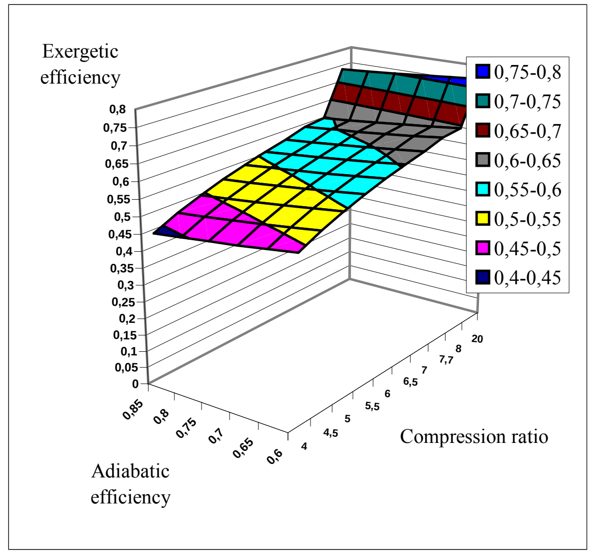

Fig 12 Variation of exergetic efficiency[1]

3 CONCLUSIONS

1. Calculus and analytical confirmation of results for the indicated power of Atlas-Copco GA-250 helical screw compressor with oil injection indicates that their values are within the limits stated at design (as presented in figure 4 and in the technical characteristics).
2. The equation for the calculus of indicated power of the equipment was developed using mathematical modeling of the working process and critical

4. Nomogram obtained and presented in figure 5 and 6 are important both in theory and practice, since using them assures a quick estimation of value of the intake power in distinct working conditions.
5. Equation (34) is original and is the contribution of the authors to the study of the exergetic efficiency of the helical screw compressors.
6. As, the adiabatic efficiency and the compression ratio raise, the losses as a function of the irreversibility of compression process π_{ir} amplifies. Examining the numerical values and the separate study of the value matrix of the ratio $\frac{\pi_{ir}}{l_t}$ as well as the specific work values and the losses due to irreversibility [1], the conclusion is: as, the adiabatic efficiency raises, the exergetic efficiency decreases;
7. For specific values of the parameters π, η_{ad} and k, study reveals that the exergetic efficiency decreases as the suction temperature raises.
8. Losses as a result of the irreversibility of the process can amplify, at the limit, when the exergy of the work is lost entirely due to irreversibility, then $\eta_{Ex} = 0$.
9. A separate study of the ratio $\frac{\pi_{ir}}{l_t}$ value matrix, the specific work and the losses due to irreversibility, reveals that l_t decreases more than π_{ir}, providing the explanation for the increase of the ratio $\frac{\pi_{ir}}{l_t}$ along the increase of the adiabatic

efficiency, and accordingly the exergetic efficiency decreases. Exergetic efficiency reaches the value $\eta_{Ex} = 1$ when $\pi_{ir} = 0$, for completely reversible processes.

10. Limits for the values of exergetic efficiency are between 0.438 ÷ 0.773, which are consistent with the actual working conditions. The value given for the compression ratio $\pi = 20$ is not a commonly used value, but it was used to illustrate in the plot the trend of the exergetic efficiency.

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