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REDUCTION OF ENERGY CONSUMPTION AND GREENHOUSE GAS EMISSIONS IN OIL INJECTED HELICALSCREW COMPRESSORS

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Abstract: *In order to have a real picture of the exergetic performances of the compressors and highlight the contribution of each loss to the exergetic yield, we performed a series of experimental determinations "in site", then based on a calculation model, we drew up the real hourly exergetic balances related to the KAESER compressor ESD 375. Based on the presented breviary, we approached a case study, used to justify energetically the feasibility of replacing old compressors, with many hours of operation, with compressors equipped with heat recuperators and compressed air dryers. In this way, the following were determined: the Carnot factor, the amount of heat given off, the exergy harnessed, the useful heat, the exergy supplied by the network, the increase in the efficiency of energy use, the annual operating time of the compressor, the annual energy economy, the production of GHG. In addition, the performed calculations have the role of justifying the installation of heat exchangers on compressors in pneumatic systems. In this work, the calculation breviary is presented on the basis of which the numerical calculation programs were made and then the effects of heat recovery in the two helical compressors with oil injection were quantified.*

Keywords: *energy efficiency, compressor, greenhouse gas emissions*

1. Introduction

Oil-injected screw compressors are widely used in industrial applications due to their efficiency and reliability. However, as global emphasis shifts toward sustainability and energy efficiency, reducing the energy consumption and greenhouse gas (GHG) emissions of these machines has become a key focus. Several strategies can be implemented to achieve these reductions, combining technological innovation with improved operational practices [1].

Oil-injected screw compressors operate using two helical rotors that compress air in a chamber. Oil is injected into the compression chamber to perform multiple functions:

- Lubrication: Reduces friction between the rotors, enhancing efficiency.
- Sealing: Improves sealing between rotors and the housing to prevent air leakage.
- Cooling: Absorbs heat generated during compression, lowering the temperature and reducing energy losses [2].

These compressors are often chosen for their durability and performance in demanding environments, but optimizing their energy use can significantly lower their environmental footprint [3].

Since energy consumption is closely tied to GHG emissions, the key to reducing emissions in oil-injected screw compressors lies in improving energy efficiency and utilizing clean energy sources.

Reducing the energy consumption and greenhouse gas emissions of oil-injected screw compressors can be achieved through a combination of technological advancements, better maintenance, and optimization strategies. [4, 5].

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2. Calculation summary for the exergetic balance of compressors [6, 7, 8, 9, 10, 11, 12, 13, 14]

Currently, energy is defined as the sum between two components with different transformation possibilities – exergy and anergy. Electrical energy and mechanical energy are considered equal to exergy. Thermal energy is composed of exergy plus anergy. Exergy has unlimited transformation capacity, while anergy (eg the energy of the external environment) has no transformation capacity. Since electrical energy, mechanical energy and thermal energy are simultaneously involved in electro-compressors, we considered that the exergetic balance provides a critical picture closer to reality. [15]

The exergetic balance establishes the equality between the exergy entered in the balance outline and the algebraic sum between the useful exergy and the exergy losses caused by the constructive imperfections (mechanical losses) and the imperfections of the thermodynamic processes (irreversibility). The calculation summary used in the calculation of the exergetic balance components is presented in the following.

The mechanical work on the compressor shaft is partially found in the useful effect, the rest serving to cover losses caused by external irreversibility related to heat transfer, at finite temperature differences, with the ambient environment:

$$\pi_e = \pi_{qc} + \pi_{qr} + \pi_{\Delta T} \tag{1}$$

and, respectively, the losses due to internal irreversibility, a consequence of the lamination of the gas on the flow path:

$$\pi_l = \pi_{la} + \pi_{lr} \tag{2}$$

The main notations used in the study are the following:

Pressure increase ratio:

$$H = \frac{p_2}{p_1} \tag{3}$$

Cylinder pressure increase ratio:

$$\beta = \frac{p_r}{p_a} = \frac{H}{(1-\psi_a)\cdot(1-\psi_r)} = \frac{H}{\psi}; \tag{4}$$

The relative pressure drop coefficient during suction:

$$\psi_a = \frac{\Delta p_a}{p_1} \tag{5}$$

Coefficient of relative pressure drop during discharge:

$$\psi_r = \frac{\Delta p_r}{p_r} \tag{6}$$

$\psi = (1-\psi_a)\cdot(1-\psi_r)$ - the global coefficient of pressure losses in the suction and discharge processes; [11, 12, 13, 14]

T_0 – ambient temperature;

$T_1 = T_a$ – air temperature at suction;

$T_2 = T_c$ – air temperature at the end of compression;

T_r – the temperature of the air leaving the cooler;

n_c - the polytropic compression exponent;

k - the adiabatic exponent

R – constant nature of air;

η_m - mechanical efficiency of the compressor;

η_{Ee} - the effective exergetic efficiency of the compressor;

φ_a – coefficient that takes into account the heat input in the suction process;

φ_r – coefficient that takes into account the heat loss in the discharge process.

Losses caused by the irreversibility of the work processes of the helical compressor were determined as follows:

- the loss caused by the lamination of the gas at suction:

$$\pi_{la} = R \cdot T_0 \cdot ln\frac{1}{1-\psi_a} \tag{7}$$

- the loss caused by the lamination of the gas during discharge:

$$\pi_{lr} = R \cdot T_0 \cdot ln\frac{1}{1-\psi_r} \tag{8}$$

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- the loss caused by the irreversibility of heat transfer at finite temperature difference during compression:

$$\pi_{qc} = \frac{R}{k-1} \cdot \frac{k-n_c}{n_c-1} \cdot \left(T_c - T_a - T_0 \cdot ln\frac{T_c}{1-T_a}\right) \tag{9}$$

- the loss caused by the irreversibility of heat transfer at a finite temperature difference during recharging:

$$\pi_{qr} = \frac{k}{k-1} \cdot R \cdot \left(T_c - T_r - T_0 \cdot ln\frac{T_c}{1-T_r}\right) \tag{10}$$

- the loss caused by the finite difference between the discharge and suction temperatures:

$$\pi_{\Delta T} = \frac{k}{k-1} \cdot R \cdot \left(T_2 - T_a - T_0 \cdot ln\frac{T_2}{T_a}\right) \tag{11}$$

We define the effective exergetic efficiency of the compressor as follows:

$$\eta_{Ee} = \eta_{Ee} \cdot \eta_m = 1 - \frac{\sum_{j=1}^{6}\pi_j}{|l_e|} = 1 - \sum_{j=1}^{6}\bar{\pi}_j \tag{12}$$

3. KAESER ESD 375 COMPRESSOR. Quantifications of heat recovery effects

The numerical calculation program based on the calculus breviary contains input data, data that were obtained based on "in site" measurements and data taken from the specialized literature [16, 17, 18, 19, 20, 21].

Input data:

$\psi_a = 0.0001$; $\psi_r = 0.0003$; $\eta m = 0.82$; $\phi_a = 1$; $\phi_r = 1{,}04$; $k = 1.4$; $R_g = 0.287\ kJ/(kg \cdot K)$;

$T_a = 282.5$ – aspiration temperature, K;

$T_0 = 282.5$ – environmental temperature, K;

$T_c = 383$ – temperature at the end of compression, K;

$p_a = 0.966$ – aspiration pressure, bar;

$p_c = 8.1$ – compression pressure, bar;

$Q = 1867\ m_N^3/h$;

$$n_c = \frac{1}{1-\frac{ln\left(\frac{T_a}{T_c}\right)}{ln\left(\frac{p_a}{p_c}\right)}},\ n_c = 1.167$$

$$\Psi = (1-\psi_a)(1-\psi_r),\ \Psi = 1;$$

$$H_m = \frac{p_c}{p_a},\ H_m = 8.385$$

$$\beta = \frac{H_m}{\Psi},\ \beta = 8.388$$

$$T_a = \phi_a \cdot T_0, T_a = 282.5\ \text{K}$$

$$T_c = T_a \cdot \beta^{\frac{(n_c-1)}{n_c}};\ T_c = 383.022\ \text{K}$$

$$T_r = \frac{T_c}{\phi_r},\ T_r = 368.29\ \text{K}$$

Using the relationships presented in the calculus breviary and developing a numerical calculation program, we calculated the energies that enter the balance sheet, the useful energy and the energy losses. These are presented synthetically, numerically and percentage in table 1 and represented graphically in fig. 1.

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Table 1. Summary table for the real hourly exergy balance of the analyzed compressor – Kaeser esd 375 [average load]

EXERGY ENTERED INTO THE CONTOUR			**EXERGY OUT OF THE CONTOUR**		
Name	kWh	%	Name	kWh	%
The exergy supplied from the network to the compressor assembly	172.55	100	**USEFUL EXERGY**		
			Useful exergy of compressed air	111.44	64.58
			Useful exergy of the cooling system	7.81	4.53
			TOTAL USEFUL EXERGY	119.25	69.11
			LOST EXERGY		
			Losses through suction lamination	1.07	0.62
			Losses due to lamination during discharge	1.02	0.59
			Heat loss during compression	15.61	9.05
			Heat losses during discharge	3.27	1.90
			Heat losses in isobaric cooling	9.82	5.69
			Mechanical losses	4.05	2.35
			Mechanical losses in the cooling system	0.99	0.57
			Fluid losses in the cooling system	0.86	0.50
			Volume losses in the cooling system	0.69	0.40
			Copper losses when operating the cooling system	1.51	0.88
			Iron losses when operating the cooling system	0.99	0.57
			Mechanical losses when operating the cooling system	0.47	0.27
			Copper losses in the compressor motor	4.37	2.53
			Iron losses in the compressor motor	5.20	3.01
			Mechanical losses	3.38	1.96
			TOTAL EXERGY LOST	53.30	30.89
TOTAL	172.55	100	TOTAL	172.55	100

Next, we performed a quantification of the effects of heat recovery on the KAESER ESD 375 compressor:

Exergy losses with the heat given off during compression – 15.61 kWh
Exergy losses with heat given up in isobaric cooling – 9.82 kWh
Exergy losses with the heat given off during recharging – 3.27 kWh

Total exergy losses with heat given up Ex = 28.70 kWh

The temperature of the compressed air entering the heat recovery system: T_{ica} = 95 ^{0}C = 368 K
Water temperature at the entrance to the heat recovery system: T_{iwater} = 45 ^{0}C = 318 K
Water temperature at the exit of the heat recovery system: T_{ewater} = 75 ^{0}C

The Carnot factor, calculation for the amount of heat related to the heat exergy [22, 23]:
$Fc = 1 - (T_{iwater}/T_{ica}) = 1 - (318/368) = 1 – 0.864 = 0.136$
The relationship between heat exergy and heat quantity $Ex = Q \cdot Fc$
The amount of heat given off $Q = Ex/Fc = 28.70/0.136 = 211$ $kWh_{thermal}$
Taking into account the thermal resistances of the heat exchanger and pipes, the transfer efficiency is Rt=0.80
Harnessed exergy Ex.val. = Rt · Ex = 0.80 · 28.70 = 22.96 kWh
Useful heat $Q_{useful} = Q \cdot Rt = 211 \cdot 0.8 = 169$ $kWh_{thermal}$
Exergy supplied from the network: Ex. entered = 172.55 kWh

Increasing the efficiency of energy use – 13.30 %
(ΔEfic.ut.en. = Ex.val./Ex.entered = 22.96/172.55 = 0.133).
Annual operating time of the compressor [24]: 4500 hours/year
Annual energy savings – 4500 ore/ year · 22.96 kWh/h = 103320 kWh/ year = 103.320 MWh/ year
1 MWh coal-fired power plant product [25] = 850 kg CO_2
GES production= 103.320 · 850 = 87822 kg CO_2/year = 87.822 tone CO_2/year

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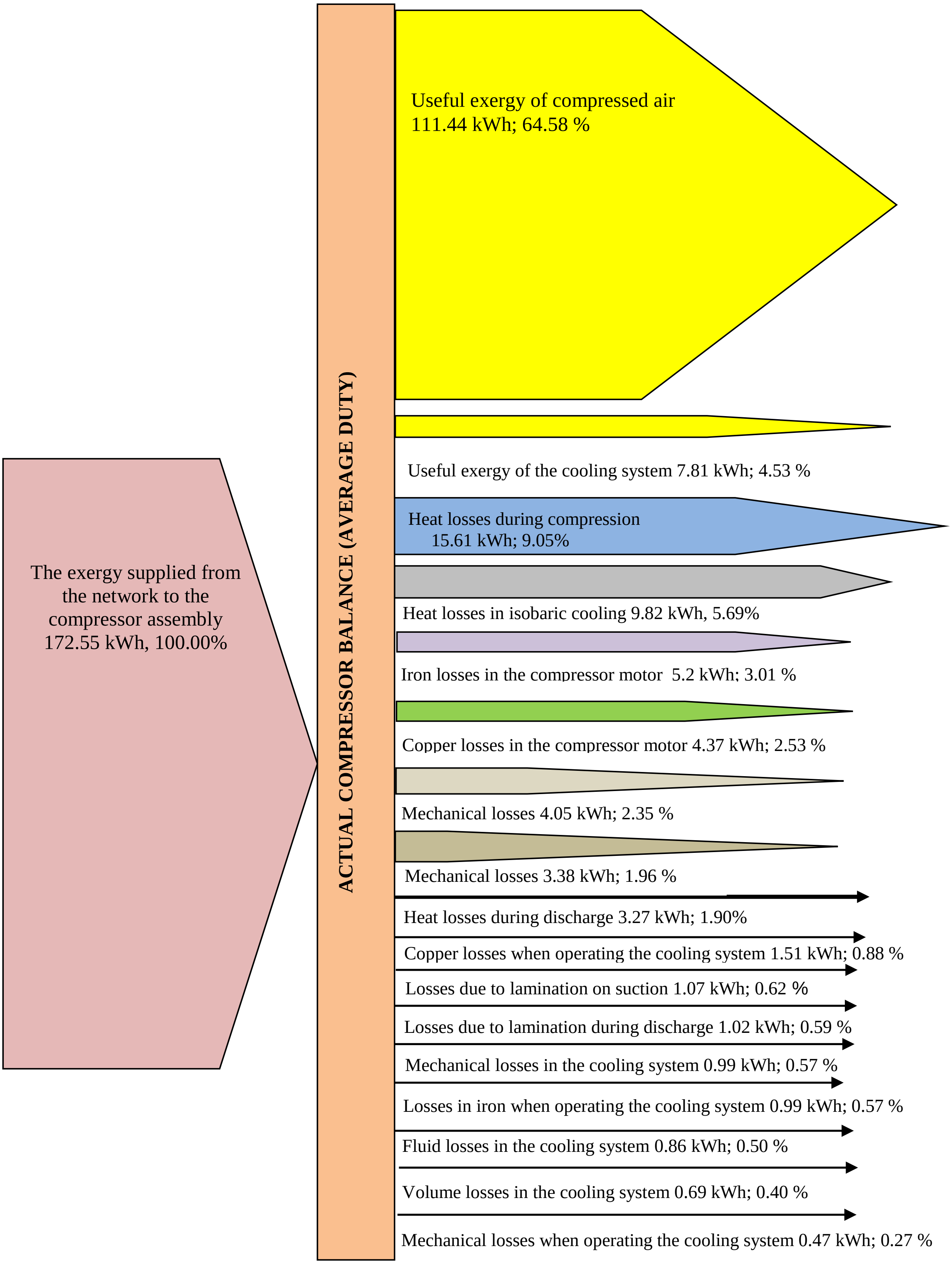

Fig. 1. Sankey Diagram

4. Conclusions

The justification for making these calculations responds to some assumed targets:

- reduction of final energy consumption;
- reduction of greenhouse gas emissions.

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The measure integrates into the principles of the circular economy (utilization of recovered heat, in other processes), contributing to the modernization and increase of the efficiency of the analysed energy systems.

The recovery and valorisation of a secondary energy resource contributes to the achievement of some components of the ecological transition, energy transition and sustainable development (economic efficiency, ecological responsibility and social solidarity).

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