SCREW COMPRESSORS USED FOR THE PRODUCTION OF COMPRESSED AIR INSIDE A COAL MINE

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ABSTRACT: *The present paper highlights that he technical measures destined to increase the energetic efficiency are feasible and profitable, the first steps consisting in the implementation of a series of measures to organise and meter the consumption and to assess the cost of compressed air followed by the development of the consumption management structures and investments. The energetic efficiency of pneumatic systems is low, the case studies highlighting the possibility to increase it as long as a series of technical solutions are implemented allowing thus energetic economies of 5 ÷ 50 %. Unfortunately the actual community market conditions and the decision mechanisms do not allow the practical application of such an important energy saving potential. The use of pneumatic energy in industry as well as in the tertiary system is quite frequent representing 10% of the industrial electricity consumption in the European Union. The working method is constituted by the creation of hourly real and optimum exergy flow diagrams for the compressors being used.*

KEY WORDS: exergy flow diagrams, compressed air, pneumatic systems.

1. INTRODUCTION

The constructive characteristics of helical screw compressors

Helical screw compressors are comprised by the helical screw compressors category. These machines represent a balanced operation with isothermal efficiencies and superior flow coefficients compared to other types of compressors. There is a wide range constructive and functional types of screw compressors, therefore their number has considerably increased during the past few years. They operate with a variety of fluids which may be gas, dry vapors or multiphase mixtures. The screw compressor is unable to have a dry operation (without any oil injection), with oil injection or using other fluids during gas compression. In order to obtain a maximum value for its efficiency a special design and operation is therefore imposed.

The screw compressor is composed of two rotors, a leading one and a driven one. The rotors of the screw compressor present a number of lobes, each pair of lobes creating a helical channel. The choice of the number of lobes depends on the necessary flowing surface.

2. The energetic balance for the Atlas Copco compressor system

Average values of the experimentally determined thermos-fluidic parameters

ATLAS COPCO GA 250 type compressor

ψ a := 0.15

ψ r:=0.09
η m :=0.84
ϕa := 1.05
ϕr :=1.04
k :=1.4
Rg : = 0.287
kJ/kg*K
Ta:= 283.5K - temperature during aspiration
T0: = 287,3K - environmental temperature
Tc: = 415K - temperature at the end of compression
Tr:= 348K - outlet temperature
pa: = 0.934 - aspiration temperature, bar
pc := 6.3 - compression temperature, bar
Q:= 2337.5
m_N^3/h

$$nc := \frac{1}{1-\frac{\ln\left(\frac{Ta}{Tc}\right)}{\ln\left(\frac{pa}{pc}\right)}}$$

nc = 1.249
Ψ:= (1 - ψa) (1 - ψr)
Ψ = 0.773

$$Hm := \frac{pc}{pa}$$

Hm = 6.745

$$\beta := \frac{Hm}{\Psi}$$

β = 8.72
Ta = ϕa · T0
Ta = 301.665K

$$Tc := Ta \cdot \beta^{\frac{(nc-1)}{nc}}$$

Tc = 464.822K

$$Tr := \frac{Tc}{\phi r}$$

Tr := 446.945K
1T0 := Rg·T0·ln (Hm)
1T0 = 157.3926 kJ/kg

It has proceeded therefore to the determination of the losses caused by the irreversibility of the operation process of the screw compressor, as follows:

- The loss caused by the lamination of the gas in the aspiration valve:

$$\pi_{la} = R \cdot T_0 \cdot \ln\frac{1}{1-\psi_a} \qquad \textbf{(1)}$$

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

$$\pi_{lr} = R \cdot T_0 \cdot \ln\frac{1}{1-\psi_r} \qquad \textbf{(2)}$$

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

$$\pi_{qc} = \frac{R}{k-1}\cdot\frac{k-n_c}{n_c-1}\cdot\left(T_c - T_a - T_0\ln\frac{T_c}{T_a}\right) \textbf{(3)}$$

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

$$\pi_{qr} = \frac{k}{k-1}\cdot R\cdot\left(T_c - T_r - T_0\ln\frac{T_c}{T_r}\right) \quad \textbf{(4)}$$

- The loss caused by the finite difference between the discharge and respectively the aspiration temperature:

$$\pi_{\Delta T} = \frac{k}{k-1}\cdot R\cdot\left(T_2 - T_a - T_0\ln\frac{T_2}{T_a}\right) \textbf{(5)}$$

The effective exergy efficiency of the screw compressor may be expressed as follows:

$$\eta_{Ee} = \eta_{Ee}\cdot\eta_m = 1 - \frac{\sum_{j=1}^{6}\pi_j}{|l_e|} = 1 - \sum_{j=1}^{6}\grave{\pi}_j \ (6)$$

Table 1. The real hourly exergy balance of the ATLAS COPCO GA-250 screw compressor system

EXERGY ENTERED INTO THE OUTLINE			**OUTLINE OUTPUT EXERGY**		
Name	kWh	%	Name	kWh	%
Exergy supplied from the network to the motor of the compressor	252.79	96.13	**USEFUL EXERGY**		
			Useful exergy of compressed air	132.14	50.25
			Useful exergy of cooling air	5.796	2.20
			Total useful exergy	137.936	52.45
			LOST EXERGY		

			Losses through lamination during aspiration π_{la}	11.25	4.28
			Losses through lamination during discharge π_{lr}	6.53	2.48
			Losses with the heat given off during compression π_{qc}	14.16	5.38
			Losses with the heat given off during discharge π_{qr}	5.57	2.12
			Losses with the heat given off during isobaric cooling $\pi_{\Delta T}$	27.27	10.37
			Mechanical losses π_m	21.8	8.29
			Losses due to air humidity	15.71	5.97
Exergy supplied from the network to the motors of the ventilators	10.18	3.87	Ventilator mechanical losses p_{mv}	0.696	0.26
			Ventilator fluidic losses p_{fv}	1.294	0.49
			Ventilator volume losses p_{vv}	0.552	0.22
			Losses in copper at the ventilator motor $p_{Cu\,v}$	1.1	0.42
			Losses in iron at the ventilator motor $p_{Fe\,v}$	0.52	0.21
			Mechanical losses at the motor of the ventilator p_{mv}	0.22	0.08
			Losses in copper at the motor of the compressor $p_{Cu\,c}$	7.402	2.81
			Losses in iron at the motor of the compressor $p_{Fe\,c}$	5.96	2.26
			Mechanical losses p_{mc}	5	1.91
			Total losses	125.032	47.55
TOTAL	**262.97**	**100**	**TOTAL**	**262.97**	**100**

Table 2. The hourly optimum exergy balance of the ATLAS COPCO GA-250 screw compressor system

EXERGY ENTERED INTO THE OUTLINE			**OUTLINE OUTPUT EXERGY**		
Name	kWh	%	Name	kWh	%
			USEFUL EXERGY		
Exergy supplied from the network to the motor of the compressor	207.33	95.32	Useful exergy of compressed air	132.14	60.75
			Useful exergy of cooling air	5.796	2.66
			Total useful exergy	137.936	63.41
			LOST EXERGY		
			Losses through lamination during aspiration π_{la}	11.25	5.17
			Losses through lamination during discharge π_{lr}	6.53	3

			Losses with the heat given off during compression π_{qc}	2.832	1.31
			Losses with the heat given off during discharge π_{qr}	1.114	0.52
			Losses with the heat given off during isobaric cooling $\pi_{\Delta T}$	5.454	2.52
			Mechanical losses π_m	21.8	10.02
			Losses due to air humidity	7.85	3.62
Exergy supplied from the network to the motors of the ventilators	10.18	4.68	Ventilator mechanical losses p_{mv}	0.696	0.32
			Ventilator fluidic losses p_{fv}	1.294	0.59
			Ventilator volume losses p_{vv}	0.552	0.25
			Losses in copper at the ventilator motor $p_{Cu\,v}$	1.1	0.51
			Losses in iron at the ventilator motor $p_{Fe\,v}$	0.52	0.23
			Mechanical losses at the motor of the ventilator p_{mv}	0.22	0.1
			Losses in copper at the motor of the compressor $p_{Cu\,c}$	7.402	3.4
			Losses in iron at the motor of the compressor $p_{Fe\,c}$	5.96	2.74
			Mechanical losses p_{mc}	5	2.29
			Total losses	79.574	36.59
TOTAL	**217.51**	**100**	**TOTAL**	**217.51**	**100**

Table 3. Revision table for the real hourly exergetic balance of the INGERSOLL RAND SSR-250 MV screw compressor unit

EXERGY ENTERED INTO THE OUTLINE			**OUTLINE OUTPUT EXERGY**		
Name	kWh	%	Name	kWh	%
Exergy supplied from the network to the motor of the compressor	283.46	94.63	**USEFUL EXERGY**		
			Useful exergy of compressed air	127.49	42.80
			Useful exergy of cooling air	8.622	2.89
			Total useful exergy	136.112	45.69
			LOST EXERGY		
			Losses through lamination during suction π_{la}	18.54	6.22
			Losses through lamination during discharge π_{lr}	8.64	2.90
			Losses with the heat given off during compression π_{qc}	14.75	4.95
			Losses with the heat given off during discharge π_{qr}	4.61	1.55

			Losses with the heat given off during isobaric cooling $\pi_{\Delta T}$	33.46	11.23
			Mechanical losses π_m	30.06	10.09
			Losses due to air humidity	21.81	7.32
Exergy supplied from the network to the motors of the ventilators	14.475	5.37	Ventilator mechanical losses p_{mv}	0.906	0.3
			Ventilator fluidic losses p_{fv}	1.696	0.57
			Ventilator volume losses p_{vv}	0.726	0.25
			Losses in copper at the motor of the ventilator $p_{Cu\,v}$	1.405	0.47
			Losses in iron at the motor of the ventilator $p_{Fe\,v}$	0.82	0.27
			Mechanical losses at the motor of the ventilator p_{mv}	0.3	0.11
			Losses in copper at the motor of the compressor $p_{Cu\,c}$	7.36	2.47
			Losses in iron at the motor of the compressor $p_{Fe\,c}$	9.5	3.18
			Mechanical losses p_{mc}	7.24	2.43
			Total losses	161.823	54.31
Total	**14.475**	**100**	**Total**	**297.935**	**100**

Table 4. Revision table for the optimum hourly exergetic balance of the INGERSOLL RAND SSR-250 MV screw compressor unit

EXERGY ENTERED INTO THE OUTLINE			**OUTLINE OUTPUT EXERGY**		
Name	kWh	%	Name	kWh	%
Exergy supplied from the network to the motor of the compressor	230.329	94.08	**USEFUL EXERGY**		
			Useful exergy of compressed air	127.49	52.08
			Useful exergy of cooling air	8.622	3.52
			Total useful exergy	136.112	55.6
			LOST EXERGY		
			Losses through lamination during suction π_{la}	18.54	7.57
			Losses through lamination during discharge π_{lr}	8.64	3.53
			Losses with the heat given off during compression π_{qc}	2.95	1.2
			Losses with the heat given off during discharge π_{qr}	0.93	0.38
			Losses with the heat given off during isobaric cooling $\pi_{\Delta T}$	6.702	2.74
			Mechanical losses π_m	30.061	12.29
			Losses due to air humidity	10.912	4.46
Exergy supplied from the network	14.475	5.92	Ventilator mechanical losses p_{mv}	0.907	0.38
			Ventilator fluidic losses p_{fv}	1.697	0.69

to the motors of the ventilators			Ventilator volume losses p_{vv}	0.727	0.29
			Losses in copper at the motor of the ventilator $p_{Cu\,v}$	1.406	0.58
			Losses in iron at the motor of the ventilator $p_{Fe\,v}$	0.82	0.33
			Mechanical losses at the motor of the ventilator p_{mv}	0.3	0.12
			Losses in copper at the motor of the compressor $p_{Cu\,c}$	7.36	3
			Losses in iron at the motor of the compressor $p_{Fe\,c}$	9.5	3.88
			Mechanical losses p_{mc}	7.24	2.96
			Total losses	108.692	44.4
Total	**14.475**	**100**	**Total**	**244.804**	**100**

3. CONCLUSION

Although the demand of compressed air at Lonea Coal Mine is ensured by modern efficient compressors manufactured by famous companies (i.e. ATLAS COPCO, INGERSOLL RAND), there are still countless possibilities to increase their energetic efficiency through:

- The parallel operation of low voltage transformers on a load curve based on minimum power losses depending on the variation of the load;
- The recovery of heat given off through the cooling of compressed air;
- Actions to eliminate the humidity of discharged compressed air;
- Ensuring the creation of a compressed air consumption chronogram in order to flatten the load curve;
- The introduction of electronic control panels for the operation of compressors fitted with variable revolution motors (VSD);

Although all the measures proposed for the improvement of energetic efficiency are feasible and cost effective, the companies are not interested in applying them due to a series of organisational deficiencies such as:

- The complexity of the structures for the management of energy consumption
- The lack of metering devices for the consumption and respectively the cost of compressed air;
- The lack of information regarding possible economies;

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