POWER DEMAND OF THE TK-R 16/8 TURBOCOMPRESSOR FUNCTION OF INLET CONDITIONS

ION DOSA[1], OVIDIU BOGDAN TOMUȘ[2]

Abstract: There are indications in literature regarding the possibility of computing the power demand for the turbocompressor depending on variation of inlet air parameters. The formulas found can be applied only for small variations of inlet air parameters, and their precision is poor. This work presents the results obtained by considering a statistical dependency between the variation of power demand and the variation of inlet air parameters for "TK-R 16/8 Iolanda II" turbocompressor.

Keywords: inlet air parameters, power demand, tubocompressor, statistical dependency

1. INTRODUCTION

Parameters of inlet air are function of season and the elevation against the sea level. Air temperature and pressure can be computed using equations found in literature, [Irimie et al. 1994], and they are valid for a gaseous system in equilibrium. Experimental data shows that temperature is dropping at a rate of 1°C at every 154 m, and the value of polytropic exponent is n=1.002, [Irimie et al. 1994].

The power demand of the turbo compressor knowing the power demand for conditions denoted with index 1, and the power demand denoted with 2, can be computed for small variation of inlet air flow rate, using equation below, [Bacu et al. 1972, Brădeanu 1977]:

$$\frac{P_{a2}}{P_{a1}} = \frac{V_{a2}}{V_{a1}} \tag{1}$$

[1] *Assoc.prof., Ph.D. Eng., University of Petroşani, i_dosa@hotmail.com*
[2] *Lecturer, Ph.D. Eng., University of Petroșani, tobogdan2002@yahoo.com*

where P_{a1} is the power demand, [W], corresponding to V_{a1} inlet air flow rate, [$m^3 \cdot s^{-1}$], and P_{a2} is the power demand corresponding to V_{a2} inlet air flow rate.

Based on equation above a direct conclusion is that for different inlet conditions which influences the volumetric flow rate, the power demand of the turbocompressor will be different than the one provided by the manufacturer.

A proper assessment of power demand of the turbocompressor as function of inlet air parameters an equation must be found that has to express with good accuracy the effect of the variation of the environmental factors on the compressor power demand.

Variance of atmospheric air parameters can cause large differences in the inlet air density [Dosa 1998], so the use of the relation (1) valid for small variations of the inlet air flow rate [Brădeanu 1972] or density, can induce significant errors in the assessment of power demand.

2. POWER DEMAND OF THE TURBOCOMPRESSOR

In order to express the most important factors that influence the power demand of the turbocompressor, the power demand is computed using equation below as a function of mass air flow rate at inlet m_{asp}, inlet stagnation temperature T^*_{asp}, stagnation pressure at inlet and outlet p^*_{asp}, p^*_{ref} and the polytropic exponent of compression, n [Brădeanu 1972]:

$$P_{act} = \frac{n}{n-1} \cdot \frac{R \cdot m_{asp} \cdot T^*_{asp} \cdot \left[\left(\frac{p^*_{ref}}{p^*_{asp}} \right)^{\frac{n-1}{n}} - 1 \right]}{\eta_c \cdot \eta_{tr} \cdot \eta_m \cdot 10^3} \text{ [kW]} \tag{2}$$

where: R - gas constant for air, [$J \cdot mol^{-1} \cdot K^{-1}$], η_c - the isentropic efficiency of the turbocompressor, η_{tr} - the gear efficiency, η_m - mechanical efficiency .

Applying the above equation to a cooled multistage turbocompressor is difficult due to the fact that it is impossible to accurately measure the air parameters through its evolution at different stages of the turbocompressor.

The polytropic exponent for the single-rotor turbocompressor without cooling can be calculated using equation found in literature [Pimsner 1988], where the unknown is l_f the work of friction.

Work of friction in different components of the compressor, [Pimsner 1988], is directly proportional to an experimental coefficient and velocity.

In case of turbocompressor the compressed air is being cooled between stages and a result equation (2) cannot be applied for the multistage turbocompressor.

Achieving different degrees of cooling for the same inlet conditions results in a change of the specific volume of the compressed air that is delivered in the next stage, so even in situations where inlet conditions are the same, the power required for compressing the air has different values depending on the degree of cooling achieved in the intermediate coolers.

Computing the work of friction in various components of the turbocompressor was made in the hypothesis of the separate action [Pimsner 1988] of the different types of losses.

In the absence of data on their interaction, the assumption of a strict dependence between pressure, temperature and power values cannot be accepted.

Therefore, it is considered that the link between the analyzed parameters expressed as an equation is uncertain, the dependence of the parameters being rather a statistical dependence.

3. STATISTICAL MODEL

In order to elaborate the statistical model, the simplifying assumptions above regarding the hypothesis of the separate action [Pimsner 1988] of different types of losses will be considered. Equation (2) that analytically analyses the dependence between power demand, pressure and suction temperature suggests the analysis of the separate influence of the two variables.

Since there is a relationship between the inlet pressure and temperature, the variables are not independent, as a result, in order to avoid errors due to the multicollinearity phenomenon, the two variables will be replaced by one, the density of inlet air.

An important consequence of the hypothesis regarding the value of the polytropic exponent is the number of independent variables measured and sorting of measured data.

Consequently, besides the inlet pressure and temperature, respectively the power demand, the following data will be measured: discharge pressure; inlet flow rate, which controls the friction, and consequently the value of the polytropic exponent; inlet and outlet temperature of the cooling water.

For steady water and compressed air flow rate, the difference between inlet temperature of the cooling water and the outlet temperature reflects the cooling conditions.

Equal temperature variances lead to the conclusion that there are similar cooling conditions, so it will be considered that the specific volume of air delivered in different stages remains the same.

Turbocompressor efficiency is considered constant for a given polytropic exponent, as it is the ratio between potential mechanical work and actual compression work.

Equally, drive speed, engine output, gear efficiency is also considered constant.

In conclusion, the data will be sorted according to the inlet flow rate and the increase of cooling water temperature, i.e. the difference between the water temperature at the inlet and outlet of the cooling circuit.

In the studied case, based on the results obtained with equation (1) for small variations of the flow [Brădeanu 1977], it can be stated that choosing linear correlation for the studied variables is the most appropriate.

The quantitative evaluation of the degree of correlation is made using the simple correlation coefficient [Constantinescu et al. 1980]:

$$R_{x,y} = \frac{n \cdot \sum_{i=1}^{n} x_i \cdot y_i - \left(\sum_{i=1}^{n} y_i\right) \cdot \left(\sum_{i=1}^{n} x_i\right)}{\sqrt{\left[n \cdot \sum_{i=1}^{n} x_i^2 - \left(\sum_{i=1}^{n} x_i\right)^2\right] \cdot \left[n \sum_{i=1}^{n} y_i^2 - \left(\sum_{i=1}^{n} y_i\right)^2\right]}} \tag{3}$$

where x_i, y_i are experimentally measured data sets, and n is the number of these sets.

Due to the random scattering of experimental data it is possible that the value of the simple sampling correlation coefficient is different from 0 even for two independent variables, which is why the significance of the value obtained for $R_{x,y}$ using equation [Constantinescu et al. 1980] must be verified:

$$H_{calc} = | R_{x,y} | \sqrt{n-1} \tag{4}$$

The value obtained compares with H_α the critical value of the simple correlation coefficient.

If calculated value $H_{calc} > H_\alpha$, then with the confidence level α we can say that the variables x and y are correlated, otherwise they are independent.

Removing gross errors can be done by calculating for all n sets of values x_i, y_i the sampling dispersion s^2_x, s^2_y and the correlation coefficient.

These values are computed for the $n-1$ sets remaining after removing the set denoted with x^*, y^* suspected to be aberrant. The resulting values are denoted by s^{*2}_x, s^{*2}_y and $R^*_{x,y}$, and the value of the equation [Constantinescu et al. 1980] is calculated:

$$R = \frac{(n-1)^2 \cdot \left(1 - R^{*2}_{x,y}\right) \cdot s^{*2}_x \cdot s^{*2}_y}{n^2 \cdot s^2_x \cdot s^2_y \cdot \left(1 - R^{*2}_{x,y}\right)} \tag{4}$$

The computed value is compared to the critical value R_α [4] for different confidence levels α.

If $R < R_\alpha$ then the sets of values are considered to be affected by gross errors with confidence level α and eliminated.

4. RESULTS AND CONCLUSIONS

In order to generate the regression equation, 124 measurements were made [Dosa 1998].

The barometric pressure *B*, the ambient temperature T_a, the discharge pressure of the turbocompressor p_{ref}, the drive power demand P_{act} at the motor terminals, the cooling water temperature in the cooling circuit at inlet and outlet T_{int} and T_{ies} were measured.

Resulting data were processed using a program based on the algorithm described above, aiming to eliminate the data affected by gross errors, and finding the significance of the calculated simple correlation coefficient [Constantinescu et al. 1988] [Dosa 1998].

In Fig. 1 are presented the power demand values calculated with the equation (1) (denoted by *x*) considering the power P_{a1} = 1,95 MW and the density ρ_1 = 1.165 kg·m^{-3} corresponding to the rated inlet conditions.

Large differences between the two sets of values are observed, which justifies the use of regression equations obtained on the basis of experimental data to assess the energy effects of variation of environmental factors.

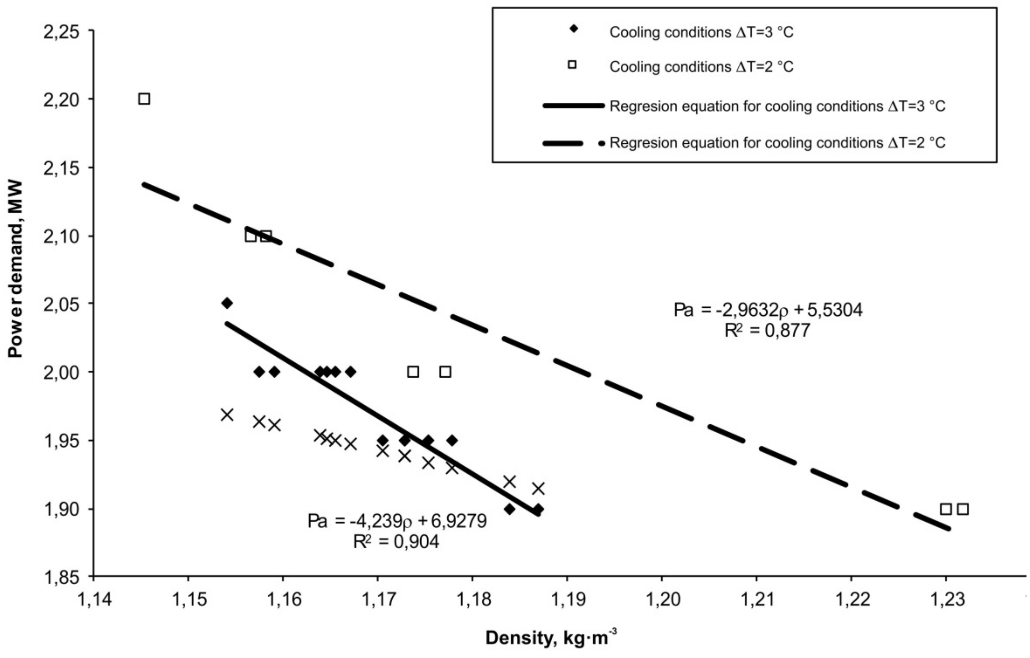

Fig. 1. Variance of power demand at a flow rate of 290.67 m^3·min^{-1}.

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