INVESTIGATION OF COMPRESSION POWER IN A THREE-STAGE CENTRIFUGAL COMPRESSOR BY THERMODYNAMIC MODELING AND NEURAL NETWORK MODELING ALGORITHM

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ABSTRACT

In this research, a three-stage centrifugal compressor with two intercoolers has been studied. According to experimental data which were taken every moment in the control room, the compressor has been modeled by using neural network modeling Algorithm and governing thermodynamic equations. Neural network modeling with accuracy of 99\% was used to examine the effect of air relative humidity and volumetric flow rate on air compressor performance. It was found that although air relative humidity causes to erosion and corrosion on the internal parts of compressor, but it has no effect on the compression power and any increase in air volumetric flow rate will be increase the compression power. According to governing thermodynamic equations, the effect of inlet air temperature to every compression stages was studied. It was found that the specific compression power will linearly as a result of any increase in inlet air temperature. The specific compression power in summer in first, second and third stages of compressor are respectively about 10\%, 6.07\% and 4.39\% more than winter. Finally the use of evaporative cooling system has been proposed in order to reduce compression power in summer, by this method the compressor compression power can be reduced up to 5\%.

KEYWORDS: Centrifugal compressor, thermodynamic modeling, relative humidity, air temperature, volumetric flow rate

The main air compressor which is studied in this article is located in Fajr Petrochemical Company of Iran. The type of Subjected Compressor is three-stage centrifugal compressor with two intercoolers that are mounted between the stages and a cooler mounted after stage three. At first, the air passes through inlet air filter. The inlet air filter consists of two high efficiency filters and pre-filter pads. Referring to the document of the equipment, it can be found that acceptable air pressure loss (pressure drop) is about 100 mm of water (10 milli bar) while air passes the filter. Inlet air filters can be washed by water, if after washing, pressure drop is greater than 5 milli bar, the filter should be replaced. Compressed air after leaving each stage, while passing through the intercoolers, is cooled by cooling water and enters the next stage of Compression. Types of intercoolers are shell and tube with floating head. Electromotor Consumption is 2850 kW and rotational speed is 2980 rpm. Inlet air volumetric Flow rate the compressor at design conditions is 25,291 normal cubic meters per hour. To avoid temperature increases, the air is cooled twice while passing compressor, between stage one and stage two and three. Finally the air exits the compressor with pressure of 6/6 bar and a temperature of about 126 °C. Ki Wook Song et.al. In 2010, worked on thermodynamic and experimental modeling of a multi-stage centrifugal compressor in order to increase the efficiency and optimization of its energy consumption and finally they could reduce energy consumption in the mentioned compressor for 5\%. Claus Hansen et.al (2008) in 2008 worked on design and modeling of centrifugal compressor by Haysy software. Tarek Abdel Salem et.al (2007) in 2007 worked on thermodynamic simulation of a multi-stage centrifugal compressor by experimental data and the equations (efficiency, pressure ratio, etc.) and compared it with computer modeling. Bosel (2009) in 2009 compared the thermodynamic analysis of single-stage and multi-stage compressors and discussed about theirs efficiency and energy. Seriar et.al (1997) in 2008 involved in the accurate thermodynamic modeling of a cooling compressor using experimental data and related equations, to develop and improve efficiency of the compressor.

THERMODYNAMIC BEHAVIOR OF THE COMPRESSOR SUBJECT OF STUDY

In this study, the thermodynamic behavior of equipment, in both summer and winter, including the most thermodynamic ranges, was considered. The minimum and maximum values for each parameter are indicated on the chart. Consumption power, inlet air volumetric Flow rate, relative humidity and inlet air density of the compressor used in this study are shown in Figures 1, 2, 3 and 4.
**Figure 1.** Power consumption of the compressor subject of study

**Figure 2.** Inlet air volumetric Flow rate of compressor

**Figure 3.** Inlet air relative humidity of compressor
2.1. First stage of compressor

For first stage of Compression, ranges of temperature and pressure of the inlet and outlet air the compressor are shown in Figures 5 and 6:

2.2. First intercooler

Figure 4. Inlet air density of compressor

Figure 5. Inlet and outlet air temperature of first stage

Figure 6. Inlet and outlet air pressure of first stage
As it is shown in Figures 7, 8 and 9, for the first intercooler, the inlet and outlet air temperature and pressure of intercooler and the inlet and outlet water temperature of intercooler were studied.

**Figure 7.** inlet and outlet air temperature of first intercooler

**Figure 8.** inlet and outlet air pressure of first intercooler

**Figure 9.** inlet and outlet water temperature of first intercooler
2.3. *second stage of compressor*

For the second stage of Compression, similar to the first stage, ranges of temperature and pressure of the inlet and outlet air the compressor were reviewed and they are shown in Figures 10 and 11:

![Figure 10. inlet and outlet air temperature of second stage](image1)

![Figure 11. inlet and outlet air pressure of second stage](image2)

2.4. *second intercooler*

In the second intercooler, similar to first intercooler, the inlet and outlet air temperature and pressure of intercooler and the inlet and outlet water temperature of intercooler were investigated (Figs. 12,13,14).
Figure 12. inlet and outlet air temperature of second intercooler

Figure 13. inlet and outlet air pressure of second intercooler

Figure 14. inlet and outlet water temperature of second intercooler

2.5. third stage of compressor
As shown in Figures 15 and 16, for the third stage of Compression similar to the first stage, ranges of temperature and pressure of the inlet and outlet air the compressor have been investigated:

![Figure 15. inlet and outlet air temperature of third stage](image1)
![Figure 16. inlet and outlet air pressure of third stage](image2)

3. Governing equation

The governing equations in this study are:

3.1. The governing equations of compression

Thermodynamics rules that used to calculate the power and efficiency of the compressors are independent of the compressor performance. The compressor can be considered as one control volume by applying these rules. According to first thermodynamics rule in the compressor, energy equation can be written as relation 1:

$$\dot{m}$$

(1)

In the centrifugal compressors, due to small surface of heat transferring, in comparison with other energy terms, the heat transfer can be neglected in equation (1). Also the terms of the potential difference on both sides of this equation can be neglected. According to the principle of mass conservation and if we assume that the $h + \frac{v^2}{2} = h_t$ equals to static enthalpy, then Compression power of a compressor equals to gradient in the fluid enthalpy from entering to exiting the compressor.

Now, by considering the air as an ideal gas, the enthalpy gradient in an ideal gas is only a function of temperature, the compressor compression power can be calculated as follows:
\[ W_{12} = \dot{m}(h_{t2} - h_{t1}) = \dot{m}(C_{p,ha}(T_2)T_2 - C_{p,ha}(T_1)T_1) \]  

(2)

It is assumed that the air humidity is constant during the air compression process due to a dryer system which is mounted between the stages of the compressor. The dryer system can control the air humidity and keep the content of humidity to be within normal range. On the other hand humidity content of the inlet air of the first stage is similar to second and third stages. Compression power of the compressor can be calculated regarding to equations 1 & 2 and inlet air temperature of the second and third stages:

\[ W = \dot{m}(h_{t6} - h_{t5}) = \dot{m}(C_{p,ha}(T_6)T_6 - C_{p,ha}(T_5)T_5) \]

(3)

3.2. enthalpy governing equation considering the effect of humidity

In the air, there is always some water vapor. The amount of water vapor in the air is defined by the vapor pressure of the air. Vapor pressure in the air, cannot be greater than the saturated vapor pressure because in this case, the water must be solid or liquid in air.

Relative humidity expresses the relationship between and the pressure of water vapor in the air and the pressure of saturated water vapor at the same temperature. Relative humidity can be calculated using the following equation 5.

\[ \phi \]  

(5)

In order to assess the influence of relative humidity, saturated water vapor should be calculated. In many references approximate relationship is used to calculate saturated water vapor pressure. To do this, the saturated vapor pressure can be calculated by a regression exponential correlation for temperature range from 273.16 to 647.14 Kelvin, through Equation 6 in which saturated water vapor pressure will be calculated in terms of bar.

\[ P_{vap}(T) = P_{cr}e^{\frac{T}{T_{cr}}[a_1 + a_2T + a_3T^2 + a_4T^3 + a_5T^4 + a_6T^5]} \]

\[ P_{cr} = 220 \text{ (bar)} \]

\[ \tau = 1 - \frac{T_1}{T_{cr}} = 647/14(k) \]

\[ a_1 = -7.85823a_2 = 1.83991a_3 = -11.7811a_4 = 22.6705 \]

\[ a_5 = -15.9393a_6 = 1.77516 \]

As the air is considered as an ideal gas, the water vapor in the air can also be considered as an ideal gas. To calculate the constant pressure heat capacity of water vapor, the equation 7 is used to establish the 200 (k) < T < 1800(k) temperature range.

\[ C \]

\[ a_w = 1.80768b_w = 17.9273E - 6 \]

\[ c_w = 68.0617E - 8d_w = -22.443E - 11 \]

The air is considered as a combination of dry and water vapor. Therefore, to calculate the constant pressure heat capacity of air, constant pressure heat capacity of dry air should also be considered.
To calculate the heat capacity of humid air, mole fraction of water vapor should be calculated. Then heat capacity of humid air is achieved:

\[ \frac{X_{\text{vap}}}{X_{\text{d}}} = \frac{P_{\text{vap}}}{P_{1}} = \frac{\phi_{\text{vap}}}{\phi_{1}} \quad (9) \]

\[ C_{p} \quad (10) \]

### 3.3. Isentropic governing equation

It is assumed that the compressor is a turbo machine; its function can be considered adiabatic. Based on conventional governing thermodynamic equations, the compressor:

\[ \eta \quad (11) \]

\( T_{1} \) is the inlet air compressor temperature and \( T_{2} \) is the outlet air compressor temperatures are the same. On the other hand, outlet air temperature is calculated from

\[ \frac{T_{2}}{T_{1}} (12) \]

\( P_{1} \) is the inlet air compressor pressure and \( P_{2} \) is the outlet air compressor pressure. The remarkable thing about the 12 is that \( \frac{C_{p,ha}}{\phi_{1}} \) in the average temperature of inlet and outlet air compressor is calculated.

### 4. Modeling

This modeling is based on isentropic efficiency Equations and compression power of each stage. Parameters affecting isentropic efficiency in a centrifugal compressor are inlet air temperature, pressure ratio of compression and shaft speed [4] shaft speed is constant and air considered as an ideal gas, the pressure ratio is related to inlet air temperature. For thermodynamic modeling of each stage, apart from the effects of humidity due to its negligible impact [4], first isentropic efficiency of each compressor is calculated using related equations by taking specific heat coefficient at constant pressure in terms of air temperature and relative humidity of air, and the graph was extracted in terms of the inlet air temperature to each stage of Compression. According to the equations, the outlet air temperature of each stage is modeled in terms of isentropic efficiency of that stage, and the outlet temperature graph was drawn in terms of air inlet temperature and compared with experimental data. Compression power of each stage was modeled by thermodynamic equations in terms of the inlet air temperature of that stage and at a constant speed per minutes and its graph was drawn and compared with its actual experimental values. Neural network modeling was performed to study the effect of the air relative humidity and the effect of inlet air volumetric Flow rate of each stage and the effect of inlet water temperature of intercoolers on the power of compression. Accordingly, Data like temperature and pressure of the inlet and outlet air of each stage, temperature of inlet and outlet water of the first and second intercoolers and inlet air volumetric Flow rate were given to the neural network as input data and the desired output were obtained from the most precise neural network. As shown in Figure 17, in this simulation, the neural network composed of four layers, 3 hidden layers (with 50, 100, 11 neurons) and an output layer (10 neurons). The regression of the mentioned neural network is shown in Figure 18 with an accuracy of 0.99.
5. Results

Based on experimental data, thermodynamic modeling and modeling by neural network algorithm, the following results were obtained:

- Effect of inlet air volumetric flow rate on the compression power was also modeled using neural networks. As shown in Figure 20, with increasing inlet air volumetric flow rate, compression power increases as well.

- Based on modeling using neural networks and data training, as shown in Figure 19, changes in the air relative humidity do not affect the compression power. It is also confirmed in the reference 4.

Figure 17. neuralnetwork modeling Algorithm

Figure 18. Regression of neuralnetwork modeling Algorithm

Figure 19. Effect of air relative humidity on compression power

- Effect of inlet air volumetric flow rate on the compression power was also modeled using neural networks. As shown in Figure 20, with increasing inlet air volumetric flow rate, compression power increases as well.
Based on the modeling, for each stage of Compression, figure of outlet air temperature can be drawn in terms of inlet air temperature. As shown in Figures 21, 22 and 23, by increasing the inlet air temperature of every stage, the inlet air temperature is always rising approximately with a constant slope. Modeling was performed with great precision and in the maximum value, in the first, second and third Compression, it showed error of 0.9, 0.68 and 0.79 % which is negligible.
Any change in temperature does not have any effect on compressor compression power [5] but in this case, the compressor compresses less air. Because the compressor always compresses a certain volume of air in a constant speed. As the temperature increases, air density decreases and mass flow rate of the compressed air reduces. Thus, thermodynamic modeling of the compressor’s compression power be determined by defining the ability of specific compression (compression power per unit mass flow rate) and by the governing thermodynamic equations in order to study the effect of inlet air temperature. As shown in Figures 24, 25 and 26, compression power on the actual and modeled state are very close and always with increasing temperature, specific compression power can be linearly increased. The largest amount of slip in the thermodynamic modeling for the first, second and third stages of Compression have been respectively 2.99, 3.54 and 2.88% that can be attributed to some reasons like assuming the air as perfect gas, and also neglecting inlet air mass flow rate in the modeling.
Figure 24. Stage 1, effect of inlet air temperature on compression power

Figure 25. Stage 2, effect of inlet air temperature on compression power

Figure 26. Stage 3, effect of inlet air temperature on compression power

6. Solution to improve the compressor performance

Fag evaporative cooling system works by spraying water using sprinklers which are inserted before the filter and cool the air before entering the filter. By this way, decrease in inlet air temperature can cause a significant reduction in compressor’s compression power and increase in its efficiency. Based on studies conducted on the method of cooling the inlet air gas turbines of Fajr Petrochemical Company, given the climate and climatic conditions of Mahshahr Special Economic Zone, Fag evaporative cooling system is the most appropriate method and it was observed that by these methods, depending
on the relative humidity of the air, the ambient air temperature can be cooled up to 15-10 °C. According to the study of economic benefits of Fag evaporative cooling method, it seems that, in the studied compressor, this method is the most appropriate strategy for cooling the inlet air the compressor, reducing compression power and increasing its efficiency. Since increase in the inlet air temperature has the greatest effects on the first compressor, therefore only the results of the first compressor are investigated. Based on the studies and modeling of the studied compressor, it was observed that if the ambient temperature can be reduced up to 15-10 °C, specific compression power of the first compressor will be decreased about 5% (Figure 27). Given that, specific compression power is the maximum in the summer, about 10 percent more than in winter, therefore this reduction of power can be an effective step towards control of ambient temperature and improvement of the compressor performance. Specific dehumidifier filters designed to be used for this reason can be applied to avoid entering of excessive humidity to the compressor and damaging its parts.

**Figure 27. effect of inlet air temperature on compression power before and after cooling**

**CONCLUSION**

Based on thermodynamic modeling and modeling by neural network algorithm in this study it was observed that increase in relative humidity cannot affect the compressor’s compression power. With increasing volumetric Flow rate of inlet air the compressor, compression can be increased. Outlet airTemperature the compressor in any stage of Compression has always linearly increased with increase in inlet air temperature. It also can be said approximately that specific compression of the first compressor can also increase linearly under influence of inlet air temperature. Compression power in summer, in the first stage of Compression, is about 10% higher than winter and in the second and third stages it is, respectively, about 6.07 and 4.93 percent more than winter. The use of evaporative cooling systems has been proposed to reduce compression power of compressor in summer. By this method we can reduce the compressor compression power up to 5%.

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