Modeling and Optimization of Industrial Multi-Stage Compressed Air System Using Actual Variable Effectiveness in Hot Regions

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Abstract
In this article, modeling and optimization of power consumption of two–stage compressed air system has been investigated. To do so, the two – stage compressed air cycle with intercooler of FAJR Petroleum Company was considered. This cycle includes two centrifugal compressors, a shell, and a tube intercooler. For modeling of power consumption, actual compressors isentropic efficiencies and intercooler thermal effectiveness are calculated from experimental data. In these equations isentropic efficiency of compressors is a function of inlet temperature and thermal effectiveness of the intercooler is a function of inlet air temperature, inlet water temperature of the intercooler and inlet volumetric flow rate of the cycle. For optimization of power consumption, the Lagrangian method is used. Power consumption and isentropic efficiency of the first and second stage compressors, thermal effectiveness of the intercooler and entropy generation of compressors are considered as the objective function and optimization conditions, respectively. In comparison to the experimental data, the modeling provided suitable accuracy. The optimization effectively reduced the power consumption of the cycle, especially in summer, in a way that the minimum and maximum reduction was 2.9% and 9.6%, respectively.

Keywords
Multi-Stage Industrial Compressed Air System, Modeling, Optimization, Variable Isentropic Efficiency, Variable Intercooler Effectiveness

1. Introduction

Air compressors are used in variety of industries to supply process requirements, to operate pneumatic tools and equipments, and to meet instrumentation needs. Compressed air systems use about 10% of total industrial-energy use in the world. Also, these systems are typically one of the most expensive utilities in an industrial facility [1-2]. Some large compressors require several kilowatt powers to run. Even for a small air compressor with displacement of 10 m^3/min, an electrical motor of power 50-100 kW is required [3-5].

The work input to a compressor is minimized when the compression process is executed in an internally reversible manner. When the changes in kinetic and potential energies are negligible, the compressor work is given by $\int vdp$ [6]. Obviously one way of minimizing the compressor work is to approximate an internally reversible process as much as possible by minimizing the irreversibility such as friction [7-8], turbulence [9-10], and nonquasi-equilibrium compression.

The extent to which this can be accomplished is limited by economic considerations. A second (and more practical) way of reducing compressor work is to keep the specific volume of the gas as small as possible during the compression process. This is done by maintaining the temperature of the gas as low as possible at the inlet of compressor and during compression since the specific volume of a gas is proportional to temperature.

There are some regions in the world with specific climatic conditions such as large differences in minimum and maximum temperatures during a year. These temperature variations can cause to considerable difference in power consumption and performance of air compressors. These conditions lead to power consumption fluctuations up to about 15% [11-14].

For large or middle power compressors, one of the most important ways to save energy is using multi-stage compression and intercooling [15-19]. The air compressed part ways to the discharge pressure, passed through an intercooler and then compressed further. To minimize specific work input or power consumption, in two stage air compression with 100% isentropic efficiency and 100% intercooler effectiveness, is readily seen to require that the stage pressure ratio rise should be the square root of overall pressure ratio in the more restrictive perfect gas model.

Lewins [20] modeled and optimized a multistage compressor with an intercooler for an ideal gas model. He used Lagrange method of optimization to find optimum condition. He showed that working in the perfect gas model, in which leads to some elementary results in an optimization of a gas turbine that gives maximum work per unit mass circulated. These are useful for giving a feel to more realistic results in gas turbine and Joule cycle theory. It has been recently shown that the essence of these results remains true, albeit with a different emphasis, when the model is relaxed to the ideal gas only. Those results have been given for a Joule cycle optimized for maximum specific work. Also he showed that if the stage duties vary, optimal condition can be calculated as function of isentropic efficiencies of compressors and effectiveness of intercoolers.

Consequently, due to the supercritical heat rejection of the trans-critical cycle and the slopes of the isotherms in the supercritical region, the highest efficiency may be achieved at pressure ratios of the first and second stage compressors significantly different from each other, depending on operating conditions [21]. In actual condition, the isentropic efficiencies of compressors are not constant and have significant variations with operational condition [22]. Compressor efficiency, pressure ratio and inlet air temperature are three important parameters in optimization of power consumption in multi-stage compressors [23].

Wu et al. [24] optimized a multistage air compression system with a shell and tube intercooler. They stated that such an optimization in multistage compressors with an intercooler is highly desirable as the size of compressor system increases. Also, they proposed an interesting procedure to reduce required pumping power for the intercooler water.

Ghorbanian and Gholamrezaei [25] used an artificial neural network to compressor performance prediction. Their results show that if one considers a tool for interpolation as well as extrapolation applications, multilayer perception network technique is the most powerful candidate. Further, for the prediction of the compressor efficiency, artificial neural network is a powerful technique.

Al Doori [26] investigated the effect of important parameters on performance of gas turbine power plant. He focused on different parameters including different compressor pressure ratios, different ambient temperature, air fuel ratio, turbine inlet temperature, and the cycle peak temperature ratio. He showed that an increase in pressure ratio in each compressor would enhance the cycle's produced work. Also, any increase in ambient temperature increases power consumptions and reduces the cycle work. Therefore, ambient temperature rising can deteriorate thermal efficiency.

Unfortunately, there is a serious shortage in literature about air compressed system in hot region and effect of inlet temperature on isentropic efficiencies of compressor and intercooler effectiveness. Therefore, in this article, power consumptions of two stage air compressed cycle in hot regions have been modeled and optimized by considering of variation of ambient temperature, variation of isentropic efficiencies and intercooler effectiveness.

2. Material and Methods

2.1. Compressed Air System Specifications

The compressed air cycle that has been investigated in this article placed in Fajr Petrochemical Company (Mahshahr-Iran) and includes two centrifugal compressors and a shell and tube heat exchanger as intercooler. Table 1 shows operating parameters of both compressors in summer condition. Investigated intercooler is a water cooled shell and tube heat exchanger by Siemens. The outlet air from intercooler would be dried with an industrial dryer to fix air moisture content. Thus, it is assumed that air moisture content would be constant in compression process. Table 2 demonstrates the operating parameters of intercooler.

Table1. Operating parameters of compressors (Summer Condition)

Parameters	First Compressor	Second Compressor
Inlet air pressure (bar)	1	1.91
Inlet air temperature (K)	321	321
Inlet air flow rate (m^3/s)	8.70	4.61
Head (kJ/kg)	74.7	62.30
Outlet air pressure (bar)	2.03	3.46
Outlet air temperature (K)	409	396
Power Consumption (kW)	848	685

Table2. Operating parameters of intercooler heat exchanger (Design Condition)

Parameters	Value
Inlet cooling water pressure (bar)	4.2
Inlet cooling water temperature (K)	296
Outlet cooling water temperature (K)	305
Outlet cooling water flow rate (m^3/s)	170

To measure thermodynamic parameters such as pressure, moisture content, and flow rate, a Testo 400 device with three probes for each parameter was used. Measurement accuracy of device is 0.1^{o}C, 0.001 bar and 0.1 m^3/hr for temperature, pressure and flow rate, respectively. Also, a Fluke 1735 power analyzer device with data logger for compression power measuring in each compressor was implemented. Pressure ratios in compressors are variable with temperature, as overall pressure of the cycle sometimes is 3.8 bar (in winter) and 3.49 bar (in summer).

2.2. Thermodynamic Modeling

Figure 1 shows the components of a two-stage compressor with a single intercooler. For the thermodynamics modeling of power consumption of compressors, assumed that the air is ideal gas and includes moisture. Also compression processes in first and second stages, compressors are adiabatic.

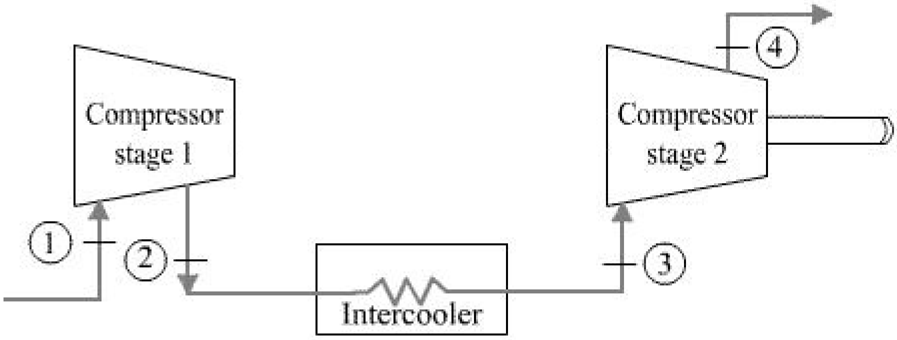

Figure1. The two-stage intercooled compressing system

2.2.1 First Compressor

We know from experimental results that the isentropic efficiency of compressor is the ratio of enthalpy differences in isentropic condition to enthalpy differences in actual conditions:

$$\eta_{12} = \frac{\Delta h_{ise}}{\Delta h_{act}} = \frac{h_{2s} - h_1}{h_2 - h_1} \tag{1}$$

By using the first law of thermodynamics and considering equation (1), the power consumption of the compressor in an ideal and actual compression can be calculated as:

$$W_{12s} = \dot{m}\left(h_{2s} - h_1\right) \tag{2}$$

$$W_{12act} = \frac{W_{12s}}{\eta_{12}} \tag{3}$$

According to equation (3), to calculate power consumption of the compressor, isentropic power consumption and isentropic efficiency of first-stage compressor based on inlet condition should be evaluated. The effects of air humidity have to be considered as well. Humid air can be considered as a mixture of dry air and water vapor. The water vapor is assumed to be near liquefying and humid air is treated as an ideal gas. By regarded equations (2) and (4), the enthalpy of humid air based on hypothesis of ideal gas model should be calculated [21]:

$$\frac{T_{2s}}{T_1} = \left(\frac{P_{2s}}{P_1}\right)^{\frac{k-1}{k}} \tag{4}$$

$$\phi = \frac{P_{vap}}{P'_{vap}} \tag{5}$$

$$P'_{vap}(T) = P_{cr}\exp\left(\frac{T_{cr}}{T}(-7.85\tau + 1.84\tau^{1.5} + 22.67\tau^{3.5} - 15.93\tau^{4} + 1.77\tau^{7.5})\right) \tag{6}$$

With evaluation of the mole fraction of water vapor in humid air, the specific heat capacity and consequently isentropic power consumption can be calculated:

$$X_{va\,p}=\frac{P_{va\,p}}{P_1}=\frac{\phi P'_{va\,p}}{P_1} \tag{7}$$

$$c_{pw}\left(T\right)=a_w+b_w\left(T\right)+c_w\left(T\right)^2+d_w\left(T\right)^3 \tag{8}$$

$$c_{pa}\left(T\right)=a_a+b_a\left(T\right)+c_a\left(T\right)^2+d_a\left(T\right)^3 \tag{9}$$

$$c_{p,ha}=X_{va\,p}c_{pw}+\left(1-X_{va\,p}\right)c_{pa} \tag{10}$$

$$W_{12act}=\frac{\dot{m}\left(c_{p,ha}(T_{2s})T_{2s}-c_{p,ha}(T_1)T_1\right)}{\eta_{12}} \tag{11}$$

By method of main square error, the isentropic efficiency of compressor as a function of inlet temperature can be calculated from experimental data with a two degree polynomial fit:

$$\eta_{12}\left(T_1\right)=0.000006948T_1^{\,2}-0.0050537T_1+1.6123 \tag{12}$$

2.2.2 Intercooler

The aim of modeling of intercooler in a compressed air system is calculating its thermal effectiveness (equation 13) and outlet air temperature.

$$\varepsilon=\frac{T_2-T_3}{T_2-T_1} \tag{13}$$

To calculate the effectiveness of intercooler, the outlet air temperature should be determined. Unlike to some studies [26-27] that assumed the effectiveness equal to quantity and constant, regarding to equation (13), thermal effectiveness couldn’t be constant.

According to recorded empirical data for this cycle, the best method for calculating the outlet air temperature and computation of thermal effectiveness is artificial neural network (ANN) method. To use ANN 256 different recorded data for intercooler were implemented.

Regarding the different networks and using trial and error method, the best network for intercooler is a feed-forward network by two hidden layer. Transfer function of the first layer is tangent sigmoid with 20 hidden neurons and second layer is log sigmoid with same number of hidden neurons.

To update weight factor and bios in the first hidden layer, trainlm function in Levenberg–Marquatd optimization method was used and Leamgdm training function has been implemented for the second layer that is a suitable method to reduce inclined plane to train and update weight and bios. To train the network, we used Train function in which the number of trained data is 70% of total number of data. 15% of data to test and 15% of them for network validation were chosen randomly.

2.2.3 Second Compressor

Power consumption of the second compressor depends on the thermal effectiveness of intercooler. Power consumption calculation procedure of this compressor is same of the first compressor. Also, since we have a dryer next to inter-cooler, inlet air moisture for the second compressor is equal to the first. The governing equation of the second compressor is similar to those of the first compressor.

Here, we can use a polynomial as a function of inlet air temperature to determine the isentropic efficiency of the second compressor, as same as the first compressor. Therefore, using experimental data, isentropic efficiency can be calculated:

$$\eta_{34}(T_3) = 0.00489246 T_3^{\,2} - 3.134729 T_3 + 572.427 \quad (14)$$

2.3 Optimization of Power Consumption

For optimization of the cycle, we introduce further Lagrange multipliers and distinguish between the efficiency of first and second stage compression. Writing the objective function as:

$$W \equiv L_0 = h_2 - h_1 + h_4 - h_3 \quad (15)$$

Together with constraints and their Lagrange multipliers

$$L_a \equiv \lambda\left[R\ln(r) - \int_{T_1}^{T_{2s}} \frac{dh}{T} - \int_{T_3}^{T_{4s}} \frac{dh}{T}\right] \quad (16)$$

$$L_b \equiv \chi\left[\eta_{12} - \frac{h_{2s}-h_1}{h_2-h_1}\right] \quad (17)$$

$$L_c \equiv \mu\left[\varepsilon - \frac{h_2-h_3}{h_2-h_1}\right] = \mu\left[\varepsilon - 1 + \frac{h_3-h_1}{h_2-h_1}\right] \quad (18)$$

$$L_d \equiv \kappa\left[\eta_{34} - \frac{h_{4s}-h_3}{h_4-h_3}\right] = \kappa\left[\eta_{34} - 1 + \frac{h_4-h_{4s}}{h_4-h_3}\right] \quad (19)$$

Then define the Lagrangian, a function of five now independent variables, as

$$L \equiv L_0 + L_a + L_b + L_c + L_d \quad (20)$$

And make the five-dimensional gradient $\nabla L = 0$ as a necessary condition. This provides: with substitution Lagrange multipliers obtained above (equation 15) and simplification it, can obtain optimization condition:

$$\frac{\eta_{12} T_{4s}}{\eta_{34} T_{2s}} = 1 + \frac{1-\varepsilon(T_{4s}-T_3)}{\eta_{34} T_3} \quad (21)$$

There is no analytical solution to find the optimum conditions, however, because it involves the temperature T_3 and the isentropic temperatures, which are not given directly by the η, ε parameters expressed in enthalpies. Nevertheless, the special case of the perfect heat exchanger ($\varepsilon = 1$) leads to somewhat a more simple optimum, since we than have $\eta_{12} T_{4s} = \eta_{34} T_{2s}$ and $T_3 = T_1$. If efficiencies are equal, then not only does $T_{2s} = T_{4s}$, but $h_{2s} = h_{4s}$ and $h_1 = h_3$.

To calculate the optimum condition of consumption power in an actual cycle using equation (21), one should follow the following scheme:

- Assume pressure ratio $r_{12} = \sqrt{r / r_o}$ for the first compressor, according to required pressure ratio and taking into account pressure losses in inter-cooler.
- Calculate parameters below:
  - $h_1(T_1)$, using temperature, moisture and flow rate of inlet air
  - T_{2s} and $h_{2s}(T_{2s})$.
  - ε, h_3 and T_3 using artificial neural network's functions.

- T_{4s} using $r_{34} = r / r_o r_{12}$ (pressure ratio in the second compressor).

- Solve equation (21) as a necessary condition for optimization of cycle's consumption power. If there would be a problem in equation satisfaction, reduce pressure ratio and outlet air temperature in the first compressor. Repeat these steps since convergence of equation 21.

3 Results and Discussion

3.1 Experimental Results and Modeling Validation

It can be observed from Figure 2 that isentropic efficiency of the first compressor is descending function of ambient temperature. From 280K to 325K ambient temperature variation, the falling of first compressor isentropic efficiency was around 5%.

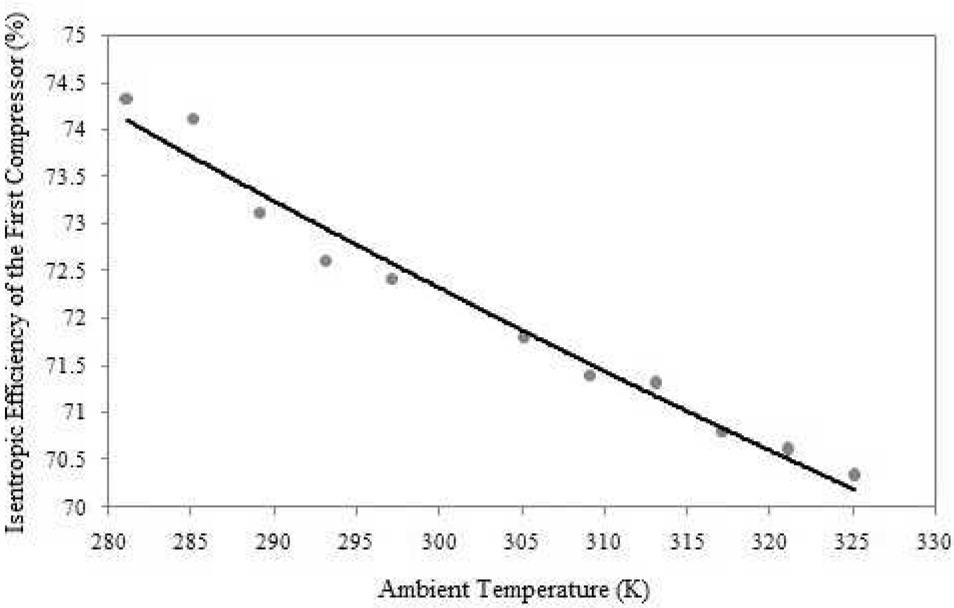

Figure2. Variation of isentropic efficiency with inlet air temperature of the first-stage compressor

We can see some results of modeling of power consumption of the first compressor with recorded data in Table 3. The results showed a good agreement with experimental data. In Mahshahr, maximum and minimum ambient temperatures are so far from each other (1^oC to 48^oC). The ambient temperature variations of this region have been showed in Figure 3. These conditions lead to power consumption fluctuations up to 14%.

Table3. Modeling results of power consumption in the first compressor versus recorded data.

T_1(K)	Q(m^3/s)	W_{12act}(kW)	W_{12mod}(kW)	Error (%)
281	6.175	773	723.10	6.47
285	6.308	791	739.30	6.46
289	6.388	793	745.13	6.45
293	6.300	790	742.14	6.45
297	6.138	775	728.17	6.44
305	6.470	804	754.95	6.43

309	6.470	804	755.15	6.43
313	6.484	794	743.42	6.43
317	6.752	821	768.88	6.42
321	6.484	794	746.24	6.42
324	6.484	794	745.30	6.44

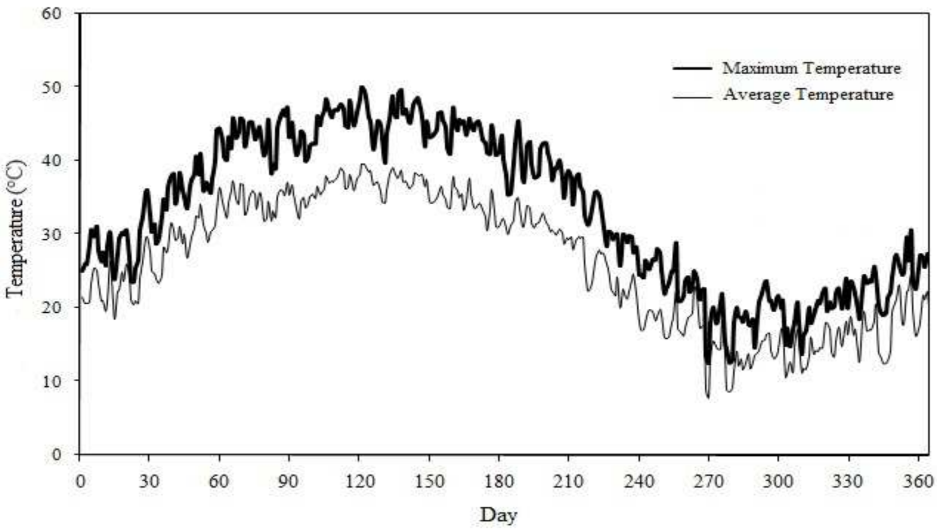

Figure3. Variations of maximum and average temperatures of Mahshahr during a year

Figure 4 states us that ANN has an appropriate response to variations of selected parameters. There is an interesting reaction from ANN against sudden change in outlet air temperature from intercooler (according to Figure 4, we have a sudden change in outlet air temperature after no. 100). This sudden change is caused by change in operational parameters because of change winter into summer season.

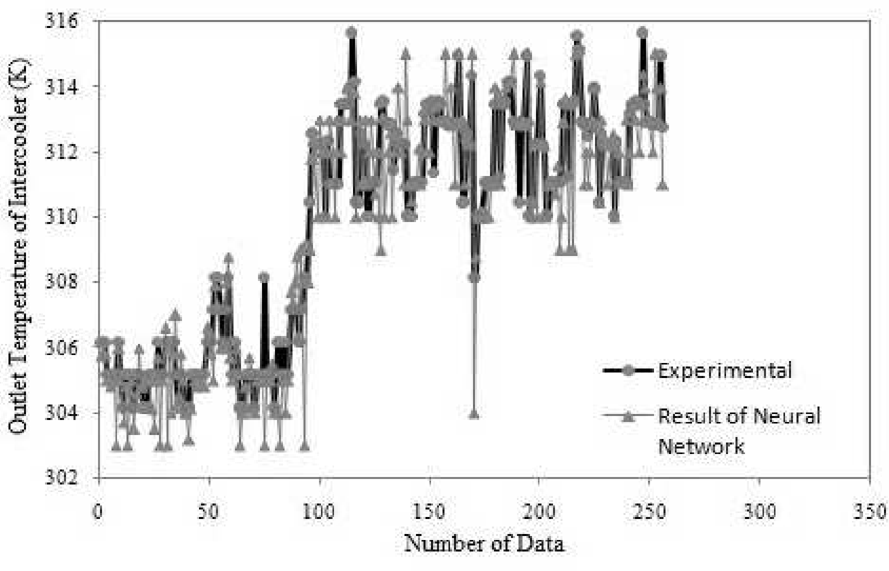

Figure4. Outlet air temperature from intercooler, comparison of ANN results with recorded data

The Table 4 shows some operational conditions of the intercooler. One can see from Table 4 that in different conditions, the falling of air pressure is constant. Point that mass flow rate of inlet water is constant. Also, it is obvious from Table 5 that ANN has a good ability to prediction of outlet air temperature from intercooler. According to Figure 5, the effectiveness of intercooler variation was from 0.7 to 1.1. It means that the inlet air temperature of second compressor can be lower than the inlet air temperature of first compressor in very hot condition. Therefore, the effectiveness can be larger than 1 in some cases. This fact has not presented in literature.

Table4. Operating conditions of the Intercooler

T_2(K)	T_3(K)	$T_{w,in}$(K)	Q_{in} (m^3/hr)	ΔP (bar)
369	303	291	22500	0.12
375	305	292	22500	0.12
384	305	295	23000	0.12
399	310	300	23700	0.12
405	310	300	23148	0.12
446	310	300	23300	0.12

Table5. ANN performance in outlet air temperature prediction of the intercooler

No. of tested data	Min. error (K)	Max. error (K)	Square mean error	Standard deviation
40	0.02	3.00	4.814	1.617

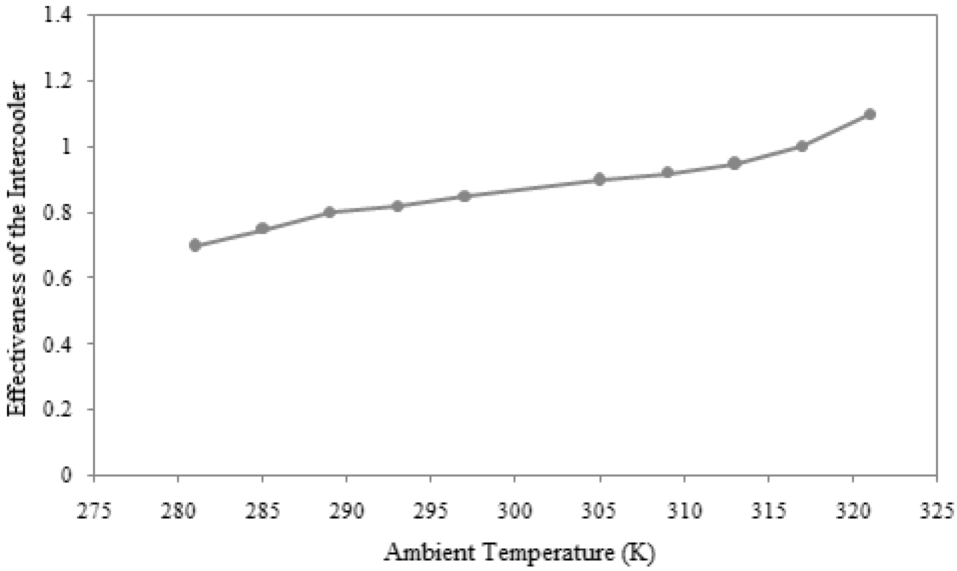

Figure5. Intercooler effectiveness variations against ambient temperature

Also, small variation in the inlet air temperature of the second compressor is an important point and it means that inlet air temperature range of this compressor is smaller than those of the first compressor, because of existence of intercooler.

Table 6 shows the comparison between modeling results and actual data of power consumption in the second compressor as well as the modeling error. The modeling error was in the range of the first compressor. Figure 6 shows the isentropic efficiency variation for second compressor against inlet air temperature. Isentropic efficiency variation for the second compressor was more limited because of narrow range of inlet air temperature variation of second compressor. In other words, the main reason of small variation in inlet air temperature for second compressor is small variation of cooling water temperature in summer and winter conditions. The maximum variation of power consumption of second compressor was around 3% in summer condition related to winter condition.

Table6. Results of the second compressor power consumption modeling versus recorded data.

T_{3act}(K)	T_{3model} (K)	W_{34act} (kW)	$W_{34model}$ (kW)	Error (%)
304	304.60	686	644.13	6.51
306	305.91	698	655.40	6.50
308	308.35	703	660.65	6.47
310	309.90	690	647.88	6.50
312	311.65	670	629.38	6.50
314	313.15	693	650.70	6.50
315	314.90	656	616.02	6.49

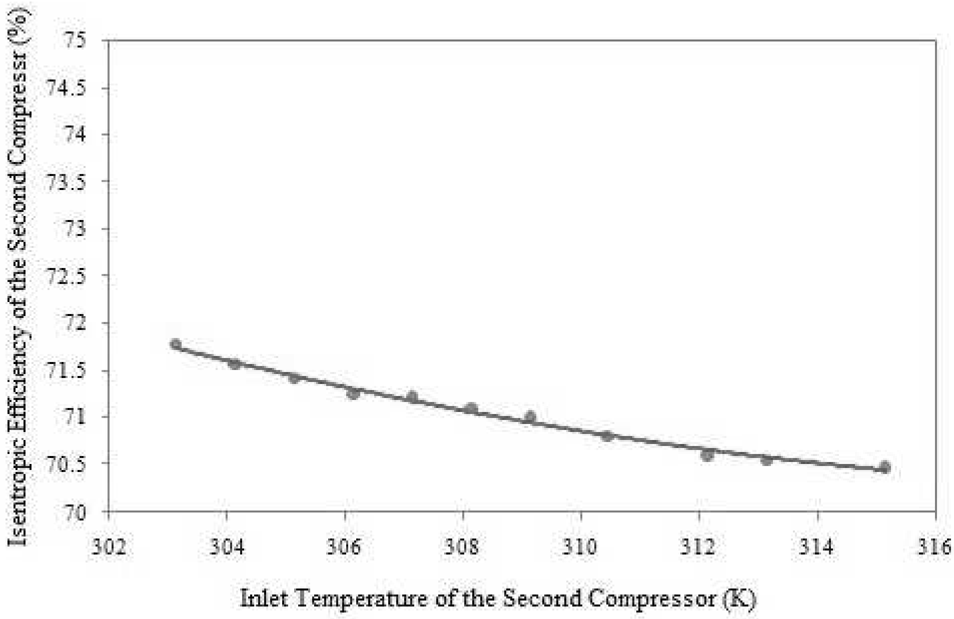

Figure6. Isentropic efficiency variation for the second compressor vs. inlet air temperature

3.2 Optimization

Operational conditions of the second compressor are depending on operational condition of the first compressor and the intercooler, strongly. We presented optimized parameters for both compressor including pressure ratios and optimized power consumptions in Tables 7 and 8. The maximum optimization for power consumption of the first and second compressor was 20% and 6% respectively. Also, Minimum and maximum optimization for power consumption of the cycle was 2.9% and 9.6% respectively.

Table7. Normal and optimized conditions for the first compressor

T_1 (K)	Q (m^3/s)	r_{12} (bar)	r_{12opt} (bar)	$W_{12model}$ (kW)	W_{12opt} (kW)
281	6.175	2.100	1.956	723.44	642.03
289	6.388	2.030	1.922	748.70	649.50
297	6.138	2.000	1.910	725.55	614.70
305	6.470	1.971	1.880	760.47	641.60

313	6.484	1.947	1.867	759.50	642.70
321	6.484	1.919	1.850	755.57	633.30
324	6.484	1.911	1.846	754.80	632.10

Table8. Normal and optimized conditions for the second compressor

T_1 (K)	Q (m^3/s)	r_{34} (bar)	r_{34opt} (bar)	$W_{34model}$ (kW)	W_{34opt} (kW)
281	6.175	1.180	1.943	650.79	693.75
289	6.388	1.180	1.911	653.00	685.14
297	6.138	1.180	1.895	611.80	630.03
305	6.470	1.782	1.876	625.98	636.40
313	6.484	1.776	1.855	647.12	660.20
321	6.484	1.776	1.848	614.43	621.34
324	6.484	1.776	1.845	608.74	612.00

Figure 7 shows a comparison between outlet air temperature variations for the first compressor at normal and optimized condition with regard to different inlet temperatures. It can be understood from figure 7 that outlet air temperature increases linearly with increase in inlet air temperature of the first compressor. At optimized condition, the outlet air temperature of first compressor was around 10K lower than normal condition.

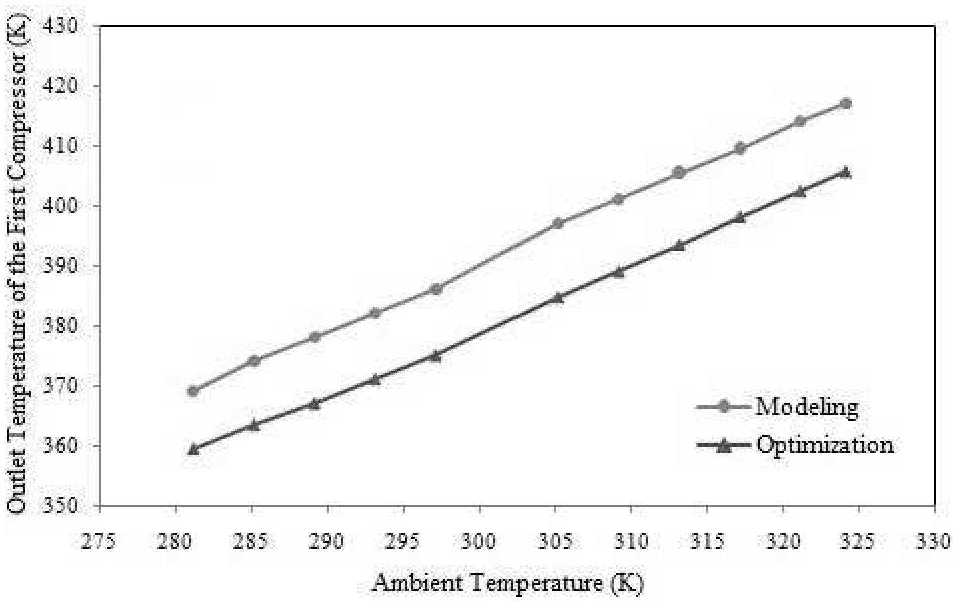

Figure7. Variations of outlet air temperature from the first compressor at normal and optimized condition

Figure 8 shows specific work consumption of the first compressor versus inlet air temperature for both modeling and optimization condition. In summer, the difference between normal and optimized condition was increased in comparison with winter.

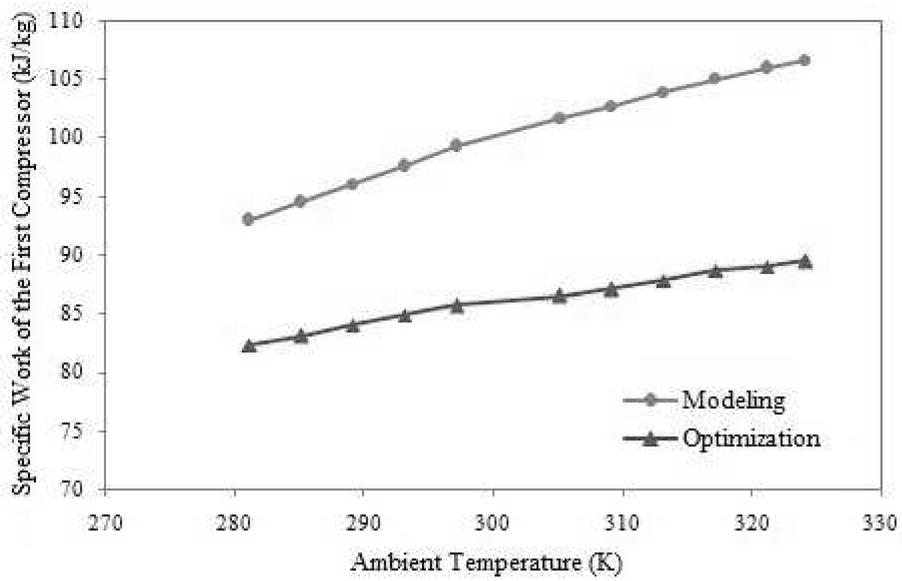

Figure8. Variations of specific work consumption of the first compressor as a function of ambient temperature at normal and optimized conditions

Using multistage compression system with an intercooler can tighten the range of temperature difference for the all compressor except the first one. Therefore, only the first compressor is affected with ambient temperature fluctuations, since intercooler existence. Figures 9 and 10 demonstrate variations of inlet and outlet air temperature of the second compressor versus ambient temperature for modeling and optimization results.

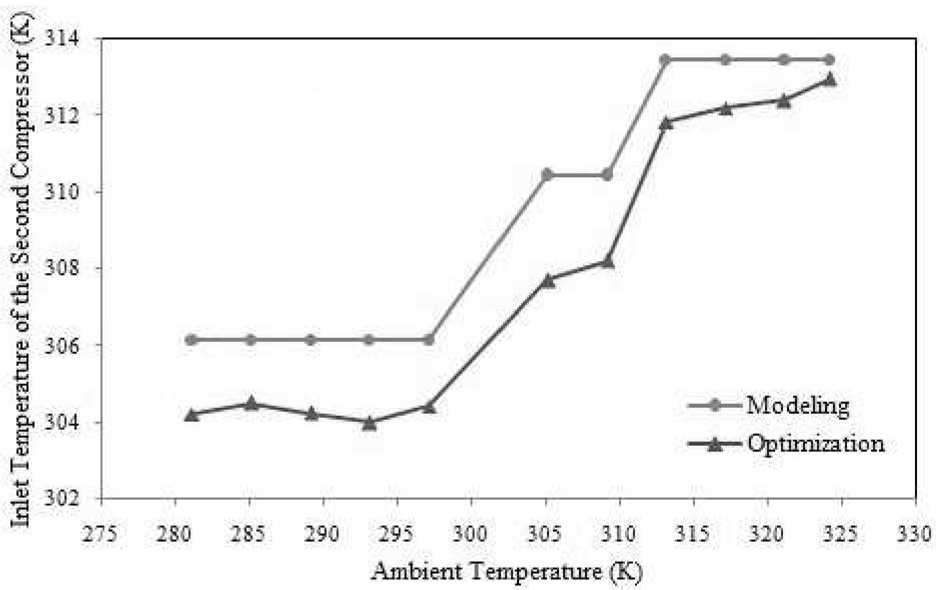

Figure9. Variations of inlet air temperature of the second compressor at normal and optimized conditions

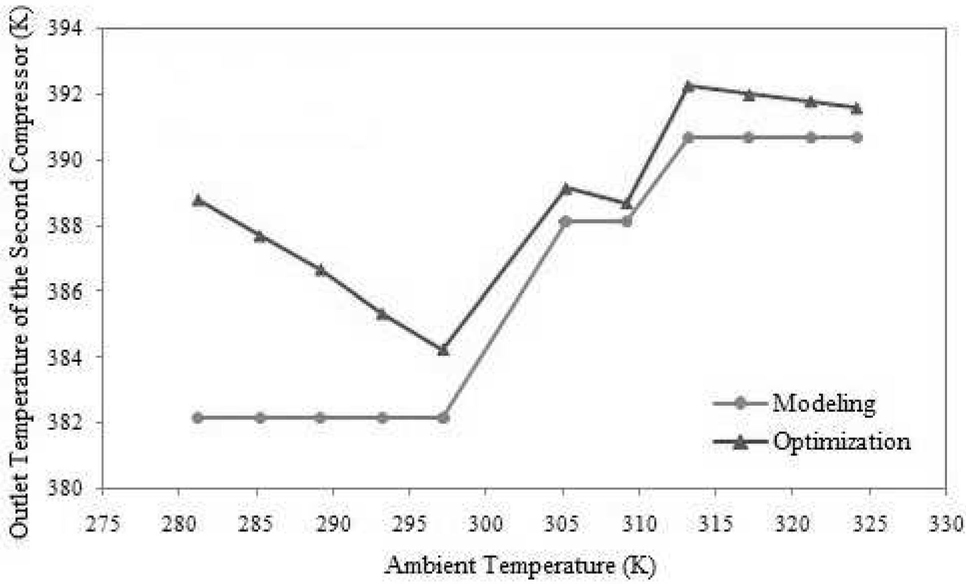

Figure10. Variations of outlet air temperature from the second compressor at normal and optimized conditions

As can be seen in Figure 9, despite ambient temperature increasing (and increase in inlet air temperature to intercooler), inlet air temperature to the second compressor has same value in some points. The main reason for this phenomenon is intercooler effectiveness. Thermal performance of intercooler is a function of ambient temperature, cooling water temperature and inlet air flow rate. According to figures 9 and 10, one can observe that optimized results about outlet air temperature from second compressor reduce sometimes with an increase in ambient temperature. This is because of difference in produced pressure ratios in the second compressor. In other words, inlet air temperature to this compressor are equal in some points, however outlet air temperature varies because of difference in optimized pressure ratios.

Figures 11 and 12 demonstrate variations of specific work consumption of the second compressor and total specific work of the cycle both versus ambient air temperature respectively.

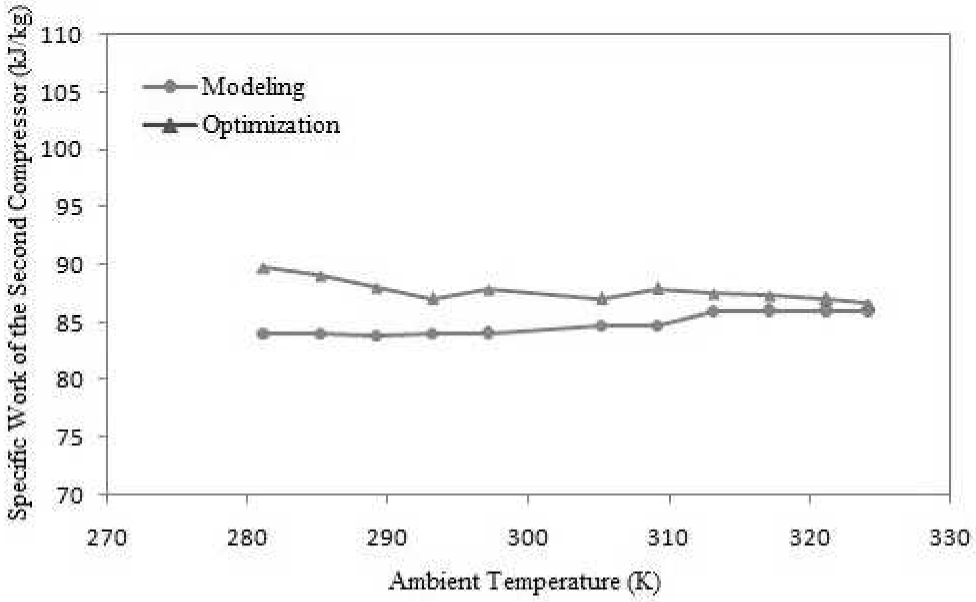

Figure11. Variations of specific work consumption of the second compressor as a function of ambient temperature at normal and optimized conditions

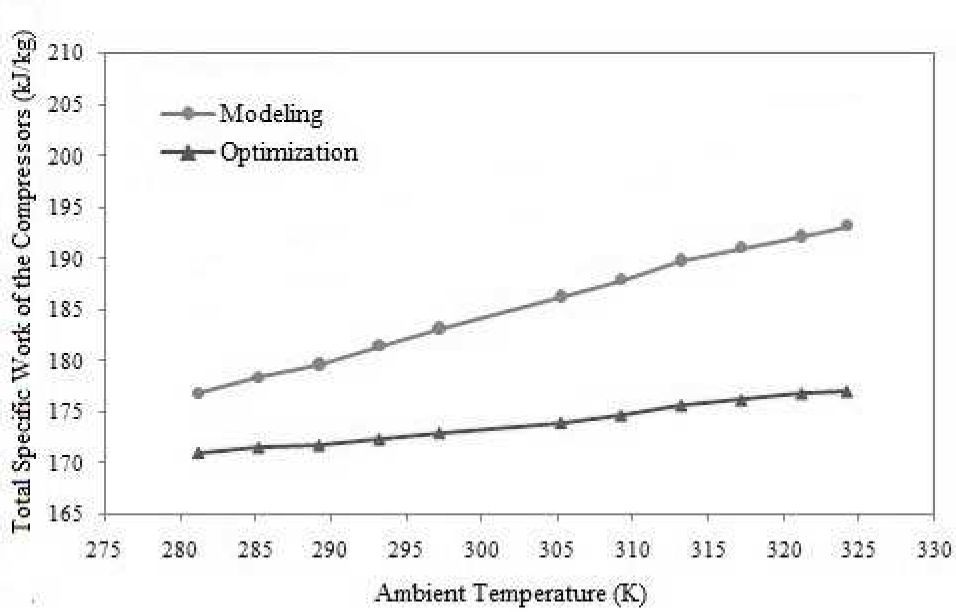

Figure12. Variations total work versus ambient temperature at normal and optimized conditions

It should be noted from figure 12 that operating conditions and the cycle's specific work consumption amounts are close to proportional modeling values at low ambient temperature (in winter), although with a considerable increase in ambient temperature (in summer) the cycle keeps out from optimized conditions. Total specific work of the cycle reduces in the range of 2.8% up to 8.72% by compressors optimization in winter and summer condition respectively. This occurs due to small variations of optimized isentropic efficiency compared to actual operating efficiency.

4. Conclusion

According to the modeling and optimization methods we can concluded that:

- Our modeling method has a satisfactory level of accuracy for both compressors' power consumptions.
- We used artificial neural network to predict outlet air temperature from intercooler. It seems that this approach is preferred to the assumption of constant thermal effectiveness for intercooler.
- The cycle's performance is too close to its optimum point in winter season. With large increase in ambient temperature (in summer), the cycle keeps away from optimum point. Total power of the cycle reduces in the range of 2.9% up to 9.6% by this optimization method in winter and summer condition, respectively.
- To optimize total work of the cycle and to supply the required pressure ratio in hot region, speed of compressors should be independent from each other. Consequently, Compressor designing for hot climate should be in form of split shaft (specially the first stage compressor) for operating compressors in different speeds and thus different pressure ratios to save more energy.
- To determine pressure ratios one should consider intercooler performance. In winter season, pressure ratio of the first compressor should be larger than the other compressor (and vice versa in summer season), because in this condition, inlet air temperature to the first compressor is smaller than of those of the second compressor.

5. Nomenclature

C_{pa} Specific heat capacity for dry air………………………………(J/Kg.K)

C_{pw} Specific heat capacity for water vapor……………………….(J/Kg.K)

$C_{p,ha}$ Specific heat capacity for humid air…………………………(J/Kg.K)

Δh_{act} The change in specific enthalpy in an actual compression… ..(J/Kg.K)

Δh_{ise} The change in specific enthalpy in an ideal compression…… .(J/Kg.K)

L_0, L_a, L_b, L_c, L_d Lagrangian functions…………………………………………..

P_{vap} Water vapor pressure…………………………………………......(KPa)

P'_{vap} Pressure of saturate water vapor………………… …………….(KPa)

Q Volumetric flow rate of inlet air to cycle…………………………(m^3/s)

R Air constant…………………………………………………….(J/Kmol.K) r

R Pressure ratio……………………………………………………………..

r_0 Pressure loss ratio in the intercooler…………………………………

T Air temperature……………………………………………………(K)

T_s Isentropic temperature…………………………………………….(K)

W_{act} Actual power consumption……………………………………(KW)

W_s Ideal power consumption………………………………………..(KW)

X_{vap} Mole fraction of water vapor in the humid air………………………....

6. Greek symbols

ε Thermal effectiveness of the intercooler…………………..………

η Isentropic efficiency of compressor……………………..…………

ϕ Relative humidity…………………..………………………………

λ,χ,μ,κ Lagrange multipliers……………………………………………...

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