Check for updates

Selection of compressors for petrochemical industry in terms of reliability, energy consumption and maintenance costs examining different scenarios

Suleyman Sapmaz[1], Fatma Canka Kilic[2*], Muharrem Eyidogan[3], Osman Taylan[4], Volkan Coban[3], Selman Cagman[3], Ibrahim Kilicaslan[1] and Durmus Kaya[1]

[1]*Department of Energy Systems Engineering, Faculty of Technology, Kocaeli University, Umuttepe, Kocaeli 41380, Turkey*

[2]*Air Conditioning and Refrigeration Technology, Department of Electrical and Energy, Kocaeli Vocational School, Kocaeli University, Mahmutpasa Mah. Mahmutpasa Cad. No: 151, Kullar/Basiskele/Kocaeli 41140, Turkey*

[3]*Department of Energy and Environmental Technologies, Karabuk University, Karabuk, Turkey*

[4]*Department of Industrial Engineering, Faculty of Engineering, King Abdulaziz University, Jeddah 21589, Saudi Arabia*

**Author for corresponding. E-mail: fatmacanka@hotmail.com*

(Received 11 June 2014; Accepted 16 December 2014)

Abstract

In this study, based on some important criteria, qualitative and quantitative evaluations were presented for the selection of compressors used in compressed air systems, in a petrochemical industry. Four different scenarios were investigated according to the main criteria considered for the selection of the compressors such as reliability, energy efficiency, investment, and maintenance costs. The types of compressor and their capacities were analyzed and compared. In Scenario I, the currently active electrically driven compressors were considered as backup compressors, instead of them, a new unit turbo compressor and two units of VSD screw compressors were purchased as active compressors. In this scenario, the initial investment cost is low and standby steam consumption is reduced due to the fact that the backup compressors are electrically driven. In Scenario II all active and backup compressors are replaced by a new one, which is proposed to be electrically driven. In this scenario, the initial investment cost is high, but energy consumption is very low. In Scenario III, all active compressors are planned to be replaced by new backup steam-driven compressors. In this scenario, although the initial investment costs and energy consumption are high, the reliability of this scenario is higher than the reliability of Scenarios I and II. In Scenario IV, all compressors are projected to be replaced by new and equivalent compressors. In this scenario, the initial investment cost and energy consumption are high, but the reliability of this scenario is the highest

comparing all other scenarios. The aim of this study is to contribute the studies on the selection of compressors by considering some important parameters as their types and capacities, in particular, for industrial enterprises that have risk of explosion like in the petrochemical industry.

Keywords: Air compressor selection, Energy consumption, Reliability, Maintenance, Carbon emissions

1. INTRODUCTION

Compressed air is kept under pressure that is greater than atmospheric pressure. It is an energy carrier obtained by compressing atmospheric air through a special machine. Electrical energy is used to produce compressed air. Compressed air is one of the most important industrial inputs after fuel, electricity and water. There are numerous industries using compressed air in a variety of applications. Compressed air is mainly used in instrumentation systems, industrial processes, and hand tools. The electric energy consumption of compressed air systems ranks third in energy costs in the production facilities (Gulec, 1999). For instance, compressed air constitutes 10% of industrial electricity consumption in the European Union (EU) (Saidur *et al*., 2010). Considering the usage of the compressed air in an industrial facility, only about 5–10% of the energy can be converted into useful energy for the production. Therefore, compressed air consumption is the most expensive form of energy usage in an industrial plant (Yang, 2009). Although compression of air is an inefficient and costly process, compressed air is substantially preferred in the applications of petrochemical processes because of its reliability against the risks of explosion.

Compressed air systems are usually composed of compressors, dryers, air receivers, dust filters, oil filters and the compressed air distribution lines. There is a large amount of moisture in the air. The moisture in the compressed air particularly damages the instruments, hand tools and processes. Therefore, it has to be removed from the system. The air that compressed in the compressors is dehumidified by passing through the dryers and then transferred to the air receiver. The air receiver responds to fluctuations in the demand effectively and provides a smooth operation of the compressor. Compressors and dryers are the main energy consumers in the system. Comparing the energy consumption of these two elements, the energy consumption of the dryer is relatively lower than energy consumption of the compressor. The main power consumer in a compressor unit is an electric motor that drives the compression element. A large portion of the input power of the engine is converted into heat energy during the compression process due to the warming air and heat released to the atmosphere. In an efficient compressor installation this heat can be used to save energy in other applications where it becomes needed. About 90% of the power used to run the compressor can be recovered as valuable heat energy (Copco, 2010).

Compressors compress air up by using different mechanical configurations. Basically, compressors are classified as positive displacement compressors and dynamic compressors. Positive displacement compressors work by forcing a given quantity of air into a chamber, namely that its volume is mechanically reduced to

compress the air. In these types of compressors, reciprocating and screw compressors are widely used. Reciprocating air compressors compress the air by taking into a piston-cylinder system. The piston-cylinder system is driven by an electric motor. Reciprocating compressors require quite large areas for their installation and produce a lot of noise. Screw compressors, as the other type of positive displacement compressors, compress the air by taking into pairs of helical screw. Screw compressors can be installed into smaller areas and they are often sold as packaged products. Although there are different types of dynamic compressors, their compression process is similar to each other. These compressors absorb the air through the impellers that rotating at high speed. The accelerated air is compressed by reducing in volume in the diffuser. This type of compressors offer higher efficiencies compared to reciprocating compressors and screw compressors at high capacities. In the past, the usage of reciprocating compressors in the compressed air systems was more prevalent. In the present day, screw compressors are now taking their place because screw compressors need small spaces and have higher efficiency values.

Given only 5–10% of the energy that consumed in the compressor is transferred to the compressed air, the studies for energy savings in compressed air systems are very important, since they provide huge economic and environmental benefits. Many studies have been conducted to support these researches in that area and they show that energy saving opportunities can be increased.

Saidur *et al*. (2010) described energy audit processes in compressed air systems by explaining the potential energy savings areas, the related calculation methods and improvements which can be made for the aim of energy savings in compressed air systems. They also explained different energy-saving measures, such as using highly efficient motors, VSD (Variable Speed Drive), leak prevention, using of outside intake air, reducing pressure drop, recovering waste heat, using of efficient nozzle, and using of variable displacement compressor to save compressed-air energy. According to their study, it has been found that for an electric motor used in a compressed-air system, a substantial amount of electric energy can be saved using high efficient motors and applying VSDs in matching speed requirements.

Yang (2009) completed some important studies on efficiency of air compressors. The objective of his study was to search out energy saving potentials in air compressor systems, in an enterprise in Vietnam. In this study it was discovered that the energy loss was due to non-optimized operations of the air compressor systems and air leakages. This study also explained that an energy efficiency investment in air compressor systems in the enterprise could be extremely cost-effective. If the enterprise invests US$ 84,000 in the air compressors to increase the efficiency performance, the investment capital cost can be recovered in six months.

In his study, Dindorf (2012) examined the potential energy savings in the compressed air line and the calculation methods. In this study, inventory and main operating parameters of the compressed air system should be constituted to define applicable energy saving methods. Hence, the compressed air systems performance indicators were defined as; specific power, annual energy cost, cost of compressed air, compressed air leaks, pressure drop in the system which can be calculated and estimated in the light of the obtained data.

Kaya *et al*. (2002) evaluated and quantified the energy losses associated with compressed-air systems, and their costs to the manufacturers. They have also showed how to reduce the cost of compressed air in existing facilities by modifications with attractive payback periods. The saving amount was US$ 2,627 with leak prevention after the studies were conducted in the facility, and using the high-efficiency motors in the systems the amount of savings was realized as US$ 4,335. Also, the amount of saving with reducing air compressor average inlet temperature was US$ 596 and with reducing the compressor air pressure, this amount reached the value of US$ 2,975. The total annual of $10,533/year energy savings can be achieved by applying these measures. The investment costs and payback periods of these energy saving potentials were reported in their study.

Saidur *et al*. (2012) aimed to identify energy saving opportunities and integrated costs of applying variable speed drives to the current applications of electrical motors in their study. Consequently, economic analysis, payback periods and the effect of current and voltage harmonics generated by VSDs have been investigated. In their study, they noted that the installation costs of VSDs are rather substantial. The cost varies between US$ 3,000 for a 5 Hp motor and US$ 45,000 for a custom engine of 300 Hp motor. The costs rise quickly for larger versions. Besides, using VSD systems assures the opportunity to save about 15–40% of the energy and extends equipment lifetime by permitting gentle start-up and shutdown. It was emphasized that VSD applications can sometimes produce harmonics in electrical distribution systems and the equipment in the systems by causing serious damage. It was also indicated that various methods were considered for preventing harmonics, but these methods are not sufficiently reliable, and come with a high cost.

The performance of water-cooled twin screw air compressors used for the Proton Exchange Membrane (PEM) fuel cell system has been examined by Li *et al*. (2009). In their research, a thermodynamic model of the working process of water-injection twin screw compressor was formed based on the equations of conservation of mass and energy. The effects of internal leakage and air–water heat transfer have been taken into account concurrently in this mathematical model. The experiments of the performance of a prototype compressor operating under various conditions have been directed to verify the model. With the increase of the rotation speed, the volumetric efficiency and discharge temperature were defined as greater, and the isothermal indicated efficiency increased at first and then decreased. The power consumption of the compressor has been an approximate linear relationship with the rotation speed. The additional power consumption for water injection occurs but it constitutes only a small fraction in the power consumption of the compressor. The results showed that the prediction the model is in good agreement with the experimental data.

Seshaihah *et al*. (2010) investigated oil-injected twin screw compressors with different types of light and heavy gas compressions on both theoretical and experimental studies. In their study, the values of efficiency, delivery rate and heat of compression of the medium and low sized compressors have been analyzed and presented. In addition, some compressors with the power of 5.5 kW and 37 kW were used to obtain a machine-independent experimental data. The compression performances of the air compressors, using argon, helium, nitrogen gases were

analyzed in their study. This study addressed the gas delivery rate and heat of compression on volumetric and power efficiency.

As they can be seen in the sources that mentioned above it is possible to save energy in compressor systems by using different methods (like air absorption from the external environment, the use of VSD, repair of leaks, reduction of pressure loss, waste heat recovery, etc.). The design of the compressor and the control methods have been studied in the different processes in the literature with various aspects. Nevertheless, on the choice of the compressors, there is no adequate work done in the literature. Energy consumption share of these systems is quite high in facilities that need a high amount of air. Thus, in addition to implementing the methods of energy recovery, the choice of the type of the compressors, their capacity, and the optimum choice of the drive systems are also important in such systems.

In this study, some important numerical evaluations were presented in detail for the selection of air compressors which will be used in compressed air systems for a petrochemical industry in Turkey. The selection of compressors has been realized by considering the annual changes of the consumption of compressed air in this industrial establishment. According to compressor types and capacities, there have been four different scenarios projected according to main parameters that considered the selection of the compressors based on; reliability, energy efficiency, investment and maintenance costs. The aim of this study is to contribute the studies on the selection of compressors by considering some important parameters as their types and capacities, in particular, for industrial enterprises that have risk of explosion like in the petrochemical industry.

There are significant number of publications about the design of the compressor, efficiency calculations, energy recovery methods as it can be given in the literature section of this study. However, on the choice of the compressors, there is no study can be found in the literature.

By taking some energy recovery measures in the compressors there is a considerable amount of energy saving can be provided. In addition, during installation of the compressor, the selection of the type of the compressors, their capacity, and the drive control methods assures the some important gain in the long term. As it is mentioned above this work has been realized in a petrochemical plant. And in this industrial establishment compressed air has the critical importance for the operations. The study has been carried out in order to determine the compressor combination in the plant that the ex-proof compressors have been used in. Also it is aimed that required qualities (pressure) and quantity (flow rate) for the facility has been assured as in a safe, reliable and economical way for the production.

2. COMPARISON OF SCREW COMPRESSORS AND TURBO COMPRESSORS

Nowadays, reciprocating compressors are widely used in the compression of high pressure and special gases applications. However, reciprocating compressors are not preferred in the applications of medium pressures with larger capacities. Reciprocating compressors can work at a pressure of 12 bars and about 100 Nm^3/h air flow rate. The average air consumption of the establishment that has been examined in this study is 12,500 Nm^3/h. To meet the need of high pressured air in the system, reciprocating

compressors have not been preferred due to their relatively low air production capacities. There has been high quality of oil free air (for quality 2nd, ISO 8573 with regard to the maximum oil concentration = 0.1 mg/m^3) (Ertas, 2007). The assessments have been conducted in the direction of the selection of oil-free compressors. Therefore, the following assessments have been carried out based on oil-free turbo and screw compressors.

The evaluation and testing standards of turbo and screw compressors are different. While dynamic compressors are tested in accordance with the ASME PTC 10 standard, displacement compressors are tested according to the ISO 1217 standards. Considering their standards differences between the two compressors, the comparison of turbo and screw compressors should be based on practical data. According to the ASME PTC 10 standards, the turbo compressor's air inlet temperature is accepted as 35 °C and the power value of this compressor's own catalog refers to the shaft power. In this case, the engine efficiency is not added to the calculation of power. In addition, oil heating system and oil circulating pump are not driven by the compressor engine, as they are fed by an external source. Thus, in addition to the compressor motor's energy consumption, there is external energy consumption considered in the evaluations. On the other hand, the screw compressors are tested as package power and the related values are conveyed to their catalogues. In screw compressors, all auxiliary equipment is available in the package. As for turbo compressors, some equipment is optional. Therefore, the additional equipment cost cannot be included in the costs of investment and energy consumption.

The maintenance requirements of turbo compressors and screw compressors are different due to their different operating principles. For example, in the oil-free screw compressors, the coating material (Polytetrafluoroethylene (PTFE)) on the screw elements wears out over time. In the region of the abrasion, the gap between the screws increases, some leaks will occur in the compressor, which causes to reduce the compressor's efficiency and increase specific energy consumption. In addition, due to the fact that the screw element cannot be coated again after 40,000 hours operating time, it needs to be replaced. The replacement of the screw elements is about one-third of cost of the screw compressor's sale price.

Imbalanced rotating elements in the compressors can cause mechanical vibrations. A turbo compressor's impeller works at high speeds, which causes vibration and risks of deviations in the rotor. While screw compressors' rotation speed is around 20,000 rpm, the turbo compressors rotating speed is around 50,000 rpm. In such high-speeds, some dust particles can cause corrosions on the impeller continuously. In this case, as a precaution, epoxy coating is applied to the impeller for preventing it from abrasion. Also, the 0.2 μm filters are used to prevent dust from entering into turbo compressors. On the other hand, these filters have a greater risk of blockage when comparing to the 5 μm filters used in the screw compressors, because they are much thinner.

It is a technological necessity to use sliding bearings in turbo compressors because they work at very high speeds. These kinds of bearings need to be lubricated before the start up. As for screw compressors, ball bearings are used in these systems. Theoretically, the life of sliding bearings is endless but the ball bearings should be

regularly replaced. The replacement intervals and costs of the ball bearings are also taken into account in the analysis of screw compressors.

The need of oil is very high in turbo compressors. While the volume of oil tank of turbo compressors is around 235 liters, screw compressors have oil tanks which have the volume varying in-between 35 and 50 liters. Based on their durability, the stainless steel oil cooler is used in screw compressors. As for turbo compressors, the copper coolers were used for the assurance of thermal efficiency. When comparing these materials, copper has a high risk of corrosion. As for the usage of stainless steel it can be taken into account that stainless steel requires more heat transfer area than the other.

Compressors can be graded according to their specific energy consumption, at different pressures. An energy efficient one must be made to meet most of the demands. At high capacities, specific energy consumption of turbo compressors is lower than the specific energy consumption of screw compressors. In terms of energy consumption, turbo compressors are more advantageous than screw compressors because they can work at full load operating conditions, 4,000 Nm^3/h and above of the flow rate. At partial loads, turbo compressors VSD (Variable Speed Driver) use is not available. As it is well known, a VSD increases the efficiency by granting the motors to be operated at the ideal speed for different load conditions. In addition, VSDs reduce engine electricity consumption by 30–60% in a lot of applications (Saidur *et al*., 2012). Instead of using a VSD, the load reduction is achieved by throttling the inlet or outlet clack, in turbo compressors. VSD applications cannot be taken place in turbo compressors; therefore, turbo compressors are less efficient than screw compressors at partial loads. Screw compressors can be controlled through VSDs. This assures that the specific energy consumption is lower due to the usage of a VSD in a screw compressor compared to the turbo compressors at variable loads.

The air flow control is carried out by applying different methods according to the type of the compressor. The simplest methods of capacity controls are start/stop and by-pass control. In the method of start/stop control system, when air tank pressure reaches to the high set pressure, compressor motor is completely stopped. When it comes to the low set pressure, the compressor motor starts running and the air production begins. A start/stop control system can be used in small-sized compressors, but in big sized compressors, this control system should not be used in the application that has frequent cycling because repeated starts and stops can cause overheating and damage in the engine. Therefore, other compressor components require more frequent maintenance and care (Marshall *et al*., 2010). As for the method of by-pass control, there are two options; either the produced air is discharged into the atmosphere or returned to the compressor suction. In addition, there is a modified version of this control system, which is called as load/unload control. This system can be used in positive displacement compressors. When air tank pressure reaches to the high set pressure, the air inlet valve closed and compressed air production is stopped, but the motor continues to run. Lower amount of air generation is still provided even the system is unloaded. When air pressure reaches to the low set pressure, air inlet valve is opened and air production is started again. As for dynamic compressors the capacity control is more difficult than positive displacement compressors. In these systems, capacity control is provided by throttling the inlet valves and outlet valves. Another capacity

control method that used in dynamic compressors is bypass control method. In this method, the generated air is discharged into the atmosphere. The compressor energy consumption will be reduced partially due to its running against to the atmospheric pressure.

3. EXISTING SYSTEM DESCRIPTION AND MEASUREMENT METHODS

The compressed air system contains ten units of compressors in the examined petrochemical plant. Five units of these compressors are used as active compressors and they run continuously. Each active compressor has a vapor-driven back up compressor and has the same capacity. The group of active compressors is constituted four units of screw compressors and a unit of turbo-electrically driven compressor. During the study of the compressor replacement, it has been decided that the electrically-driven screw compressor 450 A and its backup vapor-driven screw compressor 450-B remain on site, because they are relatively new and have the higher efficiency comparing to others. Existing air compressors are placed in two different locations in the facility. Therefore, the fluctuations in the compressed air demand are to be properly addressed in the system. The existing compressor system is displayed in Figure 1.

The air capacity of current system has been measured to be 12,500 Nm^3/h. The average air demand has been around the level of 12,700 Nm^3/h, which is higher than this measured value. As being a system peaks the air demand was measured as 14.000 Nm^3/h in the last two years. This value indicates that some of the backup compressors have been put into use. The air consumption of existing system can be seen in Figure 2.

In this survey, power consumption, steam flow rate, air flow rate have been measured. Fluke 435-II energy analyzer has been used for the analysis of the energy consumption of electrically driven compressors. To determine steam flow rate entering the turbine, a Delta-top DO64P brand flow meter has been used in

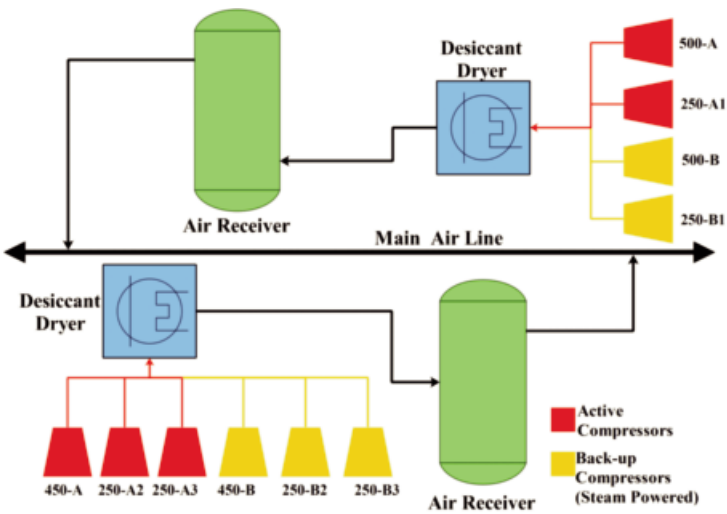


Figure 1. The existing compressor system.

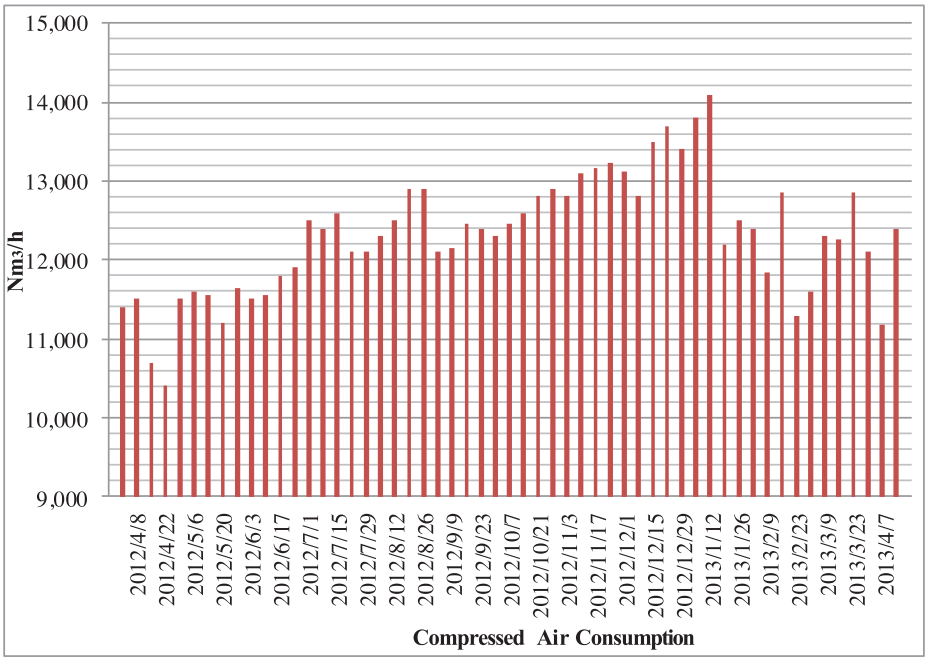


Figure 2. The air consumption of the existing system.

vapor-driven compressors. Pressurized air flow sent by the compressor was measured by a VP Flow Scope brand flow meter.

4. THE CREATION OF COMPRESSOR SELECTION SCENARIOS WITH RESPECT TO COMPRESSED AIR CONSUMPTION

The first scenario (Scenario I) is the process of handling the existing active electrically driven compressors as backup compressors. Thereupon, it has been suggested that a unit of turbo compressor and two units of VSD screw compressors are to be purchased as active compressors and two unit of compressor is planned to remain in the system. The compressor 450-A is considered as an active and compressor 450-B is considered as a backup compressor. The specific energy consumption of the turbo compressors is lower than screw compressors. Therefore, most of the constant load value will be provided by turbo compressors with high capacities. The purchased VSD compressors will be able to easily meet the demand of the system variability by mounting them in two different compressor rooms. In this case the backup compressors will not incur any investment cost. In case of emergency, the compressor group will be fed with a generator against power cuts. For this purpose, it is planned to purchase a power generator with a capacity of 1,425 kVA to support backup compressors. While this scenario exhibits low investment cost, the reliability of the backup system is not high. The steam compressors can be put into use in a very short time, but generator-powered electrically driven compressors needs certain period of time to be activated. The names and drive systems of the Scenario I compressors can be seen in Table 1.

Table 1. The names and drive systems of the compressors for Scenario I.

Scenario I				
Active Compressors				
Compressor Number	New-600-A Turbo	New-400-A VSD Screw	New-400-B VSD Screw	450-A Screw
Drive System	Electrical	Electrical	Electrical	Electrical
Back-up Compressors				
Compressor Number	500-A Turbo	250-A1 Screw	250-A2 Screw	450-B Screw
Drive System	Electrical	Electrical	Electrical	Steam

In Scenario II, the active compressor system is similar to the compressor used in Scenario I. In this scenario, it has been suggested that a unit of turbo compressor and a units of screw compressors would be purchased and it is planned that a steam-driven turbine compressor 450-B will remain in the system as a backup. The suggested names of the compressors and their drive systems are given in Table 2 in Scenario II. In an emergency case, a power generator, which has the power of 1,425 kVA will feed the compressors. As the backup compressors would be purchased recently, the initial investment cost of this scenario will increase. One of the differences between Scenario I and Scenario II is that Scenario II has higher cost of investment than Scenario I. But the 600-A and 600-B turbo compressors can be operated with equal aging process, so they will exhibit a longer service life. In other words, 600-A and 600-B turbo compressors will be operated as backup and part time actively.

The active system of Scenario III is similar to the above mentioned two scenarios' of active systems. The backup compressors that will be put in this system are selected as steam driven compressors by considering the reliability. The names and drive systems of the compressor for Scenario III are given in Table 3. It is suggested that a new unit of turbo compressor will be purchased as backup compressor for Scenario III. This makes the Scenario III a lower investment cost scenario comparing to Scenario II. Compressor 450-B and 250-B3 are the existing compressors and will stay in the system as backup compressors. All backup compressors are steam driven compressors, so that they enable the system to eliminate the need for any generator investment. Despite this, the steam consumption in the standby mode will cause an increase of the operating costs.

Table 2. The names and drive systems of the compressors for Scenario II.

Scenario II				
Active Compressors				
Compressor Number	New-600-A	New-400-A VSD	New-400-B VSD	450-A
Drive System	Electrical	Electrical	Electrical	Electrical
Back-up Compressors				
Compressor Number	New-600-B	New-425-A	450-B	
Drive System	Electrical	Electrical	Steam	

Table 3. The names and drive systems of the compressors for Scenario III.

Scenario III				
Active Compressors				
Compressor Number	New-600-A	New-400-A VSD	New-400-B VSD	450-A
Drive System	Electrical	Electrical	Electrical	Electrical
Back-up Compressors				
Compressor Number	600-B	250-B3	450-B	
Drive System	Steam	Steam	Steam	

Table 4. The names and drive systems of the compressors for Scenario IV.

Scenario IV				
Active Compressors				
Compressor Number	New-425-A Screw	New-400-A VSD Screw	New-400-B VSD Screw	450-A Screw
Drive System	Electrical	Electrical	Electrical	Electrical
Back-up Compressors				
Compressor Number	New-425-A Screw	New-425-A Screw	450-B Screw	
Drive System	Steam	Steam	Steam	

In Scenario IV, all compressors consist of screw compressors. This scenario requires purchasing an active compressor group that consists of two units of VSD screw compressors and a unit of screw compressor without a converter. The names and drive systems of the compressor for Scenario IV are given in Table 4. Compressor 450-A will still remain in the system. Two units of screw compressors will be purchased as backup compressors. All backup compressors will be vapor-driven compressors in terms of reliability. In this scenario, a significant amount waste heat recovery will be provided from the screw compressors. Nevertheless, standby steam consumption of the backup compressors will be quite high. A single unit of back up compressor will take any other compressor's place as an equivalent compressor, because all compressors in the system have the same capacity.

5. INVESTIGATION OF SCENARIOS AND THEIR CONTENT

The evaluations and classification of the compressors need to be expressed by numeric values. In this study, each suggested scenario is analyzed by comparing the compressors. In addition, the reliability, the costs, reduction of carbon dioxide of the existing system and the suggested scenarios were all evaluated.

5.1. Reliability

The reliability of a compressed air system expresses the air production in the desired pressure, flow rate, and air quality. If the air pressure drops, the instrumentation devices will be no longer in use and the hand tools cannot provide sufficient power.

The drop of air pressure in the petrochemical industry results in a risk of disrupting the production. If the control systems don't work properly, the boilers, furnaces, separation columns and hot fluid lines cannot be controlled. Therefore, reliability is vitally important to petrochemical plants. Reliability can be explained based on fundamental principles especially in any compressor failure that occur in the compressed air system, the backup compressors can be put into use immediately, to ensure the air in the required amount and conditions, and provide a continuous air production.

In this study, it was determined that the existing backup compressors were selected as steam compressors. Steam compressors' activation time is quite short. However, a significant amount of steam is spent in the standby state. It was observed that a unit of electrically driven compressor and a generator (as a group) were used as an alternative to steam turbines in the applications of backup compressors. In emergency situations, such as power cut, backup compressors are advised to be driven by a generator group.

When these systems were compared, it was determined that the duration of activation of electrically driven compressors is longer than the steam turbine compressors'. In the system with generator, first generator and then the compressor is required to be activated. In this system, the usage of a larger air receiver will reduce risks. After receiving the start instructions, the steam compressors are activated instantly. In the suggested scenarios, the backup compressors are also examined in terms of their types of drives (being steam driven or electrically driven). In the cases like Scenarios III and IV, the backup compressors are steam compressors; therefore, it is said that their reliability is higher comparing to others. In the scenarios, the steam cost of these compressors is significantly higher than the values of the first two scenarios.

The cost of steam consumption for the period of 20 years is visualized in Figure 3.

5.2. Energy consumption

The specific energy consumption of the compressors can be calculated by using the energy consumption of the compressor and the amount of air production. In the

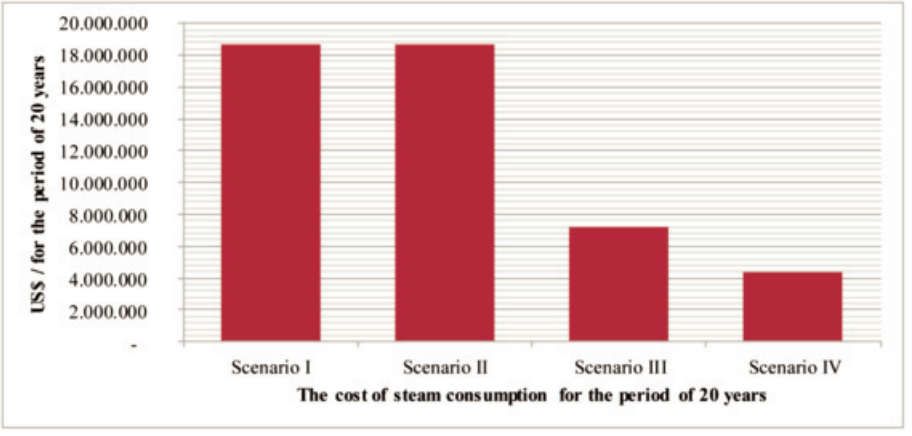


Figure 3. The cost of steam consumption for the period of 20 years.

literature, the specific energy consumption for an ideal compressor system working at 7 bar can be presented as follows;

- between 0.085 kWh/Nm^3 and 0.11 kWh/Nm^3, which is a very good ratio,
- between 0.11 kWh/Nm^3 and 0.13 kWh/Nm^3, which is an acceptable ratio,
- above 0.13 kWh/Nm^3; ratio indicates a problem in the system (Dindorf, 2012).

The specific energy consumption varies depending upon the values of altitude, humidity and temperature. The specific energy consumption has been calculated by measuring the amount of air production and energy consumption values in the existing compressor system. The measured air flow rate and power values that refer to the existing compressor system are reported in Table 5.

The specific energy consumption of the existing system has been calculated as 0.1454 KWh/Nm^3. The manufacturer catalog data have been used for the calculation of specific energy consumption of the new compressors. The specific energy consumption values that obtained from these calculations are given in Table 6. According to these data, for the first three scenarios, the specific energy consumption of the active compressor system was calculated as 0.1222 kWh/Nm^3. As for Scenario IV, this value was determined as 0.1240 kWh/Nm^3.

As it was depicted in the energy consumption tables, replacing the existing compressor system will reduce the energy consumption significantly. The specific energy consumption in Scenario IV is relatively high in the suggested system and it costs an additional amount of US$ 1,200,000 in the period of 20 years.

Table 5. The energy consumption of the existing compressor system.

Compressor	250–A1	250–A2	250–A3	450–A	500–A	Total	Specific Energy Consumption (kWh/Nm^3)
Air Flow (Nm^3/h)	1,307	1,700	1,760	1,821	3,190	9,778	**0.14536**
Measured Power (kW)	222	277	240	233	450	1,421	

Table 6. Total energy consumption and energy costs for the period of 20 years.

	Specific Energy Consumption	Energy Cost			
Unit	kwh/Nm^3	$/Nm^3	$/h	kWh/20 year	$/20 year
Existing System	0.1454	0.0490	611.92	540,888,000	97,906,677.77
Scenario I	0.1222	0.0411	514.28	454,584,000	82,284,704.42
Scenario II	0.1222	0.0411	514.28	454,584,000	82,284,704.42
Scenario III	0.1222	0.0411	514.28	454,584,000	82,284,704.42
Scenario IV	0.1240	0.0417	521.85	461,280,000	83,496,754.08

5.2.1. Carbon emission

Essentially, reduction of energy consumption is about reduction of carbon emissions. The reduction of carbon emission can be calculated depending on annual energy gain. The carbon footprint of electricity generation for Turkey is given as 0.66 kg CO_2/kwh. As for the value of the steam and the heat that have been produced in the plant, this has been calculated as 0.46 kg CO_2/kwh. Figure 4 shows the values of emission reductions achieved by comparing the existing systems. The calculations that related to carbon reduction are made according to the continuity of the existing system's operations.

As a result of the calculations, the first two scenarios suggest 6029.28 tons/year amount of CO_2 emission reduction, Scenario III suggests 3,329.04 tons/year and Scenario IV suggests 3,223.67 tons/year amount of CO_2 emissions reduction when compared with the existing system. Depending on the energy consumptions, the amount of emission reductions of Scenario IV compressors are lower than the values of the first three scenarios'. In Scenarios I and II, the amount of emissions reduction are equal because the active compressors used in these scenarios are the same.

5.2.2. The standby steam consumption of back up compressors

In a standby state of the backup compressors, they are kept in a mode as being ready to work by passing the steam through their turbines. Although this process assures the compressor system being switched very quickly in an emergency situation, it also causes a significant amount of steam consumption. It has been observed in the measurements that 10% of steam is consumed by the compressors in the standby state and at the full load. The compressor inlet temperature is 320 °C steam pressure is 38 bar,

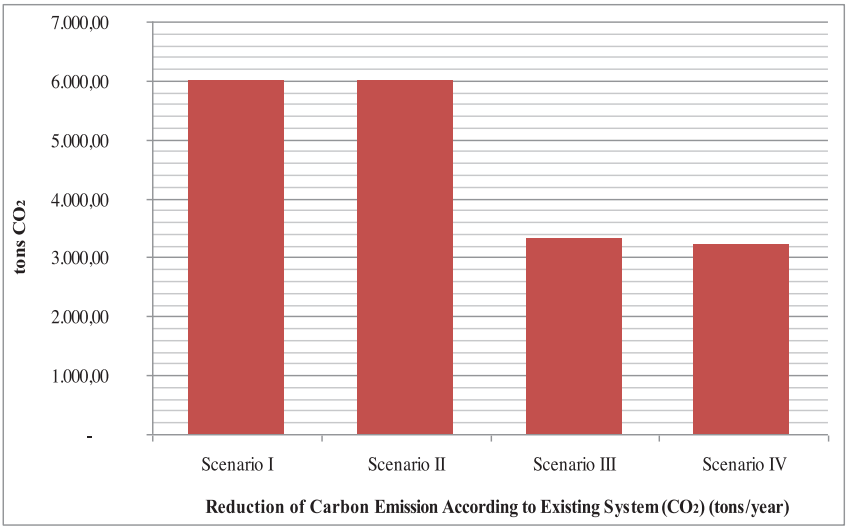


Figure 4. The annual energy consumption and emissions reduction values of suggested scenarios comparing to existing system.

respectively. The turbine outlet steam pressure is 3.5 bars and the temperature is 165 °C, respectively. The cost of steam has been taken as 0.1488 US$/ (kg/h) according to company's data and this value is used for the calculations. To prevent steam consumption in the standby state, it is necessary the backup system should be comprised of electrically driven compressors and need to be fed by an electrical turbine. The gain that is to be obtained by prevention of steam consumption in existing compressor systems is calculated by using expression (1). The amount of gain obtained by preventing the steam consumption is presented in Table 7.

$$Q = x\ (h_i - h_o) \tag{1}$$

Where;

Q = energy used in steam turbine, kW
= Mass flow rate, 0.5683 kg/s
h_i = Enthalpy of the steam, at the turbine inlet at 38 bars, 3,016 kj/kg
h_o = Enthalpy of the steam, at the turbine outlet at 3.5 bars, 2,776 kj/kg
Q = 0.5683(3,016 – 2,776) = 136 kW

In the existing system the amount of steam consumption of backup compressors will be US$ 48,716,405 in the period of 20 years. Scenario III and IV, in which backup compressors are steam driven compressors, the amount of steam consumption will be higher. In Scenario I and II, the backup compressors are electrically driven compressors; hence, the amount of steam consumption is around 1/3 of the amount of the existing system's steam consumption.

5.2.3. Waste heat recovery

As it is well known, compression of the gases is an inefficient process because it causes intense amount of heat release. Only a small amount of the energy delivered from the engine can stay in the compressed air, and the high amount of heat energy is released to atmosphere. 80 to 93% of electrical energy used in the electrically driven compressor is converted into heat (Saidur *et al*., 2012). In the applications, 60% of heat energy can be recovered and used successfully (Dindorf, 2012). This value is higher in oil-free screw compressors. As an example, around 94% of energy supplied to oil free screw compressors can be recovered (Copco, 2010). Waste heat recovery

Table 7. The steam consumption of backup compressors in the standby state.

	The steam consumption				
Unit	kg/h	kWh	US$/h	kWh/20 year	US$/20 year
Existing System	2,046.22	136.00	304.48	269,135,056	48,716,405
Scenario I	589.32	40.09	87.69	77,512,032	14,030,530
Scenario II	589.32	40.09	87.69	77,512,032	14,030,530
Scenario III	1,481.92	100.80	220.51	194,913,852	35,281,551
Scenario IV	1,702.48	115.70	253.33	223,923,650	40,532,643

Table 8. The amount of heat recovery from oil free screw compressors in the period of 20 years for all scenarios.

	Waste heat recovery			
Unit	kWh	US$/h	US$/20 year	kWh/20 year
Existing System	720	107.14	17,141,760	94,700,101.10
Scenario I	750	111.60	17,856,000	98,645,938.64
Scenario II	750	111.60	17,856,000	98,645,938.64
Scenario III	750	111.60	17,856,000	98,645,938.64
Scenario IV	975	145.08	23,212,800	128,239,720.24

units can recover the waste heat in intercooler, after cooler and oil circuit. The amount of recoverable energy is variable because the load is variable. In this study, the amount of recoverable energy from the compressors is accepted about 60% of input power. This method is only used for the calculation of oil-free screw compressors because it is not used in the calculations for the turbo compressors. Table 8 presents the returns in terms of energy amount and financial gains.

The amounts of heat recovery from the oil free screw compressors presented in Table 8 evokes the advantages of oil free screw compressors. Scenario IV assures the maximum amount of waste heat recovery because it has the highest number of screw compressors. Comparing to other scenarios, the financial gain obtained from this scenario is higher approximately by US$ 5,356,800.

5.3. Maintenance cost

Although oil-free screw compressors assure the opportunity of gaining considerable energy from waste heat recovery, their screw element needs to be replaced by a new one after working 40,000 hours of operating time because of its coating wears at the end of this period of time. If the screw element's replacement necessity is delayed, specific energy consumption will increase. Screw element replacement cost is around one-third of the price of the compressor. Considering the plant's 8,000 hours operating time during 20 years period, it has been calculated that the screws will be replaced at least three times during this period and their maintenance cost also has been taken into account. It has been projected that turbo compressors' routine maintenance cost (like oil change processes) will be around US$ 200,000 in the period of 20 years. As for oil-free screw compressors, this value will be around US$ 100,000 because the volume of oil-free screw compressors' oil tank is smaller. Given the existing system is old, maintenance costs are increased by 20% on the basis of the suggested scenarios. The maintenance costs of the compressor scenarios are shown in Figure 5.

As it has been mentioned above, the most important factor in the maintenance costs of oil-free screw compressors is the replacement of the screw element because of the abrasion. The number of oil-free screw compressors in Scenario IV is more than other scenarios; therefore, the maintenance costs are slightly higher than others.

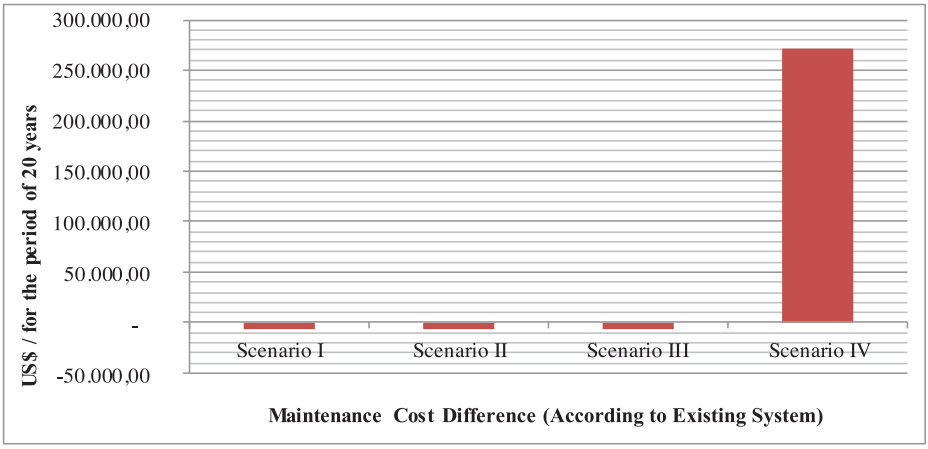


Figure 5. The maintenance costs of the scenarios' compressor systems compared to the existing system for the period of 20 years of operation (projected as difference values).

6. LIFECYCLE COST AND PAYBACK PERIODS

Payback period is a very important criterion to evaluate the feasibility of an investment accurately. The payback period of an investment is simply an indicator of determining if the return meets the expenses or not. With a simple explanation, the payback period of the investment can be described as the length of time required to recover the cost of an investment. Payback period can be calculated as follows:

$$\text{Payback Period} = \text{Initial investment}/\text{Periodic Cash Flow} \quad (2)$$

Because the studies that have been discussed in this study are indexed to the existing system, the calculated payback periods are also given by comparing to it. Operating costs are calculated by using the electrical energy consumption costs, steam cost and waste heat recovery. When calculating the investment costs, compressor prices, energy recovery unit prices, generator prices are all taken into account. By using operating and investment costs, the payback period has been calculated. These results obtained from the calculations are shown in Figure 6.

Compared to the operating costs of the current system, the benefit of the proposed compressor scenarios shown in Figure 6 for the period of a 20-year. As you can see in the graph, if you select any of the proposed options for a period of 20-year, the operating costs will be reduced at least US$ 30,000,000. Compared to Scenario III and IV, in Scenarios I and II, (with the use of turbo-compressors) operating costs will be reduced approximately US $ 20,000,000 more.

The lowest payback period was obtained from the calculations of Scenario I. In this scenario, there will be no investment for the backup compressors and also the backup compressors will be electrically driven; therefore, there will be no steam consumption. Scenario III and IV offer the longest payback periods due to their reliability, which are

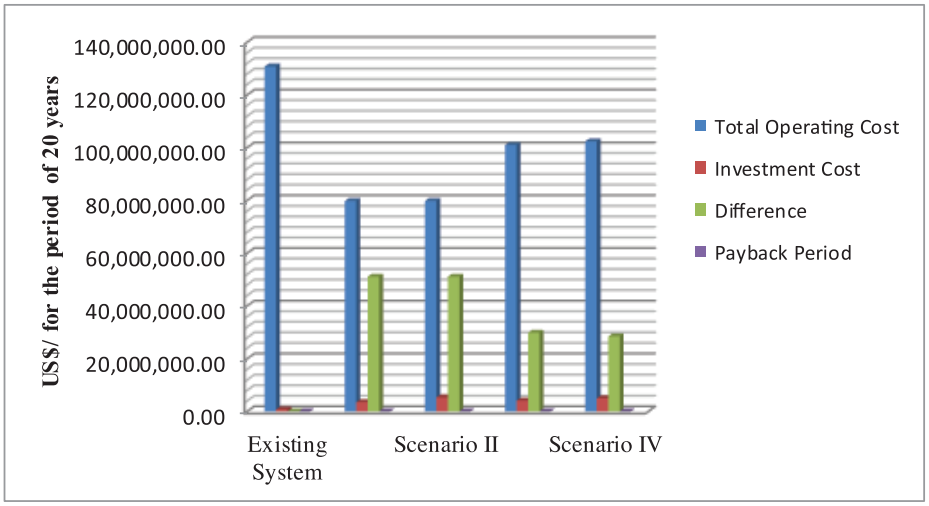


Figure 6. The operating and investment costs and payback periods of all scenarios and the existing system.

the most reliable options, they have steam backup compressors. Although Scenario IV has the advantage of the waste heat recovery because it has screw compressors, its payback period is the longest one of duration of 3.5 years.

7. CONCLUSIONS

In this study, some evaluations have been carried out for the selection of compressors that used in compressed air systems in the petrochemical industry in Turkey. These evaluations are examined and presented based on four different scenarios considering the most important parameters, which are reliability, energy efficiency, initial investment cost and maintenance costs. Compressor types and capacities are also taken into account in these four scenarios. In the direction of these examinations in the petrochemical plant, the compressed air system consists of 10 units of compressor, 5 of them are being used as active compressors. There are steam-driven backup compressors that have equivalent capacity of each of the active compressors. Active group of compressors consists of 4 units of screw compressors and a turbo electrically driven compressor. During the compressor replacement study, it has been suggested that the electrically driven screw compressor 450-A and its backup, the steam-driven screw compressor 450-B should remain on site, which are relatively new and have the higher efficiency in comparison with other compressors. The average air demand in the plant is 12,700 Nm^3/h. The peak air demand has been measured as 14,000 Nm^3/h in the system for the last two years.

The activation time of steam compressors is quite short. However, they consume significant amount of steam when they are in the standby state. In the applications of backup compressor, a unit of electrically driven compressor and generator set has been considered as an alternative to the steam turbine. When comparing these systems it can

be seen that the electrically driven compressors are activated in a longer time of period than steam turbines. In the system with the generator, principally, it is necessary that the generator should be activated first and then the compressor should be put into use. As for steam compressors, they can be operated swiftly. The backup systems of Scenario III and IV are steam driven, therefore their reliability is higher.

The specific energy consumption of the existing system has been calculated as 0.1454 kWh/Nm^3. In Scenario I, II, and III the specific energy consumption of the active compressors has been determined to be 0.1222 kWh/Nm^3. As for Scenario IV, the energy consumption is 0.1240 kWh/Nm^3. The calculations that have been realized by considering for the period of 20 years led to US$ 97,906,677.77 for the existing system, US$ 82,284,704.42 for Scenario I, II and III, US$ 83,496,754.08, for Scenario IV, respectively.

In the existing system approximately 269,135,056 kWh and US$ 49,000,000 worth of steam will be consumed by the backup compressors for the period of 20 years. In Scenario I and II, the backup compressors are electrically driven; the steam consumption will be around one-third of the steam consumption amount of the existing system. The steam consumption of Scenario III and IV, for which the backup compressors are steam driven compressors, will be close to the steam consumption of the existing system.

Although waste heat recovery is possible in screw compressors, it is not applicable in to turbo compressors in today's technology. Because Scenario IV comes with the highest number of screw compressors, the highest amount of waste heat recovery can be made in this scenario in comparison to the other scenarios. The amount of waste heat recovery present in Scenario IV is higher and it can be expressed as a difference, which is around 29,593,782 kWh and US$ 5,356,800 for the period of 20 years comparing to the other scenarios.

In today, most of the world's CO_2 emissions are produced by the conversion industry. Therefore, even energy savings studies that can be made in the compressor systems play a major role in reducing emissions. In the first two suggested scenarios, 6,029.28 tons of more emissions are expected to be gained compared to the current situation. If the Scenario III and IV would be preferred, 3,329.04 and 3,223.67 tons of CO_2 emission reduction will be achieved respectively, when comparing to the current situation.

Because the number of screw compressors in active compressor systems of Scenario I, II and III are the same, the maintenance costs of these suggested scenarios are equal. The maintenance cost of the first three scenarios is US$ 5,755 which is less than the value of the existing system, for the period of 20 years. Due to the fact that all active compressors consists of screw compressors, the maintenance costs of Scenario IV has additional cost which is about US$ 271,750 comparing to the existing system, for the period of 20 years.

Economic provision and payback period of an investment are the most important criteria in the feasibility study. The payback periods of Scenarios I, II, III and IV, are 1.37, 2.07, 2.73 and 3.50 years, respectively in this study. Backup compressors of Scenario I consists of the former active compressors. Therefore, payback period of Scenario I is the shortest comparing to the other scenarios, because there would be no investment cost for the backup compressors for Scenario I.

REFERENCES

Copco A., 2010. Compressed Air Manual, 6th Edition. Atlas Copco Airpower NV, Wilrijk, p. 145.

Dindorf R., 2012. Estimating potential energy savings in compressed air systems. *Procedia Engineering* **39**, 204–211.

Ertas E., 2007. Basıncli havanin üretimi, sartlandirilmasi ve dagitimi: enerji verimliligi acisindan yeni muhendislik kriterleri. VIII. Ulusal Tesisat Muhendisligi Kongresi, İzmir Turkey, October 25–28, 2007.

Gulec M., 1999. Pnomatik Sistemlerde Tasarruf. I. *Ulusal Hidrolik Pnomatik Kongresi*. Izmir, Turkey.

Li J.F., Wu H.G., Wang B.M., Xing Z.W. and Shu P.C., 2009. Research on the performance of water-injection twin screw compressor. *Applied Thermal Engineering* **29(16)**, 3401–3408.

Kaya D., Phelan P., Chau D. and Sarac H.I., 2002. Energy conservation in compressed-air systems. *International Journal of Energy Research* **26(9)**, 837–849.

Marshall R. and Scales B., 2014 Compressed Air-Best Practices, Sustainable manufacturing features, 11–12/10. Compressed air controls, Excerpted and submitted by Ron Marshall and Bill Scales for the Compressed Air Challenge Available at: https://www.compressedairchallenge.org/library/articles/2010-11-CABP.pdf, Accessed: 18. 02. 2014.

Saidur R., Mekhilef S., Ali M.B., Safari A. and Mohammed H.A., 2012. Applications of variable speed drive (VSD) in electrical motors energy savings. *Renewable and Sustainable Energy Reviews* **16(1)**, 543–550.

Saidur R., Rahim N.A., Hasanuzzaman M., 2010. A review on compressed-air energy use and energy savings. *Renewable and Sustainable Energy Reviews* **14(4)**, 1135–1153.

Web Page, "http://www3.tema.org.tr/Karbonmetre/Karbonmetre.html" Accessed: 20.3.2014.

Yang M., 2009. Air compressor efficiency in a Vietnamese enterprise. *Energy Policy* **37(6)**, 2327–2337.