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DESIGN AND FABRICATION OF MOBILE AIR COMPRESSOR

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ABSTRACT

Air compressors are used as versatile tool throughout the industries for a variety of purposes. They are used for various applications in manufacturing plants such as driving pneumatic tools, air operated controlling equipment's, conveying of fly ash etc.. . Air compressors are one of the major sources of energy consumption in industries. In the present scenario, the importance of energy conservation is increasing day by day. For a manufacturing plant, improving energy efficiency or replacement of less energy efficient equipment with energy efficient ones can earn significant savings. In the manufacturing plant, it was found that compressors which are running were installed in the earlier stages. Due to aging, the efficiency of compressor decreased and it resulted in huge expense. As a solution, the performance assessment of the compressors were done. Compressor with an energy efficiency less than 35% is replaced with energy efficient Compressor. Modification in the compressor air system was done to improve efficiency. The latest trend in the automotive industry is to develop light weight vehicles. Every automotive industry is looking to reduce the weight of the vehicle as it helps in the better handling of the vehicle and increases the efficiency of the vehicle. Today, the heavy vehicles are known for producing a large amount of harmful gases like CO2, SO2 etc. which act as the major source for global warming. So research is going on to find a light weight vehicle which does not pollute the environment. One of the alternatives is the use of compressed air to generate power to run an automobile. Due to the unique and environmental friendly properties of air, it is considered as one of the future fuels which will run the vehicles. So in this paper an effort is made to study the extent of research done and the potential advantages and disadvantages of the compressed air technology.

Keywords: - Light Weight Vehicles, Compressed Air, Global Warming, Alternative Sources of Energy. INTRODUCTION In 1650, German scientist Otto

The first instance of an air compressor used for something other than fire management shows up in 1762, a "blowing cylinder" powered by a water wheel. It produced a whopping 14 psi. A water-cooled version of this appears in 1872. In 1857, workers began drilling a tunnel through the Swiss Alps by hand. In 1650, German scientist Otto von Guericke devised an air pump consisting of a single piston and cylinder. With it he was able to experiment with compressed air and vacuums. George Medhurst of England designed the first motorized air compressor in 1799 and used it in mining.A compressor is a mechanical device that increases the pressure of a gas by reducing its volume. An areducing its volume. An air compressor is a specific type of gas compressor.Compressors are similar to pumps: both increase the pressure on a fluid and both can transport the fluid through a pipe. An air compressor is a mechanical device that increases the pressure of air by reducing volume.Air is compressible, the compressor reduces the volume of air and induces pressure in the airAn air compressor converts electrical energy into kinetic energy in the form of the air

An air compressor is a device that converts power (using an electric motor, diesel or gasoline engine, etc.) into potential energy stored in pressurized air (i.e., compressed air). By one of several methods, an air compressor forces more and more air into a storage tank, increasing the pressure. When tank pressure reaches its engineered upper limit the air compressor shuts off. The compressed air, then, is held in the tank until called into use. The energy contained in the compressed air can be used for a variety of applications, utilizing the kinetic energy of the air as it is released and the tank depressurizes. When tank pressure reaches its lower limit, the air compressor turns on again and re-pressurizes the tank. An air compressor must be differentiated from a pump because it works for any gas/air, while pumps work on a liquid.

There are basically 5 types of air conditioner compressor that are commonly used in the HVAC industry

- 1. Reciprocating
- 2. Scroll
- 3. Screw
- 4. Rotary
- 5. Centrifugal

There are many choices to consider when looking to buy a new compressor for your industrial plant. The most common choices are reciprocating compressors, screw compressors, and centrifugal compressors. The best choice depends on the size of your facility and how you use compressed air.

Compressors can be classified according to the pressure delivered

- Low-pressure air compressors (LPACs)
 - which have a discharge pressure of 150 psi or less
- Medium-pressure compressors
 - which have a discharge pressure of 151 psi to 1,000 psi
- High-pressure air compressors (HPACs)
 - which have a discharge pressure above 1,000 psi

They can also be classified according to the design and principle of operation

- Single-Stage Reciprocating Compressor
- Two-Stage Reciprocating Compressor
- Compound Compressor
- Rotary-screw compressor
- Rotary Vane Compressor
- Scroll Compressor
- Turbo compressor
- Centrifugal compressor

Displacement type

There are numerous methods of air compression, divided into either positive-displacement or roto-dynamic types.

a) Positive displacement:

Positive-displacement compressors work by forcing air in a chamber whose volume is decreased to compress the air. Once the maximum pressure is reached, a port or valve opens and air is discharged into the outlet system from the compression chamber.[4] Common types of positive displacement compressors are

Piston-type

air compressors use this principle by pumping air into an air chamber through the use of the constant motion of pistons. They use one-way valves to guide air into and out of a chamber whose base consists of a moving piston. When the piston is on its down stroke, it draws air into the chamber.

2) Technical Illustration of a two-stage air compressor

Its up stroke, the charge of air is forced out and into a storage tank. Piston compressors generally fall into two basic categories, single-stage and two-stage. Single stage compressors usually fall into the fractional through 5 horsepower range. Two-stage compressors normally fall



Fig 1

3) Technical Illustration of a portable singlestage air compressor

Compressor compress into the 5 through 30 horsepower range. Two-stage compressors provide greater efficiency than their single-stage counterparts. For this reason, these compressors are the most common units within the small business community. The

capacities for both single-stage and twostage compressors is generally provided in horsepower (HP), Standard Cubic feet per Minute (SCFM)* and Pounds per Square Inch (PSI). *To a lesser extent, some compressors are rated in Actual Cubic Feet per Minute (ACFM). Still others are rated in Cubic Feet per Minute (CFM). Using CFM to rate a compressor is incorrect because it represents a flow rate that is independent of a pressure reference. i.e. 20 CFM at 60 PSI.

a) Rotary screw compressors

Use positive-displacement compression by matching two helical screws that, when turned, guide air into a chamber, whose volume is decreased as the screws turn.

b) Vane compressors

Use a slotted rotor with varied blade placement to guide air into a chamber and compress the volume. This type of compressor delivers a fixed volume of air at high pressures.

Dynamic displacement

Dynamic displacement air compressors include centrifugal compressors and axial compressors. In these types, a rotating component imparts its kinetic energy to the air which is eventually converted into pressure energy. These use centrifugal force generated by a spinning impeller to accelerate and then decelerate captured air, which pressurizes it.

Cooling

Due to adiabatic heating, air compressors require some method of disposing of waste heat. Generally this is some form of air- or watercooling, although some (particularly rotary type) compressors may be cooled by oil (that is then in turn air- or water-cooled). The atmospheric changes are also considered during cooling of compressors. The type of cooling is determined by considering the factors such as inlet temperature, ambient temperature, power of the compressor and area of application. There is no single type of compressor that could be used for any application.

Pneumatics

A system which uses compressed air is called pneumatics.It deals with the study of behavior& application of compressed airA basic pneumatic system consist of a source of compressed air, control valves, pipelines & pipe fittings and pneumatic accessories like filter, regulator and lubricator.

Single stage Reciprocating Air compressor

The type of the entire compression is carried out in a single cylinder. When piston starts moving downwards, the pressure inside the cylinder falls below atmospheric pressure that opens suction valve. The pressure of the air in the cylinder rises during compression and at the end of compression, delivery valve opens and discharges the compressed air into the receiver tank. Single stage air compressor develop pressure upto 7 bar. For higher pressures multistage compressors are suitable

Double stage Reciprocating Air compressor

It consist of two cylinders – low pressure cylinder and high pressure cylinderPiston,

crankcase, piston rod, crank, crankshaft, oil, fins etc. The fresh air is drawn inside the L.P. cylinder through inlet suction filter. This air is compressed by piston As the piston moves towards the end of cylinder, the air compression took place. The delivery valve opens and this compressed air from L.P. cylinder is directed to enter inside the high pressure cylinder. In high pressure cylinder this pressurised air is further compressed to higher pressure. The high pressure air from H.P. cylinder is then delivered to receiver through discharge valves. In this compressor, a pressure of air delivered is upto 13 bar.

Rotary vane compressor

It is positive displacement type compressor. It provides higher efficiency and flow rates over a wide range of pressure. Rotary vane compressor consist of rotor with a number of vanes inserted in the radial slots cut in rotor.

2.2CALCULATION REVIEW

The free air delivery of the compressor can be calculated by receiver filling method. The volume of the receiver is noted if mentioned on the name plate. If the receiver volume is not known it should be physically measured by pouring water from a calibrated measuring can. Dished contours cannot give correct volume when calculated analytically. The compressor is kept running on load and no load for some time so that the temperature of the compressor increases. The compressor is then stopped. The sizes

of pipes up to isolations valves is measured and volume of pipe is calculated. The volume is added to the receiver volume and called 'effective receiver volume'. The valves which isolate the compressor receiver from the delivery lines are closed. Pressure gauge reading is noted. It should read zero because all the air in the receiver is drained. The compressor is started and kept on full load. This should be kept in full load mode, if controlled by an external control panel. Time taken by the compressor, in seconds to reach a cut-off pressure is recorded. FAD = (effective volume of receiver inliters)*60*(p2-p1) (Time to reach set pressure in sec)*p0

2.3 CALCULATION OF VOLUMETRIC EFFICIENCY

The volumetric efficiency of the compressor is calculated by the ratio of actual free air delivery of the compressor to specified free air delivery of the compressor. Volumetric efficiency = (Actual Output of Compressor/Specified Output of compressor).

2.4 CALCULATION OF POWER CONSUMPTION

Power consumption is calculated by: Power consumption $=\sqrt{3*V*I*Cos\phi} V =$ Voltage (V) I = Current (A) Cos ϕ = 0.98 3. PERFORMANCE ANALYSIS OF COMPRESSORS In the manufacturing plant, twelve compressors were taken under study. The performance parameters such as free air delivery, volumetric efficiency and power consumption were calculated

SAMPLE CALCULATION

Location: Raw mill Purpose: Raw mill grease spray and free jacking K.G khosla make, three cylinder (two LP and one HP) two stage, air cooled, belt driven, oil lubricated, and electrically driven mounted compressor with start-stop regulation having the following name plate details: Internal number: 30077 Model: 2 BC 26 Rpm: 750 FAD: 736*10-3 m3/min Actual rpm was found to be 716 Safety valve, drain cock and pressure gauge were found to be ok. The compressor was found to be cutting off at 7 kg/cm2 Discharge temperature was found to be 490c Maximum current drawn at the time of unloading was found to be 7.2 to 7.4 A Receiver filling time to reach 7 kg/cm2 from atmospheric pressure was found to be 206.5 sec Theisolation valves are very close to the receiver and the additional volume because of the pipes are negligible.

Output of the compressor = (250*60*7)/206.5=508.47*10-3 m3/min Volumetric efficiency = (508.47/736)*100=69.08% Power Consumption: Full load power = 5.5 kW Intake current = 7.3 A Full load current = 11A Power consumption = $\sqrt{3}*V*I*Cos\emptyset$ Power consumption = $\sqrt{3}*415*7.3*0.98 = 5.1$ Kw Table -1: Analysis Results Compressor No: Actual FAD (m3/min) Volumetric efficiency % Power consumption (kw)

PREVENTIVE MAINTENANACE

The majority of compressors working in the plant have less efficiency. Due to cost problem, all compressors cannot be replaced. The compressor having an efficiency of 30.7 is to be replaced by energy efficient one .For the other compressors preventive maintenance practices can be followed in order to improve energy efficiency. Leakage and wastage were checked and found to be absolutely minimum. Rigorous monitoring were done by the plant

Suction filters of the compressor in raw mill & packing plant should be changed as per the details mentioned later. The design separates and holds quiet some dust before the filter element. In case of compressors all the suction filters should be like the ones fitted on the compressor used for raw mill grease spray. The only difference between these filters and the old ones is provision of two plastic cases outside which create centrifugal action and which remove lot of dust before air reaches the filter element. Routine periodic maintenance of valves, cooling. system, lubricating system, unloading system etc. Should be done Preventive maintenance should be done condition based rather than the calendar

based. In suction filters of the compressors lot of gaps are found between the filter frames and inlet filters of compressors. As a result unfiltered air is likely to enter in to the compressors. It is recommended to clean the filters more frequently and observe the pressure drop across the suction filter.

2.5 REPLACEMENT OF COMPRESSOR

The compressor having an efficiency of 30.7% is to be replaced. Selection of new compressor is very important here. At present screw compressors are more efficient than any other compressors available. The annual energy savings that the plant can acquire

OLD COMPRESSOR

Discharge pressure = 9 kg/cm 2 Motor Kw = 5.5 kW/7hp FAD =736*10 -3 m 3 /min Cut-off pressure: 6 kg/cm 2 Cut-in pressure: 5 kg/cm 2 Annual Energy cost = (Full load power (kW) *annual running hours* electricity cost/kWh) / motor efficiency Full load power = 5.5 kW Motor efficiency = 65.4 % Electricity cost is 5.25 Rs /kWh Annual energy cost = (5.5*7920*5.25) / .654 = 3,49,678 Rs NEW COMPRESSOR: Discharge pressure = 7 kg/cm 2

Motor Kw = 4 kW/5.5 hp FAD = 670*10 - 3 m 3 /min Annual Energy cost = (Full load power (kW) *annual running hours* electricity cost/kWh) / motor efficiency Full load power = 4 kW Motor efficiency = 85 % Electricity cost is 5.25 Rs /kWh Annual energy cost = (4*7920*5.25) / .85 = 1, 95,670 Rs Cost savings = 1,54,008 Rs The cut-off pressure of old compressor is at 6Kg/cm2 but the discharge pressure is 9kg/cm2.So that much power can be saved. The new compressor should have discharge pressure near to the cut-off of compressor. Also the newly replaced system have better motor efficiency than the old one.

Air Powered Engine

Prof. B.S.Patel et al. tried to develop a compressed air engine by modifying an 4-stroke, single cylinder SI engine by replacing the spark plug with a pulsed pressure valve, and using compressed air as the working fluid. The working of the engine is explained theoretically and the cost analysis is made which shows that the compressed air engine is cheap when compared to the conventional SI engine.

2.6 STUDY OF COMPRESSED AIR STORAGE SYSTEM

Clean Potential Energy for 21st Century Dr. Bharat Raj Singh and Dr. Onkar Singh conducted an experiment in which they used a vaned type novel air turbine as a prime mover for a motor bike. In this experiment they tried to gain an output of 6.50 to 7.20 HP for the starting torque requirements of 500 to 750 rpm at 4 to 6 bars air pressure to running speeds of 2000 to 3000 rpm using 2 to 3 bars air pressure. The test was conducted in HBTI Kanpur and below diagrams shows the test rig setup and its layout: It consists of an air compressor which was used to produce and store 300 psi (21 bar approx.) air and use it to impact the compressed air on the vanes of the novel air turbine. The test was conducted at different inlet pressures and the efficiencies of the turbine was found to vary from 72 to 97 %. The turbine had d/D ratio of 0.7 and the results obtained were are follows: \Box 93% to 99% with variation of 6%, at speed of rotation 500 rpm for injection pressure 20 psi to 100 psi. \Box 81.8% to 89.8% with variation of 8%, at the speed of rotation 1000 rpm for injection pressure 20 psi to 100 psi. \Box 70.8% to 84.3% with variation of 13.5%, at the speed of rotation 1500 rpm for injection pressure 20 psi to 100 psi. \Box 64.4% to 79.8% with variation of 15.4%, at the speed of rotation 2000 rpm for injection pressure 20 psi to 100 psi. 59.5% to 76.5% with variation of 17%, at the speed of rotation 2500 rpm for injection pressure 20 psi to 100 psi. \Box

56.2% to 72.9% with variation of 16.7%, at the speed of rotation 3000 rpm for injection pressure 20 psi to 100 psi. A graphgiven below was drawn for comparing Actual power with respect to theoretical power and the Speed of rotation in rpm: After conducting this research they have concluded that overall performance of air turbine for working pressure ranging from 2.7-6 bar is found varying from 72%-97%. This technology canbe used in the future automotive industry.

Kanwar J.S Gill, Surinder Pal Singh, Gurpreet Singh &Malinder Singh, "Designing and Fabrication of Intercooler and Control of Three Phase Digitalized Reciprocating Air Compressor Test Rig with Automatic Control Drive Unit", International



Conference of Advance Research and Innovation (ICARI-2015). The Air Compressor Test Rig is designed to study the characteristics of a Two Stage Reciprocating Air Compressor and the compressed airflow through flow arrangement. This unit is selfcontained and fully instrumented with mild steel frame-mounted on raised foundation, with intercooler, air stabilizing tank and air receivers. The compressor is driven by an AC Motor. To provide adequate cooling to the system, the function of the intercooler is supplied with pressure and temperature measuring instruments at the inlet and outlet. With the introduction of intercooler the volumetric efficiency has been increased to 100 %. In order to measure the air flow rate air stabilizing tank should stabilize the flow of air which is mandatory in this work. Actual volume of free air delivered by this compressor is 0.020 m3/sec with a work done of 77

N-m was the result obtained during test. Moreover it was also found that the capacity to deliver air is about 1.02 kg/minute of the compressor, when the isothermal efficiency of the compressor is 45 %. If an intercooler is specially designed it has capacity of 2.049 kilojoules/kg of heat rejection. Vijaykumar F Pipalia, Dipesh D. Shukla and Niraj C. Mehta, "Investigation on Reciprocating Air Compressors - A Review", International Journal of Recent Scientific Research Vol. 6, Issue, 12, pp. 7735-7739, December, 2015. Heating is an undesirable effect of the compression process at least as far as compressors are concerned and heat transfer is nature's way of driving systems towards stability. This has not only provided food for thought for researchers trying to understand its influence and quantify its effects, but also challenged designers to mitigate its impact and develop safe and efficient designs. Also this investigation is concerned with improving the efficiency of Two Stage Reciprocating Air Compressor by providing water cooling source, radiator coolant and ethylene glycol. The experiments with air, water and different inter coolants are performed on a Two Stage Double Cylinder Reciprocating Compressor System. SuprasannaRaoRavur, Subbareddy. E. V. Investigation "Experimental to Increase the Efficiency of the Air Compressor by Changing the Coolants in Inter Cooler", International Journal for Research in Applied Science & Engineering Technology Volume 3 Issue IX, September 2015.

The compressed air usage is increasing quicklynow-a-days. But the efficiency of compressor is low due to many reasons like location, elevation, length of pipe lines, intercooler performance, even atmospheric conditions also effects the efficiency of the compressor, which increases the power consumption of the compressor. The inter cooling is the best method to reduce the coolant. In this study we are extending the investigating by changing the temperature of water and mixing of different types of the coolants in water at different proportions. The selection of the coolants depends upon their properties like miscibility, self-ignition temperature, boiling point and exploding range. For this investigation ethylene glycol and glycerol as coolants and a two stage reciprocating air compressor fitted with shell and tube type heat exchanger is selected. This investigation shows the good arguments between the water, glycerol and ethylene glycol .KuldeepTyagi,&Er. Sanjeev Kumar, "Improved Air Compression System", International Journal of Scientific Engineering and Applied Science, Volume 1, Issue 5, August 2015. Intercooling of Air Compressors is necessary for increasing its efficiency. A shell and tube type of heat exchanger is particularly suitable as an intercooler between two compression stages of a compressor. A characteristic of heat exchanger design is the procedure of specifying a design, heat transfer area, pressure drops and checking whether the assumed design satisfies all requirements or not. The purpose of this research paper is to provide an easy and efficient way to design an intercooler for air compressor. This paper describes modeling of heat exchanger which is based on the minimization of heat transfer area and a flow chart is provided showing the designing procedure involvedWadbudhe R. C., AkshayDiware, Praful kale, "A Research Paper on Improving Performance and Development of Two Stage Reciprocating Air Compressor", International Journal of Research In Science & Engineering, Volume: 3 Issue: 2 March-April 2017. The two Stage Reciprocating Air Compressors is the mostly used type of compressor found in many industrial applications such as crucial machine in gas transmission pipelines, petrochemical plants, refineries, etc. Since there is requirement ofhigh pressure ratio, reciprocating air compressor is commonly used in locomotives. After certain period of time, unexpected failures of internal components due to miscellaneous reasons occur, which inversely affects the performance of operating system. It is

essential to establish the recommended clearances mentionedfor the various parts of the compressor. Compressor parts selection between repair and replacement is done on the basis of Dimensional Measurement whichleads to easy maintenance in economical point of view.

Pawan Kumar Gupta1, S.P.Asthana2, Neha Gupta, "A Study Based on Design of Air CompressorIntercooler", International Journal of Research inAeronautical Mechanical and Engineering, Vol.1 Issue.7, November 2013, Pages: 186-203 ISSN (ONLINE): 2321-3051. This paper presents a study on which the main objective isintercooling of air compressor which is necessary for an efficient process. Basically increase in pressure is a result of reduction of a specified volume which is also known as compression. This paper mainly discuss about reciprocating compressor which is widely used for air compression. Compression is done in more than one stage and between each stage intercooler is provided to improve the efficiency of the system. 7. Vishal P. Patil, Shridhar S. Jadhav, Nilesh D. Dhas, "Performance and Analysis of Single Stage Reciprocating Air Compressor Test Rig", SSRG International Journal of Mechanical Engineering, Volume 2, Issue 5, May 2015, ISSN: 2348 - 8360. An experimental test rig has been built to test reciprocating compressors of different size and capacity. The compressors were tested with the help of air as a working fluid .The paper provide us with much needed information regarding the efficiency of the compressors operating under the same conditions with the same system parameters. This paper also highlights reports on investigation carried out on the effect of pressure ratio on indicated power, isothermal efficiency of both compressors.

The result shows that the indicated power is increasing as the discharge pressure increases, but the isothermal efficiency of both the compressors decreasing with increase in pressure ratio. Both compressor types exhibit the same general characteristics with respect to system parameters. When the experiment was carried out for constant angular speed of compressor, no change in volumetric efficiencies observed. In addition, a comparative study was carried out for two compressors and their differences were analysed. To verify the model's goodness with the aim of predicting the compressor performance, the study seems to be useful

The fig shows Air CompressorType VT4 is a Two Cylinder, Two Stage Reciprocating Air Compressor. TheHigh Pressure Cylinder is getting heated up and can be viewed with the naked eye that the High Pressure Cylinder is becoming red in colour when it is made to run for longer period of time.

2.7 THE ROLE OF MONITORING

Monitoring and tracking system performance itself does not improve energy efficiency (Radgen and Blaustein, 2000). Nevertheless it is often the first step in improving energy efficiency for two basic reasons. Measuring air use and energy consumption is essential in determining whether actions proposed in improving energy efficiency is cost effective. Tracking of system performance is a valuable tool to detect performance degradation, or change in the nature or quantity of air use.

2.8 PRESENT STATE OF THE ART OF MONITORING RECIPROCATING COMPRESSORS

Productsavailable commercially for monitoring of reciprocating compressors provide functions for machine protection, condition monitoring and performance monitoring. For condition monitoring of the compressor the mechanical condition of the compressor is assessed by measuring parameters such as frame vibration, piston rod drop, suction and discharge valve temperature, main bearing temperature cross head

acceleration, multi event Key- Phaser signal and cylinder pressure. These systems have alarm and interlocking feature to protect machines in the event abnormal of occurrence of conditions. The reciprocating compressor monitoring products available in the market are expensive. Moreover the practices of continuous monitoring of reciprocating machines have not gained the same level of acceptance as compared to centrifugal compressors It is observed that in general continuous monitoring systems are installed in reciprocating compressors having capacity >350kW and especially in those used in hydrocarbonprocessing. Reciprocating air compressor systems ranging from 30 to 350 kW installed in various industries are far too many and the performance monitoring needs to be done periodically and using off-line methods which are reliable quick and cost effective.

2.8 PERFORMANCE INDICATING PARAMETERS OF RECIPROCATING AIR

The parameters to be monitored to asses the performance of the reciprocating compressors is grouped in to two categories. A parameter which indicates the overall performance of the compressor is called vital parameter. The categories of parameters that are used to identify the reasons for degradation of performance are called diagnostic parameters. In practice vital parameter should be measured periodically and its value is compared with bench mark values. If the value of the vital parameter deviates from the bench mark value then the diagnostic parameters needs to be measured to identify problem in the compressor.

2.8.1 SPECIFIC POWER CONSUMPTION

Specific Power Consumption is the vital parameter for assessing the overall performance of reciprocating aircompressors. Specific power consumption is defined as power consumed by the compressor to deliver one cubicmeter of free air per minute at rated pressure, which is expressed in units kW/(m3/min.). In imperial units it is expressed as kW/100cfm. FAD of the compressor and electrical power consumption are measured for calculating specific power consumption.

2.8.2 POWER LOSSES

Power losses are the indicators of how the power is lost in various parts of the compressor system. It can pinpointthe cause for degradation of compressor's overallperformance.1422 Page 3International Compressor Engineering Conference at Purdue, July 14-17, 2008

2.8.2.1 Electrical losses

It is the difference between the electrical power supplied to the electric motor and mechanicalpower supplied to the compressor crank shaft. The electrical losses indicate the energy conversion efficiency of thedrive motor and energy transmission efficiency of the belts and couplings.

2.8.2.2 Mechanical Losses

It is the difference between the mechanical power input to the compressor and indicatedpower of compression. The mechanical losses account for the frictional losses in the compressor system and powerrequired to drive accessories like lubrication pump and journal bearing oil pump.

2.8.2.3 Thermodynamic losses

It is the difference between actual indicated power and ideal or theoretical powerrequired for compression. The thermodynamic losses indicate the losses due to valve resistances offered to the flowand loss due to the deviation of compression and expansion processes from ideal process. To calculate the power loss parameters Electrical power consumption, Mechanical power consumption, Incylinderpressure and Instantaneous swept volume measurements are required.

2.8.3 AIR FLOW CAPACITY LOSSES

Air flow capacity losses are components that reduce the FAD of compressors. Capacity loss occurs due to reexpansion f high pressure air left in the clearance volume at the end of discharge process. The assessment is madeby measuring the volumetric efficiency of the compressor, which is the ratio of actual volume of air delivered andswept volume of the compressor. The capacity losses also occur due to leakages in the compressor and in the flangesand piping systems. FAD of the compressor and leakage rate measurements required for estimation of airflowcapacity losses.

METHODOLOGY

3.1 MODIFICATION OF AIR COMPRESSOR SYSTEM

In the manufacturing plant, compressors are working at corrosive and high temperature conditions. The intake air may contain dust which in turn decrease the efficiency of the compressor. Low maintenance and continuous service are extremely important in this area. So upgrading the design of the compressor unit can increase the reliability, safety and overall efficiency of the screw compressor .The effect of intake air on compressor performance should not be underestimated. Intake air that is contaminated or hot can decrease compressor performance and result in excess energy and maintenance costs. If moisture, dust, or other contaminants are present in the intake air, such contaminants can build up on the internal components of the compressor, such as valves, impellers, rotors, and vanes. Such build-up can cause premature wear and reduce compressor capacity. When inlet air is cooler, it is also denser. As a result, mass flow and pressure capability increase with decreasing intake air temperatures. Conversely, as the temperature of intake air increases, the air density decreases and mass flow and pressure capability

decrease. The resulting reduction in capacity is often addressed by operating additional compressors, thus increasing energy consumption. To prevent adverse effects from intake air quality, it is important to ensure that the location of the entry to the inlet pipe is as free as possible from ambient contaminants, such as rain, dirt, and discharge from a cooling tower. If the air is drawn from a remote location, the inlet pipe size should be increased in accordance with the manufacturer's recommendation to prevent pressure drop and reduction of mass flow. All intake air should be adequately filtered. A pressure gauge indicating pressure drop in inches of water is essential to maintain optimum compressor performance

3.2 PLACEMENT OF PREFILTER

Compressors are sometimes installed in environments where there is a great deal of airborne dirt and dust, depending on the products manufactured by our customers. So the compressors are designed to eliminate dirt by installing filters on compressor air intakes. In good environments, these filters can until without maintenance, operate auxiliary equipment inspections are performed. However, in dusty environments, the filter quickly becomes clogged, and it is necessary to stop the compressor in order to clean the filter.



Shows the pre-filter mounted on the compressor.it is more efficient than intake air filters because it can be cleaned while the compressor is running Placement of pre-filters on compressors

When an intake air filter is located at the compressor, the ambient temperature should be kept to a minimum, to prevent reduction in mass flow. This can be accomplished by locating the inlet pipe outside the room or building. When the intake air filter is located outside the building, and particularly on a roof, ambient considerations are important, but may be less important than accessibility for maintenance in inclement or winter conditions. A compressor intake air filter should be installed in, or have air brought to it from a clean, cool location. The compressor manufacturer normally supplies, or recommends, a specific grade of intake filter designed to protect the compressor. The better the filtration at the compressor inlet, the lower the maintenance at the compressor. However, the pressure drop across the intake air filter should be kept to a minimum (by size and by maintenance) to prevent and a reduction in compressor capacity. A pressure differential gauge is one of the best tools to monitor the condition of the inlet filter. If the pre-filter is installed on the compressor it removes the dust from the intake filter.it can be cleaned while the compressor is running. Additional auxiliary equipment's such as pressure gauge are not needed to measure the inlet pressure drop. Continuous and smooth air flow is provided. The air flow will be maximum at the output of the compressor.

3.3 ANALYTICAL METHODOLOGY

Volumetric Efficiency = η vol= (Va/Vs)*100 Actual work done W= WL.P + WH.P WL.P=n/n-1{P1V1[(P2/P1)n-1/n-1]}joules/cycle WH.P=n/n-1{P2V2[(P3/P1)n-1/n-1]} joules/cycle Indicated power (I.P)= actual work done in joules/cycle*N/60 Isothermal work(Wiso)= P1V1{loge[P3/P1]} joules/cycle Iso-thermal power = isothermal work in joule *N/60 Iso-thermal efficiency(ηiso-thermal)= (isothermal power/indicated power)*100

3.5 MEASUREMENTS

For performance monitoring of compressors FAD of air compressor, Electrical power consumption, Mechanical Power consumption, Indicated Power consumption, and leakage rate are to be measured. There are number oftechniques available for measurement these parameters. However, only the techniques that are suitable for offlineinsitu measurements are presented in following sections

3.5.1 FAD MEASUREMENT

Air flow in reciprocating compressor is pulsating in nature. The FAD of a reciprocating compressor is the timeaverage of the pulsating flow. In conventional differential pressure measuring devices like orifice meters andventurimeters, the pressure drop across the flow element $\Box p$ is proportional to the square of the flow rate. If these devices where subjected to pulsating flow measurement, the time average of pressure drop $\Box p$ indicated will behigher than the true value. Therefore in pulsating flow measurements the average flow rate indicated by these evices will be higher than the actual flow. Additionally if the response time of the manometer used to measure the pressure drop $\Box p$ is more than the pulsation period of the flow, square root error arises. In laboratoriesthese devices re being used to measure the FAD of reciprocating compressors by damping the pulsations of the air flow usingacoustic filters. Plint and Martyr (1995) give a formula to calculate the volume of single chamber acoustic filter. It is found that the volume of the single chamber pulsation filter required for a double acting reciprocating compressorwith swept volume of 3.15 liters is 460 liters, which is 145 times the compressor swept volume. Because of its largesize, acoustic filters are not suitable for industries. In a compressor installation where in number of compressors areto be tested, a bulky

acoustic filter cannot be moved from one compressor to other during air flow measurement.

Therefore conventional differential pressure producing flowmeters are not suitable for insitu FAD measurement.Laminar flow meters are used measuring pulsating flows. In viscous (laminar) pipe flow the pressure drop varies asfirst power of the velocity; hence the time average of the pressure drop is equal to the pressure drop correspondingto the mean flow rate. This characteristic makes it quite suitable for measuring pulsating flows. Laminar flow metersare expensive and bulky. The compressed air flow meter shown in Figure 2, is a device wherein array of various size critical flow nozzles areused. The specifications of the compressed air flow meter is given in Table 2. The flow through a sonic nozzle isconstant for a particular upstream pressure as long as the pressure ratio across the nozzle is maintained below thecritical pressure ratio of air. During the conduct of the test the compressor system is isolated from the industrial distribution network by closing outlet valve of the receiver. Then compressed air flow meter is installed in the airreceiver. Compressor under test is started; nozzles in the compressed air flow meter are opened in a particular combination till rated pressure is maintained in the receiver. At this condition the flow rate of the compressor is1422 Page 4International Compressor Engineering Conference at Purdue, July 14-17, 2008equal to the flow through the compressed air flow meter. By comparing the compressed air flow meter readings withlaminar flowmeter it is found that the error is with in 2% of FSR. Compressed air flow meter is suitable formeasurement of FAD of а reciprocating compressor.

Table 0 Specifications of compressed Air flowmeter Make M/s Compessed Air Management Impact RM Inc., Canada Model LP150 Range 800cfm Figure 1 Compressed air flow meter

3.5.2 ELECTRICAL POWER MEASUREMENTS

Electrical power consumption of a motor can be measured insitu using portable power meters. However whenmotors are driven by variable speed drives, the wave forms of voltage / current are distorted. In these installationspower measuring instruments which are capable of measuring power harmonics are to be used.

3.5.3 MECHANICAL POWER MEASUREMENT

Mechanical power is computed from torque and speed measurements. Measuring torque of reciprocatingcompressors installed in industries for offline monitoring is very difficult. Standard torque transducer models thatare readily available require substantial modifications to fix them on already installed compressors.

An innovative installation design, shown in Figure 2, developed by the authors facilitates the mounting of torque transducer in the existing machines with little modifications. The design significantly reduces the complexities involved in mounting and removing the torque transducer in the industrial compressors. The torque transducer used for the research work is inductively powered and transmit data using telemetry.

3.5.4 INDICATED POWER MEASUREMENT

Indicated power of a compressor is computed from the p-V diagram of the compressor. In-cylinder pressure and instantaneous swept volume measurements are required for drawing p-V diagram. Dynamic pressure transducersare used for measuring in-cylinder pressure. Medium and small capacity reciprocating compressors do not haveports to mount pressure transducers. It is recommended that manufacturers provide compressor ports for installingpressure transducers in the compressors at suitable locations. These pressure transducer ports can be used formounting pressure transducers during measurements and plugged when not needed. If the pressure transducer ports1422 Page 5International Compressor Engineering Conference at Purdue, July 14-17, 2008are not readily provided in the compressor indicated port assembly can be used or bedrilled at suitable locations. ports can A piezoresistive transducer is suitable for measurement of in-cylinder pressure.Piezoresistive transducers are preferable to piezoelectric transducers as they measure the absolute pressure directlyand therefore pegging is not necessary. Moreover the temperature involved in compressors is within the operatingrange of standard piezoresistive transducers.

International Compressor Engineering Conference at Purdue, July 14-17, 2008 Crank angle encoders are used presently for measurement of instantaneous swept volume of reciprocating machines. The crank angle encoders are delicate devices that require special care in installation. Construction of some of the compressors does not provide access to fix crank angle encoders, some times necessitates significant modifications also. These problems render crank angle encoders unsuitable for offline monitoring of industrial compressors. Moreover the output signals from the crank angle devices are to be post processed to convert crank angle to instantaneous volume by computing kinematic relation of the slider crank mechanism used in the reciprocating compressor.

To facilitate insitu measurement of instantaneous volume swept of reciprocating compressors new device is developed а (pannirselvam et al., 2004). This device shown in Figure 4 consists of a template cam and a non contact laser displacement sensor. The template cam has profile cut according to relationship between crank angle and swept volume. The cam is permanently fitted on the compressor crank shaft suitably and rotates in unison with the compressor shaft. The non contact laser displacement sensor measures the profile of the cam dynamically and provides voltage output proportional to the cam profile and therefore to instantaneous swept volume.

industrial compressors, inexpensive In template cam is permanently fitted on the compressor. When the measurement of instantaneous swept volume is needed, the laser displacement sensor is targeted on the cam. Therefore complicated installation is done away with. The device directly provides voltage analogous to the swept volume therefore post processing electronics is not necessary. An oscilloscope operated in X-Y mode gives p-V plot. In installations having multiple compressors, a single laser displacement sensor is enough to acquire the p-V diagram of all the compressors offline. Actual and theoretical indicated power are calculated from p-V diagram by numerical integration.

3.5.5 MEASUREMENT OF LEAKAGE RATE

Leakages from a compressor system are a cause for capacity loss. For quantifying leakage rate from a compressed air system two methods are found suitable. They are pump down test method and ON-OFF time method. During leakage tests, compressor is isolated from distribution system suitably. Pump down test method quantifies the leakage by measuring time required for pressure to drop from a high level to lower level. ON-OFF time method is suitable for reciprocating compressors fitted with constant speed control. After isolating the compressor and receiver from the distribution, compressor is started and the period during which the compressor runs at loaded condition (ON time) and unloaded condition (OFF time) is noted. Since the compressor is isolated from the distribution, Itruns in loaded condition only to meet the leakages in the system. The leakage rate is given as the ratio of ON time and total time. The leakage is expressed in percentage of rated capacity of compressor.

Progress in Experimentation

Any detailed modeling of the heat transfer process would require a detailed knowledge of the temperature variations generated by the dynamics of the valves and their interaction with the cylinder and piping. Optical sensing techniques using laser induced fluorescence have an advantage of being a noncontact type, while providing a faster response also. On the experimental side, as reported in the previous section, Brok et al. (1980), and Gerlach and Berry (1989) as well asJacobs (1976) have done some heat transfer measurements in order to estimate the effect of suction gas heating on volumetric efficiency. Measurement of temperature and heat flux insidethe cylinder of a reciprocating compressor is not an easy task. It is perhaps this reason, as well as the lack of instrumentation, which constrained earlier model developers (with the exception of Brok et al.) to validate their simulations by using only the PV cards to compare with experimental pressure - time traces. Heat transfer has only an indirect effect on the P-V card andhence such validations would not be a good test for the heat transfer models. This necessitates measurement of in-cylinder temperature - time traces for developing such models. Severalinvestigators including the author (see Shiva Prasad, 1992a, b), etc. have attempted to make such measurements in order to further understand and model the in-cylinder regenerative heat transfer process. Although these measurements were able toVijaykumar .FPipalia, Dipesh D. Shukla, and Niraj C. Mehta., Investigation On Reciprocating Air Compressors A Reviewresolve the time scales enough to identify the phase differences between near wall temperature gradients and heat transfer rates, the spatial resolution was not enough to resolve the lengthscales. Hence more work needs to be done in order to help develop as well as validate in-cylinder heat transfer models. Also, as stated in the previous section, temperature and heattransfer measurements are required in the cylinder passages as well as valve chambers and passages for developing heat transfer correlations.

Experimental Setup Description

Two stage single acting reciprocating air compressor with shell and pipe type intercooler used. Electricity Supply: Single Phase, 220 VAC, 50 Hz, 5-15 amp sockets with earth connection. Water Supply: Continuous @ 2 LPM at 1 bar. Floor Area Required:

1.5 m x 0.75 m. Setup specification Bore diameter, d = 0.0935 m Length of stroke, L = 0.078 m Diameter of orifice, do = 0.011 m Diameter of pipe, dP = 0.022 m Density of water, ρ m = 1000 kg/m3 Density of air, ρ a = 1.21 kg/m3 Co-efficient of discharge of orifice, Cd = 0.64 Energy meter constant, E.M.C= 3200 pulses / kWhr Atmospheric pressure, Pa = 1.03327 x 1005 N/m2 Radius of swinging Field Dynamometer, R = 0.16 m RPM of motor, Nm = 1440 Rpm Compressor Development Issues

One can easily visualize what the direct impact of heat transfer would be on material temperatures. Large material temperatures could be attributed to large discharge temperatures. This is indeed the case in vacuum pumps and other high pressure ratio reciprocating compressors operating at pressure ratios of 30 - 40, resulting in isentropic temperatures as high as 1200°F at the end of the compression stroke. Even in low pressure ratio compressors, suction gas heating, if not keptunder control would lead to higher inlet temperatures which gets amplified to large discharge temperatures. Suction gas heating occurs in many ways. Unloading one of the ends willresult in pushing the gas back and forth through the unloaded end many times which will tend to heat up the gas. This hot gas will mix with the fresh charge and eventually enter the loadedend at a temperature much higher than the inlet temperature in the suction piping. Since the suction gas entering the loaded end has become hotter, the discharge temperature will be muchhigher than for a cylinder with both ends loaded. The extent of suction

gas heating in this case depends on various factors like speed of the compressor, which determines the residence timeof the gas entering the unloaded end in the suction passage; ratio of the swept volume to the cylinder suction passage volume, which determines the amount of gas pushed from theunloaded end which enters the loaded end during the subsequent stroke; number and location of the suction ports with reference to the inlet nozzle, which determines the flowpattern into the loaded end, etc. In addition to unloading an end, suction gas heating could also be caused by leakage past piston rings and valves, which is of serious concern in highpressure ratio compressors, and : frictional heating near the rider bands and packing rings, which are of main concern in nonlubricated, high speed compressors. Although there is no disagreement about the harmful effects of such high discharge gas temperatures, there is still someskepticism in the compressor community about the extent of its impact on performance. Such skeptics would perhaps be satisfied by just getting rid of this heat by providing efficientcooling or developing materials which could withstand such high temperatures. This has spurred lot of research in material technology to develop materials particularly for valves, riderbands, piston and packing rings, etc. Research is also being done to develop new coatings for piston rods, cylinder liners, etc., to reduce the wear and withstand the high temperatures. Ideally, from the suction gas heating point of view, one would like to have materials/coatings for suction passages and valves which would behave as perfect insulators. Cylinder liners and other internal cylinder boundaries should act like diodes and transport heat only outwards during parts of the cycle when the gas inside becomes much hotter than the cylinder walls. In fact, development in material technology and smart which can actively adjust to materials the environment is occurring at such afast pace, the author can easily envision development of materials with sandwiched spaces between them, which can be filled with a highly conducting or insulating fluid using activecontrol, depending on the temperature gradient between the gas and the cylinder wall, which can function like diodes. The high temperature environment in compressors has also generated theneed and interest for developing lubricants for withstanding such high temperatures inside the cylinders of lubricated compressors. If external cooling is to be provided, particularlyin non lubricated cylinders, the mechanical aspect of design also becomes complex in many cases because of the complexity of the cooling passages_ Further, in multistage compressors, inter cooling becomes a necessity in many cases and in all compression systems, after cooling has to be done to meet the user's requirements. Even from the reliability point of view, high temperaturescause serious concern. For example, impact strength of a valve plate or the sealing effectiveness of a piston or packing ring depends on the temperature. Cylinder lubricants may not just lose their viscosity, but also may break up leading to deposition on valve plates and passages causing inefficient operation or even failure. Any such deposition of a lubricant, or refrigerant in refrigeration compressors, or water /liquid droplets I slugs in compressors operating at relatively low temperatures, would pose a further challenge to development of reliable wall heat transfer correlations. In addition, the lubricant may also vaporize and contaminate the discharge gas thus affecting the downstream process, unless the oil vapors are removed by using expensive filters. It is also not uncommon for cylinders and pistons to loose their concentricity because of poor lubrication, leading to operational failure and even irreparable damage to the components as a result of ceasing of the parts. Finally from a designer and developers point of view, no amount of progress in analytical modeling or experimentation would help, unless quick and easy methods are developed for accurately predicting discharge the temperature, losses

attributable to suction gas heating and its impact on volumetric efficiency and power economy. However, users of such methods should also be aware of their limitations, since reliable and accurate methods with universal applicability to all types of compressors are unlikely to be developed. Hence these analytical methods could only be used as tools for reducing thecycle time for development and one will have to rely on testing for proving the design.

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4 RESULTS AND DISCUSSION

Heat transfer changes its direction during the cycle of crankshaft depending on the gas temperature inside the cylinder. When the gas temperature is lower than the wall temperature, heat flux is positive, which means that the heat is transferred from the wall into the gas. When the compressed gas reaches the same temperature as the wall, heat flux is zero and after that it changes direction, meaning the heat is transferred from the gas into the wall. This change of direction occurs not just during the compression, but also during expansion process (Figure 3). The negative effect of heat transfer inside the cylinder is the superheating of the sucked gas, resulting in decrease of compressor efficiency.

The simulation procedure started with the expansion process moving the piston from top dead center towards bottom dead center. When the pressure force acting on suction valve overcomes the spring force, suction process begins. Usually the suction process is ended when the piston is in its bottom dead center followed by the compression and discharge. Movement of the discharge valve is again controlled by pressure and spring force. A special care must be taken when the valves are opening and closing in numerical simulation. Keeping the mesh in the gap between the valve and the seat is fundamental for CFD simulation and therefore it is not possible to close/open valves continuously, but when the gap is less than 0.01 mm, suction or discharge process is terminated.

Numerical simulation of heat transfer requires high quality mesh, especially close to the walls, where the heat transfer coefficient is calculated. The value of y+ should be kept around 1, however, when the fluid flow is not static, it is difficult to keep the high mesh quality and not to enormously increase the computational requirements. In this simulation the value of y+ was checked on each surface during the whole simulation and the average value was mostly below ten. Also the convergence of the momentum equations, continuity equation and energy equation did not overcome the 10-4 criteria.

The analysis of Müllner and Bielmeier (2008) shows that most of the heat flows between the gas and piston (around 50 %), e.g. it is 35 % for cylinder head and less than 5 % goes into the cylinder wall. The rest of the heat is transferred in out-walls, which are walls around the valve. Similar results were obtained in this paper despite the fact that boundary conditions and settings were not described in Müllner and Bielmeier (2008). The results in Table 4 show the area integrated heat fluxes during one cycle of crankshaft.

CFD analysis provides with more detailed flow field inside the cylinder including the velocity and temperature distribution, which helps to divide the heat flow more accurately among the surfaces. On the other hand, the computational cost is enormous, which is not convenient during the development process. Due to the complicated use of CFD, the objects of interest are simplified tools, such as the one presented in section 3.1. The results of complex analysis can be used in 0d model, which uses integral correlations to predict heat flux. However, it is distributed just according to the current heat transfer area. As the heat transfer area is the same for all the models, distribution of heat flux is consequently almost the same as well. The values used in

Good agreement was found for the piston surface, where the same trend can be observed – minor share on total heat flux during suction and compression and higher during discharge and expansion process. The heat flux through the cylinder wall is overestimated. The surface of cylinder wall does not play such an important role as it is predicted by 0d model, especially during the suction process. CFD predicts lower heat flux through the cylinder wall than 0d simulation and vice versa for cylinder head. During discharge process the heat flux from cylinder wall is higher again at the expense of the piston. All of this could be the consequence of high velocities of gas around the cylinder head and piston, which increase the heat flux on these surfaces.

The comparison of results from the 0d model and the CFD analysis shows that one integral correlation for all surfaces inside the cylinder is not sufficient to properly describe distribution of heat flux. The same idea of using different correlation for each process of compressor was already applied by Disconzi et al. (2012), therefore it seems reasonable to use different correlation for each surface inside the cylinder as well. Usually there are three main surfaces, as it is displayed in the

Models mentioned in section 2.1 show important differences, as it was mentioned in Tuhovcak et al. (2015) and finding correct one is a difficult task. However, some correlations show better agreement with numerical simulations than others for particular surface. Therefore, the combination of well-known correlations could help to achieve higher accuracy in heat flux predictions for simplified models used for compressors.

The comparison is divided according to heat transfer areas - head, piston and wall and shows area integrated heat fluxes. Integral correlations offer only one calculation of heat transfer coefficient for all surfaces, therefore the curve produced by each correlation is the same for head, piston and wall. On the other hand, the results from numerical analysis show area integrated heat fluxes at each surface calculated from local heat transfer coefficient. Discontinuities occurring at curves from CFD are the consequence of remeshing and interpolation results from previous time step on new mesh. Even with more iteration within one time step they did not disappear. Most of the models underestimate the heat fluxes, especially during the suction phase and at cylinder head. The same problem occurs for the piston surface, only the model of Annand predicted results in reasonable agreement with CFD simulation. For the surface of cylinder wall, only the model of Disconzi shows a sufficient accuracy. The model of Adair predicts much lower heat flux for all surfaces, especially during the suction and discharge phase.

Comparison of heat flow for particular empirical correlations and numerical simulation As it was mentioned before a combination of known models could help to increase the accuracy of heat flux prediction in simplified tools. Each process and surface would have its own correlation, based on the comparison with CFD numerical simulation. The combination of models based on presented results can be found in Table 7. On the left side there is a ratio of heat flux predicted by correlations and CFD. On the right side there are models which showed best agreement with CFD simulation. In most of the cases the models of Annand and Disconzi appear, however there is also contribution of Adair and Woschni. The results show a big difference between integral correlations and numerical model for heat transfer

prediction in CFD. More investigation is necessary, especially in CFD, to perform simulation with higher accuracy. It must be pointed out, that all of the presented models were developed either for combustion engines (Woschni, Annand) or for refrigeration compressor (Disconzi, Adair). The compressor used in this paper was working with air, which could have influenced the results. Another important thing is the CFD simulation, particularly the simulation of heat transfer, which is very demanding on mesh quality close to heat transfer surface.

5.CONCLUSION

Two Stage Reciprocating Air Compressor is gone through different intercooling processes, it can be concluded that more the surface area of intercooling more will be depression in temperature of air which will directly result in improving the efficiency of the Air Compressor. From all the results of intercooling processes, it can be concluded that the radiator intercooling with increase in size of intercooler results in better volumetric efficiency as compared to other type of intercooling. In operation of Two Stages Reciprocating Air Compressor it is possible that when costs of different parameters are considered, the method adopted by us can be used for improvement in its overall efficiency. Energy cost of a compressor is approximately 75% of its life cycle costs. Performance monitoring is a valuable tool to detect performance degradation of compressor during operation. It helps to keep the energy cost in check. Even though in large reciprocating compressors online performance monitoring systems have been installed, in medium and small capacity industrial compressors installation of on line systems is not economical. In medium and small capacity compressors, performance should be monitored periodically using off-line monitoring methods. Various parameters that are indicators of the performance of air compressors are defined. It is not

necessary to monitor all these parameters at regular intervals; instead it is enough to monitor vital parameter called specific power consumption periodically. If the specific power consumption valve decreases below the bench mark limit diagnostic parameters needs to be measured. Diagnostic parameters help to identify the power loss making components and airflow capacity loss making components. Various measuring techniques that are suitable for offline measurement of FAD of a reciprocating compressor, Electrical power consumption, Mechanical power consumption, Indicated power consumption and leakage rate are discussed. In this paper we have studied about Two Stage Reciprocating Air Compressor and one of its main component which is used in air compression system i.e. intercooler. Also, we have used very simple and time efficient algorithms for designing of intercooler for Air Compressor.

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