Optimization of Auxiliary Power Consumption of Combined Cycle Power Plant

By

Tejas Raval

08MMET08

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

May 2010

Optimization of Auxiliary Power Consumption of Combined Cycle Power Plant

Major Project

Submitted in partial fulfillment of the requirements

For the degree of

Master of Technology in Mechanical Engineering (Thermal Engineering)

By

Tejas Raval

08MMET08

DEPARTMENT OF MECHANICAL ENGINEERING

INSTITUTE OF TECHNOLOGY

NIRMA UNIVERSITY

AHMEDABAD-382481

May 2010

Declaration

This is to certify that

i) The thesis comprises my original work towards the degree of Master of Technology in Mechanical Engineering (Thermal Engineering) Engineering at Nirma University and has not been submitted elsewhere for a degree.

ii) Due acknowledgement has been made in the text to all other material used.

Tejas Raval

Certificate

This is to certify that the Major Project entitled "Optimization of Auxiliary Power Consumption of Combined Cycle Power Plant" submitted by Tejas Raval (08MMET08), towards the partial fulfillment of the requirements for the degree of Master of Technology in Mechanical Engineering (Thermal Engineering) of Nirma University, Ahmedabad is the record of work carried out by him under my supervision and guidance. In my opinion, the submitted work has reached a level required for being accepted for examination. The results embodied in this major project, to the best of my knowledge, haven't been submitted to any other university or institution for award of any degree or diploma.

Mr.Shailendra Saha
Guide,Joint General Manager,
Maintenance Department,
BPOL,
Essar Power,Hajira, Surat

Dr. R N Patel
Guide, Professor,
Mechanical Engineering Department,
Institute of Technology,
Nirma University, Ahmedabad

Prof V R Iyer
Professor and Head,
Mechanical Engineering Department,
Institute of Technology,
Nirma University, Ahmedabad

Dr K Kotecha
Director,
Institute of Technology,
Nirma University, Ahmedabad

Abstract

Energy requirement increases day by day. Thermal power plants are the major source of energy for electricity generation. They are also the major polluter of the environment. In order to reduce environmental pollution either energy consumption should be reduced or energy should be generated with higher conversion efficiency. In thermal power plants part of energy generated by the plants is being consumed by different auxiliaries. The power consumption by these auxiliaries is very high due to poor operation or bad design of the equipments. There are two different ways to improve the efficiency either by efficient operation or by facility conversion. The pumps and compressors are the main auxiliaries which consumes sizable power produced by thermal power plants. Here an attempt is made to improve the efficiency of above mention auxiliaries.

To improve the performance of the pumps various methods have been suggested in the present study. This includes impeller trimming, de-staging and installation of variable frequency drives(VFD). Similarly methods for improving compressor efficiency have also been suggested in the present study. Before modifications actual data are collected for pumps and compressors. Power and efficiency are calculated and same are compared with ideal values reported in supplier manuals. On the basis of discrepancies in above data, methods for performance improvements are suggested. The cost involve for the modifications in the existing set up is also evaluated. To check economic viability of the modification\ suggestion indices like simple payback period, benefit to cost ratio , cost of saved energy etc. have been calculated.

Acknowledgements

I find it most difficult part of the entire training to record my gratefulness to various persons to extend me more than the necessary quantum of co-operation and help in this training. The words that follow are merely the formal expression of my deep sense of indebtedness that is better treasured in thoughts than said in words.

It is pleasure giving verbal expressions to my feeling of deepest gratitude towards Dr.R N Patel (Professor,Mechanical Department, Nirma University), Mr.Shailendra Saha (Jt.General Manager,Maintenance,BPOL, ESSAR POWER, HAJIRA) and Mr.Rajesh Kumar(Jt.General Manager,Operation, BPOL, ESSAR POWER, HAJIRA) who guided and encouraged me by giving their valuable time during the entire period of training.

I would also like to express my gratitude to Prof.V R Iyer(Head of the Department, Mechanical Engineering,Institute of Technology, Nirma University) for showing enthusiasm and allowing me to work outside the university.

I am also very thankful to the entire technician team as well as all technical persons of the plant for their support and co-operation.

Lastly, I am very thankful to Essar Group for giving me this golden opportunity to complete my training.

- Tejas Raval
08MMET08

Contents

List of Tables

List of Figures

Nomenclature

ρ -Density

Q -Discharge flow

P_h -Hydraulic Power

g -Specific Gravity

H -Net Head

η -Efficiency

P_m -Motor Power

P_s -Power Input to Shaft

N -Speed of Impeller

D -Impeller Diameter

ϕ -Power Factor

V -Voltage

I -Current

A -Annual Savings

CRF -Capital Recovery Factor

B/C -Benefits to Cost Ratio

Chapter 1

INTRODUCTION

Thermal (or fossil fuel) power plants (TPP) are the major polluters of man's environment, discharging into atmosphere the flue gases of fossil fuel combustion. A natural solution of the problem of reducing pollution into the atmosphere lies in maximizing the power production without increasing the capacity of the plant there by reducing the amount of the fuel burnt. This approach can be justified both economically and ecologically. The ideal way of solving the problem would be to minimize the power consumed by the auxiliaries within the power plant.[1]

History of civilization and the progress in science and technology are closely associated with the growth of power consumption. A direct consequence of the developing heat power engineering based on combustion of fuel and of the growing amount of electric power produced is the increasing consumption of fuel-energy resources.

Power plants which burns coal, natural gas, petroleum(oil) as a fuel are called fossil fuel power plants. In fossil fuel power plant the chemical energy stored in fossil fuel and oxygen of the air is converted successively into thermal energy, mechanical energy and finally electrical energy.

Following are the types of power plants:

(1) Thermal Power Plants: Thermal power plants use coal as a fuel. In India most of the electricity production is done by thermal power plant. coal is burnt in the combustion chamber of the boiler and then the steam produced from this combustion is used to run the turbine. The biggest advantage of thermal power plant is that they can work even up to 25% overload condition. Looking over the cost of coal and it reserves not only in India but also in the whole world they can be a great source of electricity generation in next few years also.

(2) Gas Turbine Power Plants: Generally gas turbine power plants have very less efficiency as compare to other power plants. In operation working fluid(generally gas) is compressed and then expands in turbine through which power is generated. During compression process the chances of leakages are more so the efficiency of these type of power plants fall in in the range of 20% to 30%. They are generally installed in aviation, oil, gas industries and marine applications because gas turbine power plant doesn't need water for cooling and also lighter in weight. In addition to this gas turbine power plants can also be used in combination with steam turbine and heat recovery steam generator systems to enhance the total efficiency of the plant. The efficiency of these type of combined cycle power plants is up to 48% or it can be improved much more. Because of absence of ash and combustion process they are the cleanest amongst all the conventional power plants.

(3) Diesel Engine Power Plants: They are installed where supply of coal and water is not available in sufficient quantitied or where power requirement is less or as stand by stations. Generally these plants can be used up to 150 MW power generation capacity. They are very beneficial to be install at the cold place and in emergency conditions. They can respond at varying loads without any difficulty. Part load efficiency of diesel power plant also do not falls so much as compared to thermal power plant. Running (operational) cost is very high in comparison to thermal, hydro power plants. Internal combustion engine is a kind of diesel power plant. They do not require fuel handling and ash handling systems and thus installation cost is lower than other types of power plants.

(4) Nuclear Power Plants: Nuclear power generation in India is the latest concept amongst all power generation technologies. Nuclear power plants generates power from nuclear fission or fusion process of the atoms of the nuclear fuel. With both the processes a large amount of heat energy is released from the nucleus of the atom. This thermal energy is removed from the heat source(nuclear reactor core) by passing them through the fuel coolant which can be either used directly as a working fluid or indirectly to heat the another fluid. At the end this steam expands in the turbine and finally produces electricity. Though the power generation from nuclear power plant is very dangerous in terms of radiation and also very costly in comparison to conventional power generation, it is considered as one of the major source of power generation in future.

(5) Hydro Power Plants: Hydro power is a renewable source of energy which is clean, free from pollution and environment friendly. In hydro-electric power plants energy of water is utilized to rotate the turbines which in turn run the generators and produce electricity. The energy of water utilized for power generation may be kinetic or potential. The kinetic energy of water is its energy in motion and potential energy is function of the difference in level or head of water. In either case continuous availability of water is basic necessity. In India the scope of water power development is tremendous. Although present utilization of hydro power in India is relatively small with the present rate of development, it would be be estimated that hydro-electric power generation would be 60% of total power generation of India. In fact country like Norway has power generation almost from hydro base power plants.

Most of the electricity generation is done by thermal Power plants in India. Table

Table 1.1: Installed Capacity of Various Types of Power Plants in India

Type	**Capacity (MW)**
Coal Power Stations	82,343
Gas Power Stations	17,055
Oil Power Stations	1,199
Hydro Power Stations	36,863
Nuclear Power Stations	4,340
Renewable Energy Power Stations	15,427

1.1 shows the installed capacity of different types of power plants.[2] Figure 1.1 shows the percentage distribution of electricity generation by different modes. [3]

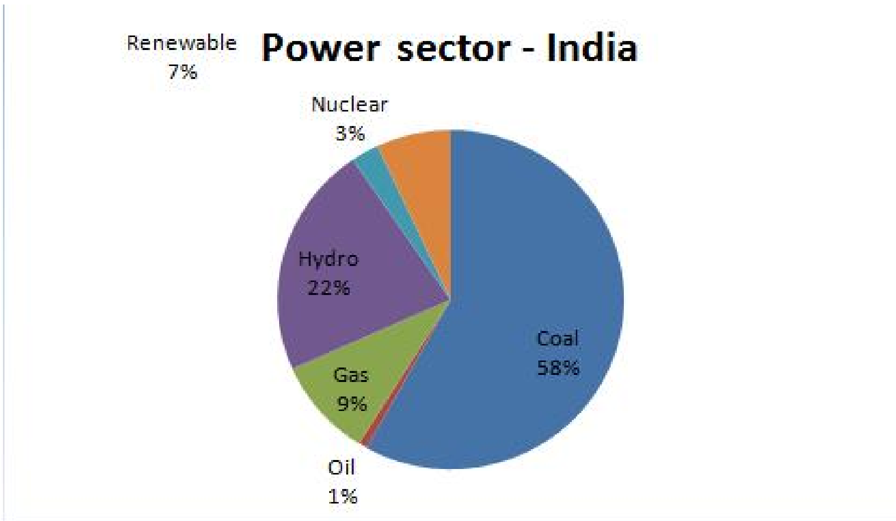

Figure 1.1: Power Sector-India

Sector wise % share is shown below the Table 1.2

Table 1.2: Sector wise % share in Power Generation In India

Sector	Capacity (MW)	% Share
State	78,378	52.5%
Central	50,522	34%
Private	28,328	13.5%

1.1 Combined Cycle

The dual advantages of efficiency and flexibility make gas turbine based cycles the dominant energy converters for the production of world's electricity today.[4] The gas turbine engine is characterized by its relatively low capital cost, compared with steam power plants. It has environmental advantages and short constructional lead time. One of the technologies adopted now-a-days for improvement is the 'Combined Cycle'. Natural gas combustion turbines and combined-cycle plants are foresighted to capture over most of the Indian new-generation market in the next decade.

The maximum Steam temperature in power cycle does not exceed beyond 600 ($^{\circ}c$), although the temperature in combustion chamber is near about 1300 ($^{\circ}c$). So there is a great decrease in availability because of heat transfer from combustion gases to steam through such a large temperature difference. By combining the high temperature power plant as a topping unit to the steam plant, a higher energy conversion efficiency from fuel to electricity can be achieved. Thus the Gas turbines are increasingly used in combination with steam cycle either to generate electricity alone as in combined cycles or to generate both electrical power and heat for industrial processes.

Open-cycle gas turbine engine exhausts higher-grade heat to the atmosphere than steam turbines. The combined cycle uses the exhaust heat from the gas turbine engine to increase the power plant output and boost the overall efficiency to more than 55%.[5], [6] At first air is compressed in the compressor of gas turbine plant. The compression of air is achieved according to Brayton cycle as shown Figure 1.2.

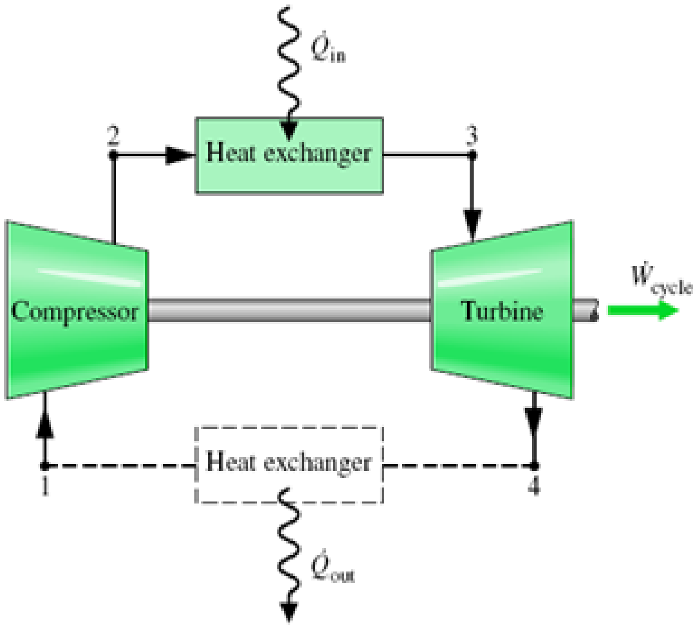

Figure 1.2: Brayton Cycle

Then the compressed air is heated in the combustor by burning fuel, where part of compressed air is used for combustion and the flue gases thus produced are allowed to expand in the turbine which is coupled with the generator.

After expanding in the gas turbine the exhaust gases are fed to the Heat Recovery Steam Generator (HRSG) to recover more heat energy. In HRSG steam is produced and then the steam is given to the inlet of the steam turbine.

The conversion of heat energy to mechanical energy with the aid of steam is based on Rankine cycle as shown in Fig. 1.3. The initial stage of working fluid is water at a certain temperature that is pressurized by a pump and fed to the boiler. In the boiler the pressurized water is heated at constant pressure. Superheated steam is expanded in the turbine which is coupled with generator.

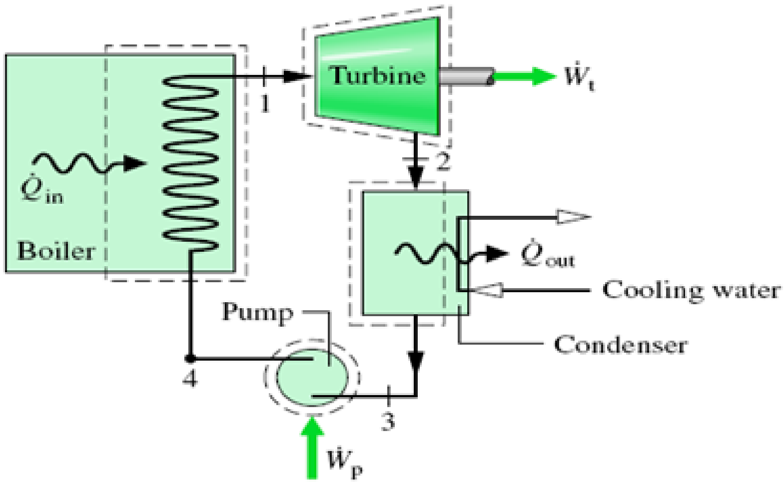

Figure 1.3: Rankine Cycle

Thus with the help of the combined cycle power plant(CCPP) more power can be generated with the same fuel capacity. Combine cycle power plant integrates two power conversion cycle-Brayton cycle (Gas turbine) and Rankine cycle (Steam turbine) with the principal objective of increasing overall plant efficiency (Fig. 1.4)

1.2 Auxiliary Equipments

The thermal power plants consist of main/power generating equipments as well as auxiliary equipments without which the power generation is not possible. While the main equipments produce power, these all additional equipments consumes power and sometimes the amount of power consumption is also very high.It is always desired to minimize these auxiliary power consumption in plant to optimize the total cost of power generation. It has been studied that auxiliary equipments in plants is focussed area where one can optimize the power consumption.

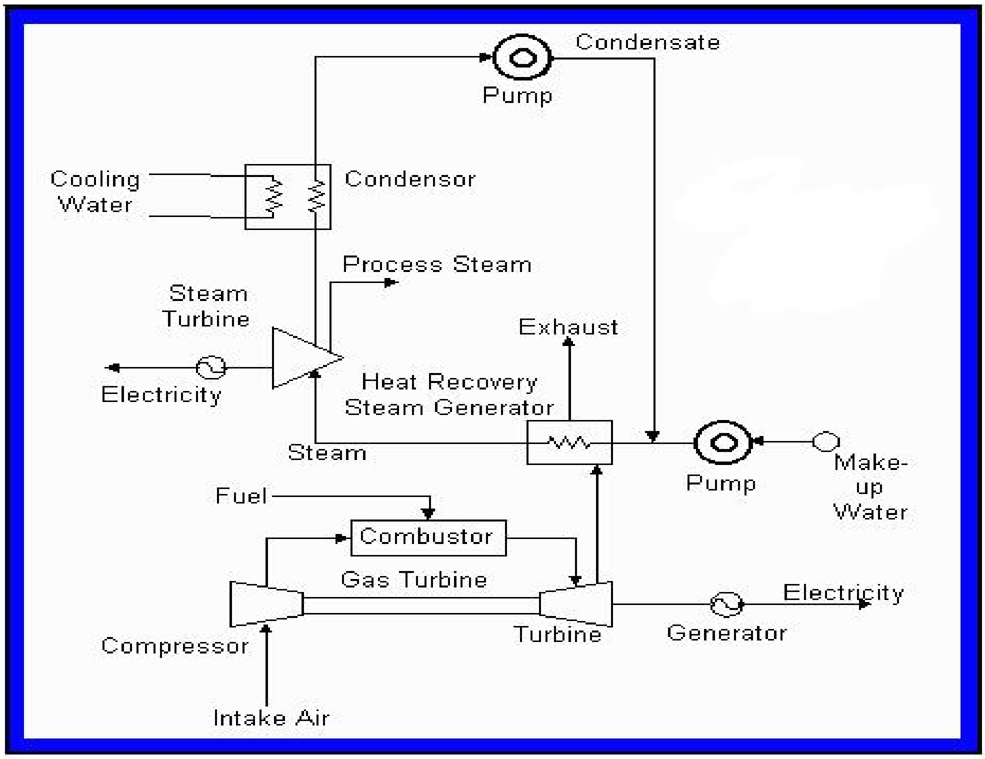

Figure 1.4: Combined Cycle Power Plant

In coal based power plants generally the auxiliary Power consumption is about 5 to 8 % while in combined cycle power plant(CCPP) the auxiliary power consumption fall in the range of 2 to 5 % of actual generating capacity.

The auxiliary power consumption optimization may be targeted by the way of optimizing the drives, increasing the efficiency of drives, harnessing the daylight , improving the natural ventilation etc.
Such auxiliary equipments are

(1) Boiler feed pumps

(2) Condensate extraction pumps

(3) Compressors

(4) Cooling towers

(5) Water treatment systems

(6) Air conditioning systems etc.

1.3 Details of the Power Plant(Profile of Power Plant)

Essar Power limited is a 1015 MW Combined Cycle Power Plant.It is situated at Hazira village near Surat in Gujarat. It has two separate power generating units named as:

(1) Essar power plant(515 MW)

(2) Bhander Power Plant.(500 MW)

The whole project is done for Bhander power plant. The plant has three Gas turbines along with three Heat Recovery Steam Generators and two Steam turbines. Details of the power plant are given in Table 1.4

The gas Turbine has dual fuel firing capability. Both Gas and Neptha can be fired depending on the availability of the fuel. Toal 318 MW power is generated from the gas turbine and the rest 182 MW is generated from the steam turbine. The exhaust flue gases from the gas turbine at 590 ($^{\circ}c$) goes to individual Heat Recovery Steam Generators(HRSG). At HRSG steam is generated at three levels, HP steam in range of 70 to 120 bar and 545 ($^{\circ}c$) temperature with a flow rate in the range of 175 to 310 TPH(Tons Per Hour), IP steam 25 bar and 300 ($^{\circ}c$) temperature with a flow rate of 18 TPH and LP steam at 7 bar and 190 ($^{\circ}c$) temperature with a flow rate of 35 TPH.

Table 1.3: Details of Essar Power Ltd, Surat

Unit	Capacity (MW)	Make
GTG – 1	100 MW	HITACHI, Japan
GTG – 2	108 MW	BHEL, Hyderabad
GTG – 3	110 MW	BHEL, Hyderabad
STG – 1	55 MW	SIEMENS,Germany
STG – 2	127 MW	SIEMENS,Sweden
Total	**500 MW**	

1.4 Organization of Thesis

The thesis has been organized in five chapters including recommendations and conclusions.

Chapter one is introduction to the Indian power sector and project work. This covers the overall picture of the project including motivation for the project work.

Chapter two is Literature review of the energy conservation in pumps and compressors which includes various methodologies adopted for energy conservation and the need for the energy conservation.

Chapter three is Observation, Results and Discussion which includes results and discussion of actual and design power consumption as well as efficiency of the boiler feed pumps, condensate extraction pumps, air compressors. Apart from this, the reasons for less efficiency and more power consumption of the auxiliaries are also found out. In this chapter, all the results obtained during the energy audit have been discussed in detail.

The name of the fourth chapter is Suggestions, Implementations and Improvements which gives details about suggestions given to plant officials for optimizing the power consumption of auxiliaries. Accordingly the implementations have been done and improvements are given in the chapter.

In chapter five conclusion and future scope of the project is given.

The thesis also includes resource of the references. This is followed by the Appendix section in which various tables are given for more details.

Chapter 2

LITERATURE SURVEY

Auxiliary power consumption is quite high in all Indian power stations and as such there is substantial scope for its minimization. In order to study, analyze and recommend remedial measures the following three sections are made for 500 MW power plant.[7]

(1)Energy conservation in Boiler Feed Pumps

(2) Energy conservation in Condensate Extraction Pumps

(3) Energy conservation in Air Compressors

2.1 Energy Conservations in Pumps

In the studies that have been conducted for energy saving, it has been seen that one of the areas of high potential energy saving is pumping systems.[8],[9],[10],[11] According to a study that the American Hydraulics Institute has made, 20% of the total energy produced by plants has been consumed by pumps in developed countries.[12] This situation has caused new research to be made to find more efficient systems in production and operation by producers and users of pumps.[13],[14],[15],[16],[17] Higher efficiency pumps itself is not enough for maximum efficiency of the pumping system.[18],[19] For pump systems, higher efficiency not only depends on a good pump

design but also on a good design of the complete system and its working conditions. Otherwise, it is inevitable that even the most efficient pump in a system that has been wrongly designed and wrongly assembled is going to be inefficient.[20],[21],[22],[23],[24],[25]
Due to high capacity of power plants, efficient operation of pumps plays very important role. It is also found that pumps do not operate at their optimal points (Best Efficiency Points) due to several reasons. The methods to improve efficiency of pumps are discussed bellow.

2.1.1 Proper Selection of the Pump as per Requirement

To provide the most active and effective system, the needs of the process should be known. Also, the flow rate-time intervals and pump head of the system should be known. The system should be selected not only to meet the needs of working at maximum capacity but also, from the economic point of view. After which the pipe installation shall be designed according to pump's requirement on high or low capacity by selecting proper pipe diameter.[26],[27]
To understand the effect of over sizing of pump we will take an example of a system. In Fig. 2.1 **green curve** is the system curve. Suppose, we have estimated our operating conditions as 500 m^3/hr flow and 50 m head, we will chose a pump curve which intersects the system curve (Point A) at the pump's best efficiency point (BEP).
But, in actual operation, we find that 300 m^3/hr is sufficient. The reduction in flow rate has to be affected by a throttle valve. In other words, we are introducing an artificial resistance in the system. Due to this additional resistance, the frictional part of the system curve increases and thus the new system curve will shift to the left -this is shown as the **red curve**.
So the pump has to overcome additional pressure in order to deliver the reduced flow. Now, the new system curve will intersect the pump curve at point B. The revised parameters are 300 m^3/hr at 70 m head. The red double arrow line shows the addi-

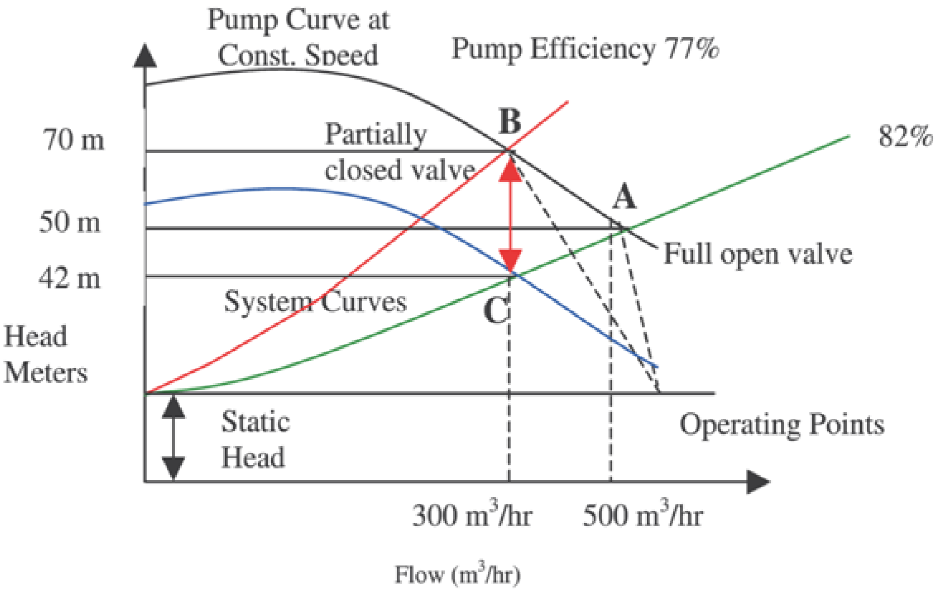

Figure 2.1: Effect of Pump Oversizing

tional pressure drop due to throttling. You may note that the best efficiency point has shifted from 82 % to 77 % efficiency.

So what we want is to actually operate at point C which is 300 m^3/hr on the original system curve. The head required at this point is only 42 meters. What we now need is a new pump which will operate with its best efficiency point at C. But there are other simpler options rather than replacing the pump. The speed of the pump can be reduced or the existing impeller can be trimmed (or new lower size impeller). The **blue pump curve** represents either of these options.

2.1.2 Impeller Trimming

To improve the pump performance impeller trimming is one of the best techniques.[28],[29],[30],[31],[32],[33]It involves machining of the impeller to reduce its diameter.[34],[35] After

the pump impeller has been trimmed geometric and kinematic similarity conditions were not completely preserved. The ratio between some characteristic dimensions (e.g. between impeller width and outlet diameter or between impeller inlet and outlet diameter etc.) changes and therefore geometric similarity is not attained. Also, kinematic similarity is not preserved at the impeller outlet because the blade angle varies with radius. At the same time similarity conditions are satisfied in many elements which include the impeller shape, disposition, number of impeller blades and kinematic conditions at the inlet like ratio between impeller width and inlet diameter.[36] It is suggested in the literature that trimming should be limited to about 75% of a pump's maximum impeller diameter. An excessive trimming can result in mismatching of impeller and casing. As the impeller diameter decreases the clearance between the impeller and the fixed pump casing increases resulting a sudden rise internal flow recirculation causing head loss and the lowering of pumping efficiency.
Specific speed is defined as:[37],[38],[39]

$$n_{sp} = [(n * \sqrt{Q}) \div H^{0.75}]$$

where n is the speed of rotation, Q is the discharge and H is the pump head.

[40] The Figure 2.2 reveals the advantages of impeller trimming. Before impeller trimming, with original impellers the power demand with two pumps running at the full load (maximum demand) flow (Qd) is Pd. This is considerable more than the power demand (Psd) with one pump running at trimmed impeller at maximum demand(Qd). Psm is presumed to be maximum available drive power , which limits the maximum flow under one pump operation with untrimmed impellers to Qms, well below Qd. The maximum power available from the single pump driver (Psm) is still greater than Psd, the trimmed impeller power at Qd.

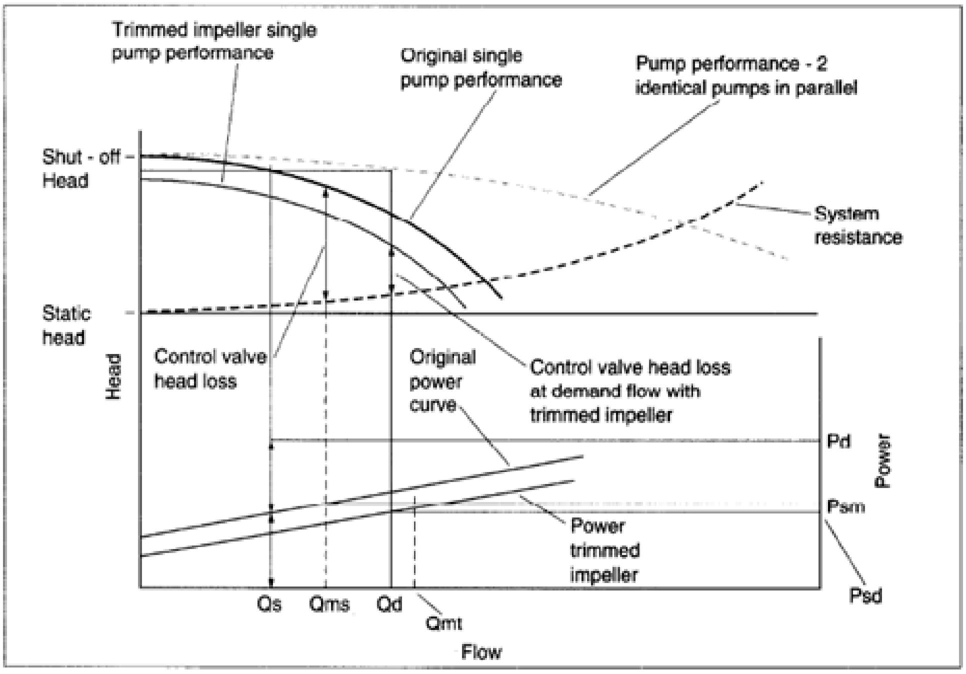

Figure 2.2: Improvements in Pumps After Impeller Trimming

2.1.3 De-staging of the pump

Pump de-staging refers to the removal of one or more impeller from the multistage pump to reduce the energy added to the system fluid. Pump de-staging offers a useful correction to pumps. Pump de-staging allows the performance curve to be moved upward achieving roughly the same effects as modification of the diameter.

Pump de-staging reduces number of stages which in turn directly lowers the amount of energy imparted to the system fluid and lowers both the flow and pressure generated by the pump in operating stages.

End users should consider a pump de-staging when:

• Many of the system's bypass valves are open, indicating excess flow available to the system equipment.

• High noise or vibration levels existence indicating excessive flow.

• A pump is operating far from its design points.

Following are the advantages of pump de-staging.

•A principal benefit of de-staging is decreasing operating and maintenance costs.

•Less fluid energy is wasted in the bypass lines and across throttle valves, or dissipated as noise and vibrations through the system.

•The actual energy saving will be a little higher than shown by the fluid energy savings due to the avoided energy lost due to inefficiencies in the pump and motor.

•In addition to energy savings, pump de-staging also reduces wear on system piping, valves, and piping supports. Excessive flow makes piping to vibrate which causes fatigue in pipe welds and mechanical joints. So over a time period welds crack and joints may become loose, making repairs necessary.

Figure 2.3 indicates the effect of de-staging. it can be seen from the figure that before de-staging, at flow rate of 220 m^3/h the corresponding head is 1800m. If any typical application requires 220 m^3/h flow rate at 1300 m of head. This can be achieved by throttling the pump but with reduced efficiency. This could also be achieved by reducing one or more number of stages from the pump. From figure it is clear that de-staging reduces the power required to operate the pump at off design parameters.

2.1.4 Installation of VFD

A variable frequency drive (VFD) is a system for controlling the rotational speed of an alternating current (AC) electric motor by controlling the frequency of the electrical power supplied to the motor. The basic function of the VFD is to act as a variable frequency generator in order to vary speed of the motor as per the user setting.

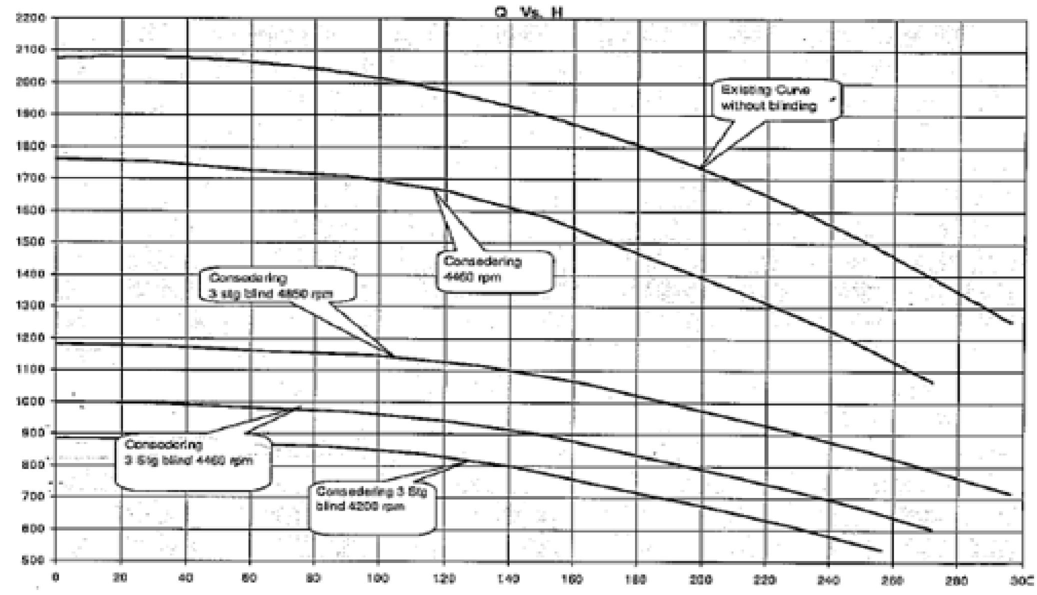

Figure 2.3: Effect of De-staging

The base speed of the motor is proportional to supply frequency. So by changing the supply frequency the motor speed can be changed.

Usually a fluid coupling is provided in between the motor and the main pump to vary the speed of the pump as per the load requirements. Generally the efficiency of fluid Coupling is in the range of 70% to 75% and with lower load the efficiency reduces to approximately 55% only. Instead if VFD is used, the speed of the motor is reduced in accordance with the flow requirement of the pump. The flow control is not obtained through regulating fluid coupling. Fluid coupling remains fully opened giving about 92% efficiency.

Following are the operational parameters which affects the performance of pump.There should also be considered while energy conservation in pumps.

Operational parameters:

1) Speed

2) NPSHR

3) Matching of pump and systems (Operating point)

4) Cooling water recirculation

5) Cavitations and recirculation [41]

6) Operation of combination pumps

7) Load variation

8) Fluid to be pumped, properties

9) Wear Ring clearance(leakage)

10) Valve problems

2.2 Energy Conservation in compressors

Although compressed air is a versatile tool used widely in industries for a variety of purposes it is typically one of the most expensive utilities in an industrial facility. Unfortunately, running air compressors (AC) often uses more energy than any other equipment. Studies show that the average compressed air typically consumes about 5% to 20% of a plant's annual electric cost.[42]

Air compressor efficiency is the ratio of energy input to energy output. Improving air compressor efficiency can yield significant savings.

The total energy use of a compressor system depends on several factors. The air compressor type, model and size are important factors in the compressor's energy consumption, but the motor power rating, control mechanisms, system design, uses and maintenance are also fundamental in determining the energy consumption of a compressed air system.

Factors affecting the Compressor Performance are:

(1) Compressor inlet temperature and pressure

(2) Compressor discharge temperature and pressure

(3) Humidity of the surrounding atmosphere

The case studies given in the literature illustrate how to identify the type of control and potential problems such as inadequate compressed air storage, over-sized compressors and compressed air leaks from the power signatures.

2.2.1 Compressor Controls

In general compressor controls seek to maintain the compressed air discharge pressure within a specified range. There are at least six common types of compressor control modes for small reciprocating and rotary air compressors as start/stop, load/unload, inlet modulation, auto-dual, variable displacement and variable speed control. [43]
Start/stop control is a frequently used method practically adopted by small reciprocating compressors. In this type of control the compressor turns itself off and draws no power as long as the discharge pressure remains above a specified level. This is the most energy-efficient type of control since the compressor runs at maximum efficiency when compressing air and turns off when not compressing air.
Most rotary compressors are unable to run in start/stop mode. Thus the compressor continues to run even as the demand for compressed air declines or ends. Most rotary compressors use either load/unload, inlet modulation, auto-dual, or variable displacement control.
With load/unload control, the compressor runs fully loaded producing compressed air at maximum efficiency until the discharge pressure reaches the upper activation pressure setting which causes the compressor to unload. When unloaded, the compressor no longer adds compressed air to the system but the motor continues to run.
With inlet modulation control, the inlet air valve to the compression device is continuously adjusted restricting the inlet air to vary the compressed air output to meet the demand on the system.
Auto-dual control is a combination of modulation and load/unload control in which

the compressor operates in modulation control down to a specified pressure and switches to load/unload control below this pressure.

With variable displacement control the effective length of the compressor rotor is varied by valves in the rotor housing.

Recently rotary compressors with variable speed motor drives have been used. Variable-speed compressors vary the speed of the compressor motor to meet the compressed air demand.[44]

Studies shows that variable torque loads such as air compression and the load on the motor varies with the cube of the motor speed. Thus decreasing the speed of the motor during periods of low compressed air demand significantly decreases the load on the motor resulting variable-speed compressors operate very efficiently at even low loads.

The case studies demonstrate that the power draw and compressed air output of industrial air compressors vary significantly according to the type of control method and the size of the compressor relative to the demand for compressed air.

Variable speed compressors have the best part-load performance, followed by compressors operating in load/unload mode and finally inlet-modulation mode. Over-sized compressors are always less efficient than properly sized compressors.

2.2.2 Ways to modify the Design (to improve the efficiency)

Save for times of need. The first aspect involves choosing a receiver or storage tank, to fit the needs of the system demand and prevent system pressure from dropping below minimum required pressure during times of peak demand. A drop in pressure will cause end tools to function improperly. The common response to the tool malfunction is to increase the system pressure. The energy used in increasing system pressure could have been saved by the use of a properly sized receiver.

Straighten the path. The second aspect of system design is the layout and design of the air delivery system. Narrow delivery lines, looping and sharp bends in the lines can create pressure drops in the system and reduce end use pressure. The common response to this is to increase compressor pressure and use more energy. This could have been avoided by better system design.

Use cooler intake air. A third design aspect that may have a significant impact on air compressor efficiency is the intake air temperature. The energy required to compress cool air is much less than that required to compress warm air. Reducing the intake temperature by moving the compressor intake outside the building and into a shaded area may drastically lower the energy required for compression.

Single vs. multiple compressors. In some systems it may be more efficient to use a series of smaller compressors rather than one larger compressor. Additional smaller compressors can be brought on-line or shut down as needed.

Recover waste heat. Recovered waste heat can be used to preheat process and boiler water, for space heating, and more.

2.2.3 Maintenance

Fix the leaks. This is the area where the most significant changes can occur. In addition of having a great impact on energy use. Improvements here are also often relatively cheap and have immediate results. Wasted air is lost through leaks in the system. Although leaks are often very small but significant amounts of air can be lost through these gaps. The air lost is proportional to the size of the orifice and a function of the air compressor supply pressure. Fig. 2.4 illustrates the amount of air lost through different orifice sizes.

[45] **Change the filters.** Another important element of the system is filters. Filters

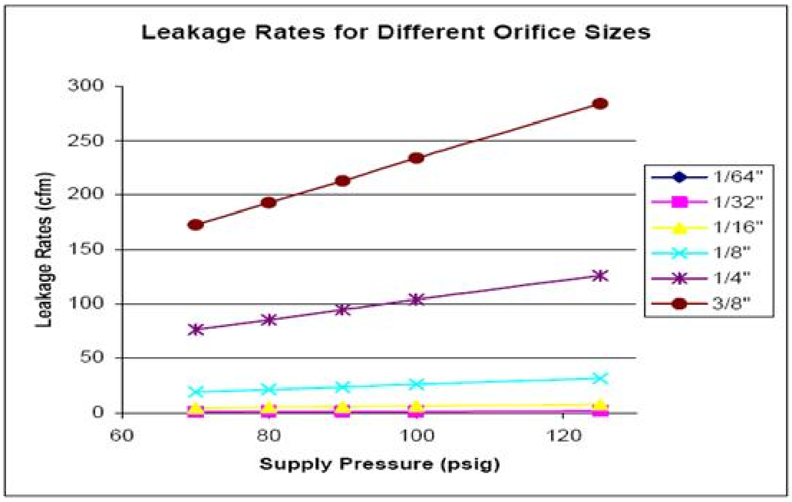

Figure 2.4: Air Compressor Leakages

are located throughout the system to ensure clean air for end uses. Often these filters are not known of or are simply not checked. Dust, dirt, moisture and grease can clog the filters leading to a pressure drop in the system. This pressure drop is not often seen for what it is and more compression energy is used to compensate for the clogged filters resulting in increased energy use.

2.3 Objective Of Present Study

Above literature study indicates that major energy loss in the thermal power plant there is due to losses of energy in auxiliaries. With rising energy cost, power plants are increasing their focus on the amount of energy expended by rotating equipment like pumps, compressor etc. Unscheduled repair and poor reliability are causing companies to lose their production and spend more money on maintenance. To conserve the energy and run the plant at high efficiency one have to optimize this power consumption and optimization becomes also mandatory for reducing the total cost of electricity generation.

This dissertation work will mainly concentrate on finding the faults and suggesting corrective measures to conserve the energy and total cost while generating the electricity in combined cycle power plants.

- To estimate the actual power consumption by various auxiliaries like pumps, compressors etc. and compare it with design data.
- To find out gap areas for energy conservation in auxiliaries.
- To suggest alternative solutions for optimization of energy consumed by auxiliaries.
- To implement above suggestions and calculate various energy economics indices to check improvements in existing system.

Suggestions and the possible area for energy conservation, methods and implementation of suggestions to optimized energy in targeted thermal power plant is discussed in subsequent sections.

Chapter 3

OBSERVATIONS,RESULTS AND DISCUSSION

To achieve the objectives mentioned in chapter 2, Essar power Ltd, Surat was selected for the study. The details of the same are given in the earlier sections. Essar power Ltd, Surat has combined cycle power plant (CCPP) of capacity 1015 MW. It was decided to optimize the auxiliary power consumption of the said power plant. The study of CCPP shows that there is an ample scope for optimizing auxiliary power consumption. In thermal power plant, the auxiliary power consumption is about 5 to 8%. This could be still higher for the power plant running at part load conditions. It also increases with the age of the power plant. Boiler feed pumps (BFP), condensate extract pumps (CEP), air compressors, air conditioning systems, cooling water systems, water treatments plants, effluent treatment plants, ventilation systems etc. are auxiliary systems which consume considerable power produced by CCPP. However, pumps and compressors consume significant amount of power and hence they were selected as auxiliary equipments to be optimized in this present study.

The following methodology was adopted:

• Identification of auxiliary equipments such as Boiler Feed Pumps (BFP), Condensate Extraction Pumps (CEP), and air compressors.

• Collection of design data of such equipments

• Measurement of actual data

• Calculation of power and efficiency for these equipments

• Comparison of actual data with design data

• Identification of causes for poor performance of said equipments

• Recommendation for improvement of performance

• Implementation of suggestions

• Improvement in efficiency and power consumption of auxiliary equipment

3.1 Energy Conservation in Pumps

Collection of Design Data:

CCPP at Essar Power Ltd, Surat consist of nine BFP, five CEP and two air compressors. The design data for these systems were recorded from manuals provided by the suppliers. The design parameters such as discharge pressure (kg/cm^2), discharge flow (m^3/h), water inlet temperature to the pump ($^\circ c$) and current drawn by the motor have been recorded. Actual above data were measured and same have been tabulated as shown in Table 3.1.

Power consumption and efficiency calculation require pump inlet and outlet pressure, flow rate of water and density at given temperature. Temperature of water at delivery end is essential for density calculation. To calculate actual power consumed by prime mover , current and voltage were measured. The detailed calculations are shown in subsequent sections.

Table 3.1: Design and Actual Parameters of BFP and CEP

		Delivery Pressure (kg/cm^2)	Flow Rate (m^3/h)	Temperature of water at pump outlet $(^\circ c)$	Current $(Amp.)$
BFP - 1	Design	121.8	100	160	65
	Actual	100	90	160.4	41.6
BFP - 2	Design	121.8	130	160	65
	Actual	100	90	160.4	44.4
BFP - 5	Design	172.7	185	200	155
	Actual	88.04	168.48	157	92.07
BFP - 6	Design	172.7	185	200	155
	Actual	95	182.57	164.4	104
BFP - 7	Design	100.47	120.4	160	53
	Actual	101.8	103	157.5	39.5
BFP - 8	Design	105.5	120.4	160	53
	Actual	101.9	103	157.5	39.7
BFP - 9	Design	100.47	120.4	160	53
	Actual	107.92	99.84	163	37.96
CEP - 1	Design	26	285	50	33
	Actual	26.03	230	43	28.93
CEP - 3,5	Design	26	300	50	34.5
	Actual	24.75	207.28	47.7	25.9
CEP - 4	Design	26	300	50	34.5
	Actual	24.4	230	41.8	28.2

3.2 Power and Efficiency Calculations

This CCPP consist of nine BFP and five CEP. Power consumption and efficiency of all BFP and CEP is shown below.

BFP-1

Pump hydraulic power,

$$\text{Ph} = \rho * \text{g} * \text{Q} * \text{H}$$

$$= [907.45(kg/m^3) * 9.8(m/s^2) * 90(m^3/h) * 922(m)]$$

$$= [907.45(kg/m^3) * 9.8(m/s^2) * 90(m^3/s) * 922(m)/3600]$$

$$\textbf{Ph = 204.98 kW}$$

[Density= According to Temp. Of fluid From steam table]

Motor Power

$$\text{Pm} = 1.73 * \text{V} * \text{I} * \text{COS}\ \phi$$

$$= 1.73 * 6600(\text{V}) * 41.6(\text{Amp.}) * 0.9$$

$$\textbf{Pm = 427.48 kW}$$

Power input to the shaft,

$$\text{Ps} = \text{Motor Power} * \text{Efficiency of Motor}$$

$$= 427.48 * 0.94$$

$$\textbf{Ps = 401.83 kW}$$

Efficiency of Pump,

η = [(Hydr. Power)/(shaft power)] * 100%

= [Ph / Ps] * 100%

$$= [204.98/401.83] * 100\%$$

$$\eta = 51.01\%$$

BFP-2

Pump hydraulic power

$$\text{Ph} = \rho * \text{g} * \text{Q} * \text{H}$$
$$= [907.45(kg/m^3) * 9.8(m/s^2) * 90(m^3/h) * 922(m)]$$
$$= [907.45(kg/m^3) * 9.8(m/s^2) * 90(m^3/s) * 922(m)/3600]$$
$$\textbf{Ph = 204.98 kW}$$

[Density= According to Temp. Of fluid From steam table]

Motor Power

$$\text{Pm} = 1.73 * \text{V} * \text{I} * \text{COS } \phi$$
$$= 1.73 * 6600(\text{V}) * 44.4(\text{Amp.}) * 0.9$$
$$\textbf{Pm = 456.26 kW}$$

Power input to the shaft,

$$\text{Ps} = \text{Motor Power} * \text{Efficiency of Motor}$$

$$= 456.26 * 0.94$$

$$\textbf{Ps = 428.88 kW}$$

Efficiency of Pump,

η = [(Hydr. Power)/(shaft power)] * 100%

= [Ph / Ps] * 100%

$$= [204.98/428.88] * 100\%$$

$$\eta = 47.79\%$$

BFP-5

Pump hydraulic power

$$\text{Ph} = \rho * \text{g} * \text{Q} * \text{H}$$

$$= [910.74(kg/m^3) * 9.8(m/s^2) * 168.48(m^3/h) * 700(m)]$$

$$= [910.74(kg/m^3) * 9.8(m/s^2) * 168.48(m^3/s) * 700(m)/3600]$$

$$\text{Ph} = 292.39 \text{ kW}$$

[Density= According to Temp. Of fluid From steam table]

Motor Power

$$\text{Pm} = 1.73 * \text{V} * \text{I} * \text{COS } \phi$$

$$= 1.73 * 6600(\text{V}) * 92.07(\text{Amp.}) * 0.9$$

$$\mathbf{Pm = 946.12 \text{ kW}}$$

Power input to the shaft,

$$\text{Ps} = \text{Motor Power} * \text{Efficiency of Motor} = 946.12 * 0.96$$

$$\mathbf{Ps = 908.28 \text{ kW}}$$

Efficiency of Pump,

$$\eta = [\ (\text{Hydr. Power})/(\text{shaft power})\ ] * 100\%$$

$$= [\ \text{Ph} / \text{Ps}\ ] * 100\%$$

$$= [292.39/908.28\ ] * 100\%$$

$$\eta = 32.19\%$$

BFP-6

Pump hydraulic power

$$\text{Ph} = \rho * \text{g} * \text{Q} * \text{H}$$

$$= [903.3(kg/m^3) * 9.8(m/s^2) * 182.57(m^3/h) * 770(m)]$$

$$= [903.3(kg/m^3) * 9.8(m/s^2) * 182.57(m^3/s) * 770(m)/3600]$$

$$\mathbf{Ph = 345.68 \text{ kW}}$$

[Density= According to Temp. Of fluid From steam table]

Motor Power

$$\text{Pm} = 1.73 * \text{V} * \text{I} * \text{COS } \phi$$
$$= 1.73 * 6600(\text{V}) * 104(\text{Amp.}) * 0.9$$
$$\textbf{Pm = 1068.72 kW}$$

Power input to the shaft,

$$\text{Ps} = \text{Motor Power} * \text{Efficiency of Motor}$$

$$= 1068.72 * 0.96$$

$$\textbf{Ps = 1029.17 kW}$$

Efficiency of Pump,

$$\eta = [\ (\text{Hydr. Power})/(\text{shaft power})\ ] * 100\%$$
$$= [\ \text{Ph} / \text{Ps}\ ] * 100\%$$

$$= [345.68/1029.17] * 100\%$$

$$\eta = 33.58\%$$

BFP-7

Pump hydraulic power

$$\text{Ph} = \rho * \text{g} * \text{Q} * \text{H}$$
$$= [909.39(kg/m^3) * 9.8(m/s^2) * 103(m^3/h) * 947(m)]$$
$$= [909.39(kg/m^3) * 9.8(m/s^2) * 103(m^3/s) * 947(m)/3600]$$
$$\textbf{Ph = 241.46 kW}$$

[Density= According to Temp. Of fluid From steam table]

Motor Power

$$Pm = 1.73 * V * I * COS\ \phi$$
$$= 1.73 * 6600(V) * 39.5(Amp.) * 0.9$$
$$\mathbf{Pm = 405.90\ kW}$$

Power input to the shaft,

$$Ps = \text{Motor Power} * \text{Efficiency of Motor}$$

$$= 405.90 * 0.94$$

$$\mathbf{Ps = 383.57\ kW}$$

Efficiency of Pump,

$$\eta = [\ (\text{Hydr. Power})/(\text{shaft power})\ ] * 100\%$$
$$= [\ Ph\ /\ Ps\ ] * 100\%$$

$$= [241.46/383.57] * 100\%$$

$$\eta = 62.95\%$$

BFP-8

Pump hydraulic power

$$Ph = \rho * g * Q * H$$
$$= [909.39(kg/m^3) * 9.8(m/s^2) * 103(m^3/h) * 947(m)]$$
$$= [909.39(kg/m^3) * 9.8(m/s^2) * 103(m^3/s) * 947(m)/3600]$$
$$\mathbf{Ph = 241.46\ kW}$$

[Density= According to Temp. Of fluid From steam table]

Motor Power

Pm = 1.73* V * I * COS ϕ

= 1.73 * 6600(V) * 39.7(Amp.) * 0.9

Pm = 407.96 kW

Power input to the shaft,

Ps = Motor Power * Efficiency of Motor

$$= 407.96 * 0.94$$

Ps = 385.522 kW

Efficiency of Pump,

η = [(Hydr. Power)/(shaft power)] * 100%

= [Ph / Ps] * 100%

$$= [241.46/383.522] * 100\%$$

$$\eta = 62.97\%$$

BFP-9

Pump hydraulic power,

Ph = ρ * g * Q * H

$$[= 904.15(kg/m^3) * 9.8(m/s^2) * 99.84(m^3/h) * 1001(m)]$$

$$[= 904.15(kg/m^3) * 9.8(m/s^2) * 99.84(m^3/s) * 1001(m)/3600]$$

Ph = 245.98 kW

[Density= According to Temp. Of fluid From steam table]

Motor Power,

$$\text{Pm} = 1.73 * \text{V} * \text{I} * \text{COS } \phi$$
$$= 1.73 * 6600(\text{V}) * 37.96(\text{Amp.}) * 0.9$$
$$\mathbf{Pm = 390.084\ kW}$$

Power input to the shaft,

$$\text{Ps} = \text{Motor Power} * \text{Efficiency of Motor}$$
$$= 390.08 * 0.94$$
$$\mathbf{Ps = 366.67\ kW}$$

Efficiency of Pump,

$$\eta = [\ (\text{Hydr. Power})/(\text{shaft power})\ ] * 100\%$$
$$= [\ \text{Ph} / \text{Ps}\ ] * 100\%$$
$$= [245.98/366.67\ ] * 100\%$$
$$\eta = \mathbf{67.01\%}$$

CEP-1

Pump hydraulic power,

$$\text{Ph} = \rho * \text{g} * \text{Q} * \text{H}$$
$$= [991.08(kg/m^3) * 9.8(m/s^2) * 230(m^3/h) * 260.3(m)]$$
$$= [991.08(kg/m^3) * 9.8(m/s^2) * 230(m^3/s) * 260.3(m)/3600]$$

Ph = 161.522 kW [Density= According to Temp. Of fluid From steam table]

Motor Power

$$\text{Pm} = 1.73 * \text{V} * \text{I} * \text{COS } \phi$$
$$= 1.73 * 6600(\text{V}) * 28.93\ (\text{Amp.}) * 0.9$$
$$\mathbf{Pm = 297.29\ kW}$$

Power input to the shaft,

Ps = Motor Power * Efficiency of Motor

$$= 297.290 * 0.94$$

Ps = 279.45 kW

Efficiency of Pump, η = [(Hydr. Power)/(shaft power)] * 100%
= [Ph / Ps] * 100%

$$= [161.522/279.45] * 100\%$$

$$\eta = 57.79\%$$

CEP-3 and CEP-5

Pump hydraulic power,

Ph = ρ * g * Q * H

$$= [988.97(kg/m^3) * 9.8(m/s^2) * 207.28(m^3/h) * 238.5(m)]$$

$$= [988.97(kg/m^3) * 9.8(m/s^2) * 207.28(m^3/s) * 238.5(m)/3600]$$

Ph = 133.09 kW

[Density= According to Temp. Of fluid From steam table]

Motor Power

Pm = 1.73* V * I * COS ϕ

= 1.73 * 6600(V) * 25.9(Amp.) * 0.9

Pm = 266.15 kW

Power input to the shaft,

Ps = Motor Power * Efficiency of Motor

$$= 266.15 * 0.94$$

Ps = 250.18 kW

Efficiency of Pump, η = [(Hydr. Power)/(shaft power)] * 100%

= [Ph / Ps] * 100%

$$= [133.09/250.18] * 100\%$$

$$\eta = 53.19\%$$

CEP-4

Pump hydraulic power,

$$\text{Ph} = \rho * \text{g} * \text{Q} * \text{H}$$

$$= [991.08(kg/m^3) * 9.8(m/s^2) * 230(m^3/h) * 244(m)]$$

$$= [991.08(kg/m^3) * 9.8(m/s^2) * 230(m^3/s) * 244(m)/3600]$$

$$\mathbf{Ph = 151.40\ kW}$$

[Density= According to Temp. Of fluid From steam table]

Motor Power

$$\text{Pm} = 1.73 * \text{V} * \text{I} * \text{COS}\ \phi$$

$$= 1.73 * 6600(\text{V}) * 28.2(\text{Amp.}) * 0.9$$

$$\mathbf{Pm = 289.78\ kW}$$

Power input to the shaft,

$$\text{Ps} = \text{Motor Power} * \text{Efficiency of Motor}$$

$$= 289.78 * 0.94$$

$$\mathbf{Ps = 272.40\ kW}$$

Efficiency of Pump, η = [(Hydr. Power)/(shaft power)] * 100%

= [Ph / Ps] * 100%

$$= [151.40/272.40] * 100\%$$

$$\eta = 55.58\%$$

Table 3.2: Design and actual data of Air Compressor

		Suction Pressure (kg/cm^2)	Delivery Pressure (kg/cm^2)	Flow Rate (m^3/h)	Current ($Amp.$)
Air compressor	Design	1	8.7	534	113
	Actual	1	7.5	528	90

3.3 Energy Conservation in Air Compressors

CCPP at Essar Power Ltd, Surat is having two air compressors. The design data for these systems were recorded from manuals provided by the suppliers. The design parameters such as free air delivery(FAD- m^3/h), suction as well as discharge pressure (kg/cm^2) and current drawn by the motor have been recorded. Actual above data were measured and same have been tabulated as shown in Table 3.2

Compressor performance should be checked in terms of free air delivered, power consumption and pressure of delivered air (at outlet) by the compressor. To evaluate the compressor performance the power consumed should be calculated while considering the outlet pressure. The free air delivered is measured by the flow meter and the pressure at outlet is measured by putting pressure gauge at outlet.

$$\text{Isothermal power (KW)} = \text{P1} \times \text{Q} \times \ln(r) \div 36.7$$
$$= 1.04 \times 534 \times \ln(7) \div 36.7$$
$$= 30.78 \text{ kW}$$

Where,

P1= Suction pressure in kg/cm^2

Q= Free air delivery in m^3/h

r=compression ratio

Isothermal Efficiency = Isothermal power $\div compressorinputpower$

Table 3.3: Air loss through leakages in the compressed air systems

Location	**Air Loss (cu Ft./Year)**	**Loss (Rs.)**
Phase – 1	**25,69,73,704**	**5,65,354**
Phase – 2	**29,12,86,801**	**5,93,786**
Phase – 3	**22,19,72,640**	**5,02,231**

Here motor power is 56 kW and motor efficiency is 93% so ,

$$\text{Compressor input power} = 56 \times 0.93$$
$$= 52.08 \text{ kW}$$

$$\text{Isothermal efficiency} = 30.78{\times}100 \div 52.08$$

$$= 59.10\%$$

$$\text{Specific Power Requirement Of compressor} = \text{motor power} \div discharge flow$$
$$= 56 \div 534$$
$$= 0.1048 \text{ kW / m}^3\text{/h}$$

In order to optimize the power consumption of the air compressor system, the leakages must be sort out. The leakages were found out manually by applying the detergent solution on the air pass line of whole the plant. Soap bubbles were observed at the leakage spots.

The detailed data of the compressed air leakages are given in Appendix -1.
Leakages were found out as per data given by manufacturer of the compressor which are given in Table 3.4. [46]

3.4 Performance Analysis

Power and efficiency were calculated on the basis of actual data collected for the auxiliary equipments listed in Table 3.5. Possible reasons for less efficiency and higher

Table 3.4: Cost of leakage of compressed air

SR NO.	**Leakage Diameter(mm)**	**Air Loss (cu feet/leak/year)**	**Cost (Rs.)**
1	9.52	12,26,53,440	2,69,838
2	6.34	5,45,12,640	1,19,928
3	3.16	1,36,28,160	29,982
4	1.58	34,01,798	7,484
5	0.78	8,49,139	1,868

power consumptions at part load are also given in Table 3.5. From above data calculation have been carried out for power and efficiency which are listed in Table 3.5. Possible reasons for the same have also been tabulated in Table 3.5.

On the basis of the above study, the suggestions were made and same have been implemented by Essar Power Ltd. The details are given in chapter 4.

Table 3.5: Power and Efficiency for BFPs and CEPs

		Power (kW)	Efficiency %	REMARKS POSSIBLE REASONS
BFP - 1	Design	385	75.5	Part load eff. is poor. Temp. is high in spite of dis. Pre. Is lower. Pump over sizing Impeller diameter
	Actual	204.98	51.01	
BFP - 2	Design	541	75.5	
	Actual	204.98	47.79	
BFP - 5	Design	1024	73	Part load eff. is extremely poor. No. Of stage Load variation Pump over sizing operation of combination of pump
	Actual	292.19	32.19	
BFP - 6	Design	1024	73	
	Actual	345.68	338.58	
BFP - 7	Design	500	73	Part load eff. is comparatively good. Dis. Pre. Is higher than what require. Throttling operation of combination of pump
	Actual	241.46	62.95	
BFP - 8	Design	500	73	
	Actual	241.46	62.97	
BFP - 9	Design	500	73	
	Actual	245.98	67.01	
CEP - 1	Design	260	73	Dis. Pre. is higher than what require. Throttling Impeller diameter
	Actual	161.52	57.79	
CEP - 4	Design	260	73	Pump over sizing Impeller diameter
	Actual	151.4	55.58	
CEP - 3,5	Design	260	73	Pump over sizing Operation of combination of pump
	Actual	133.09	53.19	

Chapter 4

SUGGESTIONS, IMPLEMENTATIONS AND IMPROVEMENTS

4.1 Suggestions

After calculating the power and efficiency of the equipments, the reasons for less efficiency and more power consumption are listed out. In order to optimize the power consumptions of the auxiliaries, several suggestions are given which will be implemented later on.
To improve the performance of the pumps, three effective solutions are given as below.

(1) De-staging of the Pump:
Before modification, BFP1 and BFP2 of CCPP at Essar Power Ltd, Surat were consisting of 15 stages. The actual and design data for all BFPs are given in Table 3.1. These pumps were running at part load operation throughout the year. The required parameters were set by throttling of the pump resulting in very poor part load efficiency and higher power consumption. It was studied from the characteristic curves

of the pumps in phase one (BFP1 and BFP2) that 12 stages would be sufficient to met with required operating parameters. However, only BFP1 was selected for de-staging. Following calculations show the savings by de-staging of BFP1.
Present Pump Running Cost at Required Flow:

$$\text{Electrical Input Power P=}(V * I * cos\phi * sqrt3)$$
$$= 6.6 * 104 * 0.85 * 1.732$$
$$= 1010 \text{ kW}$$

$$\text{Electrical Input Power} =$$

$$\frac{\text{Pump shaft power(kw)}}{\text{motor*Fluid coupling*gear box}}$$

$$= [600 \div (0.96 * 0.92 * 0.98)]$$

(from Ref .Curves suggested by the manufacturer)

$$= 693.34 \text{ kW}$$

$$\text{Energy saving after De-staging} = (1010 - 694)$$
$$= 316 \text{ kW}$$

$$\text{Energy savings per Year=}$$

$$\text{Savings per Hour } * \text{ Hours per day } * \text{ No of Days}*Availlability\ factor * unit\ cost$$

$$= 316 * 24 * 365 * 0.85 * 3$$

(3 Rs. is considered as selling prize of 1 unit)

$$= 70{,}58{,}808 \text{ INR}$$

$$\text{Simple Payback Period (SPP)} = \frac{Investment\ cost(C_0)}{Annual\ savings(A)}$$

$$= (697093 * 3)/(70, 58, 808 * 2)$$

= 0.1481 Yrs.

[**Note:- CCPP at Essar Power Ltd. has 3 phases. De-staging of phase one was suggested. This phase consists of 3 pumps running two at a time and one standby. Hence for cost calculations cost of de-staging of 3 pumps has been taken whereas for annual savings, savings by two pumps has been considered.**]

Capital Recovery Factor (CRF)
(at 20 % of discount rate and assuming life of pump 15 years)
= 0.2139

$$\text{B/C Ratio} = \frac{\text{A}}{\text{CRF}*\text{Cost}}$$

$$= (7058808 * 2)/(0.2139 * 697093 * 3)$$

$$\text{B/C} = 31.56$$

$$\text{Cost Of saved Energy} = \frac{C_0 * CRF}{Electricity\ Saved(kWh)}$$

$$= (697093 * 3 * 0.2139)/(2352936 * 2)$$

$$= 0.095 \text{ Rs./kWh}$$

Since simple payback period is very small with higher benefit to cost ratio, the modification is economically viable. The cost of saved energy is also very less in comparison to cost of electricity (Rs 3 per kWh).

(2)Trimming of the Pump Impeller:
BFP 4 of phase two was selected for impeller trimming to enhance the efficiency. Actual and design data for above mentioned pump indicates that at most of the time

this pump runs at 50 % load condition. This section illustrates the energy saving opportunity by impeller trimming.

$$\text{Electrical Input Power} = P = V * I * \cos\emptyset * \sqrt{3}$$

$$= 6.6 * 104 * 0.85 * 1.732$$

$$= 1010 \text{ kW}$$

$$\text{Electrical Input Power} =$$

$$\frac{\text{Pump shaft power(kw)}}{\text{motor*Fluid coupling*gear box}}$$

$$= [390 * 2 \div (0.96 * 0.92 * 0.98)]$$

(from Ref .Curves suggested by the manufacturer)

$$= 901.21 \text{ kW}$$

$$\text{Energy saving after De-staging} = (1010 - 901.21)$$

$$= 108 \text{ kW}$$

$$\text{Energy savings per Year} =$$

$$\text{Savings per Hour } * \text{ Hours per day } * \text{ No of Days} * Availlability\ factor * unit\ cost$$

$$= 108 * 24 * 365 * 0.85 * 3$$

(3 Rs. is considered as selling prize of 1 unit)

$$= 24{,}12{,}504 \text{ INR}$$

$$\text{Simple Payback Period (SPP)} = \frac{Investment\ cost(C_0)}{Annual\ savings(A)}$$

$$= (1500000)/(70, 58, 808 * 2)$$

$$= 0.1481 \text{ Yrs.}$$

[Note:- CCPP at Essar Power Ltd. has 3 phases.Impeller trimming of phase two was suggested. This phase consists of 3 pumps running two at a time and one standby. Hence for cost calculations cost of impeller trimming of 3 pumps has been taken whereas for annual savings, savings by two pumps has been considered.]

Capital Recovery Factor (CRF)

(at 20 % of discount rate and assuming life of pump 15 years)

$$= 0.2139$$

$$\text{B/C Ratio} = \frac{\text{A}}{\text{CRF}*\text{Cost}}$$

$$= (2412504 * 2)/(0.2139 * 1500000)$$

$$\text{B/C} = 15.038$$

$$\text{Cost Of saved Energy} = \frac{C_0 * CRF}{Electricity\ Saved(kWh)}$$

$$= (1500000 * 0.2139)/(8, 04, 168 * 2)$$

$$= 0.1994 \text{ Rs./kWh}$$

Since simple payback period is very small with higher benefit to cost ratio, the modification is economically viable. The cost of saved energy is also very less in comparison to cost of electricity (Rs 3 per kWh).

(3) Installation of Variable Frequency Drive(VFD):

To improve the performance of the pumps of Phase three of CCPP, Esaar Power Ltd, Surat installation of VFD was suggested. The benefits of this modification are discussed in this section.

Before Installation of VFD :

At base load power consumption = 1010 kW

Motor efficiency is 96 % so,

Power output by motor(Input to Fluid Coupling)= 0.96 * 1010 kW

= 970 kW

At part load Fluid coupling efficiency is 74 % so,

Power output from coupling/ Net power input to Gear box = 970 kW * 0.74 = 717.8 kW

After Installation of VFD :

After installation of VFD Fluid coupling can be operated at fully open condition of about 92 %.

Considering 4 % loss at VFD, total power consumption = 1010kW * 0.92 * 0.96

= 892 kW

Net savings using VFD = (892 kW - 718 kW)

= 174 kW

Net savings per year = 174 kW * 24 * 365 * 0.80

= 12,19,392 kWh

Net savings in INR per year = 1219392kWh * 3 INR

= 36,58,176 INR

$$\textbf{Simple Payback Period(SPP)} = \frac{\text{Investment cost}(C_0)}{\text{Annual savings}(A)}$$

$$= (11250000 * 2)/(3658176 * 2)$$

$$= 3.075 \text{ Yrs.}$$

[Note:- CCPP at Essar Power Ltd. has 3 phases. Installation of VFD of phase three was suggested. This phase consists of 3 pumps running two at a time.]

Capital Recovery Factor (CRF) [at 20 % of discount rate and assuming life of VFD 10 years]

$$= 0.2385$$

$$\text{B/C Ratio} = \frac{\text{A}}{\text{CRF}*\text{Cost}}$$

$$= (36,58,176 * 2)/(0.2385 * 11250000 * 2)$$

$$= 1.3634$$

$$\text{Cost Of Saved Energy} = \frac{C_0 * CRF}{Electricity\ Saved(kWh)}$$

$$= (11250000 * 2 * 0.2385)/(1219392 * 2)$$

$$= 2.200 \text{ Rs./kWh}\_$$

SPP, B/C ratio and Cost of saved energy indicates that the installation of VFD is economically viable.

The summery of all above modifications is given in the Table 3.6.

Table 4.1: Summery of suggestions for improving pump performance

SR NO.	Suggested Method	Annual savings (Rs/ year)	Simple Payback Period (Years)	Benefits/ Cost (B/C Ratio)	Cost of Energy Saved(Rs/kWh)
1	De-staging	70,58,808	0.1481	31.56	0.095
2	Impeller Trimming	24,12,504	0.3108	15.038	0.1994
3	Installation of VFD	36,58,176	3.075	1.3634	2.200

4.2 Implementation of Suggestions

After suggesting all above recommendations were implemented.

(1) De-staging of the Pump:

$$\text{Electrical Input Power P=}(V * I * cos\phi * sqrt3)$$
$$= 6.6 * 104 * 0.85 * 1.732$$
$$= 1010 \text{ kW}$$

$$\text{Electrical Input Power} =$$

$$\frac{\text{Pump shaft power(kw)}}{\text{motor*Fluid coupling*gear box}}$$

$$= [692 \div (0.96 * 0.92 * 0.98)]$$

(from Ref .Curves suggested by the manufacturer)

$$= 799.53 \text{ kW}$$

$$\text{Energy saving after De-staging} = (1010 - 800)$$
$$= 210 \text{ kW}$$

$$\text{Energy savings per Year=}$$

$$\text{Savings per Hour} * \text{Hours per day} * \text{No of Days}*Availlability\ factor * unit\ cost$$

$$= 210 * 24 * 365 * 0.85 * 3$$

(3 Rs. is considered as selling prize of 1 unit)

= 46,90,980 INR

$$\text{Simple Payback Period (SPP)} = \frac{Investment\ cost(C_0)}{Annual\ savings(A)}$$

$$= (697093 * 3)/(46,90,980 * 2)$$

$$= 0.2229 \text{ Yrs.}$$

Capital Recovery Factor (CRF)

(at 20 % of discount rate and assuming life of pump 15 years)

= 0.2139

$$\text{B/C Ratio} = \frac{\text{A}}{\text{CRF*Cost}}$$

$$= (46,90,980 * 2)/(0.2139 * 697093 * 3)$$

$$\text{B/C} = 20.97$$

$$\text{Cost Of saved Energy} = \frac{C_0 * CRF}{Electricity\ Saved(kWh)}$$

$$= (697093 * 3 * 0.2139)/(15,63,660 * 2)$$

$$= 0.2860 \text{ Rs./kWh}$$

[Note:- CCPP at Essar Power Ltd. has 3 phases. De-staging of phase one was suggested. This phase consists of 3 pumps running two at a time and one standby. Hence for cost calculations cost of de-staging of 3 pumps has been taken whereas for annual savings, savings by two pumps has been

considered.]

(2)Trimming of the Pump Impeller:

$$\text{Electrical Input Power } = P = V * I * \cos\emptyset * \sqrt{3}$$

$$= 6.6 * 104 * 0.85 * 1.732$$

$$= 1010 \text{ kW}$$

$$\text{Electrical Input Power} =$$

$$\frac{\text{Pump shaft power(kw)}}{\text{motor*Fluid coupling*gear box}}$$

$$= [400 * 2 \div (0.96 * 0.92 * 0.98)]$$

(from Ref .Curves suggested by the manufacturer)

$$= 924.32 \text{ kW}$$

$$\text{Energy saving after De-staging} = (1010 - 924.32)$$

$$= 85 \text{ kW}$$

$$\text{Energy savings per Year=}$$

$$\text{Savings per Hour } * \text{ Hours per day } * \text{ No of Days}*Availlability\ factor * unit\ cost$$

$$= 85 * 24 * 365 * 0.85 * 3$$

(3 Rs. is considered as selling prize of 1 unit)

$$= 18{,}98{,}730 \text{ INR}$$

$$\text{Simple Payback Period (SPP)} = \frac{Investment\ \ cost(C_0)}{Annual\ \ savings(A)}$$

$$= (1500000)/(18, 98, 730 * 2)$$

$$= 0.3950 \text{ Yrs.}$$

[Note:- CCPP at Essar Power Ltd. has 3 phases.Impeller trimming of phase two was suggested. This phase consists of 3 pumps running two at a time and one standby. Hence for cost calculations cost of impeller trimming of 3 pumps has been taken whereas for annual savings, savings by two pumps has been considered.]

Capital Recovery Factor (CRF)

(at 20 % of discount rate and assuming life of pump 15 years)

$$= 0.2139$$

$$\text{B/C Ratio} = \frac{\text{A}}{\text{CRF}*\text{Cost}}$$

$$= (18, 98, 730 * 2)/(0.2139 * 1500000)$$

$$\text{B/C} = 11.83$$

$$\text{Cost Of saved Energy} = \frac{C_0 * CRF}{Electricity\ \ Saved(kWh)}$$

$$= (1500000 * 0.2139)/(6, 32, 910 * 2)$$

$$= 0.2534 \text{ Rs./kWh}$$

Since simple payback period is very small with higher benefit to cost ratio, the modification is economically viable. The cost of saved energy is also very less in comparison to cost of electricity (Rs 3 per kWh).

(3) Installation of Variable Frequency Drive(VFD):

After Installation of VFD :

After installation of VFD Fluid coupling can be operated at fully open condition of about 92 %.

Considering 4 % loss at VFD, total power consumption = 1010kW * 0.92 * 0.96

= 892 kW

Net savings using VFD = [892 kW -(407*2) kW]

= 78 kW

Net savings in INR per year = 78 * 24 * 365 * 0.80 * 3 INR

= 16,39,872 INR

$$\textbf{Simple Payback Period(SPP)} = \frac{\text{Investment cost}(C_0)}{\text{Annual savings(A)}}$$

$$= (11250000 * 2)/(16,39,872 * 2)$$

= 6.86 Yrs.

[**Note:- 2 VFD are to be installed in the phase-2 because at a time 2 pumps are running and so minimum 2 no. of VFD are required.**]

Capital Recovery Factor (CRF) [at 20 % of discount rate and assuming life of VFD 10 years]

= 0.2385

$$\text{B/C Ratio} = \frac{\text{A}}{\text{CRF}*\text{Cost}}$$

$$= (16,39,872 * 2)/(0.2385 * 11250000 * 2)$$

= 0.611

Table 4.2: Savings of Compressed air leakages

Location	Air Loss (cu Ft./Year)	Annual Savings(Rs/year)
Phase – 1	25,69,73,704	5,65,354
Phase – 2	29,12,86,801	5,93,786
Phase – 3	22,19,72,640	5,02,231

$$\text{Cost Of Saved Energy} = \frac{C_0 * CRF}{Electricity\ Saved(kWh)}$$

$$= (11250000 * 2 * 0.2385)/(5,46,624 * 2)$$

$$= 4.908 \text{ Rs./kWh\_}$$

After finding all leakages, they must be sealed in order to reduce the deliver air duty of the compressor. The leakages were sealed by fitting the pipe joints, sticking the sealing packages, painting and at last if necessary by replacing the pipe or pipe joints. Loss due to the leakages have overcome as shown in Table 3.7.

4.3 Improvements

As discussed in the earlier sections, the exhaustive data were collected for the auxiliary equipments and suggesting same have been analyzed. The suggestions for improvements in efficiency and power were made and same have been implemented. After implementation of the above recommendations, power, efficiency etc, were again calculated. For economic viability economic indices like SPP, B/C ratio and cost of saved energy were calculated. Results are tabulated in Table 4.3. It indicates that in all four recommendations SPP is less with sizable annual savings. Cost of saved energy is also very less in comparison with existing energy cost Rs. 4 per kWh.

Table 4.3: Comparison of suggested and implemented data

SR NO.	Suggested Method	Annual Savings - suggested (Rs/ year)	Annual Savings-implemented (Rs/ year)	Simple Payback Period-suggested	Simple Payback Period-implemented
1	De-staging	70,58,808	46,90,980	0.1481	0.2229
2	Impeller Trimming	24,12,504	18,98,730	0.3108	0.3950
3	Installation of VFD	36,58,176	16,39,872	3.075	6.86
4	Compressor Leakages	16,61,371	16,61,371		

Chapter 5

CONCLUSION AND FUTURE SCOPE

5.1 Conclusion

It is very clear from earlier discussion that the reduction in auxiliary power consumption by improving their efficiency is need of a day for thermal power plants. Here an attempt is made to optimize the auxiliary power consumption by pumps and compressors. The major conclusions derived from the present study are as follows:

1) The study of operating parameters of the present auxiliaries show that the efficiency of the said system is poor and power consumption is very high at off design conditions in comparison to standard values available in the literature provided by the manufacturers.

2) The performance analysis was carried out on the basis of comparison of actual power consumption with that of designed data.

3) Performance analysis reveals that part load efficiency of pumps is very poor. Per-

formance of the pumps becomes very poor as the load on the system varies.

4) Possible reasons for that poor performance have also been found out. The suggestions for improvements in efficiency and power for pump like de-staging, impeller trimming and installation of VFD were given and also been implemented.

5) After implementation of the above recommendations, power, efficiency etc, were again calculated. For economic viability economic indices like SPP, B/C ratio and cost of saved energy were calculated. Results indicate that in all four recommendations SPP is less with sizable annual savings. Cost of saved energy is also very less in comparison with existing energy cost Rs. 4 per kWh.

5.2 Future Scope

In the present work the auxiliaries like boiler feed pumps, condensate extraction pumps and air compressors have been optimized for higher energy efficiency. In order to further improve the efficiency of thermal power plants following are the suggestions which may be taken up as the scope for future work.
Optimization of air conditioning systems by reducing the losses and by improving the performance of various elements.
Water is an essential element of thermal power plants operating on rankine cycle. However the increased salt concentration in drum water may results in to fouling which in turn reduces heat transfer. The detailed analysis is required to find out the solution.
The optimization of cooling tower parameters is also required which may result in to higher plant efficiency.

Appendix A

Leakages-Air Loss-Loss in Rs. (BPOL)

	Leakages-Air Loss-Loss in Rs. (BPOL)			
SR NO.	**Location of leakage**	**Size (mm)**	**Air loss (Cu Ft./year)**	**Loss (Rs.)**
1	Water Softening plant Softener-2 Valve-19	1.58	34,01,598	7,484
2	Water Softening plant-Softener-5 Valve-17	0.793	8,49,139	1,868
3	Water Softening plant-Softener-6 Valve-17	0.793	8,49,139	1,868
4	Water Softening plant-Softener-6 Valve-18	0.793	8,49,139	1,868
5	Water Softening plant-Softener-6 Valve-19	0.793	8,49,139	1,868
6	Water Softening plant-Softener-7 Valve-24	0.793	8,49,139	1,868
7	Water Softening plant-Softener-7 Valve-17	0.793	8,49,139	1,868
8	Water Softening plant-Softener-8 Valve-24	0.793	8,49,139	1,868
9	Water Softening plant-Softener-9 Valve-24	0.793	8,49,139	1,868
10	Water Softening plant-Softener-5 Valve-21	0.793	8,49,139	1,868
11	Water Softening plant-Near PSF blower in three pipes(Top Side)	1.58	34,01,598	7,484
12	Water Softening plant-Softener-11 Valve-24	0.793	8,49,139	1,868
	Phase -1			
13	Auxiliary Station Building(BPOL)- Filter downstream-1 isolation valve in handle (BV-99)	3.175	1,36,28,160	29,982
14	Auxiliary Station Building(BPOL)- Filter downstream-2 isolation valve (BV-100)	0.793	8,49,139	1,868
15	Receiver Upstream -1 valve Blind (Top Side)	0.793	8,49,139	1,868

16	Drier - 2B Pressure Filter T-P Gauge	0.793	8,49,139	1,868
17	Drier - 2A Upper side Near Flange in Instrument Air Line	0.793	8,49,139	1,868
18	Drier - 2B Bottom side Near Pressure Gauge in Instrument Air Line	0.793	8,49,139	1,868
19	Drier - 1A Left side near inlet in Instrument Air Line	0.793	8,49,139	1,868
20	Drier - 1A Upper side near inlet in Instrument Air Line	0.793	8,49,139	1,868
21	Drier - 1A Upper side in Pressure Regulator (Back side)	0.793	8,49,139	1,868
22	Between Drier 2A -2B in Instrument Air Line	0.793	8,49,139	1,868
23	Between Drier 2A -2B in Instrument Air Line	0.793	8,49,139	1,868
24	Drier 2B Plug	1.58	34,01,598	7,484
25	Receiver Tank-2 in Pressure Gauge	1.58	34,01,598	7,484
26	Outside of building Near HRSG-1 Near Pressure Heating station (HP-SH/ TCV-4326) in valve	1.58	34,01,598	7,484
27	Outside of building Near HRSG-1 Near Pressure Heating station (HP-SH/ TCV-4326) in coupling	1.58	34,01,598	7,484
28	Near HP-SH TCV-4326 Upstream in instrument Air Line	0.793	8,49,139	1,868
29	Near HP-SH TCV-4326 Upstream in Pressure Regulator(Both In Upstream-downstream)	0.793	8,49,139	1,868

30	Near Flash Tank to IBD CV in phase- 1	0.793	8,49,139	1,868
31	Quenching Line for IBD CV main isolation valve	1.58	34,01,598	7,484
32	On top of HRSG-1 HP Economizer Bypass CV	1.58	34,01,598	7,484
33	On top of HRSG-1 HP Economizer Bypass CV in Upstream	0.793	8,49,139	1,868
34	On top of HRSG-1 CPH Bypass Valve-1 main header in elbow & Instrument Air Line	3.175	1,36,28,160	29,982
35	On top of HRSG-1 CPH Bypass Valve-1 main header in Instrument Air Line **NOTE:- Whole Line should be checked(34 & 35)**	3.175	1,36,28,160	29,982
36	On top of HRSG-1 in main Instrument Air Line	1.58	34,01,598	7,484
37	Near CPH Bypass Valve (Half Valve) in instrument Air Line	0.793	8,49,139	1,868
38	In valve Near Chimney(HRSG-1) Stack	0.793	8,49,139	1,868
39	Near HP Start up Vent in Instrument Air Line(Phase -1)	0.793	8,49,139	1,868
40	Near HP Start up Vent in valve Upstream & downstream (Phase -1)	1.58	34,01,598	7,484

41	Near Air Receiver Tank- 1 In Instrument Air Line to APU Isolation Valve	1.58	34,01,598	7,484
42	GT Hall Near Host connection point in coupling-1 In PHASE -1	3.175	1,36,28,160	29,982
43	GT Hall on VTR side in Pressure Regulator in PHASE-1	1.58	34,01,598	7,484
	NOTE:-At left ,Outside from GT Hall Leakage no. 44 to 57			
44	At left ,Outside from GT Hall in Instrument Air Line Near LP Bypass Line in pipe joint	3.175	1,36,28,160	29,982
45	Near LP Bypass PCV-5002 in Receiver Tank on top side	1.58	34,01,598	7,484
46	In Pressure Gauge of other Receiver and also at top side	1.58	34,01,598	7,484
47	Near Receiver tank downstream Elbow (very big leakages)	3.175	1,36,28,160	29,982
48	Near LP Bypass control valve near Pressure Gauge top side-ABB	0.793	8,49,139	1,868
49	Near LP Bypass control valve in small pipe joint coupling	0.793	8,49,139	1,868
50	Near HP Bypass near gate in valve pipe joint (Back side)	0.793	8,49,139	1,868
51	Near HP Bypass near gate in valve	1.58	34,01,598	7,484
52	Near MOV -4607 back side in pipe joint	1.58	34,01,598	7,484
53	Near MOV -4902 left side in pipe joint	1.58	34,01,598	7,484

54	Near HP Bypass Receiver tank in valve	0.793	8,49,139	1,868
55	Upstream of HP Bypass Receiver tank in valve	1.58	34,01,598	7,484
56	Near other Receiver tank in isolation valve	0.793	8,49,139	1,868
57	Down to Receiver tank in pipe joint	3.175	1,36,28,160	29,982
58	Near Ejector leader towards down going in CEP in pipe joint	1.58	34,01,598	7,484
59	Near Ejector leader towards up going in CEP in pipe joint	0.793	8,49,139	1,868
60	Towards FW Heater inlet line in valve	1.58	34,01,598	7,484
61	Near Bypass to LP Drum LVL/CV 4125 downstream in pipe joint	1.58	34,01,598	7,484
62	Near CEP discharge mini recirculation CV 4122 in pipe joint (at end)	1.58	34,01,598	7,484
63	Near CEP discharge mini recirculation CV 4122 in valve	3.175	1,36,28,160	29,982
64	Down to stairs (in ground) HP Bypass TCV	3.175	1,36,28,160	29,982
65	In STG-1 near GSS in pipe joints (Near aux. PRDS drain condenser ejector)	1.58	34,01,598	7,484
66	In STG-1 near GSS in valve	1.58	34,01,598	7,484

67	In STG-1 near gland sealing line in pipe joint tapping	1.58	34,01,598	7,484
	In STG- 1 near condenser flash box in series of many valves 68 to 72			
68	In 4th valve	1.58	34,01,598	7,484
69	In 3rd valve	1.58	34,01,598	7,484
70	In 5th valve	1.58	34,01,598	7,484
71	In 6th valve	1.58	34,01,598	7,484
72	In pipe joint near 5th and 6th valve	1.58	34,01,598	7,484
73	Near GSS TCV in pipe joint	1.58	34,01,598	7,484
74	Near GSS TCV in other valve as well as in pipe joint	0.793	8,49,139	1,868
TOTAL-		**104.517**	**25,69,73,704**	**5,65,354**

References

[1] Vitaly A. Prisyazhniuk. Alternative trends in development of thermal power plants. *Applied thermal engineering*, 2007.

[2] www.indiastat.com. February, 2010.

[3] www.indiastat.com. February, 2010.

[4] Yousef S. H. Najjar. Efficient use of energy by utilizing gas turbine combined system. *Applied thermal engineering*, 2000.

[5] R. Farmer. 227 mw frame 9fa will power epon's 1.7 gw combined cycle. *Gas turbine world*, June 2003.

[6] H. Reinheimer. 650 mw combined cycle plant for gandhar, india.

[7] S.C.KAUSHIK I.P.S. PAUL, S.P.SABBERWAL. Optimum economy and efficiency in energy consumption during start-up and shut-down of 210mw thermal power stations.

[8] Sozbir N Sarac HI Brawn DM Yigit KS, Callil. Experimental investigation of optimum centrifugal pumps speeds. *In Second international conference on pumps and fans, vol. II*, October 1995.

[9] Sozbir N Yigit KS Sarac HI, Guven R. New method for simulation of centrifugal pump plants. December 1993.

[10] Bar-Cohen A Yigit KS, Arik M. New chf enhancement techniques:passive impeller micropump and gravity driven fluid flow. *In 5th world conference on experimental heat transfer, fluid mechanics, and thermodynamics*, 24-28 September 2001.

[11] Toklu E Yigit KS Calli I, Oztuirk R. An experimental study on front declined open impeller vane in circulation pumps. *In 10th Turkish national conference on thermal sciences and technologies, Vol. 1*, 6-8 September 1995.

[12] Ertoz AO. Energy efficiency in pumps. *In: 6th national installations engineering and congress*, 2003.

[13] *The pump terms guide no. 2*. POMSAD, 1997.

[14] *ISO 3555. Fundamentals of acceptance experiments for radial, mixed flow and axial pumps, class B,no.3*. POMSAD, 1998.

[15] *ISO 2548. Fundamentals of acceptance experiments for radial, mixed flow and axial pumps, class C,no.4*. POMSAD, 1998.

[16] *ISO 9905. Technical properties of radial (centrifugal) pump, class I, no. 7*. POMSAD, 2000.

[17] *ISO 9908 Technical Properties of Radial (Centrifugal) Pump, Class III, no.8*. POMSAD, 2000.

[18] *WORLD PUMPS*. May,2007.

[19] *Improving Pumping System Performance-A sourcebook for Industry*. Hydraulic Institute, 2007.

[20] *Lifelong cost in pumps: lifelong cost for plants that using pumps.no. 12*. POMSAD, 2001.

[21] Desmur G. Kovats A. Pumps, ventilators and radial and axial compressors. 1968.

[22] Lakshminarayana B. Methods of predicting the tip clearance effects in axial-flow turbomachinery. 1970.

[23] Lakshminarayana B. End wall and profile losses in a flow-low-speed axial flow-compressor rotor, no.108. 1986.

[24] Cherkassky VM. *Pumps fans compressors*. Mir publishers, 1980.

[25] Kovats DA. *Design and performance of centrifugal and axial flow pumps and compressors*. The Macmillan Company, 1964.

[26] Kaya D. An experimental study on regaining the tangential velocity energy of axial flow pump. *Energ Convers Manage*, 2003.

[27] Olgun H Tiris M Kaya D, Sarac HI. A study of axial flow pump impellers-effects of pump design parameters on its performance. *In Proc the twelfth international symposium on transport phenomena*, 2000.

[28] *Performance Optimization Training Manual: Fans, Pumps, and Blowers*. US, 1998.

[29] G. Singh. Energy and environmental benefits of pump impeller trimming in commercial building heating and cooling systems. *PhD. Thesis*, 1997.

[30] *Improving Pumping SystemPerformance,ASourcebook for Industry.* Hydraulic Institute and United States Department of Energy Industrial Technologies Program, 2006.

[31] .W. Mitchell G. Singh. Energy savings from pump impeller trimming. *ASHRAE Journal 40 (4)*, 1998.

[32] I. Weybourne A.Y. Maurice. Improving the energy efficiency of pumping systems. *J. Water SRT - Aqua 50*, 2001.

[33] Mc Nally. Increasing the centrifugal pump performance by modifying the impeller. *The Mc Nally Institute Book Division*, 1998.

[34] L.M. Tsang. Theoretical account of impeller trimming of the centrifugal pump. *Proc. of the Institution of Mechanical Engineers, Part C, J. Mech. Eng. Sci.206 (3)*, 1992.

[35] I.J. Karassik. Centrifugal pump clinic. *Marcel Dekker Inc.*, 1989.

[36] L. Nelik. *Centrifugal and rotary pumps: fundamentals with applications.* CRC Press, 1999.

[37] R.R. Ross V.S. Lobanoff. *Centrifugal pumps: design and application.* Gulf Professional Publishing Press, 1992.

[38] P. Cooper C.C. Heald. (Eds.) I.J. Karassik, J.P. Messina. *Pump Handbook.* McGraw-Hill, 2001.

[39] A. Custodio L. Bachus. *Know and understand of centrifugal pumps.* Bachus Company, 2003.

[40] Stan Shiels. Shiels turbomachinery.

[41] Austanes F.F Coopern P. Cavitation in boiler feed pumps. *Power Plant Feed Pumps Symposium*, 1980.

[42] Chau D Sarac H.I. Kaya D, Phelan P. Energy conservation in compressed-air systems. *Int J Energy*, 2002.

[43] Kelly Kissock Chris Schmidt. Power characteristics of industrial air compressor. *Department of Mechanical and Aerospace engineering.*

[44] C anka Kilic F. Kaya D. Energy conservation opportunity in vsd system - a case study. *World energy engineering congress proceedings*, September 2004.

[45] Fact sheet. Energy efficiency in air compressor. *Department of energy*, January 2004.

[46] Fact sheet. Fluid loss cost money. *Hankle Loctite Pvt. Ltd.*