Effect of cooling charge air on the gas turbine performance and feasibility of using absorption refrigeration in the “Kelanitissa” power station, Sri Lanka

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Abstract

One of the drawbacks of the gas turbine is that performance drops rapidly when ambient air temperature increases. This is a major drawback for gas turbines operated in a tropical country like Sri Lanka. In Colombo, commercial capital of Sri Lanka where this study was carried out, the ambient temperature typically varies between 25 °C and 32 °C.

The Kelanitissa gas turbine plant has single shaft gas turbines (GE MS5001 R) operated in open cycle which use diesel as fuel (designed for dual fuel) at a designed heat rate of 13,980 kJ/kWh and an electrical efficiency of 25.8%. The designed exhaust temperature is 513 °C.

In this study, Kelanitissa gas turbine unit was used for assessment of the performance with the changes in ambient air temperature. Two approaches were used to study this phenomenon. Firstly, the performance parameters were calculated by using actual data acquired by the operation history of the power plant. Secondly, the performance was analyzed using thermodynamic principles. Then results of the two approaches were compared.

The present performance values of the studied gas turbine, when compared to designed values, showed a very poor performance due to predominantly high ambient air temperatures. Originally designed for an efficiency of 25.8%, the maximum efficiency achieved at 33 °C was only 21.2%. This translates into a 4.6 %-point reduction in efficiency at 33 °C ambient temperature.

Estimated cooling load for the proposed inlet air cooling is 679.87 RT. Cost per unit cooling load of the reference 2-stage direct-fired absorption system is $751-721 (according to 600RT-700RT). For the worst case scenario the value of $751 per RT and exhaust system constituting 98% of the cost of a market-ready direct fired system (Broad Inc., 2008) can be used. This results in $736/RT as the cost for an absorption chiller system driven by exhaust heat. Total cost for the 679.87 RT system is $ 500,370.72 (Rs. 65Mn).

Payback period of the project is 11 years but the present value after 19 years is exceeding the project cost. Present value for 19 years is Rs. 65.86 Mn. Bringing down the temperature from an average of 27°C to the ISO value of 15 °C would give Rs. 6 Mn of annual savings.
Acknowledgments

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Nomenclature

\( \eta \) - Efficiency
\( \gamma \) - Specific heat ratio
\( Q_s \) - Sensible cooling load (kW)
\( \dot{v}_a \) - Volume flow rate of air at the ISO conditions (m\(^3\)/kg)
\( t_{aa,db} \) - Dry bulb temperature of the ambient air (°C)
\( t_{ci} \) - Compressor inlet temperature (°C)
\( \dot{v}_a \) - Specific volume of the wetted air per kilogram of dry air (m\(^3\)/kg)
\( P_{atm} \) – Ambient pressure in (kPa)
\( X_{aa,db} \) - Specific humidity of the ambient air at its dry bulb temperature (kg/kg)
\( M_v \) - Molecular weight of the water vapor (kg/kmol)
\( M_a \) - Molecular weight of the air (kg/kmol)
\( P_s \) - The saturation vapor pressure at inlet air dry bulb temperature (kPa)
\( C_{p,v} \) - Specific heat at constant pressure of the water vapor (kJ/kgK)
\( x_{c,i}^* \) - Specific humidity at the compressor inlet temperature and (kg/kg)
\( C_{p,w} \) - Specific heat of liquid water at constant pressure (kJ/kgK)
\( r_p \) – Pressure ratio of the compressor
\( T_{a,in} \) – Inlet air temperature (K)
\( T_{amb} \) – Ambient temperature
\( W_{cyc} \) - Cyclic work
\( W_t \) - Turbine work
\( W_c \) - Compressor work
\( h \) – Enthalpy
\( S \) - Entropy
\( T_f \) – Firing temperature
\( LHV \) – Lower heat value
\( \dot{m}_f \) – Fuel mass flow rate
\( \dot{m}_a \) – Air mass flow rate
\( \dot{m}_{gas} \) – Gas mass flow rate
\( (r_p)_{eopt} \) – Pressure ratio optimized to efficiency
\( (r_p)_{pwopt} \) – Pressure ratio optimized to power output
\( \eta_t \) – Turbine efficiency
\( \eta_c \) - Compressor efficiency
\( \eta_{cyc} \) – Cyclic efficiency
\( PV \) – Present Value
\( A \) – Annual return
\( r \) – Discounted rate
\( n \) – Number of years
1. Introduction

1.1 General Overview

A gas turbine (GT) is a heat engine that uses high-temperature, high-pressure gas as the working fluid. Part of the heat supplied by the gas is converted to mechanical work. In most cases, hot gas is produced by burning a fuel in air. This is why gas turbines are often referred to as "combustion" turbines. Because gas turbines are compact, lightweight, quick-starting, and simple to operate, they are widely used for power generation and in aircraft propulsion. The capacity of gas turbines ranges from micro size to approximately 500 MW maximum per unit.

Gas turbine power plants are ideal for providing certain midrange and peaking electric power to the grid for onsite power generation. Gas turbines are also responsive to load variations and are very cost effective and feasible in combined cycle operation. They are commonly used in combined cycle arrangements with steam Rankine bottoming cycle. There are some special cycles like BIGCC (biomass integrated gasification combined cycle) where a gas turbines is utilized due to its high power-to-weight ratio and high mass flow rate. Common fuels are NG (Natural gas), Diesel (HSD) or sometimes HFO (heavy fuel oil) to power the gas turbines. The simple gas turbine layout for power generation depicting main components is shown in Fig. 1 below.

![Figure 1.1 Schematic layout of a simple gas turbine plant](image-url)
1.2 Kelanitissa gas turbine Power Station

Sri Lanka is an Island with lots of variations of climates and landscape. Central part of Sri Lanka has hills and mountains with a maximum peak of 2524m above sea level. Hill side is lower in temperature due to elevation and lowest can reach down to about 4°C. And the low altitude coastal area of the Sri Lanka divides into two parts as dry zoon and wet zone. Normal average temperature of this dry zone is above 30°C and it can reach up to maximum of 40 to 45°C. Colombo is the commercial capital of Sri Lanka sustaining a population of 2.3 million which is more than 10% of the total population. Colombo is situated in the western coastal area of Sri Lanka. Average temperature is about 27°C and maximum can reach up to about 35°C. Total electrical energy generation in Sri Lanka is about 10,000 GWh/year. Island wide operated in a single grid with total grid capacity of about 3300 MW. Main sources are Thermal power (diesel, coal, Heavy Fuel oil, Naphtha), Hydro power and Wind power. Presently the total thermal power capacity of Sri Lanka is about 1600 MW consisting of open cycle gas turbine plants, Diesel engine plants, combined cycle power plants and coal power plants. Earlier the electrical energy load was catered only from the Hydro Power, but due to growing demands the Ceylon Electricity Board (CEB) established its first gas turbine power plant in Sri Lanka in 1981 – the Kelanitissa gas turbine power station.

Kelanitissa Power Station (Figure 1.2), which is subsidy of Ceylon Electricity Board has five gas turbines at 22 MW rated power each, and one gas turbine of larger capacity (110MW).
Those are grid connected and mostly started in peak hours to cater for the peak demand. Sri Lanka as a third world country, electricity from open cycle gas turbine (Diesel) is an extravagance solution for the energy demand. The CEB has to bare cost of about Rs.60/kWh ($0.50/kWh) per unit. But the unit selling price for electricity is far below the production cost. Hence it is not economical to run open cycle gas turbines and also environmental hazards due to high waste heat rejection are involved. Hence these gas turbines should be modified in such a way so that to increase their economic viability in operation and also to improve their environmental aspects. Nowadays, waste heat from gas turbines can be used constructively in several ways. For example in Combined Heat and Power plants, combined cycle power plants, chiller plants etc. All these methods can optimize the use of energy from the fuel. But if we can increase the efficiency of the existing gas turbine, it can directly deliver a cut down of energy cost. Charge air cooling is a plausible method to increase the power output and the efficiency of the existing gas turbines.

Following specifications are from the 5 identical gas turbines mentioned above:

- **Model**: GE MS5001 (Frame 5)
- **Make**: John Brown
- **Rated output of Turbine**: 22,480 kW
- **Generator Output**: 25 MVA
- **Fuel**: Auto Diesel
- **Year of Manufacture**: 1980

This is a single shaft gas turbine operated in open cycle which is using diesel as fuel (designed for dual fuel). It features an axial compressor with 16 stages and a turbine with 2 stages rotating at a speed of 5100 rpm. Firing temperature is about 2,500°F (1371°C) to 3200°F (1760°C) and turbine inlet temperature is in the range 1650°F - 1750°F (899°C - 954°C). Designed heat rate of the gas turbine is 13,980kJ/kWh at an efficiency of 25.8%. The designed exhaust temperature is 513°C.

For the study of GT performance with varying inlet temperature, one of these gas turbines was chosen as reference. The gas turbine has been designed around a limiting firing temperature (Tf), which ensures that the hot gas path parts are not subjected to excessive thermal stress, and as a measure of Tf, the exhaust temperature Tx is used in the control system. It would not be practical to directly measure Tf since this could be around 3000 °F (1649°C) but by using 12 thermocouples around the exhaust plenum we can obtain an accurate indication of Tf through Tx. As the unit is loaded up on a hot day it will reach the limiting firing temperature level at a high exhaust temperature and it operates on a low load with low fuel consumption. On a cold day the same limiting temperature will be reached at a lower Tx and it operates at maximum load with high fuel consumption. So increasing the ambient temperature, the power output proportionately
reduces while increasing the heat rate and the exhaust gas temperature. In this case the gas
turbine lowers its output with the inlet temperature by two ways. One is due to lowering of the
air mass flow rate and the other is lowering of the power by lowering the fuel in order to keep
down the Tf within its safe limit.

Figure 1.3 - Typical behavior of the GT unit 05 during the day

Figure 1.3 shows the variation of the ambient temperature in a typical day. It also shows
the variation of the compressor discharge pressure and power output. As can be seen from the
graph when ambient temperature increases, power output and the compressor discharge pressure
are reduced. From the turbine past data, it can be observed that the prevailing ambient conditions
of GT operation feature an air temperature variation between about 25°C to about 32°C.

1.3 Problem statement

One of the major drawbacks of gas turbines operating in tropical countries like Sri Lanka
is the rapid performance drop when ambient air temperature increases. In the area where the gas
turbine considered for this study is located (Kelanitissa plant near Colombo), the ambient air
temperature varies between 25°C and 32°C, which has also been recorded in the past data
acquired from the power station. From the daily readings of Kelanitissa gas turbines, it can be
noticed that full load capacity drops during hot daytime with about 1-2 MW out of the 20MW
output achieved in morning and night hours.
1.4 Objectives

This research and evaluation study will focus on the possible performance enhancement of the reference gas turbine by supplying charge air cooling. Main objective is to derive the behavior of the gas turbine with the ambient temperature variation. A thermodynamic model for the inlet air cooling performance will then be developed and approximated by simulations. Further, the use of available exhaust gas heat to drive the charge air cooler and its technical and economic feasibility will be studied for analyzing the adoption of a practical absorption refrigeration system in the Kelanitissa Power station.
2. **Methodology**

In this research the main focus was to investigate the present situation of the gas turbine inlet cooling. As the first step, literature survey was carried out to study about the inlet cooling methods, advantages and disadvantages in specific methods, theoretical background of the inlet cooling, gas turbine operation. Relationship between inlet cooling and gas turbine performance was analyzed and discussed.

One of the *Kelanitissa* gas turbine units was used for assessing the performance with the changes in ambient air temperature. Two approaches were applied to study this phenomenon. Firstly, changes were calculated by actual data acquired by the operation history of the power plant. The required data needed for the analysis was acquired from the data sheets of the power plant and from the machine manuals. These data were analyzed in view of determining the patterns of variations of performance with air temperature. Secondly, the performance was analyzed using thermodynamic principles. Then the results of the two approaches were compared.

In the next phase, energy calculations were performed to obtain the required level of the charge air cooling and to select a suitable capacity of absorption chiller. Then the costs involved were calculated. Finally, the technical and economic feasibility of the system were analyzed and discussed in details.
3. Literature Review

3.1 Background and past work

Industrial gas turbines are usually rated according to ISO conditions, which are (OMIDVAR, Bob, 2001):

1. Ambient dry bulb temperature: 15°C
2. Relative humidity: 60%
3. Wet bulb temperature: 7.2°C
4. Atmospheric pressure: 1.01325 bar (sea level)

But in real situations gas turbines may operate far from these ISO conditions. The varying ambient conditions affect the GT performance drastically. Most variable condition is the ambient air temperature. If a gas turbine works in an ambient temperature of 35°C its output drops by 20% and fuel consumption increases by 12.5% (PATEL, Pankaj K., 2003), as can be viewed in Figure 3.1.

The compressor work in a GT plant can be expressed by the equation (3.1) (B. DAWOUD, Y.H. Zurigat, J. Bortmany, 2005):

\[ Wc = T_{a,in} \cdot Cp \cdot \left( \frac{\gamma - 1}{\gamma} \right) (3.1) \]

According to this equation when inlet air temperature drops, the compressor work tends to reduce and as a result GT power output increases. Figure 3.1 shows the generic variations of the major GT parameters with respect to the designed condition when the inlet air temperature is changing. Here converging of the lines at 15°C shows the design point of the gas turbine. Figure 3.1 also shows how power output and exhaust flow decrease with increase of inlet air temperature. Also heat rate and exhaust temperature increase with the inlet temperature.

Table 3.1 shows the power and efficiency variation with the inlet temperature of four commercial models of gas turbines. Here it can be clearly noticed that the efficiency and the power output of the gas turbines drop at higher inlet temperatures. Hence, cooling of the inlet air before the compressor can increase the gas turbine efficiency and power output. For the Frame 5 model in Table 3.1, when the gas turbine inlet temperature drops from 35°C to 15°C, the power output increases by around 15% and efficiency increases by 1.6%.
Many have carried out research on gas turbine inlet air cooling and behavior of gas turbines in terms of different parameters. Some studies have been directly related to the inlet air cooling before the gas turbine and some are not directly related, for example “Integration of absorption cooling systems into micro gas turbine tri-generation systems using biogas: Case study of a sewage treatment plant” (JOAN CARLES BRUNO, Víctor Ortega-López, Alberto Coronas, 2008). These authors have carried out a study to determine the most suitable integrated configuration for tri-generation system that uses biogas and micro gas turbines. By analyzing different configurations of micro gas turbine, absorption refrigeration system and waste heat boiler, the authors have discussed different performances in different configurations.

Other studies focused on the investigation of different methods for gas turbine inlet cooling. For example, the “Improvement of simple and regenerative gas turbine using simple and ejector-absorption refrigeration” (L. GAROOOCI FARSHI, S. M. Seyed Mahmoudi & A. H. Mosafa, 2008) is a research conducted on investigating the effect of inlet air cooling by absorption refrigeration.

Figure 3.1- Design characteristics with inlet air temperature of a generic gas turbine (OMIDVAR, Bob, 2001)

<table>
<thead>
<tr>
<th>MODEL</th>
<th>GT10B</th>
<th>Frame 5 (5371 )</th>
<th>PA</th>
<th>Frame 6 ( 6561 )</th>
<th>*LM 6000PC</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet Temp</td>
<td>C</td>
<td>15</td>
<td>35</td>
<td>40</td>
<td>15</td>
</tr>
<tr>
<td>GT Power</td>
<td>MW</td>
<td>23.1</td>
<td>20.1</td>
<td>18.9</td>
<td>25.2</td>
</tr>
<tr>
<td>GTG efficiency</td>
<td>%</td>
<td>33</td>
<td>32.11</td>
<td>31.8</td>
<td>28</td>
</tr>
<tr>
<td>Decrease in power w.r.t 15 C</td>
<td>%</td>
<td>0</td>
<td>13</td>
<td>18</td>
<td>0</td>
</tr>
<tr>
<td>Decrease in efficiency w.r.t 15 C</td>
<td>%</td>
<td>0</td>
<td>3</td>
<td>4</td>
<td>0</td>
</tr>
</tbody>
</table>
According to this study, gas turbine power output and efficiency were found to increase by 6-10% and 1-5% respectively for a decrease of 10°C of inlet air temperature.

Al-Tobi, I. (2009) reported that gas turbine performance changes with inlet cooling according to a study performed with simulation software called “Turbomatch” which is another approach for analyzing this type of machines.

However, Thamir et al. (2011) reported that the efficiency was decreasing slightly with the inlet cooling, but power output was increasing. It states “gas turbine intake air cooling may cause a small decrease in efficiency because a lot of fuel is needed to bring compressor exhaust gas equal to the same gas turbine entry temperature”. This statement is based on basic energy calculation done on simple cycle gas turbine cycle based on shear theoretical basis. Practical gas turbines efficiency behave differently in the case of inlet air cooling and a formula for practical gas turbine efficiency variation is given by (BOYCE, Meherwan P., 2002) in the following equation:

\[
\eta_{\text{cyce}} = \left[ \frac{\eta_f T_f - T_{amb} \frac{r_p (\gamma - 1)}{\eta_c}}{T_f - T_{amb} \frac{r_p (\gamma - 1)}{\eta_c}} \right] \left( \frac{1}{1 - \frac{1}{r_p (\gamma - 1)}} \right) \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (3.2)
\]

According to this equation, the efficiency of actual gas turbines increases steadily with the inlet air cooling, as has been revealed in many studies.

Raquel Gareta (2004) has reported a study on the economic evaluation of gas turbine air-cooling systems in combined cycle applications associated with the economic factor in deriving their suitability. The authors have used economic factors such as fuel cost, hourly electricity tariffs, equipment maintenance and investment costs and other relevant economic variables to maximize the profits of the integrated system. This is an interesting research because today’s world energy generating companies are keen on these topics due to energy market rivalry.

When analyzing the inlet air cooling of gas turbines it is also necessary to carefully review the available charge air cooling methods applicable to gas turbines worldwide.
3.2 Charge air cooling methods

Fogging system

Fogging system sprays high pressurized water by atomizing into supper fine droplets and increases the mass flow rate to the compressor. It absorbs evaporation heat from the air similar to the evaporative cooling. This system uses de-mineralized water and sprays directly to the compressor air inlet. Major disadvantages are that this could damage the compressor if the water droplets travel to the compressor and also this method is not feasible in locations where the air humidity already is high. Generally, fogging inlet cooling gives 0.9% efficiency increase per one degree Celsius. Fogging cooling is able to increase the gas turbine power output by 5% per every 1% of fog mass flow (PATEL, Pankaj K., 2003).

Figure 3.2 - Fogging system (Isratec engineering and investment Ltd)

Figure 3.3 - Fogging nozzle (MEE, Thomas)
Evaporative cooler

Evaporative cooling is similar to the fogging system, however, water is not directly injected to the compressor inlet. Instead, inlet air is allowed to travel through a shower of water and the latent heat of water evaporation absorbs energy from the inlet air. Excess water is drained away from the system. This is very low cost method of inlet air cooling. But, the performance is limited by the wet bulb temperature reading (RH value) of the site, that is again limited by the relative humidity of ambient air.

Mechanical refrigeration system (Direct type)

Refrigerant type chiller systems can be used directly to decrease the inlet air temperature using vapor compression cycle. The inlet air directly is passed through the evaporator. By this method it is possible to bring the inlet air to any desired value, for example the standard temperature of 15°C (PATEL, Pankaj K., 2003). But it decreases the inlet air pressure and adversely affects the overall output due to the comparatively large power consumption for cooling.

![Figure 3.4 – Mechanical refrigeration system (OMIDVAR, Bob, 2001)](image)

Mechanical refrigeration system (indirect type)

Indirect system uses chilled water as cooling media at the compressor inlet. The system comprises a separate chiller system for inlet cooling.
**Mechanical refrigeration system with ice storage**

This type is used in grid systems where the electricity cost vary with time of the day. Ice is produced using mechanical chillers during low cost times and the ice is applied to cool the inlet air at high cost times.

**Mechanical refrigeration system with chilled water storage**

This method is similar to ice storage system and is at the same time a variation of the indirect refrigeration approach. The heat is transferred away from the inlet air by chilled water, which in turn is produced during low-cost times and stored in a thermally isolated tank.

**LiBr absorption chiller**

Absorption refrigeration operates by using two fluids. Cooling occurs by evaporation similar to the vapor compression cycle. A separate chilled water system can also be used to transfer the heat. Absorption refrigeration is advantageous in cooling the inlet air because it does not require mechanical input for the operation, it requires heat input. In gas turbine operation in open cycle or combined cycle we can associate this heat input by waste heat recovery. Also lower temperatures can be achieved by absorption refrigeration compared to evaporative cooling.

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Figure 3.5 – Absorption refrigeration system (OMIDVAR, Bob, 2001)
Among these inlet cooling methods, this study is carried out focused on the LiBr absorption refrigeration for the following reasons:

1. Absorption refrigeration can achieve the cooling quickly compared to vapor compression system, and high cooling load can be gained from this method.
2. As Sri Lanka is a moderately humid country, more frequently there is a high relative humidity level. Hence it is not recommended using evaporative cooling or fogging systems in Sri Lanka.
3. The absorption cooling cycle requires mechanical input only for running small pumps. It can primarily be driven from the waste heat rejected in the form of gas turbine exhaust.

On the other hand, absorption cooling has certain drawbacks when compared to other cooling methods:

1. High installation cost initially and high operation and maintenance cost as the system is a complicated system.
2. Relatively complex system, requiring specialized training and expertise for operation and maintenance.
3. Relative sizes of absorption equipment are larger if compared with the gas turbine. For example, the dimensions of a system having a cooling load of 2000 kW are approximately 41ft x 16ft x 13.5ft (approx. 1.3m x 0.53m x 0.44m) without cooling towers.
4. Thermodynamic approach on the gas turbine and inlet cooling

Thermodynamic analysis in this chapter is a brief outline of the generalized gas turbine performance, efficiency and its characteristics. Air-standard Brayton cycle is assumed for the analysis of the thermodynamic processes.

4.1 Gas turbine performance and Brayton cycle

Gas turbines can be thermodynamically approximated by the Brayton cycle. The ideal cycle is incorporated by two isobaric processes and two isentropic processes.

![Figure 4.1 – Ideal Joule-Brayton cycle](image_url)

In Figure 4.1,
1-2 is adiabatic isentropic compression which occurs in the multistage compressor;
2-3 is adiabatic constant pressure heat addition which occurs in the combustion chamber;
3-4 is adiabatic isentropic expansion in the turbine/expander section;
4-1 is adiabatic constant pressure heat rejection.

Considering the Carnot efficiency of the gas turbine cycle assuming $\dot{m}_a \gg \dot{m}_f$, specific heat ($C_p$ and $C_v$) of the air and the gas is constant throughout the cycle and pressure ratio in compressor and turbine are the same.
Power input to the compressor

\[ P_c = \dot{m}_{\text{air}} (h_2 - h_1) \]  \hspace{1cm} (4.1)

Turbine power output

\[ P_t = \dot{m}_{\text{gas}} (h_3 - h_4) \]  \hspace{1cm} (4.2)

Net work output of the cycle (simple cycle)

\[ W_{\text{cyc}} = W_t - W_c \]  \hspace{1cm} (4.3)

Energy added to the system

\[ Q_{2,3} = m_f x LHV_{\text{fuel}} = (\dot{m}_a + \dot{m}_f) h_3 - \dot{m}_a h_2 \]  \hspace{1cm} (4.4)

Thus, the thermal efficiency of the cycle is

\[ \eta = \frac{W_{\text{cyc}}}{Q_{2,3}} \]  \hspace{1cm} (4.5)

Also, with the above mentioned assumptions, following expression gives the relationship of efficiency with pressure ratio (EASTOP, T.D., 2002):

\[ \eta = 1 - \left( \frac{1}{r_p} \right)^{\frac{\gamma - 1}{\gamma}} \]  \hspace{1cm} (4.6)

Considering the real gas turbine cycle if efficiency in the compressor is \( \eta_c \) and efficiency of the turbine (expander) is \( \eta_t \) then the relation for the cycle efficiency for the firing temperature \( T_f \) is (BOYCE, Meherwan P., 2002):

\[ \eta_{\text{cyc}} = \frac{\eta_t T_f - T_{\text{amb}} \left( \frac{r_p^{\frac{\gamma - 1}{\gamma}}}{\eta_c} \right)}{T_f - T_{\text{amb}} - T_{\text{amb}} \left( \frac{r_p^{\frac{\gamma - 1}{\gamma}}}{\eta_c} \right)} \left( 1 - \frac{1}{r_p^{\frac{\gamma - 1}{\gamma}}} \right) \]  \hspace{1cm} (4.7)

The most affecting factors for the gas turbine thermal efficiency are the compression ratio of the compressor and the turbine inlet temperature (Gas turbine firing temperature). Gas turbine thermal efficiency increases dramatically with the increase of compressor compression ratio. However, it starts to decrease after a certain value of the compression ratio for the given temperatures. Higher values of pressure ratios can decrease the operating range of the compressor and it will be much more vulnerable to failures that could be caused by surges, dust and sedimentations on the blades. Hence the values for the pressure ratio should be optimized according for the application used.
Figure 4.2 shows a simple cycle gas turbines efficiency and work output variation with the firing temperature. Figure 4.3 illustrates the simple cycle gas turbine performance mapping with the pressure ratio and turbine inlet temperature.

![Figure 4.2 - Performance map of simple cycle gas turbine (BOYCE, Meherwan P., 2002)](image)

Following equation (BOYCE, Meherwan P., 2002) gives the optimum pressure ratio for maximum thermal efficiency considering fixed turbine inlet temperature and turbine and compressor efficiencies:

\[
\text{Following equation (BOYCE, Meherwan P., 2002) gives the optimum pressure ratio for maximum thermal efficiency considering fixed turbine inlet temperature and turbine and compressor efficiencies:}
\]
\[(r_p)_{eopt} = \left\{ \frac{1}{T_1 \eta_t - T_3 \eta_t + T_1} \left[ T_1 T_3 \eta_t - \sqrt{(T_1 T_3 \eta_t)^2 - (T_1 T_3 \eta_t - T_1 T_3 + T_1^2)(T_3^2 \eta_c \eta_t - T_1 T_3 \eta_t \eta_c + T_1 T_3 \eta_t)} \right] \right\}^{\frac{\gamma}{\gamma - 1}} \] ... ... ... ... ... (4.8)

This optimum pressure ratio can be found for maximum work output with turbine section and compressor section efficiency (BOYCE, Meherwan P., 2002).

\[(r_p)_{pwopt} = \left[ \frac{T_3 \eta_t \eta_c}{2T_1} + \frac{1}{2} \right]^{\frac{\gamma}{\gamma - 1}} \] ... ... ... ... ... ... ... ... ... ... ... ... ... ... ... ... ... ... ... ... ... ... (4.9)

Analyzing the two equations for the two pressure ratios \((r_p)_{eopt}\) and \((r_p)_{pwopt}\) for the pressure ratio for maximum efficiency is greater than the pressure ratio for the maximum work output, for the same values of firing temperature and turbine and compressor efficiencies.

Figure 4.4 below presents the normalized specific work output for the air-standard Brayton cycle as a function of compressor pressure ratio and of temperature increase ratio within the cycle. The influence of compressor air inlet temperature (T1) on the cycle work output can also be derived from the diagram.
For each T3/T1 value, there is an optimum pressure ratio where specific work is maximized. When T1 (Inlet temperature) decreases GT work output increases.
4.2 Cooling load estimation for gas turbine inlet cooling

To analyze the GT inlet cooling with absorption refrigeration, the cooling load is to be calculated by referring to the actual online data taken from the Kelanitissa gas turbine unit No.05.

Sensible cooling load can be calculated using the following equation (B. DAWOUD, Y.H. Zurigat, J. Bortmany, 2005):

\[
\dot{Q}_s = \frac{\dot{v}_a}{v_a} C_{p,a} (t_{aa,db} - t_{ci}) \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (4.10)
\]

\(\dot{Q}_s\) is sensible cooling load,

\(\dot{v}_a\) is volume flow rate of air at the ISO conditions. This will be based on actual data acquired from the site.

\(t_{aa,db}\) is dry bulb temperature of the ambient air and \(t_{ci}\) is compressor inlet temperature which should be cooled down to a desired value. This should be the design temperature or the ISO condition temperature or the temperature which will be considered for the cooling of inlet air.

\(v_a\) is specific volume of the wetted air per kilogram of dry air which varies with the relative humidity of the air. It can be calculated as follows (B. DAWOUD, Y.H. Zurigat, J. Bortmany, 2005):

\[
v_a = (0.287 + X_{aa,db} \cdot 0.462) \cdot \frac{T}{P_{atm}} \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (4.11)
\]

Temperature \(T\) is in Kelvin and \(P_{atm}\) is in kPa. \(X_{aa,db}\) is specific humidity of the ambient air at its dry bulb temperature (B. DAWOUD, Y.H. Zurigat, J. Bortmany, 2005).

\[
X_{aa,db} = \frac{P_s}{P_{atm}} - \frac{M_v}{M_a} \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots \ldots (4.12)
\]

\(M_v\) is molecular weight of the water vapor and \(M_a\) is molecular weight of the air.

\(P_s\) is the saturation vapor pressure at inlet air dry bulb temperature. This can be read from a chart or approximated from the following equation (B. DAWOUD, Y.H. Zurigat, J. Bortmany, 2005):
\[ P_s = 22064. \exp \left\{ -7.76451 \left( \frac{1-T}{647.14} \right) + 1.45838 \left( \frac{1-T}{647.14} \right)^{1.5} - 2.7758 \left( \frac{1-T}{647.14} \right)^3 \right\} \\
- 1.23303 \left( \frac{1-T}{647.14} \right)^6 \\
/ \left( \frac{T}{647.14} \right) \] … … … … … … … … … … … … … … … … … … … … … … … … . . (4.13)

If the designed inlet temperature is lower than the dew point, the Latent cooling load can be calculated as follows (B. DAWOUD, Y.H. Zurigat, J. Bortmany, 2005):

\[ \dot{Q}_l = \frac{\dot{V}_a}{v_a} \left\{ x_{aa,db} (C_{p,v} t_{aa,db} + r) - x_{c,i}^s (C_{p,v} t_{c,i} + r) \right. \]
\[ \left. - (x_{aa,db} - x_{c,i}^s) C_{p,w} t_{c,i} \right\} \ldots \ldots (4.14) \]

\( C_{p,v} \) is specific heat at constant pressure of the water vapor. \( x_{c,i}^s \) is specific humidity at the compressor inlet temperature and \( C_{p,w} \) is specific heat of liquid water at constant pressure.

### 4.3 Thermodynamic model analysis for Power output and Efficiency

Behavior of cycle power output and efficiency with varying ambient temperature can be thermodynamically modeled by developing the equations presented in the sections above.

Power input to the compressor:

\[ P_c = \dot{m}_{air} \cdot (h_2' - h_3) \]

\( \eta_c = 0.95 \) "Compressor efficiency"

\( \eta_t = 0.95 \) "turbine efficiency"

\( \dot{m}_{air} = 97 \) "air mass flow rate in kg/s"

Values of \( h_4 \) and \( h_2' \) are changing with the ambient temperature.

Turbine power output:

\[ P_t = \dot{m}_{gas} \cdot (h_3 - h_4') \]

Total work output of the cycle (simple cycle):

\[ W_{cy} = W_t - W_c \]

Due to the change of \( W_c \) with the ambient temperature, \( W_{cy} \) (cycle work output) is also changing with the ambient temperature. Hence the program is run in the parametric input mode which can enter inputs values of ambient temperature and get the respective values in output table – see Table 4.1 further below.
Efficiency for the actual cycle:

\[
\eta_{cyc} = \frac{\eta_i T_f - T_{amb} \frac{r_p (\gamma - 1)}{\eta_c}}{T_f - T_{amb} - T_{amb} \left( \frac{r_p (\gamma - 1)}{\eta_c} \right) \left( 1 - \frac{1}{r_p (\gamma - 1)} \right)}
\]

\( T_f = 1172 K \) Firing temperature

\( r_p = 10 \) "Pressure ratio"

\( \gamma = 1.33 \) "Specific heat ratio"

\( m_{air} \) is the air flow rate of the compressor and \( m_{gas} \) is the gas flow rate in the turbine. Air flow rate is taken as 97 kg/s (design data) and pressure ratio is taken as 10 (design data).

EES model was developed using above equation to analyze behavior of the real cycle with the ambient temperature. Since \( \eta_{cyc} \) is changing with ambient temperature parametric input mode is used in EES.

\textit{EES codes are given in Appendix I}

![EES diagram of the modeled gas turbine cycle](image)

Figure 4.5 - EES diagram of the modeled gas turbine cycle
Following results were obtained by running EES in parametric input mode:

<table>
<thead>
<tr>
<th>Ambient temperature (°C)</th>
<th>Power output (kW)</th>
<th>Efficiency (η&lt;sub&gt;cyc&lt;/sub&gt;)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15</td>
<td>27103</td>
<td>0.4401</td>
</tr>
<tr>
<td>20</td>
<td>26526</td>
<td>0.4386</td>
</tr>
<tr>
<td>25</td>
<td>25949</td>
<td>0.4372</td>
</tr>
<tr>
<td>30</td>
<td>25372</td>
<td>0.4356</td>
</tr>
<tr>
<td>35</td>
<td>24795</td>
<td>0.434</td>
</tr>
<tr>
<td>40</td>
<td>24219</td>
<td>0.4324</td>
</tr>
<tr>
<td>45</td>
<td>23645</td>
<td>0.4307</td>
</tr>
<tr>
<td>50</td>
<td>23071</td>
<td>0.4289</td>
</tr>
<tr>
<td>55</td>
<td>22499</td>
<td>0.427</td>
</tr>
</tbody>
</table>

Table 4.1 – Power output and efficiency with ambient temperature

Figure 4.6 and 4.7 show graphically the variation of power output and efficiency respectively with ambient air temperature for the modeled generic gas turbine cycle.

![Figure 4.6 - Power output variation with ambient temperature](image-url)
The graph of power output is almost linear with inlet temperature, but in fact it is curved with a slight slope which is decreasing towards the increasing temperature. Average reduction in power output per degree Celsius is about 115 kW. Cooling of inlet air from 27°C down to 15°C would increase the power output by 1.38 MW.

Figure 4.7 – Efficiency variation with ambient temperature

The efficiency also seems almost linear, but the slope is slightly increasing with the increase of temperature. Average change in efficiency for the graph range is about 0.033%/°C.
5. Analysis of data and results

This chapter presents the actual performance of the gas turbine in terms of efficiency, heat rate, active power which were determined with the help of available data.

5.1 Variation of active power with ambient temperature

In Kelanitissa power plant the different machine loads and the ambient air temperature are recorded on a daily basis when the turbines are in operation.

These machines normally run to accommodate peak power requirement and also run with synchronous condenser mode to supply the VAR requirement in the grid. Table 5.1 gives a list of data recorded in August 2011. Only the days which required full load active power are presented.

<table>
<thead>
<tr>
<th>Date</th>
<th>Time</th>
<th>Load (MW)</th>
<th>Temp (C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1/8/2011</td>
<td>10:00</td>
<td>16.5</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>12:00</td>
<td>16.5</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>14:00</td>
<td>16.5</td>
<td>31</td>
</tr>
<tr>
<td></td>
<td>16:00</td>
<td>16.5</td>
<td>31</td>
</tr>
<tr>
<td></td>
<td>18:00</td>
<td>16.8</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>20:00</td>
<td>17</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>22:00</td>
<td>17</td>
<td>29</td>
</tr>
<tr>
<td>2/8/2011</td>
<td>10:00</td>
<td>16.8</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>12:00</td>
<td>16.8</td>
<td>31</td>
</tr>
<tr>
<td></td>
<td>14:00</td>
<td>16.8</td>
<td>31</td>
</tr>
<tr>
<td></td>
<td>16:00</td>
<td>17</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>18:00</td>
<td>17</td>
<td>29.5</td>
</tr>
<tr>
<td></td>
<td>20:00</td>
<td>17</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>22:00</td>
<td>17</td>
<td>29</td>
</tr>
<tr>
<td>3/8/2011</td>
<td>10:00</td>
<td>17</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>12:00</td>
<td>16.5</td>
<td>31</td>
</tr>
<tr>
<td></td>
<td>14:00</td>
<td>16.5</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>16:00</td>
<td>16.5</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>18:00</td>
<td>17</td>
<td>28</td>
</tr>
<tr>
<td></td>
<td>20:00</td>
<td>17.5</td>
<td>26</td>
</tr>
<tr>
<td></td>
<td>22:00</td>
<td>17.5</td>
<td>26</td>
</tr>
<tr>
<td>23/08/2011</td>
<td>8:00</td>
<td>17</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>10:00</td>
<td>17</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>12:00</td>
<td>16</td>
<td>31</td>
</tr>
<tr>
<td></td>
<td>14:00</td>
<td>16</td>
<td>31</td>
</tr>
<tr>
<td></td>
<td>16:00</td>
<td>16</td>
<td>31</td>
</tr>
<tr>
<td></td>
<td>18:00</td>
<td>16.5</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>20:00</td>
<td>16.5</td>
<td>29</td>
</tr>
<tr>
<td>Time of the day</td>
<td>Temp(C)</td>
<td>Load(MW)</td>
<td></td>
</tr>
<tr>
<td>----------------</td>
<td>---------</td>
<td>----------</td>
<td></td>
</tr>
<tr>
<td>10:00</td>
<td>17</td>
<td>29</td>
<td></td>
</tr>
<tr>
<td>12:00</td>
<td>16.5</td>
<td>30</td>
<td></td>
</tr>
<tr>
<td>14:00</td>
<td>16.5</td>
<td>31</td>
<td></td>
</tr>
<tr>
<td>16:00</td>
<td>16.5</td>
<td>31</td>
<td></td>
</tr>
<tr>
<td>18:00</td>
<td>16.5</td>
<td>29</td>
<td></td>
</tr>
<tr>
<td>20:00</td>
<td>17</td>
<td>29</td>
<td></td>
</tr>
<tr>
<td>22:00</td>
<td>17</td>
<td>29</td>
<td></td>
</tr>
</tbody>
</table>

Table 5.1 – Turbine load data with time and ambient temperature

![Graph](image)

Figure 5.1 – Load variation of Unit 05 with ambient temperature
Figure 5.1 is a plot according to the data from Table 5.1, representing active power output and ambient temperature (in the same axis) variation with the time of the day during seven days as of Table 5.1.

As seen in Figure 5.1, above there are dips in the total active power output at the times when the ambient temperature is increasing.

![Figure 5.1](image)

Figure 5.2 shows a plot of active power vs. ambient temperature. According to this plot at some temperatures different readings of active power have been recorded. It can be seen in Table 5.1 that same ambient temperatures have different active power readings. To get a better interpretation of the data, values are averaged and plotted in Figure 5.3 below.

Figure 5.3 also shows the line of active power variation of turbine Unit 05 with the air temperature by simple linear regressions.
Figure 5.3 – Average active power with temperature

To predict the power output of the GT Unit 05 at designed temperature based on the past log, data can be linearized using simple linear regression. The data pattern can be observed linear in the range that has a higher number of data points where ambient temperature is 29, 30 and 31 degree Celsius. Here it has been assumed that in the period when these readings were taken the ambient air had a constant relative humidity and constant.

The data from Table 5.1 above can be linearized using the following equation for simple linear regression. If the equation is linear there exists:

\[ Y = \beta X + \alpha \]

Such that,

\[ \beta = \frac{\sum_{i=1}^{n} (x_i - \bar{x})(y_i - \bar{y})}{\sum_{i=1}^{n} (x_i - \bar{x})^2} \]

\[ \beta = \frac{\bar{x} \bar{y} - \bar{y} \bar{x}}{\bar{x}^2 - \bar{x}^2} \]

\[ \beta = \frac{486.0558 - 28.5 \times 17.0825}{815.1667 - 28.5^2} \]

\[ \beta = -0.2727 \]

And
\[
\alpha = \bar{Y} - \beta \bar{X}
\]
\[
\alpha = 17.0825 - (-0.2727) \times 28.5
\]
\[
\alpha = 24.85
\]

Hence the linearized equation is,
\[
Y = -0.2727X + 24.85
\]

Machine active power is interpreted by Y and ambient temperature is interpreted by X.

Then to get active power at 15°C, X is substituted by 15:
\[
Y = -0.2727(15) + 24.85
\]
\[
Y = 20.7595
\]

Hence the expected load at ambient temperature 15°C is 20.76 MW.

### 5.2 Heat Rate test

Heat rate is a commonly used term for expressing the energy efficiency of the heat engines. Heat rate is the fuel energy needed to generate a unit of electrical energy.

\[
Heat rate = \frac{Total \ energy \ of \ consumed \ fuel \ (kJ/kg)}{Generated \ electrical \ energy \ (kWh)}
\]

A heat rate test had been carried-out for the Kelanitissa Power Station gas turbines on 21/05/2012 (see Table 5.2 below). It has been done by the Public Utilities Commission of Sri Lanka, the organization which is responsible for energy utilities in Sri Lanka.
<table>
<thead>
<tr>
<th>Fuel Flow Meter Reading (Liters)</th>
<th>Generated Energy Meter Reading (MWh)</th>
<th>Auxiliary Energy Meter Reading (kWh)</th>
<th>Aux. Energy Meter Reading / Fin Fans (KWh)</th>
<th>Compressor Discharge Temp: °C</th>
<th>Exhaust Average Temp: °C</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Initial Readings</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Startup</td>
<td>32521106.00</td>
<td>682502</td>
<td>1072938</td>
<td>840691</td>
<td>-</td>
</tr>
<tr>
<td>Synchronizing</td>
<td>32521269.00</td>
<td>682502</td>
<td>1072942</td>
<td>840693</td>
<td>-</td>
</tr>
<tr>
<td>05 Minutes after Full Load</td>
<td>32521943.00</td>
<td>682503.5</td>
<td>1072943</td>
<td>840694</td>
<td>-</td>
</tr>
<tr>
<td>Running full load (17 MW) for 1 hour</td>
<td>32529735.00</td>
<td>682520.5</td>
<td>1072954</td>
<td>840712</td>
<td>328</td>
</tr>
<tr>
<td><strong>Reloading</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Running Part Load (15 MW) for 1/2 hour</td>
<td>32533308.00</td>
<td>682528.3</td>
<td>1072959</td>
<td>840721</td>
<td>326</td>
</tr>
<tr>
<td>Running Part Load (12 MW) for 1/2 hour</td>
<td>32536435.00</td>
<td>682534.4</td>
<td>1072964</td>
<td>840729</td>
<td>321</td>
</tr>
<tr>
<td>Running Part Load (10 MW) for 1/2 hour</td>
<td>32539286.00</td>
<td>682539.5</td>
<td>1072970</td>
<td>840738</td>
<td>315</td>
</tr>
<tr>
<td><strong>Shutdown</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>After Shutdown</td>
<td>32540104.00</td>
<td>682540.8</td>
<td>1072974</td>
<td>840741</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 5.2 – Readings of the heat rate test

Readings in Table 5.2 were acquired at 33°C and 1009 Mbar of ambient pressure. For the calculations, the calorific value of the diesel fuel is taken as 43961.4 kJ/kg.

<table>
<thead>
<tr>
<th>Load Case</th>
<th>Heat rate (kJ/kWh)</th>
<th>Efficiency (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Running full load (17 MW) for 1 hour</td>
<td>16925.85</td>
<td>21.26924</td>
</tr>
<tr>
<td>Running Part Load (15 MW) for 1/2 hour</td>
<td>16915.68</td>
<td>21.28203</td>
</tr>
<tr>
<td>Running Part Load (12 MW) for 1/2 hour</td>
<td>18929.90</td>
<td>19.01753</td>
</tr>
<tr>
<td>Running Part Load (10 MW) for 1/2 hour</td>
<td>20643.23</td>
<td>17.43913</td>
</tr>
</tbody>
</table>

Table 5.3 – Results of the heat rate test

GE MS5001 R gas turbine’s designed heat rate is 13,980 kJ/kWh. Comparing the values in Table 5.3 with the designed heat rate (13,980 kJ/kWh) value shows very poor performance of the actual GT unit 05. Efficiency values calculated for loads 17MW, 15MW, 12MW, 10MW were respectively 21.26%, 21.28%, 19.01%, 17.43% while the originally designed efficiency is 25.8%, confer with Figure 5.4 below.
It can be observed that the efficiency drops steadily with deeper part load conditions and is saturated above 15MW load. Maximum efficiency achieved at 33°C was 21.2% for the existing turbine. This type of turbine would give maximum efficiency of 26.4% (Table 3.1- Frame5) when operated at 15°C. According to the available data when the same GT is operated at 35°C, some 5.82% reduction in efficiency could be expected. However, according to this study the efficiency reduction is 4.6% (25.8%-21.2%) when operated at 33°C.

5.3 Actual cooling load estimation assuming inlet temperature 27°C

Cooling load can be estimated according to the procedure in section 4.2. For this estimation the average gas turbine inlet temperature is considered as 27°C and it is cooled down to 15°C which is ISO temperature. Gas flow rate of the gas turbine is taken as 81.63 m³/s at the rated load. Cooling load has been calculated using the EES software for convenience. Following are the steps of calculations.

**Sensible cooling load**

\[ X_{aa,db} = \frac{P_s}{\bar{\rho}} - \frac{P_s}{P_{atm}} \cdot \frac{M_v}{M_a} \]  \hspace{1cm} (4.12)

**Substituting values**

\[ M_v = 18, M_a = 28.9, P_s = 3.6, P_{atm} = 102.6675, \bar{\rho} = 0.70 \]
\[ X_{aa,db} = 0.01567 \]

\[ v_a = (0.287 + X_{aa,db} \cdot 0.462) \cdot \frac{T}{P_{atm}} \] ................................. (4.11)

substituting \( T = 300 \)

\[ v_a = 0.8598 \]

\[ \dot{Q}_s = \frac{\dot{V}_a}{v_a} C_{p,a} (t_{aa,db} - t_{c,i}) \] ................................. (4.10)

\[ \dot{v}_a = 81.63, \ t_{c,i} = 15, \ C_{p,a} = 1.004 \]

\[ \dot{Q}_s = 1143 \text{ kW} \]

Latent Cooling load

\[ \dot{Q}_l = \frac{\dot{V}_a}{v_a} \left\{ x_{aa,db} (C_{p,v} \cdot t_{aa,db} + r) - x_{c,i}^s \cdot (C_{p,v} \cdot t_{c,i} + r) - (x_{aa,db} - x_{c,i}^s) C_{p,w} \cdot t_{c,i} \right\} \] ........ (4.14)

Substituting values

\[ C_{p,v} = 1.866, \ t_{aa,db} = 27, \ r = 2500.8, \ x_{c,i}^s = 0.01049, \ C_{p,w} = 4.2 \]

\[ \dot{Q}_l = 1247 \text{ kW} \]

Total cooling load \[ Q = \dot{Q}_l + \dot{Q}_s = 2391 \text{ kW} \]

EES Codes are attached in appendix II

According to the calculations, the total cooling load is \( Q = 2391 \text{ kW} \) (679.87 RT). This required energy is freely available in the hot exhaust gas behind the turbine outlet. In the present market there are few manufacturers who supply absorption chiller systems in commercial scale. Some of them are:

- Broad USA
- Thermax Ldt.
- Trane Inc.
- York Intl.

These manufactures have compact designs to meet customers’ requirements which include chilled water circulation system, cooling water circulation system with cooling towers, etc.

Broad USA is a company that supplies absorption chiller systems up to 3400 RT of cooling load, driven by direct firing, steam or exhaust waste heat.
5.4 Economic Aspects of the project

Main focus of this section is to discuss about the economic feasibility of the project. Financial calculations are based on the projected performance discussed in the previous chapter.

Expected performance of the machines by inlet air cooling

If the gas turbine is fully served by inlet air cooling, the maximum power output expected is 20.76 MW. The efficiency improvement achieved by cooling the inlet from 27°C (average daily temperature) down to 15°C is 0.4%-points, as per the simulation results.

Economic feasibility analysis of the project

The project can be analyzed for economic feasibility by calculating the present value for the projected cost of the chiller system. It can be assumed that the equipment operates for about fifteen to twenty years after installation.

\[
PV = \frac{a}{r} \left[1 - \frac{1}{(1 + r)^n}\right]
\]

Where “a” is profit per year; “r” is discounted rate; and “n” is number of years.

From the generation data acquired from the Kelanitissa Power station, annual generation of Unit 05 is 31000 MWh. Discounted rate is taken as 6.1% for 2012 (SRI LANKA, Government of). Assuming that the above expected performance can actually be achieved by the project in reality, the annual return on investment would be:

- Expected efficiency increase = 0.4%
- Price of one liter of diesel = Rs.121.00
- Amount of diesel burnt per year = 1.229 × 10^7 L
- Amount of fuel that can be saved = 1.229 × 10^7 L × 0.004 × Rs.121
- Annual return on investment “a” = Rs.5.95Mn

From the above equation:

- PV for 15 years = Rs. 57.4Mn
- PV for 16 years = Rs. 59.7Mn
- PV for 17 years = Rs. 61.88Mn
- PV for 18 years = Rs. 63.93Mn
- PV for 19 years = Rs. 65.86 Mn
- PV for next 20 years = Rs. 67.68Mn
**Project cost estimation**

Absorption refrigeration systems are comparatively costly compared to other types of refrigerating equipment. Figure 5.5 and the tabulated data contained there present a comparison of installed cost for several types and sizes of absorption refrigeration systems.

![Absorption & Engine Drive Chiller, Cooling Tower, Cond. Water Pump & Piping](image)

**Figure 5.5 – Cost of absorption refrigerating systems (APOGEE, Interactive Inc., 2013)**

Estimated cooling load for the gas turbine inlet air cooling is 679.87 RT. From the data in Fig.5.5 above the plausible cost for a 2-stage absorption refrigeration system driven by GT exhaust heat can roughly be estimated. The cost per unit cooling load of the 2-stage direct fired absorption system is $751-721 (according to 600RT-700RT). For the worst case scenario the value $751 per RT can be taken, while the equipment supplying heating from waste exhaust gas would be about 98% of the cost of a direct fired system (Broad USA inc., 2008). Thus an overall value of $736/RT can be assumed as the specific cost for a complete exhaust driven absorption chiller system. Total cost for the needed 679.87 RT system would therefore be $ 500,370.72 (Rs. 65Mn).

Payback period of the project is 11 years but the present value after 19 years is exceeding the project cost. Present value for 19 years of operation is Rs. 65.86Mn.
5.5 System Design

This section focuses on some overall aspects of system design, in particular the extraction of energy from the GT exhaust gas, the practical use of commercial absorption refrigeration components, and the air cooling heat exchanger at the compressor inlet. The main steps of the design procedure are:

1. Selecting suitable products and manufacturers for an absorption chiller driven by exhaust gas;
2. Design of a heat exchanger for cooling the air at the inlet of the compressor;
3. Design of a hot gas inlet to the absorption refrigeration system.

Exhaust flue gas conditions of the Kelanitissa GT

GT exhaust mass flow rate is about 98.2 kg/s and the rated exhaust temperature is 513 °C. Volume flow rate of the exhaust gas is 167.2 m³.

Total heat content (also represented by the theoretical exergy potential) of the exhaust gas waste heat flow can be calculated by the enthalpy drop down to ambient temperature:

\[ Q = m_{gas} \cdot (h_4 - h_1) \]

\[ Q = 98.2 \times (600.1 - 300.6) \]

\[ Q = 29,410.9 \text{ kW} \]

Selection of the Absorption refrigerating system

There are several established companies in the world which manufacture and supply absorption refrigeration systems; they have already been presented in chapter 4.3 above. The further discussion here is focusing on the products from the manufacturer Broad Inc. from USA. Among the product range of Broad Inc., a suitable system can be selected and applied for GT inlet cooling.

Maximum cooling load required to bring the ambient temperature from an average of 27°C and typical air humidity for the region, down to 15°C, is 2391 kW (679.87 RT). Necessary heating power input to the chiller disregarding heat losses and system efficiency is therefore 2391 kW (or somewhat above that value in practice), while the available energy rate in the GT exhaust gas flow is a massive 29,410.9 kW hence completely able to satisfy the requirement as per the energy balance. However, the absorption chiller system requires the heating agent to have temperature levels according to manufacturer’s specifications. To cover a cooling load of 2391 kW, a two-stage absorption refrigeration system should be utilized. The BYE250X absorption system by Broad Inc. is the chiller that can achieve the cooling capacity required for the GT inlet cooling project, confer with Table 5.4.
## Package Hot W/Exhaust chiller Performance Data

BYH/BYE: hot water/exhaust from power generation or industrial waste streams (pumpset, enclosure data are the same as steam chiller)

### Table 5.4 – Product catalogue (Broad USA inc., 2008)

<table>
<thead>
<tr>
<th>Model</th>
<th>Cooling Capacity</th>
<th>Heating Capacity</th>
<th>Chilled Water</th>
<th>Cooling Water</th>
<th>Heating Water</th>
<th>Hot Water</th>
<th>Exhaust Consumption</th>
<th>Cooling Heating</th>
<th>Power Solution Demand</th>
<th>Unit Main Main Shell Operation</th>
<th>Ship, Operation Weight</th>
<th>Ship, Weight</th>
</tr>
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<tbody>
<tr>
<td></td>
<td>kW</td>
<td>kW</td>
<td>GPM</td>
<td>fth₂O</td>
<td>GPM</td>
<td>fth₂O</td>
<td>GPM</td>
<td>fth₂O</td>
<td>lb/h</td>
<td>lb/h</td>
<td>kW</td>
<td>kibbs</td>
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<tr>
<td>Two-stage hot</td>
<td>20</td>
<td>66</td>
<td>233</td>
<td>/</td>
<td>/</td>
<td>/</td>
<td>/</td>
<td>/</td>
<td>1.7</td>
<td>2.2</td>
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<td>/</td>
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<td>/</td>
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<td>/</td>
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<td>/</td>
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<tr>
<td>300</td>
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<td>1886</td>
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<td>/</td>
<td>673</td>
<td>765</td>
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<td>5815</td>
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<td>3143</td>
<td>5196</td>
<td>16.7</td>
<td>/</td>
<td>/</td>
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<td>1281</td>
<td>215</td>
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<td>3771</td>
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<td>16.7</td>
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<td>8314</td>
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<td>/</td>
<td>/</td>
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<td>2020</td>
<td>301</td>
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<tr>
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<td>/</td>
<td>6286</td>
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<td>/</td>
<td>2244</td>
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<td>357</td>
<td>47.8</td>
</tr>
</tbody>
</table>

Table 5.4 – Product catalogue (Broad USA inc., 2008)
Table 5.5 contains performance data for the BYE250X system design, extracted and adapted from the product catalogue offered by Broad Inc.

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Cooling capacity</strong></td>
<td>2908 kW</td>
</tr>
<tr>
<td><strong>Chilled water</strong></td>
<td></td>
</tr>
<tr>
<td>Flow rate</td>
<td>99.1 L/s</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>49.9 kPa</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>6.7 °C</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>13.7 °C</td>
</tr>
<tr>
<td><strong>Exhaust gas</strong></td>
<td></td>
</tr>
<tr>
<td>Mass flow rate</td>
<td>5.34 kg/s</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>500 °C</td>
</tr>
<tr>
<td>Outlet temperature</td>
<td>160 °C</td>
</tr>
</tbody>
</table>

Table 5.5 – Chiller performance data

**Components and layout of the system**

The major components of the system are: 1) Heat exchanger inside the gas turbine compressor air inlet, situated after the air filters; 2) Absorption refrigeration equipment; 3) Piping and inventory for supplying the GT exhaust gas as a driving energy source to the absorption chiller, including isolating valves and a blower fan; 4) Control and monitoring interface. Figure 6.1 below shows a general layout of the overall system.

For the extraction of hot gas from the exhaust duct a blower fan is used to stabilize and maintain a constant flow. The heat exchanger for inlet air cooling is to be mounted at the inlet plenum. It is essential to mount the cooler after the inlet filter house to avoid clogging of the heat exchanger and to minimize the increase in pressure drop. The inlet pressure drop caused by the heat exchanger should be minimum because the GT performance is very sensitive to the inlet pressure loss.

The blower used to extract exhaust gas should be able to handle high temperatures in the order of 500 °C. It should also allow for regulating the flow and hence to control the chiller output by variations in the exhaust gas flow supplied to the absorption system. Chilled water inlet pipe has to be thermally insulated by cladding to eliminate heat losses.
Figure 5.6 – Cumulative layout diagram of the absorption chiller system for gas turbine inlet air cooling
6. Concluding Remarks

6.1 Power Output

Efficiency and power output are the key indexes of measuring gas turbine performance and its variation with the fluctuations of inlet air temperature. Normally, gas turbine power output and efficiency increase when the inlet air temperature is decreasing.

The power required to drive the compressor decreases with the decrease of ambient temperature. In the T-S diagram it can be seen that the gap between two constant pressure lines reduces with the decreasing temperature. Hence the power load in the compressor decreases with the decrease of ambient temperature. For the Kelanitissa gas turbine, the actual power output increase is about 272.7 kW per 1°C reduction of air temperature. This is about 22.3% increase. The corresponding theoretical value for a generic gas turbine is 115.1 kW per 1°C reduction. It should be noted that, when calculating the theoretical power output in this study the possible variation of heat losses was neglected, considering them as constant with the change of inlet air temperature, hence it would not affect the final result in the comparison of performance.

![Figure 6.1 – Actual power and theoretical power variation with air temperature](image)

Figure 6.1 shows the comparison between actual power output variation (for the Kelanitissa gas turbine) and the theoretical variation for a generic gas turbine with ambient air temperature. Theoretical line has been shifted for better understanding, normalizing it to 31°C as a common start point reference.
It can be clearly noticed that the actual power responds to a temperature change more than the theoretically derived power variation. This difference could be due to the increase of mechanical losses with the increase of temperature. Most probably it could be the heat loss which is higher when the temperature is higher, causing the observed difference.

6.2 Efficiency

Theoretical efficiency was calculated using the equation 4.7. It is the equation for efficiency of the real Brayton cycle. According to the equation 4.7 ambient temperature, pressure ratio, turbine inlet temperature and ratio of specific heats are the factors affecting the efficiency. In theoretical efficiency calculation it is assumed that only the ambient temperature is varying with the charge air cooling and all the other factors are constant. From the heat rate test results presented in chapter 4.2 it is derived that the efficiency at 33°C is 21.26%. Hence we can shift the graph for efficiency to fit the actual interpretation of the efficiency variation with the ambient temperature as shown in Figure 6.2.

![Figure 6.2 – Efficiency variation (shifted) with inlet air temperature](image)

Fig. 6.2 is thus presenting the expected efficiency profile of the Kelanitissa gas turbine with the variation of inlet air (charge air) temperature range from 15°C to 55°C. The average efficiency increase is about 0.033%/°C. This average efficiency value is specific to this gas turbine model, and will be fluctuating with the actual change of inlet temperature and pressure ratio.
According to above Figures 6.1 and 6.2, the Kelanitissa gas turbine can reach up to 20.76 MW power output and 21.86% efficiency maximum at 15°C of inlet air temperature, which is the design temperature. The design performance indexes according to the manufacturer are 25.8% of efficiency and 22.4MW power output. This major difference between the original performance and the present performance is due to aging of the machine after nearly 30 years of operation.

6.3 Feasibility

According to chapter 5, 27°C is the average air temperature in Colombo year-round, and bringing it down to ISO value of 15°C gives Rs.5.9Mn of annual savings for the Kelanitissa gas turbine Unit 05 at the established operational hours. This saving is only from the decreased amount of fuel due to efficiency improvement. A value or measurement for economic return due to the increase of active power output is not considered in the financial benefits of the project. The annual power generation by the Kelanitissa plant can be increased by more than 20%. If the inlet cooling is to be done by absorption refrigeration, the capital cost for a 600RT-700RT system would be about Rs.65Mn ($500,370).

Direct payback for this system only by fuel savings is about eleven years (assuming constant fuel prices as per 2013), but the benefit from power increase should also be considered in the actual assessment of the practical feasibility if the additionally produced power can be sold to the grid at the same conditions.

6.4 Future developments

Further studies can be done on this project to increase the accuracy of feasibility assessment and to enhance the economic benefits. One option is to cool the inlet air for all five gas turbines installed at the Kelanitissa power station by using exhaust gas from one gas turbine. This method can keep cost down while achieving higher economic feasibility. It will require a chiller system with larger cooling capacity which should be enough to cool the air for all the five gas turbines in the power station. The exhaust gas from a single turbine does have the energy content to deliver such large cooling load, however, that particular machine should be kept operating with priority in order to supply precooled air to all other operating turbines at any particular moment. Again, the overall increase of power production for the entire plant should also be valorized for the financial benefits according to the actual operational hours for each of the available gas turbines in the plant.
If the Kelaniwatta power plant utilized inlet cooling from the beginning of its commercial life, the past 30 years of operation would have saved a lot of money and energy, approximately at a present value of Rs.81Mn. Waste heat can be used in many ways to optimize the use of energy; extending the proposed chiller plant into a combined district cooling and/or heating unit which could provide district cooling energy for commercial users in Colombo. A proper estimation of the capital cost for such a project, however, is outside the scope of this study.

Open cycle gas turbine operation is normally omitted in modern power generating systems. Still, there are many cases around the world where old or new gas turbines are utilized in open-cycle mode for back-up or peak loads, or for various industrial drives in hot climates which cumulatively present an enormous potential for efficiency improvement and fuel savings, with the consequent benefit of an immediate decrease of the environmental footprint for these installations. District cooling/heating systems are not an option in some locations, while the utilization of exhaust gas heat for inlet air cooling could generally be applied anywhere in hot climate zones. Deliverables of this study may be able to help optimizing the operation and the environmental performance of such existing open cycle gas turbine plants.
References


DAIBER, Paul C. Performance and Reliability Improvement for the MS5001 Gas Turbines. Atlanta: GE Power systems.


PATEL, Pankaj K. 2003. BETTER POWER GENERATION FROM GAS TURBINE. ATLANTA: ASME.


Appendix I

EES Equations for Power Output and Efficiency Calculation

\[ p_0 = 1.04166 \text{"Inlet pressure in bar"} \]
\[ p_1 = 17.7 \text{"compressor discharge pressure"} \]
\[ E_c = 0.95 \text{"Compressor efficiency"} \]
\[ E_t = 0.95 \text{"turbine efficiency"} \]
\[ m_a = 97 \text{"air mass flow rate kg/s"} \]

\[ P_c = m_a(h_2-h_1)E_c \text{"compressor power kw"} \]
\[ h_2 = \text{enthalpy(Air, p=p_1, S=s_1)"specific enthalpy in kJ/kg"} \]
\[ h_{21} = h_1 + (h_2-h_1)/E_c \]
\[ s_1 = \text{entropy(Air, T=t_0, p=p_0)} \]
\[ s_{21} = \text{entropy(Air, h=h_{21}, p=p_1)} \]
\[ h_1 = \text{enthalpy(Air, T=t_0)} \]
\[ P_t = m_{gas}(h_3-h_{41})E_t \text{"turbine power"} \]
\[ h_3 = \text{enthalpy(Air, T=899)} \]
\[ s_3 = \text{entropy(Air, h=h_3, p=p_1)} \]
\[ h_4 = \text{enthalpy(Air, p=p_0, S=s_4)} \]
\[ s_4 = \text{entropy(Air, T=899, P=p_1)} \]
\[ h_{41} = h_3 - E_t(h_3-h_4) \]
\[ s_{41} = \text{entropy(air, p=p_0, h=h_{41})} \]
\[ m_{gas} = m_a + m_f \text{"gas flow rate"} \]
\[ m_f = (h_3-h_{21})m_a/44000 \text{"Fuel flow rate"} \]

\[ P = P_t - P_c \text{ "power output"} \]

\[ Q_{in} = m_{gas}(h_3-h_{21}) \text{"Total heat input"} \]

\[ E = \left( \frac{E_T(T_f-(T_{amb}R_p^{(y-1)/y})/E_t)}{(T_f-T_{amb}-(R_p^{(y-1)/y}))/E_c}) \right)^{1-(1/R_p^{(y-1)/y})} \text{"Efficiency"} \]
\[ T_f = 899 + 273 \text{"firing temperature"} \]
\[ T_{amb} = t_0 + 273 \text{"ambient temperature"} \]
\[ y = 1.33 \text{"specific heat ratio"} \]
\[ R_p = (p_1/p_0) \text{"Pressure ratio"} \]

Formatted equations

\[ p_0 = 1.04166 \]
\begin{align*}
p_1 &= 17.7 \\
Ec &= 0.95 \\
Et &= 0.95 \\
P_c &= ma \cdot (h_2 - h_1) \\
h_2 &= h(Air, P = p_1, s = s_1) \\
h_{21} &= h_1 + \frac{h_2 - h_1}{Ec} \\
s_1 &= s(Air, T = t_0, P = p_0) \\
s_{21} &= s(Air, h = h_{21}, P = p_1) \\
ma &= 97 \\
h_1 &= h(Air, T = t_0) \\
Pt &= mgas \cdot (h_3 - h_{41}) \\
h_3 &= h(Air, T = 899) \\
s_3 &= s(Air, h = h_3, P = p_1) \\
h_4 &= h(Air, P = p_0, s = s_4) \\
s_4 &= s(Air, T = 899, P = p_1) \\
h_{41} &= h_3 - Et \cdot (h_3 - h_4) \\
s_{41} &= s(Air, P = p_0, h = h_{41}) \\
m_{gas} &= ma + mf \\
mf &= (h_3 - h_{21}) \cdot \frac{ma}{44000} \\
P &= Pt - P_c \\
Q_{in} &= mgas \cdot (h_3 - h_{21})
\end{align*}
\[
E = \left[ \frac{Et \cdot Tf - \frac{Tamb \cdot Rp}{Et} \left( \frac{y - 1}{y} \right)}{Tf - Tamb - Tamb \cdot \left( \frac{Rp}{Ec} \frac{y - 1}{y} - 1 \right)} \right] \cdot \left[ 1 - \frac{1}{Rp \left( \frac{y - 1}{y} \right)} \right]
\]

\[
Tf = 899 + 273
\]

\[
Tamb = t_0 + 273
\]

\[
y = 1.33
\]

\[
Rp = \frac{p_1}{p_0}
\]
Appendix II

EES Equations for Cooling Load Calculation

//sensible cooling load

\[
Q_s = (\frac{V_{dota}}{V_a}) \cdot C_{pa} \cdot (t_{aadb} - t_{ci})
\]

\[
v_a = (0.287 + X_{aadb} \cdot 0.462) \cdot \frac{T}{P_{atm}}
\]

\[
X_{aadb} = \left( \frac{P_s}{(P_{atm} / R_h - P_s)} \cdot \frac{M_v}{M_a} \right)
\]

//latent cooling load

\[
Q_l = (\frac{V_{dota}}{V_a}) \cdot (X_{aadb} \cdot (C_{pv} \cdot t_{aadb} + r) - X_{sci} \cdot (C_{pv} \cdot t_{ci} + r) - (X_{aadb} - X_{sci}) \cdot C_{pw} \cdot t_{ci})
\]

\[
X_{sci} = \left( \frac{P_{s1}}{P_{atm} - P_{s1}} \right) \cdot \frac{M_v}{M_a}
\]

V_{dota}=81.63
C_{pa}=\text{specheat(air,T=273)}
t_{aadb}=27
t_{ci}=15
P_{atm}=102.6675
P_s=3.6 \quad "Saturated pressure at 27C"
P_{s1}=1.7 \quad "saturated pressure at 15C"
R_h=0.70
M_v=18
M_a=28.9
C_{pv}=\text{specheat(water,T=300,P=102.6675)}
r=2500.8
C_{pw}=4.2

27+273=T

Q=Q_s+Q_l

Formatted equations

\[
Q_s = \frac{V_{dota}}{V_a} \cdot C_{pa} \cdot (t_{aadb} - t_{ci})
\]

\[
v_a = (0.287 + X_{aadb} \cdot 0.462) \cdot \frac{T}{P_{atm}}
\]

\[
X_{aadb} = \left( \frac{P_s}{P_{atm} / R_h - P_s} \right) \cdot \frac{M_v}{M_a}
\]

\[
Q_l = \frac{V_{dota}}{V_a} \cdot (X_{aadb} \cdot (C_{pv} \cdot t_{aadb} + r) - X_{sci} \cdot (C_{pv} \cdot t_{ci} + r) - (X_{aadb} - X_{sci}) \cdot C_{pw} \cdot t_{ci})
\]
Calculation results

Cpa=1.004
Cpv=1.866
Cpw=4.2
Ma=28.9
Mv=18
Patm=102.7
Ps=3.6
Ps1=1.7
r=2501
Rh=0.7
T=300
taadb=27
tci=15
Va=0.8598
Vdota=81.63
Xaadb=0.01567
Xsci=0.01049
Q=2391
Ql=1247
Qs=1143

\[
X_{sci} = \left( \frac{Ps_1}{Patm - Ps1} \right) \cdot \frac{Mv}{Ma}
\]

\[
Q = Qs + Ql
\]