POTENTIALITY OF INTRODUCING ABSORPTION CHILLER SYSTEMS TO IMPROVE THE DIESEL POWER PLANT PERFORMANCE IN SRI LANKA

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Thesis submitted in partial fulfillment of the requirements for the degree Master of Engineering

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DECLARATION OF THE CANDIDATE & SUPERVISOR

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ABSTRACT

This aims to find the potentiality of introducing absorption chiller systems to improve the diesel power plant performance in Sri Lanka. The energy efficient operation of diesel power plants is very much important for the country due to the high cost of generation of thermal electricity. Therefore waste heat of diesel power plant is utilized to run a suitable absorption chiller. The considered waste heat is mainly of exhaust and cooling water in the diesel engines of the power plant.

The performance of the power generating diesel engines is considered in two ways. That is in terms of specific fuel oil consumption (SFC) and engine deration. The SFC of the engines varies due to many factors. Since the site conditions in Sri Lanka are not in standard conditions the higher SFC and engine deration is possible. The ISO standard site conditions mean the 25°C (77°F) ambient temperature, 30% relative humidity and a model was developed to evaluate the performance of particular engines. All the temperature values in the model are given in Fahrenheit degrees (°F). It is observed that the engine SFC is low and the engine will not derate at the standard site conditions. From the model it is obvious that when the ambient temperature is 70°F (21.1°C) the engine will not derate due to the effect of humidity even though the percentage of relative humidity reaches 100. In contrast, above 133.6°F (56.4°C) ambient temperature the power plant diesel engines derate due to the effect of humidity irrespective of the value of percentage relative humidity.

The investigated model was applied to evaluate the improved performance of a diesel power plant by introducing an absorption chiller system. The building cooling load was additionally integrated to that system. Therefore it further uplifts the advantages by saving electricity of vapour compression air conditioners.

DEDICATION

I lovingly dedicate this thesis to my family, who supported me in each & every way to make this effort a success.

ACKNOWLEDGEMENT

I take this opportunity to express my sincere thanks to Prof. Rahula A. Attalage, Deputy Vice Chancellor of University of Moratuwa, Sri Lanka as my supervisor, for his great contribution to select this topic for the research project and guidance for finding data. I would also like to express my sincere thanks to Dr. Himan Punchihewa, Course Coordinator of MEng/PG Diploma in Energy Technology (2011/12), Dept. of Mechanical Engineering, University of Moratuwa, for the encouragement and guidance given to fulfill this endeavour.

I also thank Dr. Inoka Manthilake, Senior Lecturer, Dept. of Mechanical Engineering, University of Moratuwa, and Mr. Kithsiri Gamage, Mechanical Engineer, Uthuru Janani Power Station, Ceylon Electricity Board for their fullest support shown me to collect data from their data base and guidance given me to gather details from relevant authorities.

I would like to extend my thanks to friends and colleagues specially batch mates of MEng/PG Diploma in Energy Technology (2011/12) for their enormous encouragement, knowledge and help given me to make this task successful.

At last, but not least, I would like to thank my loving wife, son and parents for their tireless support and encouragement during the course of my academic career.

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LIST OF ABBREVIATIONS

Abbreviation	Description
AC	Air Conditioning
AN	Acid Number
BSFC	Break Specific Fuel Consumption
CEB	Ceylon Electricity Board
CFC	Chloro Fluoro Carbon
COP	Coefficient of Performance
DEMA	Diesel Engine Manufactures Association
HCFC	Hydro Chloro Fluoro Carbon
HP	Horse Power
HT	High Temperature
HTG	High Temperature Generator
LT	Low Temperature
PPP	Privet Power Producer
RH	Relative Humidity
RPM	Rounds Per Minute
SCV	Steam Control Valve
SFC	Specific Fuel Oil Consumption
TBN	Total Base Number
TOC	Total Operating Cost
TR	Tons of Refrigerant
UJPS	Uthuru Janani Power Station

LIST OF APPENDICES

Chapter 1

Introduction

1.1 Background

Power generation industry in Sri Lanka is being carried by Ceylon Electricity Board (CEB) and the Private Power Producers (PPP). The compositions are given in Figure 1.1 and Figure 1.2.

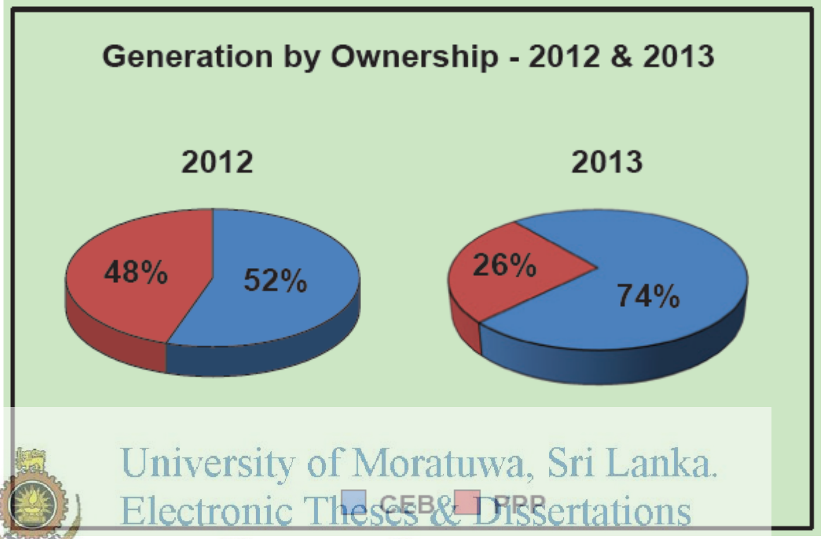


Figure 1.1: Electricity generation by ownership 2012 & 2013[1]

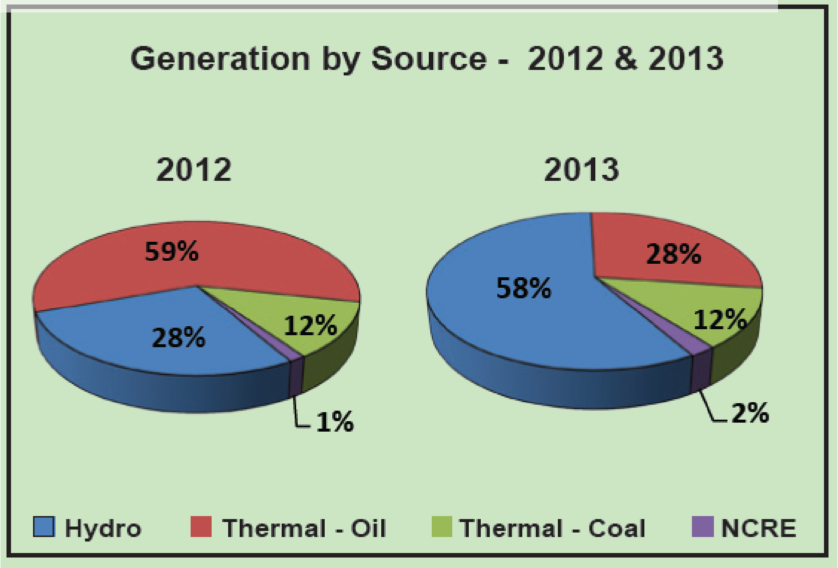


Figure 1.2: Electricity generation by source 2012 & 2013[1]

Thermal-oil power generation based plants in Sri Lanka are playing very important role to cater the demand of electricity in the country. The thermal-oil power generating plants in the country are combined cycle plants and diesel power plants which are owned by both CEB and PPPs. The diesel power stations in the country are mentioned below.

CEB is having its own diesel power stations as following running capacities [2].

Sapugaskanda Power Station A	72 MW
Sapugaskanda Power Station B	72 MW
Uthuru Janani Power Station	24 MW
Sub Total	168 MW

PPPs have diesel power plants as following running capacities.

Asia Power	51 MW
Colombo Power	60 MW
Heladhanavi	100 MW
Ace Embilipitiya	100 MW
Northern Power	27 MW
Sub Total	338 MW
Total running capacity	506 MW

Table 1.1: Total annual energy dispatch of diesel power stations in Sri Lanka [3]

Power Station	Annual Total Energy Dispatch / GWHrs		
	2012	2013	2014
Ceylon Electricity Board			
Sapugaskanda Station A	392.599	181.983	237.116
Sapugaskanda Station B	533.208	390.940	403.432
Uthuru Janani Power Station	0.000	125.298	94.649
Chunnakam Power Station	5.156	0.303	0.000
Independent Power Producers			
Asia Power	332.982	157.356	179.547
Colombo Power (Barge)	481.685	331.789	293.063
Heladhanavi	699.090	469.490	481.823
Ace Embilipitiya	621.023	395.928	463.404
Nothern Power	101.833	23.310	57.391
Aggreko – Chunnakam	47.287	0.000	0.000
Lakdhanavi	99.403	0.000	0.000
Ace Power (Matara)	85.732	0.000	0.000
Ace Power (Horana)	178.947	0.000	0.000
Total	3,578.945	2,076.397	2,210.425

*Values of year 2014 have been considered except the month of December.

1.2 Problem identification

Sri Lanka is mainly generating electricity from hydro power stations, thermal power stations and wind power plants. Recently, the thermal power generation in the country has increased due to the effect of droughts which lessen the hydro power generation and also the increasing electricity demand. Therefore, the energy efficient operation of diesel power plants is very much important for the country. Especially in a country like Sri Lanka, the ambient is relatively hot and wet which is not favorable for the diesel engines. Hence the cooling effect could be an important function for power plant diesel engines and the performances of it could depend much on it. Optimizing the fuel oil consumption of diesel power plant engines is also important to afford the unit price of electricity as fuel oil component plays the major part for the electricity pricing in the country. Therefore, the waste heat recovery system combined with a suitable absorption chiller system to improve the power plant diesel engine performance is important.

1.3 Aim and objectives

The aim is to study the performance reductions in diesel power plant and find improvement potentialities combined with suitable absorption chiller systems which suit Sri Lanka with following objectives.

1. To study the parameters that effect diesel engine performance.
2. To develop a model to apply absorption chiller systems to ensure improved performance of diesel power plants.
3. To evaluate the model in a case study.

1.4 Methodology

The methodology was mainly divided in to three numbers of phases. The details of each phase are given below.

1.4.1 Phase 1: Literature review

- Study power generating diesel engine performance for varying parameters.
- Explore waste heat recovery and absorption chiller systems.

1.4.2 Phase 2: Development of a model

- Prioritize the parameters.
- Find relation of the engine performance for those key parameters.
- Identify the process for a model.
- Develop an adequate model.

1.4.3 Phase 3: Applying the model in a case study

- Select a suitable power station and apply the whole scenario including the model.
- Evaluation and discussion

Chapter 2

Diesel Engine and Auxiliary Systems

The related studies of diesel engine and its auxiliary systems are covered under this chapter. The power generating diesel engine performances for varying parameters were included for the study. The corresponding available news paper articles, journals, books and internet were referred.

2.1 Diesel engine and working principal

The diesel engine is an internal combustion engine which initiates ignition by the heat of compression and burns the fuel that has been injected into the combustion chamber. The diesel engine is also known as a compression-ignition engine.

2.1.1 Thermodynamic cycle

This does the approximation of the pressure and the volume of the diesel engine combustion chamber.

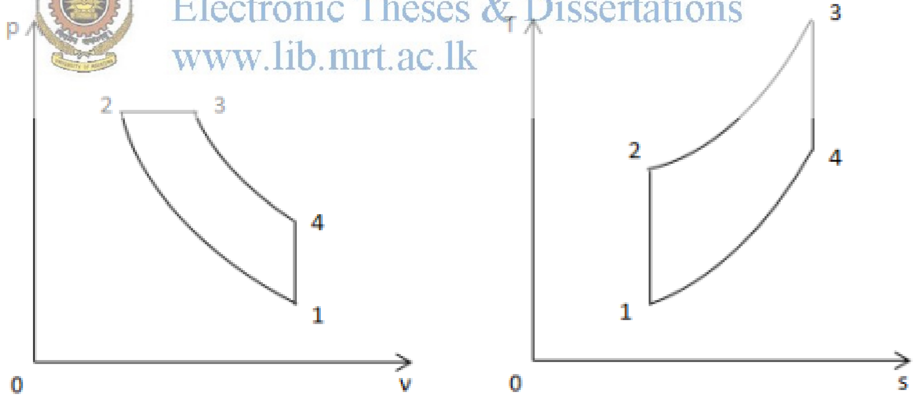


Figure 2.1: Thermodynamic cycle [4]

The piston completes four separate strokes which constitute a single thermodynamic cycle. A stroke refers to the full travel of the piston along the cylinder from Bottom Dead Center (BDC) to Top Dead Center (TDC), in either direction. The four separate strokes are termed as,

1. Intake
2. Compression
3. Power
4. Exhaust

2.1.2 Heat supplied to diesel engine

Heat supply by the fuel is basically converted in to four categories as heat converted in to work (Indicated horse power), heat rejected to the cooling water, heat rejected through exhaust gases and heat loss to the surrounding by means of radiation. The indicated horse power is the summation of break horse power and the mechanical energy loss in the diesel engine. The heat that is rejected through exhaust gases can be divided as heat in the dry exhaust gas and heat in the water vapour.

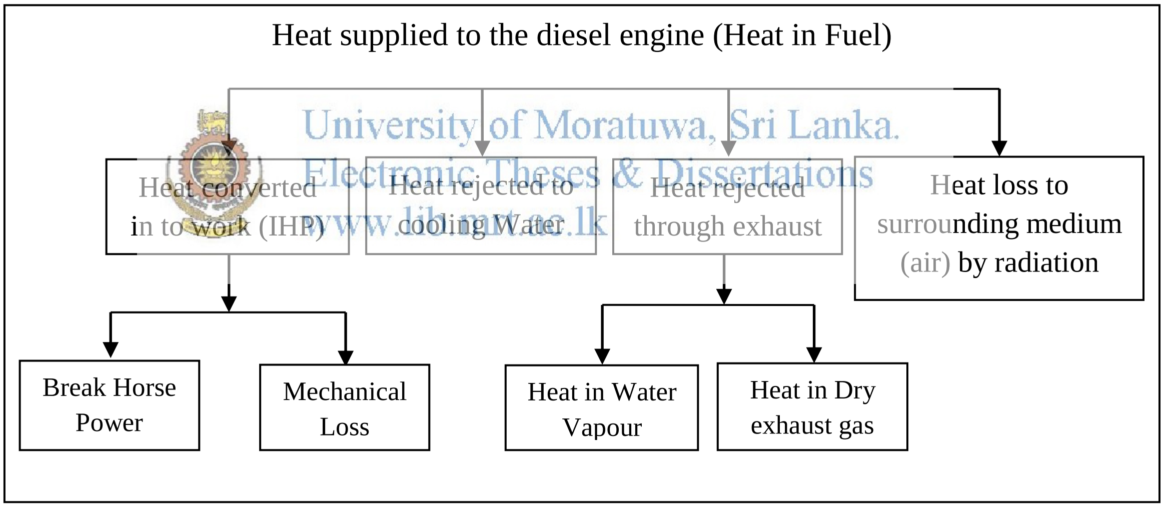


Figure 2.2: Heat supplied to the diesel engine (Heat in fuel)

2.1.3 Expected outcomes of an optimum diesel engine

The below mentioned factors are expected from an optimum diesel engine

- High thermal efficiency
- Low lube oil consumption
- Long component life time

- Fuel versatility
- Low emission level
- Easy maintenance

2.2 Engine auxiliary systems

The engine auxiliary system is very much important for engine operation. For instance, the cooling water systems, lube oil system, fuel system, turbo charging, charge air system and engine cooling methods. The effect of site conditions as ambient temperature and humidity are also very important.

2.2.1 Cooling water systems

The cooling system is a system of parts and fluid that work together to control operating temperature of the engine for optimal performance. The system is made up of passages inside the engine block and heads. Cooling water systems consist of water pump with a motor unit to circulate the coolant, a thermostat to control the temperature of the coolant, a radiator yard or cooling tower to cool the coolant, a pressure regulator to control the pressure in the system, and hoses and pipes to transfer the coolant from the engine to the radiator.

The functions of the cooling water systems fulfill can be classified as,

1. Cooling the engine components such as liners, pistons and cylinder covers, etc to maintain the temperature within the acceptable limits. Otherwise the materials used in the construction of engines are unable to withstand the thermal stresses that would be imposed upon them. In some large engines, a separate cooling water circuit is maintained to cool the cylinder liners and the piston area.
2. Cooling the lube oil in order to control the temperature and then the viscosity of lubricating oil remains within the limits which required for effective lubrication. In some diesel engines lubricating oil is used for cooling the piston. In such circumstances, heat of the lube oil is rejected by the cooling water system.

3. Cooling the charge air in the pressure charged engines, to increase the density of the combustion air entering the combustion chamber while reducing its temperature. This increases engine power by burning more fuel on each piston firing stroke whilst maintaining acceptable exhaust valve temperature.

➢ Effect of engine cooling water temperature variations

Providing sufficient cooling for the engine components and its systems is the main function of an effective cooling system. The cooling system must be purged properly to avoid vapour locks that form highly thermally loaded zones of components. High temperature of the cooling water cannot withstand the forces that would act on the engine components. However, suitable components must ensure that maximum permissible temperatures are not exceeded in any of the engine load point. The low temperature of engine cooling water system could make high wear and tear in the cylinder liner and the piston rings of the engine. It also reduces the life time of the spares [5].

➢ Effect of engine cooling water pressure variations

Increasing the cooling system pressure contributes to raise the boiling point of the coolant. Correct system pressure is required for the proper water pump seal lubrication. If the pressure increases than the set pressure point, a pressure regulator will release the pressure [6].

➢ Effect of engine cooling water chemical level variations

This primarily protects cooling water lines from the scaling, corrosion fouling and biological contamination which lead to the followings.

- Reducing maintenance cost
- Reducing equipment repair or replacement cost
- Minimizing shutdowns for cleaning and replacement of system components
- Increasing the heat transfer efficiency and energy efficiency of the system
- Reducing fuel costs for power generation

Consistent control of both the corrosion inhibitor chemicals and the key water chemistry characteristics is very important to execute above matters. If the chemical level is high in the cooling water system it directly affects the chemical cost. The best step to achieve the minimum Total Cost of Operation (TOC) and minimum stresses of the system is selecting an appropriate treatment program and operating conditions [7].

- **Effect of charge air cooling water parameter variations**

Charge air cooling water circuit is responsible to reduce the temperature of lubrication oil and charge air which required to engine combustion. Increases of charge air temperature admit less dense air with high temperature for the combustion chamber which ultimately causes poor combustion efficiency. In addition, high charge air temperature leads to high exhaust temperature also. Low temperature of charge air cooling water circuit can reduce the combustion efficiency of the diesel engine and lessen the engine loading speed also. Since the diesel engines are having high compression ratios extreme higher density of charge air can shift the operations which provide optimum efficiency [8].

In most of the diesel power plant engines, the charge air system and the lube oil system are cooled by a cooling water circuit which is called as Low Temperature (LT) cooling water system. Therefore, lubrication oil temperature varies according to the charge air cooling water temperature of the engine.

2.2.2 Lube oil system

Lubricating oil act as life blood of diesel engines and it is doing the following important functions in the diesel engine.

- Wear reduction of components such as bearings, pistons, piston rings, cylinder liners
- Cooling the engine components
- Corrosion prevention
- Cleaning pistons while preventing sludge build-up on internal surfaces

- Prevent leakages due to seal failure (Sealing the surfaces).

Engine lube oil is basically a mixture of two components as mentioned below.

- Base oil (typically 88 - 92%)
- Chemical additives (8 - 12%)

Since the base oil alone cannot provide all the expected lubricating oil functions in the diesel engines, the chemical additives play an increasingly important role in the oil formulation [9].

2.2.3 Lube oil system parameter variations

- **Effect of oxidation**

Oxidation occurs when the base oil is attacked by oxygen. The oxidation is accelerated by heat, pressure and catalytic materials in the system. Disadvantages of oxidation are forming lacquer deposits, corrosion of metal components, and increasing the oil viscosity and these disadvantages prevent the ability of lubrication.

- **Effect of nitration**

Nitration is caused by oil degradation in a reduced oxygen environment and results in nitrogenous by-products. These compounds can combine with water and form nitrous acids in the lubricant. These acids can reduce additive effectiveness and increase the rate of oil degradation by formation varnish, lacquer, sludge and engine deposits.

- **Effect of sulfation**

This is due to the sulfur in the lube oil system which reacts with the oxygen in the presence of heat. Sulfurous products form deposits, lacquer, varnish and sludge in the lube oil. They can deteriorate the additives and also can react with water to form sulfuric acids that corrode the metals and degrade the lubricant.

- **Effect of Acid Number (AN)**

Acid number is the quantity of acid or acid-like constituents in the lubricant. An increase in AN usually indicates lubricant degradation due to oxidation, nitration or the presence of acidic products from oil degradation, contamination or combustion. An increase of the AN can corrode the metal components of the system.

- **Effect of soot**

Soot is formed during the combustion process. Excess soot increases the lube oil viscosity, higher pumping costs, power loss and the risk of lubricant starvation, especially at start-up. Soot can stimulate wearing of the valve train which leads to poor fuel economy due to the variation of injection timing and valve timing.

- **Effect of fuel dilution**

Fuel dilution refers to mixing fuel with the lube oil normally due to the mechanical malfunction, leakage or abnormal operating conditions. This is harmful to the engine because there is rapid reduction in lube oil film strength due to a reduction in viscosity.

- **Effect of viscosity**

Viscosity is one of the most prominent parameter of lubricating oil. It is an indicator of the lubricant's film strength. The oil viscosity is also an indicator of contamination from soot, glycol, fuel and oxidation.

As mentioned above, the viscosity will be increased by higher levels of soot, oxidation at higher temperatures, and nitration. It does not allow the required oil amount to pass through and therefore less lubrication affects the mating surfaces. Also the required lube oil pressure cannot be maintained due to high flow resistances. Most commonly, soot accumulation in the lube oil would be the primary factor for the increment of viscosity. Filter plugging, bearing failures, sludge formation and fuel economy losses are the consequences of high viscosity in lube oil.

The right protection to the engine parts is not guaranteed by the low viscosity due to the less strength of the oil films. Lower viscosity of lube oil can make it easier to cold start the engine because it presents less resistance for moving parts. This could lead to get enhanced fuel economy while having high wear and tear.

➢ Effect of temperature variation

This can decrease the lubricating function and increase the lube oil consumption. Normally high temperature can reduce the viscosity of the lube oil. Continuous increase of lubricant temperature will affect oxidation stability causing a rise in viscosity due to varnish and sludge formation. The formation of explosive vapour from lubricating oil is greatly accelerated by a rise in the temperature of the lubricating oil. The low temperature of lube oil increases the viscosity and lessens the lubrication functions. It totally shifts the lube oil parameters from its optimum ranges [10].

➢ Effect of Total Base Number (TBN) variation

Diesel engine lubricant possesses alkalinity and it is called as total base number (TBN). TBN of the lube oil is a key factor which governs oil life in the diesel engines and it neutralize acids forming during normal engine operation. The crankcase oil must maintain an adequate level of TBN during operation to prevent the increasing of acids. The low TBN value increases the oxidation, corrosive wear, cylinder liner, ring and bearing wear. In contrast, increase in TBN would cause an increase in detergents in lube oil. The increase in detergents implies a subsequent increase in metallo-organic compounds forming ash after the detergent has been consumed.

➢ Effect of high particular matters

Anti-wear additive can be affected by the high concentration of particular matters. The increase of viscosity, filter plugging, reduction of lubricant flow, valve train wear, bearings failure and sludge could be happened. Finally, the fuel consumption will increase.

- **Effect of high water content**

Water is widely considered as one of the most destructive contamination to a lubrication oil system. The water in oil affects in two ways as water destroying physical properties and chemical properties. Bacterial growth on the water/oil interface leading to corrosion and change lubricating oil characteristics. This can lead the damage of machine components. Moreover, the lubrication function of the oil deteriorates due to high water contents [11].

2.2.4 Fuel oil system

The fuel oil system in a diesel power plant includes fuel storage tanks, fuel transfer pumps, strainers and connecting pipe works, separators, etc. Fuel transfer pumps are required to transfer fuel from delivery point to storage tanks and from storage tanks to engine. Strainers (filters) are required to ensure clean fuel to the engine. Fuel oil heaters are required to acquire relevant temperatures and viscosities of fuel.

Fuel oil delivered to the power plant is received by the storage tanks. Impurities are allowed to settle down in the storage tanks and rest is removed by separators and strainers. Oil is pumped from storage tanks or settling tanks to daily consumption tank through separators. The fuel injection system is responsible to measure adequate quantity of fuel to the engine. It also atomizes and injects the fuel into the combustion chamber.

2.2.5 Fuel oil system parameter variations

- **Effect of temperature variations**

Temperature is a very important parameter of the fuel oil system. The high temperature can course low viscosity and then oil leaks through the injection pumps. So it cannot supply the required fuel amount with right pressure. As a result, the engine will not reach its full load. In addition, the low temperature of fuel oil is also a disadvantage. It increases the viscosity and this can make high wear in the injector pumps. Less efficiency in combustion is possible due to the inadequate atomization. Finally when temperature of the fuel oil shifts from its optimum ranges it leads to poor performance of the engine.

➢ Effect of viscosity variations

Heavy fuel oils are normally purchased on the basis of a limiting viscosity due to storage, handling, or engine-related restrictions. Increasing fuel viscosity further produces incomplete combustion products and inadequate atomization in the diesel engine. In addition, due to high viscosity it can make high wear rate in the injector pumps. In contrast, low viscosity fuel oil leaks through the injection pumps and as a result less fuel pressure occurs in the system. This leads less fuel to the combustion chamber and ultimately engine does not reach to its full load output.

➢ Effect of pressure variations

Fuel pressure is an important factor to the diesel engine combustions. The required fuel pressure to the suction port of the injection pumps is always welcomed. The low fuel oil pressure leads to less atomization and fuel drops remains on the tip of the injector nozzle. These cause irregular engine operations, knocking, irregular cylinder firing and engine speed fluctuations.

2.2.6 Turbo charging system

A turbocharged engine is typically more fuel efficient than a regular engine at a given power. Turbo charging is used to increase air flow through engines, while increasing power density as mentioned below.

2.2.7 Results of turbo charging [12]

1. Power Increase: Mean effective pressure of the engine can be easily increased by 30-50%. Hence, the power output increases.
2. Fuel Economy: Because of increased turbulence, better combustion and better mixing of the fuel and air.

3. Mechanical Efficiency: The mechanical efficiency referred to maximum load is increased because the increment of frictional losses with a super charger driven directly from the engine is quite smaller in comparison to the power gained by super charging.
4. Fuel Knock: It is decreased due to increased compression pressure. The increment of inlet pressure decreases the ignition lag and this reduces the rate of pressure rise in the cylinder causing an increased smoothness of operation.
5. Volumetric efficiency: Volumetric efficiency is increased since the clearance gases are compressed by the induced charge which is at a higher pressure than the exhaust pressure.

The following additional advantages are also important.

- Lower engine mass per power unit
- Turbo charger works as a silencer
- Ambient air density variation does not affect much as in engine without turbo charging unit
- Less emissions due to harvesting energy from the exhaust gases

2.2.8 Charge air system

This includes air filters, ducts, coolers and compressor. The system supplies the required air for the combustion. Air is drawn from outside the engine room which is normally outside environment and delivered to the intake manifold through the suction air filters. Suction air filters remove the dust and other suspended impurities from air which causes the wear and tear of the engine and combustion problems. The supercharger increases the pressure of air supplied to the cylinder so that it could develop an increased power output.

2.2.9 Effect of charge air

The turbulent effect of charged air improves mixing of the fuel particles with the air. During the process of compressing air, the charge produces many effects. The temperature of the charge air is raised by compression. Therefore, it results in a higher temperature within the cylinders which is particularly beneficial to introduce better vapourization of the fuel. However, higher charge air temperature leads to reduce the density. The increase in the temperature of the charge air also affects the detonation of the fuel.

2.2.10 Charge air system parameter variations

- **Effect of temperature variations**

Temperature of the air can vary the density of it as the air is highly compressible medium. High temperature of the charge air admits less mass of air to the combustion chamber. This leads to high specific fuel oil consumption via inefficient combustion. The low temperature of charge air leads to less vapourization of fuel and it makes poor combustion efficiency again. More condensation is possible when charge air temperature goes down. Failure to remove the condensate water in the scavenge air may damage the oil film between the piston rings and the cylinder liner.

- **Effect of pressure variations**

Pressure of the charge air determines the mass flow rate to the combustion chamber. Due to high pressure, high mass flow rate of air to the combustion chamber is possible. Therefore, it can result in inefficient air fuel ratio. High pressure can also lead high stresses in the components which are not favourable for the long life operation. Low pressure of charge air leads less mass of air to the combustion chamber then it tends to the poor air fuel ratio which ultimately increases the specific fuel oil consumption.

2.2.11 Effect of ambient temperature

ISO condition of ambient temperature for the diesel engine is 25^{o}C. Air density varies inversely with its temperature. However, practically ISO condition cannot be achieved throughout the year. Hence, the engine performance varies accordingly.

High ambient temperature admits less mass of air to the engine cylinder for combustion which also leads high temperatures of exhausts.

Low ambient temperature leads inefficient combustion in the cylinders. This is mainly due to the less ignition temperatures even though the density of air is high.

The article by Hindren A. Saber1a, Ramzi R. Ibraheem Al-Barwari, Ziyad J. Talabany, (2013), Effect Of Ambient Air Temperature On Specific Fuel Consumption Of Naturally Aspirated Diesel Engine, Science and Engineering, Vol. 1(1), 1-7 show the followings.

“It was observed that the brake specific fuel consumption increased with increasing inlet air temperature while brake specific fuel consumption decreased with increasing engine torque.” (2.5LDetroit Diesel engine) [13].

In the below mentioned Figure 2.3 as an example of 50 Nm constant torque line, it is obvious that when the inlet air temperature has reached higher (10-65^{o}C) the break specific fuel consumption of the engine has gone higher (0.35-0.36 kg/kWh). The data were collected at constant engine speed of 1,500 rpm.

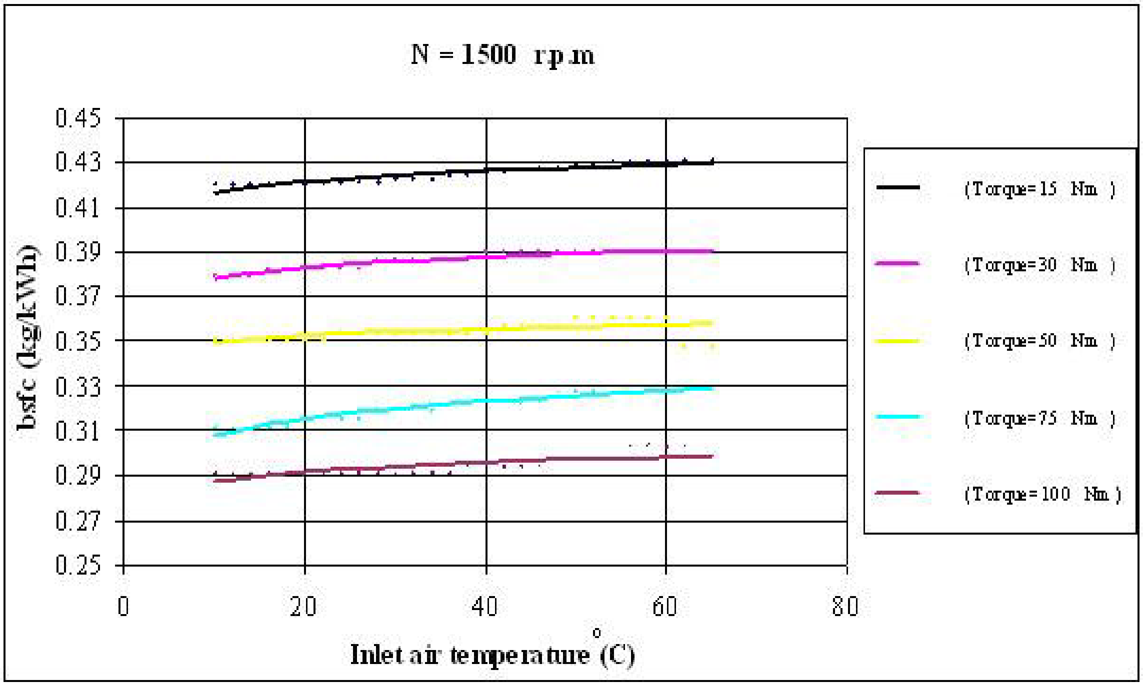


Figure 2.3: Effect of inlet air temperature on the brake specific fuel consumption, at constant engine speed (1,500 rpm) and different engine torques [13]

In the below mentioned Figure 2.4, it is obvious that when inlet air temperature has increased the break specific fuel consumption has also increased in the constant engine speed lines.

As an example, when the engine speed is 2,300 RPM the inlet air increment is 10^{o}C - 65^{o}C and the engine break specific fuel consumption has increased from 0.327 to 0.345 kg/kWh.

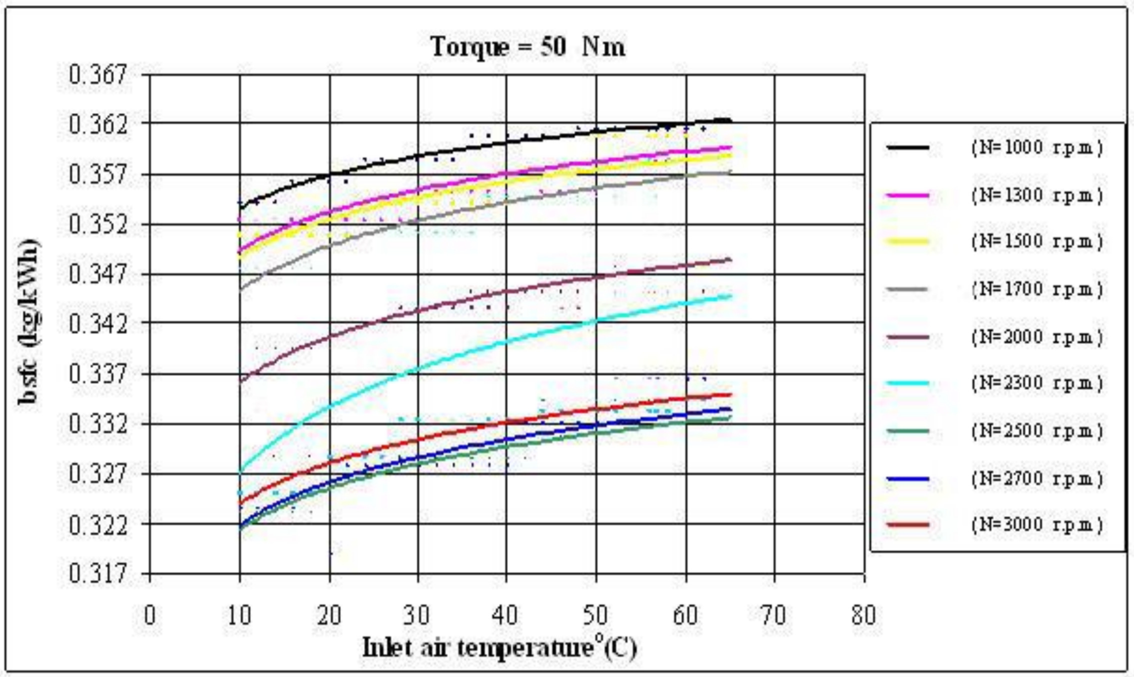


Figure 2.4: Effect of inlet air temperature on the brake specific fuel consumption, at constant engine torque (50 Nm) and different engine speeds [13]

2.2.12 Effect of Relative Humidity (RH)

Moist air is a mixture of air and water vapour. Compared to the effect of temperature, the influence of the RH of air is relatively less. The important point is that more humid air contains more moisture which occupies the space of oxygen while less humid air contains less moisture which has more oxygen for combustion. The ISO 3046 standard reference condition of relative humidity is 30% [14].

2.2.13 Engine cooling methods

- **Air cooling systems**

In these systems fins or extended surfaces are provided on the cylinder head, cylinder walls etc. The heat in the engine cylinders is conducted to the fins and from the fins, heat dissipates to the air. The amount of heat dissipated to the air basically depends on the mass flow rate of air, fin surface area and the thermal conductivity of metal used for fins.

Air cooling is possible in very cold climates where the water is in freeze. Un-necessity of radiators or cooling water pumps is the advantages of air cooling systems. The air cooling systems do not have water leakage repairs and maintaining cooling water chemical levels also. In contrast, the air cooling is comparatively less efficient. Air cooling is used in aero planes to maintain light weights and motorcycle engines where the engine is exposed to the air directly.

- **Water cooling system**

Water cooling system mainly consists of water pump, radiator, thermostat valve, fan, water jackets, and antifreeze mixtures. Water cooling can adapt uniform cooling of cylinder, cylinder head and valves. Specific fuel consumption of engine improves while having properly designed cooling water system. In the water cooling system, the engine does not need to be provided at the front end in moving vehicles to have direct contact of air. Engine is less noisy than the air cooling engines as it has water for damping noises.

Water cooling depends on the supply of water specially in cooling tower systems. The power consumption of the water pumps is a disadvantage of water cooling systems. If the water cooling system fails while the engine is continuously running, it will cause severe damages to the engine components. The water cooling system is expensive as it has been integrated many components than the air cooling systems. It requires more maintenance and care while in operation.

Chapter 3

Waste Heat Recovery Methods

The related studies of the application of waste heat recovery methods and vapour absorption chillers are addressed in this chapter. In addition, sources of waste heat of diesel engine are discussed here.

3.1 Waste heat

The waste heat means the heat energy that is exhausted to atmosphere without getting any practical use. Waste heat sources are mainly heated products in the production systems, hot combustion gases discharged to the atmosphere and heat transfer from hot equipment surfaces etc. Mainly the waste heat can be reduced by installing waste heat recovery technologies or improving equipment efficiency.

3.2 Factors affecting waste heat recovery

Important parameters that should be considered are heat quantity, temperature, composition, allowed temperatures, operating schedules and availability. These parameters analyze the usefulness of the heat for the particular practical purposes.

3.3 Waste heat recovery technologies

Various waste heat recovery technologies which improve the energy efficiency safe guarding the environment are mentioned below.

3.3.1 Recuperator

Recuperators are mostly used to recover the heat from the exhausts and heat the incoming low temperature air. Some typical applications are soaking ovens, annealing ovens, melting furnaces, reheat furnaces and incinerators.

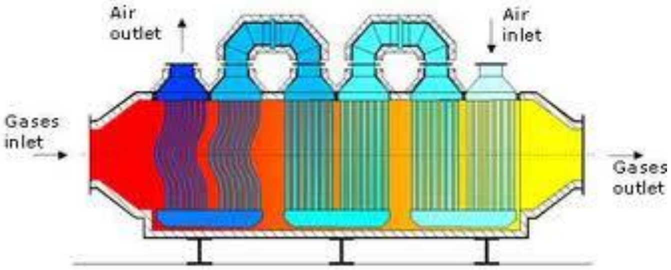


Figure 3.1: Recuperator [15]

3.3.2 Regenerator

- **Furnace regenerator**

Regenerative furnaces consist of two chambers in which hot and cold air flows alternatively. When the combustion exhausts pass through one channel, the wall around it (bricks) absorbs heat from the combustion gases and increases its temperature. Once it gets heated the combustion gases are diverted through the other chamber and incoming combustion air is diverted through the heated chamber. Then heat is transferred to the combustion air entering the furnace. Two chambers are used; one chamber absorbs heat from the exhaust gases while the other transfers heat to the combustion air.

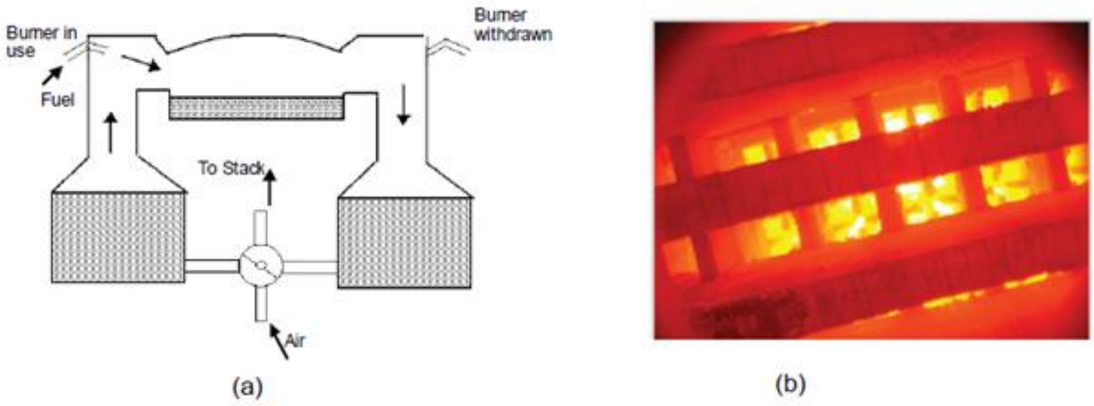


Figure 3.2: (a) Regenerative furnace diagram, (b) Checkerwork in glass regenerative furnace [16]

- **Rotary regenerator/Heat wheel**

Rotary regenerators also referred as heat wheels use a rotating porous disc placed across two parallel ducts. The wheel rotates between the two ducts and transfers heat from the hot gas duct to the cold gas duct (Figure 3.3).

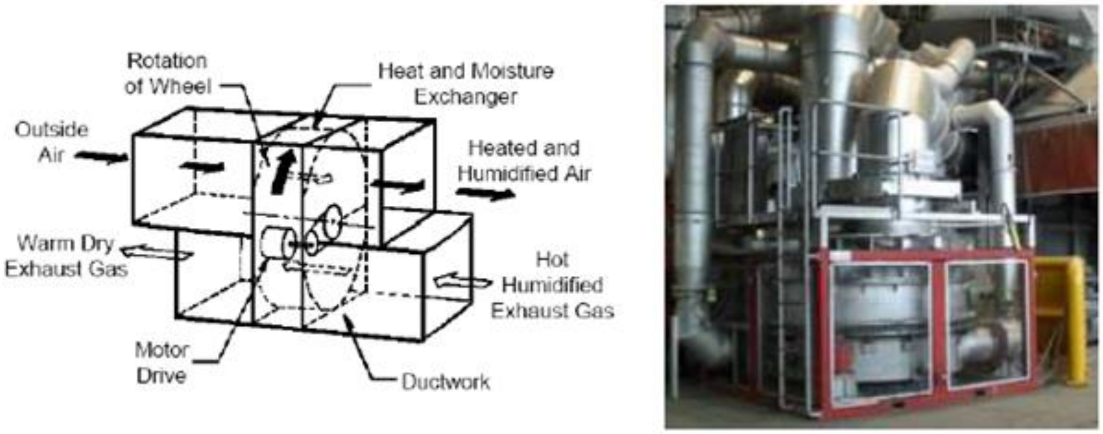


Figure 3.3: (a) Rotary Regenarator, (b) Rotary Regenerator on a Melting Furnace [16]

3.3.3 Finned tube heat exchangers/Economizers

The finned tube heat exchangers/ economizers consist of a round tube with attached fins that maximize surface area and heat transfer rates (Figure 3.4). Mostly, the cold liquid flows through the tubes and receives heat from hot exhaust gases flowing across the fins. Finned tube heat exchangers are used to recover heat from exhaust gases for heating liquids. Applications of finned tube heat exchangers/ economizers include boiler feed water preheating, hot process liquids, hot water for space heating, or domestic hot water.

Figure 3.4: Finned tube heat exchangers [17]

3.3.4 Shell and tube heat exchangers

The shell and tube heat exchangers consist of bundle of small parallel tubes. These tubes are packed within a cylindrical shell (Figure 3.5). One fluid flows through the tubes while the other fluid flows through the shell which is over the tube bundle. The tube walls transfer the heat energy from the hotter fluid to the cold fluid. The Buffells are placed inside the shell to ensure that flow is passing properly across each section. Depending upon the requirement of the heat exchanger one or more tube passes can be achieved. Shell and tube heat exchangers are the most widely used in all type of heat exchangers. They are relatively low cost compared to other types and are available in various ranges of size, capacity, material of construction.

Figure 3.5: Shell and tube heat exchangers [18]

3.3.5 Waste heat boilers

Waste heat boilers can be water tube or gas tube boilers in which hot exhaust gases are used to generate steam or hot water. The exhaust gas is from a diesel engine, a gas turbine, an incinerator or any other source of waste heat. The use of finned tubes inside the boiler extends the heat transfer areas and allows a more compact size. The conventional waste heat boiler cannot generate superheated steam. If superheated steam is needed, an external supper heater is required.

3.4 Vapour absorption chiller systems

Vapour absorption refrigeration cycles absorb the heat when the refrigerant evaporates at lower pressures and reject the heat from the refrigerant at a relatively higher pressure. Unlike the electrical energy in the more familiar vapour compression cycle vapour absorption chillers utilize heat energy. This is because electric chiller employs a mechanical compressor whereas the absorption chillers use heat source and do not use a mechanical compressor. This makes the system very attractive when there is a cheap source of heat, such as solar heat or waste heat. Absorption chillers have recently gained widespread acceptance because they can operate with industrial waste heat streams.

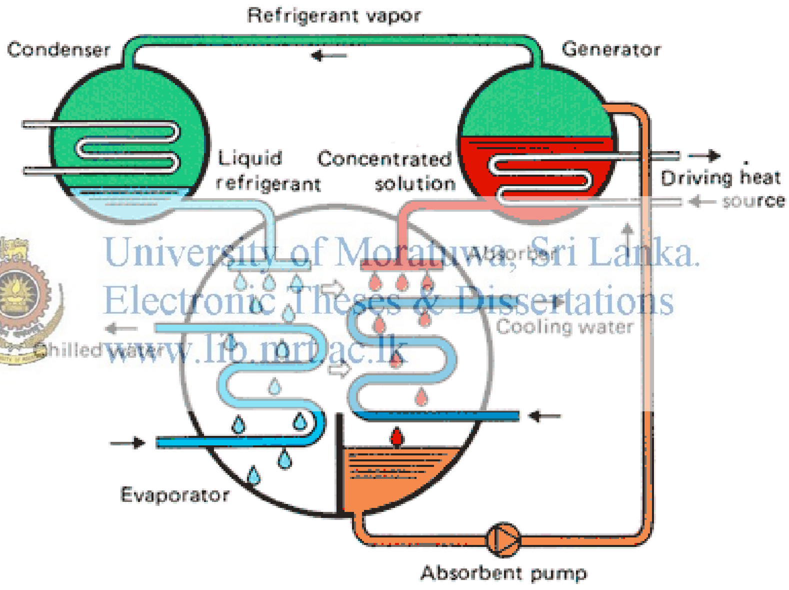


Figure 3.6: Simplified absorption cycle [19]

In the simplest absorption system design, the absorption machine consists of 4 basic components, generator, condenser, evaporator and absorber.

The absorption machine operates under two pressures

1. High pressure refrigerant separation side corresponding to the condenser-generator
2. Low pressure absorption process in vacuum corresponding to evaporator-absorber

3.4.1. Vapour absorption chiller classification

Absorption chillers are classified as single effect absorption chillers, double effect absorption chillers and triple effect absorption chillers.

"Single-effect absorption chillers have coefficient of performance (COP) of approximately 0.6-0.8 out of an ideal 1.0. Since the COPs are less than one, the single-effect chillers are normally used in applications that recover waste heat such as waste steam from power plants or boilers. Double-effect absorption chillers have COPs of approximately 1.0 out of an ideal 2.0. While not yet commercially available, prototype triple effect absorption chillers have calculated COPs from 1.4 to 1.6. The COP metric is also applied to electric chillers. However, since COP is based on site energy, it is not good for comparing gas and electric chiller efficiencies." [20]

"Absorption systems (Lithium bromide and Ammonia) have a higher complexity than conventional air conditioning (AC) systems, but do offer a potential increase in efficiency if high temperature waste heat is available."[21]

3.4.2 Applications of absorption chiller systems

Specific applications where absorption chillers have a substantial advantage

1. Organizations that large amount of heat is usually dump to the surrounding as waste heat can be converted to useful refrigeration
2. During the simultaneous need of heat and power (cogeneration system), absorption chillers can utilize the thermal energy to produce chilled water.
3. For facilities where the electrical supply is expensive, unreliable, or unavailable, it is easier to achieve heat input. Absorption chillers use very little electricity in comparision to an electric motor driven compression cycle chiller.
4. Elimination of the use of Chloro Fluoro Carbon (CFC) and Hydro Chloro Fluoro Carbon (HCFC) refrigerants
5. Low maintenance

3.4.3 Choice of refrigerant- absorption pairs

The refrigerant and absorption pair depends on the required application. The below discussion lies on the Lithium Bromide water vapour absorption system and Ammonia water vapour absorption systems.

➢ **Lithium Bromide-water absorption systems**

A fluid pair comprising lithium bromide salt solution as absorbent and water as refrigerant is vastly used for absorption chiller applications. Water as refrigerant is much more volatile in comparison to lithium bromide which is practically non-volatile at the operating temperatures and pressures in the system. Hence it is feasible to separate the refrigerant from the absorbent for proper evaporation of the refrigerant in the evaporator.

A concentrated lithium bromide solution has great affinity for water. Since the operating pressures are low in the lithium bromide-water absorption system, the pumping cost is low and also the wall thickness of the shell gets reduced in comparison to the ammonia-water fluid combination.

The evaporation temperature must be kept above the freezing point of water and therefore the temperature of chilled water of such systems cannot be less than 5^{o}C (41^{o}F).

Due to the corrosive environment of lithium bromide the inhibitors need to be added to the system to protect the metal parts of the system.

➢ **Ammonia-water absorption systems**

A fluid pair comprising ammonia as refrigerant and water as absorbent is mainly used for low temperature refrigeration applications. Ammonia water as working media has following advantages.

- High affinity
- High stability
- Moderate operating pressure

- Non-corrosive to carbon steel

Water has great affinity for ammonia and can dissolve enormous amounts of ammonia vapour and there is no solidification problem encountered with ammonia refrigerant over a wide range of evaporating temperatures even up to very low temperature applications.

Since water also evaporates when the ammonia water solution is heated in the 'generator', the ammonia vapour from the 'generator' is mixed with water vapour. If the ammonia vapour mixed with water-vapour reaches the condenser, condensation of water vapour will interfere with the evaporation of the ammonia liquid in the evaporator and reduce the refrigeration capacity. A separate rectifier is generally provided to overcome this problem [22].

3.4.4 Limitations of Lithium Bromide-water and Ammonia-water systems

Lithium Bromide absorption chillers are commonly used to provide chilled water where steam or hot water is available. The single effect and double effect Lithium Bromide absorption chillers are available with the COP 0.6 and 1.2 respectively. The double effect unit does cost about 1/3 more than the single effect unit and it also requires a higher pressure steam. The COP of an aqua Ammonia system is less than a single stage Lithium Bromide system while its initial cost is several times higher. Therefore, lithium Bromide is generally preferred than the aqua Ammonia systems [23].

The primary problems in the absorption chillers are crystallization and vacuum leaks. When the machine operates too close to the saturation temperature of the lithium bromide solution and the lithium bromide begins to precipitate out of the solution (crystallization). While this will not damage the machine, it usually requires application of external heat to get the lithium bromide back in to the solution. But in the modern absorption chillers have electronic controls which prevent the chiller from operating at temperatures and concentrations which allow crystallization.

In the ammonia vapour absorption chiller systems, ammonia is corrosive to copper and in the lithium bromide absorption refrigeration system, lithium bromide is corrosive in nature. In that sense they reduce the overall life of the system.

The working pressures of the absorption refrigeration cycle are very low. In case of the lithium bromide system these pressures are so low that even the expansion valve is not required since the drop in pressure of the refrigerant due to its flow which is good enough to produce its expansion. Due to this reason, the refrigeration system should be sealed thoroughly so that no atmospheric gases would enter the refrigeration system. The compression refrigeration systems are also packed tightly in order to prevent the leakage of the refrigerant to the atmosphere.

The vacuum leaks of the absorption chillers are serious problems which adversely affect the efficiency of the machine causing corrosion in the unit. Due this reason, the possible points which could admit air such as valves must be repaired and maintained as necessary. In addition to these, absorption chillers are relatively trouble free, partly due to the fact that there are less moving parts and also less vibration [24].

Though the running cost of the absorption refrigeration system is much lesser than the vapour compression system, its initial capital cost is much higher. Therefore, initial high investment demotivates the buyers due to their financial difficulties [25].

3.4.5 Operating log with parameters

These help the operator to recognize both normal and abnormal machine conditions. In addition, they contribute to plan the preventive maintenance schedule and diagnose machine problems. A typical operating log sheet with parameters is shown by Figure 3.7.

Re.	Items	Unit	Date : / /		
1.	Time	:	:	:	:
2.	Chilled water flow rate	m^3/hr			
3.	Chilled water inlet & outlet temp.	°C (/)	/	/	/
4.	Chilled water inlet & outlet pressure	kg/cm^{2}g	/	/	/
5.	Hot / Cooling water flow rate	m^3/hr			
6.	Hot / Cooling water inlet & outlet temp.	°C (/)	/	/	/
7.	Hot / Cooling water inlet & outlet pressure	kg/cm^{2}g	/	/	/
8.	Generator temp.	°C			
9.	Generator vapour temp.	°C			
10.	Generator fluid level	O			
11.	Absorber fluid level	O			
12.	Evaporator fluid level	O			
13.	Steam pressure before SCV	kg/cm^{2}g			
14.	Steam pressure in HTG header	kg/cm^{2}g			
15.	Steam control valve opening	%			
16.	Pressure in storage tank	mmHg			
17.	Pressure in shell	mmHg			
18.	Attained vacuum of purge pump	mmHg			
19.	Steam Consumption	kg/hr			

Figure 3.7: Record readings in accordance with the operating log at frequent intervals [26]

3.5 Waste heat sources of diesel engine

The recovered amount of heat will depend on the investment for the heat recovery equipment. It is obvious that total energy concept can result in overall plant thermal efficiency. The following methods can be utilized for heat recovery. The heat of the exhaust gases and cooling water is not wasted to the environment and it is fed to waste heat boilers to supply heating systems, hot water consumers, drying or cooling systems, as discussed below.

1. Heat in the jacket water can be recovered in a heat exchanger to obtain hot water.
2. Hot water can also be obtained by using the heat both in the jacket water and exhaust gases. (The jacket water heat exchanger acts as a pre heater).
3. Low pressure steam can also be generated by utilizing the heat both in the jacket water and exhaust gases. The jacket water heat exchanger acts as feed water heater and exhaust gas heat exchanger serving as exhaust heat boiler.
4. Hot air can be obtained by enclosing the engine exhaust silencer and forcing the air through the enclosure by means of a fan. The obtained hot air can be used for heating the buildings.

The recovered heat can be used for processes and air conditionings.

3.5.1 Quantifying the waste heat

The amount of waste heat in the diesel engine could be calculated by using the well-known Q= mCT equation, where m is the mass of the sample, T is the temperature (measured in degrees Kelvin), and C is known as the specific heat capacity. Specific heat capacity is a parameter which contains all the information about the material and phase of the sample.

3.5.2 Measuring the waste heat

Temperatures of the inlet and outlet of the cooling water system and their flow rates should be measured. The temperature of the exhaust gas after turbocharger should also be measured with the flow rates. Ambient temperature and engine load is also very important to measure in order to evaluate the behavior of the parameters.

3.6 Absorption chiller system applications with waste heat in diesel engines

It is clear that absorption refrigeration systems have received a considerable interest among the research society during last years and a substantial amount of work has been published on the subject. Even though several investigations have been devoted to vapour absorption refrigeration systems, only few past studies have addressed the concept of driving absorption refrigeration systems by waste heat from internal combustion engines. The way which pre cooling and inter cooling configurations of combustion air affect the engine performance is mentioned below.

“The theoretical performance of four different configurations of a turbocharged diesel engine and absorption refrigeration unit combination when operating in a high ambient day temperatures of 35-40 °C. The results were compared” [27]

Table 3.1: Performance of the engine at 35°C ambient temperature for different configurations [27]

Engine Configuration	No charge air cooling	With pre cooler only	With inter cooler	With pre and inter cooler
Engine speed [RPM]	2000	2000	2000	2000
Inlet manifold temp. [K]	434	414	334	329
Power [kW]	154.8	159.6	165	167.4
Exhaust temp. [K]	1000	951	890	860
BSFC [gm/kW hr]	211.4	207.4	200.7	198.2
Air entering system [kg/s]	0.09135	0.09607	0.09834	0.1023
Air leaving system [kg/s]	0.0958	0.101	0.103	0.107
Break thermal efficiency [%]	38.51	39.25	40.56	41.0
Energy to exhaust [%]	38.35	38.48	36.49	34.05
Energy to coolant [%]	18.15	17.37	14.96	14.71

The above table clearly shows that when the air cooling is absent the fuel consumption goes high. Inter cooling is more efficient than pre cooling even though the both configurations reduce the Break Specific Fuel Consumption (BSFC). When the combustion air is cooled it gives favorable BSFC while the break thermal efficiency increases.

The Table 3.2 shows cooling potentiality based on engine rating. It implies that when the engine rating increases the cooling potentiality also increases.

Table 3.2: Cooling potentiality based on engine rating [28]

Engine Rating (kW)	Cooling capacity on (Exhaust + Jacket water)/ kW
300	100-110
500	175-200
1000	300-350
1500	425-500
2000	525-600

Indicative and may vary as per engine waste heat parameters

Chapter 04

Developing a Model

Specific fuel consumption and engine deration are addressed under this chapter while developing a model. Mathlab and Mintab softwares were aided to develop the model.

4.1 Prioritize the parameters

By considering all the above discussed parameters, a way of utilizing a waste heat absorption chiller for the diesel engine performance improvement was found.

The engine performance has been defined in ISO standard conditions. So when the site condition is vary from the standard the performance changes.

The ISO standard

Standard Reference conditions:
ISO 3046–1: 2002; ISO 15550: 2002

Air temperature	25 ^{o}C
Air pressure	100 kPa
Relative humidity	30 %
Cooling water temperature before charge air cooler	25^{o}C

The engine and its auxiliary system operating parameters cannot be changed from its optimum ranges because it is worthless for the operation and maintenance of the diesel power plant. The site condition deviation from the standard conditions of a diesel power plant is an unavoidable circumstance since actual power station site conditions are not always standard conditions. As much as actual site conditions are closer to the standard conditions performance of the engine falls closer to the standard condition performances.

4.2 Specific fuel oil consumption

The actual site condition of a diesel power plant could vary from the ISO 3046–1: 2002 standard and the specific fuel oil consumption can be also changed accordingly. Basically ambient temperature, pressure and RH could change the combustion efficiency and also the specific fuel consumption of the engine. The engine expects dry and high dense air with adequate pressure to the combustion chamber. If the ambient temperature increases relative to the ISO 3046-1: 2002 standard ($25^{o}C$) the density of the air will reduce and it will lead to the poor mass of Oxygen for the diesel engine combustion.

The high RH of the site again increases the amount of water vapour in the air and it reduces the oxygen amount in that particular volume of air. Therefore, it leads to poor combustion efficiency. Moreover, water vapour absorbs the combustion heat and it further reduces the combustion temperature. Hence, this leads to higher fuel oil consumption and ultimately increases the specific fuel oil consumption.

When the site ambient conditions different from the standard reference conditions given in clause 5 of ISO 15550:2002 the specific fuel consumption will differ from that declared for the standard reference conditions and shall be recalculated for or from the standard reference conditions.

Following equations can be used if other methods are not declared by the manufacturer.

$$SFC_x = \beta\ SFC_r$$

β -Fuel consumption recalculation factor

SFC_x - Specific Fuel oil consumption at site

SFC_r -Specific Fuel oil consumption at standard reference conditions

Where $$\beta = k/\alpha$$

α -Power adjustment factor

k -Ratio of indicated power

$$\alpha = k - 0.7(1 - k)\left[\frac{1}{\eta m} - 1\right]$$

η_m - Mechanical efficiency

$$k = \left(\frac{px - a\o x\ psx}{pr - a\o r\ psr}\right)^{m} \left(\frac{Tr}{Tx}\right)^{n} \left(\frac{Tcr}{Tcx}\right)^{s}$$

P_x - Site total barometric pressure (kPa).

P_r - Standard reference total barometric pressure, 100 kPa.

a - Factor

$\Phi_X\ P_{SX}$ - Site water vapour pressure (kPa)

$\Phi_r\ P_{Sr}$ - Standard reference water vapour pressure (kPa)

m - Total barometric pressure ratio factor

T_r - Standard reference suction air temperature (K).

Tx - Site suction air temperature (K).

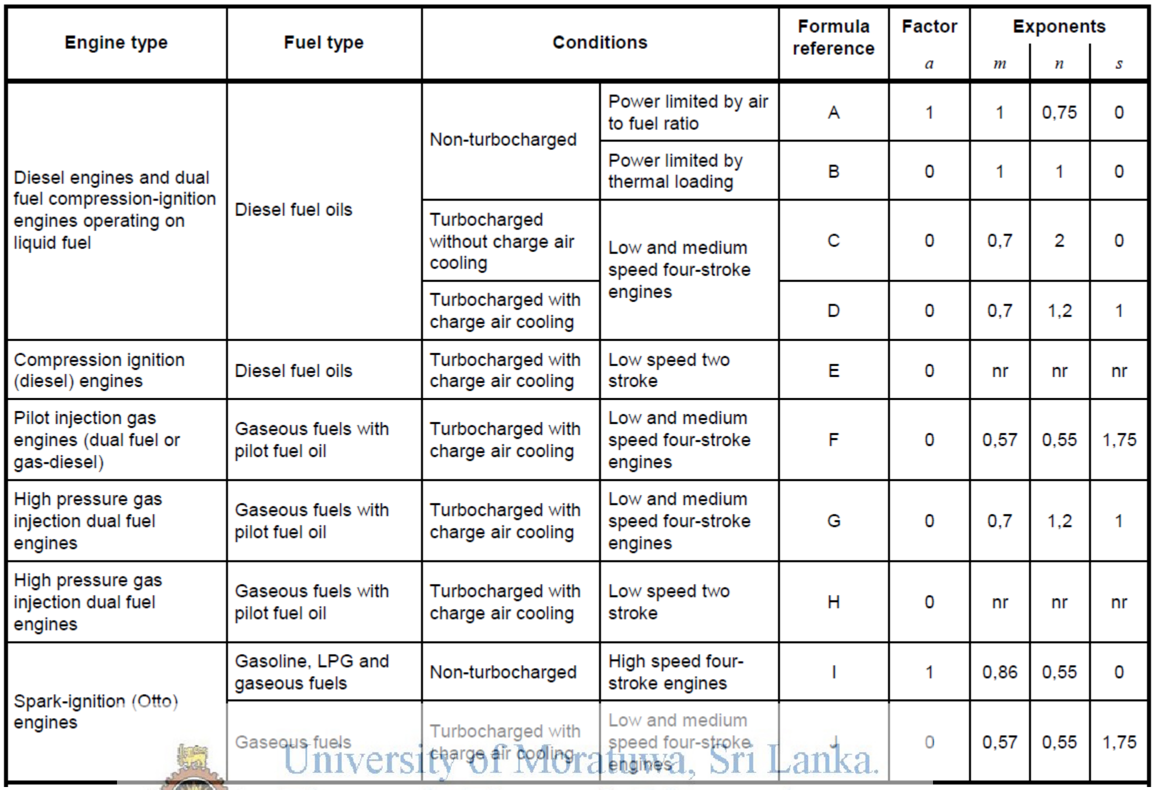
n - Suction air temperature ratio factor

T_{cr} - Standard reference charge air coolant temperature (K).

T_{cx} - Site charge air coolant temperature (K)

s - Charge air coolant temperature ratio factor, 1

Table 4.1: Numerical values for SFC [29]



Engine type	Fuel type	Conditions		Formula reference	Factor	Exponents		
					a	*m*	*n*	*s*
Diesel engines and dual fuel compression-ignition engines operating on liquid fuel	Diesel fuel oils	Non-turbocharged	Power limited by air to fuel ratio	A	1	1	0,75	0
			Power limited by thermal loading	B	0	1	1	0
		Turbocharged without charge air cooling	Low and medium speed four-stroke engines	C	0	0,7	2	0
		Turbocharged with charge air cooling		D	0	0,7	1,2	1
Compression ignition (diesel) engines	Diesel fuel oils	Turbocharged with charge air cooling	Low speed two stroke	E	0	nr	nr	nr
Pilot injection gas engines (dual fuel or gas-diesel)	Gaseous fuels with pilot fuel oil	Turbocharged with charge air cooling	Low and medium speed four-stroke engines	F	0	0,57	0,55	1,75
High pressure gas injection dual fuel engines	Gaseous fuels with pilot fuel oil	Turbocharged with charge air cooling	Low and medium speed four-stroke engines	G	0	0,7	1,2	1
High pressure gas injection dual fuel engines	Gaseous fuels with pilot fuel oil	Turbocharged with charge air cooling	Low speed two stroke	H	0	nr	nr	nr
Spark-ignition (Otto) engines	Gasoline, LPG and gaseous fuels	Non-turbocharged	High speed four-stroke engines	I	1	0,86	0,55	0
	Gaseous fuels	Turbocharged with charge air cooling	Low and medium speed four-stroke engines	J	0	0,57	0,55	1,75

4.3 Engine deration

This describes about the engine rating to be checked for operation under standard conditions of ambient temperature and humidity. If conditions at site differ from the prescribed standard condition with respect to any of these factors, the engine will not be able to carry its full load as previously determined. Then it is needed to calculate deration allowance.

4.3.1 Ambient temperature

Deration for high ambient temperature is obvious as warm air has less density than the cold air. Therefore, high temperature ambient contains low weight of Oxygen which is required for the combustion in the engine cylinder.

British practice states that deration is not necessary for ambient temperatures below 85°F(29.5°C), but the corresponding figure for American standards is usually 90°F (32°C). For higher temperatures, the deration percentage of between 2 and 3% must be allowed for every 10°F above the datum [30].

$$\text{Thus,} \quad Y = 0.2\,T - 17$$

Y - Deration percentage due to ambient temperature variations

T - Temperature in °F

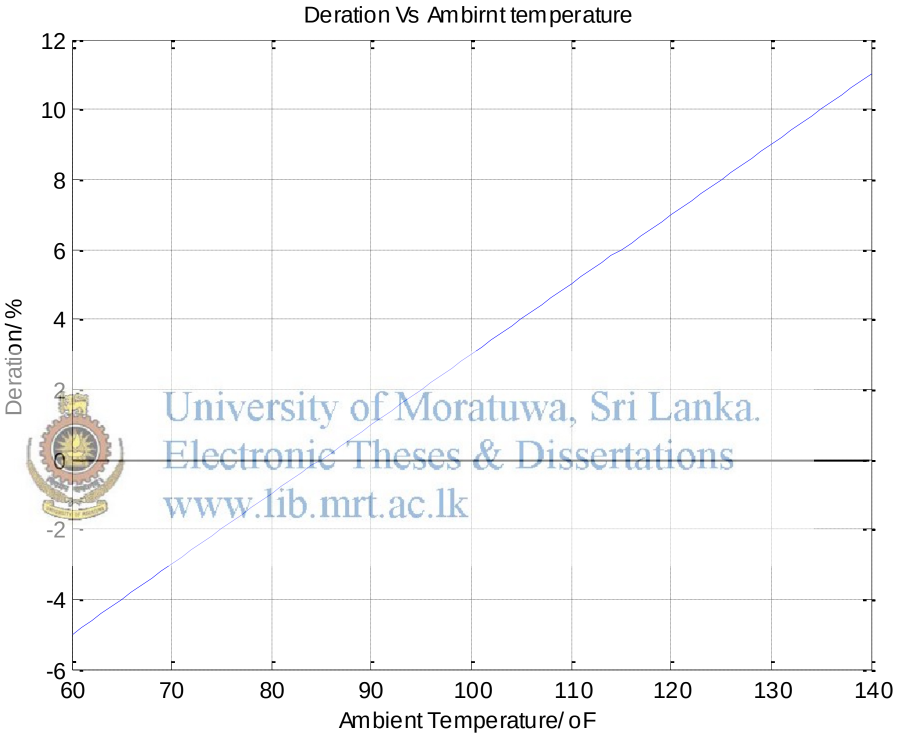


Figure 4.1: Graphical interpretation of Percentage of deration Vs ambient temperature

4.3.2 Altitude

The reason for deration being necessary for high altitude operation is similar to that for high ambient temperature. This means the air is less dense and therefore less Oxygen is available in the engine cylinder to support combustion.

The British standard requires 500 feet above sea level to start deration while the American ratings are usually more generous in this respect. The Diesel Engine Manufactures Association (DEMA) standard requires deration to start at 1500 feet above sea level. Hence above the datum line deration is at the rate of approximately 3.5 % per 1000 feet for normally aspirated engines and approximately 2.5 % per 1,000 feet for turbocharged engines. The diesel power stations in Sri Lanka are well below the 500 feet sea level and so altitude deration allowance is not applicable [30].

$$\text{Thus, } Y = \frac{5}{2000}X - \frac{5}{4}$$

Y - Deration percentage due to altitude

T - Altitude in feet

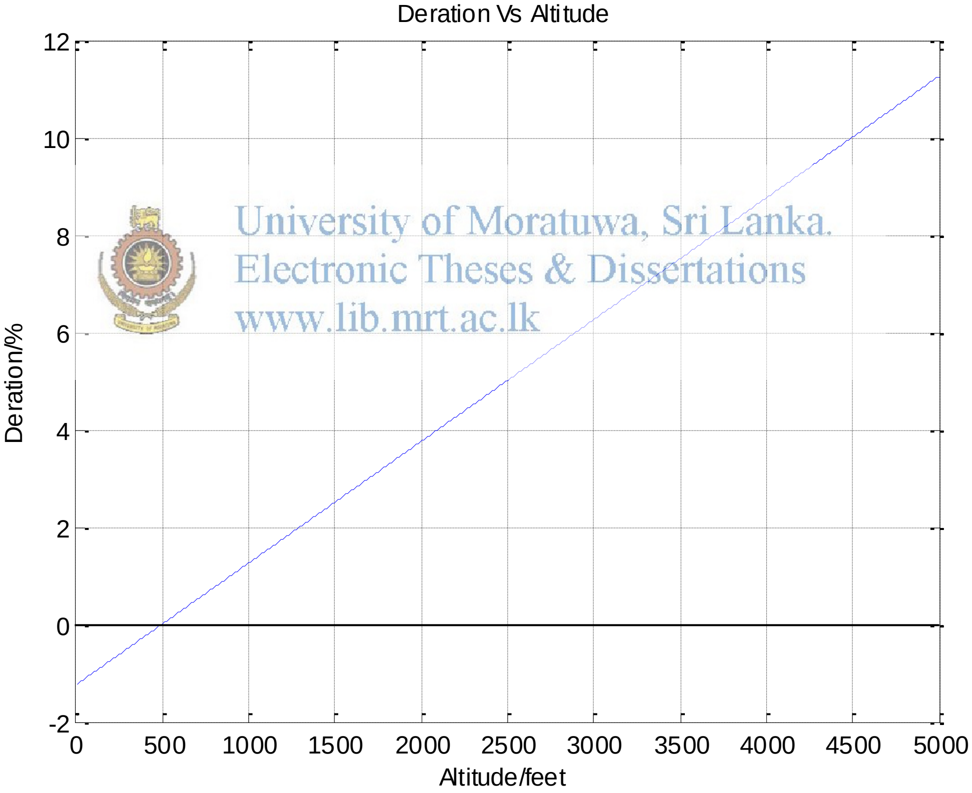


Figure 4.2: Graphical interpretation of percentage of deration Vs altitude

4.3.3 Humidity

The humidity of atmosphere is a measure of the percentage of water vapour which is held in suspension in the air. When the water vapour occupies a certain space in the humid air, an equal volume of air is displaced. Water vapour is useless for combustion purpose as it lessens the amount of Oxygen. Hence, a deration percentage must be allowed to compensate for the Oxygen lost in the displaced air.

Table 4.2: Percentage of deration Vs percentage of humidity at constant atmospheric temperatures [30]

Atmospheric Temperature	Percentage Humidity									
	10	20	30	40	50	60	70	80	90	100
	Percentage Deration									
85						0.5	1.0	1.5	2.0	2.4
90					0.4	1.0	1.6	2.2	2.7	3.3
95				0.2	0.9	1.6	2.2	2.9	3.6	4.2
100				0.7	1.5	2.2	3.0	3.8	4.6	5.3
105			0.3	1.2	2.1	3.0	3.9	4.8	5.7	6.6
110			0.7	1.8	2.8	3.8	4.9	5.9	6.9	8.0
115			1.2	2.4	3.6	4.8	6.0	7.2	8.4	9.6
120		0.4	1.7	3.1	4.5	5.9	7.3	8.6	10.0	11.4
125		0.8	2.3	3.9	5.5	7.1	8.7	10.2	11.8	13.4

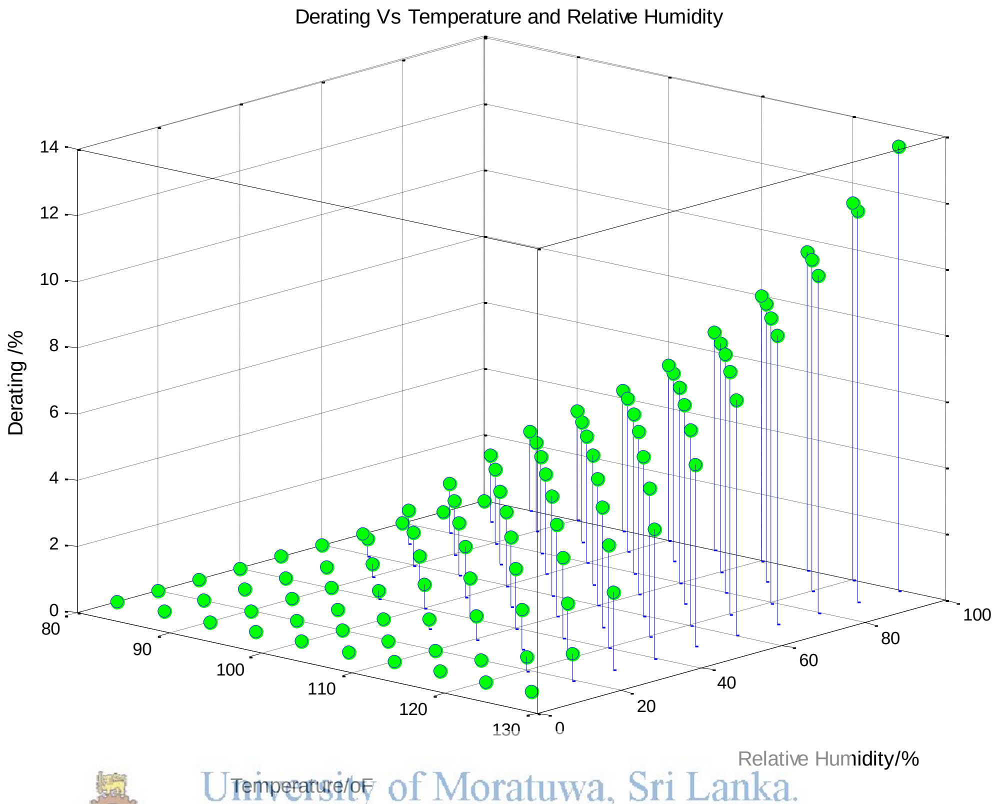


Figure 4.3: Graphical interpretation of percentage of deration Vs percentage of humidity at constant atmospheric temperatures in Table 4.2 (Refer Appendix A)

- **Model for deration**

3^{rd} order regression equations for Percentage Deration (Y) Vs Percentage Humidity were deduced at constant temperatures by the values in the Table 4.2.

According to that, when the temperature is constant at 85^{o}F,

Deration Percentage (Y) Vs Relative Humidity (X) is given below with the aid of Min Tab software and Mathlab software.

$Y= -4.6296 \times 10^{-6} X^3 + 0.00095238 X^2 -0.013823X -1.1079$ ----> T85

Regression Analysis is S = 0.0140859, R^2 = 100.0%, R^2 (adj) = 100.0%, P= 0.000

Generally, coefficient of determination (R^2) should be high ($R^2 > 80\%$). As R^2 in above T85 equation is 100.0% we can conclude that the fitted model explains 100% of the observed variability. The P value (significant) should be less than 0.05 for a better model. Since the P value is 0.000 in the fitted model (T85 equation) it can be used for forecasting as well.

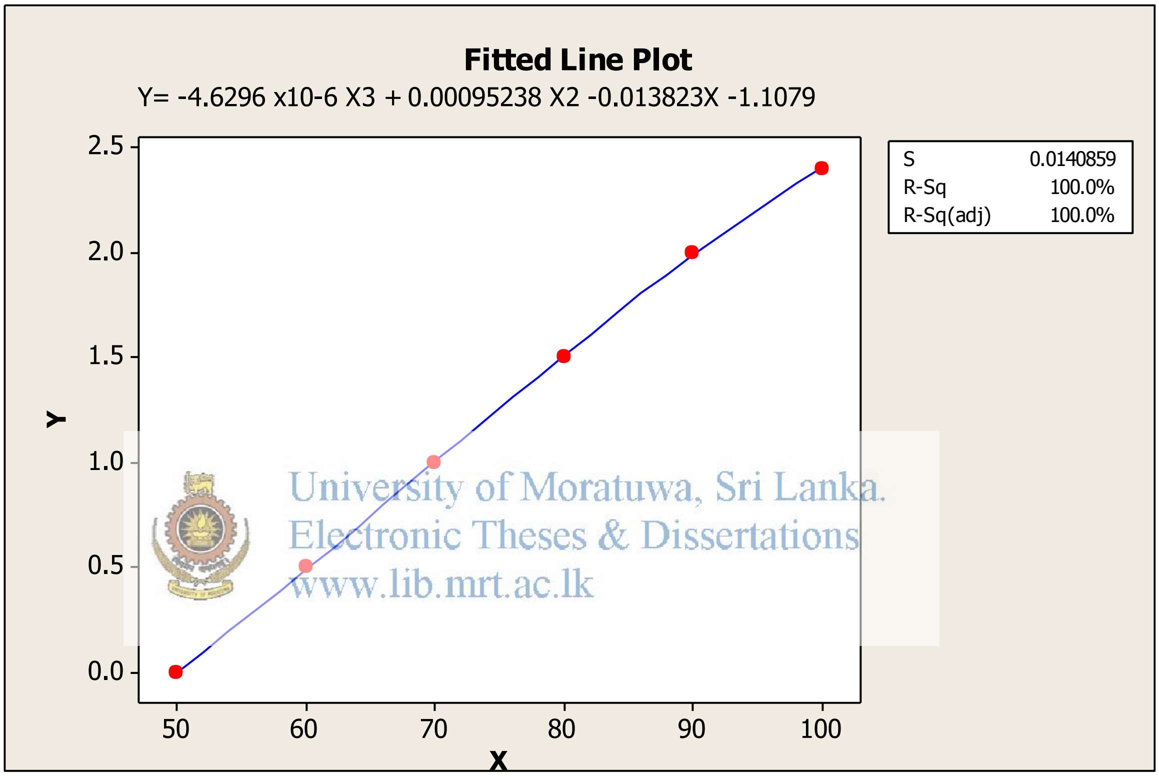


Figure 4.4: Fitted line plot of T85 equation

The above calculations for constant temperatures at 90°F, 95 °F, 100 °F, 105 °F, 110 °F, 115 °F, 120 °F, 125 °F are mentioned in Appendix B.

The critical value of the RH (the Percentage humidity which gives zero percentage deration) at the particular (constant) ambient temperature is the point where the deration is zero. It can be taken by having the solutions of above 3rd order polynomials as calculated below.

$0= -4.6296 \times 10^{-6} X^3 + 0.00095238 X^2 -0.013823X -1.1079$ ----> **T85**

Solutions are X1=178.9987, X2= -26.3024, **X3=50.1103**

Three numbers of solutions could be taken from the above 3rd order polynomial. The solution means the percentage of RH at which the deration is zero. By having the solution of T85, 3rd order polynomial, it is obvious that at 85°F of ambient temperature the zero percentage deration can be obtained when the percentage of RH is at 50.1103% or below.

$0=-5.5556 \times 10^{-6} X^3+0.0012262 X^2- 0.03004X- 0.42381$ ----> **T90**

Solutions are X1=190.1719, X2= -9.9151, **X3=40.4574**

$0=-8.5859 \times 10^{-6} X^3+0.0018409X^2-0.059455X+0.30714$ ----> **T95**

Solutions are X1=176.2779, X2= 6.3941, **X3=31.7377**

$0=-2.2727 \times 10^{-6} X^3+0.00047294X^2+0.045974X-1.7405$ ----> **T100**

Solutions are X1=272.0964, X2= -93.9563, **X3=29.9560**

$0=-7.0707 \times 10^{-6} X^3+0.0014545X^2-0.0038384X-0.53333$ ----> **T105**

Solutions are X1=201.1449, X2= -17.2171, **X3=21.7803**

$0=-3.0303 \times 10^{-6} X^3+0.00063961X^2+0.06105X-1.5071$ ----> **T110**

Solutions are X1=27.2639, X2= -86.8467, **X3=20.6543**

$0=-5.0564 \times 10^{-21} X^3+1.3582 \times 10^{-18} X^2+0.12X-2.4$ ----> **T115**

Solutions are X1=4871580942, X2= -4871580693, **X3=19.9978**

$0=-8.5664 \times 10^{-6} X^3+0.0016445X^2+0.041696X-0.74333$ ----> **T120**

Solutions are X1=212.9173, X2= -33.2159, **X3=12.2695**

$0=-6.9347x10^{-6}\ X^3+0.0013298X^2+0.080204X-1.0633$ ----> **T125**

Solutions are X1=237.7024, X2= -57.2161, **X3=11.2740**

From the solutions of above 3rd order polynomials, the RH values more than 100% and minus were rejected since those values do not practically exist.

Table 4.3: Relevant RH and constant temperature of the particular polynomial for zero percentage deration

No.	**Relative Humidity (RH)/ %**	**Constant Ambient Temperature (T)/ °F**
T85	50.1103	85
T90	40.4574	90
T95	31.7377	95
T100	29.9560	100
T105	21.7803	105
T110	20.6543	110
T115	19.9978	115
T120	12.2695	120
T125	11.2740	125

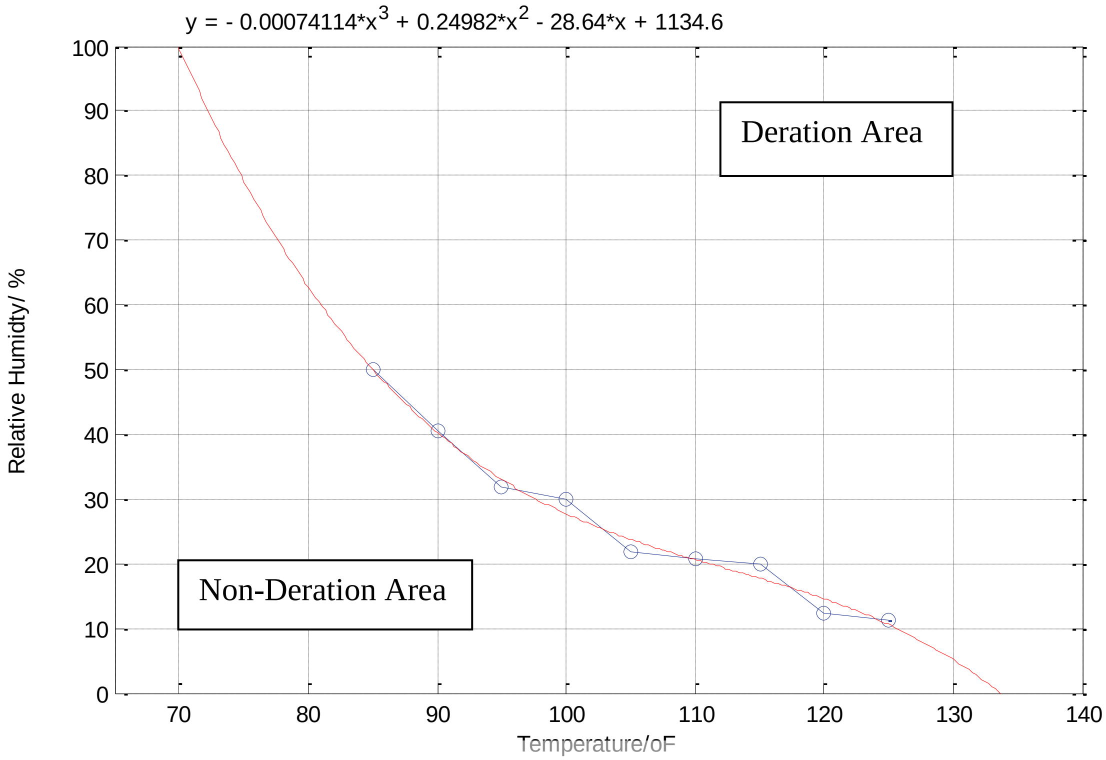


Figure 4.5: Relative humidity Vs temperature

Polynomial Regression Analysis: RH (Y) versus Temperature (X)

The regression equation is

$Y = 1134.6 - 28.64X + 0.24982\ X^2 - 0.000741\ X^3$ **----> (M)**

S = 2.07746 R^2 = 98.4% R^2 (adj) = 97.4%, P=0.000

As R-sq is 98.4% it can be concluded that the fitted model explains 98.4 % of the observed variability. The statistic R^2 indicates the percentage of variability of the dependent variability explained by the fitted model. The P value (significant) should be less than 0.05 for a better model. Since the P value in the fitted model (equation M) is 0.000, it can also be used for forecasting.

Figure 4.5 clearly shows that when the ambient is hot and wet the engine derates while the dry and cool ambient does not derate the diesel engine.

In the above Figure 4.5, it is obvious that when the temperature is 70^{o}F the RH percentage is 100%. This means that when the temperature goes down further from 70^{o}F the engine does not derate due to the humidity factor (deration percentage is zero) even though the RH percentage reaches 100%. This can be concluded by the below mentioned 3rd order polynomial of Deration (Y) Vs RH (X).

$Y = 0\,X^3 + 0\,X^2 - 0X - 0$ ----> T70

4.4 Summary

4.4.1 Specific fuel oil consumption at the site conditions

$$SFC_x = \beta\ SFC_r$$

β - Fuel consumption recalculation factor

SFC_x - Specific Fuel oil consumption at site

SFC_r - Specific Fuel oil consumption at standard reference conditions

Where $\beta = k/\alpha$

α - Power adjustment factor

k - Ratio of indicated power

$$\alpha = k - 0.7(1-k)\ [\frac{1}{\eta m} - 1]$$

η_m - Mechanical efficiency

$$k = \left(\frac{px - a\phi x\ psx}{pr - a\phi r\ psr}\right)^m \left(\frac{Tr}{Tx}\right)^n \left(\frac{Tcr}{Tcx}\right)^s$$

η_m - Mechanical efficiency

P_x - Site total barometric pressure (kPa).

P_r - Standard reference total barometric pressure, 100 kPa.

a - Factor,0

$\Phi_x\ P_{sx}$ - Site water vapour pressure (kPa)

$\Phi_r\ P_{Sr}$ - Standard reference water vapour pressure (kPa)

m - Total barometric pressure ratio factor

T_r - Standard reference suction air temperature (K)

Tx - Site suction air temperature (K)

n - Suction air temperature ratio factor

T_{cr} - Standard reference charge air coolant temperature (K)

T_{cx} - Site charge air coolant temperature (K)

s - Charge air coolant temperature ratio factor, 1

4.4.2 Deration percentage calculation

- **Deration percentage due to ambient temperature**

$$Y = 0.2\,T - 17$$

Y - Deration Percentage Due to Ambient temperature variations

T - Temperature in ^{o}F

- **Deration percentage due to altitude**

$$Y = \frac{5}{2000}\,X - \frac{5}{4}$$

Y - Deration Percentage Due to Altitude

X - Altitude in feet

- **Deration percentage due to relative humidity**

By utilizing below mentioned 3rd order polynomials, the deration percentage of a diesel engine for the site conditions can be calculated.

$Y= 0\,X^3 + 0\,X^2 - 0X - 0$ ----> T70

$Y= -4.6296\times10^{-6}\,X^3 + 0.00095238\,X^2 - 0.013823X - 1.1079$ ----> T85

$Y= -5.5556\times10^{-6}\,X^3 + 0.0012262\,X^2 - 0.03004X - 0.42381$ ----> T90

$Y= -8.5859\times10^{-6}\,X^3 + 0.0018409X^2 - 0.059455X + 0.30714$ ----> T95

$Y= -2.2727\times10^{-6}\,X^3 + 0.00047294X^2 + 0.045974X - 1.7405$ ----> T100

$Y= -7.0707\times10^{-6}\,X^3 + 0.0014545X^2 - 0.0038384X - 0.53333$ ----> T105

$Y= -3.0303\times10^{-6}\,X^3 + 0.00063961X^2 + 0.06105X - 1.5071$ ----> T110

$Y= -5.0564\times10^{-21}\,X^3 + 1.3582\times10^{-18}\,X^2 + 0.12X - 2.4$ ----> T115

$Y= -8.5664\times10^{-6}\,X^3 + 0.0016445X^2 + 0.041696X - 0.74333$ ----> T120

$Y= -6.9347\times10^{-6}\,X^3 + 0.0013298X^2 + 0.080204X - 1.0633$ ----> T125

By interpolating the constants of above corresponding 3rd order polynomials, the required 3rd order polynomial for a given temperature can be calculated.

Total Deration = Deration due to the effects of (Ambient Temperature + Altitude + Relative Humidity)

4.4.3 Flow chart of the performance evaluation

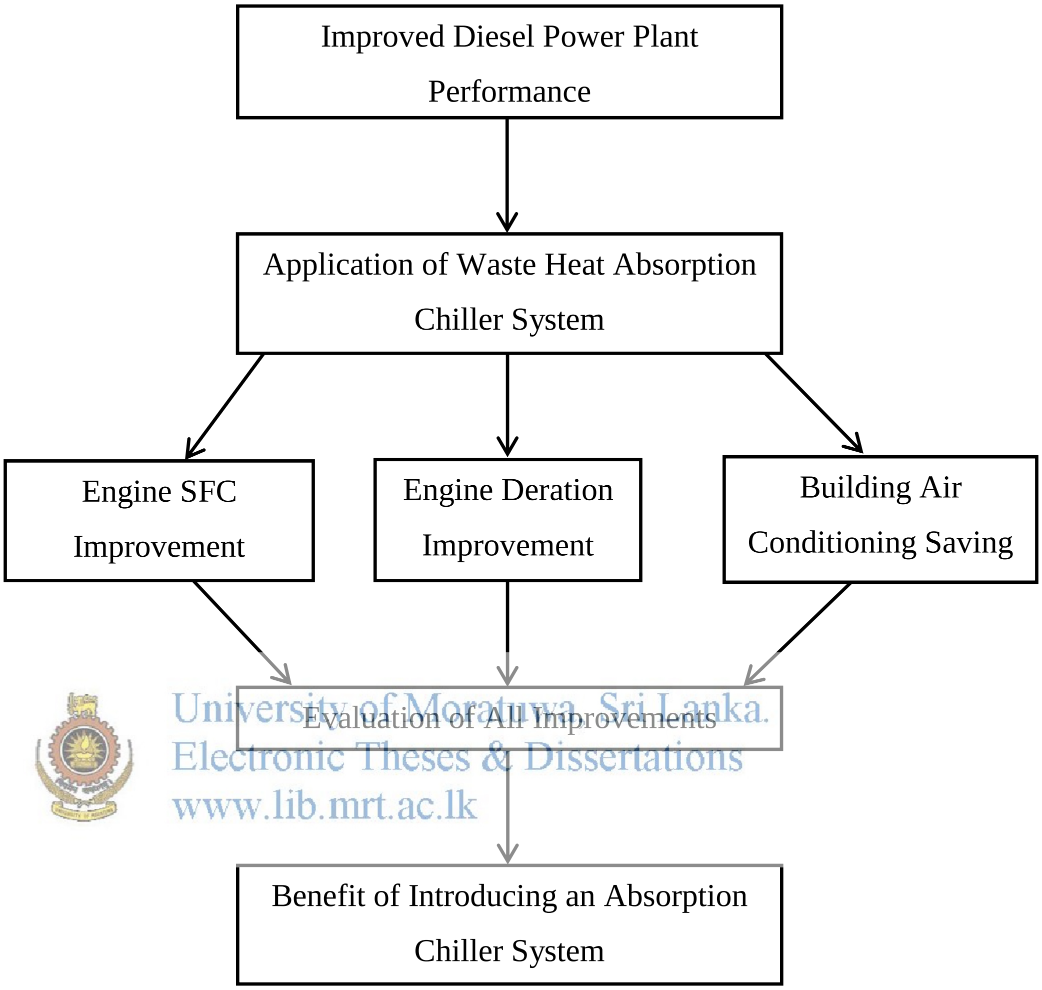


Figure 4.6: Flow chart of the performance evaluation

4.5 Selection of vapour absorption chiller

It is suitable to use lithium bromide and water absorption system on the basis of the non toxicity of lithium bromide and water, low volatility and absence of requirement of an extra component as well as lower installation maintenance and operation cost relative to aqua ammonia system.

It is obvious that single effect absorption chillers have less COP than the double effect absorption chillers. Therefore, it is worthy to use double effect lithium bromide and water absorption chiller to improve the diesel power plant performance. Moreover, in the double effect absorption chillers the quality of the waste heat source has to be higher than the single effect absorption chillers. In that sense if the quality of the waste heat available in the diesel power plant does not match the required quality for the chiller, it needs to improve the quality of the waste heat to cater the double effect absorption chiller. Otherwise, it is necessary to step back to the single effect lithium bromide and water absorption chiller.

Chapter 05

Case Study

In this chapter the developed model has been applied to a practical situation. The Uthuru Janani Power Station (UJPS) located in Jaffna peninsula was selected for the case study.

5.1 Uthuru Janani Power Station (UJPS)

UJPS at Chunnakum (24MW) consists of three diesel engines. This heavy fuel operated power plant was installed in Jaffna and commercial operation of the plant commenced on 01 January2013. The Uthuru Janani power plant has the capacity of generating 176 million units of electricity annually, saving the CEB Rs. 1,350 million [31].

5.2 Data collection

The hourly temperature readings and RH values were obtained during the period of 0.00hrs on 12.09.2014 to 24.00hrs on 13.09.2014 and they are tabulated in Table 5.1.

Table 5.1: Site RH and temperature readings at UJPS (0.00hrs on 12.09.2014 to 24.00hrs on 13.09.2014)

Time (Hrs)	0.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00
Temperature/ºC	27.9	27.7	27.6	27.4	27.4	27.3	27.5	29.2	30.3	32.2
RH %	71.4	79.8	81.1	82.1	80.1	82.1	81.2	74.6	70.4	62.6

Time (Hrs)	10.00	11.00	12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00
Temperature/ºC	34.3	34.3	34.5	36.0	35.0	34.1	32.7	30.9	29.6	29.2	28.8	28.4	28.2
RH %	55.6	55.6	54.5	45.5	47.8	51.7	60.8	66.4	73.3	70.9	73.7	75.4	76.7

Time (Hrs)	23.00	00.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00	11.00
Temperature/ºC	27.9	27.8	27.5	27.5	27.4	27.4	27.4	27.5	29.1	31.8	31.9	32.2	32.5
RH %	78.5	81.0	81.6	81.6	82.1	81.2	82.9	83.2	76.2	65.3	65.2	60.7	59.4

Time (Hrs)	12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00	23.00
Temperature/ºC	32.8	32.8	34.4	34.4	32.6	31.3	29.7	28.9	28.6	28.3	28.1	28.0
RH %	52.6	51.0	45.5	48.1	60.3	67.2	70.4	73.3	76.5	78.2	78.8	79.6

5.2.1 Instrument details

Instrument : Humidity and temperature transmitter HMT 337

Order Code : HMT330 7S1D011XCAB100A0EAAB1A1

Serial number : G4910021

Manufacturer : Vaisala Oyj. Finland

Calibration Date : 7th December 2011

Temperature measurement

Range: -70 ... + 180 ºC (-94 ... +356 ºF)

Typical accuracy: 0.1 ºC (0.18 ºF)

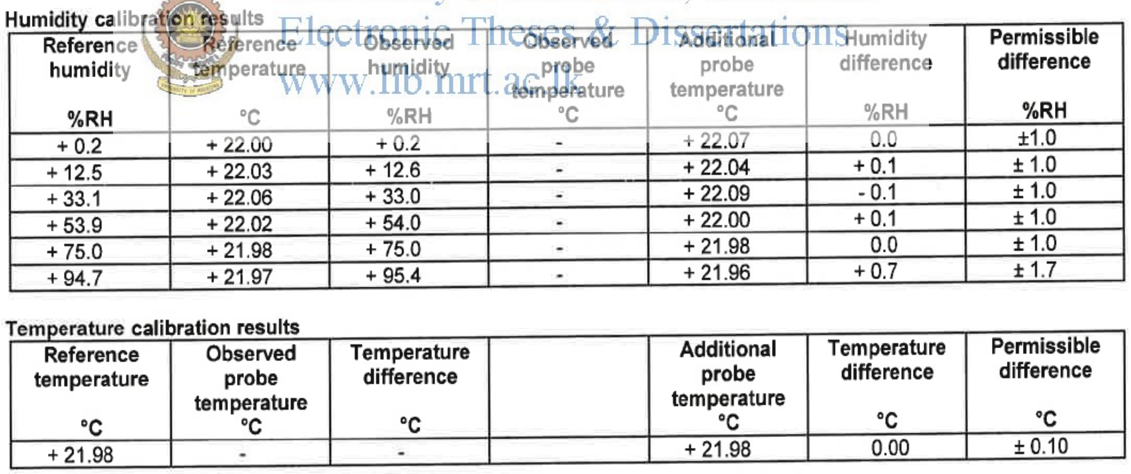
Sensor: Pt100 PRT DIN IEC 751 class 1/4 B

Cable length: 2 m, 5 m, and 10 m

Pressure tight: up to 7 bar

Probe material: stainless steel

Table 5.2: Humidity and temperature instrument calibration results



Humidity calibration results

Reference humidity %RH	Reference temperature °C	Observed humidity %RH	Observed probe temperature °C	Additional probe temperature °C	Humidity difference %RH	Permissible difference %RH
+ 0.2	+ 22.00	+ 0.2	-	+ 22.07	0.0	±1.0
+ 12.5	+ 22.03	+ 12.6	-	+ 22.04	+ 0.1	± 1.0
+ 33.1	+ 22.06	+ 33.0	-	+ 22.09	- 0.1	± 1.0
+ 53.9	+ 22.02	+ 54.0	-	+ 22.00	+ 0.1	± 1.0
+ 75.0	+ 21.98	+ 75.0	-	+ 21.98	0.0	± 1.0
+ 94.7	+ 21.97	+ 95.4	-	+ 21.96	+ 0.7	± 1.7

Temperature calibration results

Reference temperature °C	Observed probe temperature °C	Temperature difference °C		Additional probe temperature °C	Temperature difference °C	Permissible difference °C
+ 21.98	-	-		+ 21.98	0.00	± 0.10

5.3 Evaluating the SFC and fuel oil saving

The engine full load is 8.924MW at the ISO standard site conditions and the specific fuel oil consumption is 202.6 g/kWh.

In order to achieve the improved performance of the diesel engine the ambient temperature needs to be brought to 25°C and RH to 30%. By making the account of instrument permissible difference -1% of RH and -0.1°C of temperature (refer Table 5.2) the Table 5.3 was produced.

Table 5.3: Site humidity and temperature readings at UJPS adjusted for -1% of RH permissible difference and -0.1°C of temperature permissible difference of the instrument (0.00hrs on 12.09.2014 to 24.00hrs on 13.09.2014)

Time (Hrs)	0.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00
Temperature/°C	27.8	27.6	27.5	27.3	27.3	27.2	27.4	29.1	30.2	32.1
RH %	70.4	78.8	80.1	81.1	79.1	81.1	80.2	73.6	69.4	61.6

Time (Hrs)	10.00	11.00	12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00
Temperature/°C	34.2	34.2	34.4	35.9	34.9	34.0	32.6	30.8	29.5	29.1	28.7	28.3	28.1
RH %	54.6	54.6	53.5	44.5	46.8	50.7	59.8	65.4	72.3	69.9	72.7	74.4	75.7

Time (Hrs)	23.00	00.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00	11.00
Temperature/°C	27.8	27.7	27.4	27.4	27.3	27.2	27.3	27.4	29.0	31.8	31.8	32.1	32.4
RH %	77.5	80.0	80.6	80.6	81.1	80.2	81.9	82.2	75.2	64.3	64.2	59.7	58.4

Time (Hrs)	12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00	23.00
Temperature/°C	32.7	32.7	34.3	34.3	32.5	31.2	29.6	28.8	28.5	28.2	28.0	27.9
RH %	51.6	50.0	44.5	47.1	59.3	66.2	69.4	72.3	75.5	77.2	77.8	78.6

The below calculation has been done for 0.00 hrs on 12.09.2014 reading of Table 5.3

$$SFC_x = \beta\, SFC_r \qquad \text{---->(A)}$$

Where, $\beta = k/\alpha$

$$\alpha = k - 0.7(1-k)\left[\frac{1}{\eta m} - 1\right]$$

$$k = \left(\frac{px - a\o x\, psx}{pr - a\o r\, psr}\right)^m \left(\frac{Tr}{Tx}\right)^n \left(\frac{Tcr}{Tcx}\right)^s$$

Px = 100 kPa

Pr = 100 kPa

a = 0 according to the given numerical value table in the ISO 3046-1:2002 standard.

m = 0.7 according to the given numerical value table in the ISO 3046-1:2002 standard.

n = 1.2 according to the given numerical value table in the ISO 3046-1:2002 standard.

s = 1 according to the given numerical value table in the ISO 3046-1:2002 standard.

The above constants are for low and medium speed four strokes turbocharged compression ignition liquid diesel oil engines with charge air cooling. (Refer Table 4.1: Numerical values for SFC, Formula reference D in the table)

Tr = 298 K

Tx = 273+27.8 K

Tcr= 298 K

Tcx= 273+47 K (Charge air inlet cooling water is 47^{o}C)

$$k = \left(\frac{100 - 0}{100 - 0}\right)^{0.7} \left(\frac{298}{300.8}\right)^{1.2} \left(\frac{298}{320}\right)^{1}$$

k = 1 x 0.9888 x 0.9313

k = 0.9208

$$\alpha = 0.9208 - 0.7(1 - 0.9208)\left[\frac{1}{0.8} - 1\right]$$

α = 0.9069

β = k/ α

β = 0.9208/0.9069

β = 1.0153

According to the site conditions, the specific fuel oil consumption can be recalculated by Equation A.

SFC_{site} = β x SFC_{r} ---->(A)

= 1.0153 x 202.6 g/kWh

SFC_{site} = 205.69 g/kWh

If the standard conditions are achieved the anticipated fuel saving of that particular hour is calculated below (At the ISO standard conditions the engine load is 8.924 MW and SFC is 202.6 g/kWh in UJPS),

Numbers of kWh produced per hour = 8,924x 1 kWh
= 8,924 kWh
Saving from 8,924 kWh = 8,924 x (205.69-202.60) g
= 27,575.16 g
From all three engines = 24,320.00 x 3 g
= 82,725.5 g
= 82.73 kg

The calculated SFC values for particular ambient temperatures are given below.

Table 5.4: Relevant SFC and hourly fuel oil saving

Time (Hrs)	0.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00
Temperature/°C	27.8	27.6	27.5	27.3	27.3	27.2	27.4	29.1	30.2	32.1
RH %	70.4	78.8	80.1	81.1	79.1	81.1	80.2	73.6	69.4	61.6
Calculated SFC /g	205.69	205.66	205.65	205.61	205.61	205.60	205.63	205.90	206.07	206.38
Fuel saving/kg	82.73	81.92	81.65	80.58	80.58	80.32	81.12	88.35	92.90	101.20

Time (Hrs)	10.00	11.00	12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00
Temperature/°C	34.2	34.2	34.4	35.9	34.9	34.0	32.6	30.8	29.5	29.1	28.7	28.3	28.1
RH %	54.6	54.6	53.5	44.5	46.8	50.7	59.8	65.4	72.3	69.9	72.7	74.4	75.7
Calculated SFC /g	206.71	206.71	206.75	206.99	206.83	206.68	206.46	206.17	206.96	205.90	205.84	205.77	205.74
Fuel saving/kg	110.03	110.03	111.10	117.53	113.25	109.23	103.34	95.58	116.73	88.35	86.74	84.87	84.06

Time (Hrs)	23.00	00.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00	11.00
Temperature/°C	27.8	27.7	27.4	27.4	27.3	27.2	27.3	27.4	29.0	31.8	31.8	32.1	32.4
RH %	77.5	80.0	80.6	80.6	81.1	80.2	81.9	82.2	75.2	64.3	64.2	59.7	58.4
Calculated SFC /g	205.69	205.68	205.63	205.63	205.61	205.60	205.61	205.63	205.88	206.33	206.33	206.38	206.43
Fuel saving/kg	82.73	82.46	81.12	81.12	80.58	80.32	80.58	81.12	87.81	99.86	99.86	101.20	102.54

Time (Hrs)	12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00	23.00
Temperature/°C	32.7	32.7	34.3	34.3	32.5	31.2	29.6	28.8	28.5	28.2	28.0	27.9
RH %	51.6	50.0	44.5	47.1	59.3	66.2	69.4	72.3	75.5	77.2	77.8	78.6
Calculated SFC /g	206.47	206.47	206.73	206.73	206.44	206.23	205.98	205.85	205.80	205.76	205.73	207.71
Fuel saving/kg	103.61	103.61	110.57	110.57	102.80	97.18	90.49	87.01	85.67	84.60	83.80	83.26

Total saving from 0.00hrs on 12.09.2014 to 24.00hrs on 13.09.2014 is equal to the total of the hourly fuel oil saving (total saving per two days).

Total fuel saving (per two day) = 4,466.66 kg

Total fuel saving (per day) = 2,233.33 kg/day

By accounting the instrument permissible difference for +1% of RH and +0.1^{o}C of temperature of the Table 5.1, the Table 5.5 was produced.

Table 5.5: Site humidity and temperature readings at UJPS adjusted for +1% of RH permissible difference and +0.1^{o}C of temperature permissible difference of the instrument (0.00hrs on 12.09.2014 to 24.00hrs on13.09.2014)

Time (Hrs)	0.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00
Temperature/^{o}C	28.0	27.8	27.7	27.5	27.5	27.4	27.6	29.3	30.4	32.3
RH %	72.5	80.9	82.2	83.2	81.2	83.2	82.3	75.6	71.4	63.6

10.00	11.00	12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00
34.4	34.4	34.6	36.1	35.1	34.2	32.8	31.0	29.7	29.3	28.9	28.5	28.3
56.6	56.6	55.5	46.5	48.8	52.7	61.8	67.4	74.3	71.9	74.7	76.4	77.7

23.00	00.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00	11.00
28.0	27.9	27.6	27.6	27.5	27.5	27.5	27.6	29.2	31.9	32.0	32.3	32.6
79.5	82.0	82.6	82.6	83.1	82.2	83.9	84.2	77.2	66.3	66.2	61.7	60.4

12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00	23.00
32.9	32.9	34.5	34.5	32.7	31.4	29.8	29.0	28.7	28.4	28.2	28.1
53.6	52.0	46.5	49.1	61.3	68.2	71.4	74.3	77.5	79.2	79.8	80.6

The Table 5.6 has been produced by calculating the SFC values and fuel oil saving amounts for the particular ambient temperatures in Table 5.5.

Table 5.6: Relevant SFC and hourly fuel oil saving amounts for Table 5.5

Time (Hrs)	0.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00
Temperature/ºC	28.0	27.8	27.7	27.5	27.5	27.4	27.6	29.3	30.4	32.3
RH %	72.5	80.9	82.2	83.2	81.2	83.2	82.3	75.6	71.4	63.6
Calculated SFC /g	205.73	205.69	205.68	205.65	205.65	205.63	205.66	205.93	206.11	206.41
Fuel saving/kg	83.80	82.73	82.46	81.65	81.65	81.12	81.92	89.15	93.97	102.00

10.00	11.00	12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00
34.4	34.4	34.6	36.1	35.1	34.2	32.8	31.0	29.7	29.3	28.9	28.5	28.3
56.6	56.6	55.5	46.5	48.8	52.7	61.8	67.4	74.3	71.9	74.7	76.4	77.7
206.75	206.75	206.78	207.02	206.86	206.71	206.49	206.20	206.00	205.93	205.87	205.80	205.77
111.10	111.10	111.91	118.33	114.05	110.03	104.14	96.38	91.02	89.15	87.54	85.67	84.87

23.00	0.00	1.00	2.00	3.00	4.00	5.00	6.00	7.00	8.00	9.00	10.00	11.00
28.0	27.9	27.6	27.6	27.5	27.5	27.5	27.6	29.2	31.9	32.0	32.3	32.6
79.5	82.0	82.6	82.6	83.1	82.2	83.9	84.2	77.2	66.3	66.2	61.7	60.4
205.73	205.71	205.66	205.66	205.65	205.65	205.65	205.66	205.92	206.35	206.36	206.41	206.46
83.80	83.26	81.92	81.92	81.65	81.65	81.65	81.92	88.88	100.40	100.66	102.00	103.34

12.00	13.00	14.00	15.00	16.00	17.00	18.00	19.00	20.00	21.00	22.00	23.00
32.9	32.9	34.5	34.5	32.7	31.4	29.8	29.0	28.7	28.4	28.2	28.1
53.6	52.0	46.5	49.1	61.3	68.2	71.4	74.3	77.5	79.2	79.8	80.6
206.51	206.51	206.76	206.76	206.47	206.27	206.01	205.88	205.84	205.79	205.76	205.74
104.68	104.68	111.37	111.37	103.61	98.25	91.29	87.81	86.74	85.40	84.60	84.06

Total saving from 0.00hrs on 12.09.2014 to 24.00hrs on 13.09.2014 is equal to the total of the hourly fuel oil saving (total saving per two days),

Total fuel saving (per two day) = 4,482.65 kg

Total fuel saving (per day) = 2,241.32 kg/day

When instrument permissible value was considered the fuel saving for the period from 0.00hrs on 12.09.2014 to 24.00hrs on13.09.2014 were 4,466.66 kg and 4,482.65 kg. The difference was 15.99 kg. Due to the measuring instrument permissible difference the possible maximum percentage fuel oil saving error was calculated below.

$$\text{Percentage of maximum error} = \frac{15.99}{4{,}466.66} \times 100$$

$$= 0.36\%$$

5.4 Engine deration evaluation

Engine deration needs to be considered in three ways. Those are the ambient temperature, site altitude and humidity. At the UJPS the site is around 3m above the sea level. Therefore, the altitude factor for the UJPS is not effective for the deration of the engines. Deration due to the ambient temperature and humidity need to be evaluated at site.

5.4.1 Deration due to the ambient temperature effect

The deration due to the ambient temperature effect were calculated for measured temperature readings and tabulated. The deration for ambient temperature readings adjusted for -1°C of temperature permissible difference of instrument were calculated and tabulated in Table 5.7

Table 5.7: Deration percentage calculated for ambient temperature readings adjusted for -0.1°C of temperature permissible difference of the instrument (0.00hrs on 12.09.2014 to 24.00hrs on 13.09.2014)

Hour	Constant temperature/ °C	Constant temperature/ °F	Deration/ %
00.00	27.8	82.0	0.0000
01.00	27.6	81.7	0.0000
02.00	27.5	81.5	0.0000
03.00	27.3	81.1	0.0000
04.00	27.3	81.1	0.0000
05.00	27.2	81.0	0.0000
06.00	27.4	81.3	0.0000
07.00	29.1	84.4	0.0000
08.00	30.2	86.4	0.2800
09.00	32.1	89.8	0.9600

Continue..

Hour	Constant temperature/ ^{o}C	Constant temperature/ ^{o}F	Deration/ %
10.00	34.2	93.6	1.7200
11.00	34.2	93.6	1.7200
12.00	34.4	93.9	1.7800
13.00	35.9	96.6	2.3200
14.00	34.9	94.8	1.9600
15.00	34.0	93.2	1.6400
16.00	32.6	90.7	1.1400
17.00	30.8	87.4	0.4800
18.00	29.5	85.1	0.0200
19.00	29.1	84.4	0.0000
20.00	28.7	83.7	0.0000
21.00	28.3	82.9	0.0000
22.00	28.1	82.6	0.0000
23.00	27.8	82.0	0.0000
00.00	27.7	81.9	0.0000
01.00	27.4	81.3	0.0000
02.00	27.4	81.3	0.0000
03.00	27.3	81.1	0.0000
04.00	27.2	81.0	0.0000
05.00	27.3	81.1	0.0000
06.00	27.4	81.3	0.0000
07.00	29.0	84.2	0.0000
08.00	31.8	89.2	0.8400
09.00	31.8	89.2	0.8400
10.00	32.1	89.8	0.9600
11.00	32.4	90.3	1.0600
12.00	32.7	90.9	1.1800
13.00	32.7	90.9	1.1800
14.00	34.3	93.7	1.7400
15.00	34.3	93.7	1.7400
16.00	32.5	90.5	1.1000
17.00	31.2	88.2	0.6400
18.00	29.6	85.3	0.0600
19.00	28.8	83.8	0.0000
20.00	28.5	83.3	0.0000
21.00	28.2	82.8	0.0000
22.00	28.0	82.4	0.0000
23.00	27.9	82.2	0.0000
Average	30.1	86.1	0.5283

The deration due to the ambient temperature is applicable when the ambient temperature is above 85^{o}F. In the Table 5.7, when the ambient temperature is below 85^{o}F deration has been considered as zero. The average value of the deration percentage is 0.5283%.

Scatter plot of deration Vs measured temperature readings for permissible value of -0.1 °C of instrument using Matlab software is shown in Figure 5.1. The points which show on or below x=0 axis are considered as zero deration percentage in the Table 5.7.

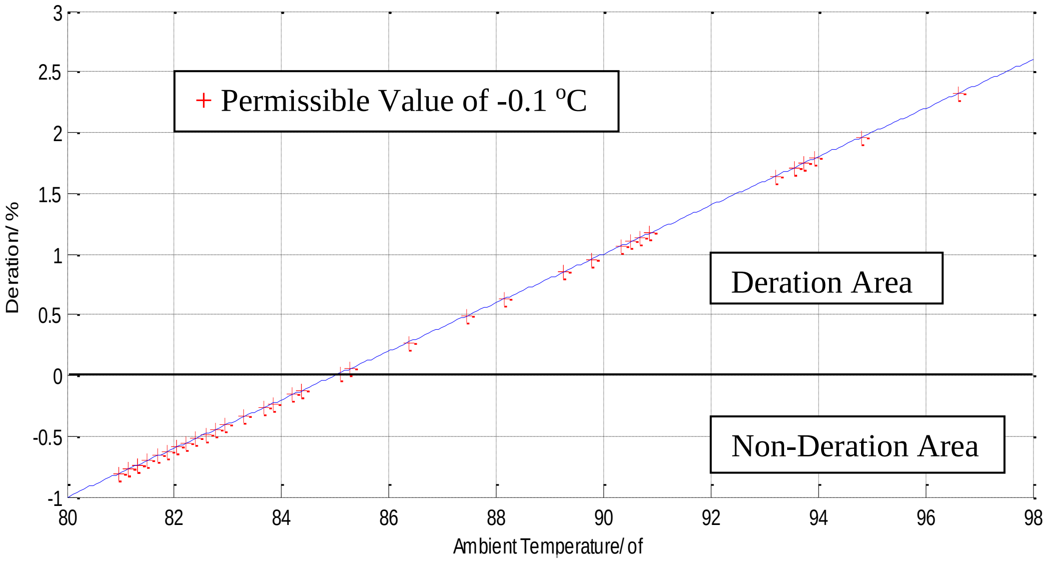


Figure 5.1: Scatter plot of deration Vs measured temperature readings permissible value of -0.1 °C instrument using Matlab software

The deration for ambient temperature readings adjusted for +0.1°C of temperature permissible difference of instrument have also been calculated and tabulated in Table 5.8.

Table 5.8: Deration percentage calculated for ambient temperature readings adjusted for +0.1°C of temperature permissible difference of the instrument (0.00hrs on 12.09.2014 to 24.00hrs on 13.09.2014)

Hour	Constant temperature/ °C	Constant temperature/ °F	Deration/ %
00.00	28.0	82.4	0.0000
01.00	27.8	82.0	0.0000
02.00	27.7	81.9	0.0000
03.00	27.5	81.5	0.0000
04.00	27.5	81.5	0.0000
05.00	27.4	81.3	0.0000
06.00	27.6	81.7	0.0000

Continue..

Hour	Constant temperature/ ^oC	Constant temperature/ ^oF	Deration/ %
07.00	29.3	84.7	0.0000
08.00	30.4	86.7	0.3400
09.00	32.3	90.1	1.0200
10.00	34.4	93.9	1.7800
11.00	34.4	93.9	1.7800
12.00	34.6	94.3	1.8600
13.00	36.1	97.0	2.4000
14.00	35.1	95.2	2.0400
15.00	34.2	93.6	1.7200
16.00	32.8	91.0	1.2000
17.00	31.0	87.8	0.5600
18.00	29.7	85.5	0.1000
19.00	29.3	84.7	0.0000
20.00	28.9	84.0	0.0000
21.00	28.5	83.3	0.0000
22.00	28.3	82.9	0.0000
23.00	28.0	82.4	0.0000
00.00	27.9	82.2	0.0000
01.00	27.6	81.7	0.0000
02.00	27.6	81.7	0.0000
03.00	27.5	81.5	0.0000
04.00	27.5	81.5	0.0000
05.00	27.5	81.5	0.0000
06.00	27.6	81.7	0.0000
07.00	29.2	84.6	0.0000
08.00	31.9	89.4	0.8800
09.00	32.0	89.6	0.9200
10.00	32.3	90.1	1.0200
11.00	32.6	90.7	1.1400
12.00	32.9	91.2	1.2400
13.00	32.9	91.2	1.2400
14.00	34.5	94.1	1.8200
15.00	34.5	94.1	1.8200
16.00	32.7	90.9	1.1800
17.00	31.4	88.5	0.7000
18.00	29.8	85.6	0.1200
19.00	29.0	84.2	0.0000
20.00	28.7	83.7	0.0000
21.00	28.4	83.1	0.0000
22.00	28.2	83.8	0.0000
23.00	28.1	82.6	0.0000
Average	30.3	86.5	0.5600

When the ambient temperature is above 85^{o}F, the deration due to the high ambient temperature occurs. Therefore, in the above Table 5.8, when the ambient temperature is below 85^{o}F deration was zero. The average value of the deration percentage is 0.5600%.

The scatter plot of deration Vs measured temperature readings using Matlab software for ambient temperature readings in the above Table 5.8 is shown in Figure 5.2. The points which shows on or below x=0 axis should be considered as zero deration percentage.

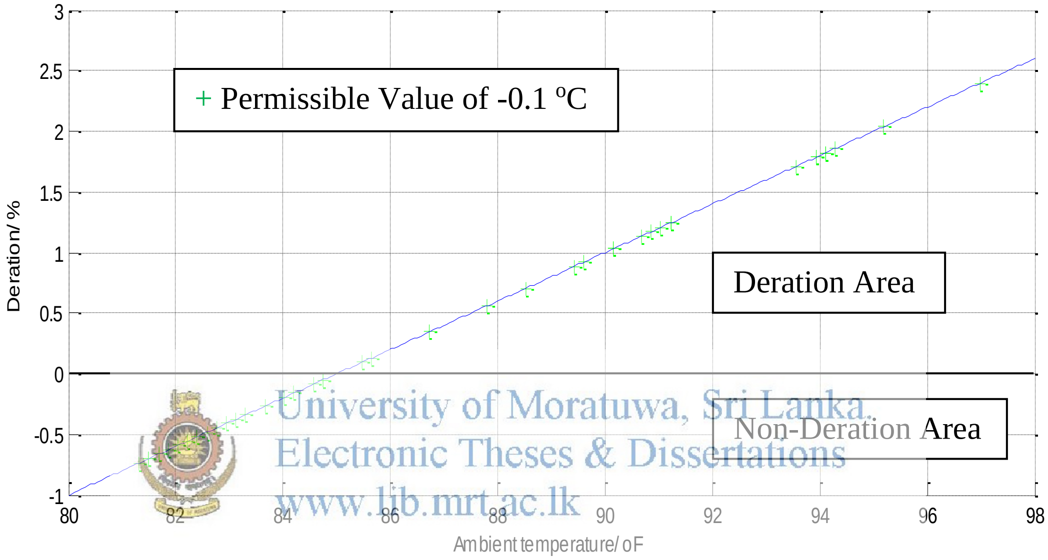


Figure 5.2: Scatter plot of deration Vs measured temperature readings for permissible value of +0.1 ^{o}C instrument using Matlab software

The difference in average percentage deration in the both cases (In Table 5.7 and Table 5.8) = 0.5600 - 0.5283 %

= 0.0317 %

Therefore, the maximum error could occur = $\frac{0.0317 \times 100}{0.5283}$ %

= 6.00 %

5.4.2 Deration due to the altitude effect

The UJPS site is around 3m above the sea level. Therefore, the altitude factor of the UJPS is not considerable for the deration of the engines.

5.4.3 Deration due to the relative humidity effect

Figure 5.3 shows all the points plotted in Table 5.3 and Table 5.5 for ambient temperature vs relative humidity. The red color shows the values in Table 5.3 while the green color shows the values in Table 5.5. From the Figure 5.3, it is obvious when the ambient temperature and RH is closer to the ISO standard site conditions the points move towards the boarder which separates deration and non-deration area. This means while RH reduces deration effect due to the humidity reduces.

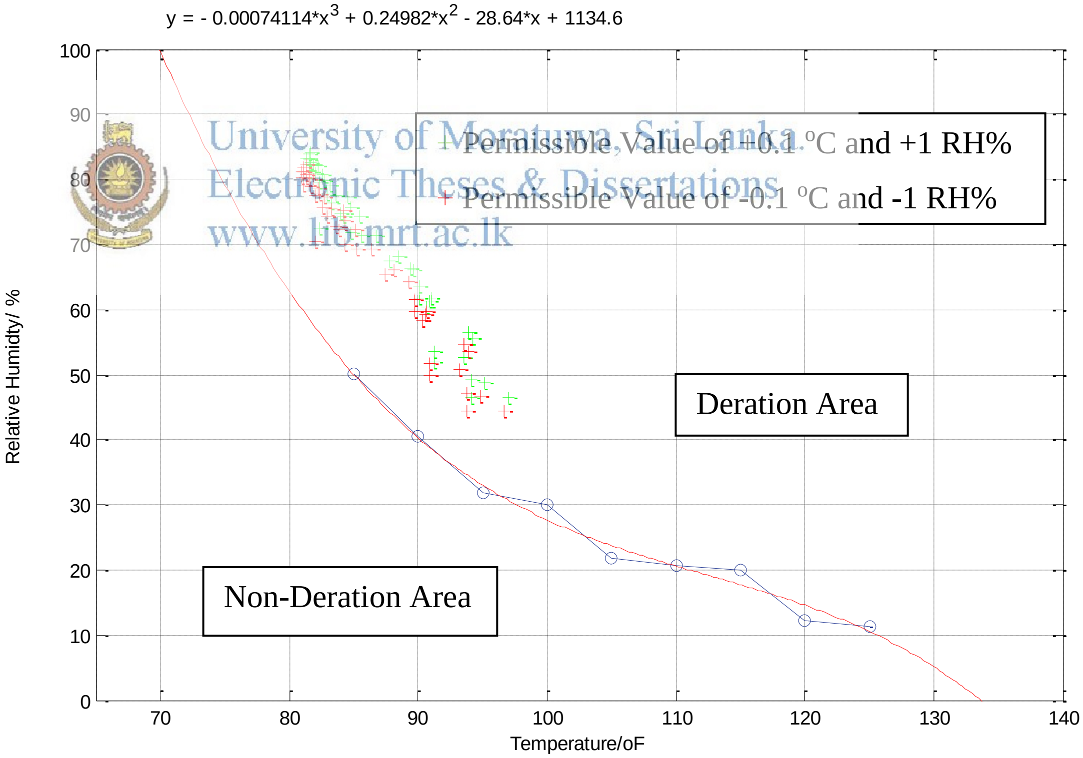


Figure 5.3: Scatter plot of measured temperature and RH points using Matlab software considering the permissible values of the instrument (Refer appendix C)

- **Calculating the amount of deration**

At the ISO standard condition the engine full load is 8.924 MW. According to the calculations it shows below mentioned amount of deration percentage. The number of kilo watt hours derated can be calculated from that percentage of deration. Calculation of the percentage of deration during the 00.00 hour on 12.09.2014 is given below.

Temperature is 27.8 ^{o}C and RH is 70.4%.

The temperature 27.8 ^{o}C = 82.0 ^{o}F

It is needed to interpolate and derive the equation of deration percentage vs RH at (27.8 ^{o}C) 82.0 ^{o}F ambient temperature. Therefore, the equation T85 and T70 was taken which is adjacent to the value 82.0^{o}F.

$$Y = -4.6296 \times 10^{-6} X^3 + 0.00095238 X^2 - 0.013823X - 1.1079 \text{ ----> T85}$$

$$Y = 0 X^3 + 0 X^2 - 0X - 0 \text{ ----> T70}$$

$$Y = a X^3 + b X^2 - c X - d \text{ ----> T82}$$

At T= 82.0 ^{o}F (27.8 ^{o}C) and d constants of the above 3rd order polynomial are interpolated as mentioned below.

Value a,

$$\frac{(-4.6296-0)}{85-70} \times 10^{-6} = \frac{(-4.6296-a)}{85-82} \times 10^{-6}$$

$$a = -3.7037 \times 10^{-6}$$

Value b,

$$\frac{(-0.00095238-0)}{85-70} = \frac{-0.00095238-b}{85-82}$$

$$b = 0.00076190$$

Value c,

$$\frac{-0.013823-0}{85-70} = \frac{-0.013823-c}{85-82}$$

$$c = -0.011058$$

Value d,

$$\frac{-1.1079-0}{85-70} = \frac{-1.1079-d}{85-82}$$

$$d = -0.8863$$

By applying the above calculated a, b, c and d constants the 3rd order polynomial at 82.0 °F ambient temperature is given below.

$$Y = -3.7037\text{x}10^{-6}\,X^{3} + 0.00076190X^{2} -0.011058X -0.8863 \qquad \text{---->}\ T82$$

By using the above T82, the deration at 82°F and 70.4% RH is calculated below.

$$Y = -3.7037\ \text{x}10^{-6}\,(70.4^{3}) + 0.00076190\,(70.4^{2}) -0.011058\,(70.4) - 0.8863$$

$$Y = -1.6877 + 3.7761 - 0.7785 -0.8863$$

$$Y = 0.8190\%$$

Accordingly, all the a, b, c and d constants of the 3rd order polynomials for the hourly readings in Table 5.3 and Table 5.5 are calculated and is tabulated in Table 5.9 and Table 5.10 respectively.

Table 5.9: Calculated a, b, c and d constants of the 3rd order polynomials and percentage of derations (According to the data in Table 5.3)

Hour	Constant temperature/ °C	Constant temperature/ °F	RH %	a	b/ x 10^{-4}	c/ x 10^{-2}	d	Deration/ %
00.00	27.8	82.0	70.4	-3.7037	7.6190	-1.1058	-0.8863	0.8190
01.00	27.6	81.7	78.8	-3.6111	7.4286	-1.0782	-0.8642	1.1320
02.00	27.5	81.5	80.1	-3.5494	7.3016	-1.0598	-0.8494	1.1623
03.00	27.3	81.1	81.1	-3.4259	7.0476	-1.0229	-0.8198	1.1585
04.00	27.3	81.1	79.1	-3.4259	7.0476	-1.0229	-0.8198	1.0851
05.00	27.2	81.0	81.1	-3.3950	6.9841	-1.0137	-0.8125	1.1481
06.00	27.4	81.3	80.2	-3.4876	7.1746	-1.0413	-0.8346	1.1459
07.00	29.1	84.4	73.6	-4.4444	9.1428	-1.3270	-1.0636	1.1404
08.00	30.2	86.4	69.4	-4.8889	10.2905	-1.8364	-0.9164	1.1313
09.00	32.1	89.8	61.6	-5.5186	12.1525	-2.9391	-0.4512	1.0597

Continue..

Hour	Constant temperature/ ºC	Constant temperature/ ºF	RH %	a	b/ x 10^{-4}	c/ x 10^{-2}	d	Deration/ %
10.00	34.2	93.6	54.6	-7.7374	16.6878	-5.1219	0.1025	1.0493
11.00	34.2	93.6	54.6	-7.7374	16.6878	-5.1219	0.1025	1.0493
12.00	34.4	93.9	53.5	-7.9192	17.0567	-5.2984	0.1463	0.9811
13.00	35.9	96.6	44.5	-6.5657	14.0315	-2.5718	-0.3481	0.7075
14.00	34.9	94.8	46.8	-8.4647	18.1631	-5.8278	0.2779	0.6610
15.00	34.0	93.2	50.7	-7.4950	16.1961	-4.8866	0.0440	0.7529
16.00	32.6	90.7	59.8	-5.9798	13.1226	-3.4158	-0.3215	1.0498
17.00	30.8	87.4	65.4	-5.0741	10.8381	-2.1607	-0.7795	1.0236
18.00	29.5	85.1	72.3	-4.6481	9.5786	-1.4147	-1.0942	1.1332
19.00	29.1	84.4	69.9	-4.4444	9.1428	-1.3270	-1.0636	0.9581
20.00	28.7	83.7	72.7	-4.2284	8.6984	-1.2625	-1.0119	1.0429
21.00	28.3	82.9	74.4	-3.9815	8.1905	-1.1888	-0.9528	1.0568
22.00	28.1	82.6	75.7	-3.8889	8.0000	-1.1611	-0.9306	1.0878
23.00	27.8	82.0	77.5	-3.7037	7.6190	-1.1058	-0.8863	1.1088
00.00	27.7	81.9	80.0	-3.6728	7.5555	-1.0966	-0.8789	1.1988
01.00	27.4	81.3	80.6	-3.4876	7.1746	-1.0413	-0.8346	1.1608
02.00	27.4	81.3	80.6	-3.4876	7.1746	-1.0413	-0.8346	1.1608
03.00	27.3	81.1	81.1	-3.4259	7.0476	-1.0229	-0.8198	1.1585
04.00	27.2	81.0	80.2	-3.3950	6.9841	-1.0137	-0.8125	1.1154
05.00	27.3	81.1	81.9	-3.4259	7.0476	-1.0229	-0.8198	1.1876
06.00	27.4	81.3	82.2	-3.4876	7.1746	-1.0413	-0.8346	1.2201
07.00	29.0	84.2	75.2	-4.3827	9.0159	-1.3086	-1.0488	1.2019
08.00	31.8	89.2	64.3	-5.4074	11.8239	-2.7445	-0.5333	1.1530
09.00	31.8	89.2	64.2	-5.4074	11.8239	-2.7445	-0.5333	1.1473
10.00	32.1	89.8	59.7	-5.5186	12.1525	-2.9391	-0.4512	0.9512
11.00	32.4	90.3	58.4	-5.7374	12.6308	-3.1805	-0.3800	0.9277
12.00	32.7	90.9	51.6	-6.1011	13.3685	-3.5335	-0.2922	0.6057
13.00	32.7	90.9	50.0	-6.1011	13.3685	-3.5335	-0.2922	0.5205
14.00	34.3	93.7	44.5	-7.7980	16.8108	-5.1807	0.1171	0.4535
15.00	34.3	93.7	47.1	-7.7980	16.8108	-5.1807	0.1171	0.5915
16.00	32.5	90.5	59.3	-5.8586	12.8767	-3.2982	-0.3507	0.9999
17.00	31.2	88.2	66.2	-5.2222	11.2762	-2.4202	-0.6701	1.1544
18.00	29.6	85.3	69.4	-4.6852	9.6881	-1.4796	-1.0669	1.0064
19.00	28.8	83.8	72.3	-4.2592	8.7619	-1.2717	-1.0193	1.0317
20.00	28.5	83.3	75.5	-4.1049	8.4444	-1.2256	-0.9823	1.1392
21.00	28.2	82.8	77.2	-3.9506	8.1270	-1.1796	-0.9454	1.1699
22.00	28.0	82.4	77.8	-3.8271	7.8730	-1.1427	-0.9159	1.1583
23.00	27.9	82.2	78.6	-3.7654	7.7460	-1.1243	-0.9011	1.1723
Average	30.1	86.2	68.5					1.0256

Table 5.10: Calculated a, b, c and d constants of the 3rd order polynomials and percentage of derations (According to the data in Table 5.5)

Hour	Constant temperature/ °C	Constant temperature/ °F	RH %	a	b/ x 10^{-4}	c/ x 10^{-2}	d	Deration/ %
00.00	28.0	82.4	72.5	-3.8271	7.8730	-1.1427	-0.9159	0.9355
01.00	27.8	82.0	80.9	-3.7037	7.6190	-1.1058	-0.8863	1.2446
02.00	27.7	81.9	82.2	-3.6728	7.5555	-1.0966	-0.8789	1.2849
03.00	27.5	81.5	83.2	-3.5494	7.3016	-1.0598	-0.8494	1.4681
04.00	27.5	81.5	81.2	-3.5494	7.3016	-1.0598	-0.8494	1.3820
05.00	27.4	81.3	83.2	-3.4876	7.1746	-1.0413	-0.8346	1.2568
06.00	27.6	81.7	82.3	-3.6111	7.4286	-1.0782	-0.8642	1.2671
07.00	29.3	84.7	75.6	-4.5370	9.3333	-1.3547	-1.0857	1.2641
08.00	30.4	86.7	71.4	-4.9444	10.4548	-1.9337	-0.8753	1.2741
09.00	32.3	90.1	63.6	-5.6162	12.3849	-3.0628	-0.4092	1.2077
10.00	34.4	93.9	56.6	-7.9192	17.0567	-5.2984	0.1463	1.1757
11.00	34.4	93.9	56.6	-7.9192	17.0567	-5.2984	0.1463	1.1757
12.00	34.6	94.3	55.5	-8.1617	17.5484	-5.5337	0.2048	1.1437
13.00	36.1	97.0	46.5	-6.0606	12.9372	-1.7283	-0.5119	0.8724
14.00	35.1	95.2	48.8	-8.3334	17.8618	-5.5238	0.2252	0.8149
15.00	34.2	93.6	52.7	-7.7374	16.6878	-5.1219	0.1025	0.9055
16.00	32.8	91.0	61.8	-6.1617	13.4914	-3.5923	-0.2776	1.2007
17.00	31.0	87.8	67.4	-5.1482	11.0572	-2.2905	-0.7248	1.1782
18.00	29.7	85.5	74.3	-4.7222	9.7976	-1.5445	-1.0395	1.2848
19.00	29.3	84.7	71.9	-4.5370	9.3333	-1.3547	-1.0857	1.0788
20.00	28.9	84.0	74.7	-4.3210	8.8889	-1.2901	-1.0340	1.1612
21.00	28.5	83.3	76.4	-4.1049	8.4444	-1.2256	-0.9823	1.1797
22.00	28.3	82.9	77.7	-3.9815	8.1905	-1.1888	-0.9528	1.2007
23.00	28.0	82.4	79.5	-3.8271	7.8730	-1.1427	-0.9159	1.2286
00.00	27.9	82.2	82.0	-3.7654	7.7460	-1.1243	-0.9011	1.3093
01.00	27.6	81.7	82.6	-3.6111	7.4286	-1.0782	-0.8642	1.2785
02.00	27.6	81.7	82.6	-3.6111	7.4286	-1.0782	-0.8642	1.2785
03.00	27.5	81.5	83.1	-3.5494	7.3016	-1.0598	-0.8494	1.2753
04.00	27.5	81.5	82.2	-3.5494	7.3016	-1.0598	-0.8494	1.2417
05.00	27.5	81.5	83.9	-3.5494	7.3016	-1.0598	-0.8494	1.3050
06.00	27.6	81.7	84.2	-3.6111	7.4286	-1.0782	-0.8642	1.3390
07.00	29.2	84.6	77.2	-4.5061	9.2698	-1.3454	-1.0784	1.3344
08.00	31.9	89.4	66.3	-5.4445	11.9334	-2.8094	-0.5059	1.2903
09.00	32.0	89.6	66.2	-5.4815	12.0429	-2.8743	-0.4785	1.3062
10.00	32.3	90.1	61.7	-5.6162	12.3849	-3.0628	-0.4092	1.0967
11.00	32.6	90.7	60.4	-5.9798	13.1226	-3.4158	-0.3215	1.0850
12.00	32.9	91.2	53.6	-6.2829	13.7373	-3.7100	-0.2484	0.7422
13.00	32.9	91.2	52.0	-6.2829	13.7373	-3.7100	-0.2484	0.6536
14.00	34.5	94.1	46.5	-8.0404	17.3025	-5.4160	0.1756	0.5899
15.00	34.5	94.1	49.1	-8.0404	17.3025	-5.4160	0.1756	0.7359

Continue..

Hour	Constant temperature/ ºC	Constant temperature/ ºF	RH %	a	b/ x 10^{-4}	c/ x 10^{-2}	d	Deration %
16.00	32.7	90.9	61.3	-6.1011	13.3685	-3.5335	-0.2922	1.1598
17.00	31.4	88.5	68.2	-5.2778	11.4405	-2.5175	-0.6290	1.3011
18.00	29.8	85.6	71.4	-4.7407	9.8524	-1.5769	-1.0258	1.1454
19.00	29.0	84.2	74.3	-4.3827	9.0159	-1.3086	-1.0488	1.1585
20.00	28.7	83.7	77.5	-4.2284	8.6984	-1.2625	-1.0119	1.2659
21.00	28.4	83.1	79.2	-4.0432	8.3175	-1.2072	-0.9676	1.2849
22.00	28.2	83.8	79.8	-3.9506	8.1270	-1.1796	-0.9454	1.2810
23.00	28.1	82.6	80.6	-3.8889	8.0000	-1.1611	-0.9306	1.2943
Average	30.3	86.5	70.5					1.1654

The difference in average percentage deration
of the both cases (Table 5.9 and Table 5.10) = 1.1654 - 1.0256 %
= 0.1398 %

The maximum error that could occur = $\frac{0.1398 \times 100}{1.0256}$ %
= 13.63 %

The resultant deration due to the ambient temperature effect and the humidity effect adjusted for -1% of RH and -0.1ºC of temperature permissible differences of the instruments (0.00hrs on 12.09.2014 to 24.00hrs on13.09.2014) is calculated below.

Deration due to the ambient temperature effect = 0.5283 %
Deration due to the relative humidity effect = 1.0256%
Total resultant deration percentage = Deration due to the ambient temperature effect + Deration due to the relative humidity effect
= 0.5283 % + 1.0256%
= 1.5539 %

The resultant deration due to the ambient temperature effect and the humidity effect adjusted for +1% of RH and +0.1^{o}C of temperature permissible differences of the instruments (0.00hrs on 12.09.2014 to 24.00hrs on13.09.2014) is calculated below.

Deration due to the ambient temperature effect = 0.5600 %

Deration due to the relative humidity effect = 1.1654%

Total resultant deration percentage = Deration due to the ambient temperature effect + Deration due to the relative humidity effect

= 0.5600 % + 1.1654%

= 1.7254 %

The calculation of deration for -1% of RH and -0.1^{o}C of temperature permissible differences of the instruments are given below,

Maximum load at ISO standard conditions = 8.924 MW

Deration percentage = 1.5539 %

The calculated maximum load at site $= \frac{8.924 \text{ x } (100-1.5539)}{100}$

= 8.785 MW

Therefore, the increment of engine capacity = 8.924 - 8.785

= 0.139MW

Derated power of one engine = 139 kW

Derated power of all engines = 139 x 3

= 417 kW

Accordingly, the calculation of deration for +1% of RH and +0.1°C of temperature permissible differences of the instruments is given below,

Average deration percentage = 1.7254 %

At site calculated maximum load $= \frac{8.924 \times (100-1.7254)}{100}$

= 8.770 MW

Therefore, increment of engine capacity = 8.924 - 8.770

= 0.104MW

Derated power of one engine = 154 kW

Derated power of all engines = 154 x 3 kW

= 462 kW

From the above calculations, it shows that when the instrument permissible difference is -1% for RH and -0.1°C for temperature, the derated power was 417 kW while having 462 kW of derated power at permissible difference +1% for RH and +0.1°C for temperature.

5.5 Calculation of required cooling load

According to the above evaluations by shifting the UJPS site conditions to the ISO 3046-1 standard, it is possible to save fuel oil consumption. Since the engines are not much derating, it does not considerably improve the output of the engine power. In addition, the building air conditioning can be introduced to gain an additional advantage of electricity saving.

5.5.1 Intake air mass flow rate

Since UJPS is having W32V20 engines, the cylinder bore and the stroke are 32cm and 40cm respectively with 20 numbers of cylinders per engine.

Stroke of the cylinder = 0.40 m

Bore of the cylinder = 0.32 m

No. of cylinders = 20

Rounds per Minute (RPM) = 750

Volume of a cylinder $= \Pi r^2 L$ m^3

$= \Pi$ x(0.16^2) x 0.4 m^3

= 0.0322 m^3

Volume of 20 cylinders = 0.644 m^3

Number of suction revolutions per minute = 750/2

= 375

Number of suction revolutions per second = 375/60

Induced air volume $= 0.644 \times \frac{375}{60}$ m^3/s

Theoretical air consumption = 4.025 m^3/s

Volumetric efficiency = 80%

Actual charge air volume flow rate to the cylinders = 4.025 x0.8 m^3/s

= 3.22 m^3/s

Assuming air as an ideal gas, volume of charge air could be calculated

$$\frac{P_1V_1}{T_1} = \frac{P_2V_2}{T_2}$$

Charge air pressure is 2.82 bar and temperature is 56 ^{o}C

$$\frac{2.82 \times 3.22}{273+56} = \frac{1 \text{x} V_2}{273+25}$$

$$V_2 = 8.22 \text{ m}^3/\text{s}$$

Density of air at 25^{o}C and 1 bar is taken as 1.1839 kg/m^3 [32].

Then the mass flow rate is = 8.22 x 1.1839

= 9.732 kg/s/engine

5.5.2 Weight of dry air

At RH 30% and ambient temperature 25^{o}C weight of dry air was calculated to find the cooling load using phychrometric chart.

Partial pressure of water vapour at 30% and 25^{o}C = 0.94 kPa

$$\text{Specific Humidity} = \frac{0.622 \text{x} \text{Partial Pressure of Water vapour}(P_s)}{\text{Partial Pressure of Dry Air }(P_{da})}$$

Partial Pressure of water vapour(P_s) [33] = 0.94 kPa

$$= \frac{0.622 \text{ x } 0.94}{(100-0.94)}$$

$$= 0.0059 \text{ kg/ kg}_{da}$$

Therefore, amount of dry air in the 9.732 kg/s moist air

$$= \frac{9.732}{1+0.0059}$$

$$= 9.675 \text{ kg}_{da}/\text{s}$$

5.5.3 Cooling load required to condition the combustion air

It is needed to know the highest enthalpy value of suction air in the environment to estimate the required maximum cooling load to evaluate the capacity of the vapour absorption chiller. When the ambient temperature and RH increase, the enthalpy of air increases. The values of Table 5.10 have been taken to evaluate, because in the Table 5.10 the hourly ambient temperature and RH are higher than the values in the Table 5.9. The relevant enthalpy values of air are calculated according to the temperature and RH (Appendix D) and tabulated in the Table 5.11. The maximum enthalpy value has been considered to estimate the required capacity of the vapour absorption chiller to condition the combustion air.

Table 5.11: Calculated enthalpy of ambient air (According to the data in Table 5.10)

Hour	Constant temperature/ ^{o}C	Constant temperature/ ^{o}F	RH/ %	Enthalpy/ kJ/kg
00.00	28.0	82.4	72.5	72.3
01.00	27.8	82.0	80.9	76.8
02.00	27.7	81.9	82.2	77.2
03.00	27.5	81.5	83.2	76.9
04.00	27.5	81.5	81.2	75.8
05.00	27.4	81.3	83.2	76.6
06.00	27.6	81.7	82.3	76.8
07.00	29.3	84.7	75.6	79.3
08.00	30.4	86.7	71.4	80.8
09.00	32.3	90.1	63.6	82.3
10.00	***34.4***	***93.9***	***56.6***	***84.6***
11.00	***34.4***	***93.9***	***56.6***	***84.6***
12.00	34.6	94.3	55.5	84.4
13.00	36.1	97.0	46.5	81.3
14.00	35.1	95.2	48.8	79.9
15.00	34.2	93.6	52.7	80.3
16.00	32.8	91.0	61.8	82.8
17.00	31.0	87.8	67.4	80.2
18.00	29.7	85.5	74.3	80.0
19.00	29.3	84.7	71.9	76.8
20.00	28.9	84.0	74.7	77.1
21.00	28.5	83.3	76.4	76.7
22.00	28.3	82.9	77.7	76.7
23.00	28.0	82.4	79.5	76.7
00.00	27.9	82.2	82.0	77.8
01.00	27.6	81.7	82.6	77.0
02.00	27.6	81.7	82.6	77.0
03.00	27.5	81.5	83.1	76.9
04.00	27.5	81.5	82.2	76.4
05.00	27.5	81.5	83.9	77.4
06.00	27.6	81.7	84.2	78.0
07.00	29.2	84.6	77.2	80.0
08.00	31.9	89.4	66.3	82.9
09.00	32.0	89.6	66.2	83.2
10.00	32.3	90.1	61.7	80.8
11.00	32.6	90.7	60.4	80.9
12.00	32.9	91.2	53.6	76.4
13.00	32.9	91.2	52.0	75.1

Continue..

Hour	Constant temperature/ ^oC	Constant temperature/ ^oF	RH/ %	Enthalpy/ kJ/kg
14.00	34.5	94.1	46.5	75.8
15.00	34.5	94.1	49.1	78.1
16.00	32.7	90.9	61.3	82.0
17.00	31.4	88.5	68.2	82.4
18.00	29.8	85.6	71.4	78.4
19.00	29.0	84.2	74.3	77.2
20.00	28.7	83.7	77.5	78.2
21.00	28.4	83.1	79.2	78.1
22.00	28.2	83.8	79.8	77.7
23.00	28.1	82.6	80.6	77.8

The maximum enthalpy value is 84.6 kJ/kg and it has been occurred when the ambient temperature is at 34.4^oC and RH is 56.6%. Therefore, according to the above calculations 9.732 kg/s of air flow rate with 56.6% of RH and 34.4^oC of ambient temperature is needed to be conditioned to the 30% of RH and 25^oC of ambient temperature in order to calculate the required maximum cooling load.

By referring the psychrometric chart at point 56.6% of RH and 34.4^oC of ambient temperature the enthalpy is 84.6 kJ/kg and at 30% of RH, 25^oC of ambient temperature the enthalpy is 40.1 kJ/kg (Appendix E).

Therefore, the enthalpy difference = 84.6-40.1 kJ/kg
= 44.5 kJ/kg

Needed energy for 9.675kg/s of air mass flow rate = 44.5 kJ/kg x 9.675 kg/s
= 430.5 kW

For all 3 engines = 430.5 kW x 3
= 1,291.5 kW

5.5.4 Cooling load required to reduce the charge air cooling water inlet

The required amount of energy to bring down the charge air cooling water inlet temperature from 47^{o}C to 25^{o}C is calculated below,

Mass flow rate (m) of charge air cooling water is 16.67 kg/s

Cooling Load (Q) $= m\ C\ \Delta T$

Specific heat capacity (C) $= 4.2$ kJ/kg^{o}C

Q $= 16.67 \times 4.2 \times (47-25)$ kW

Q $= 1{,}540.31$ kW

Cooling Load required for all 3 engines $= 1{,}540.31 \times 3$ kW

$= 4{,}620.93$ kW

The required heat energy needs to be rejected to reduce the temperature of the charge air cooling water inlet from 47^{o}C to 25^{o}C is approximately 4,650 kW for all 3 engines.

The cooling water inlet temperature at the charge air cooler is 47^{o}C. However, the cooling water outlet temperature at the radiator yard varies from 35^{o}C to 40^{o}C depending on ambient temperature. The 40^{o}C temperature cooling water is considered from radiator outlet in order to design the cooling load for water cooling by an absorption chiller which can cater the worst case scenario (instead of 35^{o}C). The 47^{o}C to 40^{o}C (to 35^{o}C) temperature reduction can be achieved using the radiator yard. The radiator yard cooling water outlet (around 35^{o}C to 40^{o}C) could be directly sent through an absorption chiller and get down the temperature further to 25^{o}C. Thereafter, it needs to send the charge air cooling water of 25^{o}C temperature to the charge air cooler inlet to meet the ISO standard. Required cooling load to reduce charge air cooling water temperature from 40^{o}C to 25^{o}C is calculated below.

Cooling water flow rate through the charge air cooler is 16.67 kg/s/engine.

Cooling Load (Q) $= m\ C\ \Delta T$

Cooling Load per engine $= 16.67 \times 4.2 \times (40-25)$ kW

Q = 1,050.21 kW
For all three engines = 1,050.21 x 3 kW
Cooling Load for all three engines = 3,150.63 kW

This is the required heat energy that needs to be rejected in order to reduce the charge air cooling water temperature from 40^{o}C to 25^{o}C.

5.5.5 Building cooling load

In the UJPS the air conditioning of the building is being done by split type air conditioning units. The numbers of units with their cooling capacities in the UJPS are given in Table 5.12.

Table 5.12: Summary of the air conditioning units at UJPS

No.	Cooling Capacity/ BTU	Electricity Demand kW	Cooling Capacity/ kW	Quantity/ Numbers	Total Cooling Capacity/ kW	Total Electricity Demand kW
01	12,000	1.17	03.52	14	49.28	16.38
02	18,000	1.75	05.27	16	84.32	28.00
03	24,000	2.15	07.03	11	77.33	23.65
04	75,000	7.68	22.00	06	132.00	46.08
Total					342.93	114.11

For air conditioning purposes, the UJPS needs 342.93kW of cooling load. The required cooling capacity of the absorption systems are given in Table 5.13 for each category.

Table 5.13: Summary of the required cooling loads to introduce absorption chiller system

No	Description	Required cooling capacity/ kW
01	To reduce the temperature and Relative Humidity to 25^{o}C and 30%	1,291.50
02	For the building cooling purposes	342.93
03	For charge air cooling (16.61 kg/s from 40^{o}C to 25^{o}C)	3,150.63
Total		4,785.06

5.6 Steam available in the exhaust boilers

UJPS is utilizing exhaust energy of only one engine at a time to generate steam through a gas tube boiler for the whole station steam consumption. The details of the steam system are mentioned below.

Steam temperature = 170^{o}C
Steam pressure = 7 bar
Maximum steam flow rate of the boiler of an engine = 2,500kg/h
= $\frac{2,500}{3,600}$ kg/s
= 0.694 kg/s

The currently operating boiler was set to 40% of exhaust flow rate level to cater the normal plant steam consumption. Hence, the required total steam for the operation of the whole plant is calculated below.

The steam required for plant operations = 0.4 x 2,500 kg/h
= 1,000 kg/h
Balance of unutilized steam in the running boiler = 2,500-1000 kg/h
= 1,500 kg/h

Therefore, the balance of 1,500 kg/h of steam is available in addition which can be utilized as waste heat source for vapour absorption chillers. The waste heat in the exhaust of remaining both engines can also be utilized. Hence, the availability of total steam flow rate is calculated as mentioned below.

Steam available in the running boiler = 1,500 kg/h
Steam available in the other two engines = 2,500 x 2
= 5,000 kg/h

Total steam available = 5,000 + 1,500 kg/h

= 6,500 kg/h

5.7 Energy in HT cooling water for vapour absorption chiller

The single stage hot water chillers mostly demand 98°C of hot water as the waste heat energy source and 10°C of reduction of hot water is expected. The temperature of the engine HT water is 96°C. According to that, the energy available for a single stage hot water vapour absorption chillers is calculated below. (Appendix F)

HT water mass flow rate available for all three engines are 32.14 m^3/h,

Heat energy available,Q = m C ΔT

= 32.14 x 4.2 x (96-86) kW

= 1,349.88 kW

5.8 Introducing the vapour absorption chiller

In addition to the building cooling the fuel saving advantage is 4,466.66 kg per two days.

Total fuel saving per day by all 3 engines at UJPS = $\frac{\text{4,466.66 kg}}{2}$ + 0.36% tolerance

= 2,233.33 kg + 0.36% tolerance

Density of fuel = 970kg/m^3

Total fuel saving per day = 2,233.33 / 970

= 2.3024 m3 + 0.36% tolerance

= 2,302.4 litres + 0.36% tolerance

LKR 90/= per HFO litre

Saving per day = LKR 207,216/= + 0.36% tolerance

= LKR 207,216/= to LKR 207,962/=

If we consider performance data of Broad X Non- Electric BS 500 steam absorption chiller of Broad Central Air Conditioning it shows following parameters. (Appendix G)

Table 5.14: BS 500 model steam chiller performance data [34]

Model	Cooling Capacity kW	Chill water flow rate m^3/h	Pressure drop kPa	Cooling water flow rate m^3/h	Pressure drop kPa	Steam consumption Kg/h	Total Power Demand kW
BS 500	5,815	714	60	1,221	70	6227	275.2

General Conditions

1. Rated saturated steam pressure: 0.8 MPa, rated condensate temp: 95°C

2. Rated chilled water outlet/inlet temp: 7°C/14°C 3. Rated cooling water outlet/inlet temp: 37°C/30°C

4. Lowest permitted outlet temperature for chilled water: 5°C

5. Lowest permitted inlet temperature for cooling water: 10°C

6. Steam pressure upper limit 110%

7. Adjustable chilled water flow rate: 50%~120%

8. Pressure limit for chilled W, cooling W: 0.8MPa (except special order)

9. Adjustable load: 5%~115%

10. Machine room ambient temperature: 5~43°C, humidity ≤ 85%

11. Standard climate conditions for cooling operation: 36°C, relative humidity 50% (wet bulb 27°C)

12. Rated cooling COP: 1.41

13. Life design: 25 years

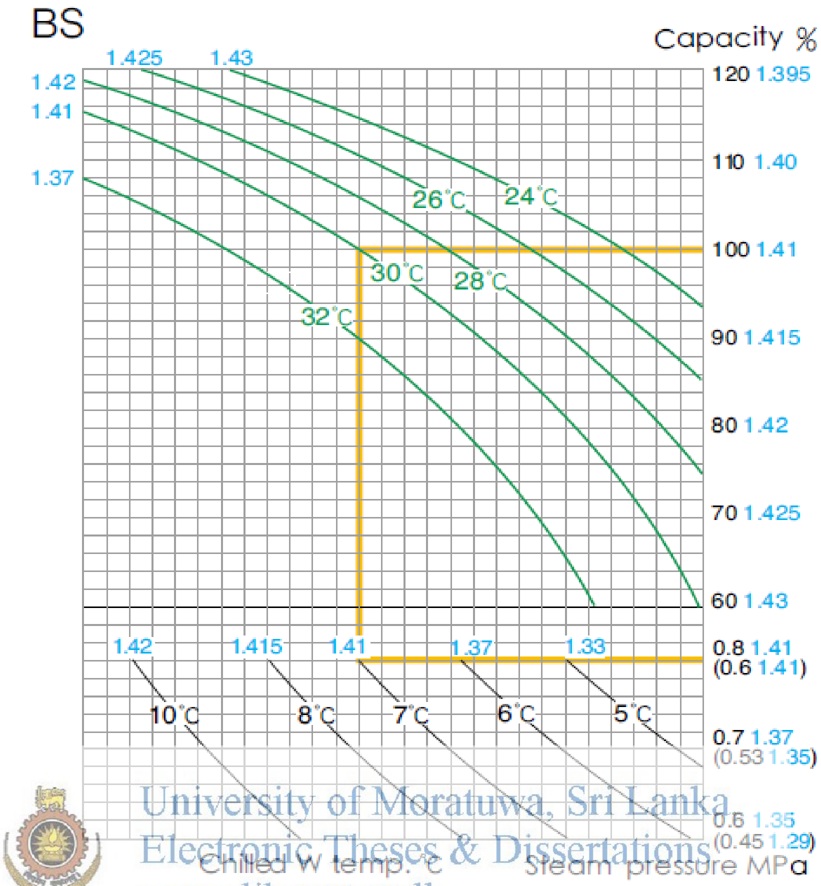


Figure 5.4: Model selection curves, chilled/cooling water temp, cooling capacity, COP [34]

According to the above Figure 5.4 for the 7 bar steam, 7^{o}C chill water temperature and 30^{o}C condenser temperature the power output is around 92.5% of the rated output.

Therefore, the actual output cooling load = 5,815 x 1.15 x 0.925 kW

(Multiplied by 1.15 since the maximum capacity is 115% of nominal capacity)

= 6,185.70 kW

By introducing the above absorption chiller, it is obvious that required cooling load of 4,785.06 kW is achievable. According to the Table 5.14, the required steam flow rate for the absorption chiller is 6,227 kg/h and it is available in the UJPS as unutilized steam.

5.8.1 Net electricity consumption (Appendix G)

Total power demand for the chiller system = 275.2 kW

Total power demand for the air handling unit has been taken as 15% of the power consumption of the chiller system.

Total power demand for air handling unit = 275.2 x0.15 kW
= 41.28 kW
Savings gained by building cooling = 114.11 kW
Balance consumption = 275.2 + 41.28 -114.11 kW
Net electricity consumption = 202.37 kW
Consumption per day = 202.37 x 24 kWh
= 4,856.88 kWh

Average electricity selling price per unit is LKR 17.93 [1]

Additional Electricity cost = LKR 4,856.88 x 17.93
= LKR 87,083.85 per day

5.8.2 Investment and payback period calculation (Appendix H)

Installation cost of the new vapour absorption chiller is given below.

- **Cost for the chiller installations**

Total cost for the chiller = EUR 670,000.00
Total cost for the pump set = EUR 203,000.00
Total cost for the machine room = EUR 35,000.00
Cost for the chiller installations = EUR 908,000.00
At conversion rate of 1.00 EUR is LKR 165.93 as at 18-11-2014 [35]
= LKR 150,664,440.00

- **Local costs**

Total cost for the cooling tower = LKR 6,000,000.00
Piping = LKR 8,000,000.00

Electrical connections	= LKR 5,000,000.00
Heat exchangers	= LKR 6,000,000.00
Cost for the chiller installations	= LKR 150,664,440.00
Total Investment	= LKR 175,664,440.00
Price saving due to fuel saving	= LKR 207,216/= per day
Net saving	= LKR 207,216.00 - Rs. 87,083.85 per day
	= LKR 120,132.15 per day
Net savings per month	= LKR 120,132.15 x 30
	= LKR 3,603,964.50 per month

The annual increase in electricity tariff and annual fuel price increment has been taken as 15%.

Table 5.15: Payback period calculation of investment

Year	Month	Monthly Saving/ LKR	Recover of Investment/ LKR
	0	0	-175,664,440.00
First year	1	3,603,964.50	-172,060,475.50
	2	3,603,964.50	-168,456,511.00
	3	3,603,964.50	-164,852,546.50
	4	3,603,964.50	-161,248,582.00
	5	3,603,964.50	-157,644,617.50
	6	3,603,964.50	-154,040,653.00
	7	3,603,964.50	-150,436,688.50
	8	3,603,964.50	-146,832,724.00
	9	3,603,964.50	-143,228,759.50
	10	3,603,964.50	-139,624,795.00
	11	3,603,964.50	-136,020,830.50
	12	3,603,964.50	-132,416,866.00
Second year	1	4,144,559.18	-128,272,306.83
	2	4,144,559.18	-124,127,747.65
	3	4,144,559.18	-119,983,188.47
	4	4,144,559.18	-115,838,629.29
	5	4,144,559.18	-111,694,070.11
	6	4,144,559.18	-107,549,510.93

Continue..

Year	Month	Monthly Saving/ LKR	Recover of Investment/ LKR
Second year	7	4,144,559.18	-103,404,951.75
	8	4,144,559.18	-99,260,392.57
	9	4,144,559.18	-95,115,833.38
	10	4,144,559.18	-90,971,274.20
	11	4,144,559.18	-86,826,715.02
	12	4,144,559.18	-82,682,155.84
Third year	1	4,766,243.06	-77,915,912.79
	2	4,766,243.06	-73,149,669.73
	3	4,766,243.06	-68,383,426.67
	4	4,766,243.06	-63,617,183.61
	5	4,766,243.06	-58,850,940.55
	6	4,766,243.06	-54,084,697.49
	7	4,766,243.06	-49,318,454.43
	8	4,766,243.06	-44,552,211.37
	9	4,766,243.06	-39,785,968.31
	10	4,766,243.06	-35,019,725.25
	11	4,766,243.06	-30,253,482.19
	12	4,766,243.06	-25,487,239.13
Forth year	1	5,481,179.52	-20,006,059.61
	2	5,481,179.52	-14,524,880.09
	3	5,481,179.52	-9,043,700.57
	4	5,481,179.52	-3,562,521.05
	5	5,481,179.52	1,918,658.47

The recovery period of the investment is three years and five months. When ±10% of investment price variation in the market is considered, the anticipated variation in recovery period is calculated as follows,

10 % of LKR 175,664,440.00 investment = LKR 17,566,444.00

During the 4th year the monthly return = LKR 5,481,179.52

Recovery period due to 10% price variation $= \frac{\text{LKR } 17{,}566{,}444.00}{\text{LKR } 5{,}481{,}179.52}$

= ±3.20 Months

When the ±10 % of investment price variation is expected in the market there will be around three months of variation in the calculated recovery period of the investment.

Chapter 6

Conclusion and Discussion

The overall objective of this research is to study potentiality of introducing absorption chiller systems to improve the diesel power plant performance in Sri Lanka. In the sense it was identified that the diesel engine performance could be drown back due to the tropical site conditions.

The below mentioned three streams were considered with the intention of introducing absorption chiller cooling systems in order to improve diesel power plant performance.

1. Specific fuel oil consumption
2. Engine deration
3. Building cooling

Three issues mentioned above have to be overcome by utilizing waste heat via vapour absorption chiller to improve the diesel engine performance. The current engine performance at site conditions was evaluated. Therefore, according to the prepared scenario, the specific fuel oil consumption and engine deration was studied in UJPS.

In the case study it clearly shows that the engines at UJPS are having high fuel oil consumptions in comparison to the ISO conditions due to the prevailing site conditions at UJPS. The engine deration is not a considerable issue because the site conditions are not that much worse for a considerable deration for the engines. However, the specific fuel oil consumption is considerably high. In addition, building cooling load has been integrated to the absorption chiller cooling system to accumulate the benefits.

In order to introduce the vapour absorption chiller to the diesel power plant the identified cooling loads were as given below

- Combustion air conditioning to the ISO standard site conditions
- Indoor air conditioning
- Charge air cooling water inlet temperature reducing

If we cater the required total load by means of a waste heat absorption chiller the payback period of the investment was 3 years and 5 months and also the monthly saving was LKR 3,603,964.50. The price variation of the investment was given as ±10% and so there could be variation of payback period by three months from the above calculated result.

According to the above calculations, it was noted that considerable advantages can be gained by introducing an absorption chiller system to improve the Uthuru Janani diesel power plant performance in Jaffna peninsula.

Finally from the model it was observed that when the ambient conditions are much worse the potentiality of improving the diesel power plant engine performance by introducing vapour absorption chiller is high. It is in terms of both the SFC and engine deration. As a further requirement in depth study incorporating more precise cost of sub-systems could be carried out in view of the economic feasibility.

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Appendix A: Graphical interpretation of Table 4.2 in Mathlab software

Table 4.2: Percentage of deration Vs percentage of humidity at constant atmospheric temperatures [30]

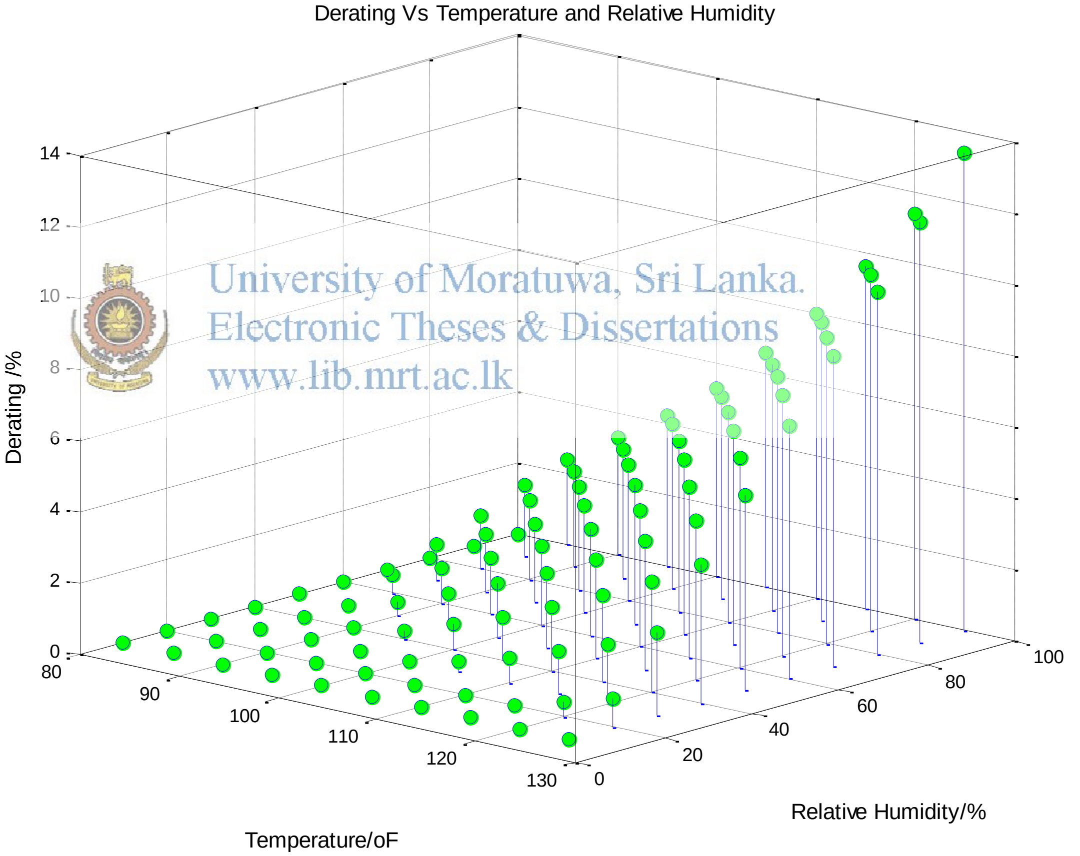
Atmospheric Temperature	Percentage Humidity									
	10	20	30	40	50	60	70	80	90	100
	Percentage Derating									
85						0.5	1.0	1.5	2.0	2.4
90					0.4	1.0	1.6	2.2	2.7	3.3
95				0.2	0.9	1.6	2.2	2.9	3.6	4.2
100				0.7	1.5	2.2	3.0	3.8	4.6	5.3
105			0.3	1.2	2.1	3.0	3.9	4.8	5.7	6.6
110			0.7	1.8	2.8	3.8	4.9	5.9	6.9	8.0
115			1.2	2.4	3.6	4.8	6.0	7.2	8.4	9.6
120		0.4	1.7	3.1	4.5	5.9	7.3	8.6	10.0	11.4
125		0.8	2.3	3.9	5.5	7.1	8.7	10.2	11.8	13.4

```
X =
    80    80    80    80    80    80    80    80    80    80
    85    85    85    85    85    85    85    85    85    85
    90    90    90    90    90    90    90    90    90    90
    95    95    95    95    95    95    95    95    95    95
   100   100   100   100   100   100   100   100   100   100
   105   105   105   105   105   105   105   105   105   105
   110   110   110   110   110   110   110   110   110   110
   115   115   115   115   115   115   115   115   115   115
   120   120   120   120   120   120   120   120   120   120
   125   125   125   125   125   125   125   125   125   125

y =
    10    20    30    40    50    60    70    80    90   100
    10    20    30    40    50    60    70    80    90   100
    10    20    30    40    50    60    70    80    90   100
    10    20    30    40    50    60    70    80    90   100
    10    20    30    40    50    60    70    80    90   100
    10    20    30    40    50    60    70    80    90   100
    10    20    30    40    50    60    70    80    90   100
    10    20    30    40    50    60    70    80    90   100
    10    20    30    40    50    60    70    80    90   100
    10    20    30    40    50    60    70    80    90   100
```

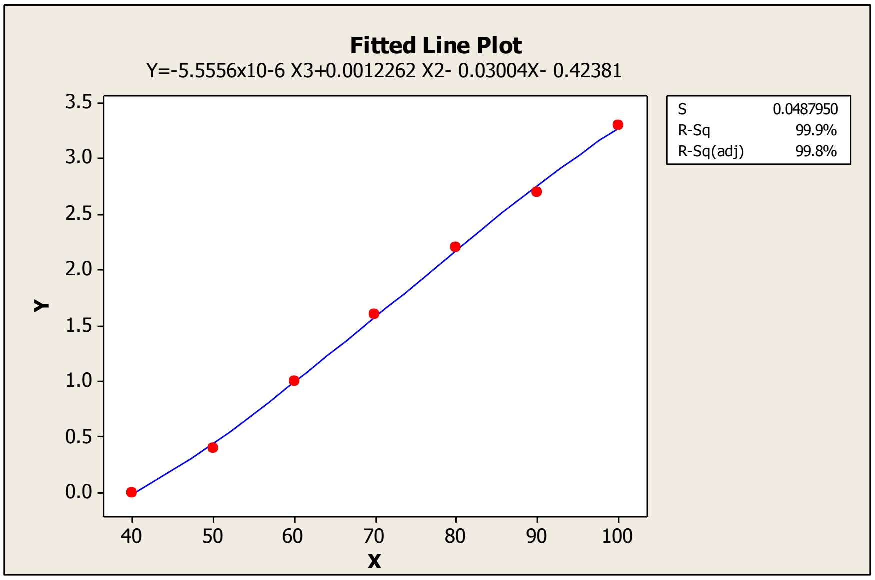
```
z =
     0        0        0        0        0        0         0        0        0         0
     0        0        0        0        0   0.5000    1.0000   1.5000   2.0000    2.4000
     0        0        0        0   0.4000   1.0000    1.6000   2.2000   2.7000    3.3000
     0        0        0   0.2000   0.9000   1.6000    2.2000   2.9000   3.6000    4.2000
     0        0        0   0.7000   1.5000   2.2000    3.0000   3.8000   4.6000    5.3000
     0        0   0.3000   1.2000   2.1000   3.0000    3.9000   4.8000   5.7000    6.6000
     0        0   0.7000   1.8000   2.8000   3.8000    4.9000   5.9000   6.9000    8.0000
     0        0   1.2000   2.4000   3.6000   4.8000    6.0000   7.2000   8.4000    9.6000
     0   0.4000   1.7000   3.1000   4.5000   5.9000    7.3000   8.6000  10.0000   11.4000
     0   0.8000   2.3000   3.9000   5.5000   7.1000    8.7000  10.2000  11.8000   13.4000

>> stem3(x,y,z,'MarkerFaceColor','g')
>> grid on
```



Appendix B: Calculations of the model

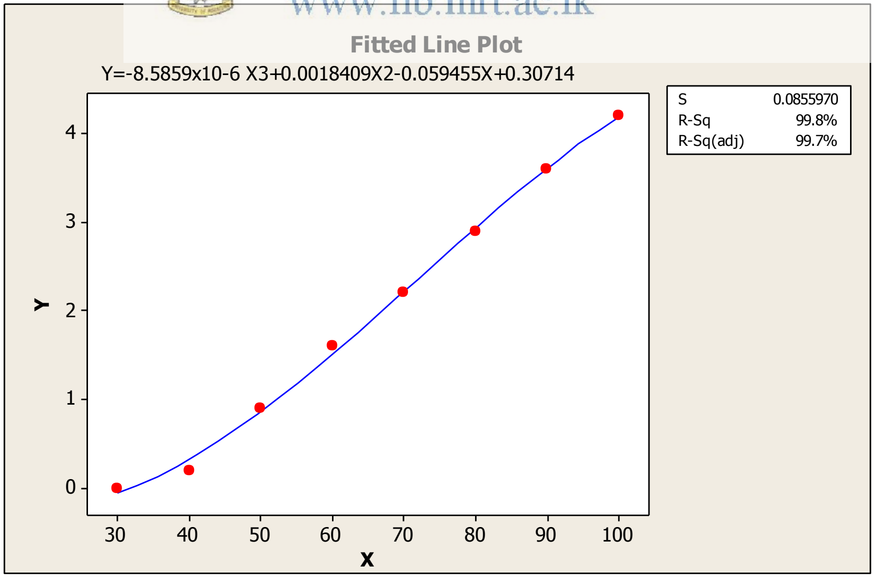
When the temperature is maintaining at 90^{o}F,



Y=-5.5556x10-6 X^3+0.0012262 X^2- 0.03004X- 0.42381------------> T90

Regression Analysis is S = 0.0487950 R^2= 99.9% R^2 (adj) = 99.8% P= 0.000

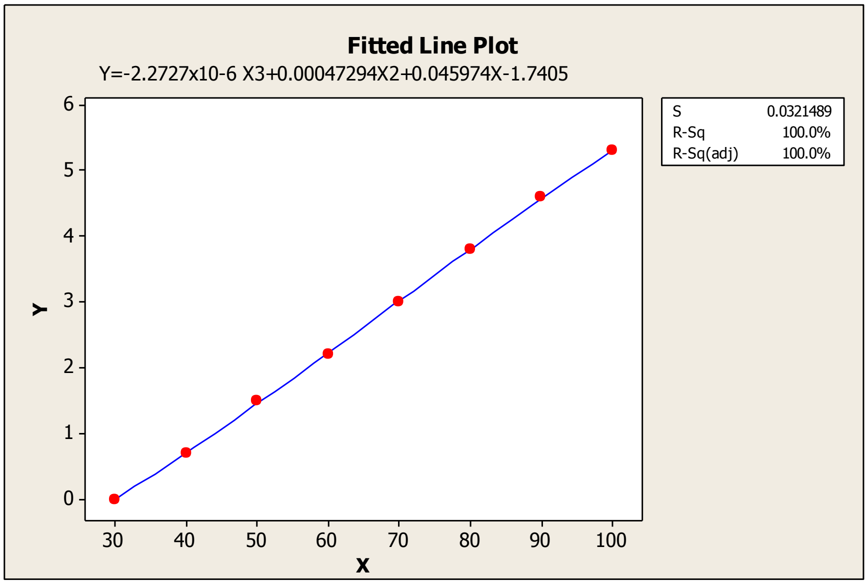
When the temperature is maintaining at 95^{o}F,



Y=-8.5859x10-6 X^3+0.0018409X^2-0.059455X+0.30714------------> T95

Regression Analysis is S = 0.0855970 R^2= 99.8% R^2 (adj) = 99.7% P= 0.000

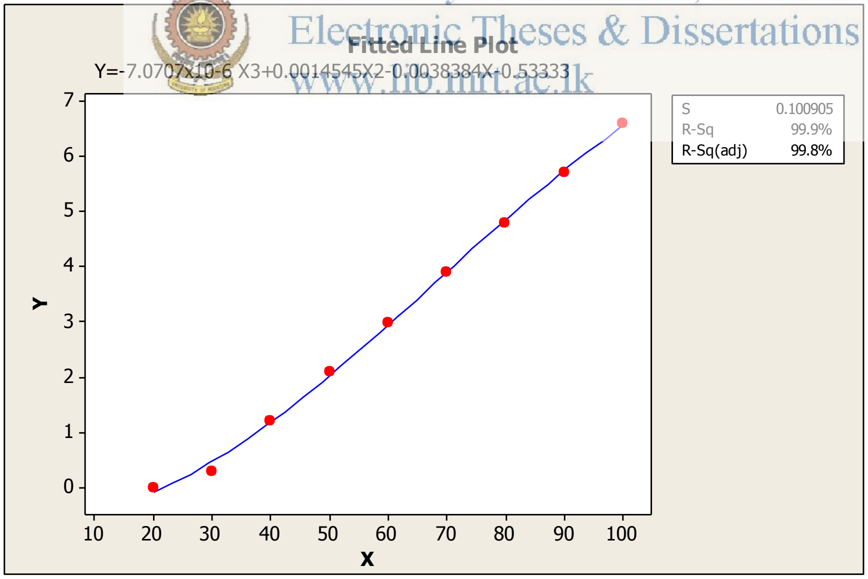
When the temperature is maintaining at 100°F,



Y=-2.2727x10-6 X^3+0.00047294X^2+0.045974X-1.7405----------> T100

Regression Analysis is S = 0.0321489 R^2 = 100.0% R^2 (adj) = 100.0% P= 0.000

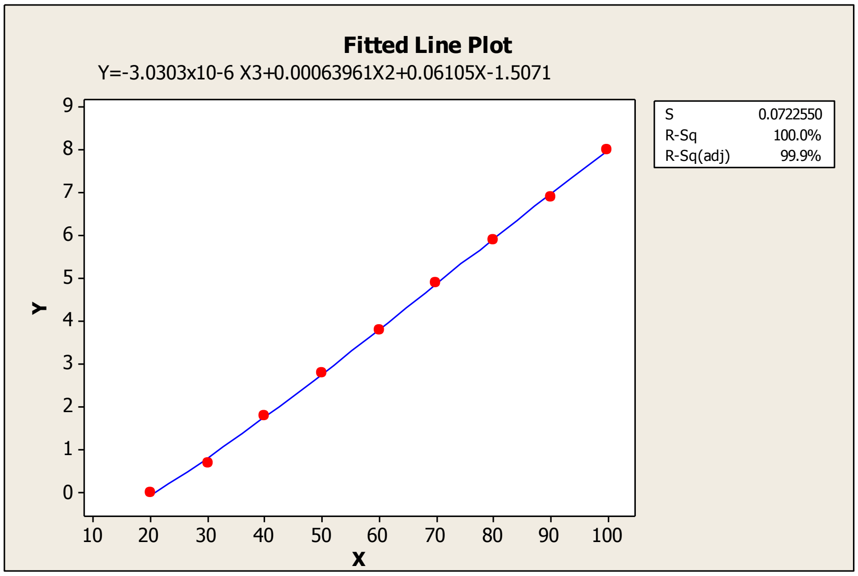
When the temperature is maintaining at 105°F,



Y=-7.0707x10-6 X^3+0.0014545X^2-0.0038384X-0.53333---------> T105

Regression Analysis is S = 0.100905 R^2= 99.9% R^2 (adj) = 99.8% P= 0.000

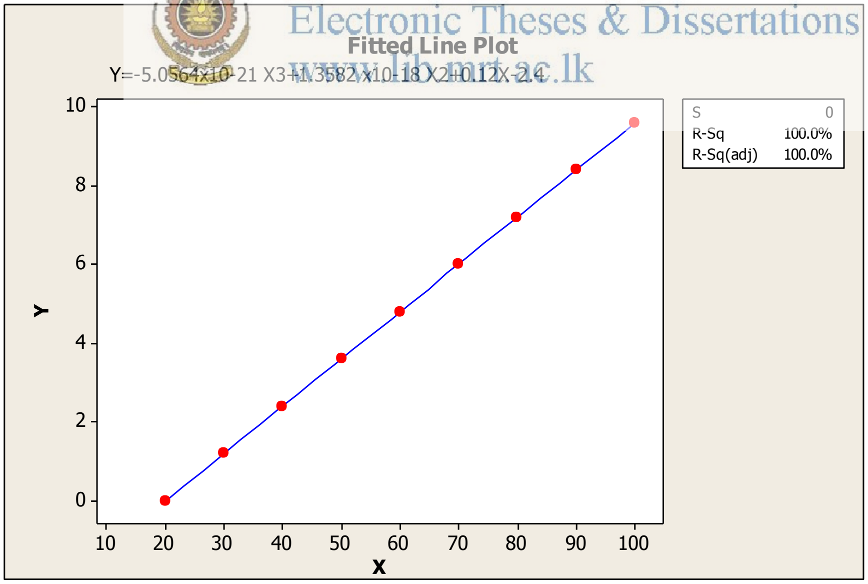
When the temperature is maintaining at 110°F,



Y=-3.0303x10-6 X^3+0.00063961X^2+0.06105X-1.5071-----------> T110

Regression Analysis is S = 0.0722550 R^2= 100.0% R^2 (adj) = 99.9% P= 0.000

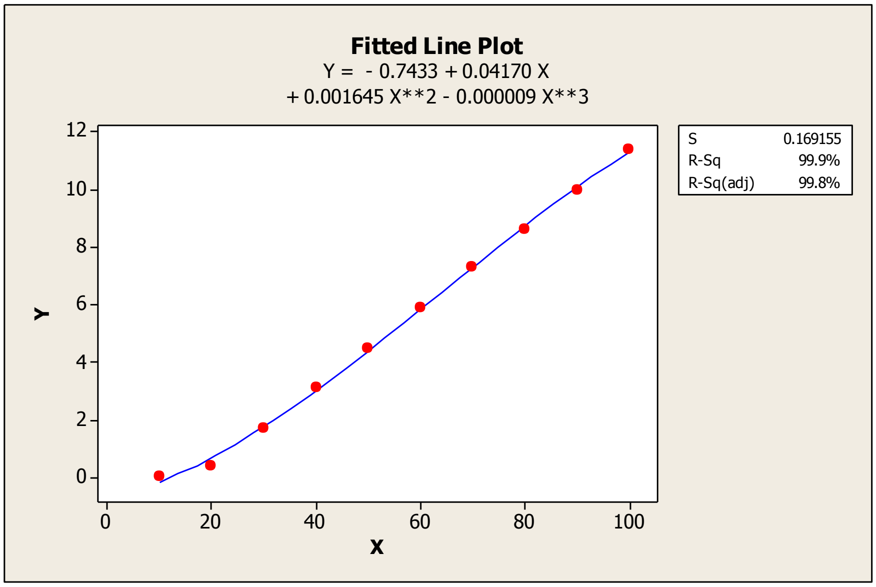
When the temperature is maintaining at 115°F,



Y=-5.0564x10-21 X^3+1.3582 x10-18 X^2+0.12X-2.4---------------> T115

Regression Analysis is S = 0 R^2= 100.0% R^2 (adj) = 100.0% P= 0.000

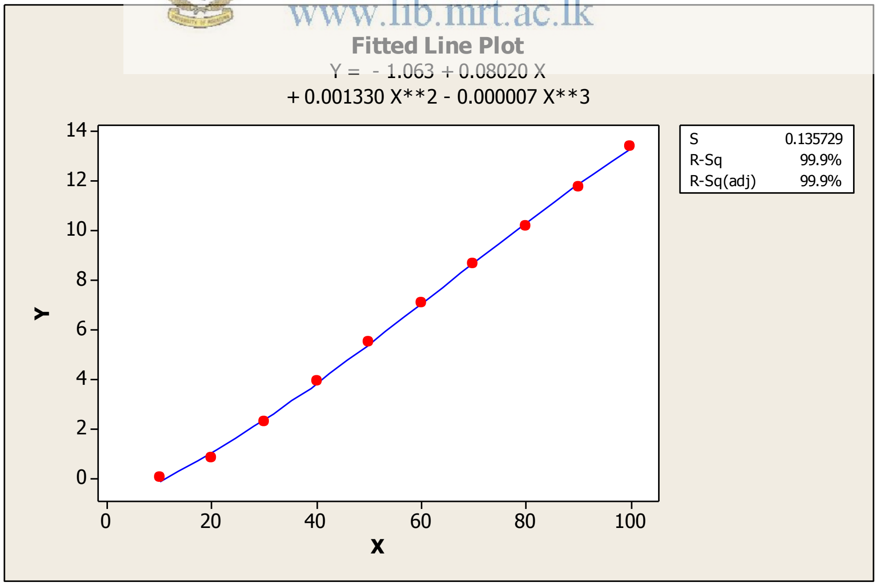
When the temperature is maintaining at 120^{o}F,



Y=-8.5664x10-6 X^{3}+0.0016445X^{2}+0.041696X-0.74333----------> T120

Regression Analysis is S = 0.169155 R^{2} = 99.9% R^{2} (adj) = 99.8% P= 0.000

When the temperature is maintaining at 125^{o}F,



Y=-6.9347x10-6 X^{3}+0.0013298X^{2}+0.080204X-1.0633------------> T125

Regression Analysis is S = 0.135729 R^{2} = 99.9% R^{2} (adj) = 99.9% P= 0.000

Appendix C: Scatter plot of measured temperature and RH points using Matlab software

```
x =
Columns 1 through 15
 82.0400   81.6800   81.5000   81.1400   81.1400   80.9600   81.3200   84.3800   86.3600
89.7800   93.5600   93.5600   93.9200   96.6200   94.8200
Columns 16 through 30
93.2000   90.6800   87.4400   85.1000   84.3800   83.6600   82.9400   82.5800   82.0400
81.8600   81.3200   81.3200   81.1400   80.9600   81.1400
Columns 31 through 45
81.3200   84.2000   89.2400   89.2400   89.7800   90.3200   90.8600   90.8600   93.7400
93.7400   90.5000   88.1600   85.2800   83.8400   83.3000
Columns 46 through 48
82.7600   82.4000   82.2200

y =
Columns 1 through 15
70.4000   78.8000   80.1000   81.1000   79.1000   81.1000   80.2000   73.6000   69.4000
61.6000   54.6000   54.6000   53.5000   44.5000   46.8000
Columns 16 through 30
50.7000   59.8000   65.4000   72.3000   69.9000   72.7000   74.4000   75.7000   77.5000
80.0000   80.6000   80.6000   81.1000   80.2000   81.9000
Columns 31 through 45
 82.2000   75.2000   64.3000   64.2000   59.7000   58.4000   51.6000   50.0000   44.5000
47.1000   59.3000   66.2000   69.4000   72.3000   75.5000
Columns 46 through 48
77.2000   77.8000   78.6000

>> scatter (x,y,'+','r')
```

```
p =

Columns 1 through 15

82.4000   82.0400   81.8600   81.5000   81.5000   81.3200   81.6800   84.7400   86.7200

90.1400   93.9200   93.9200   94.2800   96.9800   95.1800

Columns 16 through 30

93.5600   91.0400   87.8000   85.4600   84.7400   84.0200   83.3000   82.9400   82.4000

82.2200   81.6800   81.6800   81.5000   81.5000   81.5000

Columns 31 through 45

81.6800   84.5600   89.4200   89.6000   90.1400   90.6800   91.2200   91.2200   94.1000

94.1000   90.8600   88.5200   85.6400   84.2000   83.6600

Columns 46 through 48

83.1200   82.7600   82.5800


q =

Columns 1 through 15

72.5000   80.9000   82.2000   83.2000   81.2000   83.2000   82.3000   75.6000   71.4000

63.6000   56.6000   56.6000   55.5000   46.5000   48.8000

Columns 16 through 30

52.7000   61.8000   67.4000   74.3000   71.9000   74.7000   76.4000   77.7000   79.5000

82.0000   82.6000   82.6000   83.1000   82.2000   83.9000

Columns 31 through 45

84.2000   77.2000   66.3000   66.2000   61.7000   60.4000   53.6000   52.0000   46.5000

49.1000   61.3000   68.2000   71.4000   74.3000   77.5000

Columns 46 through 48

79.2000   79.8000   80.6000


>> scatter (p,q,'+','g')
```

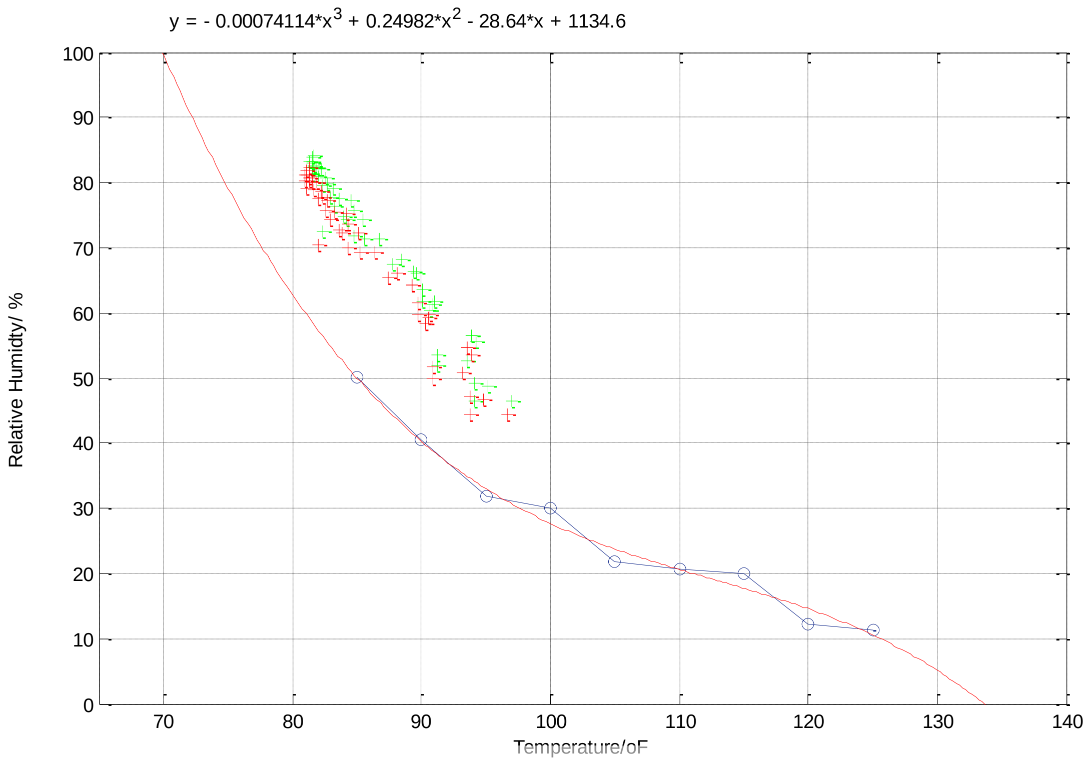


Figure 1: Scatter plot of measured temperature and RH points with permissible value of the instrument using Matlab software

Appendix D: Psychrometric chart enthalpy calculations

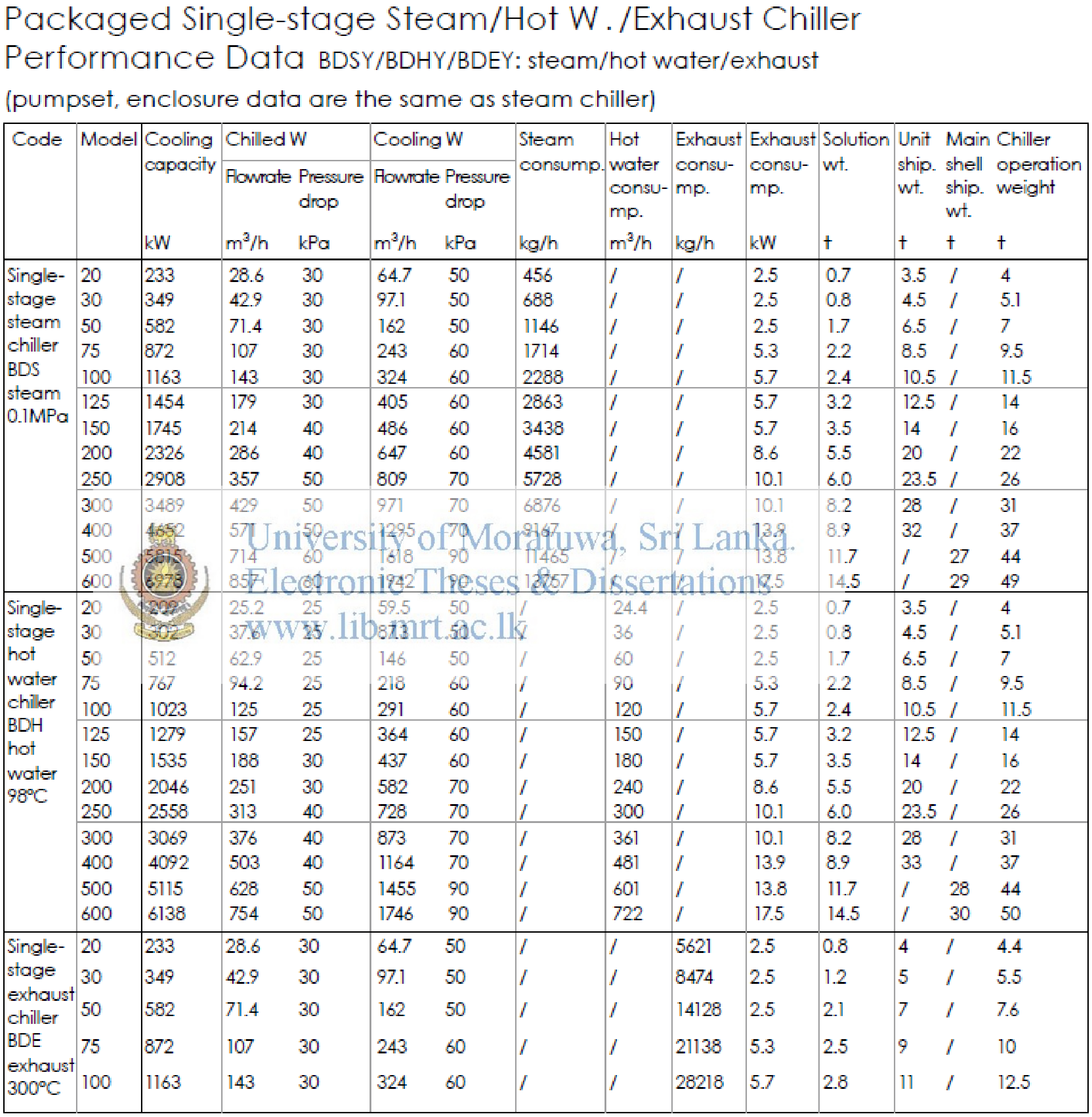
(For 28^{o}C and 72.5 % RH)

Appendix E: Psychrometric chart cooling load calculations

(For 25^{o}C and 30 % RH and 34.4^{o}C and 56.6 % RH)

Appendix F: Hot water chiller performance data

NOTE: Extracted from Model Selection and Design manual of Broad Central Air Conditioning (Absorption LiBr+H2O), Broad X Non- Electric Chiller



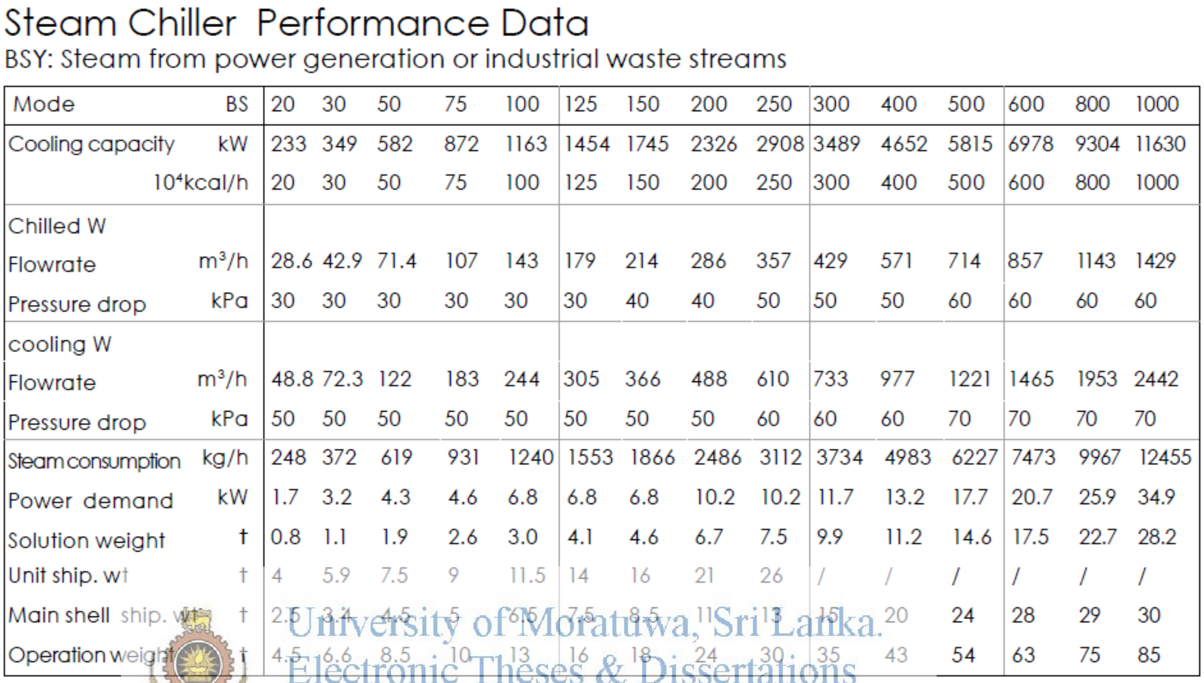
Packaged Single-stage Steam/Hot W./Exhaust Chiller Performance Data BDSY/BDHY/BDEY: steam/hot water/exhaust

(pumpset, enclosure data are the same as steam chiller)

Code	Model	Cooling capacity	Chilled W		Cooling W		Steam consump.	Hot water consu-mp.	Exhaust consu-mp.	Exhaust consu-mp.	Solution wt.	Unit ship. wt.	Main shell ship. wt.	Chiller operation weight
			Flowrate	Pressure drop	Flowrate	Pressure drop								
		kW	m^3/h	kPa	m^3/h	kPa	kg/h	m^3/h	kg/h	kW	t	t	t	t
Single-stage steam chiller BDS steam 0.1MPa	20	233	28.6	30	64.7	50	456	/	/	2.5	0.7	3.5	/	4
	30	349	42.9	30	97.1	50	688	/	/	2.5	0.8	4.5	/	5.1
	50	582	71.4	30	162	50	1146	/	/	2.5	1.7	6.5	/	7
	75	872	107	30	243	60	1714	/	/	5.3	2.2	8.5	/	9.5
	100	1163	143	30	324	60	2288	/	/	5.7	2.4	10.5	/	11.5
	125	1454	179	30	405	60	2863	/	/	5.7	3.2	12.5	/	14
	150	1745	214	40	486	60	3438	/	/	5.7	3.5	14	/	16
	200	2326	286	40	647	60	4581	/	/	8.6	5.5	20	/	22
	250	2908	357	50	809	70	5728	/	/	10.1	6.0	23.5	/	26
	300	3489	429	50	971	70	6876	/	/	10.1	8.2	28	/	31
	400	4652	571	50	1295	70	9167	/	/	13.9	8.9	32	/	37
	500	5815	714	60	1618	90	11465	/	/	13.8	11.7	/	27	44
	600	6978	857	60	1942	90	12757	/	/	17.5	14.5	/	29	49
Single-stage hot water chiller BDH hot water 98°C	20	209	25.2	25	59.5	50	/	24.4	/	2.5	0.7	3.5	/	4
	30	302	37.6	25	87.3	50	/	36	/	2.5	0.8	4.5	/	5.1
	50	512	62.9	25	146	50	/	60	/	2.5	1.7	6.5	/	7
	75	767	94.2	25	218	60	/	90	/	5.3	2.2	8.5	/	9.5
	100	1023	125	25	291	60	/	120	/	5.7	2.4	10.5	/	11.5
	125	1279	157	25	364	60	/	150	/	5.7	3.2	12.5	/	14
	150	1535	188	30	437	60	/	180	/	5.7	3.5	14	/	16
	200	2046	251	30	582	70	/	240	/	8.6	5.5	20	/	22
	250	2558	313	40	728	70	/	300	/	10.1	6.0	23.5	/	26
	300	3069	376	40	873	70	/	361	/	10.1	8.2	28	/	31
	400	4092	503	40	1164	70	/	481	/	13.9	8.9	33	/	37
	500	5115	628	50	1455	90	/	601	/	13.8	11.7	/	28	44
	600	6138	754	50	1746	90	/	722	/	17.5	14.5	/	30	50
Single-stage exhaust chiller BDE exhaust 300°C	20	233	28.6	30	64.7	50	/	/	5621	2.5	0.8	4	/	4.4
	30	349	42.9	30	97.1	50	/	/	8474	2.5	1.2	5	/	5.5
	50	582	71.4	30	162	50	/	/	14128	2.5	2.1	7	/	7.6
	75	872	107	30	243	60	/	/	21138	5.3	2.5	9	/	10
	100	1163	143	30	324	60	/	/	28218	5.7	2.8	11	/	12.5

Appendix G: Steam chiller performance data

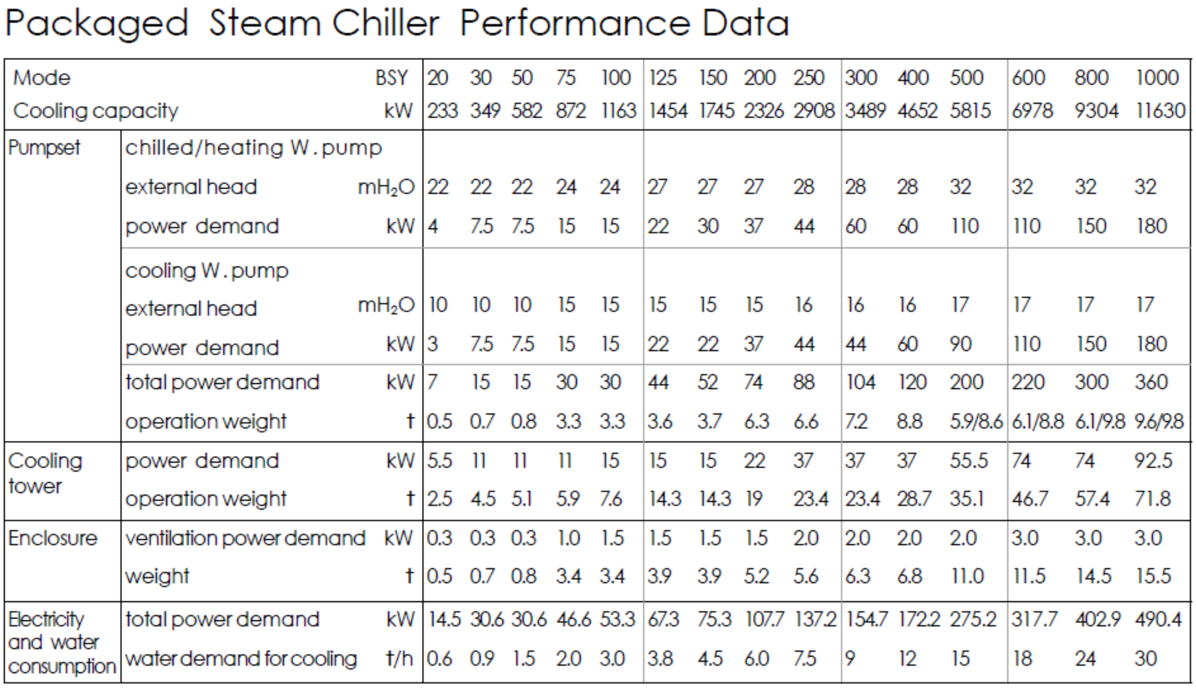
NOTE: Extracted from Model Selection and Design manual of Broad Central Air Conditioning (Absorption LiBr+H2O), Broad X Non- Electric Chiller



Steam Chiller Performance Data

BSY: Steam from power generation or industrial waste streams

Mode	BS	20	30	50	75	100	125	150	200	250	300	400	500	600	800	1000
Cooling capacity	kW	233	349	582	872	1163	1454	1745	2326	2908	3489	4652	5815	6978	9304	11630
	10^4kcal/h	20	30	50	75	100	125	150	200	250	300	400	500	600	800	1000
Chilled W																
Flowrate	m^3/h	28.6	42.9	71.4	107	143	179	214	286	357	429	571	714	857	1143	1429
Pressure drop	kPa	30	30	30	30	30	30	40	40	50	50	50	60	60	60	60
cooling W																
Flowrate	m^3/h	48.8	72.3	122	183	244	305	366	488	610	733	977	1221	1465	1953	2442
Pressure drop	kPa	50	50	50	50	50	50	50	50	60	60	60	70	70	70	70
Steam consumption	kg/h	248	372	619	931	1240	1553	1866	2486	3112	3734	4983	6227	7473	9967	12455
Power demand	kW	1.7	3.2	4.3	4.6	6.8	6.8	6.8	10.2	10.2	11.7	13.2	17.7	20.7	25.9	34.9
Solution weight	t	0.8	1.1	1.9	2.6	3.0	4.1	4.6	6.7	7.5	9.9	11.2	14.6	17.5	22.7	28.2
Unit ship. wt	t	4	5.9	7.5	9	11.5	14	16	21	26	/	/	/	/	/	/
Main shell ship. wt	t	2.5	3.4	4.5	5	6.5	7.5	8.5	11	13	15	20	24	28	29	30
Operation weight	t	4.5	6.6	8.5	10	13	16	18	24	30	35	43	54	63	75	85

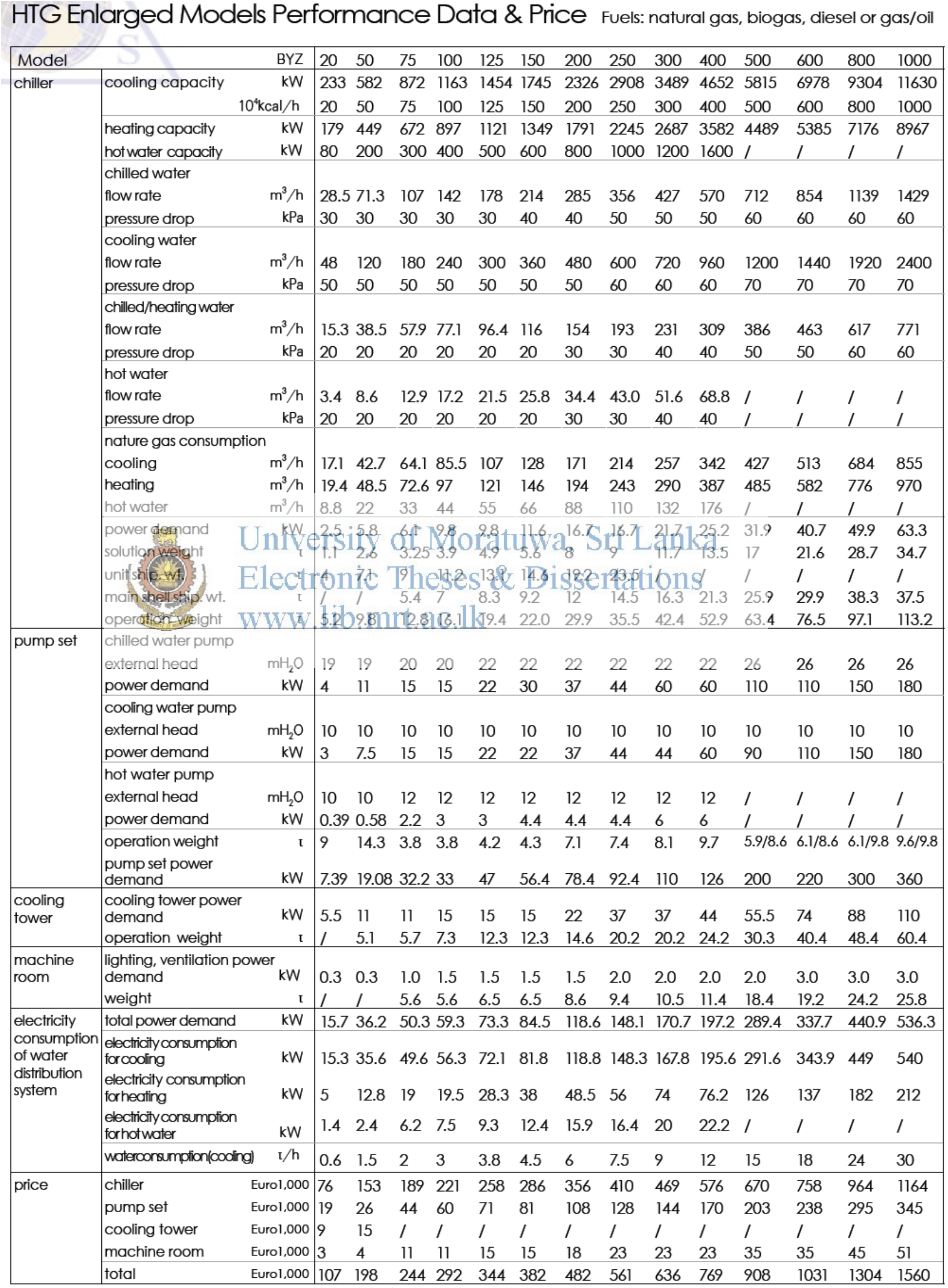


Packaged Steam Chiller Performance Data

Mode		BSY	20	30	50	75	100	125	150	200	250	300	400	500	600	800	1000
Cooling capacity		kW	233	349	582	872	1163	1454	1745	2326	2908	3489	4652	5815	6978	9304	11630
Pumpset	chilled/heating W.pump																
	external head	mH_2O	22	22	22	24	24	27	27	27	28	28	28	32	32	32	32
	power demand	kW	4	7.5	7.5	15	15	22	30	37	44	60	60	110	110	150	180
	cooling W.pump																
	external head	mH_2O	10	10	10	15	15	15	15	15	16	16	16	17	17	17	17
	power demand	kW	3	7.5	7.5	15	15	22	22	37	44	44	60	90	110	150	180
	total power demand	kW	7	15	15	30	30	44	52	74	88	104	120	200	220	300	360
	operation weight	t	0.5	0.7	0.8	3.3	3.3	3.6	3.7	6.3	6.6	7.2	8.8	5.9/8.6	6.1/8.8	6.1/9.8	9.6/9.8
Cooling tower	power demand	kW	5.5	11	11	11	15	15	15	22	37	37	37	55.5	74	74	92.5
	operation weight	t	2.5	4.5	5.1	5.9	7.6	14.3	14.3	19	23.4	23.4	28.7	35.1	46.7	57.4	71.8
Enclosure	ventilation power demand	kW	0.3	0.3	0.3	1.0	1.5	1.5	1.5	1.5	2.0	2.0	2.0	2.0	3.0	3.0	3.0
	weight	t	0.5	0.7	0.8	3.4	3.4	3.9	3.9	5.2	5.6	6.3	6.8	11.0	11.5	14.5	15.5
Electricity and water consumption	total power demand	kW	14.5	30.6	30.6	46.6	53.3	67.3	75.3	107.7	137.2	154.7	172.2	275.2	317.7	402.9	490.4
	water demand for cooling	t/h	0.6	0.9	1.5	2.0	3.0	3.8	4.5	6.0	7.5	9	12	15	18	24	30

Appendix H: Chiller performance data and prices

NOTE: Extracted from Model Selection & Design Manual of Broad X Absorption Chiller.



HTG Enlarged Models Performance Data & Price

Fuels: natural gas, biogas, diesel or gas/oil

Model		BYZ	20	50	75	100	125	150	200	250	300	400	500	600	800	1000
chiller	cooling capacity	kW	233	582	872	1163	1454	1745	2326	2908	3489	4652	5815	6978	9304	11630
		10^4kcal/h	20	50	75	100	125	150	200	250	300	400	500	600	800	1000
	heating capacity	kW	179	449	672	897	1121	1349	1791	2245	2687	3582	4489	5385	7176	8967
	hot water capacity	kW	80	200	300	400	500	600	800	1000	1200	1600	/	/	/	/
	chilled water															
	flow rate	m^3/h	28.5	71.3	107	142	178	214	285	356	427	570	712	854	1139	1429
	pressure drop	kPa	30	30	30	30	30	40	40	50	50	50	60	60	60	60
	cooling water															
	flow rate	m^3/h	48	120	180	240	300	360	480	600	720	960	1200	1440	1920	2400
	pressure drop	kPa	50	50	50	50	50	50	50	60	60	60	70	70	70	70
	chilled/heating water															
	flow rate	m^3/h	15.3	38.5	57.9	77.1	96.4	116	154	193	231	309	386	463	617	771
	pressure drop	kPa	20	20	20	20	20	20	30	30	40	40	50	50	60	60
	hot water															
	flow rate	m^3/h	3.4	8.6	12.9	17.2	21.5	25.8	34.4	43.0	51.6	68.8	/	/	/	/
	pressure drop	kPa	20	20	20	20	20	20	30	30	40	40	/	/	/	/
	nature gas consumption															
	cooling	m^3/h	17.1	42.7	64.1	85.5	107	128	171	214	257	342	427	513	684	855
	heating	m^3/h	19.4	48.5	72.6	97	121	146	194	243	290	387	485	582	776	970
	hot water	m^3/h	8.8	22	33	44	55	66	88	110	132	176	/	/	/	/
	power demand	kW	2.5	5.8	6.1	9.8	9.8	11.6	16.7	16.7	21.7	25.2	31.9	40.7	49.9	63.3
	solution weight	t	1.1	2.6	3.25	3.9	4.9	5.6	8	9	11.7	13.5	17	21.6	28.7	34.7
	unit ship. wt.	t	4	7.1	9	11.2	13.8	14.6	18.2	23.5	/	/	/	/	/	/
	main shell ship. wt.	t	/	/	5.4	7	8.3	9.2	12	14.5	16.3	21.3	25.9	29.9	38.3	37.5
	operation weight	t	5.2	9.8	12.8	16.1	19.4	22.0	29.9	35.5	42.4	52.9	63.4	76.5	97.1	113.2
pump set	chilled water pump															
	external head	mH_2O	19	19	20	20	22	22	22	22	22	22	26	26	26	26
	power demand	kW	4	11	15	15	22	30	37	44	60	60	110	110	150	180
	cooling water pump															
	external head	mH_2O	10	10	10	10	10	10	10	10	10	10	10	10	10	10
	power demand	kW	3	7.5	15	15	22	22	37	44	44	60	90	110	150	180
	hot water pump															
	external head	mH_2O	10	10	12	12	12	12	12	12	12	12	/	/	/	/
	power demand	kW	0.39	0.58	2.2	3	3	4.4	4.4	4.4	6	6	/	/	/	/
	operation weight	t	9	14.3	3.8	3.8	4.2	4.3	7.1	7.4	8.1	9.7	5.9/8.6	6.1/8.6	6.1/9.8	9.6/9.8
	pump set power demand	kW	7.39	19.08	32.2	33	47	56.4	78.4	92.4	110	126	200	220	300	360
cooling tower	cooling tower power demand	kW	5.5	11	11	15	15	15	22	37	37	44	55.5	74	88	110
	operation weight	t	/	5.1	5.7	7.3	12.3	12.3	14.6	20.2	20.2	24.2	30.3	40.4	48.4	60.4
machine room	lighting, ventilation power demand	kW	0.3	0.3	1.0	1.5	1.5	1.5	1.5	2.0	2.0	2.0	2.0	3.0	3.0	3.0
	weight	t	/	/	5.6	5.6	6.5	6.5	8.6	9.4	10.5	11.4	18.4	19.2	24.2	25.8
electricity consumption of water distribution system	total power demand	kW	15.7	36.2	50.3	59.3	73.3	84.5	118.6	148.1	170.7	197.2	289.4	337.7	440.9	536.3
	electricity consumption for cooling	kW	15.3	35.6	49.6	56.3	72.1	81.8	118.8	148.3	167.8	195.6	291.6	343.9	449	540
	electricity consumption for heating	kW	5	12.8	19	19.5	28.3	38	48.5	56	74	76.2	126	137	182	212
	electricity consumption for hot water	kW	1.4	2.4	6.2	7.5	9.3	12.4	15.9	16.4	20	22.2	/	/	/	/
	waterconsumption(cooling)	t/h	0.6	1.5	2	3	3.8	4.5	6	7.5	9	12	15	18	24	30
price	chiller	Euro1,000	76	153	189	221	258	286	356	410	469	576	670	758	964	1164
	pump set	Euro1,000	19	26	44	60	71	81	108	128	144	170	203	238	295	345
	cooling tower	Euro1,000	9	15	/	/	/	/	/	/	/	/	/	/	/	/
	machine room	Euro1,000	3	4	11	11	15	15	18	23	23	23	35	35	45	51
	total	Euro1,000	107	198	244	292	344	382	482	561	636	769	908	1031	1304	1560