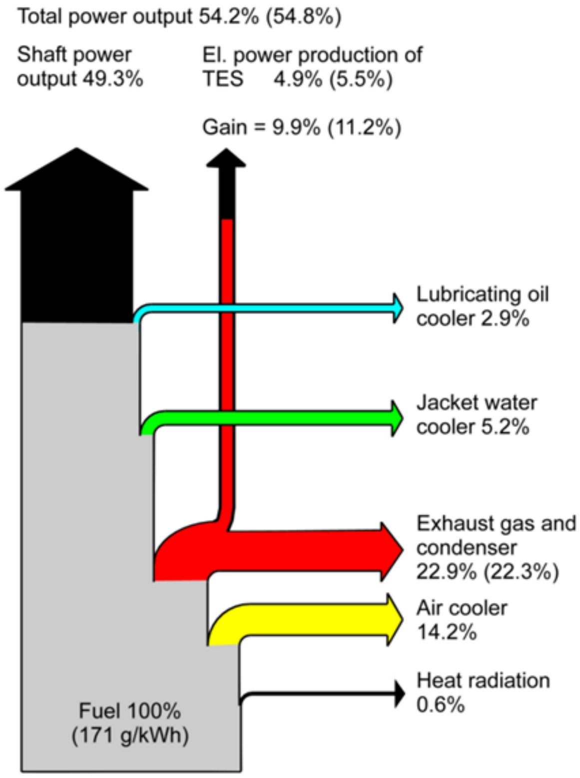
KTH Industrial Engineering
and Management

A waste heat recovery steam power generation system for ACE Power Embilipitiya (Pvt) Ltd, Sri Lanka

Udayani Priyadarshana Weerasiri



Master of Science Thesis

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KTH Industrial Engineering and Management	**Master of Science Thesis EGI-2014-053MSC EKV1034** **A waste heat recovery steam power generation system for ACE Power Embilipitiya (Pvt) Ltd, Sri Lanka** Udayani Priyadarshana Weerasiri

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Abstract

In this study, the heat recovery from exhaust gas at the ACE Power Embilipitiya (Pvt) Ltd (APE) in Sri Lanka was conceptually proposed and evaluated. APE has an installed capacity of 100 MW comprising 14 units of 7.5MW medium speed diesel engines fired with heavy fuel oil. There is only a minimum recovery of waste heat in the plant at the moment, only for fuel preheating, whereas waste heat recovery (WHR) boilers of 750kW$_{th}$ are equipped on eight engines. The larger portion of the waste heat is dumped into the environment without being used in any reasonable way.

The objective of this work was to design a HRSG system for the remaining six engines to recover maximum possible heat from the exhaust gas and select a suitable steam turbine according to the heat demand capacity of the proposed HRSG, for generating additional power and thus converting the APE plant into a sort of a combined cycle.

At the initial stage of the investigation, the amount of recoverable waste heat was estimated by evaluating the known parameters of the engines at fully loaded condition. The maximum theoretical waste heat recovery potential from the exhaust gas stream of one engine was calculated as 9807.87 MJ/h, equivalent to a heat rate of 2724.4 kW.

The modelling and optimization of the proposed HRSG was done using the Engineering Equation Solver (EES) software, considering technical and practical limitations such as pinch point temperature difference, approach point temperature difference, terminal temperature difference and sulphur dew point in the stack.

A commercially available steam turbine with a power output of 3.579 MW was selected as the optimum steam turbine for the desired conditions, utilising 12884.4 MJ/h of recovered waste energy amounting to 21.89% of the total available energy in the flue gas.

Acknowledgment

My sincere gratitude goes to Dr.N.S.Senanayake, Mr Ruchira Abeyweera and Ms I.U Attanayake for their continued support and guidance for the completion of this thesis.

I thank the staff of Ace Power Embilipitiya (Pvt) Ltd, the staff of ICBT, OUSL and all the friends and colleagues who supported and encouraged for the accomplishment of this task.

Last but not least I thank my parents and family for their support and guiding me spiritually.

List of Abbreviations

APE	Ace Power Embilipitiya (Pvt)Ltd
API	American Petroleum Institute deg.
CPC	Ceylon Petroleum Cooperation
CHP	Combine heat and power
EE	Energy Efficient
EES	Engineering Equation Solver
GHG	Greenhouse gases
HHV	Higher Heating Value
HRSG	Heat recovery steam generator
LHV	Lower heating Value
$\dot{m}$	Mass flow
P	Power
Q	Energy
s	Specific gravity
S	Sulphur content by weight
t	Temperature
WHR	Waste heat recovery
IPP	Independent Power Producers

Subscripts

appr	approach
e	electrical
fw	feed water
gas	Exhaust gas
is	isentropic
l	losses
pp	pinch point
sh	superheated steam
st	steam

Contents

1 Introduction

1.1 General overview

ACE Power Embilipitiya (Pvt) Ltd (APE) is an independent power producer in Sri Lanka with a capacity of 100 MW net electrical power output, supplying electricity to the national grid with a minimum guaranteed energy amount of 697.7 GWh per year, which is equivalent to nearly 7000 hours per year of full-load operation.

The power plant was commissioned in 2005 and operates on a 10-year power purchasing agreement with Ceylon Electricity Board (CEB). The plant is equipped with 14 Caterpillar 16CM32C engine-generator units running on heavy fuel oil, each with a rated power capacity of 7.5 MW; and the generated electricity is supplied to the Embilipitiya grid station of CEB via a 132 kV transmission line. The entire operations and maintenance work of the power plant is carried out with local expertise under ISO 9001:2008, ISO 14001:2004, OHSAS 18001 management systems and 5S industrial housekeeping standard.

The overall efficiency of the energy conversion typically falls below 40% with much of the energy lost as waste heat taken away by the exhaust gas and sunk by the other waste heat flow streams of the turbocharged Caterpillar 16CM32C engines. In an attempt to recover waste heat, APE has installed eight numbers of thermal oil boilers operated with waste heat recovery (WHR) from the exhaust gas of eight engine units and the recovered heat is used for process heating of heavy fuel oil. The exhaust heat of the remaining six APE engine units are left unattended and this study was focused on the feasibility and design procedures for Heat Recovery Steam Generation (HRSG) units for the remaining six engines to recover maximum possible heat from the exhaust gas with the purpose of additional electricity generation by a steam cycle; and selecting a suitable standard steam turbine according to heat estimated recovery capacity of the above HRSG.

Exhaust gas from six engines (Engine no 9 to 14) as shown in Figure 1.2 is expected to be used for HRSG, since other eight engines have WHR boilers. Figures 1.1 and 1.2 show the present layout and exhaust gas pipe arrangement of the plant.

1.2 Statement of the problem

With the climate patterns becoming more unpredictable due to global warming Sri Lanka has to rely on thermal electricity generation for the reliability of the national electricity grid especially during drought or less rainy seasons. The country is 100% dependent on expensive imported coal and oil for thermal power generation as alternative to the locally available hydro power.

Therefore, it is of vital importance that the thermal power plants of Sri Lanka make the most out of the imported energy sources of fuel. At the same time, protecting the environment as much as possible, by reducing the GHG emissions is very much a timely necessity.

Since eight engine units out of the 14 engine units at ACE Power Embilipitiya (Pvt) Ltd are already equipped with 750kW thermal oil boilers and only the other six engines have no existing WHR method, it has obviously been decided by the plant management that a new proper WHR system should be assessed only for the remaining six engines.

The main goal is to enhance the overall efficiency of fuel utilization through optimized waste heat recovery for electricity generation by a steam bottoming cycle, thus also minimizing the release of rejected heat to the environment.

During normal full-load running conditions one Caterpillar 16CM32C engine emits heated exhaust gas at 300^{o}C~320 ^{o}C with a volume flow rate of 40904 $m^{3/}h$. It was estimated that the energy waste due to current practice is approximately 235388.9 MJ per 24 hrs of operation as thermal energy lost to the environment with the hot exhaust gas, which corresponds to a 2724.4 kW thermal load of rejected heat just by the exhaust gases per each engine unit. For the six engines that lack any heat recovery, the total potential for recoverable heat rate from exhaust gases would therefore sum up to roughly 16.35 MW.

This study completely disregarded any possibilities for waste heat utilization from other available low-temperature waste heat streams from the engines (such as jacket water cooling and charge air cooling) due to the clearly expressed unwillingness of the ACE plant management to consider any feasible investment options other than high-temperature heat recovery from exhaust gas only.

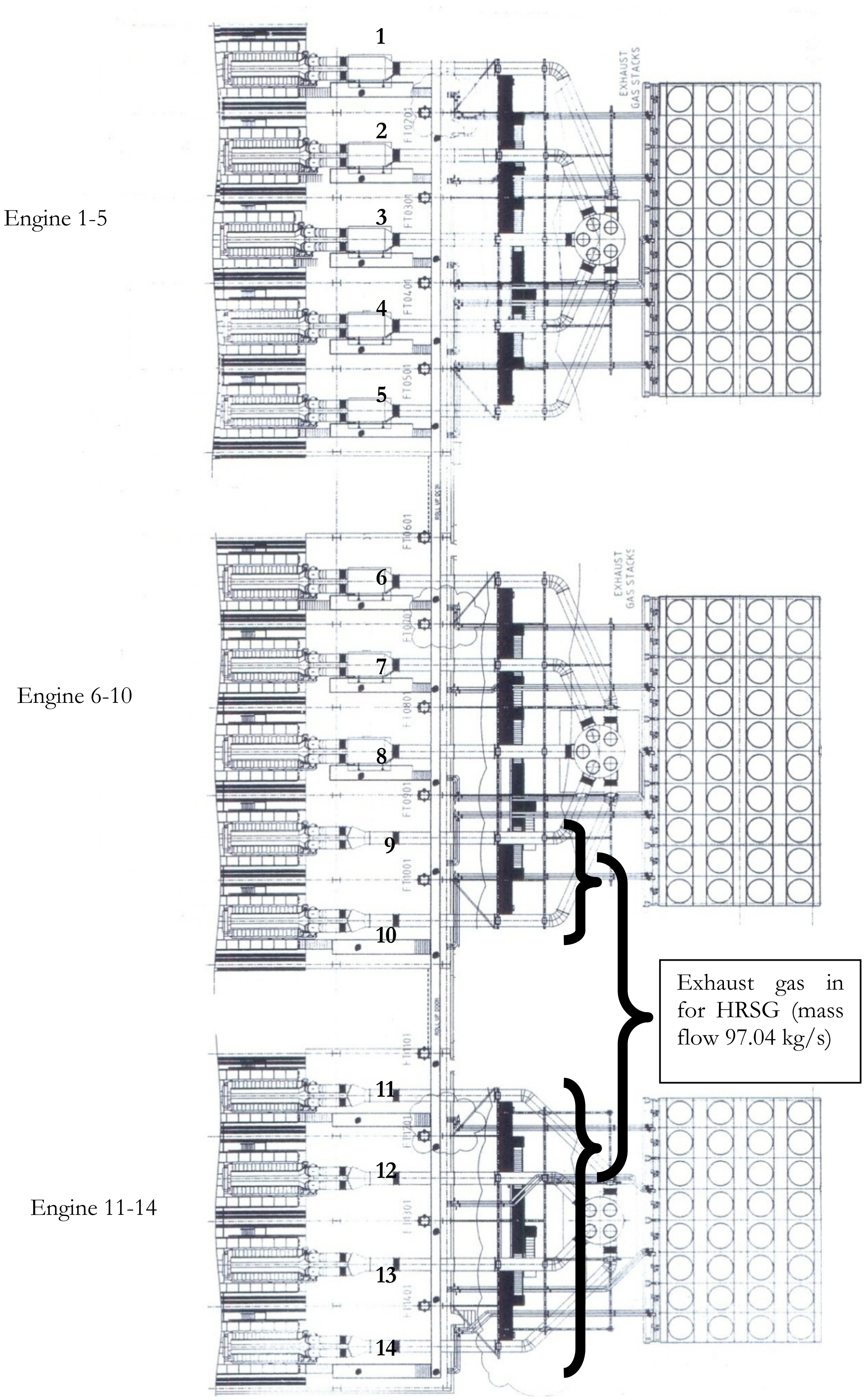


Figure 1.1: Plant layout

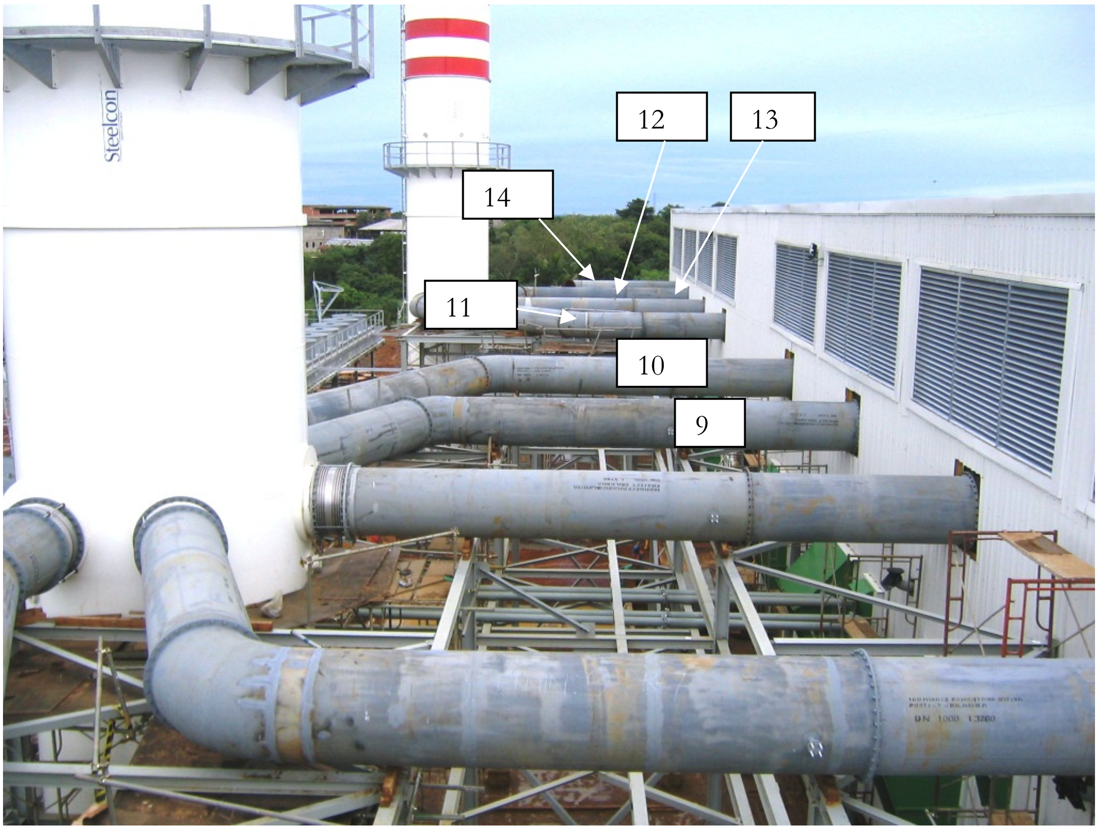


Figure 1.2: Exhaust gas out pipes from power house

1.3 Aims and Objectives

1.3.1 Aim

The aim of this work is to investigate the possibility of increasing the energy efficiency of the Ace Power Embilipitiya (Pvt) Ltd thermal power plant by suitably utilizing the waste heat of the exhaust gasses produced by 6 of the engine units, which is at present directly released to the environment.

1.3.2 Objective

The objective of the study is to model a HRSG and select a matching commercially available steam turbine for improving the performance of the thermal power plant by optimizing the waste heat recovery from the exhaust gas streams of the available HFO-fired medium speed four-stroke diesel engines. Additional power would be generated by a technically and economically optimum solution in the form of a steam bottoming cycle.

2 Literature Review

2.1 Background

The electricity demand of Sri Lanka is met by hydro, thermal, wind and solar power plants owned and operated by both Ceylon Electricity Board and the private sector various independent power producer (IPP) companies alike.

The total installed capacity of electricity generation of Sri Lanka exceeds 3000 MW and the majority of the installed capacity belongs to CEB. The distribution of power generation facilities in Sri Lanka between CEB and IPPs is shown in Figure 2.1. It is noteworthy that the share of installed capacity of CEB and the share of hydro power generation facilities have been increasing annually during the recent years.

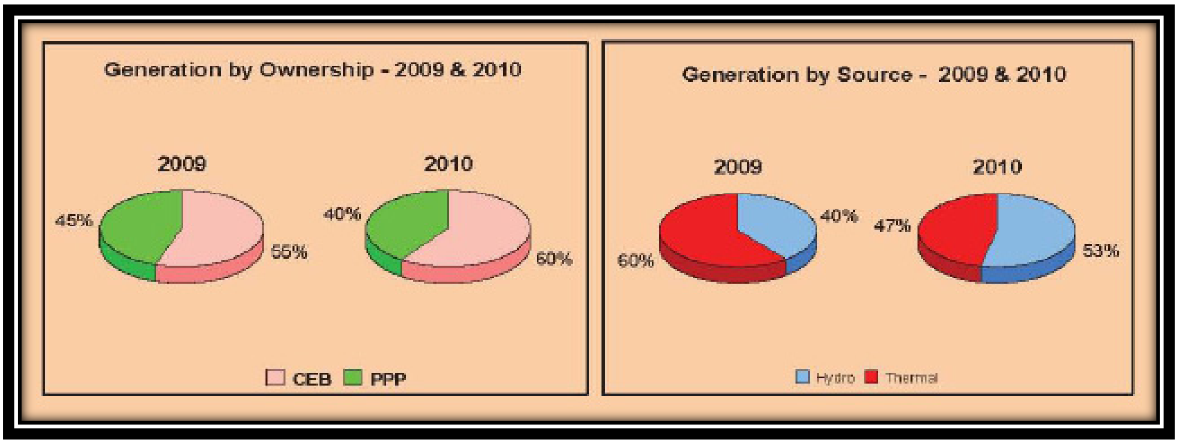


Figure 2.1: Ownership of plants and power generation source distribution in Sri Lanka.

At present over 10 active IPPs are operating in Sri Lanka. Some of IPPs with the highest individual installed capacities are presented in Table 2.1 below.

Table 2.1 Independent Power Producers (IPPs): Plant Details

Owned by	Location	Capacity (MW)	Fuel Type
AES (Kelanitissa) (Pvt) Ltd.	Kelanitissa	163	Auto Diesel
HelaDanavi (Pvt) Ltd.	Putlam	100	Furnace Oil 180cst
ACE Power (Embilipitiya) Ltd.	**Embilipitiya**	**100**	**Furnace Oil 180cst**
Colombo Power (Pvt) Ltd.	Colombo Port	60	Furnace Oil 180cst
Asia Power (Pvt) Ltd.	Sapugaskanda	51	Furnace Oil 380cst
LakDanavi (Pvt) Ltd.	Sapugaskanda	22	Furnace Oil 180cst
ACE Power (Matara) Ltd.	Matara	20	Furnace Oil 180cst
ACE Power (Horana) Ltd.	Horana	20	Furnace Oil 180cst
Agrico (Pvt) Ltd.	Jaffna	15	Auto Diesel

ACE Power is one of the leading IPPs of Sri Lanka with 3 different plants, including the Embilipitiya site (APE) with an installed capacity of 100 MW employing 14 units of Caterpillar 16CM32 engines that run on heavy fuel oil.

2.2 Caterpillar 16CM32 engine

APE uses the Caterpillar Diesel Engine 16CM32 as the prime mover for driving alternators to generate electricity. The 16CM32 is a medium speed, 4-stroke, 16-cylinder, V-type with 32 cm cylinder diameter, turbocharged intercooled marine engine with direct fuel injection. The major characteristics of the engine could be tabulated as given in Table 2.2.

Table 2.2 Mechanical specification according to the supplier

Caterpillar Diesel Engine 16CM32	
Design	16 cylinder V-type arrangement 750 rpm, Direct injection Compressed air starting Rotation counter clockwise Exhaust gas turbocharging and charge air cooling Operating on heavy fuel oil, 180 cSt at 50 ^{o}C
Rated power output	7520 kW
Specific fuel consumption	177 g/kWh at 100% mechanical engine output
Specific fuel consumption at the generator terminals	194.2 g/kWh at 100% electrical output

2.3 Heat Balance of the engine unit

A medium speed diesel engine unit built for power generation is meant to convert the chemical energy of the fuel into useful mechanical energy to drive an alternator. The typical efficiency of this energy conversion measured from fuel chemical energy to electrical output at the alternator's terminals, is usually above 40% depending on the type and size of the prime mover. It is noteworthy that among the different energy flows out of the engine including the waste heat streams, the engine exhaust gas alone carries up to 30% of the total primary fuel energy [12].

In the case of typical medium speed internal combustion engines about 28% [1] of the total energy is rejected with the hot exhaust gas. As illustrated in Figure 2.2, heat energy is lost to the environment without useful work in the forms of:

1. Exhaust gas
2. Jacket cooling water or jacket cooler
3. Lubrication oil cooler
4. Charge air cooler or intercooler
5. Radiation loss

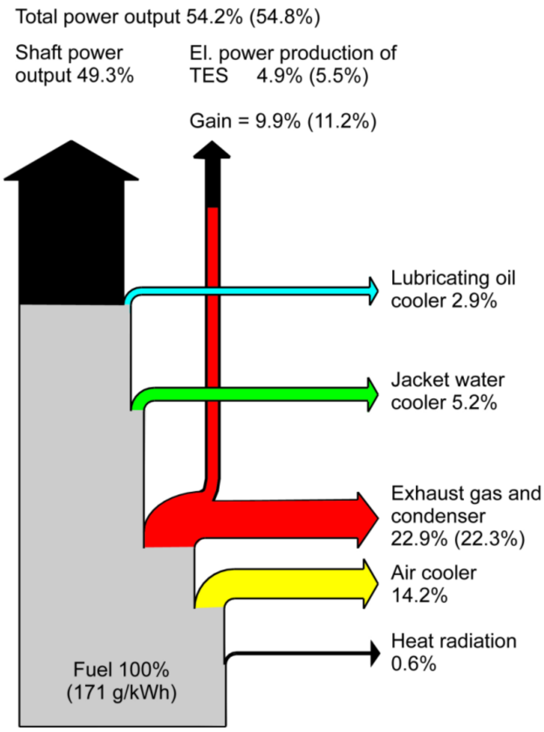


Figure 2.2 Typical heat balance of a large marine diesel engine with exhaust gas heat recovery for additional power generation [1]

Heat is generated by the processes of fuel combustion or chemical oxidation reaction. Majority of thermal power plant industry involving medium speed diesel engine-generator units have not concentrated enough on WHR but are designed to dump waste heat to the environment without recovering for any economic purpose. Feasibility of recovery of waste heat is governed by the facts of temperature levels of various waste heat streams and the investments and economies involved.

During this study, in the context of APE, it was found that a large quantity of hot flue gas is generated from the diesel engines that are utilized at APE. Further it was clear that even a small attempt to recover some of the waste heat if not all the heat would still bring significant cost benefits by saving on primary fuel and by reducing GHG emissions.

The existing waste heat recovery thermal boilers at APE recover a maximum of 60 MWh of waste heat energy annually. The amount of waste heat energy recovered depends on the factors of engine load and process heat demand of APE.

2.4 Waste Heat Recovery

The rising costs of energy and the global warming in recent years have highlighted the need to develop advanced energy systems to increase the efficiency and reduce the emissions. One of the methods that are extensively used is the waste heat recovery. Waste heat recovery is the process of yielding energy from potential high energy sources that are lost to the environment without any useful work usually following an industrial energy conversion process.

There is a great scope to recover waste heat from various industries and to generate power or process heat using a heat recovery steam generator (HRSG). The technological know-how of waste heat recovery is indeed a subject that has been subjected to extensive research and development. Some of the most widely used waste heat recovery technologies are listed below.

2.4.1 Recuperators

A recuperator is a counter flow type of heat exchanger that can be installed across a gas stream with potential recoverable heat energy like exhaust gas. It is manufactured in vertical, horizontal and cellular configurations and is able to recover waste heat with effectiveness up to 99%.

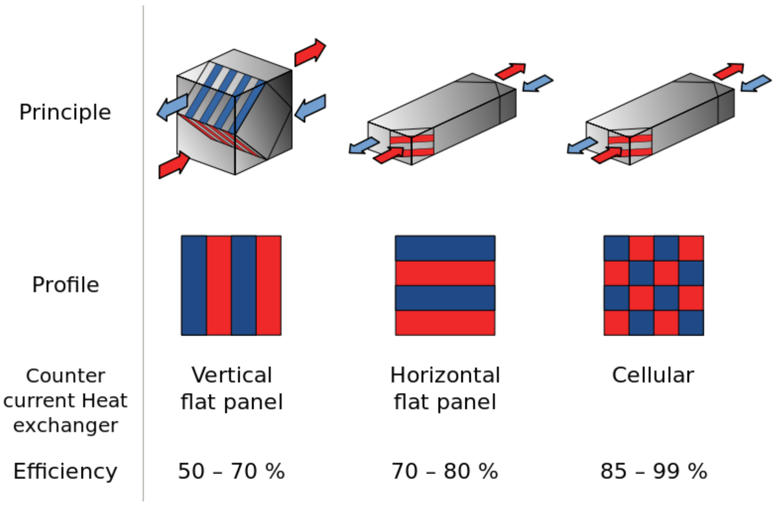


Figure 2.3 Major configurations of Recuperators [2]

2.4.2 Heat wheel

Heat Wheel is a rotary type of heat exchanger which can be used for recovering heat energy from a warm gas flow in a regenerating (cyclic) mode of operation. It is most efficient as a counter flow heat exchanger.

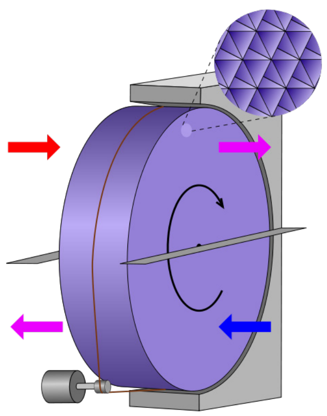


Figure 2.4 Operation of a heat wheel rotary heat exchanger [3]

2.4.3 Heat pipe

A heat pipe is used for recovering waste heat from a hot reservoir to a cold reservoir. It uses the principals of thermal conductivity and phase transition for transferring energy at a sustained temperature level.

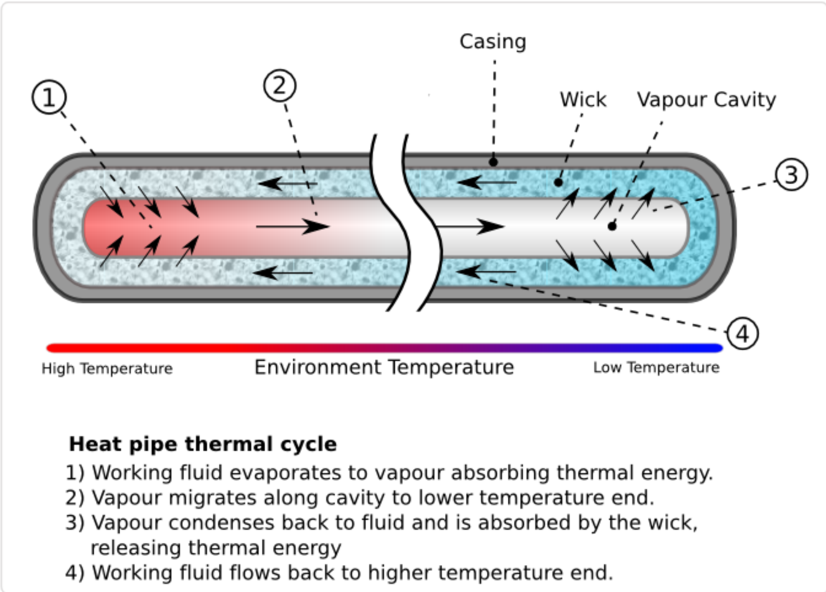


Figure 2.5 WHR process of a heat pipe [4]

2.4.4 Economizers

An economizer is a heat exchanger used for recovering energy within a system, most often in applications of heating up of water by the heat contained in exhaust gases.

2.4.5 Shell and tube heat exchangers

Shell and tube heat exchangers are the most common type of waste heat exchangers used in the industry. It is a proven and efficient heat recovery arrangement where the colder fluid is usually flowing inside the tubes.

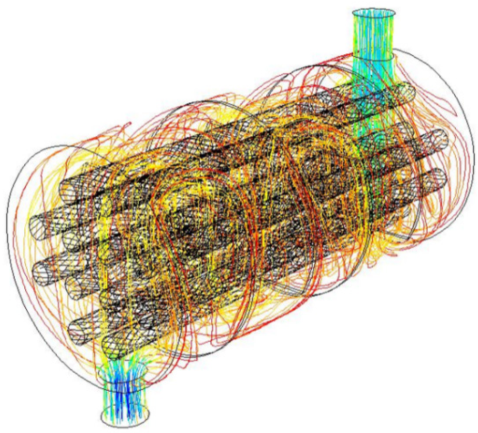


Figure 2.6 WHR simulation in a shell and tube type of heat exchanger [5]

2.4.6 Plate Heat Exchangers

Plate Heat Exchangers use metal plates as heat exchanging surfaces. They differ from conventional heat exchangers as they utilize more surface area for heat exchange process than any other WHR system.

2.4.7 Run around coil exchangers

A run-around coil exchanger connects the hot and cold reservoirs of a process system by pipework, where the hot fluid usually runs inside the tube coil while the cold fluid occupies the space around the tube coil.

2.4.8 Waste heat recovery boilers

Waste heat recovery boilers are used to capture heat energy directly from a waste heat source like exhaust gas. The hot waste fluid passes around consecutive tube bundles such as economizer, evaporator and superheater, while inside the tubes water is flowing and steam is being produced. Alternatively, thermal oil may be heated up on the secondary side of the heat exchanger without boiling.

Waste heat recovery arrangements may also be involved in utilizing low-temperature heat with the help of advanced systems such as heat pumps and thermocompressors.

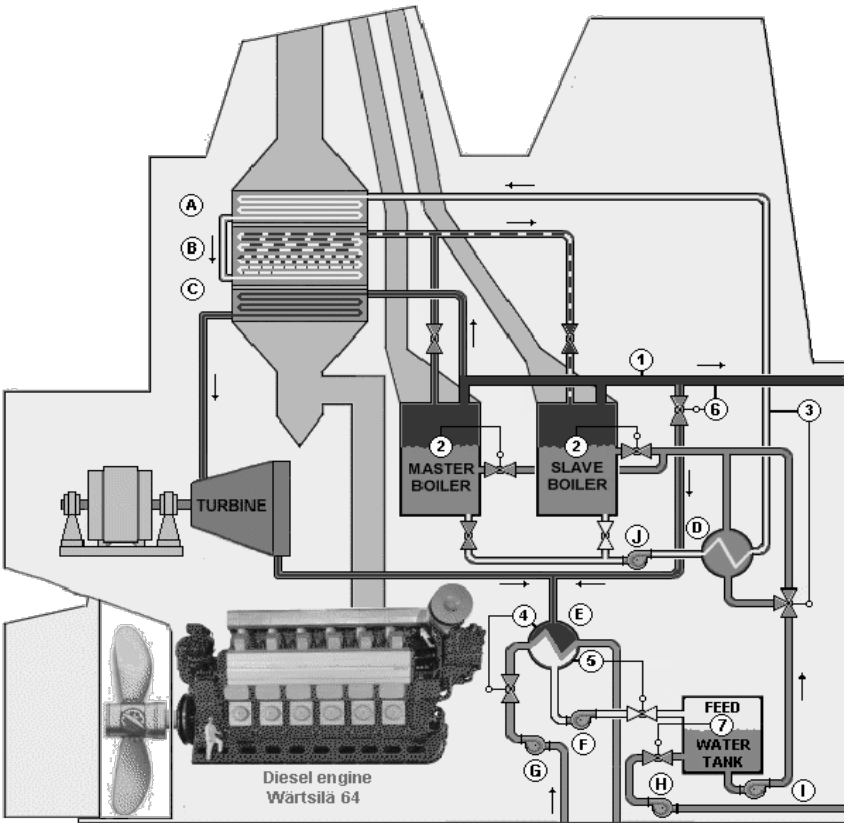
2.5 Application of Exhaust Gas Boilers for heat recovery

Caterpillar 16CM32 engine units at APE fall into the category of medium speed marine engines with a shaft rotational speed of 750 rpm. A typical medium speed marine engine loses approximately as much as 30% of it's heat energy with exhaust gas. The engine units at APE are turbocharged and the exhaust gas turbochargers are capable of recovering some of this heat loss but most of the waste heat energy is lost with the exhaust gas.

Exhaust Gas Boilers (EGB) meant for medium speed engine heat recovery systems are designed to capture heat energy from engine exhaust gas at medium temperature levels and limited pressure loss. Therefore EGBs are designed with finned or pinned boiler tubes at the gas side in order to effectively capture and transfer heat from relatively high quantities of flue gas.

The effectiveness of an EGB largely depends on the condition of the heat transferring surface. Therefore periodic soot blowing of the boiler is essential to keep the heat transferring surface free from soot deposits which may decrease steam production considerably and could lead to a soot fire if unattended for a considerable period.

The application of a typical heat recovery system used onboard a ship employing a medium speed marine engine is given in Figure 2.7. The waste heat from the engine exhaust gas is converted to steam energy and a steam turbine is driven from it in turn driving an alternator to produce electricity. The generated steam is also used for onboard process heating and whenever the steam production exceeds the demand the excess steam is dumped to a dumping condenser.



A- Economizer section of the Waste Heat Recovery Boiler.
B- Evaporator section of the Waste Heat Recovery Boiler.
C- Superheated section of the Waste Heat Recovery Boiler.
D- Heat exchanger.
E- Condenser
F- Condensate pump
G- Cooling water pump
H- Make-up water pump
I- Boiler feed water pump
J- Boiler water circulation pump
K- Alternator

1. Steam pressure control
2. Water level control
3. Economizer inlet temperature control.
4. Condenser pressure control
5. Condenser level control
6. Steam dump control valve
7. Feed water tank level control

Figure 2.7 A marine WHR system for medium speed diesel engines [6]

2.6 Steam Turbine

A Steam turbine is a prime mover capable of converting steam energy to mechanical energy by means of steam expansion through an orifice and converting the enthalpy of steam into mechanical energy by the impulse or reaction of steam against the turbine blades.

Steam turbines can be classified according to the following criteria depending on their operation and design.

By the action of steam:

Impulse, Reaction, Impulse and reaction combined.

The number of stages and pressure reduction involved:

Single stage, Multi-stage, Whether there is one or more revolving vanes separated by stationary reversing vanes.

The direction of steam flow:

Axial, Radial, Mixed, Tangential, Helical, Re-entry

The inlet steam pressure:

High pressure, Medium pressure, Low pressure

The final exit pressure or condenser temperature level:

Cold Condensing, Warm Condensing / Backpressure, Non-condensing

The source of steam:

Extraction, Accumulator

Impulse turbines rely on the kinetic energy of a fluid flow by changing the direction of a high velocity fluid flow resulting in an impulse to spin the turbine. The pressure head of the fluid is converted to velocity head before reaching the rotor blades by accelerating the fluid in a nozzle (stator blades).

A reaction turbine utilizes also a certain pressure drop of the working fluid through the rotor blades, so that some of the acceleration of fluid happens also in the rotor thus generating reaction forces. Reaction turbines often achieve higher efficiencies and are more adapted to certain applications but need a higher number of stages for the same overall pressure drop.

Modern turbine designs often incorporate both impulse and reaction stages to a varying degree whenever applicable.

The major characteristics between the impulse and reaction type turbines can be summarized as shown in Figure 2.8 and Table 2.3 below.

A typical application of a steam turbine to drive an electrical generator is shown in Figure 2.9. The high pressure steam entering the turbine undergoes expansion in usually three separately designed sections of the turbine i.e., high pressure, medium pressure and a low pressure turbine section respectively.

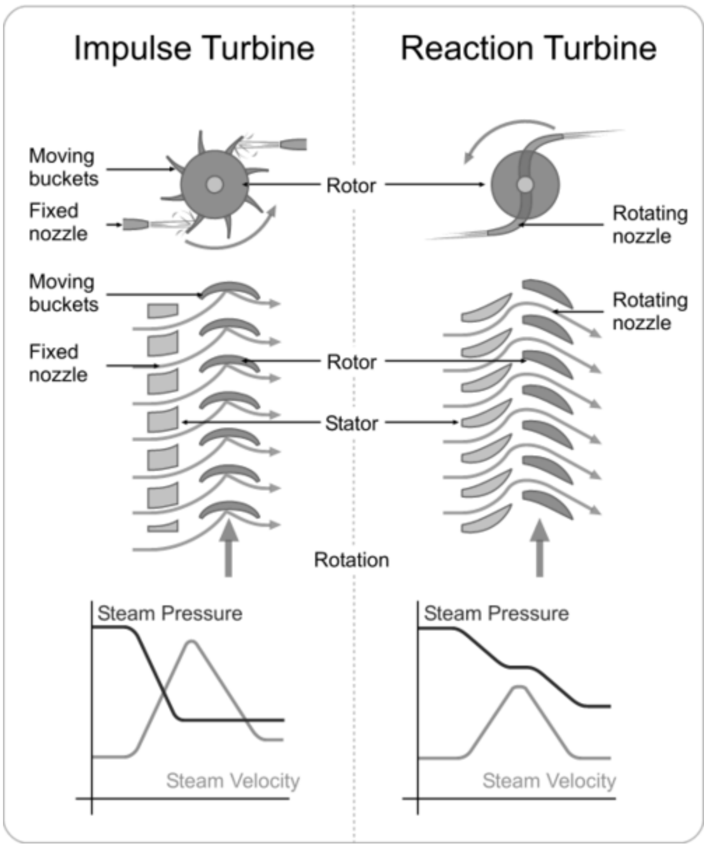


Figure 2.8 Design variations of impulse and reaction turbines [7]

Table 2.3 Characteristics of impulse and reaction turbines

Impulse Turbine	**Reaction Turbine**
Nozzle and moving blades are in series	Fixed blades and moving blades. No nozzles
Pressure falls in nozzle	Pressure falls also in rotor blades
velocity (or kinetic energy) of steam increases only in nozzle (stator blades)	velocity (or kinetic energy) of steam increases both in stator and rotor blades
Needs compounding to increase the efficiency	No compounding needed to increase efficiency
High pressure drop per stage	Low pressure drop per stage
Less number of stages	Higher number of stages
Efficiency is lower	Efficiency is higher
Requires less space	Requires more space
Blade manufacturing is easier	Blade manufacturing is more difficult

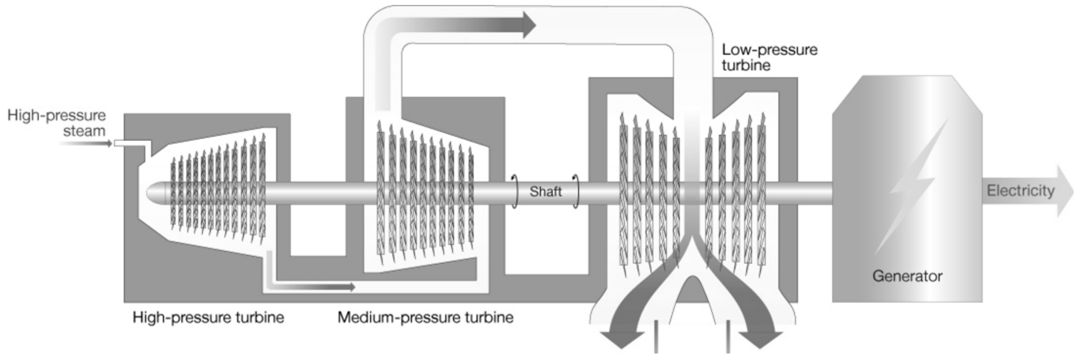


Figure 2.9 Industrial application of a steam turbine for WHR [8]

2.7 The principle of the waste heat recovery cycle [9]

A waste heat recovery steam generator recovers heat from a hot gas source like exhaust gas and produces pressurized steam. They can be operated in either the cogeneration mode or the combined-cycle mode. In the cogeneration mode, steam produced from the HRSG is mainly used for process applications, whereas in the combined-cycle mode, power is generated via a steam turbine driven electrical generator.

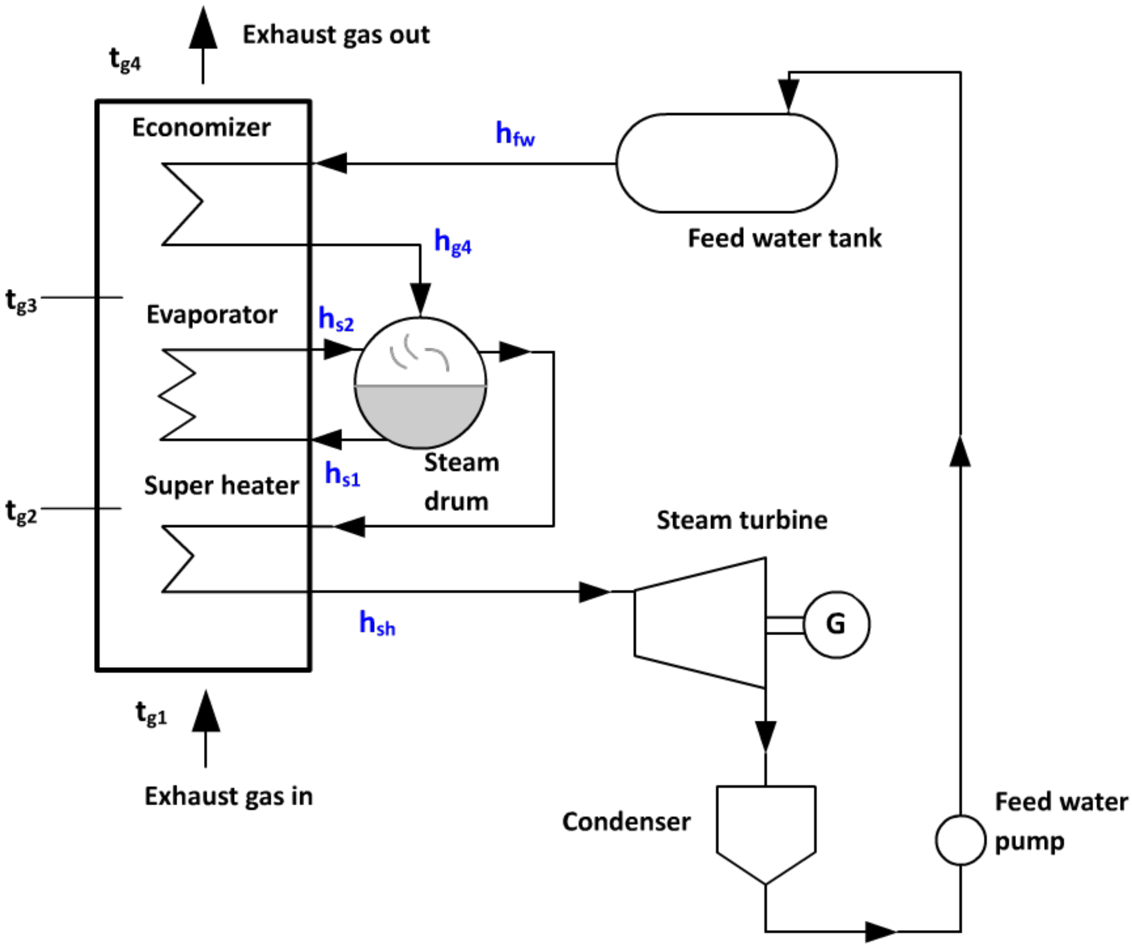


Figure 2.10 Schematic diagram of a basic heat recovery cycle

2.7.1 Condenser

Condensers are very commonly used in steam power plants. A surface condenser is designed as a sealed container filled with a large number of tubes. This type of condenser is a shell and tube heat exchanger, mainly. Vapour is separated from the cooling water and for this reason, the idea of the surface condenser is to keep by without the need of water treatment and reuse feedwater with high purity. The amount of cooling water required to condense the incoming steam would generally be of amount 50 to 80 times the steam flow, typically with a very tight temperature difference. Within the range of absolute pressure 0.1-0.4 bar, condenser offers low back pressure at the outlet from the turbine in general. Vacuum is created in a cold condenser, giving the plant a good economy; thus maximizing the thermal efficiency of the plant. The proper operation of the condenser is critical for the availability and efficiency off the steam plant.

2.7.2 Heat Recovery Steam Generator

Heat recovery steam generators are useful for recovering heat from hot flue gases. In general, a heat recovery steam generator (HRSG) is situated next to the hot flue gas exhaust duct of a power plant. Without extracting the heat from the furnace exhaust flue gas of high temperature from the outside is burned fuel, HRSG is a boiler mainly. In general, whether coming from a diesel engine or a gas turbine, the hot flue gases are led straight into the HRSG, which should not exert too high of a backpressure to the gas path. Economizer, evaporator and superheater are the three main components of the HRSG. The preheating of feedwater, the boiling process and the superheat of dry steam, respectively, follow the hot gas path in a counter-current flow, where the pinch-point temperature difference plays the most important role for the distribution of the heat exchange surfaces, as shown in Figures 2.10 and 2.11.

2.7.3 Economizer

The economizer heats up the feed water close to saturated water temperature by gaining heat from the low-temperature flue gas. Economizer located in last part of the HRSG where the exhaust gas has the lowest temperature. An economizer is a forced-flow, once-through, convection heat-transfer device to which feed water is supplied at a pressure slightly above that in the steam-generating section and at a rate corresponding to the steam output of the unit. The feed water flow is heated up somewhat below the saturated water temperature because evaporation in the economizer tubes is undesired. The heated water then enters the boiler steam drum. The difference between the water exit temperature of the economizer (which is the same as the evaporator entry temperature) and the saturated water temperature is usually termed approach temperature difference.

The approach temperature difference makes sure that the feedwater does not boil already in the economizer tubes, and usually is in the range of 10 – 25 °C below the saturated water temperature at the given pressure.

2.7.4 Evaporator

The evaporator further heats up the feedwater coming from the economizer up to the proper saturated water conditions and evaporates the entire mass flow to saturated steam, providing energy equal to the latent heat of the water. The evaporator is always the middle part of the HRSG and the gas temperature is higher than that of the economizer. Water is vaporized into steam, steam generation occurs in the evaporator. Required heat transfer is the amount needed to change the phase of water from liquid to vapour (steam). The pinch point temperature difference is relevant to the evaporator part of the HRSG and shows the minimum temperature difference between the hot gas flow and the saturated water temperature. Larger pinch point temperature difference would lead to smaller necessary heat exchange surface for the evaporator, while a smaller pinch point temperature difference would allow for a deeper recovery of heat from the exhaust gas stream but would require a larger heat exchange surface with consequently higher costs and higher pressure drop, see Fig. 2.11. The pressure drop on the flue gas side should be kept at a minimum and is usually at 25 to 40 mbar.

2.7.5 Superheater

The superheater is the first heat exchange surface on the gas path of the HRSG where the gas is hottest, and consequently the last heat exchange surface on the water/steam side, where the saturated steam us brought up to superheated conditions to a temperature governed by the available hot gas temperature and the selected limit for the temperature difference being a direct function of the heat exchange surface and thus investment costs. Superheating may be parallel or counter-flow type and economic considerations must be taken into account depending on if used in low or high heat. Metal alloy material cost is higher than for the lower temperature heat exchange sections. Sometimes the HRSG may use supplementary firing in the gas duct, which increases the steam production but subjects the superheater to even higher temperatures.

The heat transfer between the exhaust gas and the water/steam side of the HRSG, including the pinch point location, is illustrated in Figure 2.11.

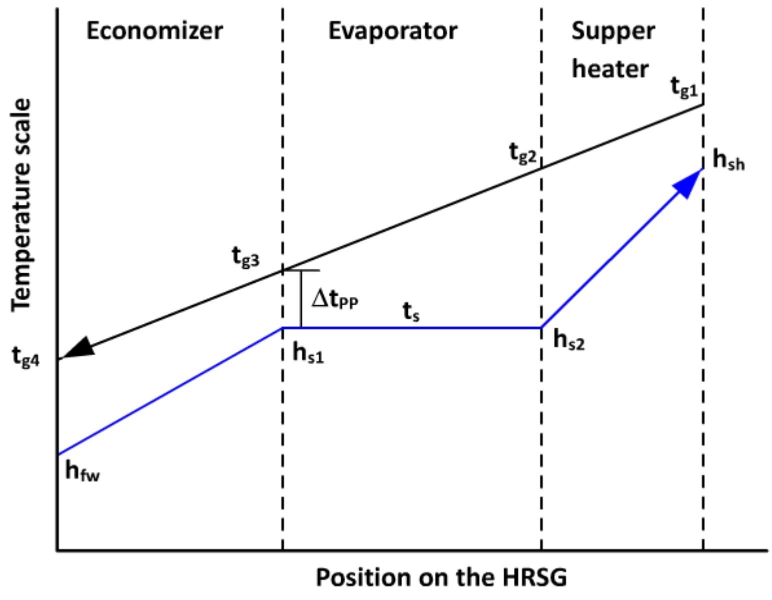


Figure 2.11 Heat transferred from exhaust gas to water (steam) in the HRSG in economizer, evaporator and superheater sections.

2.8 Types of HRSG

HRSGs can be classified according to their design, operation and also application.

- Fired and Unfired

The steam generated in a HRSG may also be used for process heating apart from it's main function of driving a steam turbine. The load condition of the primary waste heat source, i.e. an internal combustion engine or a gas turbine, could affect the process heating at times of low load running by not meeting the desired steam demand. Therefore some HRSGs are fitted with separate fuel burners at the inlet duct of the HRSG, usually fired with the same fuel as the mail engine, to be used at reduced waste heat supply conditions and are classified as supplementary-fired HRSGs.

- Single and Multiple Pressure Operation

Smaller HRSG units are typically designed only with a single pressure level of steam generation in order to decrease complexity and ensure lowest possible costs. Larger units and especially combined cycles aiming at high efficiency would enhance the heat recovery in the HRSG by using multiple (usually 2 or 3) pressure levels of steam generation.

- Horizontal and vertical

Based on the position of the waste heat gas flow, the HRSG can be designed horizontally or vertically. Although both the designs carry similar manufacturing costs the different geometry has the following advantages over one-another, as listed in Table 2.4.

Table 2.4 Comparison of Horizontal and Vertical HRSGs

Horizontal	Vertical
Require 30% larger footprint area	Requires less ground space
Higher number of expansion joints	Lesser number of expansion joints
Less structural requirements	Higher and complex structural requirements
Difficult to maintain and inspect	Easier to maintain and inspect

3 Methodology

The waste heat recovery potential from exhaust gas of Caterpillar 16CM32C engine was investigated as the first step of the study. At the initial stage the quantity of recoverable waste heat was calculated by comparison between an installed waste heat thermal oil boiler at APE and an engine without a waste heat recovery method at fully loaded condition.

A calculation was performed taking into account the calorific value of fuel used and the operating conditions in order to find the maximum theoretical waste heat recovery potential of the six available engines of APE.

The approach for WHR from Caterpillar 16CM32C engines at the APE site was selected as a HRSG citing the possibility of selecting a suitable steam turbine from the market and producing additional electrical power thus converting the plant into a combined cycle.

A preliminary design modelling of the HRSG was done considering the approach point and the pinch point variables, which affect the steam temperature profile and steam production rate. The pinch point temperature difference was selected as $8^{o}C$ and special care was taken to avoid temperature cross situations.

The optimization of the modelled HRSG was performed using the Engineering Equation Solver software package (EES). Initially, trial runs were carried out in EES to arrive at the maximum possible power output without compromising on size and cost by varying the pinch point and other parameters.

A second set of EES trial runs were carried out to find out the optimum steam pressure for the HRSG keeping the pinch point temperature constant at $8^{0}C$. The steam pressure was varied from 5 bar to 17 bar for the trials and the limiting factor for setting the maximum possible power output was taken as the sulfur dew point temperature of the outgoing exhaust gas after heat recovery.

A commercially available steam turbine was selected and evaluated with maximum possible power output according to availability to suit the context of APE.

4 WHR potential of Caterpillar 16CM32 engine

The Caterpillar 16CM32 engine number 14 and engine number 2 at the APE location were compared for the potential of waste heat recovery. The latter is equipped with a WHR boiler for fuel preheating, which was used for the calculation of WHR potential of the system; and the other engine operates with no installed method for WHR.

The engine number 14, with no WHR boiler installed showed a steady flow of exhaust gas at an average temperature of 320 ^{0}C with the engine fully loaded.

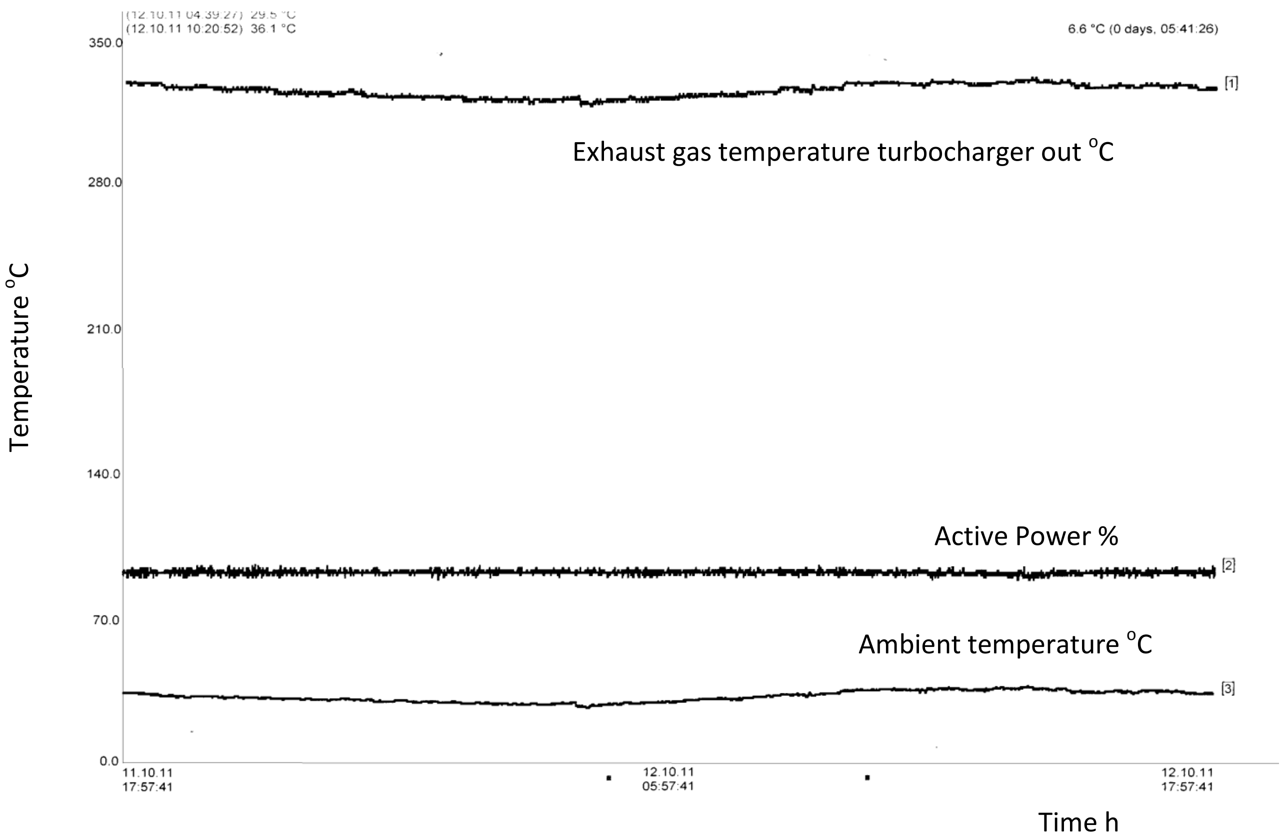


Figure 3.1 Graph of engine exhaust temperature after the turbocharger, active power and ambient air temperature.

The exhaust gas system of engine number 2 which is equipped with a thermal WHR boiler showed a sharp difference between temperatures before and after the WHR boiler.

Therefore, it is obvious that the Caterpillar 16CM32 engine can be practically used for effective recovery of waste heat from exhaust gas by any type of a waste heat boiler unit.

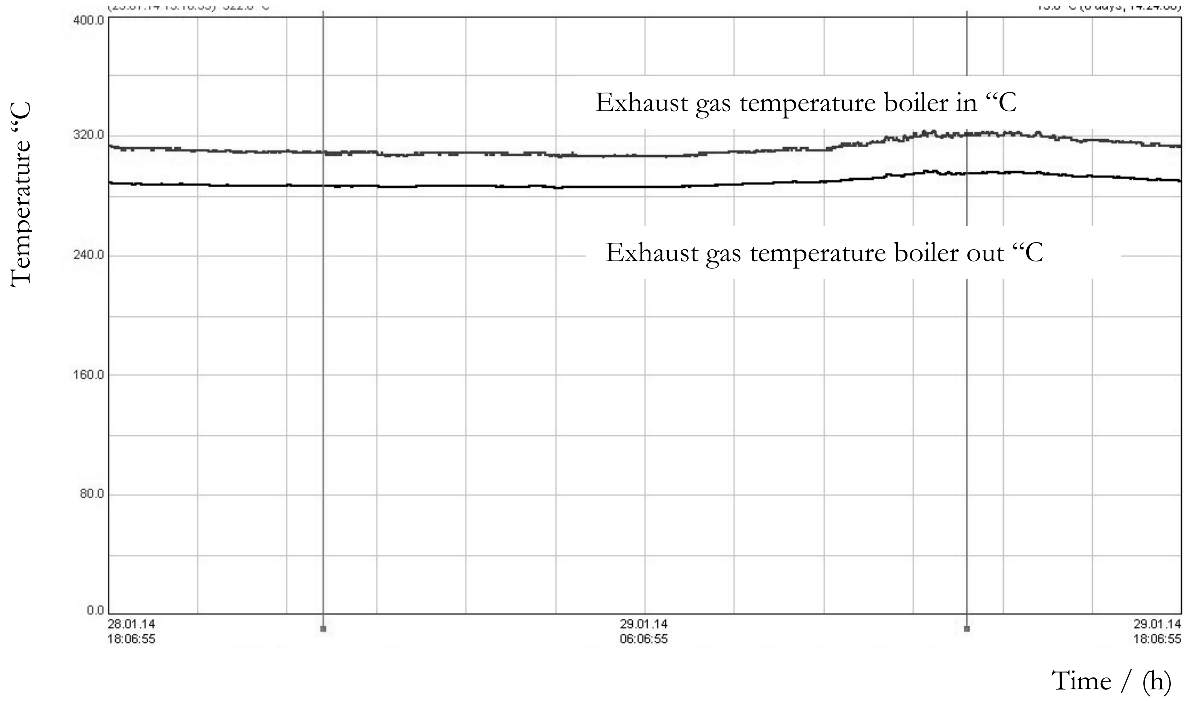


Figure 3.2 Graph of Exhaust temperature of engine number 02 before and after the existing WHR boiler

4.1 Equations for Estimating Waste Heat

In order to calculate the WHR potential of the engine units at APE, the following method was used [10].

The Lower Heating Value of the fuel used was calculated using the following equation.

$$LHV=HHV-91.23\times\%H_2 \qquad (1)$$

Where

LHV = Lower Heating Value

HHV = Higher Heating Value given in equation (2)

$$HHV=17884+57.5\times {}^{o}API-102.2\times\%S \qquad (2)$$

^{o}API = Density of the fuel as expressed in terms American Petroleum Institute degrees. API is given by the equation (3)

$$s=\frac{141.5}{131.5+{}^{0}API} \qquad (3)$$

Where s = specific gravity of the fuel

%S = Sulfur % in the fuel

$\%H_2$ = Percentage of H_2 in the fuel which is given by the equation (4)

$$\%H_2 = F - \frac{2122.5}{^0API + 131.5} \qquad (4)$$

Where F is a factor that depends on the value of API:

$F = 24.5$ for $0 \leq {}^0API \leq 9$

$F = 25$ for $9 \leq {}^0API \leq 20$

$F = 25.20$ for $20 \leq {}^0API \leq 30$

$F = 25.45$ for $30 \leq {}^0API \leq 40$

The heat energy generated by combustion is given by the equation (4)

$$Q_f = \dot{m}_f LHV_f \qquad (5)$$

Where

m_f = mass flow rate of fuel

Basic calculation for losses

$$Q_l = Q_f - Q_e \qquad (6)$$

Where

Q_l = Energy loss

Qe = Electrical energy output

Energy extracted from flue gas;

$$Q_g = \dot{m}_g c_p \left(t_{g1} - t_{g4}\right) \qquad (7)$$

Where

m_g = mass flow rate of flue

t_{g1} = Exhaust gas temperature at the inlet the HRSG

t_{g4} = Exhaust gas temperature at the exit of the HRSG

Energy extracted percentage;

$$Q_{recovery} = \frac{Q_g}{Q_f} \qquad (8)$$

4.2 Energy calculation (for one engine)

WHR energy calculation for a single engine unit at APE could be performed as follows.

This calculation is based on one engine unit without any existing exhaust gas heat recovery. The required data is taken from the DNV fuel testing report (Appendix 1).

%S = 2.26

s = 0.9659

From equation (3)

$$^{0}API=\frac{141.5}{0.9659}-131.5$$

$$^{0}API=14.995$$

Therefore, from equation (4);

The value of F was taken as 25, corresponding to the API range

$$9\leq {}^{0}API\leq 20$$

$$\%H_2=25-\frac{2122.5}{14.995+131.5}$$

$$\%H_2=10.51\%$$

From equation (2)

$$HHV = 17884+57.5\times 14.995-102.2\times 2.26$$

$$HHV = 18515.2405$$

From equation (1)

$$LHV = 18515.2405-91.23\times 10.51$$

$$LHV = 17556.4132 \text{ Btu/lb}$$

$$LHV = 40.83 \text{ MJ/kg}$$

Applying equation (5) to determine the heat energy generated by fuel combustion:

$$\dot{m}_f = 1464.29 \text{ kg/h}$$

$$Q_f = \dot{m}_f LHV_f = (1464.29)(40.83)$$

$$Q_f = 59787.36 \text{ MJ/h}$$

Electrical energy

$$Q_e = 6.916 \text{ MWh} = 24897.6 \text{ MJ}$$

Heat energy not converted to electrical energy;

$$Q_l = Q_f - Q_e = 59787.36-24897.6 = 34889.76 \text{ MJ/h}$$

In order to estimate the heat energy recovered the equation (7) was used:

$$Q_g = \dot{m}_g c_p \left(t_{g1} - t_{g4}\right)$$

t_{g1}= flue gas temperature into HRSG 308^{0}C = 586.4 ^{0}F (see Appendix 2)

t_{g4}= flue gas temperature at exit of HRSG

This temperature is the lowest temperature in the HRSG, and was taken at a minimum above the sulfur dew point temperature corresponding to the sulfur content in the fuel (Appendix 2 and 4).

$$t_{g4} = 275 \text{ F} = 135 \text{ }^{o}\text{C}$$

c_p= 1.05 kJ/kgK

ρ_g= 1.32 kg/m^3 (see Annexure 2)

$\dot{m}_g$= 53993.28 kg/h (at 0^{0}C, 1013 Pa)

$$Q_g = (53993.28)(1.05)(308-135)$$

$$Q_g = 9807.87 \text{ MJ/h}$$

From equation (8)

$$Q_{recovery} = 16.4\%$$

Therefore, the Caterpillar 16CM32 engine unit at APE has a waste heat recovery potential of 16.4%, as a share of the fuel energy input into the engine that is later rejected through the exhaust gas flow.

5 Modelling of HRSG

5.1 HRSG temperature profiles and steam generation.

Starting point for the engineering of the HRSG is the evaluation of the steam temperature profile and gas and steam generation function. For firing steam generator convention alone can be from assuming an outlet gas temperature and steam flow rate desired to calculate the required amount of supplementary fuel to meet the demand for steam. Due to a lower inlet gas temperature and a large gas/steam ratio, the unfired HRSG behaves differently. If assuming arbitrary gas temperature or steam generation rate, "temperature cross situations" may occur. Figures 5.1 and 5.2 show the generic layout and the temperature profile of a HRSG consisting of economizer, evaporator and superheater at a single pressure.

Two variables that affect the steam temperature profile and steam production rate are the approach point and the pinch point temperature difference, as given in Fig. 5.2. The pinch point is the difference between the temperature of saturated steam and the gas temperature leaving the evaporator. Approach point is the difference between the temperature of the water entering the evaporator and temperature of saturated steam. Selection of these two variables will affect the size of the economizer and evaporator heat exchangers and therefore the cost of the HRSG. For typical unfired HRSG hardware that can be built and shipped economically in terms of size and complexity, both the pinch and approach point temperature difference vary between 7 °C to 25 °C, usually [11].

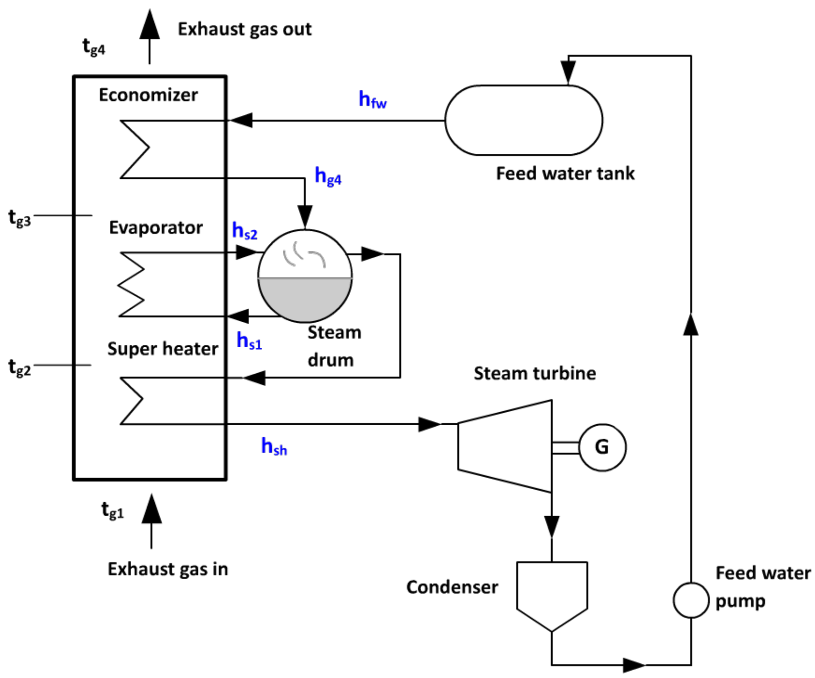


Figure 5.1: Temperature and enthalpy parameters in a HRSG with a steam cycle

For a selected approach point temperature difference as 8°C, two conditions must be met in order to avoid temperature cross situations: namely $t_{g3}>t_s$ and $t_{g4}>t_{fw}$. Optimization of power output of steam turbine considering above situation can be done by using Engineering Equation Solver.

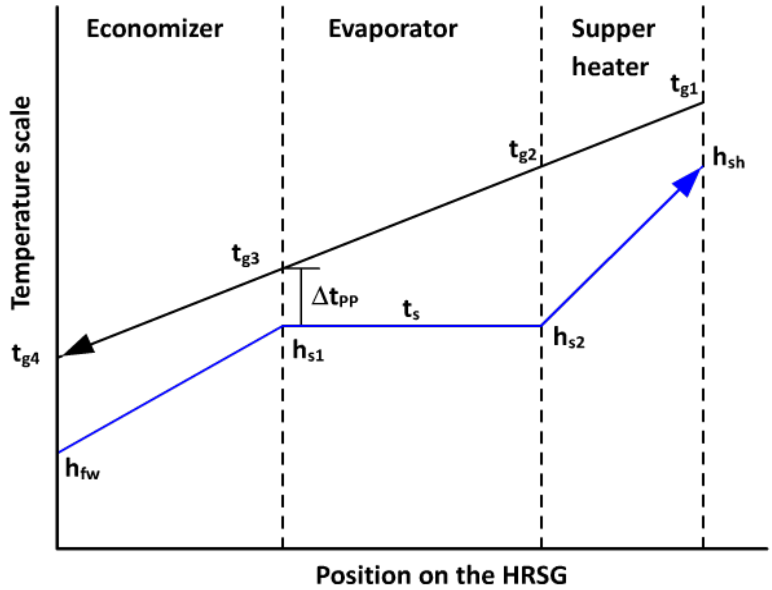


Figure 5.2 HRSG temperature profile (Steam side and Gas side)

5.1.1 Modelling parameters

Known parameters related to the exhaust gas of all available engines together:

Exhaust gas mass flow rate (m_g) = 97.04 kg/s
Exhaust gas input temperature (t_{g1}) = 308°C average

For calculations the following parameters were assumed as given below.

Steam cycle:

Inlet steam: superheated at 17 bar and 280°C
Condenser pressure (p_{out}) = 0.07 bar
Steam turbine power output (P_{st}) = 3 MW
Isentropic efficiency of the steam turbine (η_{st}) = 0.93
Mechanical efficiency of the steam turbine (η_m) = 0.98

HRSG:

Approach temperature (Δt_{app}) = 8°C

[The temperature of water entering the evaporator is 8°C less than the boiling temperature of steam at 17 bar]

Heat losses of HRSG (h_l) to surroundings = 2%

5.1.2 Optimizing parameters

In order to bring the parameters to an optimum level keeping the plant operations unhindered, the following optimization process was carried out. The possible steam turbine power output was calculated in MW and the relevant Pinch Point temperature difference was selected in ^{0}C.

Calculation according to optimized value can be expressed as shown below (cf. Fig.5.2).

The pinch point temperature difference, Δt_{pp}:

$$\Delta t_{pp} = t_{g3} - t_s \qquad (9)$$

t_s is found in saturation tables for water/steam at pressure of 17 bar, = 204.34 ^{0}C.

Applying energy balance between t_{g1} and t_{g3};

$$\dot{m}_{gas}\bar{c}_p\left(t_{g1}-t_{g3}\right)=\dot{m}_{st}\left(h_{sh}-h_{appr}\right) \qquad (10)$$

In the equation (10) t_{g3} or the steam mass flow, m_{st} are not known. In order to calculate the gas temperature t_{g3}, we first have to calculate the steam mass flow. The steam mass flow is found using heat balance for the steam turbine.

$$P_{st}=\dot{m}_{st}\left(h_{sh}-h_{out}\right)\eta_m \qquad (11)$$

P_{st}=Steam turbine power out put

h_{out}=Enthalpy of steam at turbine out

$\dot{m}_{st}$=Steam mass flow rate

h_{sh}= Enthalpy of super heated steam

η_m=turbine mechanical efficiency

The superheated steam enthalpy (17 bar, 280°C) is found in h-s diagram or in steam tables.

The enthalpy out from the turbine is found using an h-s diagram, by first finding the isentropic outlet enthalpy. Thereafter the real outlet enthalpy can be calculated from equation (12).

$$h_{out}=h_{sh}-\eta_{is}\left(h_{sh}-h_{out,is}\right) \qquad (12)$$

h_{out}=Enthalpy of steam at turbine out
h_{sh}= Enthalpy of superheated steam
η_{is}=Isentropic efficiency of turbine
$h_{out,is}$=Isentropic outlet enthalpy of steam

$$h_{out}=h_{sh}-\eta_{is}\left(h_{sh}-h_{out,is}\right)$$

$$h_{out}=2885-0.93(2985-2064)$$

$$h_{out}=2128 \text{ kJ/kg}$$

The steam mass flow is calculated from (11)

$$P_{st}=\dot{m}_{st}\left(h_{sh}-h_{out}\right)\eta_m$$

h_{sh}= enthalpy of superheated steam at 17 bar and 280°C = 2985 kJ/kg

h_{out}= enthalpy of saturated steam at 0.07bar = 2164 kJ/kg

$$3000 = \dot{m}_{st}(2985-2164)0.98$$

$$\dot{m}_{st}= 3.727 \text{ kg/s}$$

The gas temperature t_{g3} can now be calculated from (10)

$$\dot{m}_{gas}\bar{c}_p\left(t_{g1}-t_{g3}\right)=\dot{m}_{st}\left(h_{sh}-h_{appr}\right)$$

h_{sh}=enthalpy of super heated steam at 17 bar and 280 °C

$$h_{app}=c_p\left(t_s-t_{app}\right)=4.187(204.3-8)=820.7 \text{ kJ/kg}$$

$$(97.04)(1.05)\left(308-t_{g3}\right)=3.727\left(2985-820.7\right)$$

$$t_{g3}=222.60^{o}C$$

The pinch point temperature difference is:

$$\Delta t_{pp}= t_{g3} - t_s=222.6-204.3= 18.29^{o}C$$

The stack temperature can be calculated with a similar heat balance as in the previous task, this time with a span over the whole steam cycle

$$\dot{m}_{gas}\bar{c}_p\left(t_{g1}-t_{g4}\right)\left(1-h_l\right)=\dot{m}_{st}\left(h_{sh}-h_{fw}\right) \ldots\ldots(13)$$

It can be assumed that the specific heat is the same from t_{g1} to t_{g3} as from t_{g1} to t_{g4}
The feed water enthalpy can be found with the temperature after condenser, i.e. saturated water at 0.07 bar.
The same enthalpy is found for saturated water at 0.07 bar. The change of liquid water enthalpy across a pump is neglected; therefore the feedwater enthalpy at 17 bar pressure is the saturated water enthalpy after the condenser.

The stack temperature can be calculated from (13):

$$\dot{m}_{gas}\bar{c}_p(t_{g1}-t_{g4})(1-h_l)=\dot{m}_{st}(h_{sh}-h_{fw})$$

h_{sh}= enthalpy of super heated steam at 17 bar and 280 °C

h_{fw}= enthalpy of saturated water at 0.07 bar

$$90(1.05)(308-t_{g4})(1-0.02)=3.727(2985-163.4)$$

$$t_{g4}=194.4^{o}C$$

Heat balance over economizer:

$$\dot{m}_{gas}\bar{c}_p(t_{g3}-t_{g4})=\dot{m}_{st}(h_{appr}-h_{fw})\text{..(14)}$$

$\dot{m}_{gas}$= Flue gas mass flow rate from six diesel engines

$\bar{c}_p$= specific heat of flue gas

t_{g3}= Flue gas temperature after evaporator

t_{g4}= Flue gas output temperature from HRSG

$\dot{m}_{st}$. = Steam mass flow rate of HRSG

h_{appr}= Enthalpy at approach point temperature

h_{fw}= Feedwater enthalpy

Heat balance over Evaporator:

$$\dot{m}_{gas}\bar{c}_p(t_{g2}-t_{g3})=\dot{m}_{st}(h_{s2}-h_{appr})\text{...(15)}$$

$\dot{m}_{gas}$= Flue gas mass flow rate from six diesel engines

$\bar{c}_p$= specific heat of flue gas

t_{g2}= Flue gas temperature after economizer

t_{g3}= Flue gas temperature after evaporator

$\dot{m}_{st}$ = Steam mass flow rate of HRSG

h_{s2}= Enthalpy of steam after the evaporator

h_{appr}= Enthalpy at approach point temperature

t_{g2} can be calculated by using equation 15,

$\dot{m}_{gas}\bar{c}_p(t_{g2}\text{-}t_{g3})=\dot{m}_{st}(h_{s2}\text{-}h_{appr})$

h_{s2}= enthalpy of saturated steam at 17 bar

h_{appr}= 4.18x (196.3) = 820.7

90x1.05 (t_{g2}-222.6) =3.727x (2795-820.7)

t_{g2}=300.5^{o}C

Heat balance over super heater:

$\dot{m}_{gas}\bar{c}_p(t_{g1}\text{-}t_{g2})=\dot{m}_{st}(h_{sh}\text{-}h_{s2})$……………………………………………………………….(16)

$\dot{m}_{gas}$= Flue gas mass flow rate from six diesel engines

$\bar{c}_p$= specific heat of flue gas

t_{g1}= Flue gas input temperature to HRSG

t_{g2}= Flue gas temperature after economizer

$\dot{m}_{st}$ = Steam mass flow rate of HRSG

h_{sh}= Enthalpy of super heated steam

h_{s2}= Enthalpy of steam after the evaporator

For optimization pinch point temperature should fall between 8 ^{0}C and 16^{0}C. It is apparent that when the optimization is done with the pinch point temperature close to 8^{0}C the power output from HRSG is higher but the large scale size and high cost makes it uneconomical. On the other hand when the pinch point temperature gets closer to 16^{0}C the power output of HRSG gets limited. Above calculation indicate that exhaust gas out from HRSG is 194.4 ^{o}C. This means we can recover more energy.

Therefore, in order to investigate the attainable maximum possible power output with steam temperature at 280^{0}C and 17 bar, the following trial runs were carried out using EES.

Table 5.1 Pinch point temperature difference according to turbine power output (other design parameters remain same) (See Appendix 5)

	P_{st} [MW]	t_{ppdif} [C]	h_{fw} [kJ/kg]	t_{g1} [C]	t_{g2} [C]	t_{g3} [C]	t_{g4} [C]	t_s [C]	$\dot{m}_{st\ al,st}$ [kg/s]
Run 1	3	18.29	163.4	308	300.5	222.6	194.4	204.3	3.727
Run 2	3.1	15.44	163.4	308	300.2	219.8	190.7	204.3	3.851
Run 3	3.2	12.59	163.4	308	300	216.9	186.9	204.3	3.975
Run 4	3.3	9.748	163.4	308	299.7	214.1	183.1	204.3	4.099
Run 5	3.4	6.903	163.4	308	299.5	211.2	179.3	204.3	4.224
Run 6	3.5	4.057	163.4	308	299.2	208.4	175.5	204.3	4.348
Run 7	3.6	1.212	163.4	308	299	205.6	171.7	204.3	4.472
Run 8	3.7	-1.634	163.4	308	298.7	202.7	167.9	204.3	4.596
Run 9	3.8	-4.48	163.4	308	298.5	199.9	164.2	204.3	4.72
Run 10	3.9	-7.325	163.4	308	298.2	197	160.4	204.3	4.845
Run 11	4	-10.17	163.4	308	298	194.2	156.6	204.3	4.969
Run 12	4.1	-13.02	163.4	308	297.7	191.3	152.8	204.3	5.093
Run 13	4.2	-15.86	163.4	308	297.5	188.5	149	204.3	5.217
Run 14	4.3	-18.71	163.4	308	297.2	185.6	145.2	204.3	5.342
Run 15	4.4	-21.55	163.4	308	297	182.8	141.4	204.3	5.466
Run 16	4.5	-24.4	163.4	308	296.7	179.9	137.7	204.3	5.59
Run 17	4.6	-27.24	163.4	308	296.5	177.1	133.9	204.3	5.714
Run 18	4.7	-30.09	163.4	308	296.2	174.3	130.1	204.3	5.838
Run 19	4.8	-32.94	163.4	308	296	171.4	126.3	204.3	5.963
Run 20	4.9	-35.78	163.4	308	295.7	168.6	122.5	204.3	6.087
Run 21	5	-38.63	163.4	308	295.5	165.7	118.7	204.3	6.211

According to Table 5.1, when steam turbine power is 3.3MW pinch point temperature difference is 9.748°C and when steam turbine power is 3.4MW pinch point temperature difference is 6.903 °C.

Therefore, it is evident that the HRSG modeling is feasible to reach a considerable power output between 3.3-3.4MW in the context of APE with the pinch point temperature difference ranging close to 8^0C.

Secondly, EES run was carried out to find out the optimum steam pressure in the HRSG with the pinch point temperature kept constant at 8^0C.The steam pressure was varied between 5 bar to 17 bar. The limitation factor for setting the maximum possible power output was taken as the sulfur dew point temperature of the outgoing exhaust gas after the heat recovery.

Table 5.2 Turbine power output according to superheated steam pressure

	P_{st} [MW]	t_{ppdif} [C]	h_{fw} [kJ/kg]	t_{g1} [C]	t_{g2} [C]	t_{g3} [C]	t_{g4} [C]	t_s [C] $\dot{m}_{st}$	$\dot{M}_{total,st}$ [kg/s]	p_{sh} [bar]
Run 1	3.837	8	163.4	308	291.3	159.9	129.5	151.9	5.782	5
Run 2	3.835	8	163.4	308	292	164.2	133.4	156.2	5.66	5.6
Run 3	3.827	8	163.4	308	292.7	168.1	137	160.1	5.546	6.2
Run 4	3.814	8	163.4	308	293.3	171.8	140.4	163.8	5.44	6.8
Run 5	3.798	8	163.4	308	293.9	175.2	143.6	167.2	5.339	7.4
Run 6	3.779	8	163.4	308	294.4	178.4	146.6	170.4	5.244	8
Run 7	3.758	8	163.4	308	294.9	181.5	149.5	173.5	5.154	8.6
Run 8	3.735	8	163.4	308	295.4	184.3	152.2	176.3	5.068	9.2
Run 9	3.711	8	163.4	308	295.8	187	154.9	179	4.985	9.8
Run 10	3.685	8	163.4	308	296.2	189.6	157.4	181.6	4.905	10.4
Run 11	3.658	8	163.4	308	296.6	192.1	159.9	184.1	4.828	11
Run 12	3.631	8	163.4	308	297	194.5	162.2	186.5	4.754	11.6
Run 13	3.603	8	163.4	308	297.3	196.7	164.5	188.7	4.683	12.2
Run 14	3.574	8	163.4	308	297.6	198.9	166.8	190.9	4.613	12.8
Run 15	3.544	8	163.4	308	297.9	201	168.9	193	4.546	13.4
Run 16	3.515	8	163.4	308	298.2	203.1	171	195.1	4.48	14
Run 17	3.485	8	163.4	308	298.5	205	173.1	197	4.416	14.6
Run 18	3.454	8	163.4	308	298.8	207	175	199	4.354	15.2
Run 19	3.423	8	163.4	308	299.1	208.8	177	200.8	4.293	15.8
Run 20	3.393	8	163.4	308	299.3	210.6	178.9	202.6	4.234	16.4
Run 21	3.361	8	163.4	308	299.6	212.3	180.8	204.3	4.176	17

Lowest possible steam pressure indicates the highest turbine output and highest steam mass flow rate (see Table 5.2).

It is seen that EES Run 3 determines the best criteria for the HRSG modeling by surpassing the sulfur dew point which is 135^0C and maintaining a safe stack temperature.

Therefore, the thermodynamically optimum design criterion for HRSG is as follows:

Steam Inlet Pressure: 6.2 bar

Steam flow rate: 5.546 kg/s

Pinch point temperature difference: 8^0C

Estimated maximum power output of the steam turbine: 3.827 MW

5.1.3 Selection of a suitable commercial steam turbine

A low parameter steam turbine to suit the calculated optimum conditions was selected referring data given in Appendix 7 as follows.

Model S4-0.5 (See Appendix 7, [26]) seems the best selection in terms of power output (4 MW) but the fact that it requires low inlet steam pressure (5 bar) in turn brings the stack temperature to 129.5^{0}C according to Table 5.2, which is below the sulfur dew point and therefore is not acceptable as a technically feasible selection in the APE context.

Therefore the Model S3.69-1.27 is chosen instead as the best suitable steam turbine for the setup with a power output between 3.574-3.603MW at a steam inlet pressure of 12.7 bar and maintaining the pinch point temperature at 8^{0}C as calculated in Table 5.2.

According to above model, adapting the chosen steam turbine to the best matching conditions of the designed HRSG, would result in the following expected performance as shown in Table 5.3 below;

Power output of steam turbine = 3.579 MW
Steam pressure required = 12.7 bar

Table 5.3 Power output of the selected steam turbine, practically adapted to the best matching HRSG parameters [26]

									$\dot{m}_{st}$	
	P_{st}	t_{ppdif}	h_{fw}	t_{g1}	t_{g2}	t_{g3}	t_{g4}	t_s	$\dot{M}_{total,st}$	p_{sh}
	[MW]	[C]	[kJ/kg]	[C]	[C]	[C]	[C]	[C]	[kg/s]	[bar]
Run 1	*3.579*	*8*	*163.4*	*308*	*297.6*	*198.6*	*166.4*	*190.6*	*4.625*	*12.7*

6 Discussion and Conclusions

This work considered the conceptual modeling of a HRSG for the APE diesel engine plant in Sri Lanka, to improve the energy performance and overall efficiency of the power plant. The waste heat recovery potential from only the exhaust gas of the Caterpillar 16CM32C engines was investigated as a first step. At the initial stage the quantity of recoverable waste heat was calculated by comparison between an installed waste heat thermal oil boiler at APE and an engine without a waste heat recovery method at fully loaded condition. A calculation was performed taking into account the calorific value of fuel used and the operating conditions in order to find the maximum theoretical waste heat recovery potential of the six available engines of APE. It was found that the energy potential by recoverable process heat from one engine unit per hour is (Q_g) 9807.87MJ.

In the APE site, six engine units are available for energy recovery. According to technical and practical limitations such as pinch point temperature difference, approach point temperature difference, terminal temperature difference and sulfur dew point of stack, modeling process of HRSG was done using the Engineering Equation Solver software. The approach point temperature was selected as 8°C and avoiding temperature cross situations.

The Optimization of the modelled HRSG was performed using again the Engineering Equation Solver software. Trial runs were carried out to arrive at the maximum possible power output without compromising on size and cost. A second set of EES trial runs were carried out to find out the optimum steam pressure for HRSG keeping the pinch point temperature constant at 8^0C. The steam pressure was varied from 5 bar to 17 bar for the trials and the limiting factor for setting the maximum possible power output was taken as the sulfur dew point temperature of the outgoing exhaust gas after heat recovery.

In the modelling process, the results indicate that lowest possible steam pressure and highest steam temperature gives maximum power output from steam turbine. However, the commercially available small scale steam turbines in the market do not exactly match with our requirement. Then a steam turbine was selected with a maximum possible power output according to availability. Rated power output of selected turbine is 3.579 MW.

From the chosen steam turbine, 12884.4 MJ of energy was recovered per hour and this amount is 21.89% of recoverable total energy of flue gas per hour basis.

In conclusion,

Energy potential by recoverable process from one Caterpillar 16CM32C engine is (Qg) 9807.87 MJ per hour.

The approach point temperature was selected as 8°C in order to avoid temperature cross situations.

The optimization of the modeled HRSG was performed using the Engineering Equation Solver software.

The maximum possible power output without compromising on size and cost and the optimum steam pressure for the HRSG keeping the pinch point temperature constant at 8^0C were investigated using EES.

Estimated theoretical maximum power output of the steam cycle: 3.827 MW.

The rated power output of the selected optimum commercially available steam turbine is 3.69 MW (at 12.7 bar inlet pressure).

Power output of selected steam turbine in practice, if adapted to the desired parameters according to the HRSG design, would be 3.579 MW.

The recovered energy from the selected turbine is 12884.4 MJ which is 21.89% of total energy of flue gas on per hour basis.

7 Appendices

Appendix 1: From DNV petroleum service report (taken by APE for every fuel batch change in CPC)

DNV Petroleum Services - Fuel Analysis Report dated: 29-Sep-2011

Installation: ACE POWER EMBILIPITIYA

Sample Number	SNG1126407
Product Type	(HFO)
Sampling Date	26-SEP-2011
Sampling Point	STORAGE TK - KOLONNAWA
Sampling Method	CONTINUOUS DRIP
Sent From	COLOMBO
Date Sent	26-SEP-2011
Arrived at Lab	27-SEP-2011
Supplier	CEYLON PC
Sample Reference	TANK # 47

Seal Data DNVPS, SEAL INTACT, 5194386

Receipt Data	Unit	
Source Of Data*		Q.Cert
Density @ 15°C	kg/m^3	965.9
Viscosity @ 50°C	mm^2/s	171.5
Sulfur	% m/m	2.26

Test Parameter	Unit	Result
Net Calorific Value	MJ/kg	40.92

Appendix 2: From TUV industry service report (taken by APE as annual requirement)

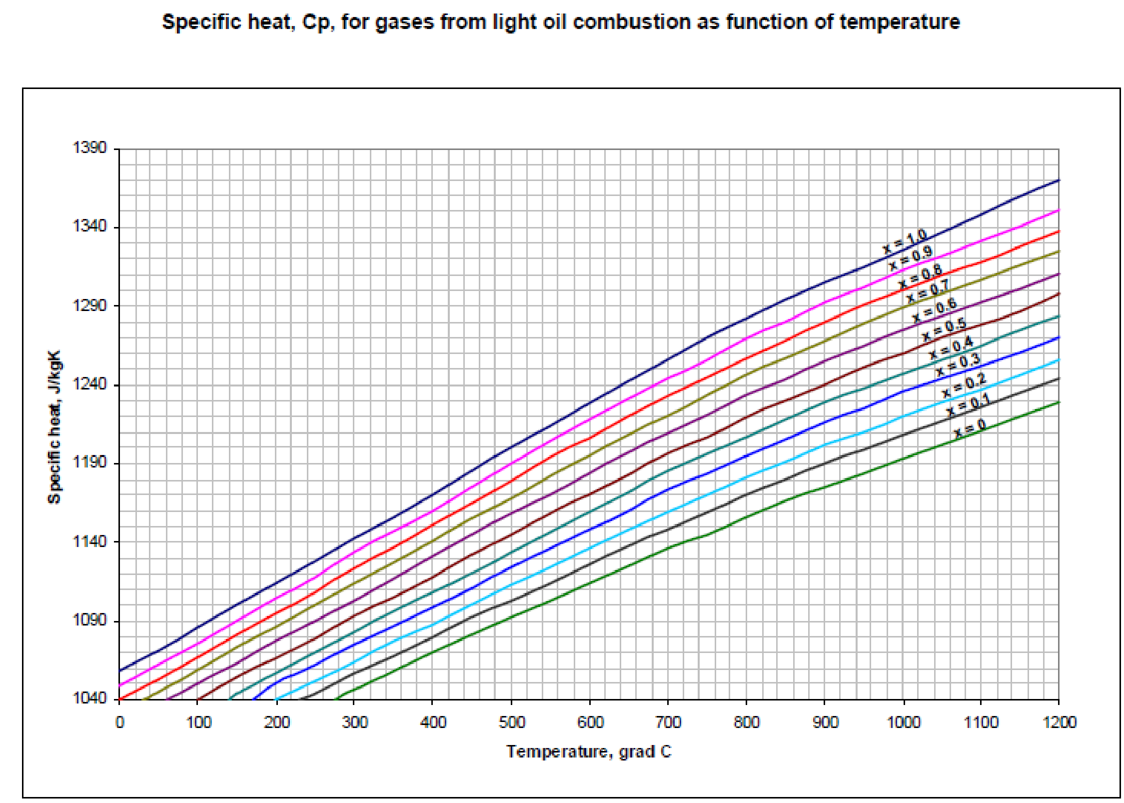
7.1.1 Engine No. 14

Test Number		1	2
Filter number		N10	N11
Date		09.03.09	09.03.09
Time		11:00 AM	11:38 AM
		11:30 AM	12:08 PM
Ancillary Parameters of Exhaust Gas			
Oxygen at dry condition	Vol.%	13,6	13,6
Carbon dioxid (calculated) at dry condition	Vol.%	5,5	5,5
Density dry (0°C ,1013hPa)	kg/m3	1,320	1,320
Barometric pressure	hPa	1002	1002
Static pressure in stack	hPa	8,0	8,0
Exhaust gas temperature	Grd.C	308	308
Exhaust gas humidity x	kg/kg	0,05	0,05
Exhaust gas humidity F	Vol.%	7,4	7,4
Density of exhaust gas(0 °C ,1013 hPa, wet)	kg/m3	1,283	1,283
Density of exhaust gas(wet)	kg/m3	0,601	0,601
Value of the square root of the dynamic pressure	••••	1,825	1,825
Flow speed of exhaust gas	m/s	33,3	33,3
Cross section of measuring plane	m2	0,785	0,785
Exhaust gas volume flow rate wet	m3/h	94125	94125
Exhaust gas volume flow rate (0 °C ,1013hPa, wet)	m3/h	44109	44109
Exhaust gas volume flow rate (0 °C, 1013hPa, dry)	m3/h	40904	40904
Ancillary Parameters of Sampling System			
Static pressure	hPa	82	66
Temperature gasmeter	Grd.C	12	31
Sampling Volume dry (gasmeter)	m3	1,259	1,214
Sampling Volume dry (0 °C ,1013hPa)	m3	1,096	1,007
Sampling Volume wet (0 °C,1013hPa)	m3	1,182	1,086
Sampling volume flow at probe	m3/h	5,045	4,635
Cross section (probe)	mm2	38,5	38,5
Sampling volume vilocity in probe	m/s	36,4	33,5
Total Mass	mg	*)	66,6
Sampling duration	min	30	30
Results			
Particulate matter (wet)	mg/m3	*)	28,7
Particulate matter (0 °C,1013hPa, wet)	mg/m3	*)	61,3
Particulate matter (0 °C,1013hPa, dry)	mg/m3	*)	66,1
Mass flow	kg/h	*)	2,71

*) The filter of the first sampling couldn´t be validated. There was a damage of the filterbox, caused by transport.

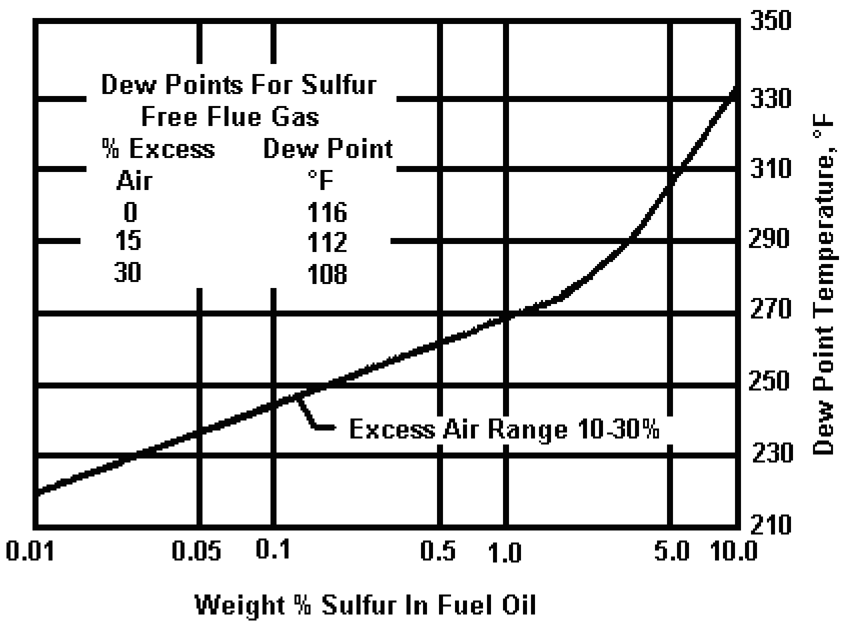
Aitken Spence Power

Appendix 3:



Appendix 4:

Acid dew point temperature for fuel with different sulfur percentage



Appendix 5: EES program code for the thermodynamic calculations

```
p_sh=17[bar];c_p=1.05[kJ/kg·K];P_st=3[MW];t_sh=280[C];p_out=0.07[bar];eta_is=0.93;eta_m=0.98;m_dot_gas=
90[kg/s];t_g1=308[C];hl=2"%"; C=4.18[kJ/kg·C]

{The pinch point temperature difference, t_ppdif is}

{t_ppdif=8[C]}

t_ppdif= t_g3-t_s

t_apprdif=8[C]

t_appr=t_s-t_apprdif

t_s=T_SAT(Steam,P=p_sh*(convert(bar,kPa)))  {For the pressure of 17 bars}

h_s1=ENTHALPY(water,x=0,P=p_sh*(convert(bar,kPa)))

h_s2=ENTHALPY(steam,x=1,P=p_sh*(convert(bar,kPa)))


"In order to calculate the gas temperature tg3we can make a heat balance over the  evaporator and the
superheater, as we knowevaporation data and superheat data on  the steam side: "


m_dot_gas *c_p*(t_g1-t_g3)=m_dot_st*(h_sh-h_appr) "----(1)"


{In the equation with do not know t_g3or the steam mass flow, m_st.In order to calculate  the gas temperature tg3,
we first have to calculate the steam mass flow}

{The steam mass flow can be found via a heat balance over the turbine}

P_st*(convert(MW,kW))=m_dot_st*(h_sh-h_out)*eta_m "---(2)"


s=ENTROPY(Steam,T=t_sh,P=p_sh*(convert(bar,kPa)))

h_outis=ENTHALPY(Steam,S=s,P=p_out*(convert(bar,kPa)))

{The superheated steam enthalpy (17bars, 280°C) is found in h-s diagram or in steam}

h_sh=ENTHALPY(Steam,T=t_sh,P=p_sh*(convert(bar,kPa)))

{The enthalpy out from the turbine is found in a h-s diagram, by first finding the isentropic outlet enthalpy.
Thereafter the real outlet enthalpy can be calculated}

h_out=h_sh-eta_is*(h_sh-h_outis) "----(3)"

{The stack temperature can be calculated with a similar heat balance as in the previous task, this time with a span
over the whole steam cycle}

m_dot_gas*c_p*(t_g1-t_g4)*((100-hl)/100)=m_dot_st*(h_sh-h_fw)

{It can be assumed that the specific heat is the same from tg1to tg3 as from tg1to tg4. }

h_fw=ENTHALPY(water,x=0,P=p_out*(convert(bar,kPa)))


{h_appr=ENTHALPY(Water,T=t_appr,P=p_sh*(convert(bar,kPa)))}

h_appr=C*t_appr


{heat balance  Evaporator}

m_dot_gas*c_p*(t_g2-t_g3)=m_dot_st*(h_s2-h_appr)
```

$p_{sh} = 17$ [bar] $c_p = 1.05$ [kJ/kg·K] $P_{st} = 3$ [MW] $t_{sh} = 280$ [C]

$p_{out} = 0.07$ [bar] $\eta_{is} = 0.93$ $\eta_m = 0.98$ $\dot{m}_{gas} = 90$ [kg/s]

$t_{g1} = 308$ [C] $hl = 2$ % $C = 4.18$ [kJ/kg·C]

$$t_{ppdif} = t_{g3} - t_s$$

$$t_{apprdif} = 8 \quad [C]$$

$$t_{appr} = t_s - t_{apprdif}$$

$$t_s = T_{sat}\left[\text{Steam}, P = p_{sh} \cdot \left|100 \cdot \frac{kPa}{bar}\right|\right]$$

$$h_{s1} = \mathbf{h}\left[\text{water}, x = 0, P = p_{sh} \cdot \left|100 \cdot \frac{kPa}{bar}\right|\right]$$

$$h_{s2} = \mathbf{h}\left[\text{Steam}, x = 1, P = p_{sh} \cdot \left|100 \cdot \frac{kPa}{bar}\right|\right]$$

In order to calculate the gas temperature tg3we can make a heat balance over the evaporator and the superheater, as we knowevaporation data and superheat data on the steam side:

$$\dot{m}_{gas} \cdot c_p \cdot [t_{g1} - t_{g3}] = \dot{m}_{st} \cdot [h_{sh} - h_{appr}] \quad \text{----}(1)$$

$$P_{st} \cdot \left|1000 \cdot \frac{kW}{MW}\right| = \dot{m}_{st} \cdot [h_{sh} - h_{out}] \cdot \eta_m \quad \text{---}(2)$$

$$s = \mathbf{s}\left[\text{Steam}, T = t_{sh}, P = p_{sh} \cdot \left|100 \cdot \frac{kPa}{bar}\right|\right]$$

$$h_{outis} = \mathbf{h}\left[\text{Steam}, s = s, P = p_{out} \cdot \left|100 \cdot \frac{kPa}{bar}\right|\right]$$

$$h_{sh} = \mathbf{h}\left[\text{Steam}, T = t_{sh}, P = p_{sh} \cdot \left|100 \cdot \frac{kPa}{bar}\right|\right]$$

$$h_{out} = h_{sh} - \eta_{is} \cdot [h_{sh} - h_{outis}] \quad \text{----}(3)$$

$$\dot{m}_{gas} \cdot c_p \cdot [t_{g1} - t_{g4}] \cdot \left[\frac{100 - hl}{100}\right] = \dot{m}_{st} \cdot [h_{sh} - h_{fw}]$$

$$h_{fw} = \mathbf{h}\left[\text{water}, x = 0, P = p_{out} \cdot \left|100 \cdot \frac{kPa}{bar}\right|\right]$$

$$h_{appr} = C \cdot t_{appr}$$

$$\dot{m}_{gas} \cdot c_p \cdot [t_{g2} - t_{g3}] = \dot{m}_{st} \cdot [h_{s2} - h_{appr}]$$

SOLUTION

Unit Settings: SI C kPa kJ mass deg

$C = 4.18$ [kJ/kg·C] $c_p = 1.05$ [kJ/kg·K] $\eta_{is} = 0.93$

$\eta_m = 0.98$ $hl = 2$ $h_{appr} = 820.7$ [kJ/kg]

h_{fw} = 163.4 [kJ/kg]	h_{out} = 2164 [kJ/kg]	h_{outis} = 2102 [kJ/kg]
h_{s1} = 872 [kJ/kg]	h_{s2} = 2795 [kJ/kg]	h_{sh} = 2985 [kJ/kg]
$\dot{m}_{gas}$ = 90 [kg/s]	$\dot{m}_{st}$ = 3.727 [kg/s]	p_{out} = 0.07 [bar]
p_{sh} = 17 [bar]	P_{st} = 3 [MW]	s = 6.77 [kJ/kg·K]
t_{appr} = 196.3 [C]	$t_{apprdif}$ = 8 [C]	t_{g1} = 308 [C]
t_{g2} = 300.5 [C]	t_{g3} = 222.6 [C]	t_{g4} = 194.4 [C]
t_{ppdif} = 18.29 [C]	t_s = 204.3 [C]	t_{sh} = 280 [C]

No unit problems were detected.

Appendix 6: Calculation Equations

$$HHV=17884+57.5\times {}^{o}API-102.2\times\%S \quad (1)$$

$$LHV=HHV-91.23\times\%H_2 \quad (2)$$

$$\%H_2=F-\frac{2122.5}{{}^{0}API+131.5} \quad (3)$$

$$s=\frac{141.5}{131.5+{}^{0}API} \quad (4)$$

$$\Delta t_{pp}= t_{g3}-t_s \quad (9)$$

$$\dot{m}_{totalgas}.\bar{c}_p\left(t_{g1}-t_{g3}\right)=\dot{m}_{totalst}.\left(h_{sh}-h_{appr}\right) \quad (10)$$

$$P_{st}=\dot{m}_{totalst}.\left(h_{sh}-h_{out}\right)\eta_{m+g} \quad (11)$$

$$h_{out}=h_{sh}-\eta_{is}\left(h_{sh}-h_{out,is}\right) \quad (12)$$

$$\dot{m}_{total\ gas}.\bar{c}_p\left(t_{g1}-t_{g4}\right)(1-hl)=\dot{m}_{totalst}.\left(h_{sh}-h_{fw}\right) \quad (13)$$

$$\dot{m}_{totalgas}.\bar{c}_p\left(t_{g3}-t_{g4}\right)=\dot{m}_{totalst}.\left(h_{appr}-h_{fw}\right) \quad (14)$$

$$\dot{m}_{totalgas}.\bar{c}_p\left(t_{g2}-t_{g3}\right)=\dot{m}_{totalst}.\left(h_{s2}-h_{appr}\right) \quad (15)$$

$$\dot{m}_{totalgas}.\bar{c}_p\left(t_{g1}-t_{g2}\right)=\dot{m}_{totalst}.\left(h_{sh}-h_{s2}\right) \quad (16)$$

Appendix 7: Commercially available steam turbine models of the required size and input parameters [26] (Qingneng Power, http://www.qnpower .com / product html /152.html)

NO.	Model	Capacity (MW)	Speed (r/min)	Inlet			Exhaust Pressure (Mpa)	Weight (t)	Overall Dimensions LxWxH(mm)
				Pressure (MPa)	Dryness ( )	Temp ( )			
1	S1.0-0.3	1	3000	0.3		200	0.008	14.6	3220x2150x1750
2	S1.3-0.36	1.3	3000	0.36		180	0.013	18.2	4550x2300x2600
3	S1.5-0.14	1.5	3000	0.14	0.995		0.0072	18.7	3200x2300x2532
4	S1.5-0.16	1.5	3000	0.16	0.995		0.0088	18.7	3200x2300x2532
5	S1.5-1.7	1.5	3000	1.7		240	0.098	9.8	4120x2650x2530
6	S2-0.6	2	5600	0.6		275	0.0061	12	4257x2145x2375
7	S2.6-1.08	2.6	3000	1.08	0.995		0.009	17.7	3500x2850x2500
8	S3-0.5	3	3000	0.5	0.995		0.009	16	3250x2850x2500
9	S3-0.5	3	3000	0.5		230	0.009	16	3250x2850x2500
10	S3-0.5	3	3000	0.5		270	0.008	15.7	3500x2250x1750
11	S3.69-1.27	3.69	3000	1.27		300	0.007	17.7	3500x2850x2500
12	S3.9-1.08	3.9	3000	1.08	0.995		0.009	17.7	3500x2850x2500
13	S4-0.5	4	3000	0.5		230	0.01	16.3	3300x2840x2500
14	S5-1.0	5	3000	1		260	0.01	17.9	3510x2830x2485
15	S6-0.5	6	3000	0.5		230	0.01	38.1	5000x3900x2610
16	S6-0.5	6	3000	0.5	0.995		0.01	38.1	5000x3900x2610
17	S6-1.0	6	3000	1		230	0.009	42.1	5160x3900x2600
18	S6-1.0	6	3000	1	0.995		0.01	42.1	5160x3900x2600
19	S8-1.0	8	3000	1		260	0.01	42.9	5160x3900x2600
20	S8-1.0	8	3000	1	0.995		0.01	42.9	5160x3900x2600
21	S9-1.35	9	3000	1.35		310	0.005	43.2	5190x3900x2590
22	S10-0.981	10	3000	0.981		300	0.008	43.9	5200x3900x2590
23	S10-1.0	10	3000	1		260	0.01	43.8	5160x3900x2600
24	S10-1.0	10	3000	1	0.995		0.01	43.9	5160x3900x2600
25	S12-0.785	12	3000	0.785		415	0.0073	45	5205x3770x2450
26	S12-1.0	12	3000	1	0.995		0.01	42.9	5160x3900x2600
27	S12-1.25	12	3000	1.25		315	0.01	39.1	4910x3900x2600

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