Waste Heat Recovery

HOUSSAM ACHKOUDIR
NAOWAR HANNA

KTH Industrial Engineering
and Management

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Study of the efficiency potential of water based Rankine WHR system with a piston expander

Houssam Achkoudir

Naowar Hanna

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Studie av verkningsgrad potentialen för ett vatten baserat Waste Heat Recovery system med kolvexpander

Houssam Achkoudir, houssam@kth.se
Naowar Hanna, naowar@kth.se

Godkänt	Examinator	Handledare
2011-02-25	Hans-Erik Ångström, KTH	Gustav Ericsson, KTH
	Uppdragsgivare Scania AB	Kontaktperson Johan Linderyd, Scania

Sammanfattning

En ånganna monterades i EGR (Exhaust Gas Recirculation) slingan på en 12,7 liters Scania Euro V motor (DC1306). En modell som beskriver Rankine cykel togs fram med vatten som köldmedium i simuleringsverktyget GT-Power. Ångpannan i GT-power modellen kalibrerades m.h.a experimentell data.

Simuleringarna visade att det optimala ångtrycket, det trycket där högst effekt kan erhållas från expandern, är beroende av EGR temperaturen. Det innebär att ju högre EGR inloppstemperatur desto högre optimalt ångtryck. EGR temperaturen i punkt 2 i ESC cykeln är 514˚ C för denna motor, vilket resulterar i ett optimalt tryck på 120 bar enligt simuleringen. Vidare analyserades den optimala överhettningsgraden, vilket innebär antalet grader som ångan uppvärms vid konstant tryck efter att allt vatten har förångats. Simuleringarna visar att högst effekt i expandern erhålls då ångan överhettas 10 grader, alltså den lägsta överhettningsandelen. Detta beror på att ångeffekten från ångpannan är högst vid lägst andel överhettningsgrad, vilket beror på ett högre vatten flöde.

Simuleringen visar att EGR temperaturen är viktigare än EGR flödet. Ett sätt att öka EGR temperaturen är genom att tillsatselda. Detta innebär att spruta in bränsle i avgasröret. Beräkningar visar dock att det är lönsammare att spruta in bränslet direkt i förbränningsrummet. Att öka EGR temperaturen med 150˚C skulle resultera i ca 2,5 *kW* ökning i erhållen effekt från exandern. Att spruta in samma dieselflöde i förbränningsrummet skulle medföra en effektökning med ca 5,2 *kW* från motorn.

Vid det optimala ångtrycket samt 10 graders överhettad ånga minskar bränsleförbrukningen för punkt 2 i ESC cykeln med 1,4 %. WHR (Waste Heat Recovery) systemet verkningsgrad ligger på 18,4 %. Vid montering av ytterliggare en ångpanna efter turbinen för att på så vis utnyttja energin i avgasflödet, istället för bara i EGR gaserna, skulle bränsleförbrukningen minska med 3,41 %.

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Study of the efficiency potential for a water based Waste Heat Recovery system with piston expander

Houssam Achkoudir, houssam@kth.se
Naowar Hanna, naowar@kth.se

Approved 2011-02-25	Examiner Hans-Erik Ångström, KTH	Supervisor Gustav Ericsson, KTH
	Commissioner Scania AB	Contact person Johan Linderyd, Scania

Abstract

An evaporator was mounted in the EGR loop of a 12,7 liter Scania Euro V engine (DC1306). A model describing the Rankine cycle was developed with water as refrigerant in the simulation tool GT-Power. The evaporator in the GT-Power model was calibrated with experimental data.

The simulations showed that the optimal vapor pressure where the maximum power available from the expander is obtained depends on the EGR temperature. Higher EGR inlet temperature leads to increased optimal vapor pressure. The EGR temperature in case 2 of the ESC cycle is 514 °C for the engine above, this result in an optimal vapor pressure of 120 bar according to the simulation. The optimum level of superheating was analyzed, which means the amount of degrees the vapor temperature is raised at a constant pressure after all the water is evaporated. The simulations show that the highest power in the expander was obtained when the steam was superheated by 10 degrees, i.e. the lowest level of superheating. The steam power after the evaporator is highest at the lowest level of superheating, because of the higher refrigerant flow.

Simulations show that the EGR temperature has a bigger impact than the EGR flow. One way to increase the EGR temperature is by supplementary burning, which means injecting fuel into the exhaust pipe. Calculations show that it is more profitable to inject fuel directly into the combustion chamber. Increasing the EGR inlet temperature with 150 °C would result in 2,5 *kW* higher power output from the expander. Injecting the same fuel flow in the combustion chamber the engine power output increases with 5,2 *kW*.

Operating point 2 in the ESC cycle reduces the fuel consumption with 1,4 % if run at the optimal steam pressure of 120 bar and 10 degrees of superheated vapor. The reduction of the fuel consumption would be 3,41 %, if the power in the exhaust mass flow would be utilized by integrating another evaporator after the turbine.

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We also thank Sten Dahlman who build the evaporator and Peter Platell (Ranotor) for letting us use the evaporator and helped us to understand the Rankine cycle.

Table of contents

Nomenclature

HC - Unburned hydrocarbons

GHG - Green House Gases

WHR - Waste Heat Recovery

EGR - Exhaust gas recirculation

GT-Power – Is an industry standard engine simulation tool that is specially designed for both steady state and transient simulation

L - Liqiud

V - Vapor

CP - Critical Point

Hp - Horse power

T-S diagram – Temperature Entropy diagram

PID regulator – Proportional Integral Derivate regulator

ESC cycle – European Stationary Cycle

VGT – Variable Geometry Turbocharger

$\dot{Q}_{in}$ - Heat flow rate to the system (energy per unit time)

$\dot{Q}_{out}$ - Heat flow rate from the system (energy per unit time)

$\dot{W}$ - Mechanical power consumed by or provided to the system (energy per unit time)

E_{Steam} - Steam power (power input to the expander)

A_{HT} - Heat transfer area

A_F - Flow area

r - Radius

n_{tubes} - Number of tubes

L_{Tube} - The length of a micro tube, which is $3\ m$

E_{Evap} - The power output from the evaporator

Δh_{Evap} - The change in specific enthalpy across the evaporator

$\dot{m}_{ref}$ - The mass flow of the refrigerant

η_{EGR} - Evaporator's temperature efficiency based on the cooling ability

η_{Steam} - The evaporator's efficiency could also be described as how good the evaporator is in terms of heating up the refrigerant

$T1_{cool}$ – The inlet cooling water temperature to the condenser

$P1_{Cool}$ – The inlet cooling water pressure to the condenser

$\dot{m}_{cool}$ - The cooling water mass flow

$T2_{cool}$ – The cooling water temperature out from the condenser

$P2_{Cool}$ – The cooling water pressure out from the condenser

$T1_{Ref}$ – The water inlet temperature (after the condenser)

$P1_{ref}$ - The water inlet pressure (after the condenser)

$\dot{m}_{ref}$ - The mass flow in the refrigerant circuit

$T2_{Ref}$ - The inlet temperature of the refrigerant (equal to $T1_{Ref}$)

$P2_{ref}$ - The water pressure after the pump

$T2_{EGR}$ - The outlet temperature of the EGR

$P2_{EGR}$ - The outlet pressure of the EGR

$T1_{EGR}$ - The inlet temperature of the EGR

$P1_{EGR}$ - The inlet EGR pressure

$\dot{m}_{EGR}$ - The mass flow for the EGR

$T3_{Ref}$ - The steam temperature

$P3_{Ref}$ - The steam pressure

$T4_{Ref}$ - The temperature after the expander (inlet temperature to the condenser)

$P4_{Ref}$ - The pressure after the expander (inlet pressure to the condenser)

η_{T} - The turbines isentropic efficiency

E_{Exp} - The expander power output

$E_{Condenser}$ - The heat load in the condenser

$\Delta h_{Condenser}$ - The change in specific enthalpy across the condenser

C - The turbulent coefficient

m - The turbulent exponent

Nu -The Nusselts number

Re - The Reynolds number

Pr - The Prandels number

h - The film coefficient

L_{ref} - The reference length

k - The thermal conductivity

C_p - The heat capacity

μ - The dynamic viscosity

ρ_{cool_water} - The cooling water density

U - The velocity of the cooling water

V_s – The pump/turbine displacement

η_v - The volumetric efficiency for the pump/turbine

n – Revolution per seconds for the pump/turbine

$\dot{m}$ – The mass flow for the pump/turbine [*kg/s*]

ρ - The inlet density for the pump/turbine

RFC - Reduction of fuel consumption

E_{Pump} - The pump power

E_{Engine} - The power the engine requires at the current loading point

η_{WHR} - The WHR efficiency

h_1 - The enthalpy of the refrigerant in liquid state

h_2 - The steam enthalpy, provided by moliers diagram if temperature and pressure is known

P_{Steam} - The steam pressure

P_{Water} - The water pressure while the refrigerant is in liquid state

h'_{20° - Enthalpy for saturated water at 20 ˚C

ν_{water} - The specific volume of water at 20 ˚C

E_{EGR} - The power input to the system (to the evaporator)

$C_{p_{EGR}}$ - The heat capacity for EGR

$\dot{m}_{Diesel}$ - The diesel mass flow

H_{Diesel} - The diesel specific energy [*J/kg*]

x - The degrees the EGR is to be raised

η_{Engine} - The efficiency of a diesel engine (assumed to be 40 %)

P1- The refrigerant pressure before the evaporator

P2- The steam pressure after the evaporator

T2-1- The steam temperature after the evaporator

T2-2- The steam temperature after the evaporator (two thermocouples)

T_A-N – Sixteen thermocouples from A-N mounted on the micro tubes inside the evaporator

1. Introduction

The increasing oil price is putting pressure on the industry to reduce the fuel consumption, and at the same time try to achieve the rigorous emission legislations. Also an increased focus on CO_2 and GHG (Green House Gases) driven by the concern for global warming is a strong driver to increase the efforts to develop technologies that improves fuel consumption. One way to do it is to employ a hybrid system. Hybrid generally utilizes kinetic energy recovery, the energy can be used to generate electricity and charge an electric storage device. It would work if the aim is to drive in the city where there is a lot of start/stop, however the power output for a long haul truck is constant and high, so the battery in hybrid does not last for long.
Another way is to take advantage of the heat energy in the exhaust gases, by either use turbo compound or Rankine cycle. This report is based on using the Rankine cycle. By using the Rankine cycle, the heat energy from the exhaust gas can be used to heat up a pressurized fluid into vapor and obtain power by expanding the vapor. The power can be used to assist the engine by adding torque to the engine output. The energy can also be used in a hybrid system, it can be used to generate electricity and charge a battery.
The aspect has been interesting for many companies, because there is a lot of energy that is wasted and can be used. However there is still a lot to understand and evaluate about the WHR system for the companies that invest in it. It is important to understand the effect of the different parameters on the WHR system.

This project will evaluate and discuss the possibilities with a proposed WHR Rankine system based on water as working fluid and a special design evaporator capable of handling high steam pressures. To be able to do that, a 6 cylinder 12,7 liter Scania diesel engine (DC1306) with an evaporator mounted on the EGR-root were run. The experimental data obtained from the tests have been used to calibrate a model in GT-Power [1]. In order to investigate the performance of the entire WHR system, a condenser, pump, receiver and expander has been integrated into the model.

1.1 Background

In a previous project about WHR system [2], a Rankine cycle with water as refrigerant was studied. This was done by simulating an evaporator in GT-Power, then mounting a physical evaporator in the EGR loop of the engine. The evaporator replaced the EGR cooler, and the EGR cooler was moved downstream the evaporator because of high EGR outlet temperature after the evaporator. The simulation and experimental results was then compared.

The purpose of this master thesis is to simulate the entire Rankine cycle, this means integrating a pump, expander, receiver and condenser into the model from the previous project. The objective has been to understand the influence of different parameters, understand fuel consumption potential for a complete system and the performance of the special designed evaporator.

1.1.1 Rankine cycle

The Rankine cycle [8] is about converting heat into work, figure 1 describes the Rankine cycle.

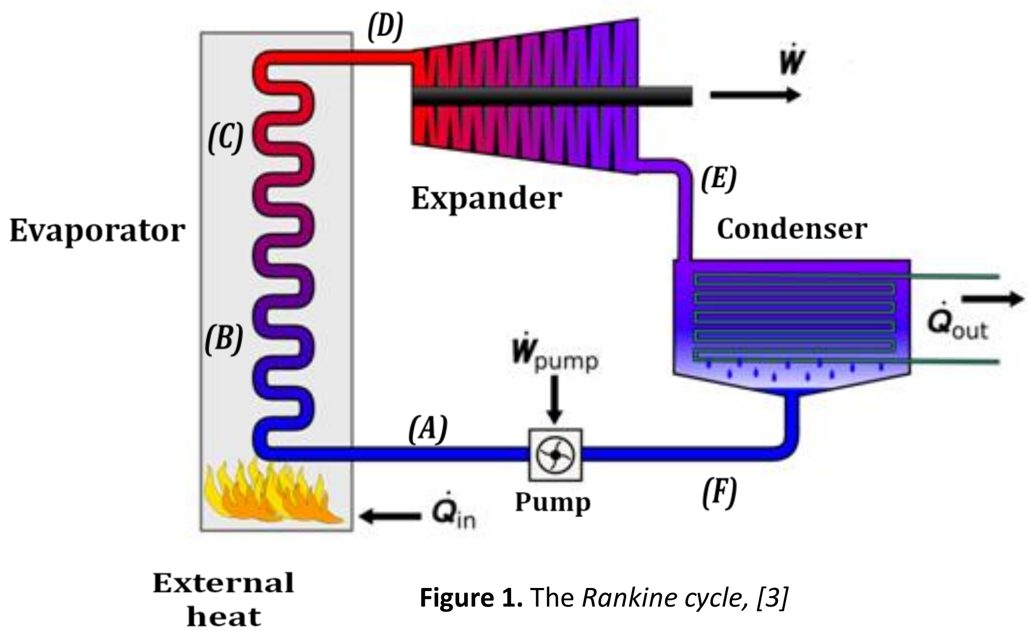


Figure 1. The *Rankine cycle, [3]*

In stage (F), the working fluid has a low pressure and is pressurized by a pump into a high pressure phase. In stage (A) the pressurized fluid enters a boiler that is heated by a external heat source, $\dot{Q}_{in}$, and the fluid turns in to vapor at a certain input energy. The transformation from liquid to vapor depends on the pressure and temperature of the working fluid, see figure 4. The vapor is expanded from stage (D) to (E) which generates a certain power output $\dot{W}$.
The vapor is condensed in the condenser. A certain heat flow rate $\dot{Q}_{out}$ is required to condense the vapor which in this case is water cooled. A decrease in temperature and pressure will occur and the vapor is condensed into liquid and back to stage (F).

The heat source in this case is the waste heat in the EGR gases, which normally is cooled by the coolant water. The energy losses are the power input to the pump and the energy needed to condense the vapor. The energy needed to condense the vapor can be solved in several ways, like using the cooling water from the engine. Mounting the evaporator in the EGR loop also means reducing the cooling demands on the engine since theoretically there is no need for the EGR cooler. This power can instead be used to condense the vapor.

The refrigerant circuit in GT-Power is illustrated in figure 2. The receiver in the figure is the tank which is used to collect the water in the system.

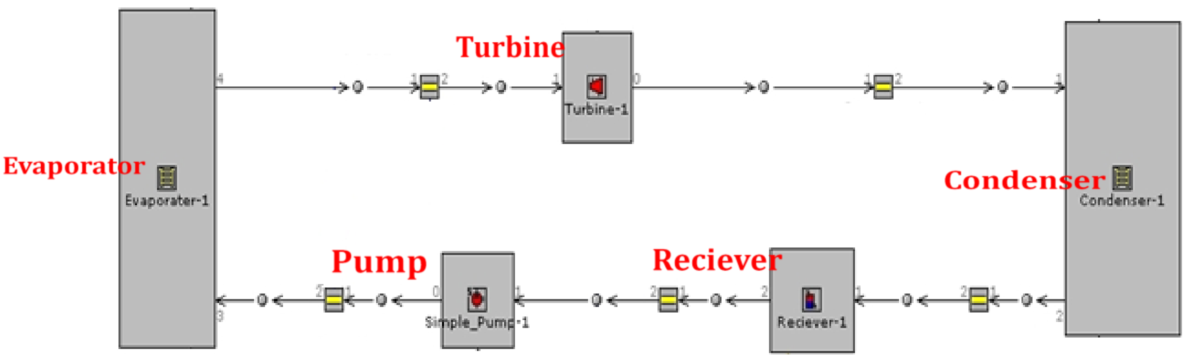


Figure 2. *WHR system*

1.1.2 Superheated vapor

The different phases of the refrigerant inside the evaporator are described with a temperature-enthalpy diagram (hT-diagram), see figure 3.
The first part of the red line, A-B, in figure 3 represents the refrigerant heating up until the boiling point is reached. The second part, B-C, illustrates the enthalpy needed to evaporate all the refrigerant i.e. only steam exists at point C. The final part, C-D, is the superheated part, showing the steam superheat at constant pressure. As can be seen most of the input energy is consumed during the second part i.e. the evaporation of the refrigerant.

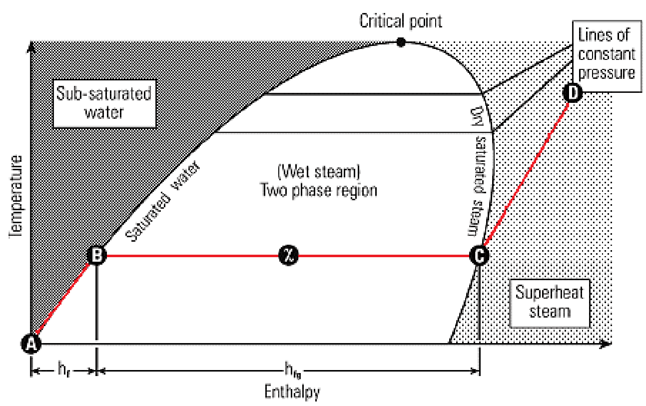


Figure 3. *Different states* of the refrigerant

The letters (A-D) in figure 1 and 3 describes the same phase.
It is important that the mass flow entering the expander is fully evaporated. To achieve this, the steam will always be at least 10 degrees superheated. It is also important to understand the relationship between the temperature and pressure of the refrigerant, figure 4 shows the boiling temperature for several pressures for water i.e. when the water starts to evaporate.

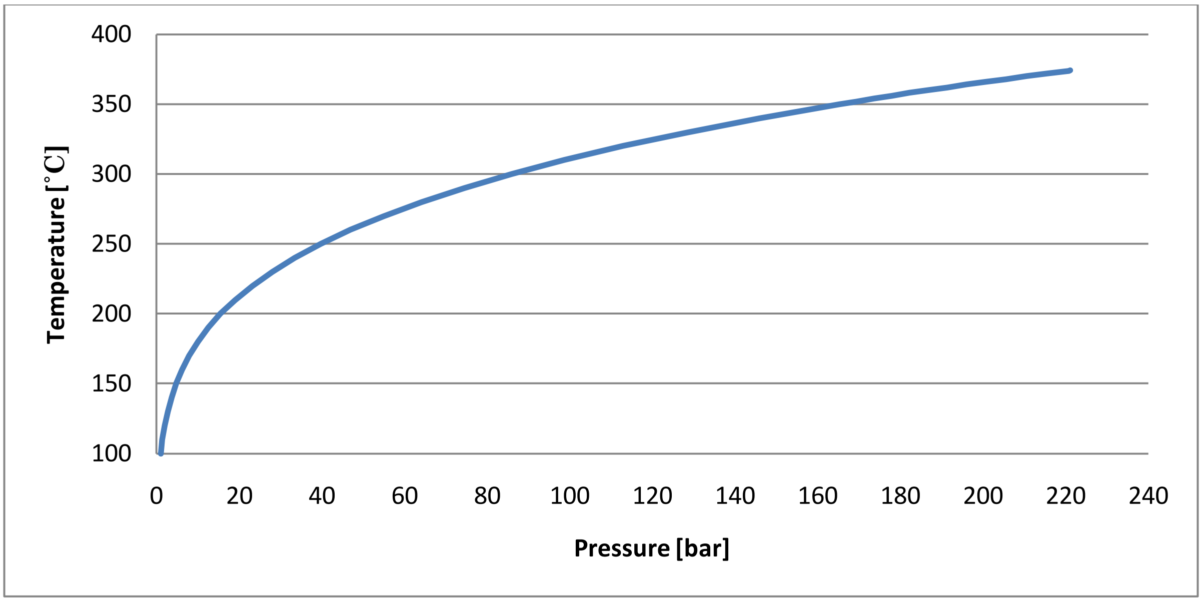


Figure 4. *Boiling temperatures for different water pressures [4]*

1.1.3 Working fluid

Working fluids for WHR system are according to the slope of the saturated vapor line in the temperature-entropy diagram, usually divided into three types: dry, wet and isentropic. Corresponding rankine cycles for the three types are shown in figure 5 [5]. L and V stands for liquid and vapor, CP stands for critical point.

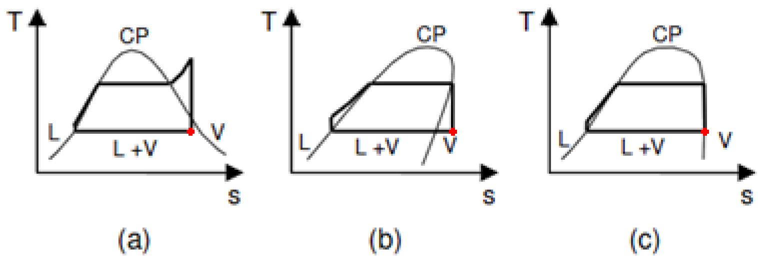


Figure 5. *Three fluid types: (a) wet fluid; (b) dry fluid; (c) isentropic fluid*

The wet fluid is in general inorganic fluid, such as water, ethanol, ammonia etc. As seen in the picture above, at the end of the expansion (the red point in the picture), the wet fluid generally ends up in the two phase area (L+V). The dry fluid however has a positive slope for the saturated vapor line, and at the end of the expansion process the fluid ends as superheated vapor. Some examples of a dry fluid are Benzene and R245fa. The isentropic fluid, the saturated vapor line is almost vertical in the diagram for most temperature range, and therefore the expansion process stays as saturated vapor.

If a dry fluid is chosen as the working fluid for a Rankine cycle, a recuperator can be used after the expander to utilize the remaining energy in the superheated vapor and improve the cycle efficiency. The disadvantage is the enlarged cost and the packaging of the system can be difficult due to increased number of components. The less dry the fluid is, the smaller the size of the recuperator is required.

1.1.4 Subcritical and supercritical cycle

Depending on the temperature and pressure of a fluid, it can either be subcritical or supercritical. Supercritical cycle is defined by pressures larger than the refrigerants critical point and subcritical by pressures below its critical point. The refrigerants T-S diagram can show if the cycle is subcritical or supercritical, assuming the temperature and pressure is known. Subcritical and supercritical cycles for two different working fluids will be explained below.

Figure 6 shows a T-S diagram over a subcritical cycle for a dry fluid, R245fa, [6]. The efficiency is $18{,}1\ \%$ at an operation parameter of: pump pressure $=\ 19{,}3\ bar$, condensation pressure $=\ 3\ bar$, evaporation temperature $= 120°C$, condensation temperature $= 45°C$ and a maximum superheating temperature at $350°C$ though R245fa becomes thermally unstable at higher temperatures.

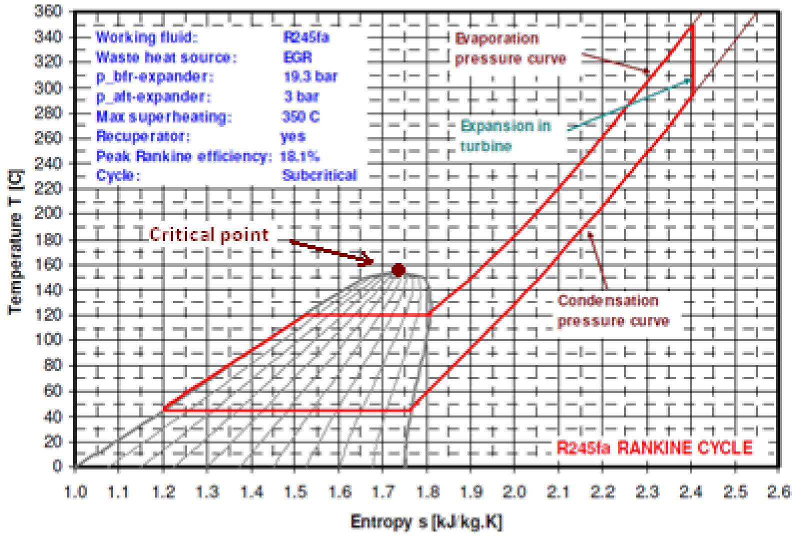


Figure 6. *T-s diagram of a subcritical rankine cycle for R245fa*

Supercritical cycle of the same fluid is shown in figure 7. The Rankine cycle efficiency is 15,8 % at an operation parameter of: pump pressure = 100 bar, condensation pressure = 3 bar. As seen in the picture below, the superheating is controlled such that the end of the expansion process enters two phase state, and therefore the superheating temperature is $190°C$. Subcritical cycle for this type of fluid has a higher efficiency than supercritical cycle, though a recuperator is used in the subcritical cycle. As seen in the figure, the expansion ratio is too large for a single-stage turbine, and therefore some other kind of expander is needed or advanced turbine with higher expansion ratio.

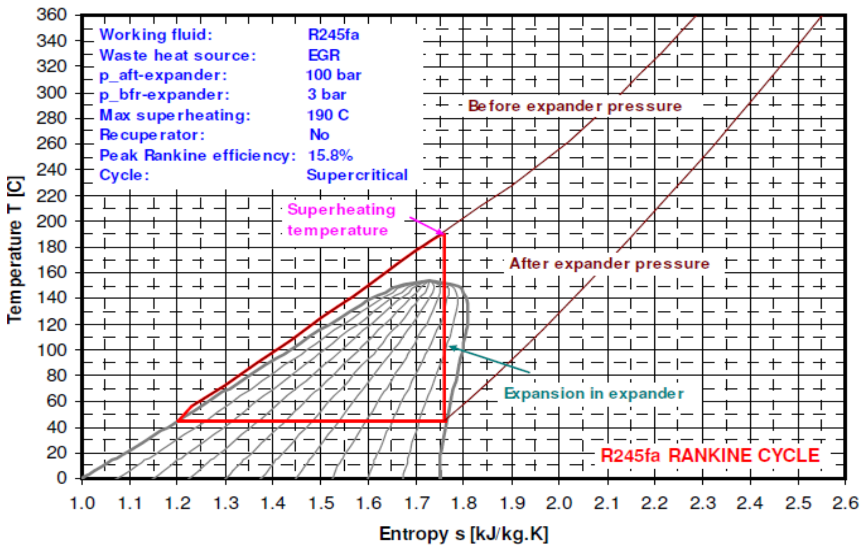


Figure 7. *T-s diagram of a supercritical rankine cycle for R245fa*

The same cycles are shown in figure 8-9 [6] but for a wet fluid, ethanol. In the subcritical stage for ethanol, the evaporation temperature, cycle efficiency and superheating temperature are targeted to be the same as in R245fa.

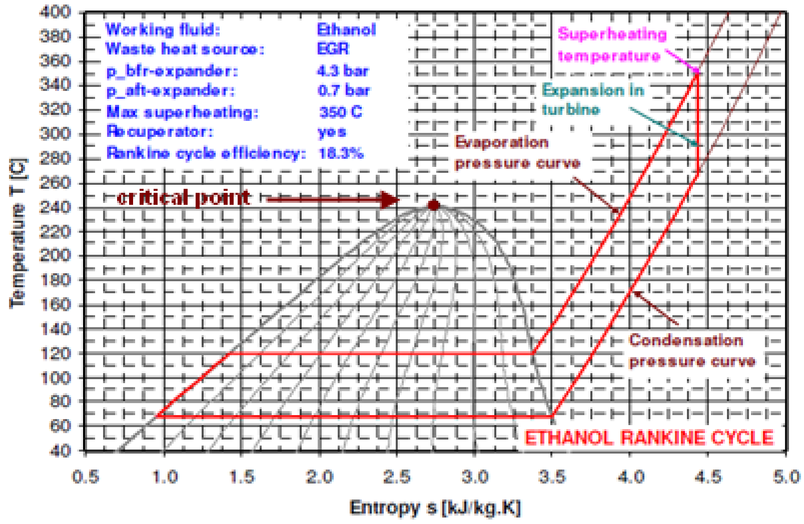


Figure 8. *T-s diagram of a subcritical rankine cycle for ethanol*

In the supercritical stage, the operation parameters were: pump pressure = 80 bar, condensation pressure = 1,08 bar, condensation temperature = 80°C and the superheating temperature = 306°C. Under these operation parameters, the cycle efficiency was 25,5 %. Although a recuperator is used in the subcritical cycle, the supercritical cycle have a higher efficiency. That is because the level of superheating in subcritical cycle and supercritical cycle is closer for a wet fluid than for a dry fluid. As described before, a dry fluid has a positive slope for the saturated vapor line, and therefore the dry fluid's level of superheating is limited by the end of expansion state. However, a simple turbine can still not handle the expansion ratio.

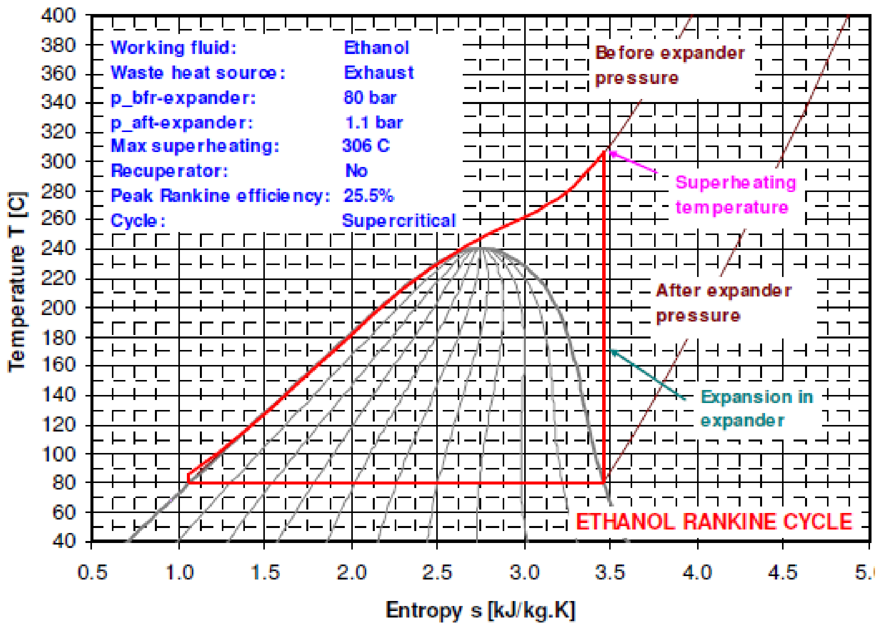


Figure 9. *T-s diagram of a supercritical rankine cycle for ethanol*

In this project the working fluid will be water, which has the highest thermal stability and wetness. The working parameter for the pump will be large, up to 180 bars. At those high pressures a 1-stage turbine will not be sufficient.

1.1 Problem definition

The purpose of this master thesis is to understand the impact following parameters have on the waste heat recovery system:

- Refrigerant: Pressure and flow
- EGR: Temperature and flow
- The degree of superheating

A model was built where the Rankine cycle can be simulated, the model was used for conducting a study of parameters effect. There is also a need for experimental data to validate the model, therefore an evaporator will be mounted in the EGR loop of the engine.

The question is how much can the fuel consumption be reduced?

1.2 Boundaries

In this master thesis some boundaries have been made:

- Power input to the system is only obtained from the EGR gases
- Only water is used as a refrigerant
- The piston expander is replaced by a turbine in GT-Power
- During the experimental tests only the evaporator is mounted in the EGR loop, while in the simulations the entire Rankine system is modeled.
- In GT-Power the investigated steam pressure are between 20-180 bar while in the experimental run the steam pressure is up to 100 bar. The evaporator is designed to tolerate up to 250 bar.
- The investigated EGR temperatures are between 290-700˚ C and the the investigated EGR flows vary from 37-130 *g/s.*
- During the experimental tests only the operating points in the ESC cycle were run.

1.4 Sources of error

The pressure drop is modeled just before the evaporator instead of over the evaporator in GT-Power. This leads to a more stable model due to that the pressure drop only occurs while the refrigerant is in the liquid state.
Assuming a constant refrigerant flow at two different pressure levels, then the pressure drop should be higher at the lower pressure level. This depends on the higher density of the steam at the lower pressure level. This part is not taken into consideration in the pressure drop just before the evaporator in the GT-Power model. The pressure drop is only dependent of the mass flow. This means that the power required by the pump should be a bit higher at low pressure levels when

comparing to the current results. Since the pump power is 10 times smaller than the expander power, the affect on the final result can be neglected.

The EGR is assumed to be consisting of only air in the GT-Power model, this should have no bigger effect since the component in GT-Power is calibrated with the data provided from the experimental tests.

In the engine test there was a filter in the water inlet pipe, prior to the evaporator. This filter was clogged by some black particles, even though the water was distillated and salt free. It is believed to be an organic substance. This substance in the filter led to an increase in the pressure drop with approximately 7-10 bar in some operation points of the ESC cycle.

The piston expander isentropic efficiency is assumed to be constant due to lack of data. Otherwise, the isentropic efficiency of the piston expander is dependent on the filling factor. A higher level of filling decreases the isentropic efficiency due to lower expansion rate in the piston expander.

The steam power E_{Steam} provided from GT-Power is 1-2 % too high. This depends on the assumption made that the heat transfer to the walls from the EGR equals the steam power, this means neglecting some heat losses. But since the losses are of such a small magnitude the effect on the final result should be minimal.

The EGR pressure drop during the experimental test could differ a bit since there is no pressure transducer after the evaporator, the pressure just prior of the turbine was used.

2. Method

The first task was to build a model in GT-Power that represent the Rankine cycle with water as the refrigerant. Results from experimental tests where the evaporator was mounted in the EGR loop of a Scania euro V engine was used to calibrate the evaporator in the GT-Power model.

2.1 GT-Power

In a prior project course [2] a model was developed with the simulation tool GT-Power. The model describes the heat transfer inside the evaporator between the refrigerant and the diesel exhaust gases for different inlet temperatures and flows, see figure 10.

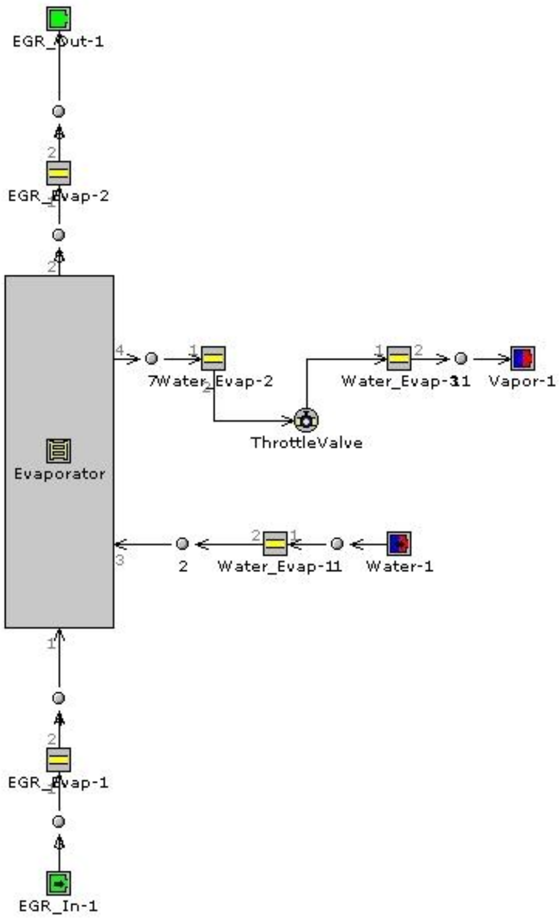


Figure 10. *Basic model of the evaporator developed in project course*

In this master thesis further components will be introduced in the model so the entire Rankine cycle could be described. A pump, receiver, expander and a condenser were integrated into the old model.

2.1.1 Evaporator

The evaporator is designed and manufactured by Ranotor [7]. It possesses a special cylindrical design and geometry which is made to tolerate up to 250 bar of steam pressure, see figure 11. The refrigerant flows in micro tubes with small diameter. The purpose of the small diameter is to always have laminar flow. The refrigerant should be equally distributed to 48 micro tubes inside the evaporator. The exhaust gases flows around these micro tubes and heats up the water. The 48 micro tubes are divided into two groups inside the evaporator, one group consisting of 21 micro tubes and the other group of 28 micro tubes. This means the water has two inlets to the evaporator and the steam exits through two outlets that comes together 20 *cm* after the evaporator. At each steam exit there is a thermocouple mounted into the flow, the data from these sensors will be analyzed in order to understand if the water is equally distributed between the two groups. There is also 16 thermo couples mounted inside the evaporator.

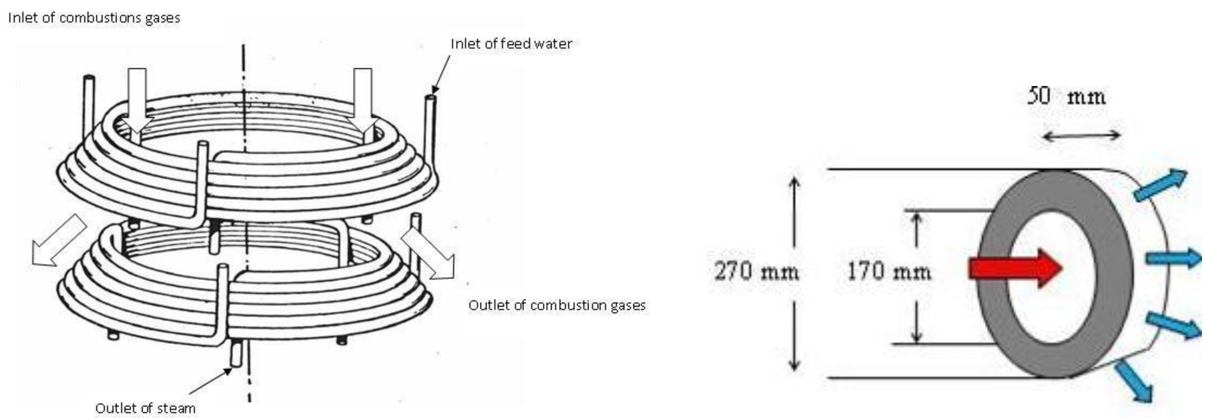


Figure 11. *The dimensions of Ranotors counter flow evaporator*

In order to calibrate the evaporator in GT-Power the physical evaporator was mounted in the EGR loop of a 360 hp 12.7 liter Scania engine (DC1306). Different parameters like the refrigerant flow, EGR-flow, EGR-temperature and the refrigerant pressure were varied in several tests. The test results were used to calibrate the evaporator in GT-Power, see appendix 8.4.

Other needed input data in GT-Power like heat transfer area A_{HT} and flow area A_F were calculated according to equations (1-2):

$$A_{HT} = 2 \cdot \pi \cdot r \cdot L_{tube} \cdot n_{tubes} \tag{1}$$

$$A_F = \pi r^2 \tag{2}$$

Where *r* is the radius of the micro tube, L_{tube} is the length of the micro tube and n_{tubes} is the number of the tubes.

The power output E_{Evap} is calculated with equation 3:

$$E_{Evap} = \dot{m}_{ref} \cdot \Delta h_{Evap} \tag{3}$$

Δh_{Evap} represent the change in specific enthalpy across the evaporator and $\dot{m}_{ref}$ is the mass flow of the refrigerant.

Finally the evaporator's temperature efficiencies η_{EGR} and η_{Steam} was calculated with the help of equation 4-5:

$$\eta_{EGR} = \frac{\mathrm{T1_{EGR}} - \mathrm{T2_{EGR}}}{\mathrm{T1_{EGR}} - \mathrm{T2_{Ref}}} \tag{4}$$

$$\eta_{Steam} = \frac{T3_{Ref} - T2_{Ref}}{T1_{EGR} - T2_{Ref}} \tag{5}$$

$T1_{EGR}$ is the inlet temperature of the EGR, $T2_{EGR}$ the outlet temperature of the EGR, $T3_{Ref}$ the steam temperature and finally $T2_{Ref}$ the inlet temperature of the refrigerant.

2.1.2 Expander

The high pressure levels required in a Rankine cycle with water as the refrigerant are a problem while using turbines, there is no single stage turbine that can handle these pressure levels. One possible solution is to use a piston expander and a study has been made at KTH where a piston expander has been developed for these kinds of applications [9]. The isentropic efficiency for the piston expander depends on the volume flow rate of the vapor. This means a lower volume flow rate result in a larger expansion rate and therefore higher isentropic efficiency for the expander. While a higher volume flow rate leads to a decreased expansion rate which affects the isentropic efficiency negatively, see figure 12 [10].

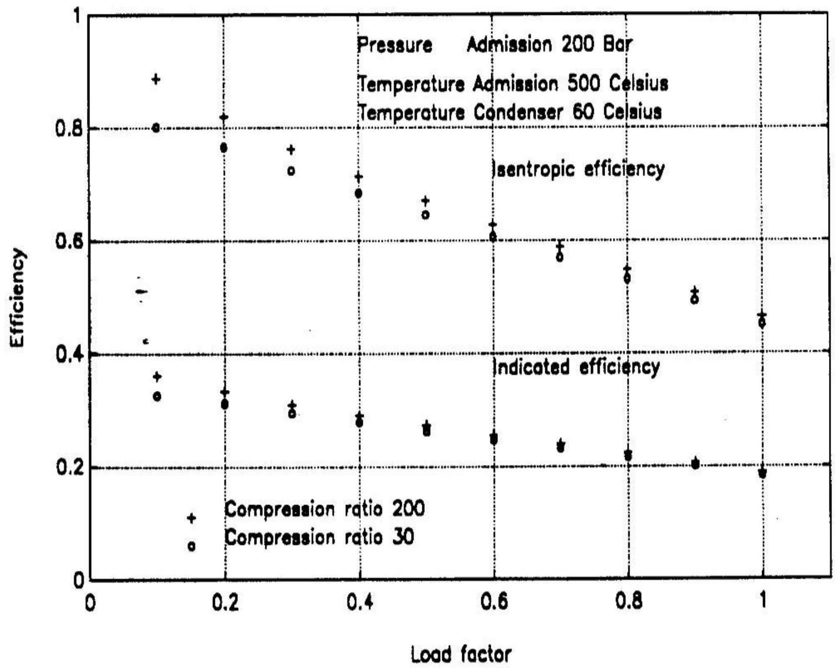


Figure 12. *The isentropic efficiency of the expander as function of the load factor, [5]*

Ideally it is best to run the expander with a low torque i.e. small filling and a high speed. This will lead to a high isentropic efficiency. But lack of time makes it hard to build a piston expander model in GT-

Power. Therefore a simple turbine with a constant geometry will be used. The turbine's speed will represent the filling factor i.e. when the filling factor is high the speed will increase and vice versa. Too calculate a relationship between the speed of the turbine and the isentropic efficiency is to complex, therefore it was decided to use a constant isentropic efficiency regardless of the load applied on the turbine in GT-Power. The isentropic efficiency η_T was chosen to 65 %, see appendix 8.2.

If the steam leaving the expander is in the two phase area i.e. not superheated (see figure 5 and 49), a turbine wouldn't have worked since the turbine blades get damaged when they come in contact with water. In case of the piston expanders this will not cause any problems. This also means a higher power output from the expander since more energy can be utilized from the steam.

The expander power E_{Exp} is calculated in GT-Power with equation 6:

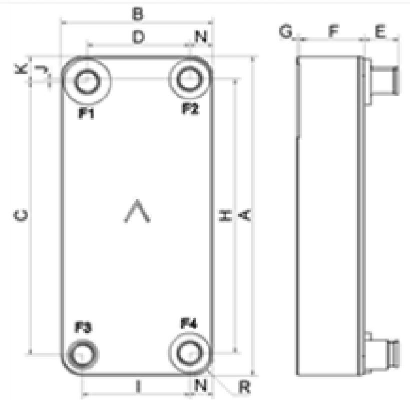
$$E_{Exp} = \Delta h \cdot \eta_T \cdot \dot{m}_{refr} \tag{6}$$

The change in specific enthalpy across the turbine is Δh, $\dot{m}_{refr}$ is the refrigerants mass flow and η_{isen} is the turbines isentropic efficiency.

2.1.3 Condenser

A water cooled condenser often has an elongated tank with tubes inside it. The refrigerant flows around the tubes while the cooling water flows inside the tubes. To acquire the condensers geometry and characteristic, see figure 13, a calculation tool is used. The calculation tool, SSP G7, is provided by SWEP, [11].

DIMENSIONAL DATA



	Unit	Value	Tolerance
A	mm	525	+/-2
B	mm	243	+/-1
C	mm	448	+/-1
D	mm	164	+/-1
E	mm	54,2 (opt. 27,1)	+/-1
F	mm	78,7	+2%/-1,5%
G	mm	4,00	+/-1
H	mm	450	+/-1
I	mm	171	+/-1
J	mm	4,50	
K	mm	42,0	
N	mm	37,5	
R	mm	35,0	

Figure 13. The condenser, B200T, dimensions

The specification was 50 kW of heat load, maximum power in the EGR gases, and a working pressure of 1,1 bar. The heat load $E_{Condenser}$ was calculated with equation 7:

$$E_{Condenser} = \dot{m}_{ref} \cdot \Delta h_{Condenser} \tag{7}$$

To calculate some important parameters, like the turbulent coefficient, *C,* and the turbulent exponent, *m,* equation 8 was used:

$$Nu = C \cdot Re^{m} \cdot Pr^{1/3} \tag{8}$$

To calculate the Nusselts number *Nu and* prandels number *Pr,* equations 9-10 was used:

$$Nu = \frac{h \cdot L}{k} \tag{9}$$

$$Pr = \frac{\mu \cdot C_p}{k} \tag{10}$$

Where K is the thermal conductivity, *h* is the film coefficient and C_p is the heat capacity.

The reference length L_{ref} is another parameter required by GT-Power when building up the condenser, equation 11 is used to calculate L_{ref}:

$$L_{ref} = \frac{\mu \cdot Re}{\rho_{cool\_water} \cdot U} \tag{11}$$

Where *U* is the velocity of the cooling water, *Re* is the Reynolds number given by the calculation tool (SPG 7), ρ_{cool_water} is the density and μ is the dynamic viscosity.

In equation 8 the turbulent exponent *m* is assumed to be 0,6. According to GT-power it should be less than 1. Therefore it is now possible to calculate the turbulent coefficient *C.* The flow area and the heat transfer area are calculated in the same way as for the evaporator, see equation 1-2. For full data see appendix 8.1.

The cooling water mass flow is kept constant at 1,697 *kg/s,* this value was provided by the calculation tool, [ref ssp] meaning the condenser maximum heat load is 50 *kW*. The water inlet temperature to the condenser is also kept constant at 91,5°C regardless of loading case. This means the cooling water outlet temperature will vary for different operating points since the cooling power needed differs for different points.

2.1.4 Pump & Receiver

The pump used in GT-Power is a simple positive displacement pump, it is speed controlled with a PID regulator, see appendix 8.2. Power required by the pump for different operating points is provided by GT-Power.

The volume of the receiver is 10 liter containing only water, see appendix 8.2.

2.2 Setting up the system

To determine the pump and the turbine displacement equation 12 was used

$$V_s[\frac{m^3}{s}] = \frac{\eta_v \cdot n \cdot \rho}{\dot{m}} \qquad (12)$$

Where η_v is the volumetric efficiency, n the expanders/pumps revolutions per second, $\dot{m}$ the mass flow in *kg/s* and ρ the inlet density. To determine the inlet density before the turbine and the pump a very short simulation is run. This short simulation makes it possible to see what the density is prior to the pump and the turbine. The mass flow is decided to 10 *g/s* based on experimental tests performed during the project course. The rotational speed is set so that the turbine has a speed much higher than the pump. That depends on the geometry between the turbine and the pump, theirs rotational speed cannot be the same.
In order to obtain a stable system three critical things must be accomplished:

- The initial composition in the different components must be right
- The pump must be regulated regarding speed
- The turbine must be regulated regarding speed

If the system initially contains only water then there is no room for expansion i.e. when the water boils into vapor. This will lead to uncontrollable increase in pressure in the system. Therefore initially the correct composition in the pipes is only water before the evaporator and mostly vapor after the evaporator. In the receiver there is only water while it is equal amount of water and vapor in the condenser and the evaporator.

To be able to control the steam pressure in the system a PID regulator is calibrated and integrated in the model. The input to the regulator is the total pressure measured just before the evaporator and the output is the turbine rotational speed, see figure 14.

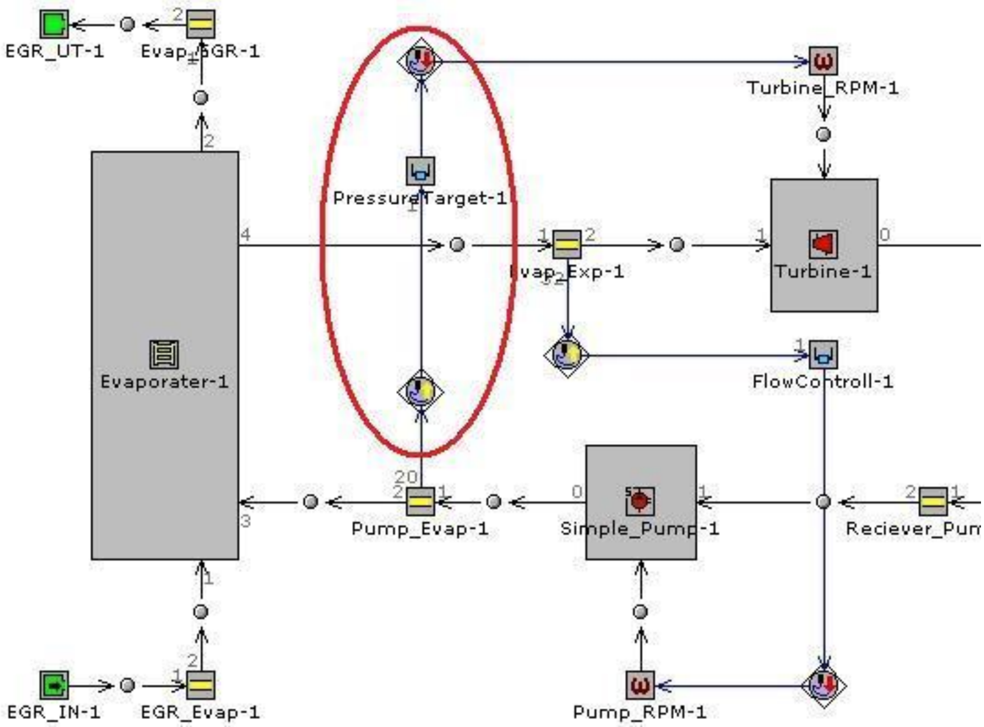


Figure 14. *The marked part represents the steam pressure PID regulator in the model*

If the steam pressure in the system is lower than the target then the turbines rotational speed will be decreased in order to raise the pressure and vice versa.
As mentioned earlier it is also desirable to control the refrigerant flow in the system. This is done by integrating another PID regulator where the input is the temperature after the evaporator. The output of this regulator is the pumps rotational speed, see figure 15.

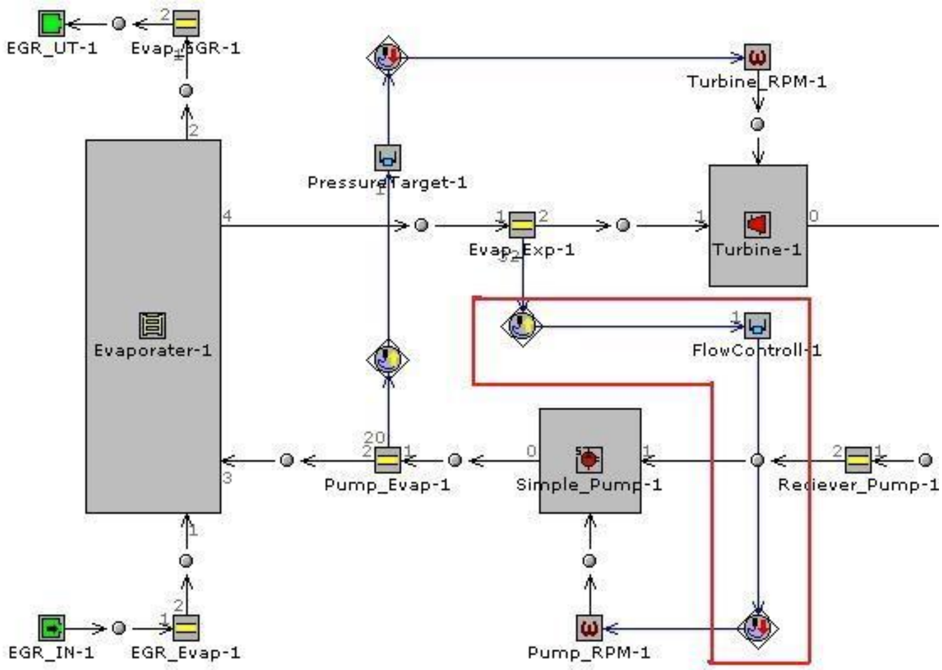


Figure 15. *The marked part represents the steam temperature PID regulator in the model*

That means the target of the PID regulator is set to a desired temperature of the vapor, if the vapor temperature after the evaporator should be too high the PID regulator will increase the pump rotational speed which will raise the refrigerant flow. That will decrease the steam temperature after the evaporator. This is done until the temperature after the evaporator equals the target of the PID regulator. If the temperature is low after the evaporator then the pumps flow will be decreased by the PID regulator.

In figure 16 the entire model can be seen, the evaporator is the big component to the left and the condenser is the big component to the right and the closed circuit contains the refrigerant, which is water.

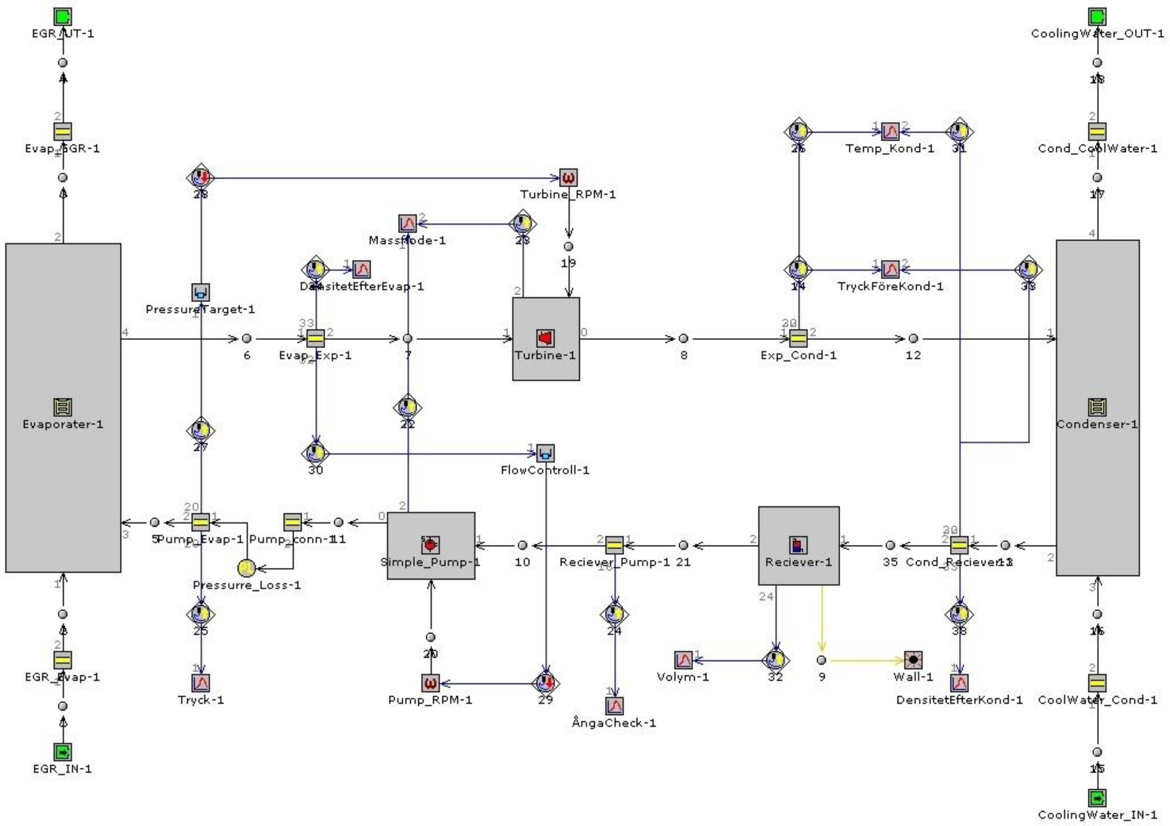


Figure 16. The improved model describing the Rankine cycle

2.3 The engine test

In the first section of this chapter the experimental set-up is described, while the operating points and the measurement is described in the second part.

2.3.1 Experimental set-up

The experimental set up, see figure 17, is similar to the one described in the prior project course. The refrigerant flows in an open system where the first end is composed by the receiver and the pump while the second end by the throttle valve. The other circuit contains the EGR provided by the engine, it flows inside the evaporator through the EGR cooler and back to the engine. The steam pressure was controlled by the throttle valve placed after the evaporator while the pump is controlled by a frequency converter. The steam was then let out into the atmosphere.

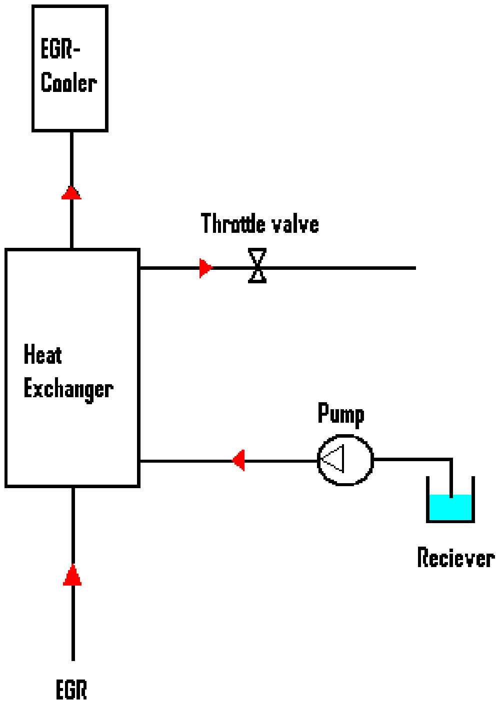


Figure 17. *The experimental layout*

In figure 18 the evaporator can be seen mounted, marked in red, in the EGR loop replacing the EGR cooler. Originally the idea was to replace the EGR cooler but the EGR did not cool down enough to be reused in the combustion chamber. This depends on the evaporator's heat transfer area being too small. Therefore the EGR cooler was reinstalled downstream the evaporator to provide the necessary cooling of the EGR, marked in blue.

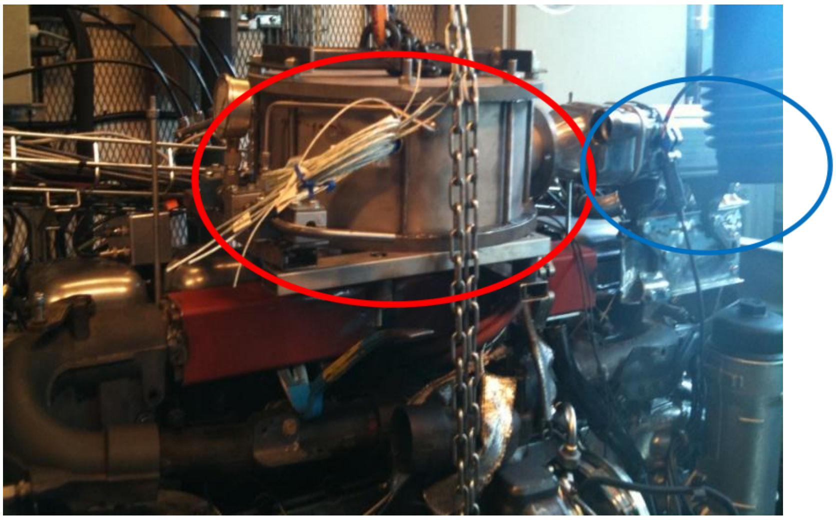


Figure 18. *The evaporator mounted on the Scania engine*

The pump and the tank can be seen in figure 19-20. This was an open system with the steam being let out in the atmosphere, the tank was manually filled with softened water.

Figure 19. *The refrigerant tank*

Figure 20. *The refrigerant pump*

Finally figure 21 illustrates the steam being led out through the pipe to the atmosphere.

Figure 21. *The superheated steam being let out in the atmosphere*

2.3.2 Operation points and testing

In the prior project the ESC-cycle were run, but the water mass flow meter was not functioning correctly which lead to incorrect data of the refrigerant flow. Also a thermocouple was mounted on the pipe, instead of in the pipe, which gave an error in vapor temperature after the evaporator. In this master thesis a new mass flow meter has been integrated into the system and two new thermocouples were mounted in the flow which leads to more accurate data of the vapor temperature. Therefore the ESC-Driving cycle will be repeated once again for more accurate data, see table 1.

Table 1. *The different operating points tested in the ESC-cycle*

Operating point	Speed [RPM]	Tourque [Nm]	EGR Flow [g/s]	EGR Temperature [˚C]	EGR Pressure ahead of the evaporator [bar]
2	1225	1840	77,9	514	1,516
3	1450	878	71,4	385	1,664
4	1450	1318	90,3	434	1,188
5	1225	956	51,7	428	1,515
6	1225	1434	66,3	478	1,036
7	1225	478	36,9	325	1,083
8	1450	1700	109,8	474	1,66
9	1450	439	60,1	295	1,274
10	1675	1517	122,4	463	1,613
11	1675	379	75,7	293	1,359
12	1675	1137	111,6	411	1,216
13	1675	758	89,3	360	1,737

The ESC cycle operating points could also be seen in figure 22. Divided in four groups, each group contains three operating points.

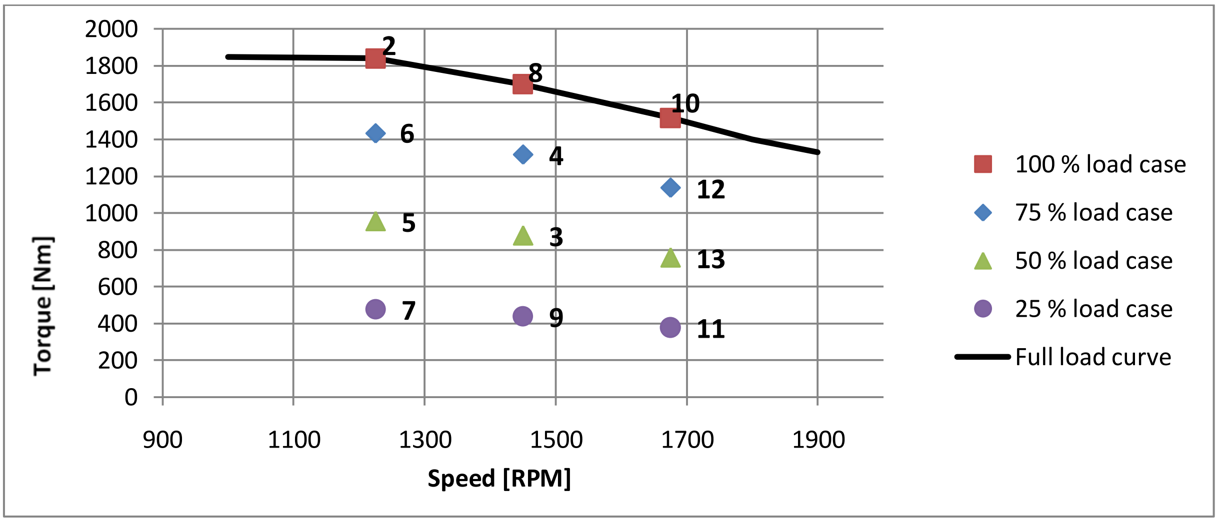


Figure 22. *Torque and speed of the different operating points of the ESC cycle*

There was a desire of increasing the EGR flow and EGR temperature separately while keeping the pressure and temperature of the refrigerant constant to understand the influence of the above mentioned parameters on the power output from the evaporator.
Several attempts were made to increase the EGR flow for some of the operating points in table 1, this was done by controlling the VGT (Variable Geometry Turbocharger) which leads to increased backpressure from the turbine. However if the EGR flow rate was increased more than 1-2 % from the original settings the smoke would increase drastically. More smoke means the evaporator would get clogged (soot). The impact from the enlarged amount of soot is a big decrease in heat transfer inside the evaporator. There was no possibility in this test setup of increasing the injection pressure to work against the increased soot formation.

To understand the influence of different parameters on the power output of the evaporator some tests were done, see table 2. The parameters included in the test were:

- EGR-temperature and flow
- Refrigerant flow
- Pressure in the closed loop
- Percentage superheated vapor

Table 2. *The different tests done to analyze different parameters affect*

Test number	Operating point (ESC)	Torque [Nm]	Speed [RPM]	Pressure [bar]	Flow Refrigerant [g/sec]
1	10	1517	1675	100	6-9,8
2	10	1517	1675	60	10,2-11,6
3	5	956	1225	60	1,3-3,9
4	5	956	1225	30	2,9-4,1
5	2	1852	1225	100-90-80	constant
6	4	1318	1450	100-90-80	constant

Test number 1-4 represent two different operating points, each points is tested for two different system pressures and every pressure is run for 4-5 different refrigerant flows, all flows resulting in superheated vapor. The purpose of this test is to understand if the refrigerant flow or the rate of superheated vapor is better in terms of the evaporators power output.

Test number 5-6 also represents two different loading points, this time each load point is tested for three different system pressures. However the flow is kept constant for all pressures, as table 2 shows. The intentions with these tests are to evaluate the effect of different pressures in the evaporator.

As mentioned earlier the EGR temperature cannot be varied and the EGR flow can be increased a very small amount, so in order to compare the impact of the EGR temperature and the EGR flow existing load points in the ESC cycle will be used.

2.4 Performance calculation

The reduction of fuel consumption for different operating points, *RFC [%],* is calculated according to equation 13:

$$RFC = \frac{E_{Exp} - E_{Pump}}{E_{Engine}} \tag{13}$$

Where E_{Exp} is the power output from the expander, E_{Pump} the power required by the pump and E_{Engine} the power the engine produces at the current loading point. While the WHR efficiency η_{WHR} is calculated with equation 14:

$$\eta_{WHR} = \frac{E_{Exp} - E_{Pump}}{E_{Steam}} \tag{14}$$

Where E_{Steam} is the power input to the expander provided by GT-Power as the heat transfer to walls i.e. the power required to heat up, evaporate and finally superheat the steam. The steam power E_{Steam} for the experimental results is calculated with equation 15:

$$E_{Steam} = \dot{m}_{Ref} \cdot (h_2 - h_1) \tag{15}$$

Where h_2 is the steam enthalpy provided by moliers diagram [4] since the temperature and pressure of the steam is known, while h_1 is the enthalpy of the refrigerant in liquid state and is calculated with equation 16:

$$h_1 = h'_{20^\circ} + v_{water} \cdot (P_{Steam} - P_{Water}) \tag{16}$$

Where P_{Steam} is the steam pressure, P_{Water} the pressure while the refrigerant is in liquid state and h'_{20° the enthalpy for saturated water at 20 ˚C while v_{water} is the specific volume of water in *m^3/kg.* The evaporator's efficiency is temperature based, the numerator in equation 4 represent the used evaporator i.e. how much the EGR was cooled for different load points. While the denominator in the same equation illustrates an evaporator with extremely large heat exchange surfaces i.e. the EGR would be cooled to the same degree as of the water inlet temperature $T1_{Ref}$.

$$\eta_{EGR} = \frac{T1_{EGR} - T2_{EGR}}{T1_{EGR} - T2_{Ref}} \tag{4}$$

The evaporator's efficiency could also be described as how good the evaporator is in terms of heating up the refrigerant, equation 5 was used:

$$\eta_{Steam} = \frac{T3_{Ref} - T2_{Ref}}{T1_{EGR} - T2_{Ref}} \tag{5}$$

The power input E_{EGR} to the system is defined as the maximal power the evaporator could pick up from the EGR, this means the EGR gases can maximally be cooled to the same degree as the refrigerants inlet temperature, equation 17 was used:

$$E_{EGR} = (T1_{EGR} - T2_{Ref}) \cdot \dot{m}_{EGR} \cdot C_{p_{EGR}} \tag{17}$$

2.4.1 Supplementary burning

One method of increasing the EGR inlet temperature is through supplementary burning, this means injecting fuel in the exhaust pipe. The EGR temperature will increase while the fuel burns, the more fuel injected the higher the temperature rise will be. One requirement to facilitate supplementary burning is that the oxygen content in the exhaust gases is high enough to oxidize the injected fuel. This varies for different load points.
In order to calculate the diesel mass flow required to heat up the EGR to a desired temperature, equations 18 was used:

$$\dot{\mathrm{m}}_{\mathrm{Diesel}} = \frac{\mathrm{C}_{\mathrm{p}_{\mathrm{EGR}}} \cdot x \cdot \dot{m}_{EGR}}{\mathrm{H}_{\mathrm{Diesel}}} \tag{18}$$

Where H_{Diesel} *J/kg* is the diesel specific energy, *x* is the degrees the EGR is to be raised. The efficiency of this operation is assumed to be 100 % i.e. no losses.

The required diesel flow is then used to calculate how much power, E_{Engine}, would be obtained if the injection was directly in the combustion chamber. This was made with the help of equation 19:

$$E_{Engine} = \dot{m}_{Diesel} \cdot H_{Diesel} \cdot \eta_{Engine} \tag{19}$$

The efficiency η_{Engine} of the diesel engine is assumed to be 40 %.

3. Result

In this chapter the results from the experimental tests and the GT-Power simulations will be presented. The results presented below is for operating point 2 in the ESC cycle unless otherwise noted, see table 1. For other operating points the results will be available in appendix 8.3. Also one requirement for all the tests/simulations are that the steam entering the expander is superheated.

3.1 Experimental results

The purpose of this experimental test was to understand the evaporator. As mentioned earlier some tests were done where the water flow and the refrigerant pressure was varied. As figure 23 illustrates a constant flow for different steam pressures result approximately in the same power output from the evaporator. The small difference in power depends on the mass flow measuring device, the flow fluctuate which leads to difficulties in acquiring exactly the same flow for different pressures.

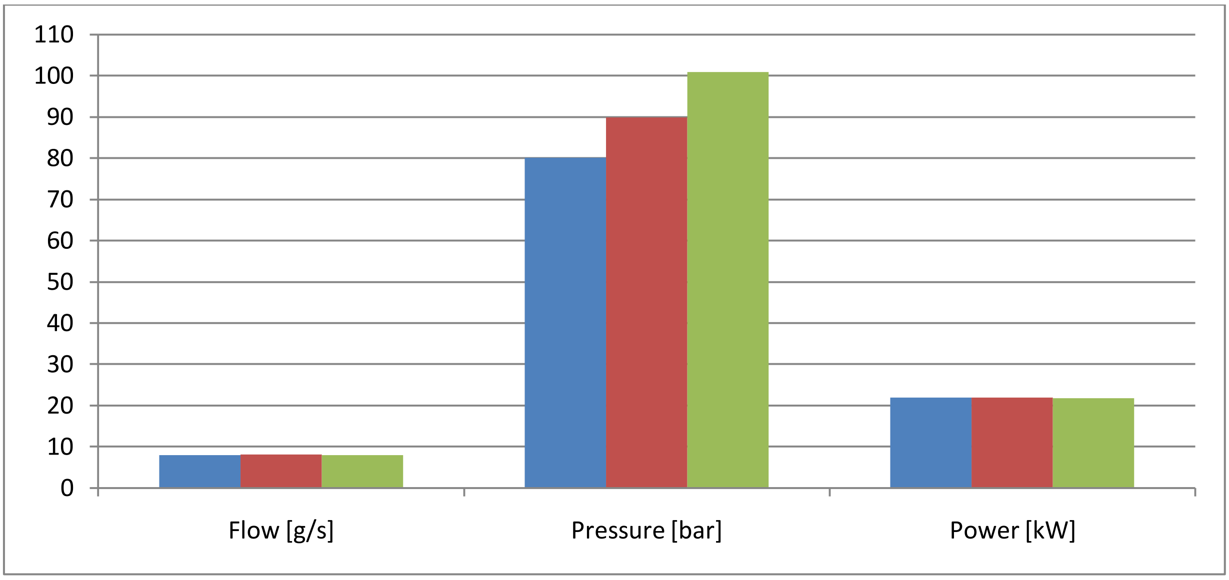


Figure 23. *Varying the refrigerant pressure while the refrigerant flow is constant*

This makes it hard to analyze the steam temperatures and the EGR outlet temperatures for different pressures. Therefore the same test was simulated with an exact refrigerant mass flow, the result was that lower refrigerant pressure leads to larger heat transfer from the exhaust gases. But the evaporator's performance improved at higher steam pressures because the steam temperatures increased. It is important to mention that the difference is small, a couple of degrees at the same flow, as can be seen in figure 24.

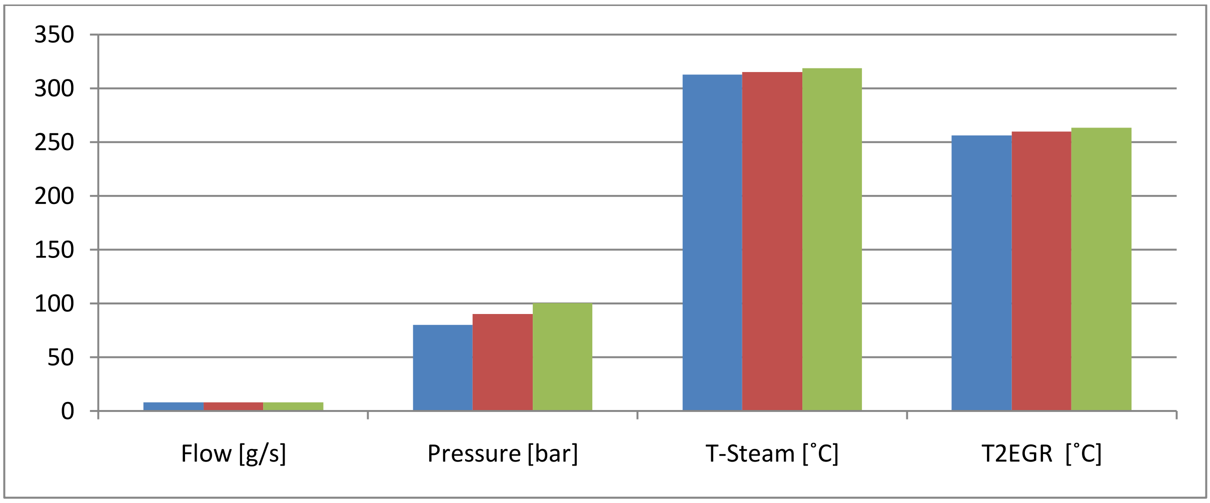


Figure 24. *Flow, pressure and steam temperature of the refrigerant and the EGR outlet temperature*

The next step was to keep the refrigerant pressure constant while varying the refrigerant flow, see figure 25. Five different refrigerant flows were tested for the same pressure. All flows resulting in different levels of superheated vapor. An increase of the refrigerant flow will decrease the steam temperature after the evaporator. As can be seen in figure 25 increasing the refrigerant flow will result in higher steam power, when looking at equation 3 this result indicates that the refrigerant flow has a bigger effect than the steam temperature on the steam power. This test was performed on operating point 10, see table 1. The power input to the evaporator is almost constant during these tests, 57,8-58,5 *kW*. The variation depends on different EGR temperatures from test to test, though the difference is 10 degrees as maximum.

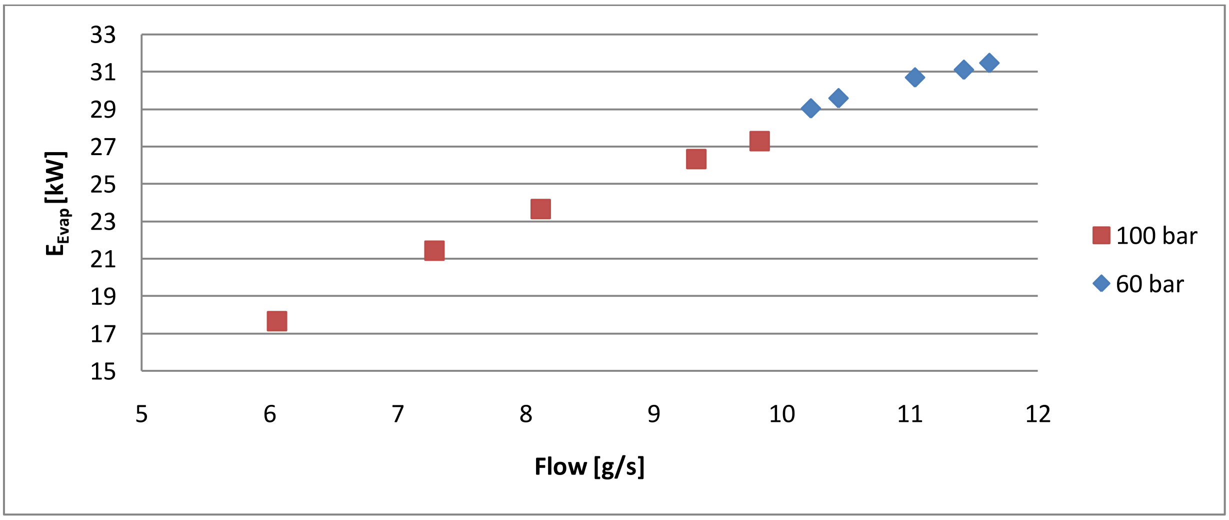


Figure 25. *Steam power output from the evaporator for different refrigerant mass flows for operating point 10*

Further analyze of figure 25 shows that the steam power also increase for lower refrigerant pressure. This depends on that the boiling temperature for water increase with increasing pressure, the boiling temperature at 60 bar is 277˚ C and at 100 bar 311˚ C, see figure 4. This means when running at 60 bar a larger mass flow could be applied and still acquire superheated vapor. A higher refrigerant mass flow also means the cooling of the diesel exhaust gases would increase, resulting in more energy utilized from the exhaust gases, see figure 26. Another interesting fact is the decreasing margin between the steam temperature and the EGR temperature for lower steam pressures and lower levels of superheating.

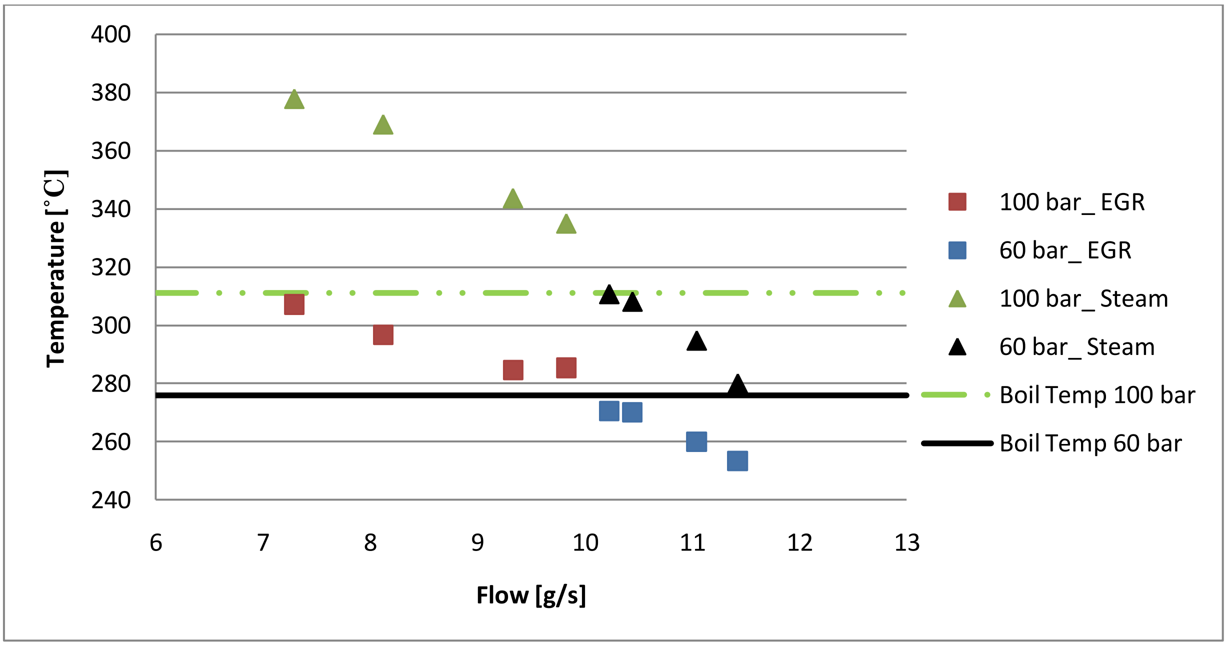


Figure 26. *The refrigerant flow versus the EGR-outlet/steam temperature after the evaporator, for operating point 10*

To maximize the power output from the evaporator the steam pressure must be as low as possible and the steam must be just superheated i.e. highest refrigerant mass flow possible according to the result above.

During the experimental tests the upper limit of the steam pressure was set to 100 bar. The evaporator wasn’t tested for higher steam pressures out of safety.
The steam temperature, the refrigerant mass flow and the EGR outlet temperature from the experimental test are presented in table 3 (see table 1 for full data of the operating points of the ESC cycle). For EGR inlet temperatures higher than 450 ˚C the water flow is approximately a tenth of the EGR flow, while for EGR temperatures lower than 300 ˚C the water flow is approximately 3 % of the EGR flow.

Table 3. *EGR temperatures and EGR flows for the ESC cycle*

Operating point	EGR flow [g/s]	Water flow [g/s]	EGR inlet temperature [˚C]	EGR outlet temperature [˚C]	Steam temperature [˚C]	Steam Pressure [bar]	Load case [%]
2	77,9	7,9	500	284	329	100	100
3	71,4	4	384	248	282	60	50
4	90,3	6,7	436	274	312	90	75
5	51,7	3,47	416	255	288	60	50
6	66,3	6,4	467	267	310	90	75
7	36,9	1,16	306	193	200	13	25
8	109,8	9,91	483	277	326	100	100
9	60,1	2,14	296	192	217	20	25
10	122,4	10,57	459	275	325	100	100
11	75,7	2,29	290	200	225	20	25
12	111,6	6,7	407	265	308	90	75
13	89,3	3,68	355	248	277	60	50

The 100 % load points were run at a steam pressure of 100 bar, the 75 % load points were run at a steam pressure of 90 bar , the 50 % load points at 60 bar while the 25 load points were run at highest possible steam pressure that would result in superheated steam. The steam pressure was chosen so the full load points would have the highest steam pressure. The steam pressure was then decreased for the lower loading cases as can be seen in table 3.

The EGR power E_{EGR} and the evaporator's EGR efficiency η_{EGR} for the ESC cycle operating points can be seen in figure 27, the dots representing the 100 % load points, stars present the 75 % load points, triangles illustrating the 50 % load points and finally the squares stands for the 25 % load cases.

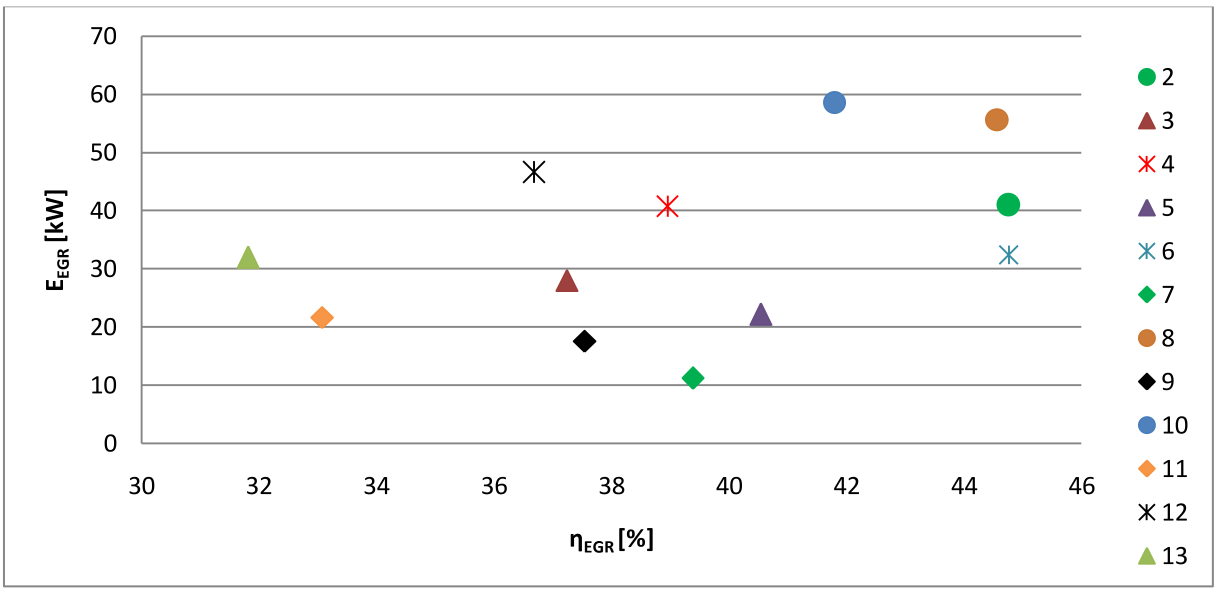


Figure 27. *The evaporator's efficiency based on the EGR cooling*

The efficiency varies between operating points in the same load case, this is due the different EGR inlet temperature. Since the strategy was to keep a constant steam pressure and relatively constant steam temperatures, this means only the refrigerant flow will separate the different operating point in the same load case, as can be seen in table 3. As mentioned earlier a higher EGR temperature results in a higher percentage refrigerant flow when comparing EGR flows. For example looking at the 75 % load case it shows that operating point 6 have efficiency of approximately 45 % while operating point 4 has 39 % and point 12 has a efficiency of 37 %. Comparing the EGR inlet temperatures in table 3 shows that they differ by 40-60 degrees in favor for operating point 6, which means that the steam pressure applied for operating points 4 and 12 is too high. This can also be seen if analyzing other operating points, for example operating point 7. The efficiency for this point is approximately 40 % which is larger than several other operating points, even though the EGR temperature of point 7 is lower. This is due to the steam pressure for point 7 is 13 bar. Compared with the relatively low EGR temperature the steam pressure is really low.

The evaporator's efficiency based on the steam temperature η_{Steam} is about 30 % higher than the EGR efficiency shown above. This efficiency describes how good the evaporator is at heating up the steam while the EGR efficiency describes how good the evaporator is at cooling down the EGR. One interesting fact is that the points with high EGR efficiency are the points with low steam efficiency. For example operating point 13 has the lowest EGR efficiency, about 32 %, while the steam efficiency is about 77 %, which is the highest percentage of the entire ESC cycle, see figure 28.

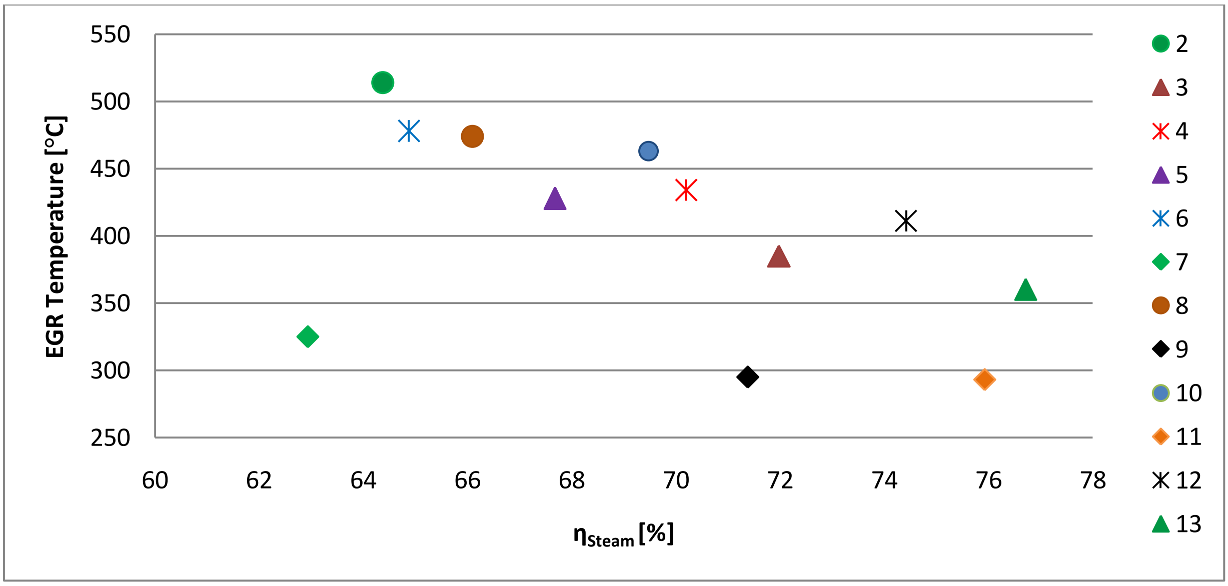


Figure 28. *The evaporator's efficiency based on the steam temperature*

3.1.1 Transient response

A transient test run was performed to understand the evaporator's response. The engine speed and torque was varied according to figure 29, four operating points were run. All the load levels and the speed levels in the ESC driving cycle were covered as the torque curve and the speed curve indicates. P1 and P2 is the refrigerant pressure before and after the evaporator, T2-1 and T2-2 is the steam temperature measured after the evaporator. T-EGR is the EGR inlet temperature while T2-EGR is the EGR outlet temperature.

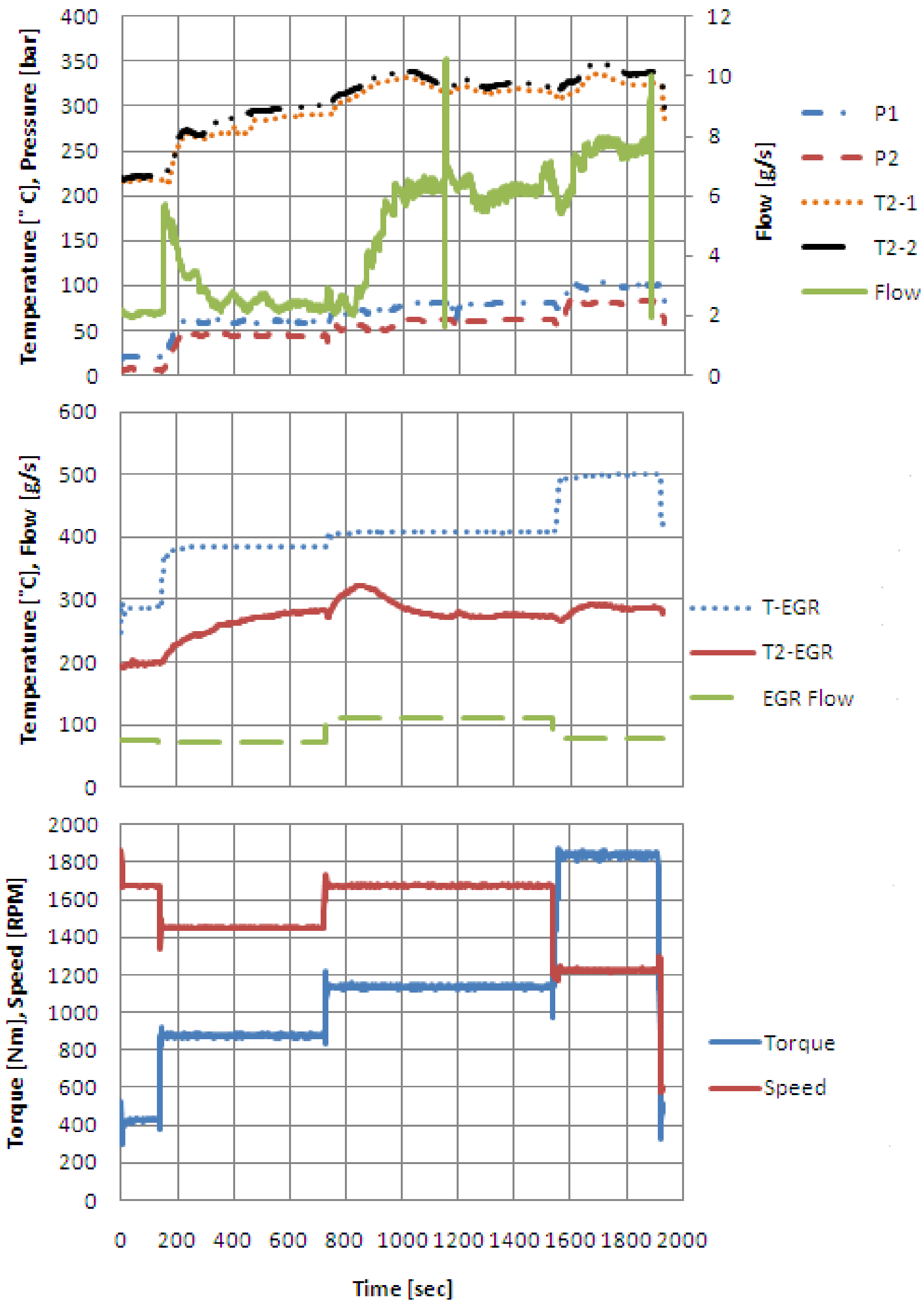


Figure 29. *The engine, refrigerant and the EGR temperature and flow*

The change in EGR inlet and outlet temperature and the EGR flow can be seen in figure 29, T-EGR is the inlet temperature of the EGR and T2-EGR is the outlet temperature of the EGR. Figure 29 also present the water pressure P1 prior to the evaporator, the steam pressure P2 after the evaporator, the steam temperature T2-1 -T2-2 after the evaporator and the refrigerant flow. The steam pressure was chosen as the highest possible pressure resulting in superheated steam. Comparing the x-axis it is possible to see how "slow/fast" the evaporator is, a change in torque and speed is performed according to figure 29 one can see how long time after the change in load point is done the steam takes to settle in. The first change is done after approximately 120 seconds, the steam temperature and pressure settles in at approximately 200 seconds. This means the process takes about 80 seconds to settle in.

The two thermocouples mentioned in the experimental setup can be seen in figure 30, the margin between the two is a couple of degrees prior to evaporation (t=0 to t=x_1) and during evaporation (t=x_1 to t=x_2). Then the difference between them grows to about 10 degrees when all the water is evaporated and the steam starts to superheat. This means one of the two micro tubes group gets all the water evaporated before the other group, the steam has lower density meaning it takes more place. This result in more water shuffled over to the second group, therefore the margin increase between the two groups.

Although it is important to mention that the difference in degrees lays as maximum 10 degrees, meaning the refrigerant is still pretty equally distributed.

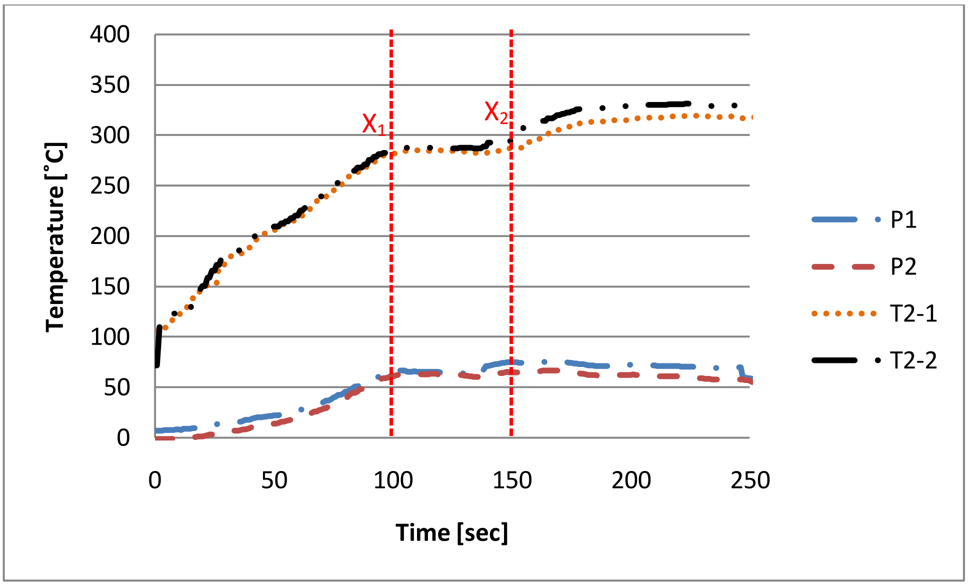


Figure 30. *The two temperature sensors and the pressure before and after the evaporator*

The same thing occurs while analyzing the thermocouples mounted on the micro tubes inside the evaporator, see figure 31. The difference is a couple of degrees before evaporation starts, but the margin increases due to evaporation. There are 16 thermocouples but only five of them are presented. This way it is easier to understand. Thermocouple T_G always has the highest steam temperature while T_C always gives the lowest steam temperature.

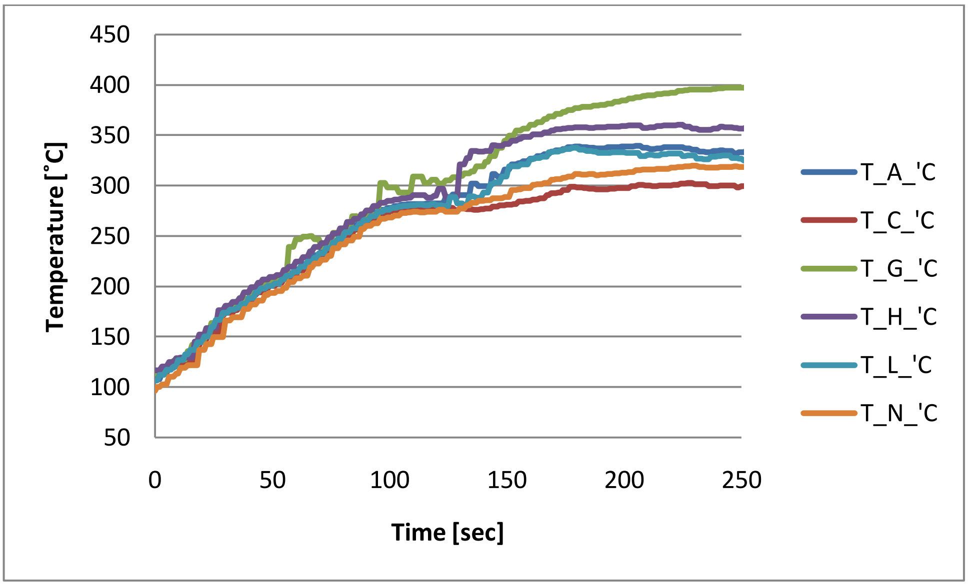


Figure 31. *The thermocouples inside the evaporator*

During these test the engine load is constant resulting in a constant EGR flow and temperature. The operating point is 2, see table 1.

3.2 GT-Power Results

As mentioned earlier a model was developed in GT-Power to simulate the Rankine cycle with water as the refrigerant. As figure 32 illustrates there is a cooling circuit, an EGR circuit and finally the refrigerant circuit. In the refrigerant circuit there are four different states, each state is defined by the pressure *P*, mass flow $\dot{m}$ and the temperature *T*. The refrigerant mass flow $\dot{m}_{Ref}$ is constant i.e. the same mass flow at all states, while $T1_{Ref}$ is equal to $T2_{Ref}$ because water is incompressible.

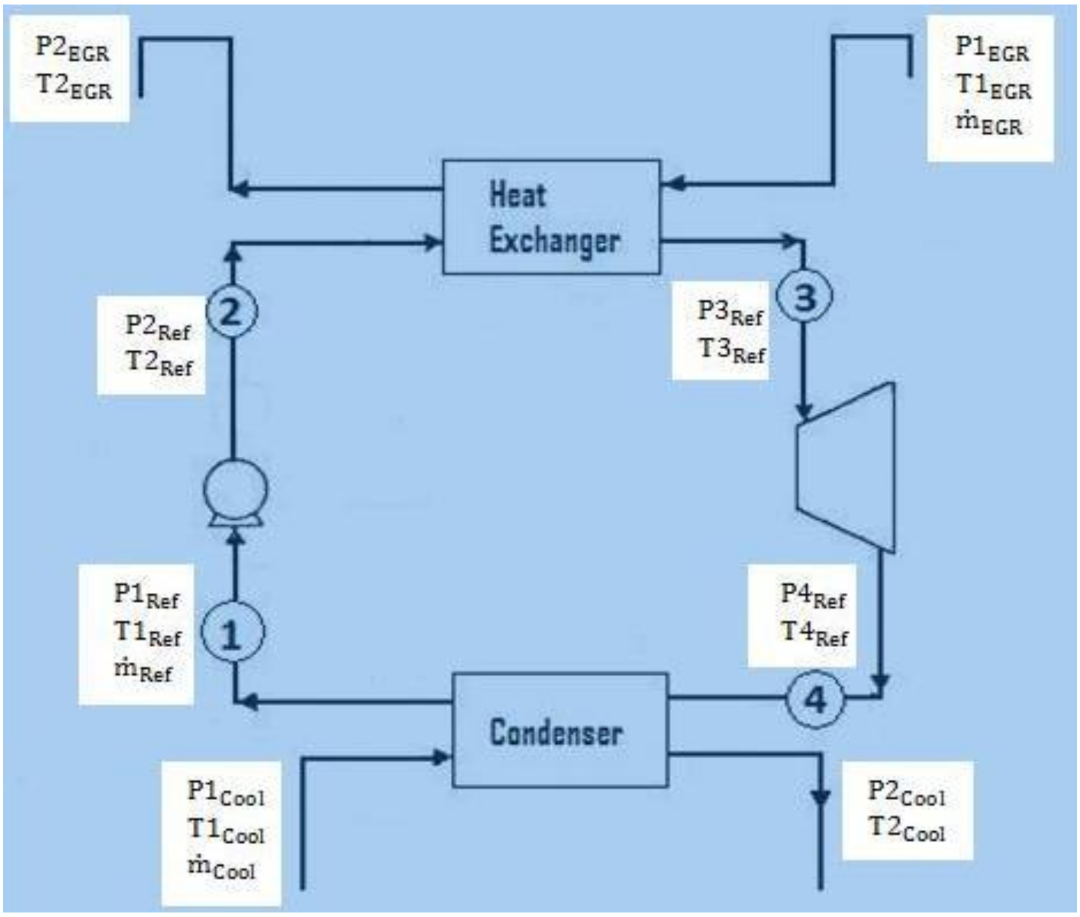


Figure 32. *The rankine layout*

The cooling water mass flow to the condenser is kept constant, 1,697 *kg/s* regardless of load case, while the inlet temperature $T1_{cool}$ is 91,5° C. Regarding the EGR circuit it will be varied in order to understand the effect of the EGR mass flow and the EGR temperature. Whereas the cooling pressure $P1_{Cool}$ is assumed to be 1,5 bar and the EGR pressure $P1_{EGR}$ is acquired from the engine test, see table 1.

3.2.1 EGR temperature variation

In this simulation the steam pressure $P3_{Ref}$ and the steam temperature $T3_{Ref}$ is kept constant, se table 4.

Table 4. *Values of the steam pressure and steam temperature*

Parameter	Simulation 1	Simulation 2
$P3_{Ref}$ [bar]	150	90
$T3_{Ref}$ [˚C]	359	320,5
$T1_{EGR}$ [˚C]	514-550-600-650-700	514-550-600-650-700

While the EGR pressure $P1_{EGR}$ and mass flow $\dot{m}_{EGR}$ is also kept constant. The EGR inlet temperature $T1_{EGR}$ is varied according to table 4 in this test.

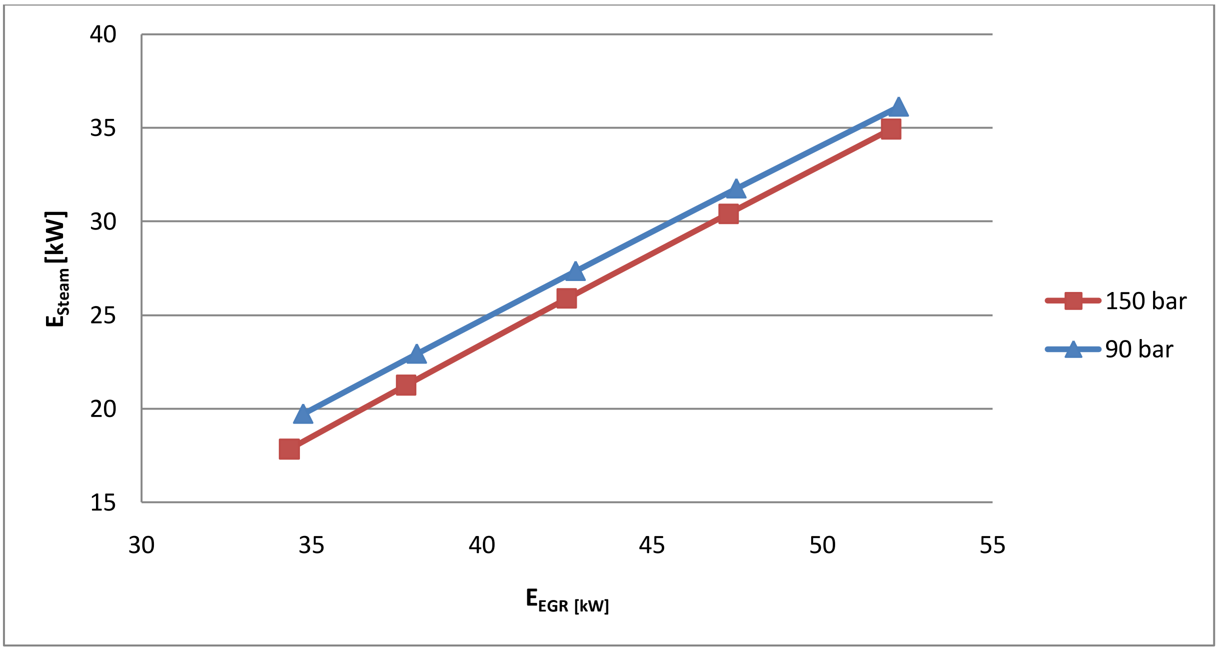


Figure 33. Steam power for different EGR inlet temperatures

As the EGR inlet temperature rises the EGR power input increases, which leads to increased steam power. This gives more power in the steam entering the piston expander. Analyzing figure 33 also shows that the steam power is larger for 90 bar, confirming the engine test result. Because the boiling temperature is lower at 90 bar compared to 150 bar the refrigerant mass flow can be increased resulting in a higher steam power output after the evaporator.

Studying the expanders output power shows for every 5 *kW* gain in EGR power the expander power output increase with approximately 0,8-1 *kW*. As can be seen in figure 34, the 150 bar line overtake the 90 bar line for higher EGR inlet temperature despite the steam power input to the expander is higher for the 90 bar line.

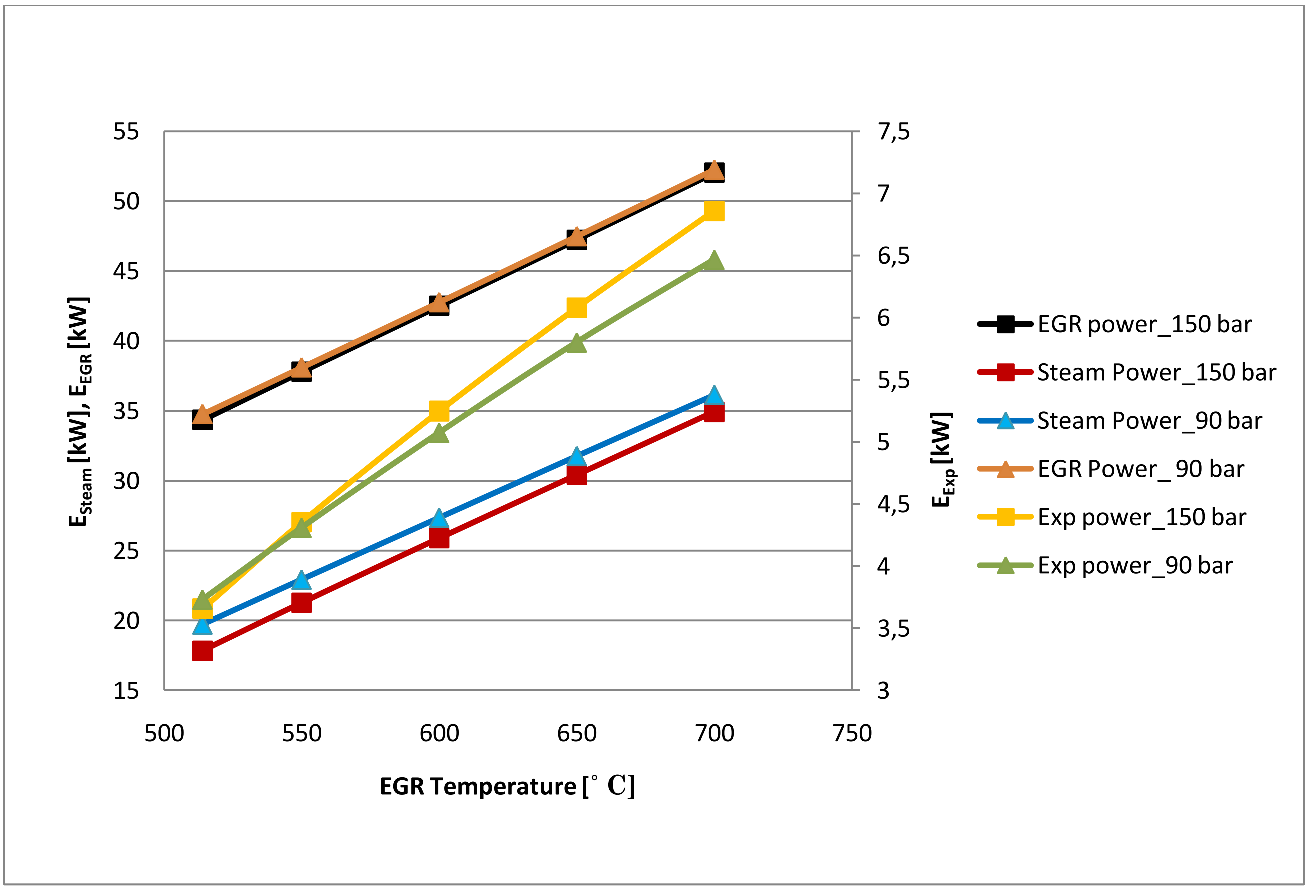


Figure 34. *EGR power, steam power and expander power for different EGR temperatures and refrigerant pressure*

This depends on the thermal efficiency of the Rankine cycle, it increases with the steam pressure according to figure 35. Another explanation is the relationship between the EGR temperature and the steam pressure as will be shown later in the report. To maximize the power output from the piston expander the optimal steam pressure needs to be determined. If a low steam pressure is applied then a large amount of steam power is obtained after the evaporator, but also low thermal system efficiency. If a very high pressure is applied then the amount of steam power provided by the evaporator decreases, but a high system thermal efficiency is acquired. So if the energy input is increased then the optimal pressure should also increase. This can be seen in the growing difference in terms of the expander's power output, see figure 34.

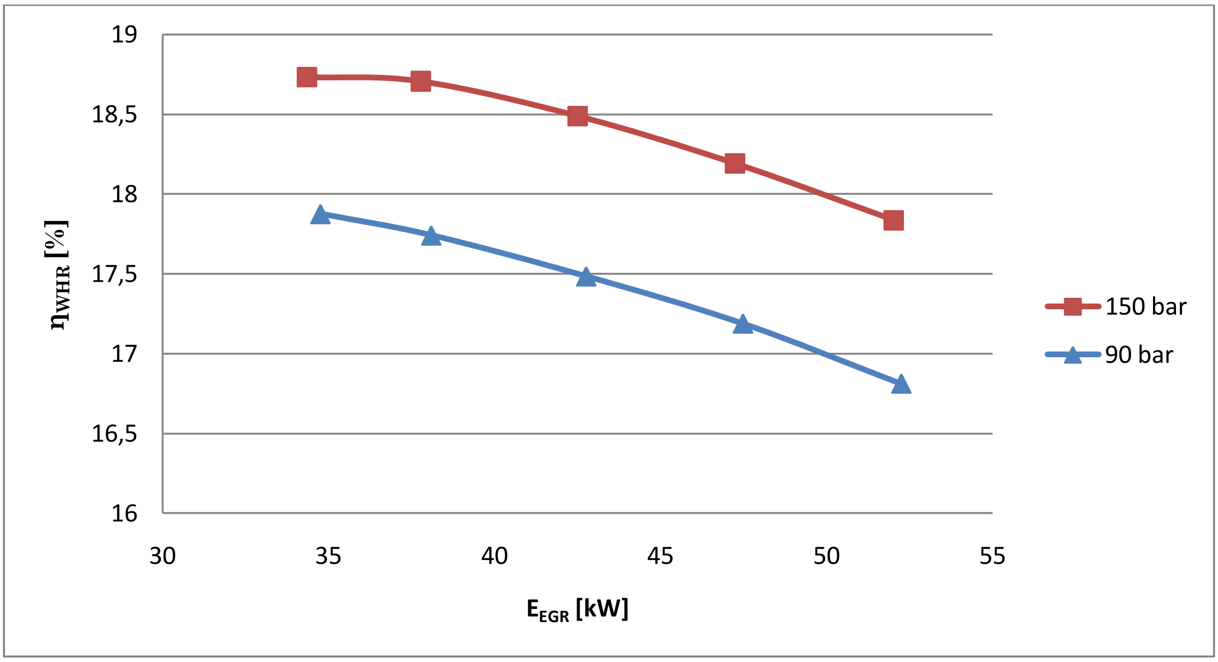


Figure 35. *The Rankine thermal efficiency η_{WHR} for different steam pressures*

Also it is interesting to see the thermal efficiency decreasing when the energy input is growing. This depends on the fact that the model was calibrated for certain energy input, when the energy input grow then the expander doesn't utilize enough energy out of the superheated steam. This can be observed when analyzing the steam pressure $P4_{Ref}$ after the expander, see figure 36.

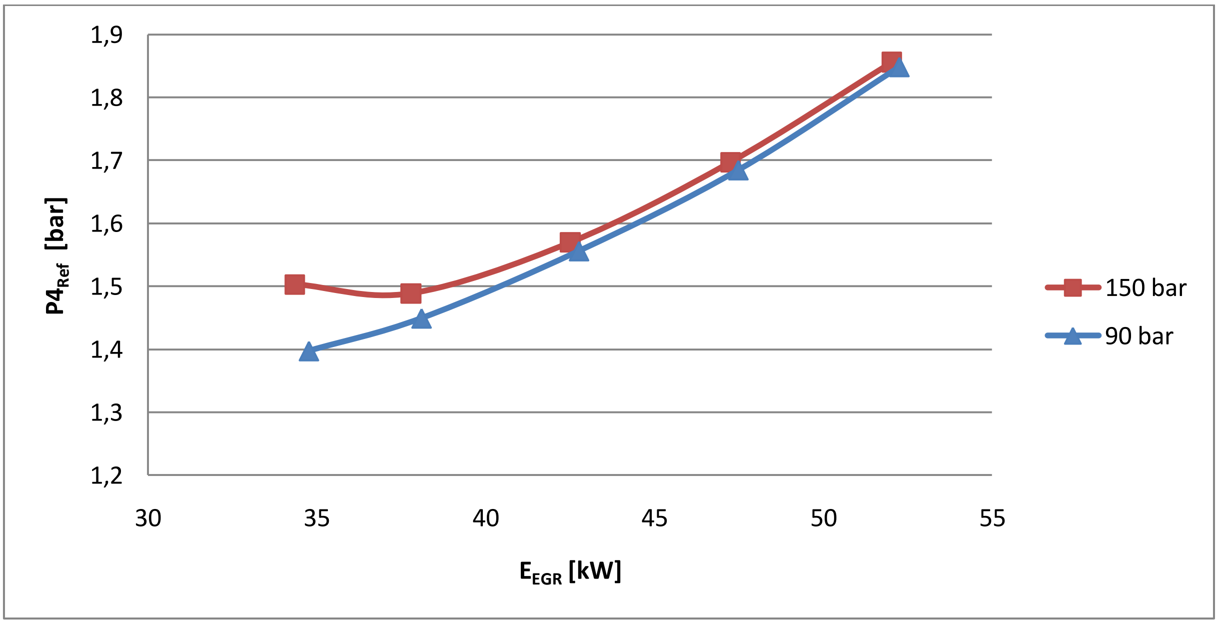


Figure 36. *The steam pressure $P4_{Ref}$ prior to the condenser for different levels of EGR power*

To explain the lift in the expander's output power the evaporator's efficiency was analyzed, see figure 37. This shows that increasing the EGR temperature results in higher evaporator efficiency. Also the 90 bar line have higher efficiency because the boiling temperature is lower and therefore a higher refrigerant mass flow can be applied leading to a higher steam power.

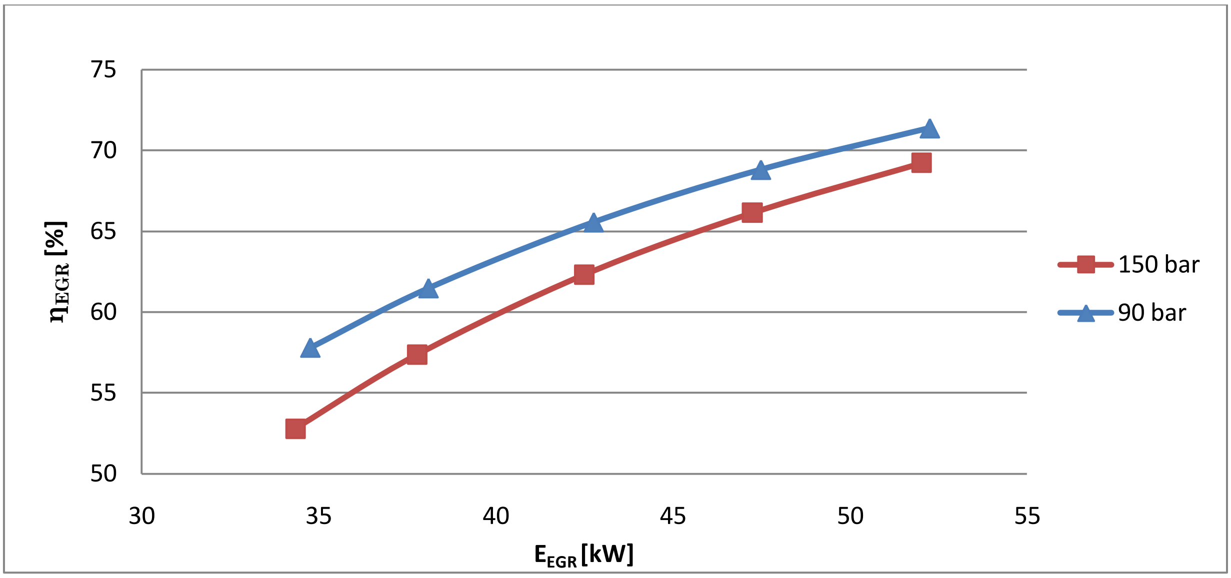


Figure 37. *The efficiency of the evaporator for varying energy inputs*

Another interesting observation is that the EGR gases cools down more if the inlet temperature is increased see figure 38. This depends on the growing refrigerant mass flow, as mentioned earlier the steam temperature $T3_{Ref}$ was kept constant so when the energy input is increased the mass flow also grow. A higher refrigerant mass flow leads to a larger cool down of the EGR gases.

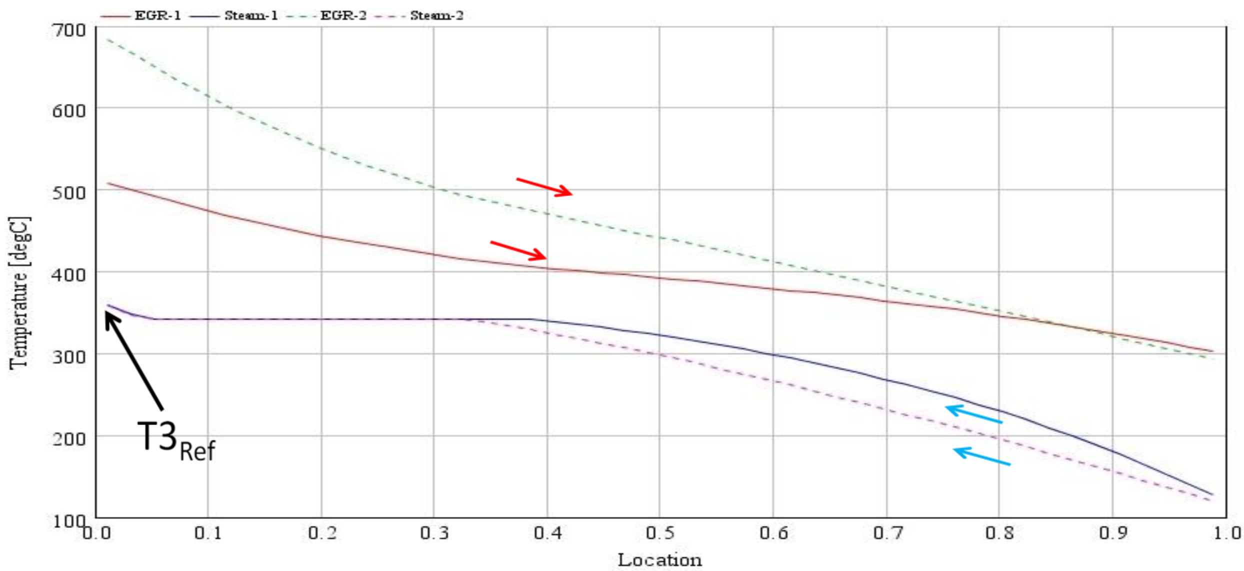


Figure 38. *The outlet temperature of the exhaust gases at 150 bar pressure*

Higher EGR temperatures can best be obtained through supplementary burning, which means burning diesel in the exhaust system prior to the evaporator. A calculation has been made to investigate if it is profitable to inject diesel in the exhaust system in order to increase the EGR temperature, or if it is better to inject in the combustion chamber, see figure 39.

The figure illustrates the expander's output power increase if the EGR gases temperature would be increased with 50, 100 and 150 degrees. A calculation is made using equation 18-19 with the same diesel flow injected directly in the combustion chamber. The increase in engine output power is compared to the expander increase in output power. The result speaks clearly in favor of injecting diesel into the combustion chamber instead of supplementary burning.

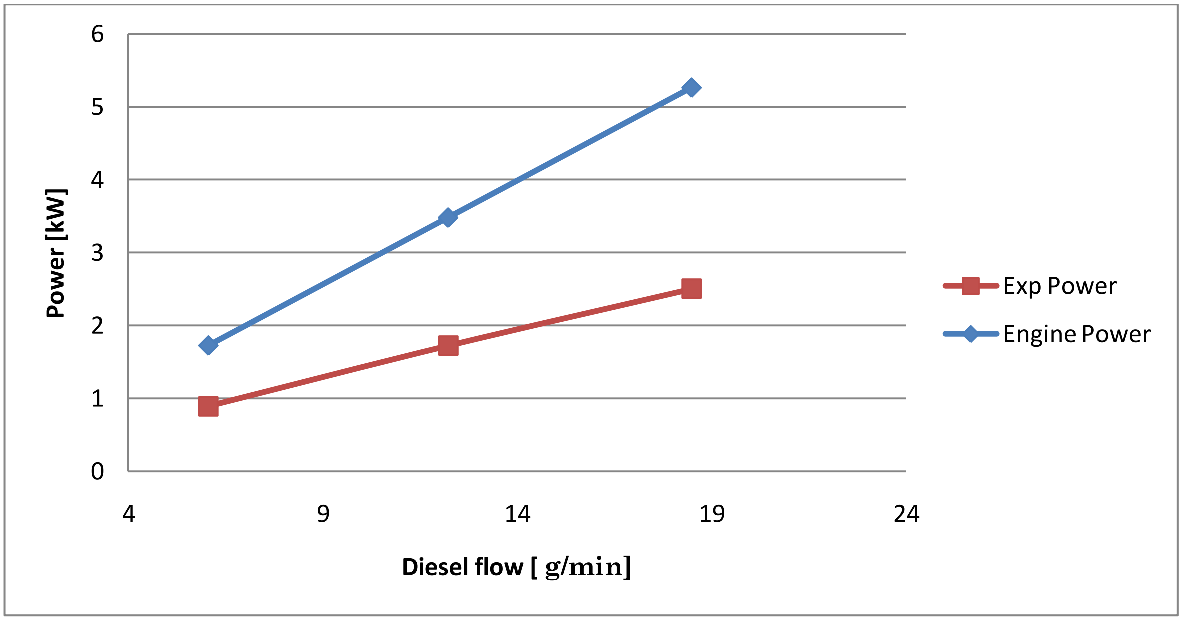


Figure 39. *Engine power and expander power for the same amount of injected diesel*

3.2.2 EGR Flow variation

The EGR flow $\dot{m}_{EGR}$ was varied and the power output from the expander analyzed, the EGR flow was varied according to table 5.

Table 5. *The EGR flow variation*

Simulation	Steam pressure [bar]	EGR Flow [g/s]
1	150	77,9-90-100-110-120-130
2	90	77,9-90-100-110-120-130

In figure 40 the steam power can be seen as a function of the EGR power. The same logic applies here as for the EGR temperature variation.

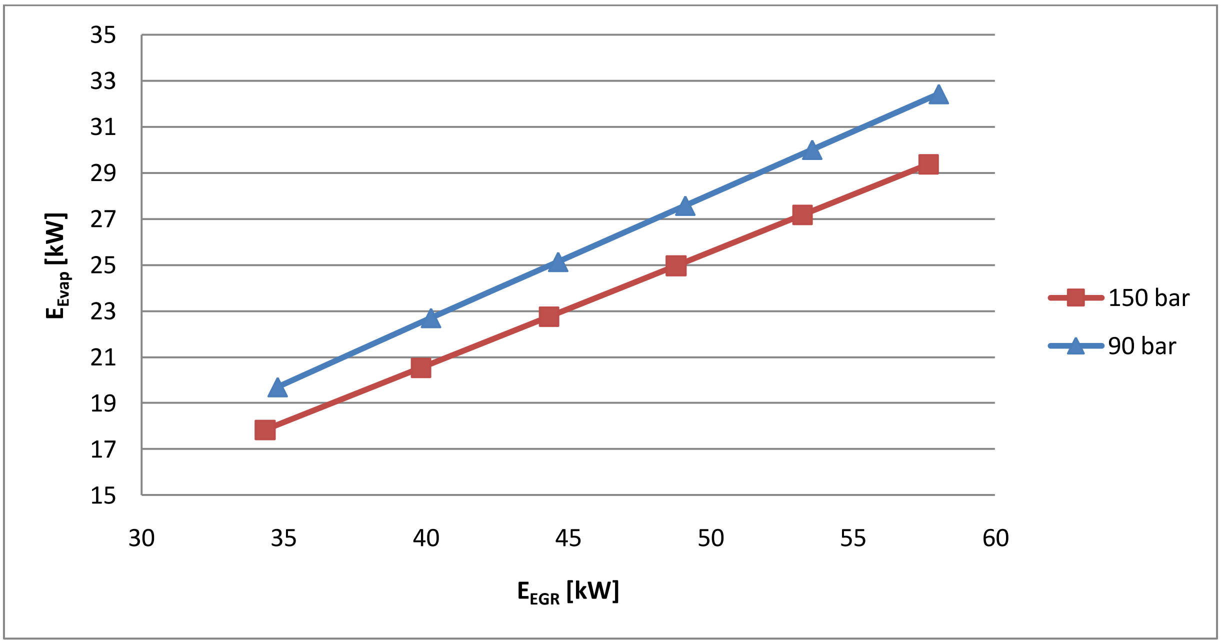


Figure 40. *Steam power for several EGR flows and refrigerant pressure*

As figure 41 illustrates, if the EGR power is increased with approximately 5 kW then the expander's power output is increased with approximately 0,5 kW. Similar to the EGR temperature increase figure it is possible to see that the 90 bar line result in higher expander power for lower energy input. But when the energy input grow it is possible to see the 150 bar equal the 90 bar line, as mentioned earlier if the energy input to the system increases then the optimal steam pressure also increases.

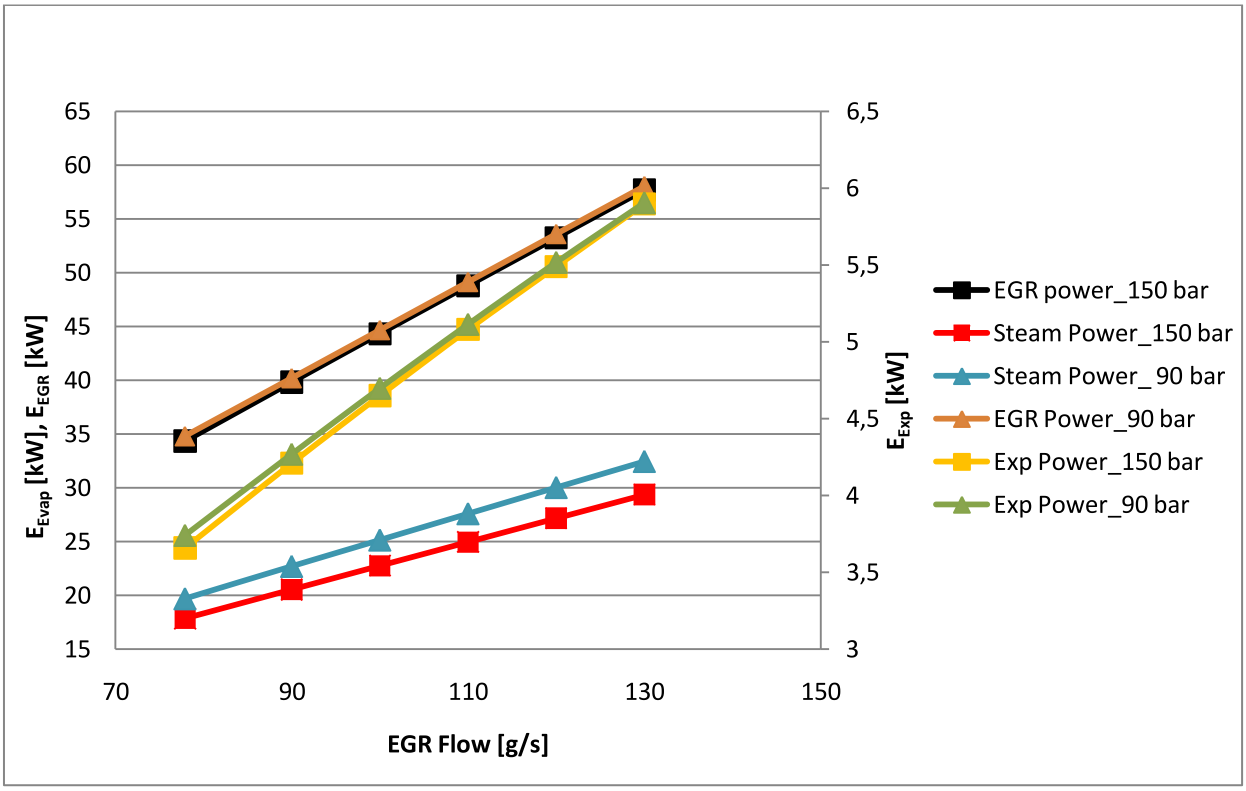


Figure 41. *Expander power, EGR power and steam power for different EGR flows*

Analyzing the effect of the increased EGR flow on the evaporator's efficiency is interesting because it is desirable to understand if this evaporator is over/under dimensioned. As table 5 illustrates the flow has been raised in several steps. The total increase in EGR flow is 52 *g/s*, which is an increase with 67 % of the default EGR flow. Analyzing figure 42 shows that the evaporators efficiency decrease with approximately 1 % when comparing the original EGR flow with the final value of the EGR flow for the same steam pressure. If the evaporator would be placed on an engine with larger EGR flow rate or in the exhaust pipe (not only the EGR), this would mean that the efficiency is practically the same i.e. this evaporator is over dimensioned for the used DC1306 engine. Also it is possible to see the difference in efficiency between the two pressure levels, approximately 5 % in favor of the 90 bar steam pressure level. The difference between 90 bar and 150 bar is the boiling temperature, it is lower for 90 bar, see figure 4. A lower boiling temperature results in a higher refrigerant mass flow which leads to a larger cooling of the EGR gases, that means the numerator in equation 4 grows. This leads to larger evaporator efficiency.

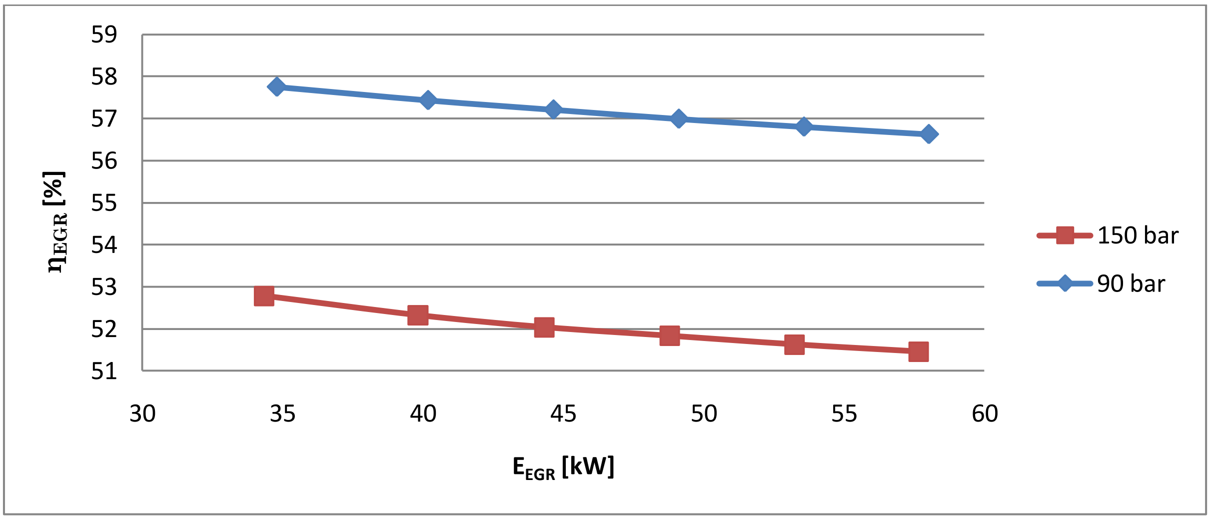


Figure 42. *The evaporator's efficiency for different energy input*

3.2.3 Fuel consumption reduction

To understand the impact of such a system analysis has to be made about fuel savings, figure 43 represents the reduction of fuel combustion for several power inputs to the waste heat recovery system. Increasing the power input is done with two different methods, either through supplementary firing which leads to higher EGR temperatures or by increasing the EGR rate resulting in a higher EGR mass flow. Starting from the default settings for operating point 2 and then keeping the mass flow constant while increasing the temperature a couple of steps and vice versa. The results clearly shows that the EGR temperature have a bigger impact than the EGR flow. Also it is possible to decrease the fuel consumption with up to 2,6 % if the EGR temperature is raised approximately 200 degrees, this percentage could increase if the exhaust gases was used instead of only the EGR gases.

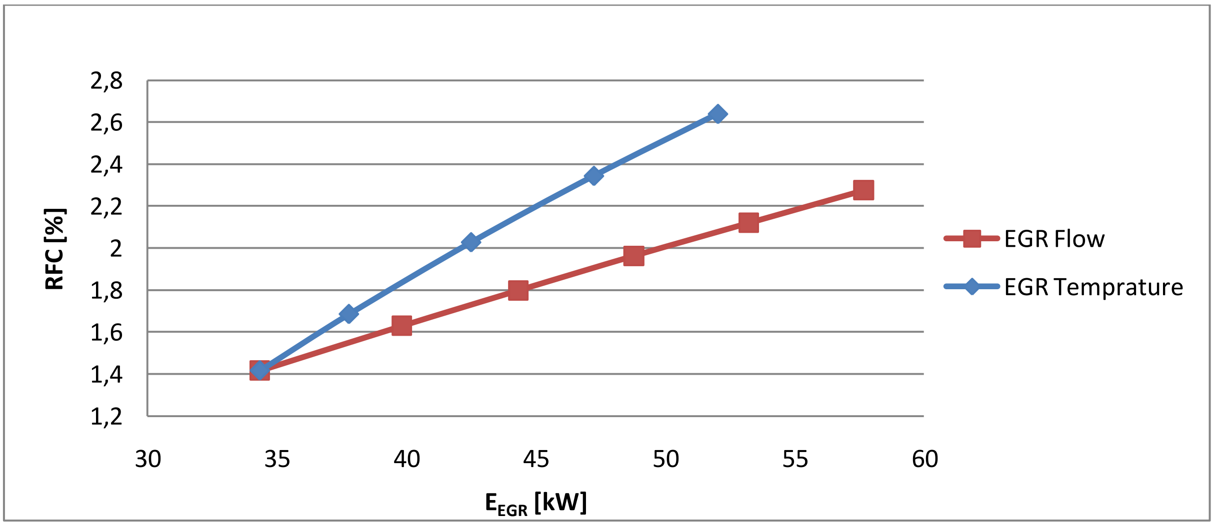


Figure 43. *Reduction of fuel consumption for different energy inputs*, 150 bar

3.2.4 Optimizing the system

In order to maximize the power output of the expander two optimizations must be done. Assuming a constant energy input to the system, $\dot{m}_{EGR}$ and $T1_{EGR}$ constant, one must understand whether it is better to have a high/low refrigerant mass flow. A high mass flow means lower steam temperature and vice versa. Also a high refrigerant mass flow results in a larger steam power after the evaporator compared to the case with lower refrigerant flow as learned earlier. But as mentioned earlier the expander's isentropic efficiency is dependent on the filling factor. So if a higher mass flow strategy is used it means a lower steam temperature which leads to higher steam density in the expander. But if a lower mass flow strategy is approached that means more superheated steam and therefore lower steam density. A higher mass flow means the filling factor in the expander will increase, a lower steam density means that the vapor takes more place inside the expander leading to larger filling factor. Therefore a simulation was run were the steam pressure $P3_{Ref}$ was varied and the steam temperature $T3_{Ref}$ was varied for every corresponding pressure. The purpose of this simulation was to understand which strategy result in highest power output from the expander but also to understand which strategy leads to highest filling factor in the expander.

For a constant energy input different pressure levels have been simulated, 40-180 bar. Also for every pressure, four different refrigerant flows have been simulated. For example at 100 bar the boiling temperature is 311˚ C. As figure 44 indicate the first simulation will result in steam temperature at 321 ˚ C, the second 331˚ C, the third 351˚ C and the final simulation 371˚ C which corresponds to 10-60 degrees superheated steam. The result is presented in terms of the piston expander's power output in figure 44. The optimal system pressure for point 2 is 120 bar.

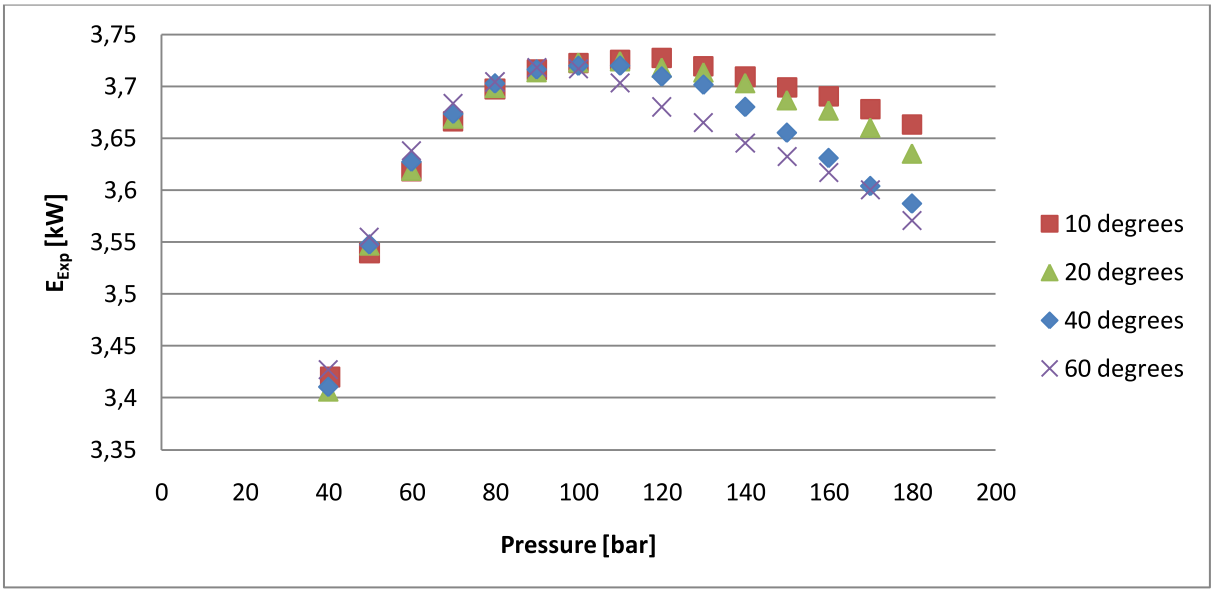


Figure 44. *The expander power for several pressures and temperatures*

As mentioned earlier the thermal efficiency of a Rankine cycle is proportional to the steam pressure, which means a higher pressure results in higher thermal efficiency, see figure 45. Higher level of superheating result to a higher system efficiency compared with a lower level of superheating. In order to acquire a higher steam temperature the refrigerant flow must be reduced. This means lower pump power and steam power as shown earlier. Since the WHR efficiency takes the pump- and steam -power into consideration, see equation 15, this leads to a higher efficiency compared to just superheating the steam. But the disadvantage is the decreased steam power entering the expander, compared to the just superheated steam, leading to a lower power output from the expander, see figure 46.

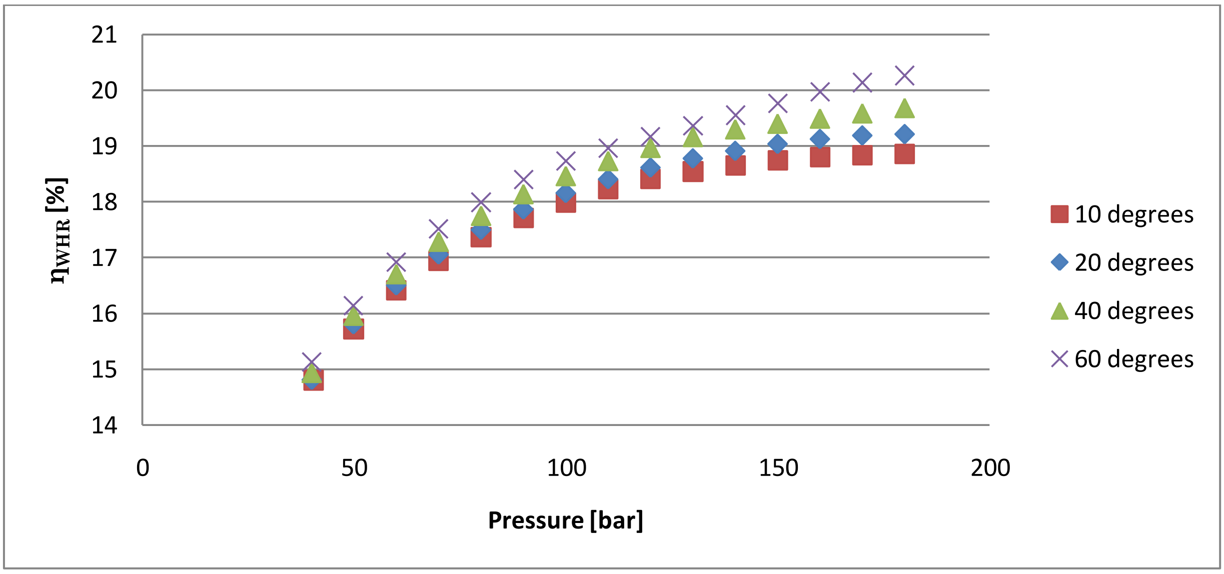


Figure 45. *The thermal efficiency of the system at different pressures and temperatures*

Still the highest expander power is not acquired at 180 bar (see figure 44), this depends on the steam power before the expander, see figure 46.

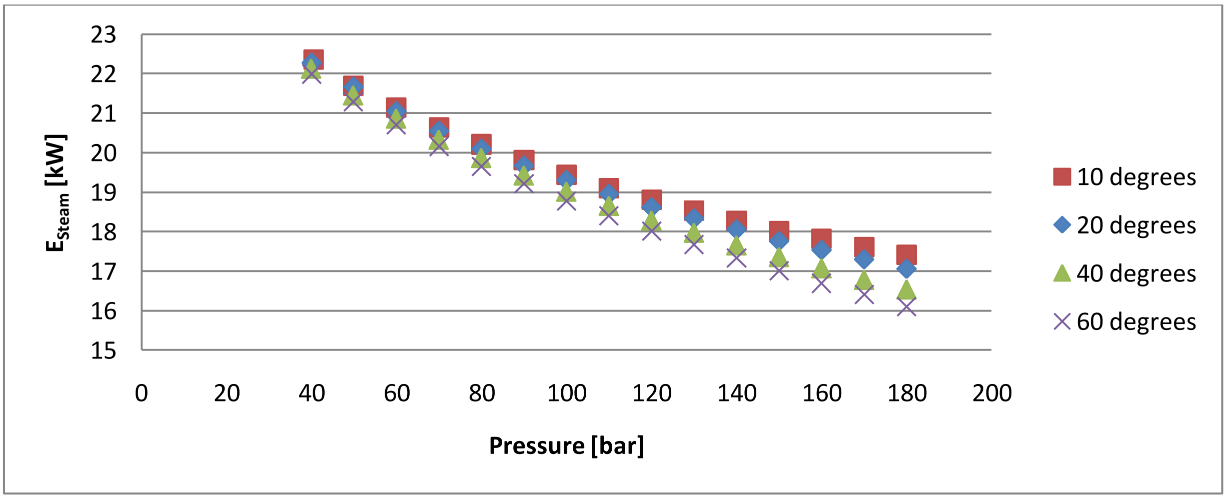


Figure 46. *Steam power in front of the expander*

As figure 46 shows highest steam power is obtained at the lowest pressure, this depends on the boiling temperature. Because of the low boiling temperature the mass flow can be increased. Also it is possible to see that roughly superheated steam result in highest power output from the evaporator.

As described earlier the expander is replaced by a speed regulated turbine in GT-Power, this means that the turbines speed reveals the filling factor i.e. if the turbines speed increases when the filling factor increases and vice versa. Analyzing figure 47 shows that a higher pressure level gives a lower filling factor which would mean higher isentropic efficiency. Also it is possible to see that the margin between the different levels of superheated steam decrease with higher steam pressure. But as figure 44 shows the best power output from the expander is obtained at 120 bar for the lowest level of superheated vapor, this means it is best to run with lowest level of superheated vapor. If taking the higher isentropic efficiency, for higher steam pressures, into account as a function of filling factor the optimal operating pressure will increase some. In this study the isentropic efficiency has been set to a constant value of 65 %. The dependence of the filling factor has not been considered.

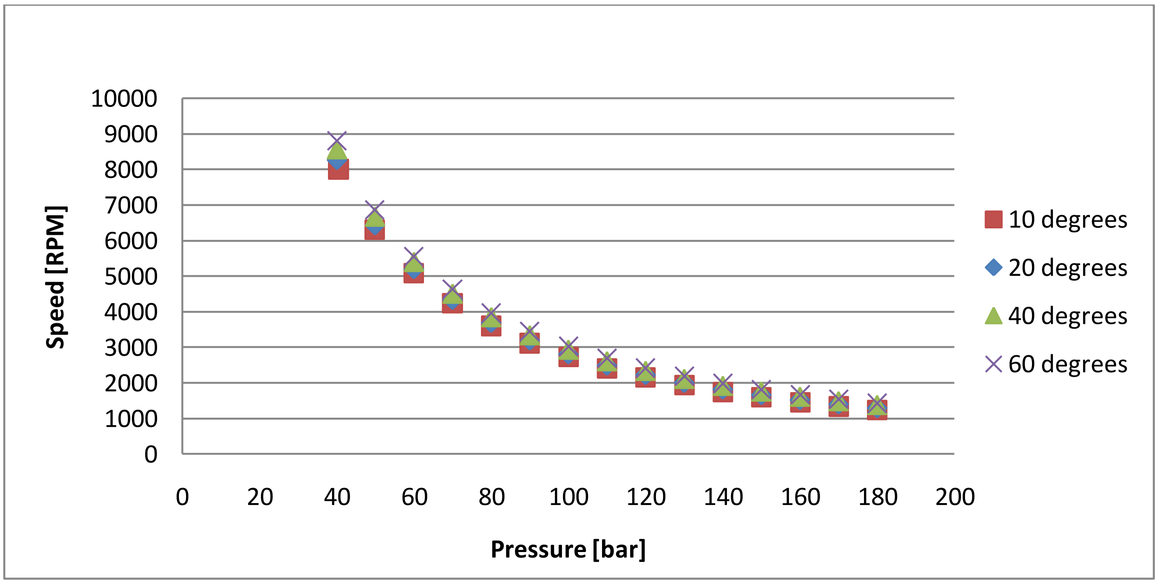


Figure 47. The expander speed for several refrigerant pressure and temperatures

The second optimization of this system is to decide the optimal steam pressure for different EGR inlet temperatures. Optimal steam pressure is the pressure that results in the highest power output from the expander. It is known that low steam pressures results in the largest steam power output from the evaporator but at the same time with higher steam pressure a higher system thermal efficiency is acquired. Therefore a simulation was run with varied steam pressure for different EGR inlet temperature, see appendix 8.3 for full result. As can be seen in figure 48 the steam pressure should be chosen depending on the EGR inlet temperature, a higher temperature leads to higher steam pressure.

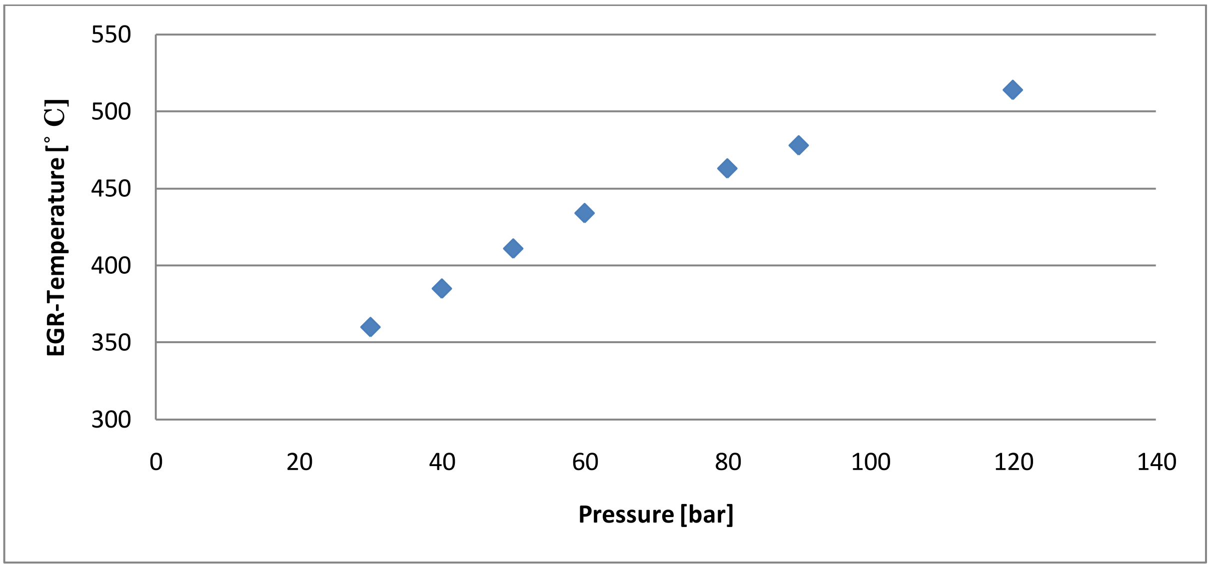


Figure 48. *Optimal steam pressure for different EGR inlet temperature*

3.2.5 The ESC driving cycle

In this chapter several operating points from table 1 will be presented. The simulation strategy was to decide the steam pressure with the help of figure 48 while the level of superheating is chosen to 10 degrees superheated steam according to the result. This should result in maximizing the power output from the expander.

The steam power for the different operating points can be seen in table 6, as mentioned earlier a higher EGR temperature result in higher steam pressure. The highest steam pressure that results in superheated vapor is chosen for operating point 9, 25 % load case, this doesn't necessarily mean the optimal pressure, meanwhile the highest steam pressure is 120 bar for operating point 2. The highest refrigerant flow is 11,2 *g/s* for point 10. The pressure drop of the refrigerant is 7,9 bar for the highest mass flow.

Table 6. *The refrigerant properties*

Operating point	Steam Power [kW]	Steam pressure [bar]	Refrigerant Flow [g/sec]	Refrigerant Pressure drop [bar]	Steam Temperature [˚C]
2	19,2	120	8,15	6,25	335
3	11,5	40	4,7	4,4	260
4	18,2	60	7,46	5,77	286
6	15,9	90	6,56	5,19	313
8	25,1	90	10,39	7,9	313
9	5,2	18,3	2,19	2,9	222
10	27,25	80	11,21	8	305
12	19,9	50	8,13	6,23	274
13	12,5	30	5	4,48	244

In table 7 it is possible to see how much of the steam power that is extracted by the expander. Also the system efficiency is presented, as can be seen in tables 6-7 a higher steam pressure result in higher WHR efficiency. Finally the reduction in fuel consumption is presented for the corresponding operating point. Operating point 10 stands for the largest saving with 1,7 % at the actual engine load case.

Table 7. *Illustrates required/obtained power of the different components for several operating points*

Operating point	Pump Power [kW]	Expander Power [kW]	WHR Efficiency [%]	RFC [%]
2	0,26	3,72	18,4	1,46
3	0,05	1,8	15,2	1,31
4	0,12	3,1	16,4	1,48
6	0,158	2,93	17,4	1,51
8	0,259	4,53	17	1,65
9	0,013	0,78	14,7	1,15
10	0,25	4,77	16,6	1,7
12	0,115	3,22	15,6	1,55
13	0,043	1,78	13,9	1,31

The EGR power input to the evaporator can be seen in table 8. The EGR power varies from 12 *kW,* 25 % load case, to 50 *kW*, 100 % load case. The EGR cools down only 82 degrees for point 9. This depends on the low EGR inlet temperature but also the low refrigerant flow. The EGR pressure drop is the pressure drop over both the evaporator and EGR cooler.

Table 8. *Presents the cooling of the EGR and EGR power for several operating points*

Operating point	T-EGR [˚C]	ΔT EGR [˚C]	EGR Flow [g/s]	EGR Pressure Drop [bar]	EGR Power [kW]
2	514	227	77,9	0,78	36,16
3	385	148	71,4	0,3	22,6
4	434	186	90,3	0,6	33,6
6	478	221	66,3	0,5	28,3
8	474	211	109,8	0,8	46,1
9	295	82	60,1	0,04	12
10	463	205	122,4	0,82	49,9
12	411	164	111,6	0,55	38,7
13	360	126	89,3	0,35	25,9

The cooling power required is illustrated in table 9, it varies from 4,5 *kW* up to 21,8 *kW*. The temperature and pressure of the condenser can also be seen in table 9. For example, the pressure differs with 0,4 bar when comparing point 10 and 9. This is due to the filling factor in the expander, for the full load point the volume flow rate is much higher than for the 25 % load point. This results in higher condenser pressure for point 10 since the expander doesn't utilize enough energy of the superheated steam.

Table 9. The temperature and pressure of the condenser, the cooling power for several operating points

Operating point	Condenser Temperature $T4_{REF}$ [˚ C]	Condenser Pressure $P4_{REF}$ [Bar]	$T1_{Ref}$ [˚C]	$P1_{Ref}$ [bar]	Cooling Power [kW]
2	108,1	1,35	103	1,3	15,3
3	106,4	1,27	102	1,26	9,6
4	107,8	1,33	103	1,31	14,8
6	106,9	1,29	102	1,26	12,6
8	110,9	1,48	104	1,42	20
9	102	1,1	100	1,1	4,5
10	112,1	1,54	105	1,47	21,8
12	108,9	1,38	103	1,35	16,3
13	106	1,25	102	1,24	10,4

Figure 49 represents the T-S diagram for operating point 2. In stage (A) the refrigerant is in liquid state just ahead of the evaporator. While stages (B)-(D) illustrates the refrigerant heat up until it reach the boiling point, evaporating and finally superheating in the evaporator. Stage (E) represents the expansion phase. At the end of the expansion the refrigerant is in two phase state, 15 % water and 85 % steam in this case. The final stage (F) represents the condenser where the refrigerant leaving the condenser is only in liquid form and sub cooled by two degrees, see also figure 1 and figure 5 for better understanding.

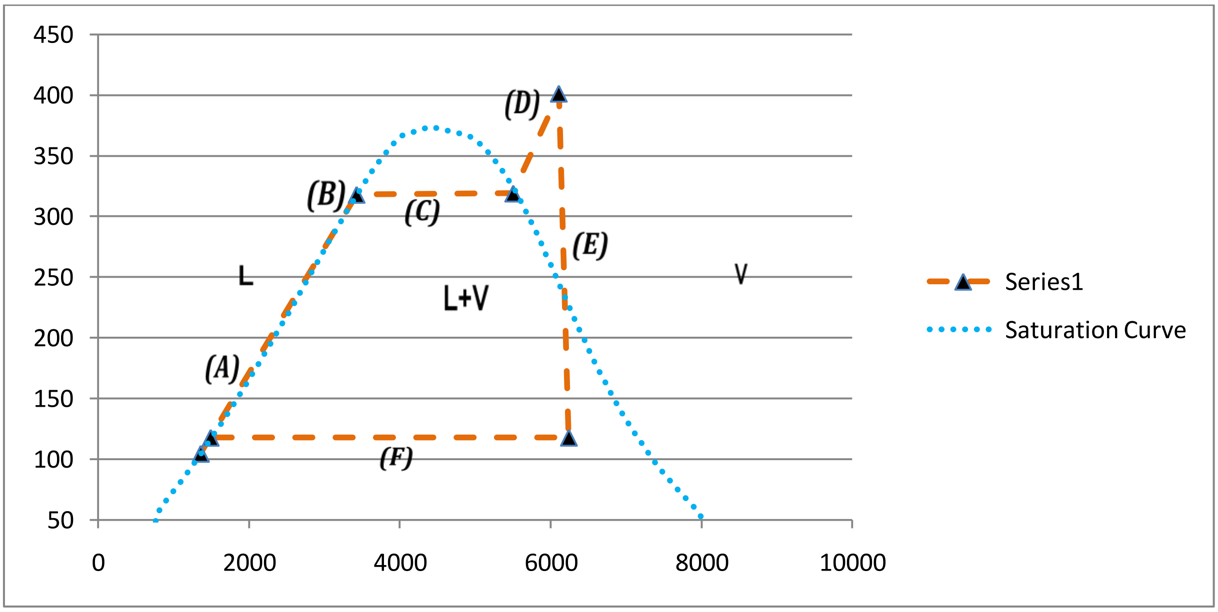


Figure 49. *T-S diagram for operating point 2*

3.2.6 Exhaust flow

This simulation represents the case where an evaporator would be mounted in the exhaust pipe, after the turbine. This means a larger exhaust flow, compared with the EGR flow. The disadvantage is the low exhaust temperatures since they cool down 80-150 degrees compared to the EGR temperatures. This simulation was performed for three different operating points from the ESC cycle. The power, pressure, temperature and the flow of the steam can be seen in table 10. The optimal steam pressure is 30 bar for operating point 2 according to figure 48. The level of superheating is increased since just superheated steam didn't work in the model. This is because of the big change in the energy input which the model is not calibrated for. As for the steam pressure for operating points 3-4, the lowest working pressure is applied since there is no data in this master thesis for the optimal steam pressure for low exhaust/EGR temperature. The same evaporator used in the EGR loop is now placed in the exhaust pipe, considering the large increase in the exhaust mass flow the evaporator is under dimensioned leading to a decreasing efficiency.

Table 10. *The refrigerant properties*

Operating point	Steam Power [kW]	Steam pressure [bar]	Refrigerant Flow [g/sec]	Refrigerant Pressure drop [bar]	Steam Temperature [˚C]
2	34,7	31,8	13,6	8,7	280
3	18,2	22	7,3	5,4	240
4	31,96	23,8	12,5	8,4	270

The expander power is increased for all the three points compared to only the EGR as an energy input, this leads to increased reduction of the fuel consumption.

Table 11. *Illustrates required/obtained power of the different components*

Operating point	Pump Power [kW]	Expander Power [kW]	WHR Efficiency [%]	RFC [%]
2	0,13	4,74	13,3	1,95
3	0,047	2,35	12,68	1,73
4	0,099	4,38	13,39	2,14

Table 12 illustrates the exhaust gas properties, as can be seen the highest exhaust gas temperature is 360 ˚C while the exhaust mass flow is much higher than the EGR flows. This results in increased exhaust power input energy to the system. The evaporator's efficiency $\eta_{Exhaust}$ is based on how much power it utilizes from the exhaust, it can be seen in table 12. The exhaust evaporator efficiency $\eta_{Exhaust}$ for operating point 2 is 20 % lower than the EGR evaporator efficiency η_{EGR}, see figure 37. As learned earlier the evaporator's efficiency increases if the steam pressure decreases, since the refrigerant flow can be increased leading to more energy utilized from the exhaust/EGR power. The steam pressure in the exhaust pipe is about 30 bar while 90 bar in the EGR pipe. This means that the evaporator is highly under dimensioned.

Table 12. *The exhaust gas properties*

Operating point	T-Exhaust [˚C]	ΔT Exhaust [˚C]	Exhaust Flow [g/s]	$\eta_{Exhaust}$ [%]	Exhaust Power [kW]
2	360	112	265	36,4	77,3
3	305	86	192	35,8	43,79
4	320	103	256	40,7	63,38

The cooling power required in the condenser increases because of the increased steam power entering the expander.

Table 13. The temperature and pressure of the condenser, the cooling power for several operating points

Operating point	Condenser Temperature $T4_{REF}$ [˚ C]	Condenser Pressure $P4_{REF}$ [Bar]	$T1_{Ref}$ [˚C]	$P1_{Ref}$ [bar]	Cooling Power [kW]
2	116	1,75	107	1,66	28,9
3	108	1,36	103	1,33	15,43
4	106	1,25	103	1,24	26,9

Finally table 14 represents two evaporators, one in the EGR loop and the other in the exhaust pipe. When combining the energy output from the expander for both the evaporators result in an increased reduction of the fuel combustion.

Table 14. *The reduction of fuel consumption for both evaporators*

Operating point	Total RFC [%]
2	3,41
3	3,04
4	3,62

3. Discussion

If one just looks at the evaporator the strategy should be low steam pressure and just superheated steam i.e. high refrigerant mass flow. This would result in the largest steam power after the evaporator, according to figures 25 and 46.
The purpose is to achieve a high output power from the expander, but with the strategy mentioned above this will not happen. As figure 45 indicate the system thermal efficiency rise with increasing steam pressure, therefore a high steam pressure is preferable. The disadvantage is the decreased steam power from the evaporator. Therefore an optimum needs to be found where the power output from the evaporator and the thermal efficiency at the prevailing steam pressure results in the largest power obtained from the expander. Figure 44 shows the optimal pressure for the operating point.

Assuming a constant steam pressure the superheating degree has to be decided in order to obtain the largest expander power. If one only looks at the evaporator at a constant steam pressure, roughly superheated vapor would lead to the highest power output, see figure 44. One disadvantage is the increased cooling power needed, the same logic applies at the condenser as for the evaporator i.e. higher refrigerant flow results in higher cooling power. Analyzing figure 47 though shows that the expander speed i.e. filling factor is lower for the roughly superheated vapor, but also that the margin is decreasing for higher steam pressures. This means for even higher pressures, up to 250 bar the larger level of superheated steam may have a lower expander speed if the trend remains. Looking at figure 44 clearly suggest that the optimal pressure is way lower than 250 bar, which should mean that the optimum level of superheating is precisely superheated steam. The issue is the constant isentropic efficiency used for the expander, again if one looks at figure 47 one discovers that the speed of the expander is lower for higher steam pressures. This means that isentropic efficiency of the expander is higher for higher pressure, which could result in lifting the optimal steam pressure in figure 44.

If the optimal steam pressure would increase to say 250 bar, following the trend in figure 47 would suggest higher isentropic efficiency for the larger level of superheating. Therefore it is crucial to determine a relationship between the expanders speed and the isentropic efficiency in order to acquire more accurate results.

Studying figures 35 and 42 clearly speaks in the favor of the EGR temperature over the EGR flow, for every 5 kW increase in the EGR power 0,5 kW is obtained from the expander if the source of the rise was the EGR flow. While the double is obtained if the input energy lift was caused by the EGR temperature. Therefore it is favorable to increase the EGR temperature, but looking at figure 40 shows that it is better to use the required diesel flow needed to increase the EGR temperature a desired amount of degrees directly in the combustion chamber, rather than supplementary burning.

Figure 49 shows the EGR temperature versus the optimal steam pressure. Higher EGR temperature i.e. higher energy input result in higher optimal steam pressure which is logical. Because higher energy input to the system results in the possibility to increase the steam pressure and acquire a higher system thermal efficiency while still getting a reasonable power output from the evaporator. This leads to larger power output from the expander at a higher steam pressure. This is also confirmed by figures 34 and 41, when increasing the energy input to the system, the 150 bar line equals/overtakes the 90 bar line. Initially the optimal pressure is closer to 90 bar but as the energy input grows the steam pressure rises closer or even passes the 150 bar line resulting in the 150 bar line overtaking the 90 bar line in terms of expander power. Also the difference between the EGR temperature and the EGR flow is visible in this case, as an increase in EGR temperature clearly pushes the optimal pressure higher than the corresponding EGR flow. That's the reason why the 150 bar line overtakes the 90 bar line, in the EGR temperature figure, while the 150 bar line only equals the 90 bar line in the EGR flow figure.

For a constant flow figures 23 and 24 shows that the heat transfer is larger at lower steam pressure but the evaporator works better at higher steam pressures. The heat transfer from the exhaust gases is smaller at a higher steam pressure compared to a lower but the steam outlet temperature is still higher meaning the performance of the evaporator is better at higher pressures. Looking only at the evaporator the magnitude of the applied steam pressure is not important if the refrigerant mass flow is constant, since the difference in terms of output power is negligible. If one looks at the whole system then it is desirable to apply as high pressure as possible in order to raise the thermal efficiency of the system.

The reduction of fuel consumption is approximately 1,4 % in operating point 2, see figure 43. If the EGR power is increased 10 kW the reduction of fuel consumption will rise with approximately 0,2 %, if the EGR flow was the reason of the lift in EGR power. If the EGR temperature is the cause in the change in EGR power then a 5 kW lift result in 0,3 % improvement of the fuel consumption.

The evaporator manufactured by Ranotor is over dimensioned for this specific engine (DC1306). Increasing the EGR flow with approximately 67 % resulted in 1 % decrease in the evaporator's efficiency, see figure 42. While increasing the EGR temperature lifted the evaporator's EGR efficiency η_{EGR}, see figure 37. This depends on that the same amount of EGR flow enters the evaporator, but with higher temperature i.e. more energy. In the case of increased EGR flow, the mass flow entering the evaporator increases leading to less cooling of the EGR gases, due to lack of heat exchange areas.

The reduction of fuel consumption lays between 1,1-1,7 % for most of the ESC cycle operating points. This percentage could be increased either by using the entire exhaust gases instead of only the EGR gases. For example operating point 2 reduces the fuel consumption with 3,4 %, see table 14. The highest cooling power required by the condenser is about 21 kW for operating point 10, since the EGR cooling is no longer needed some power is saved and could instead be used to condense the vapor. If the required cooling power is too high, more than the engines cooling capacity, one could opt to run that specific operating point with a high level of superheating, this leads to a reduced cooling power demand from the condenser.

Figure 49 shows that at the end of expansion state the refrigerant is in the two phase area. As mentioned earlier for a piston expander this is no problem. The water percentage is 20 % as a maximum for all the simulated points in the ESC cycle. Assuming a constant energy input to the system i.e. constant operating point, then the water percentage increases with the steam pressure. As figure 47 reveals the filling factor of the expander decreases with higher steam pressures, meaning a larger percentage of water. Assuming there is a requirement of having superheated steam after the expansion phase, the solution would be to increase the level of superheating in the evaporator. The water percentage is about 10 % lower for the 60 degrees level of superheating compared to the 10 degrees level of superheating, this is for a full load operating point.

4. Conclusion

The steam pressure should be depending on the EGR temperature, a higher EGR temperature means a higher steam pressure. The steam should be roughly superheated in order to obtain the largest power output from the expander. Since the condenser is water cooled and the cooling power is obtained from the engine it could happen, probably at full load, that the engine power is not enough to condense the steam. At this situation the superheating level should be increased resulting in a reduced cooling power required by the condenser.

Supplementary burning is a method used to increase the EGR inlet temperature by injecting fuel in the exhaust pipe, resulting in increased power input to the system which leads to higher power obtained from the expander. Calculations show that it is better to inject the diesel directly into combustion chamber since the power output from the engine is larger for the same diesel flow.

The water mass flow is very low, 20 *g/s* as a maximum, which means the receiver in the GT-Power model is too large. The volume is 10 liter*,* considering the low refrigerants mass flow indicates that the receiver could be downsized.

In order for this WHR system to be profitable the entire energy in the exhaust gases must be used i.e. not only the EGR as the case was in this master thesis. Especially when analyzing point 9, 25 % load point, only 0,8 *kW* is obtained from the expander. Therefore it is crucial to use all the exhaust energy in order to push up this number.

The WHR system is best suited in application where the engine loading case is not varied too much. Since heavy duty trucks engines and marina engines often travel long distances and the engine loading case is not varied too much the WHR system is a good option in order to utilize the energy in the exhaust gases. The WHR system is a more challenging solution for inner city driving.

5. Future work

A master thesis where the primary goal is to build a model of a piston expander in GT-Power and customize the isentropic efficiency curve as a function of the expanders filling factor. Thereafter connect the piston expander, instead of the turbine, to the model and investigate if the current result still agrees. Also mount the piston expander downstream the evaporator and run some experimental result. Then compare the experimental results with the simulation result to validate the model.

Build an evaporator with bigger heat exchange area in GT-Power, in order to avoid having an under dimensioned evaporator. This leads to increased power output from the expander meaning larger reduction in fuel consumption, when compared to the current results.

Compare water to other refrigerant in order to understand if water is a sufficient working fluid.

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8. Appendix

8.1 Condenser Data

CONDENSER - Performance

Condenser: B200Tx30

Fluid Side 1 : Steam
Fluid Side 2 : Water
Flow Type : Counter-Current

Table 15. *The heat load, inlet- and outlet temperature of the condenser*

DUTY REQUIREMENTS		Side 1		Side 2
Heat load	kW		50,00	
Inlet temperature	°C	110,00		88,00
Condensation temperature (dew)	°C	101,85		
Subcooling	K	2,00		
Outlet temperature	°C	99,86		95,00
Flow rate	kg/s	0,02196		1,697
Fluid condensed	kg/s	0,02196		
Max. pressure drop	kPa	50,0		50,0

Table 16. *Pipe diameters, Reynolds number and operating pressure*

PLATE Condenser		Side 1		Side 2
Total heat transfer area	m²		3,61	
Heat flux	kW/m²		13,8	
Mean temperature difference	K		10,14	
O.H.T.C. (available/required)	W/m²,°C		6340/1370	
Pressure drop -total*	kPa	0,845		16,7
- in ports	kPa	-0,0282		0,290
Operating pressure - outlet	kPa	107		
Number of channels		14		15
Number of plates			30	
Oversurfacing	%		364	
Fouling factor	m²,°C/kW		0,575	
Port diameter	mm	60,0		53,0
Recommended inlet connection diameter	mm	From 42,2 to 94,5		
Recommended outlet connection diameter	mm	From 1,71 to 5,41		
Reynolds number				3070
Inlet port velocity	m/s	12,3		0,798

Table 17. *Physical properties of the condenser*

PHYSICAL PROPERTIES		Side 1	Side 2
Reference temperature	°C	101,86	91,50
Liquid - Dynamic viscosity	cP	0,277	0,309
- Density	kg/m³	957,0	964,4
- Heat capacity	kJ/kg,°C	4,220	4,209
- Thermal conductivity	W/m,°C	0,6796	0,6759
Vapor - Dynamic viscosity	cP	0,0123	
- Density	kg/m³	0,6269	
- Heat capacity	kJ/kg,°C	1,973	
- Thermal conductivity	W/m,°C	0,02529	
- Latent heat	kJ/kg	2,252	
Film coefficient	W/m²,°C	16300	14600
Average wall temperature	°C	99,52	98,57
Channel velocity	m/s	5,23	0,246

Heatexchanger : B200Tx30

Totals	Unit	Value
Total weight (no connections)	kg	21,8
Hold-up volume, inner circuit	dm^3	3,40
Hold-up volume, outer circuit	dm^3	3,60
Port size F1/P1	mm	60,0
Port size F2/P2	mm	53,0
Port size F3/P3	mm	42,0
Port size F4/P4	mm	53,0
NND F1/P1	mm	65
NND F2/P2	mm	58
NND F3/P3	mm	42
NND F4/P4	mm	58
Channel plate thickness	mm	0,350

8.2 Pump, Expander & Reciever

Component	Isentropic efficiency [%]	Volumetric efficency [%]	Template name in GT-Power
Pump	40	80	PumpPosDispRefrig
Expander	65	70	TurbPosDispRefrig
Reciever	-	-	RecieverDryerRefrig

8.3 GT-Power results

In this section the simulation results for the other operating points in the ESC cycle are presented.

8.3.1 EGR temperature variation

The following results are for operating point 12 in the ESC cycle.

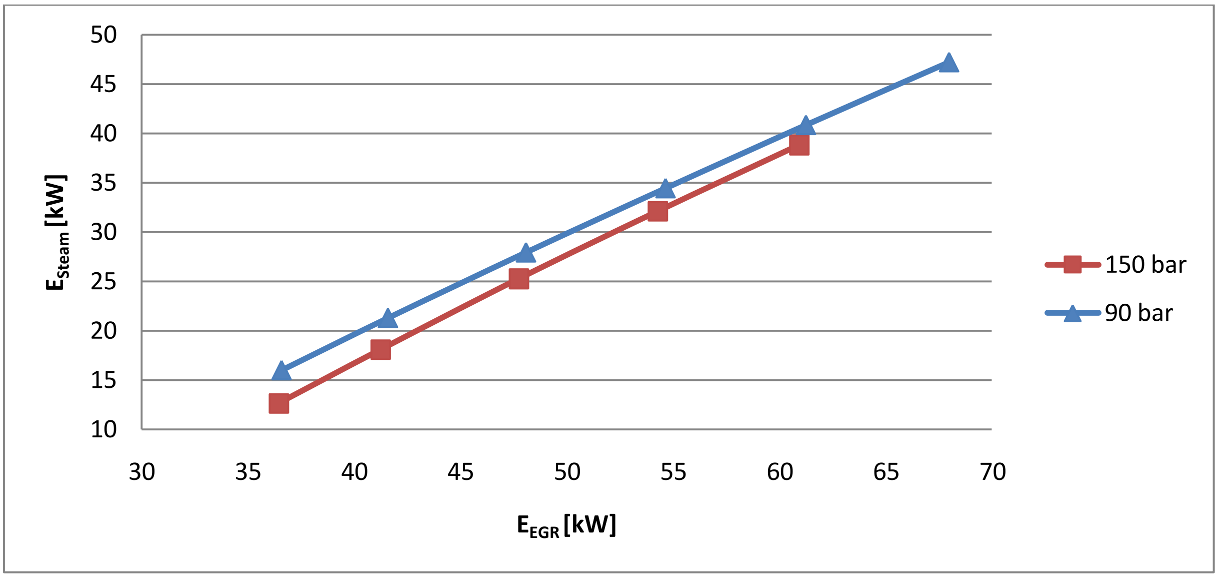

Figure 50. *The steam power and EGR power for two steam pressures*

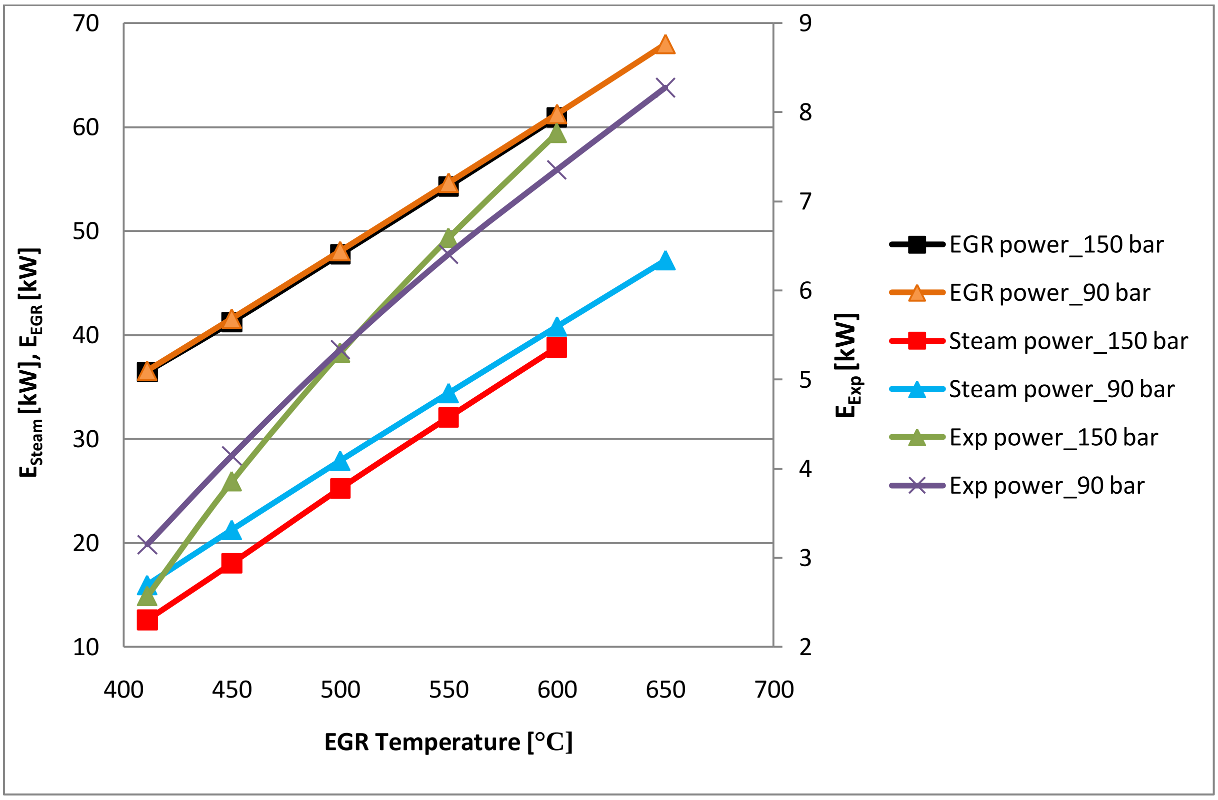

Figure 51. *The expander power, steam power and the EGR power for two different steam pressures*

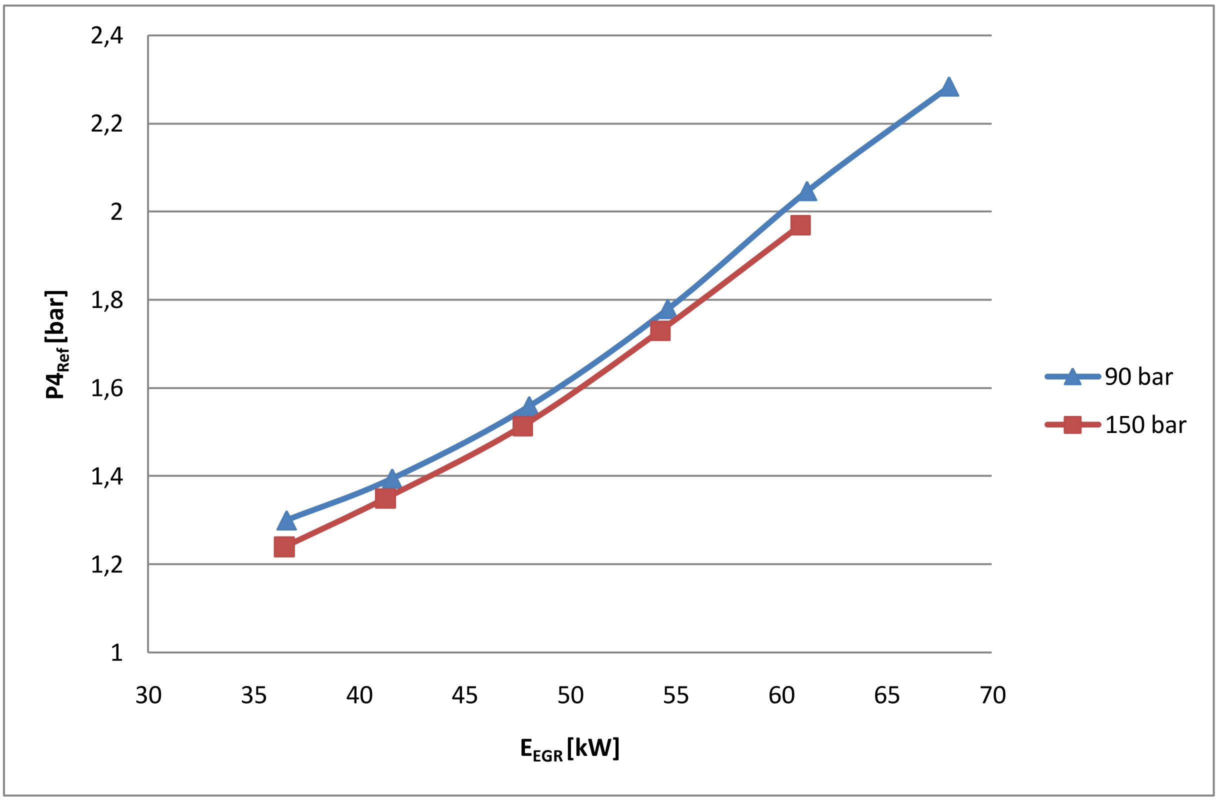

Figure 52. *The pressure ahead of the condenser*

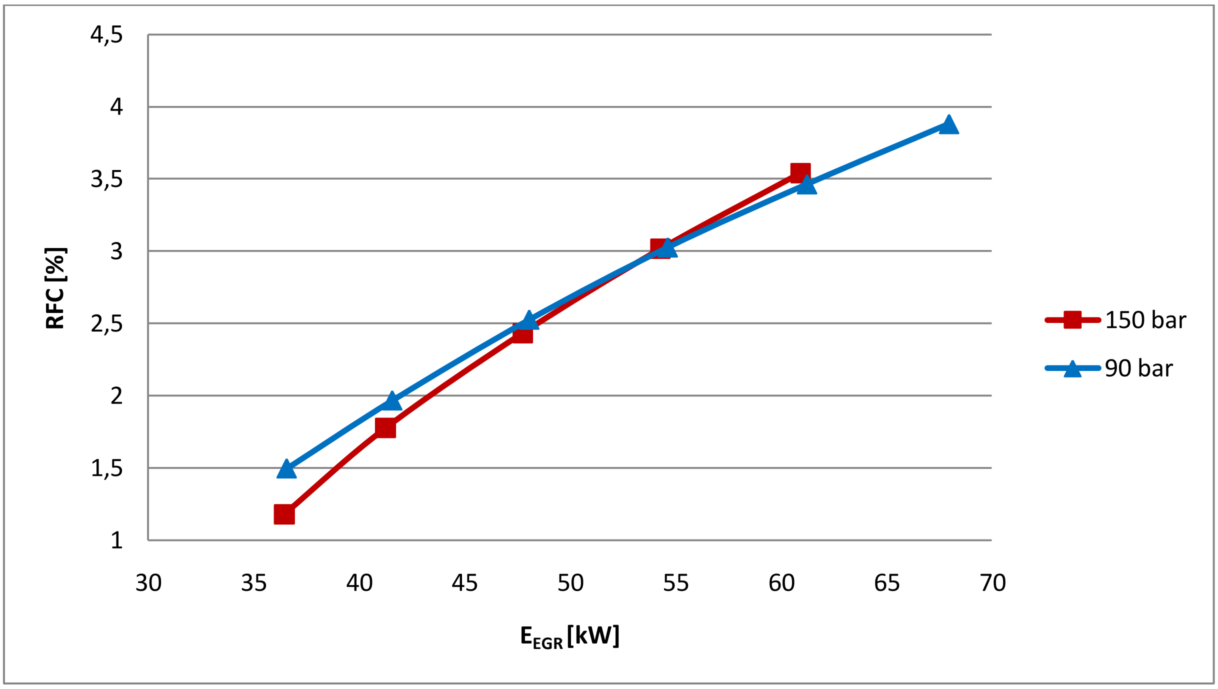

Figure 53. *The reduction in fuel consumption for two different steam pressures*

8.3.2 EGR flow variation

The following results are for operating point 12 in the ESC cycle.

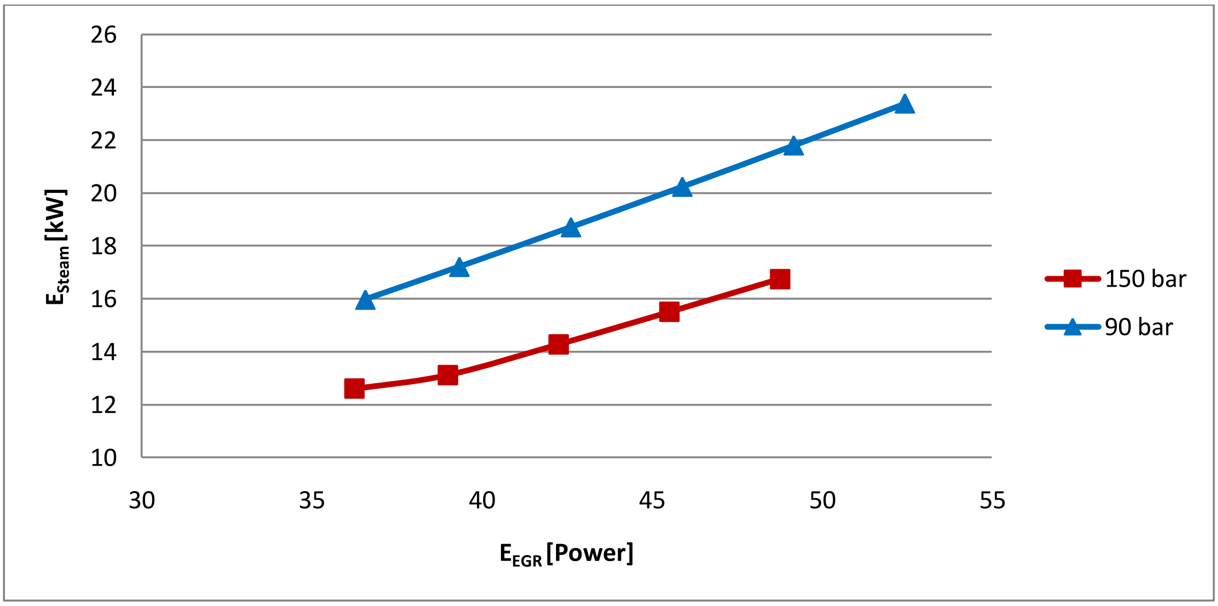

Figure 54. *EGR power and steam power for operating point 12*

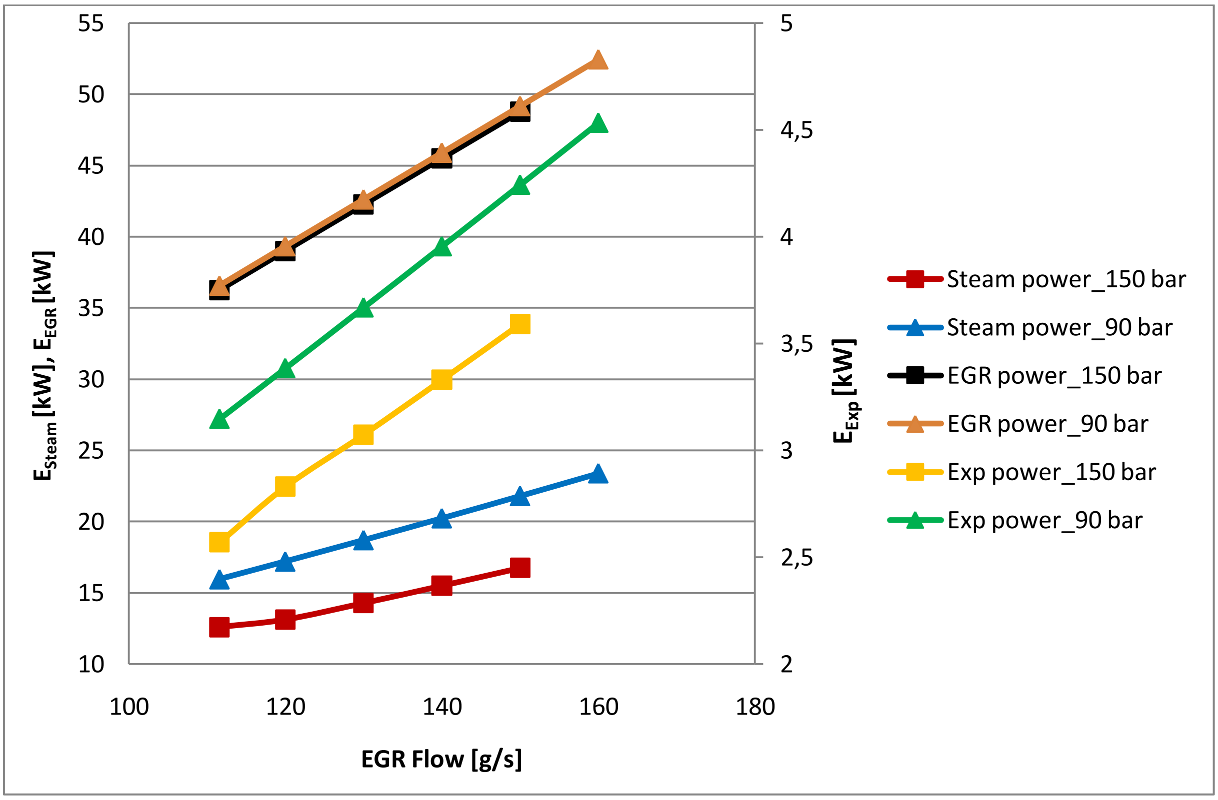

Figure 55. *EGR power, steam power and expander power for two different steam pressures*

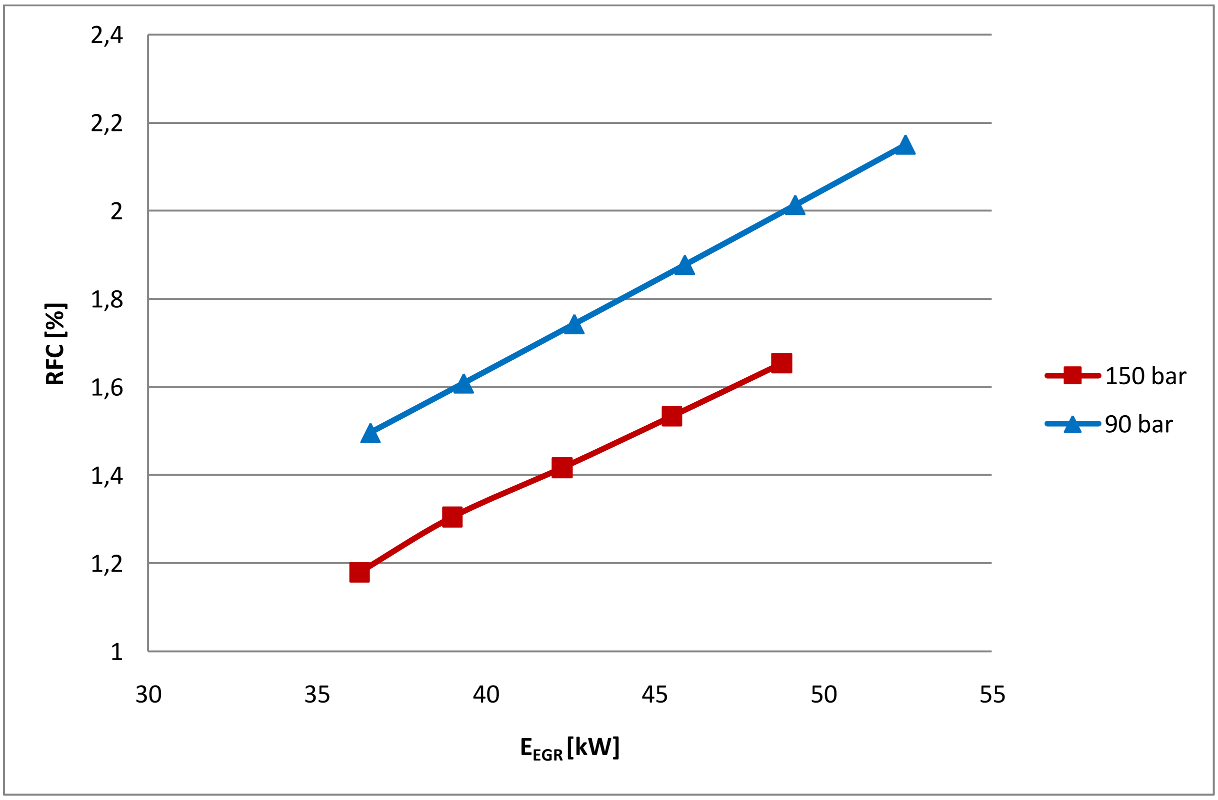

Figure 56. *The reduction in fuel consumption for two different steam pressures*

8.3.4 Optimizing the system

The result for operating points 3, 4, 6, 8, 10, 12 and 13 are presented in figures 57-69 below.

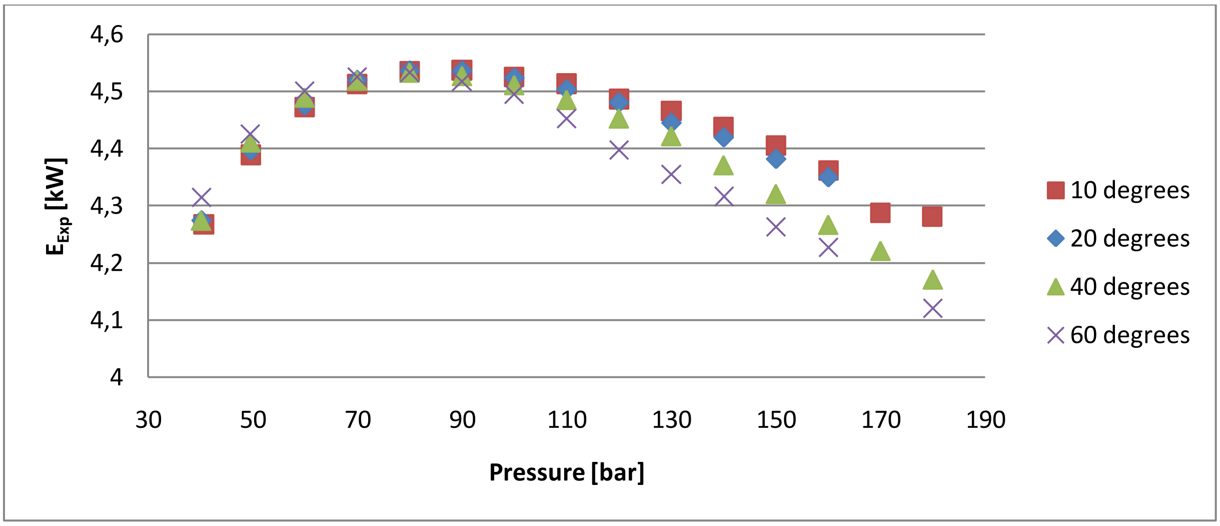

Figure 57. *Expander power for several steam pressures, point 8*

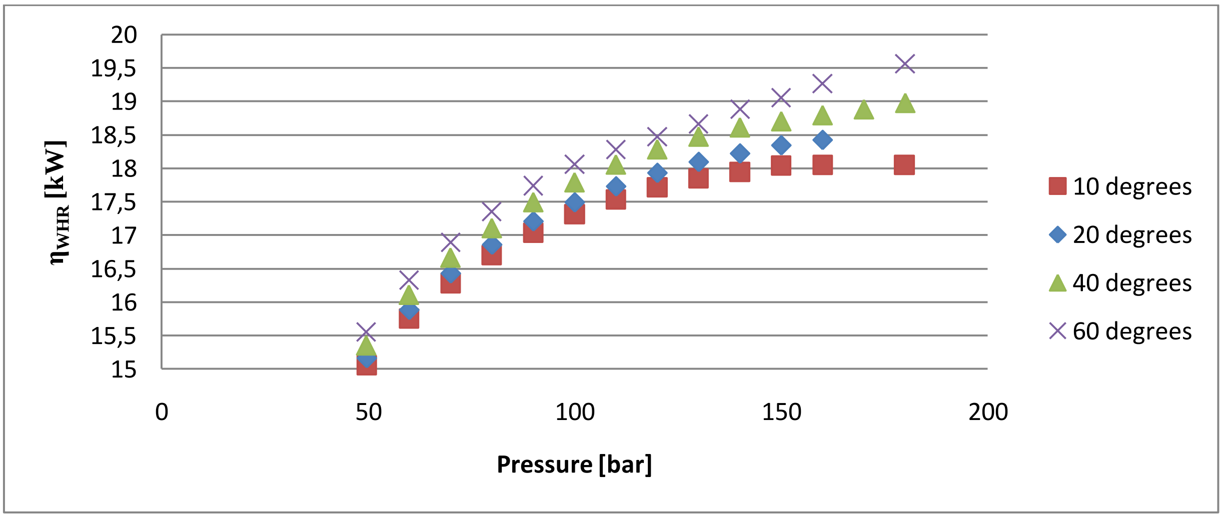

Figure 58. *The thermal efficiency of the system for different steam pressures, point 8*

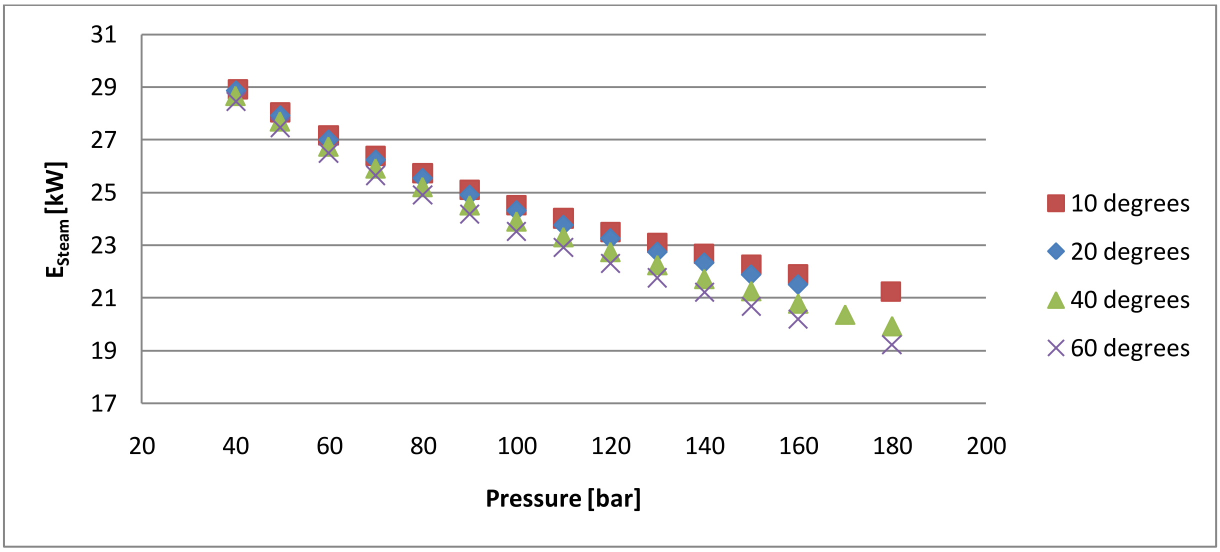

Figure 59. *The steam power for different pressures, point 8*

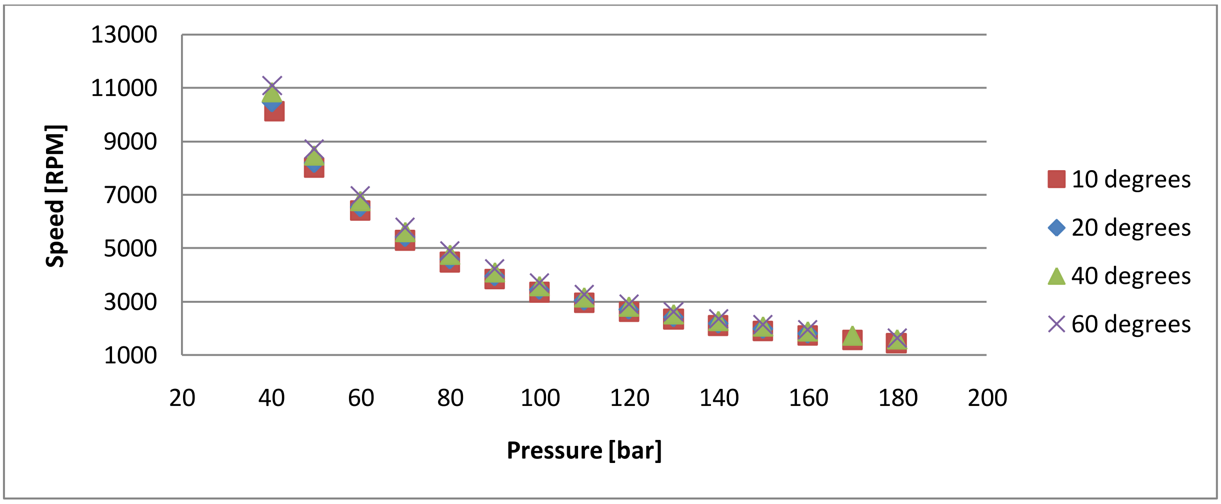

Figure 60. *The expander speed for different steam pressures, point 8*

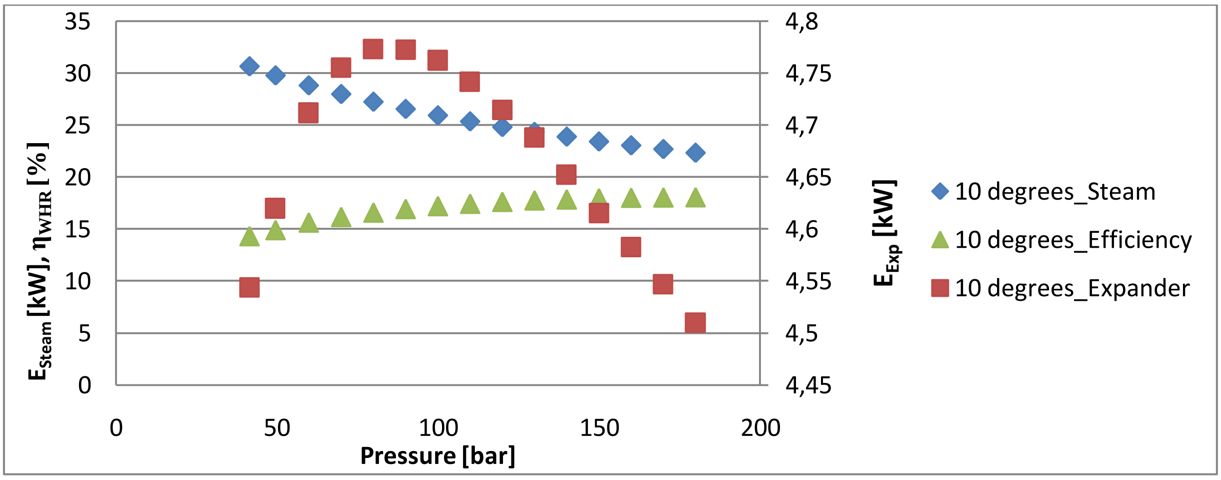

Figure 61. *The expander power, steam power and system efficiency for different pressures, point 10*

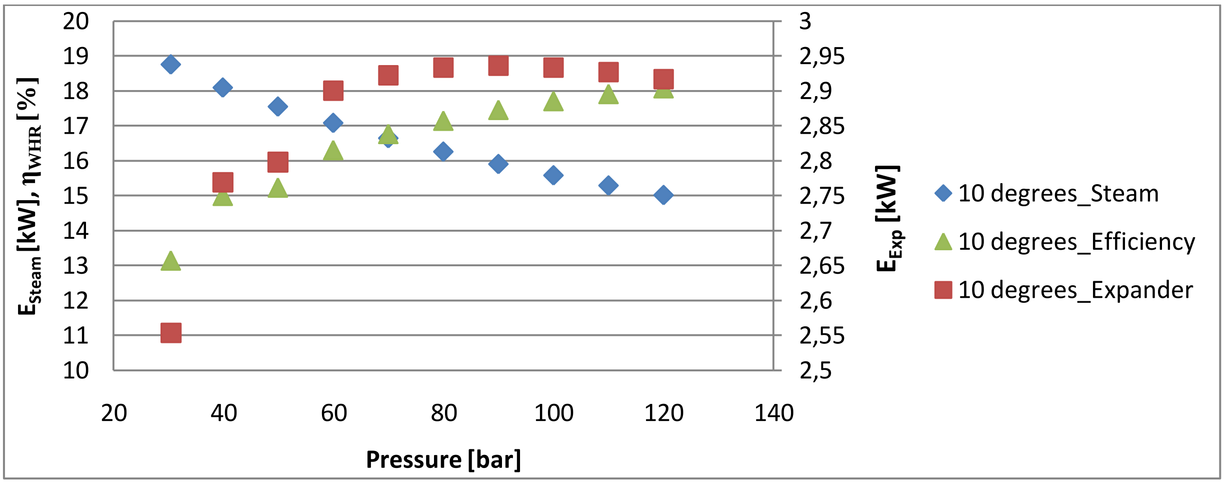

Figure 62. *The expander power, steam power and system efficiency for different pressures, point 6*

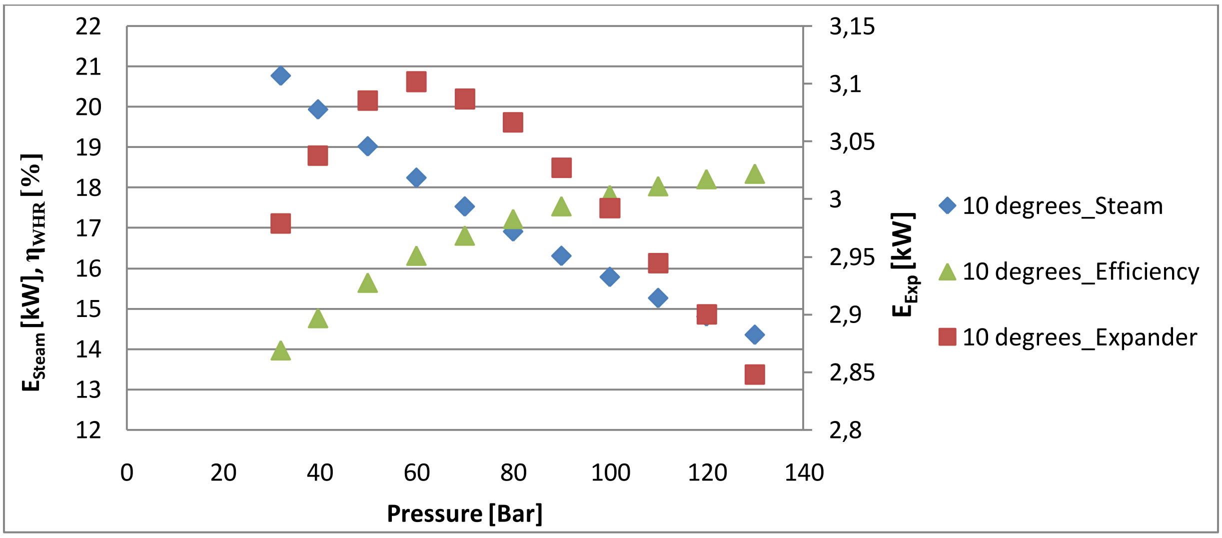

Figure 63. *The expander power, steam power and system efficiency for different pressures, point 4*

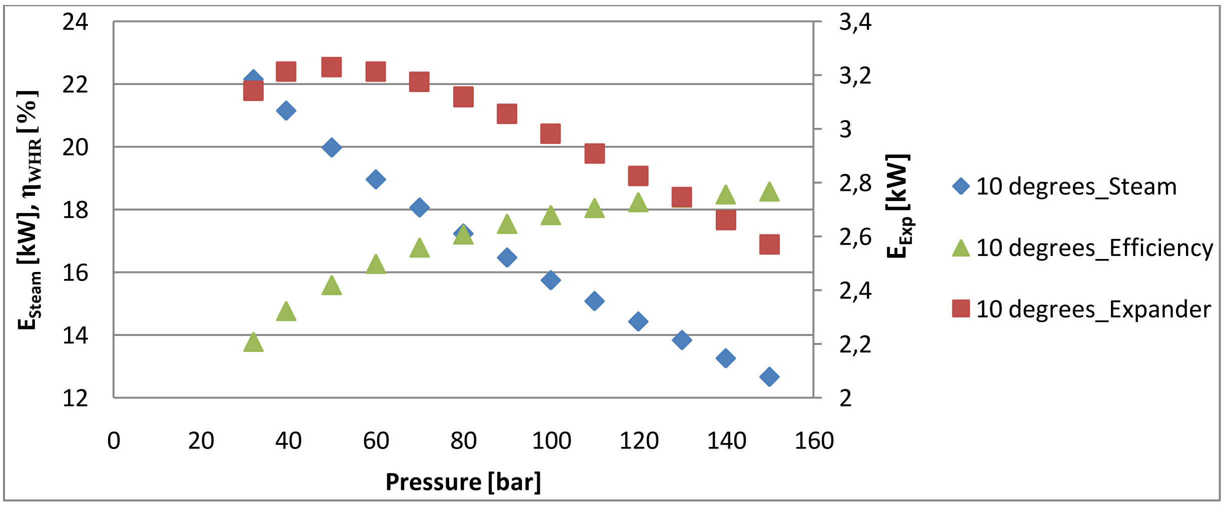

Figure 64. *The expander power, steam power and system efficiency for different pressures, point 12*

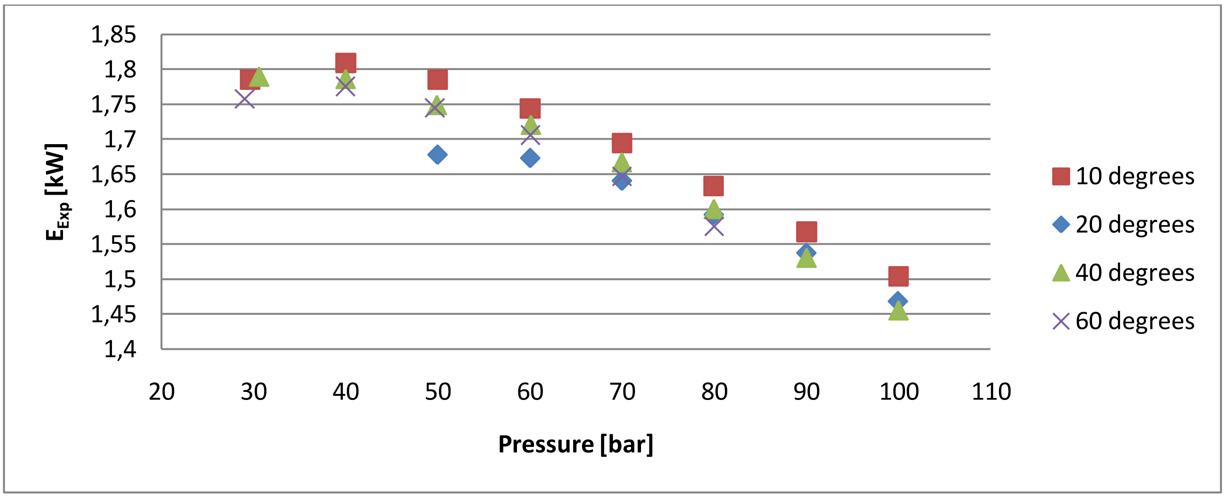

Figure 65. *Expander power for several steam pressures, point 3*

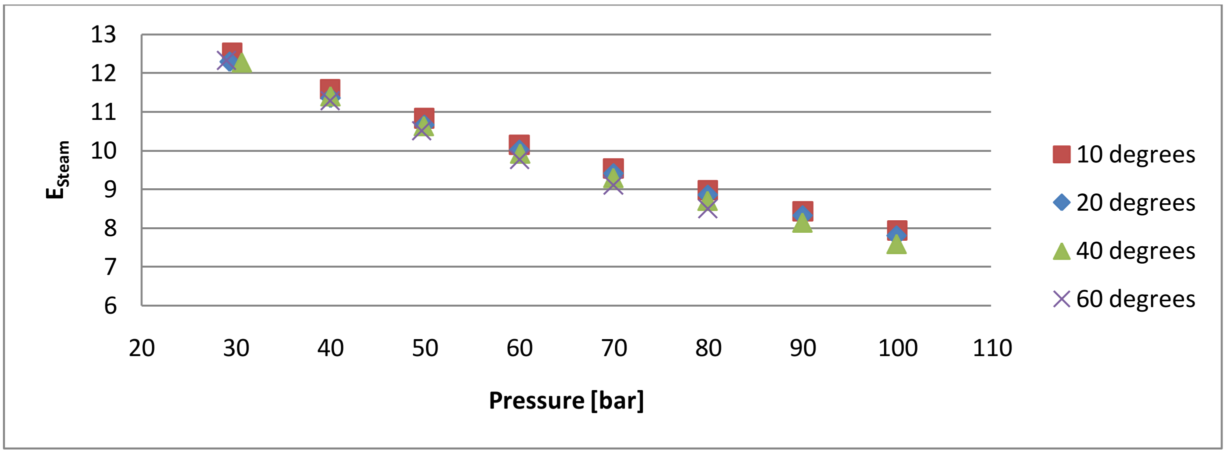

Figure 66. *Steam power for several steam pressures, point 3*

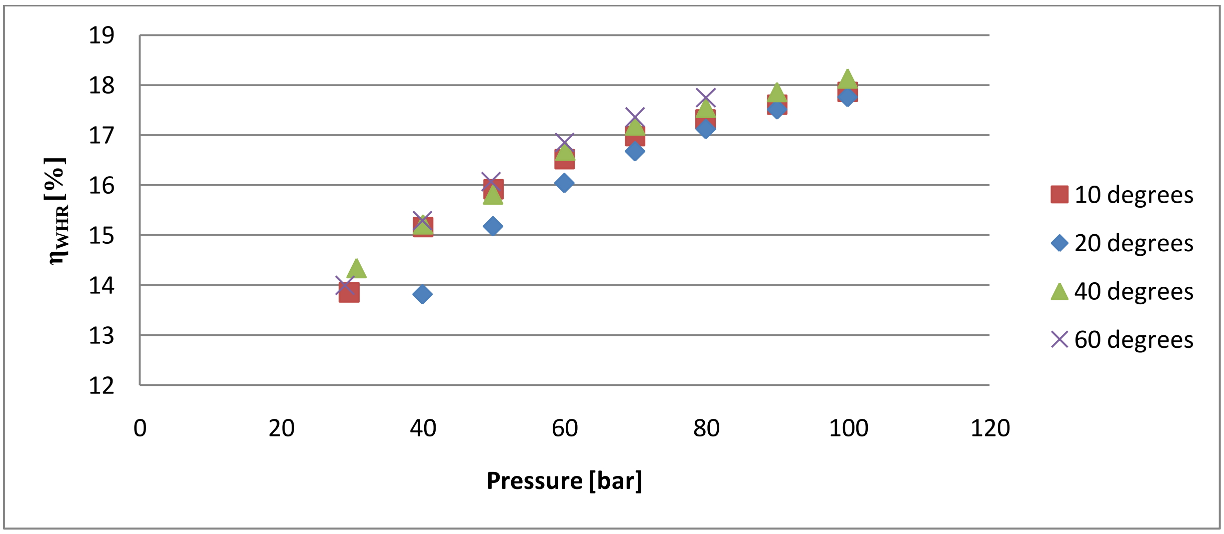

Figure 67. *The system efficiency for different steam pressures, point 3*

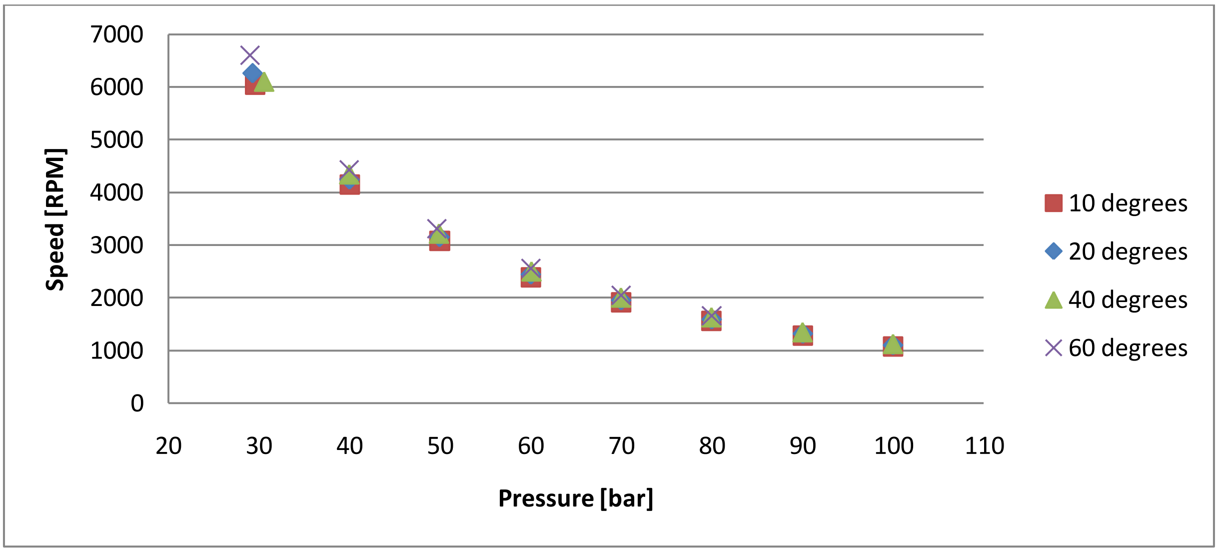

Figure 68. *The expanders speed for different steam pressures, point 3*

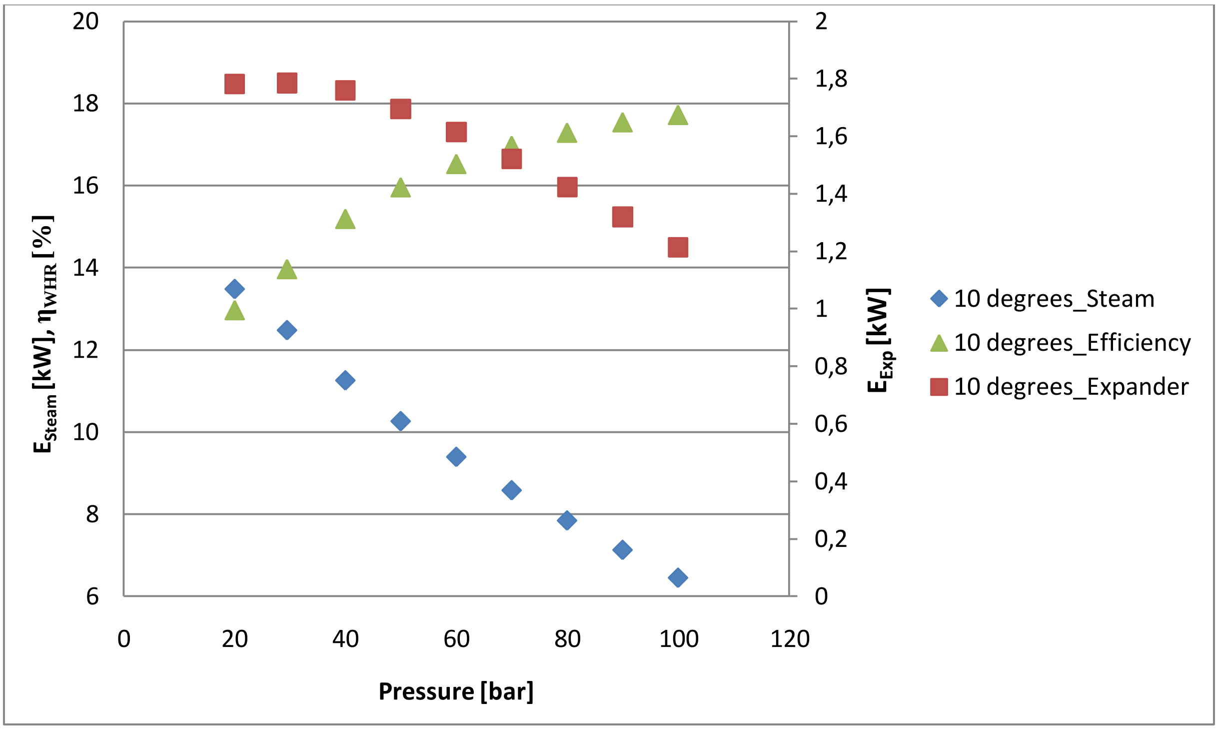

Figure 69. *The expander power, steam power and system efficiency for different pressures, point 13*

8.4 Evaporator calibration

The calibration of several points in the ESC cycle can be seen in figures 70-71.

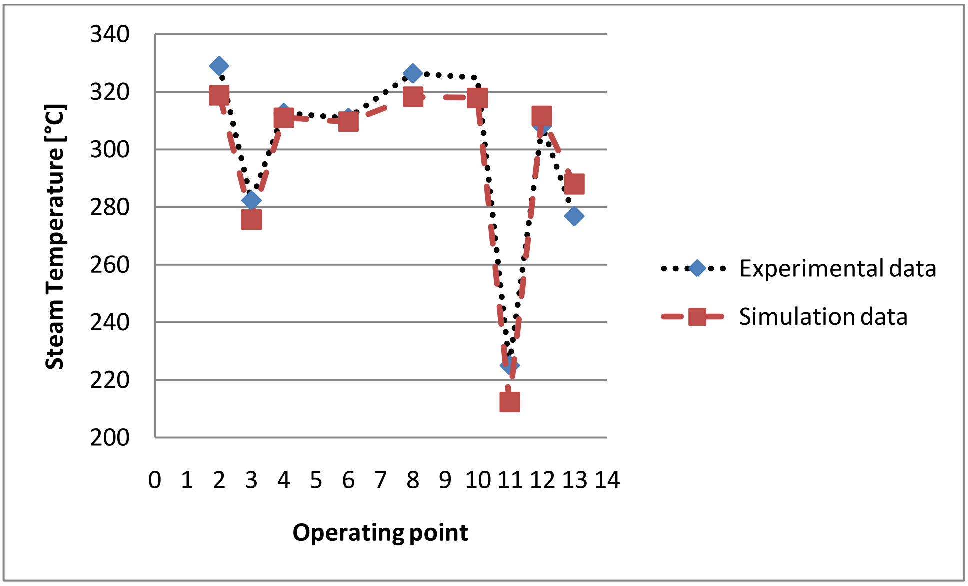

Figure 70. *The steam temperature for different load points, simulation and experimental data*

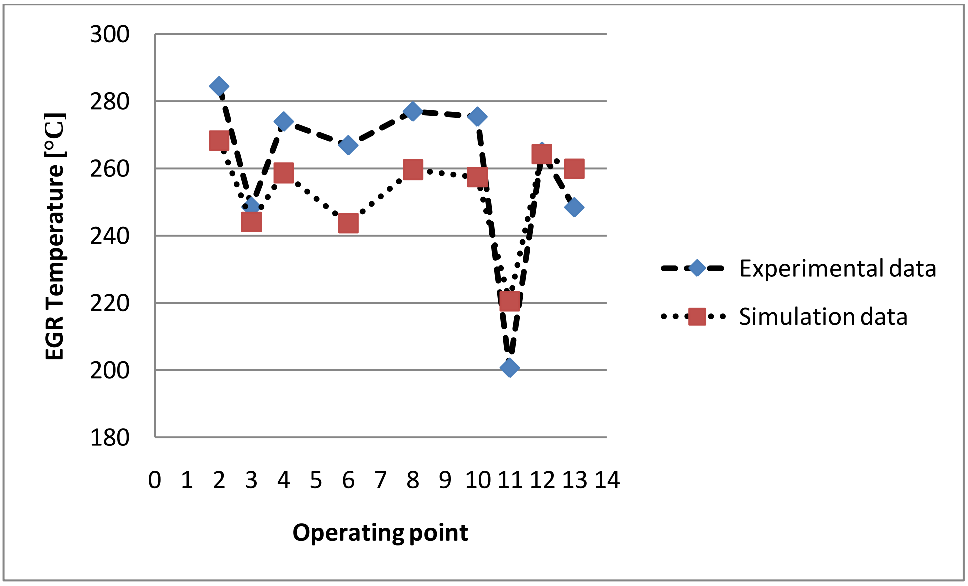

Figure 71. *The EGR outlet temperature for different load points, simulation and experimental data*