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PART LOAD BEHAVIOR OF A LARGE-SCALE AMMONIA HEAT PUMP SUPPLYING DISTRICT HEATING NETWORK

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Abstract

This master thesis work aims to display results from several different configurations and compressor types in order to have an idea in advance about the system response to these choices. The system taken as a reference is located in Nordhavn district and is composed of a heat pump, electric boilers, and thermal storage system.

To achieve high efficiency at part load it is essential to employ some capacity control systems. After a quick overview of those systems, some of them will be explained and included in the cycle. Several configurations are going to be taken into account; for each one, a different model will be used. Two different compressors type will be considered as they are commonly used in refrigeration and heat pump applications: reciprocating and screw-type compressors.

The basic heat pump model will be built with *EES* and then developed adding more detailed components. In particular, the compressor's performance will be evaluated especially at part load. This will be done using *GEA RTSelect* software with real compressor data.

Every compressor type has its relations to evaluate overall efficiency, especially regarding volumetric and isentropic efficiency. For the three all the necessary equations will be implemented into the model in order to simulate their actual performance at part load.

After the results for each case are displayed, there will be a discussion upon the various solutions pointing out advantages and disadvantages when choosing one or the other. This comparison should be made on the basis of different outcomes such as overall efficiency, control performance, reliability and application range.

Then, a definitive performance assessment will be made and the various cases will be investigated in order to detect their best possible application; there will be a discussion of the different systems. Furthermore, a quick sensitivity analysis will be carried out varying the DH forward temperature and showing the most relevant information.

Finally, all the results will be summarized in the conclusions as well as possible further investigation on the subject.

Preface

This master thesis has been developed within the European Erasmus Programme, in particular through an agreement between Università degli Studi di Padova and Denmark Technical University. This project work lasted for five months and has been carried out at the DTU campus in Lyngby. It is the final test to achieve the master degree in Energy Engineering and counts for 18 CFU at the University of Padova.

Acknowledgements

During these months I have received help from many people and therefore I would like to thank my supervisors Brian Elmegaard, Torben Ommen and in particular Wiebke Meesenburg for her patience and kindness, and my Italian supervisor Anna Stoppato.

Questa tesi è dedicata alla mia famiglia: ai miei genitori, che sono stati sempre di grande aiuto sia durante il mio percorso universitario, sia durante la mia permanenza in Danimarca, a Irene e Nicola, che mi sono stati d'esempio ognuno a suo modo. Vorrei inoltre ringraziare amici vecchi e nuovi per il loro supporto: una menzione particolare per Riccardo, Edoardo, Simone, Riccardo e Mattia. Tra le nuove persone incontrate in Danimarca meritano una citazione speciale Stefano, Elena, Giorgia Ludovica e Luca.

Contents

Nomenclature

Symbols

A_{cond}	Condenser Area
A_{evap}	Evaporator Area
$C_{p,w}$	Water Specific Heat
COP	Coefficient Of Performance
Δh_{is}	Isentropic Enthalpy Difference
Δh_{real}	Real Enthalpy Difference
Δp	Pressure Difference
η_{is}	Isentropic Efficiency
η_{vol}	Volumetric Efficiency
$\dot{m}_{ref}$	Refrigerant Mass Flow
$\dot{m}_{gw}$	Ground Water Mass Flow
$\dot{m}_{DH}$	District Heating Mass Flow
n	Rotational Speed
Q_{cond}	Condensation Heat
Q_{DH}	District Heating Heat
Q_{ev}	Evaporation Heat
ρ	Density
rpm	Revolutions Per Minute
$T_{gw,in}$	Ground Water Inlet Temperature
$T_{gw,out}$	Ground Water Outlet Temperature
$T_{DH,in}$	District Heating Water Inlet Temperature
$T_{DH,out}$	District Heating Water Outlet Temperature
V	Compressor Volume
V_{dot}	Compressor Displacement
$\dot{W}$	Electric Power

Abbreviations

BEP	Best Efficiency Point
CHP	Combined Heat and Power
COP	Coefficient Of Performance
DH	District Heating
EER	Energy Efficiency Ratio
EES	Engineering Equations Solver
ES	Energy Savings
EU	European Union
GHG	Greenhouse Gas
IE	Isentropic Efficiency
HEX	Heat Exchanger
HP	Heat Pump
HTC	Heat Transfer Coefficient
HFC	Hydrofluorocarbon
HFO	Hydrofluoroolefin
O&M	Operation and Maintenance
PR	Pressure Ratio
RES	Renewable Energy Sources
TEWI	Total Equivalent Warming Impact
VE	Volumetric Efficiency
VSD	Variable Speed Drive

1 Introduction

In the last years, the main problem to address in energy-related topics has always been reducing CO_2 emissions in order to contain global warming. Several international agreements have been made during the years. The current goal for European Union is to cut Green House Gas (GHG) emissions of 20% in 2020 and 40% in 2030 comparing to 1990s. Moreover, energy efficiency should be increased by 20% and renewable energy sources should reach 20% of total energy consumption by 2020. Both these values are set to be at 27% by 2030 [1].

A smart use of RES is very important to reach these targets, thus in Europe, many different systems are being developed. In particular, Denmark has two well-exploited sources such as biomass and wind. In 2014, wind power covered 39% of Danish electricity demand; with biomass and wind, Denmark claims to achieve independence from fossil fuel by 2050 [2]. For this, it is considered worldwide to be one of the leaders of energy transition. Copenhagen set the target to be CO_2-neutral by 2025. This target is planned to be achieved by using less energy, switching to green energy production and rethinking mobility [2].

Among other goals, the Danish strategy comprehends the coal phase-out, i.e. converting all power stations from fossil fuels to biomass by 2035. The CHP plants will all be fed with biomass and provide both electricity and heating. The heat is distributed by the DH network, which already covers almost all Copenhagen area. Different scenarios were analysed separately; wind, biomass, and hydrogen have been considered in order to point out different strategies [2].

In all these cases, heating plays a very important role because of its large percentage in the energy consumptions. Therefore, it is very important to consider solutions that can reduce energy consumptions and CO_2 emissions. In particular, heat pumps have proven to be one of the most efficient systems among others. Bach et al. [3] pointed out that these systems can be operated very efficiently when integrated with CHP and DH, especially when they are connected to the distribution line, rather than to the transmission line.

HPs are foreseen to be a possible way to stabilize intermittent energy sources such as wind and solar power, so these systems will have a bright future in the years to come. The operating strategy would follow the electricity prices: when wind power is available cheaply, it is used for electricity and as input for the heat pumps; when wind power generation is scarce, prices are higher so it is wiser to shut down HPs; in that case CHPs are operated along with heat from heat storage [4].

1.1 Problem Statement

One of the main issues to consider when designing a heat pump is its part-load behavior as it operates in very different conditions throughout the year. This is due to the fact that the power size must be chosen to supply the required heat even in the least favorable conditions, i.e. when the outside temperature is at its lowest value. It is clear that this extreme situation is likely to happen only a few times during the considered period. For this reason, all the components are actually oversized for most of the time and they have to work in part-load conditions.

Besides condensation and evaporation temperature, COP is mainly affected by compressor performance. Hereby, it should be optimized accordingly. What makes the difference between good and bad performance is to contain required electric power, which is mainly absorbed by the compressor (given that pumps usually absorb much less power). In order to do so, several capacity control methods are available: these systems allow the compressor to adapt the volume displacement or its speed to reduced power requirements. For example, when the load is 50%, capacity control makes possible to consume only 55-60% instead of 100% (with variable speed drive) [5] [6].

Heat source, forward temperature and outside temperature can consistently affect both cycle efficiency and heating demand, therefore different scenarios should and will be considered in the sensitivity analysis; in particular, various conditions will be analysed with different values of forward temperature besides the design point case. Only forward temperature will be varied because the heat source considered is groundwater and its temperature is practically steady during the year. These different scenarios will then be compared pointing out how different parameters such as COP, DH mass flow and condensation temperature vary.

1.2 Nordhavn District

Nordhavn district is situated in the northern part of Copenhagen and it will soon be transformed from an industrial area to a new residential and business area with room for more than 40000 new residents [7]. This district represents an opportunity to develop new areas in the city in a sustainable way. Many leading companies are involved in the project called *"EnergyLab Nordhavn"*. Every partner has its own duties to contributes in its area of expertise: those are for example buildings, utilities, and energy. The main goal is to create a lively urban area where all can enrich each other; all the main parts involved shall give their contribution to sustainable growth in that area. Small and large businesses, family, university and public institution are supposed to work together to make this possible [8].

Regarding energy supply, new solutions will be investigated in order to guarantee all the community's needs in a smart way. For example, exploiting as much as possible renewable sources is one of the main drives of the project. Main issues such as fluctuating power generation and supply-demand matching will be addressed. Moreover, new buildings' technologies and automation have a relevant part of energy management. The district is supposed to be a laboratory for new smart technologies and intelligent ways to develop and enrich a neighborhood without affecting inhabitants' quality of life or the environment [9].

FIGURE 1 - NORDHAVN DISTRICT FROM ABOVE **[7]**

1.3 Task

This master thesis is related to providing heating to some of the buildings of the cruise terminal in the north harbour of Copenhagen with large-scale heat pumps. This is necessary since the district is not fully covered by DH, therefore new heating solutions have to be considered. This project will be about the implementation of a heat pump based system with electric boiler and heat storage. In particular, the focus will be on the heat pump cycle and its performance at part load.

In this project, the heat pump cycle will be evaluated. The most suitable solution will be presented along with the choosing criteria. Various configurations will be considered with intense attention to part-load behavior and different capacity control systems.

One of the main problems to address is indeed to modulate mass flow when demand is not at its maximum. Achieving that would lead to high energy savings since part load operation in heat pump systems is very common. Without any kind of capacity control, the system would run always at maximum power and that energy would be wasted. Moreover, compressors running at lower speed could improve their performance and consequently achieve a higher cycle efficiency in terms of COP.

FIGURE 2 - COPENHAGEN NORTH HARBOR AREA FROM ABOVE **[9]**

2 Heat Pumps

In this chapter heat pump technology will be quickly presented with particular focus on its basic layout scheme and main components. Then, there will be an overview of HP range of application, especially in relation to DH networks, which is the case study in this project.

2.1 Thermodynamic Cycle and Components

Heat pumps are devices that transfer heat from a cold source to a hot source using electric power. The basic cycle is shown in Figure 3 and it is usually called *"inverse cycle".* The standard layout counts 4 components: compressor, condenser, evaporator and expansion valve.

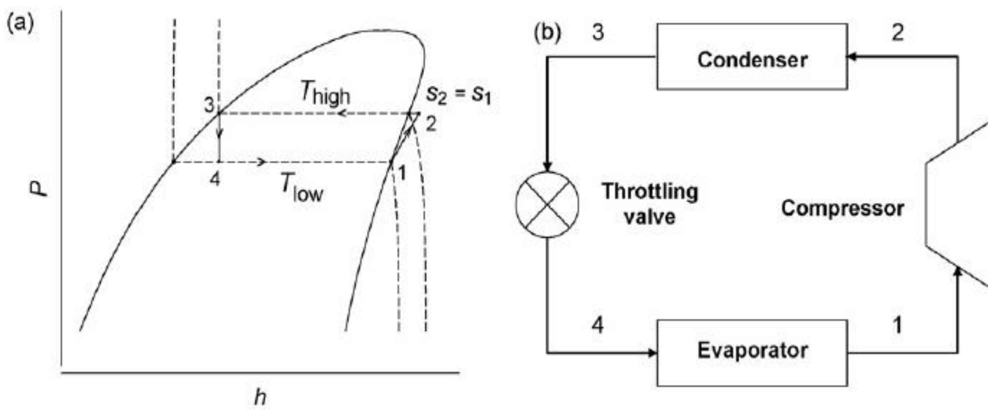


FIGURE 3 - THERMODYNAMIC CYCLE AND SYSTEM LAYOUT OF A SIMPLE HEAT PUMP **[10]**

The working fluid is called "*refrigerant*" and it can be used in heating and cooling mode both if the system is invertible. The most widely used are R134a, R32, R410A, R404A, R717 (NH_3) among others. In the last few years, interest in natural fluids as ammonia or CO_2 (R744) has grown because of the global warming issue. Common refrigerants have indeed a high GWP, so natural fluids are to be preferred. Ammonia, for example, has ODP equal to zero and good thermo-physical properties but it is toxic and flammable.

When operating a heat pump, COP (*Coefficient Of Performance*) is one of the most important parameters: achieving a high value means low energy consumption for the same output. This compares heat absorbed or released to the electric power required; in fact in an inverse cycle heat is the output, while electric power is the input. It is defined in equation 2.1.

$$COP = \frac{Q}{W} \tag{2.1}$$

where Q can be absorbed heat in refrigeration cycles or released heat in heat pumps applications and W is the electric power required. To be fair, in refrigeration it is more often used another parameter, which has the same features and is called EER (*Energy Efficiency Ratio*).
Moreover, COP can be used to compare different heat pumps. Common values range around 3-4 depending on the layout and the operating temperatures. It mainly depends on the outside temperature and on compressor's performance in the operating range. Besides COP, there is another parameter that is quite important in the process of evaluating an HP system; this is called SPF (Seasonal Performance Factor) and considers the global behavior throughout the whole year. In fact, it considers the global heat energy delivered and the electric energy required during the entire period and can be expressed as:

$$SPF = \frac{\Sigma Q}{\Sigma W} \tag{2.2}$$

To achieve a high SPF, it is not enough to have a good COP, but it is crucial to keep as low as possible different kind of losses such as idling, defrosting and heat losses. Furthermore, it is very important to maintain a low energy consumption of pumps and auxiliary machines [11]. As will be later explained in paragraph 5.1, this aspect assumes particular relevance at part load.

2.1.1 Compressor

The compressor takes the vapor from the evaporator and releases it to the condenser at a higher pressure. Pressure ratio is between the two levels in evaporation and condensation. Condensation and evaporation pressure are determined by several parameters needed to satisfy the demand such as source temperature, return temperature and heat load.

The compressor can be considered as the most important component because its performance heavily affects the overall efficiency. In particular, when the outside conditions vary the compressor must adapt by changing its pressure ratio. Every compressor has its own optimal pressure ratio, so when this changes efficiency drops. There are several types of compressors for the different operating conditions: piston type, rotary, screw, scroll and turbo compressors. These can operate in a different range of pressure ratio, volume flow, and power size [12].

2.1.2 Condenser

In the condenser, the fluid changes phase from vapor to liquid through *desuperheating*, condensation and *subcooling*. The heat released can be exploited to heat a secondary fluid as water or air and cover the heating demand. Different types are available nowadays: air-cooled, water-cooled and evaporative condensers. The water-cooled could be plate-type condensers or shell-and-tube; air-cooled is usually with finned coils in natural or forced convection [12] [13]. In order to calculate the heat transfer coefficient and then the surface needed, many empirical correlations are available in the literature. These equations apply to different flow types (*stratified, annular, etc*) and condensation models (*filmwise, dropwise*).

2.1.1 Evaporator

In the evaporator, external heat is transferred from the heat source to the working fluid: this changes its state from liquid to vapor and in most cases, it is also superheated. This last step is necessary to avoid liquid droplets in the compressor inlet and get a proper compression process. The evaporators are divided into air coolers and liquid coolers based on the heat source. Air coolers are finned batteries working in forced convection. The main configurations of the liquid coolers are the dry expansion, in which the refrigerant flows inside the tubes, and flooded types with evaporation developing outside them [13]. Depending on the heat transfer mechanism, different relations can be found in the literature to determine the overall heat transfer coefficient. As well as the condensers, evaporators are sized using this coefficient.

2.1.2 Expansion Valve

The expansion valve has basically two functions: maintaining the pressure difference and regulating the mass flow. Among several types, capillary tubes and thermostatic valves are the ones which are mainly used. For large applications like the case study, floating valves are quite common [12] [13]. This valve brings the fluid from condensation pressure to evaporation pressure and the process can be assumed isenthalpic (neglecting kinetic and potential energy variations).

FIGURE 4 - THERMOSTATIC VALVE **[14]**

2.2 Heat Pumps Applications

Heat pumps are widely used in different sectors and to supply very different demands. They can be used in residential buildings or in large-scale applications. The main advantage is that heat pumps are able to provide heat when the electricity prices are low and store energy thanks to thermal storage for later use.

Heat pumps can stabilize power markets and produce energy at a very low price during off-peak hours. are also suitable to balance power generation from renewable sources like wind and solar power. Therefore, in this project, the attention is focused on the use of heat pumps into district heating.

2.2.1 *Power Balancing*

Since wind and solar power are growing at a very fast rate, it is very important to balance these intermittent energy sources. Especially in northern European countries such as Sweden, Denmark, and the Baltic States, this problem has already been addressed investigating several possibilities. In Denmark, for example, wind power penetration is very high and to manage this power generation properly a well-planned strategy is mandatory. Among other solutions, power-to-heat installations in district heating show comforting results. In fact, these systems are able to provide great flexibility as they can operate according to the electricity price evolution. When surplus occurs, it can be used to generate heat power and store it until needed. This can often happen due to strong and sudden variations of wind power generation and in general, it is a main feature of most renewable energy sources. While solar power production can be estimated evaluating sunny hours during the day and meteorological conditions, wind power is in most cases unpredictable. Therefore, for countries which heavily relies on wind, some kind of solution to balance and manage this floating generation is strongly advised [15].

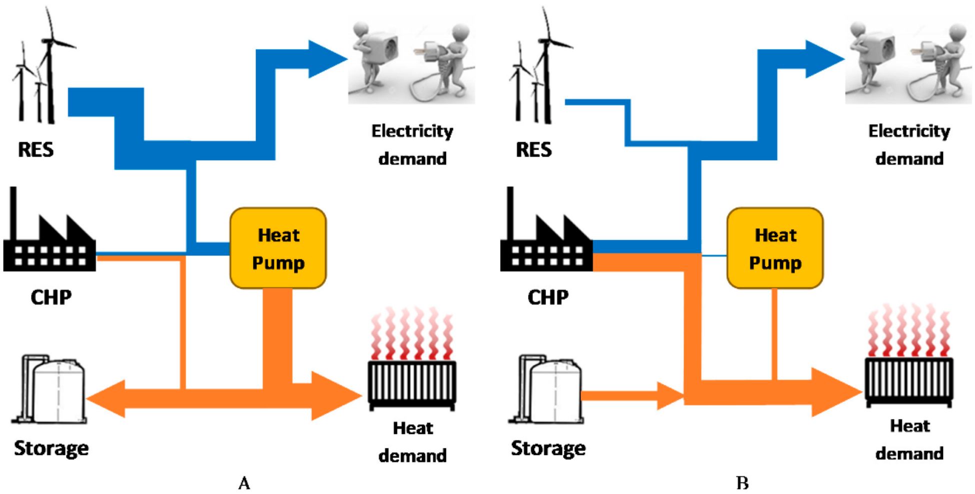


FIGURE 5 - OPERATIONAL STRATEGY DURING HIGH (A) AND LOW RES POWER GENERATION (B) **[16]**

2.2.2 *Power-to-Heat Systems*

To achieve the target of 100% renewable energy that many countries such as Denmark have set to fulfill by 2050, wind, solar and biomass energy would have to occupy a growing share in energy generation. In this scenario, CHP plants that nowadays are operated to supply both heat and electricity should decrease their production, especially the electricity share. In order to do so, the heat share has to be obtained from other sources: in particular, one solution is deploying power-to-heat systems, which convert electricity into heat. In this way, both the heat would be supplied in a more flexible way and electricity surplus from growing renewable energy sources could be used cost-effectively and managed properly [3].

Along with CHP plants, power-to heat systems can be operated supplying the district heating network. When coupled with CHP, heat power is obtained from heat recovery, which is one of the most important principles for DH networks. On the other hand, in association with large-scale heat pumps, DH is able to supply heat from electricity surplus and that is another smart way to manage the whole network. In Russia, long heating season and heavy wind power generation in the same period makes this solution very interesting; in USA, Denmark and Germany, this configuration with heat pumps and energy storage has been the subject to several studies. One of the reasons it has not been developed as it could have been is that various forward temperatures are required in different countries; the higher the temperature the less competitive the system would be. This also depends on how reliable the DH network is, meaning that low heat losses would require a lower forward temperature. Of course, it is strongly affected by outside temperature throughout the year.

Power-to-heat solutions usually have two main configurations: it could either be large-scale in existing plants or small-scale close to end users. The main focus of this project is on the first ones and some of the advantages are presented next. Large-scale configurations have greater balancing capacity, the possibility of large heat storage, better exploitation of electricity surplus and the possibility to generate heat from very different heat sources. In particular, they can exploit sources not commonly available for domestic applications; these could be sea water, sewage water or deep geothermic. On the other hand, small installations are easier to handle and supply heat requirements without needing a large distribution network. They also work with an almost steady load and therefore achieve high efficiency at any time. These usually find application in suburbs or low densely populated areas, whereas large-scale are mostly used in developed areas and large cities [15].

2.2.3 Integration in District Heating

District heating can provide heat to very large areas and is widely spread in Europe. Copenhagen, for example, has a district heating network that is able to cover 98% of the internal heat demand [4]. Through district heating, it is possible to generate power with few power plants and then dispatch it where is needed. The advantage is that high-efficiency CHP plants can be operated only when costs are low, while during windy days electricity comes from wind power and can also be used as an input for the heat pumps. In this way, overall generation costs are maintained very low. HPs have proven to be well-suited for baseload coverage but also for low loads and domestic hot water production [15].

Especially in densely populated areas, DH is very suitable, while in rural contexts and with low fuel prices it should not be considered as a competitive solution. DH is often operated with natural gas or renewable combustibles. In Europe gas is widely used because of the existing grids, but DH is still underdeveloped for buildings; only in Iceland, Denmark, Sweden, Finland, Baltic states, Poland and Russia it is long-established in urban areas [11].

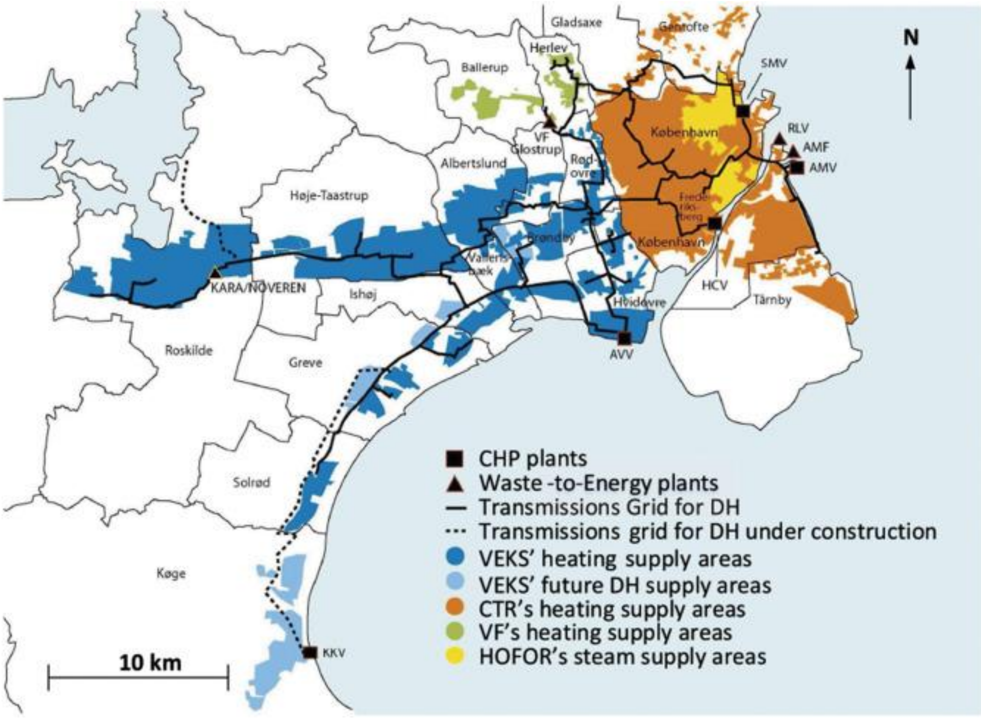


FIGURE 6 - DISTRICT HEATING IN GREATER COPENHAGEN **[3]**

District heating plays an important role when it is used together with renewable energy sources such as solar collectors, geothermal heat, and biofuels. Since DH network is already developed to work with CHP plants, it is possible to integrate the system also with solar and wind power. This process would lead to fewer CO_2 emissions and low generation costs. As can be seen in Figure 7, use of recycled heat from renewable sources has grown consistently in the past few years. Moreover, excess power generation could be used to store thermal energy thanks to the heat pumps. Thermal storages cost a lot less than electricity storage, so HP provide great advantages [4].

Several studies have been carried out upon integrating heat pumps with district heating in Denmark as in many other countries. These analyses showed that heat pumps can help reducing GHG emissions and operate the entire network more efficiently. It is known that heat pumps gain in efficiency when they operate with high-temperature source and low-temperature sink. Even if HPs are able to generate stable and efficient heat, they still need a reliable heat source, preferably with very low temperature variations. Given that, it is important to map all the available sources such as waste heat, sewage water, seawater, and others. Forward temperature has less freedom since it is usually around the same temperature level for residential heating purpose and domestic hot water generation [3] [17]. A recent study by Ommen et al. [18] has shown that increasing heat pumps capacity to the optimal size would guarantee a 1.6% saving in fuel consumption.

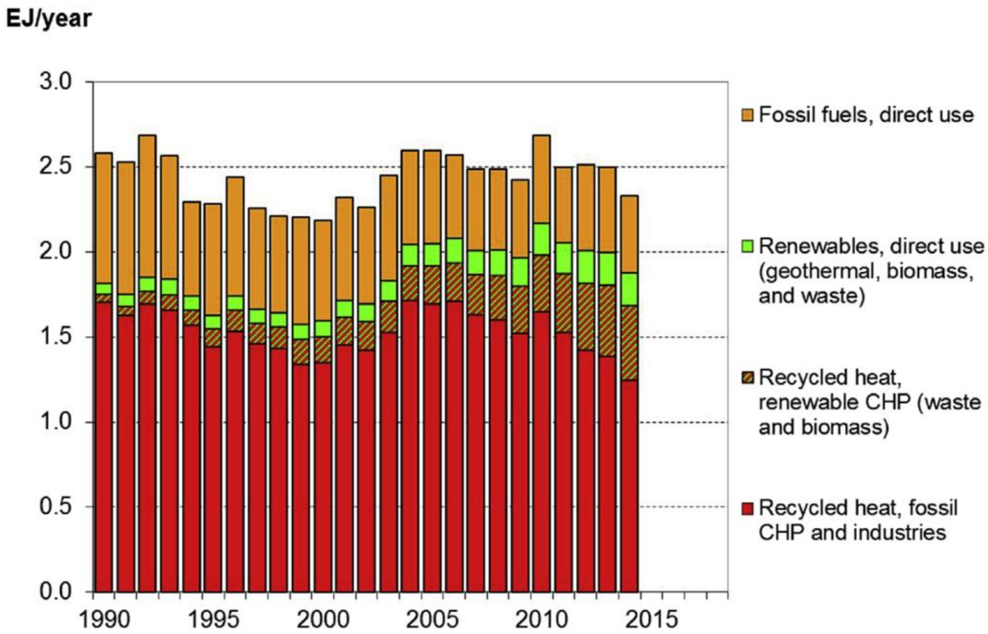


FIGURE 7 - HEAT SUPPLIED IN EU IN DH NETWORKS **[4]**

3 Refrigerants

In the last few years in refrigeration and air conditioning applications, new interest has been developed in lowering global warming impact of commonly used refrigerants. Many efforts have been put in testing innovative synthetic fluids such as HFOs [19] and developing natural fluid cycles; in particular, NH_3 and CO_2 are mainly employed in these applications [20]. Figure 8 shows the evolution of refrigerants used from the very first years: from natural fluids, there was a transition to CFCs and HCFCs that were more efficient and safe. Their main drawback was ozone layer-damaging, so it was decided to move on to 3rd generation fluids known as HFCs. As global warming started to become a major issue, large use of HFCs began to be a problem and they are right now in the process of being replaced by natural and low GWP fluids.

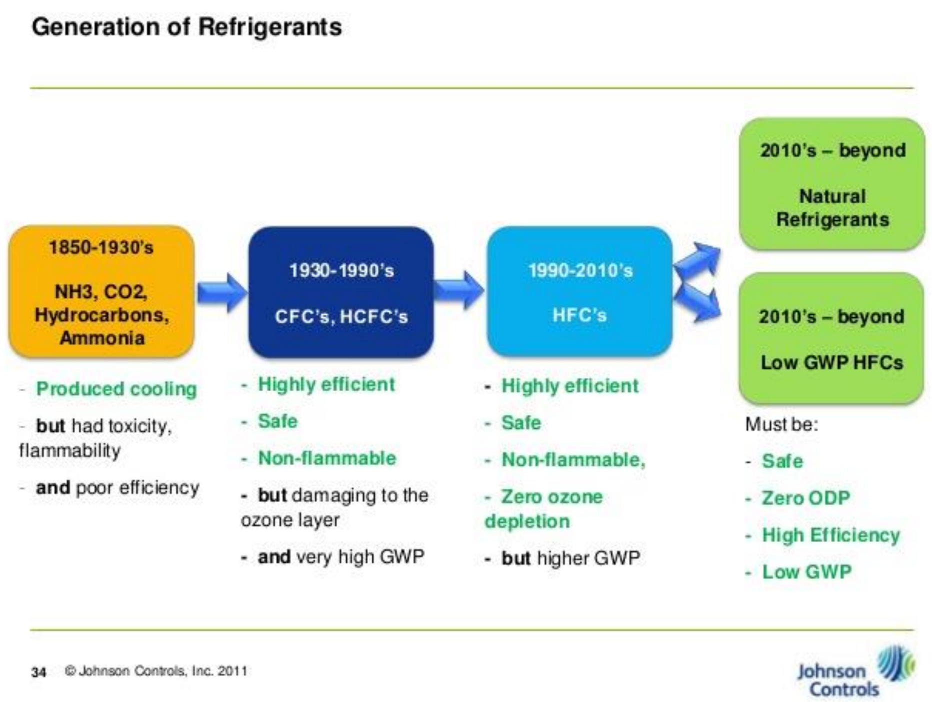


FIGURE 8 - EVOLUTION OF EMPLOYED REFRIGERANTS **[21]**

The main problem is to achieve the same cycle efficiency using fluids with lower GWP. R134a, for example, is widely used in a different range of applications due to its high efficiency but it has a relevant impact on the environment if leaks occur. On the other hand, using a low GWP fluid with low cycle efficiency does not solve the problem because higher electric power would be required for the same capacity. That would lead to higher CO_2 emissions depending on how that energy had been produced.

What has been said can be summarized showing the TEWI index (Total Equivalent Warming Impact) calculated as follows:

$$TEWI = X * GWP + \alpha_{CO2} * L * E \tag{3.1}$$

where X [kg] is the released mass of fluid, GWP is global warming potential, α_{CO2} [kg CO_2/kJ] is the released CO_2 per energy unit, L [years] is life expectancy and E is energy per time unit [kJ/year]. The first term is called "*direct effect*" and the second is the *"indirect effect"* since it does not depend on actual released fluid but it is mainly affected by cycle efficiency [22].

3.1 HFC

In refrigeration and heat pumps the most commonly used refrigerants are R134a, R407C, R410A, and R404A; they all belong to the HFC family and they have high efficiency but at the same time high GWP values. During the years they have taken the place of R22 in many applications. Right now, switching from these fluids to low impact fluids seems to be necessary in order to contain global warming; HFOs have been identified as the possible solution as long as their efficiency will be improved in the next years. These will be presented in the next paragraph.

While R134a is a pure fluid, the others are all mixtures. As a matter of fact, R407C contains a percentage of R134a and it has very similar properties to R22. For this reason, R407C had been detected as a very good choice to replace R22, which was definitively banned, at least in industrialized countries. R410A can achieve same or better performance than R407C, but they are both supposed to be replaced in the future possibly by R32. This has 50% of R410A and combines good efficiency with low GWP, being one of the best possible long-term solutions; its only drawback is related to flammability. At last, R404A reaches quite high performance, better than most HFCs but it also presents a quite relevant impact on the environment and should face a quick phase-out in the near future [14].

As it was said, the possible solutions to the global warming issue are represented by R32, HFO and natural fluids such as ammonia. These last two will be presented next.

3.2 HFO

Hydrofluoroolefins, often shortened to HFO, belong to the 4th generation of refrigerants and they have many advantages. Some of them have zero ODP, very low GWP, no or low flammability, efficiency, availability and relatively low costs. So they can combine good performance with sustainability, safety, and cost-effectiveness. R1234yf and R1234ze are currently in use while others are under development. All of them are likely to have a bright future in refrigerating and heating sector as well as in automotive air-conditioning and many companies are focusing on achieving the same efficiency as their predecessors. For this reason, the performance analysis is often made in comparison with R134a [15] [23].

In Figure 9 below it is possible to notice that R1234yf has basically the same curve as R134a and therefore it could be a possible drop-in replacement for it. Instead, R1234ze is characterised by lower vapor pressures than the other two. However, it is compatible with most materials and could be employed for other applications. It is classified as mildly flammable but it actually produces a flame only for temperature higher than 30°C [24].

R1234yf has ultra-low GWP and very similar properties to R134a; for this reason, it is the number one solution for its replacement in the years to come. Very few layout and design differences would have to be made and an important saving in GHG emissions would be achieved. Other than system efficiency, its only drawback is that it is mildly flammable, but with proper safety procedures, it can be handled avoiding any possible risks. However, for this reason, the machinery can be classified able to work with both R134a and R1234yf only if they meet precise safety requirements [25].

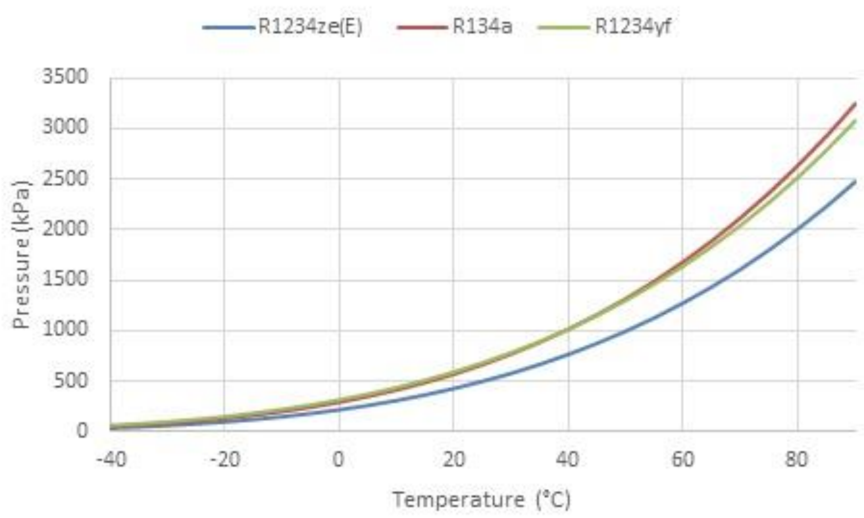


FIGURE 9 - PRESSURE-TEMPERATURE CURVE FOR HFO **[24]**

3.3 Ammonia

Engineering and technology involving ammonia are long established and the problems regarding its high toxicity and flammability have been handled with a satisfactory outcome. Unfortunately, ammonia is very corrosive and it cannot be used with copper, so only open-type screw and reciprocating compressors can be employed safely. However, when leaks occur, ammonia has the unique property of being less dense than air so it can easily reach the atmosphere without harming anyone. Moreover, it has a characteristic odor making it easy to detect when it is released [14]. NH3 is also seen as a possible replacement for 3rd generation refrigerants. Given that, in this thesis ammonia will play an important role and will be used as the main working fluid. Nonetheless, gathering some results using HFO fluids would have been interesting especially in order to compare them to ammonia, but it has not been possible due to lack of time and data. As it has been said before, HFO and natural fluids will probably replace HFC in the years to come.

Ammonia has been chosen as working fluid because it is an industrial standard and can work with high efficiency without causing harm in terms of CO_2 (GWP = 0). Different large size compressors are available for NH3 but unfortunately, it works with relatively high pressure, therefore all the other components must withstand that pressure. Moreover, since district heating temperature could require higher temperature depending on the circumstances, the pressure would increase as well. All these considerations point out that high-pressure components are required and their availability is not always granted [26].

4 Compressors

Refrigeration compressors are machines that reduce the volume of a gas increasing its pressure (except for dynamic compressors, whose mechanism will be better explained in their dedicated paragraph); they are used in refrigeration and heat pump cycles to move superheated gas from the evaporator to the condenser. They can be divided into open, hermetic and semi-hermetic.

In open compressors the device and the electric motor are separated; this configuration is less efficient due to mechanical losses but is easier to cool down (commonly with ambient air), more reliable and easy to maintain.

Hermetic and semi-hermetic have the electric motor integrated and it is not possible to access them directly. These are more efficient but if the motor drive should fail the entire machine would have to be replaced. The electric motor is cooled by the refrigerant itself affecting the thermodynamic cycle [27].

4.1 Compressor Types

Compressors can also be split up into positive displacement types and dynamic. The main focus of this project will be on the first ones. Those are divided on the base of their construction features as follows: piston, rotary (rotary vanes and rolling piston), scroll and screw compressors. Obviously, the compression process has different characteristics for each type.

Ammonia is strongly corrosive to copper, therefore, open compressors are mainly used in order to preserve the electric motor. When operating a system with ammonia as working fluid, piston, screw and scroll compressors are widely employed.

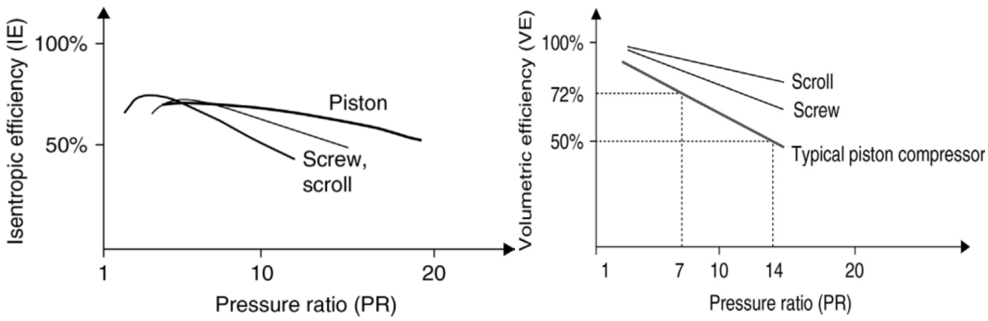


FIGURE 10 - IE AND VE FOR DIFFERENT COMPRESSOR TYPES **[13]**

4.1.1 Reciprocating Compressors

In a reciprocating compressor, gas is delivered to the high-pressure level thanks to a piston that moves alternatingly. The presence of valves for suction and discharge makes impossible to exploit the entire available volume. In addition to that, the unused volume expands during the compression process. For those reasons it is defined a theoretical volumetric efficiency that considers the effects of this unexploited volume usually called *"dead volume".* This efficiency can be estimated through the equation below:

$$\eta_{vol,th} = 1 - \frac{V_0}{V} * [(\left(\frac{p1}{p2}\right)^{\frac{1}{k}} - 1)] \tag{4.1}$$

where V_0 is the dead space, V is the swept volume, $\left(\frac{p1}{p2}\right)$ is the pressure ratio and k is the specific heat ratio.

Real volumetric efficiency is lower than the theoretical since there are also pressure losses, heat exchanges, and gas leaks. This efficiency can be calculated by different equations; one among others is Pierre equation which is reported below and it provides achievable volumetric efficiency for "good" reciprocating compressors [28]:

$$\eta_{vol} = k_1 * \left(1 + k_s * \frac{t_{2k} - 18}{100}\right) * \exp(k_2 * \frac{p1}{p2}) \tag{4.2}$$

where k_1, k_s, k_2 are coefficients given for different fluids: for ammonia those are 1.02, 0 and 0.063 respectively; t_{2k} is the inlet temperature at compressor suction. Volumetric efficiency values obtained through this equation are displayed in Figure 11.

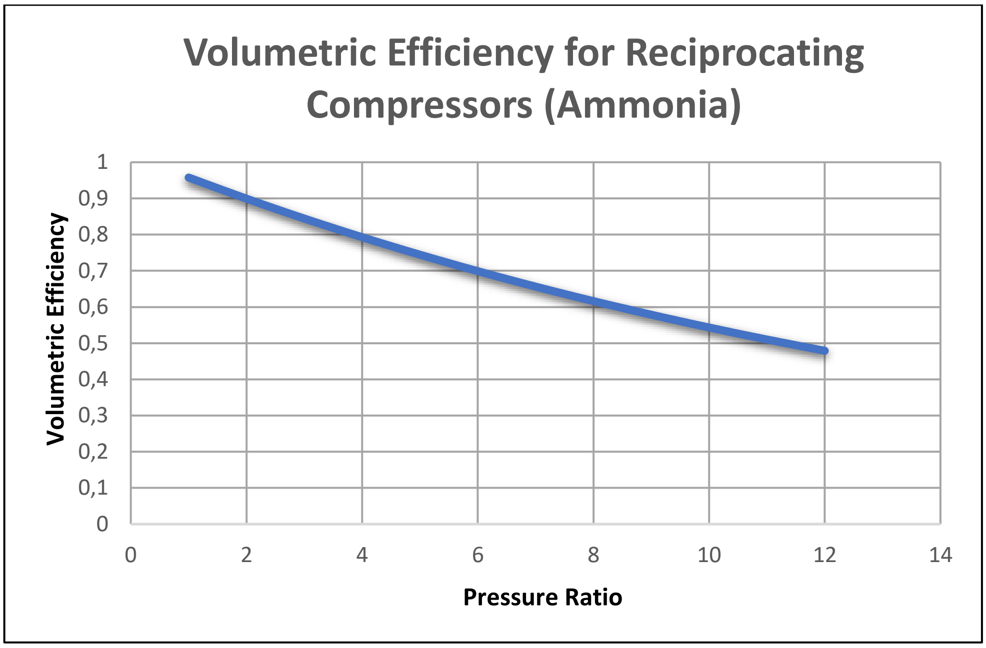


FIGURE 11 - VOLUMETRIC EFFICIENCY USING AMMONIA ACCORDING TO PIERRE

Another important parameter is isentropic efficiency, that summarises all the possible losses in the compression process such as friction, heat exchanges, and fluid leaks. This efficiency is by definition the ratio between isentropic and real work. It mainly depends on pressure ratio, but also on condensation temperature and superheating. In the particular case of ammonia, heat exchanges losses are way higher than those due to friction; for synthetic fluids, it is the opposite [29].

FIGURE 12 - OPEN-TYPE RECIPROCATING COMPRESSOR **[12]**

4.1.2 Screw Compressors

In these type gas enters at the suction side and is compressed as the screw rotates; the discharge happens when the screws end and gas is released at high pressure.

By design screw compressors have several advantages: reliability, low costs, fewer moving parts, less vibration and less refrigerant loss. All these benefits are achieved thanks to a more continuous process compared to a reciprocating compressor. This fact leads also to a long expected lifespan [12].

FIGURE 13 - SCREW-TYPE COMPRESSOR **[30]**

These devices are characterized by a built-in volume ratio, which is the ratio between volume at the inlet and at discharge. It is calculated as:

$$v_i = \frac{V_1}{V_2} \tag{4.3}$$

V_1 is the gas volume at the inlet, while V_2 is its volume when discharge port opens. This parameter strongly influences overall efficiency: for each volume ratio maximum, isentropic efficiency is obtained at different pressure ratios as can be seen in Figure 14 [29].

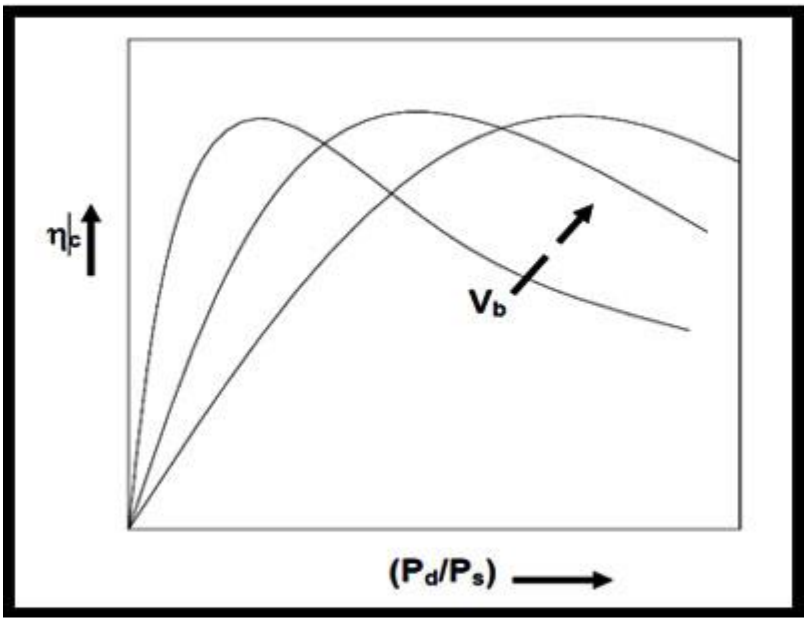


FIGURE 14 - INFLUENCE OF PRESSURE RATIO AND BUILT-IN VOLUME RATIO ON EFFICIENCY **[31]**

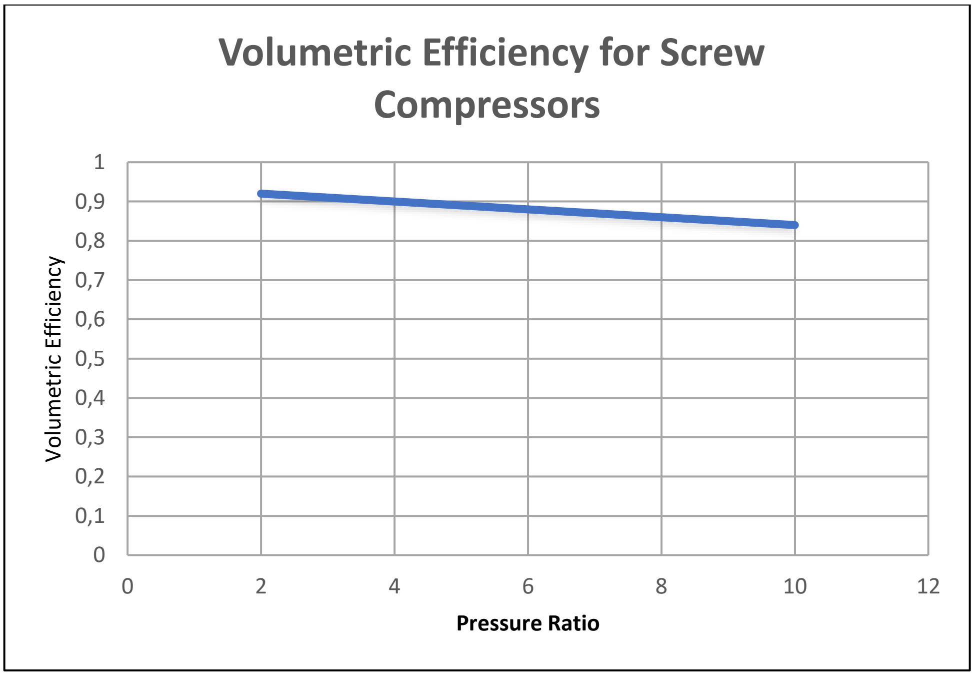


FIGURE 15 - VOLUMETRIC EFFICIENCY FOR SCREW COMPRESSORS (V_i=2.6)

As for volumetric efficiency, screw compressors are able to achieve very high values and in most cases way higher than reciprocating. This is mainly due to a more continuous compression process with very few moving parts. As well as IE, VE also depends on built-in volume ratio. An example of volumetric efficiency trend is reported below for $v_i = 2.6$. This equation has been taken from the book *“Refrigerating Engineering” by E.Granryd* and will be used for the simulation later.

4.1.3 Scroll Compressors

Scroll compressors are basically composed of two spiral-shaped components: one is a fixed scroll and the other is an orbiting scroll. As the screw compressor, this type has a built-in volume ratio and isentropic efficiency curve is very similar. On the other hand, in scroll compressors, there is no direct path between suction and discharge (which is in the core of the spiral) so gas leaks are lower than in screw-types; obviously, this leads to a higher achievable volumetric efficiency. These compressors are mostly hermetic [13].

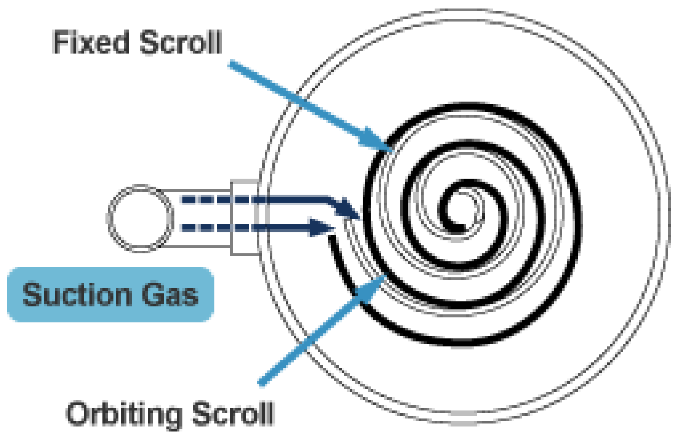


FIGURE 16 - SCHEME OF SCROLL COMPRESSOR **[32]**

Scroll compressors are often used in low power applications (1-50 kW), for that reason they will not be applied to the model in this case study; what is going to be analysed is a 800 kW system so it is out of range for scroll; nevertheless, they represent an efficient solution in residential units for example. As can be seen below, scroll have very good VE: above 90% for PR lower than 5; IE is also quite high even if it drops quite fast for high pressure ratio. Therefore, scroll compressor should be used when only low pressure ratio is required, which is often the case in their application range.

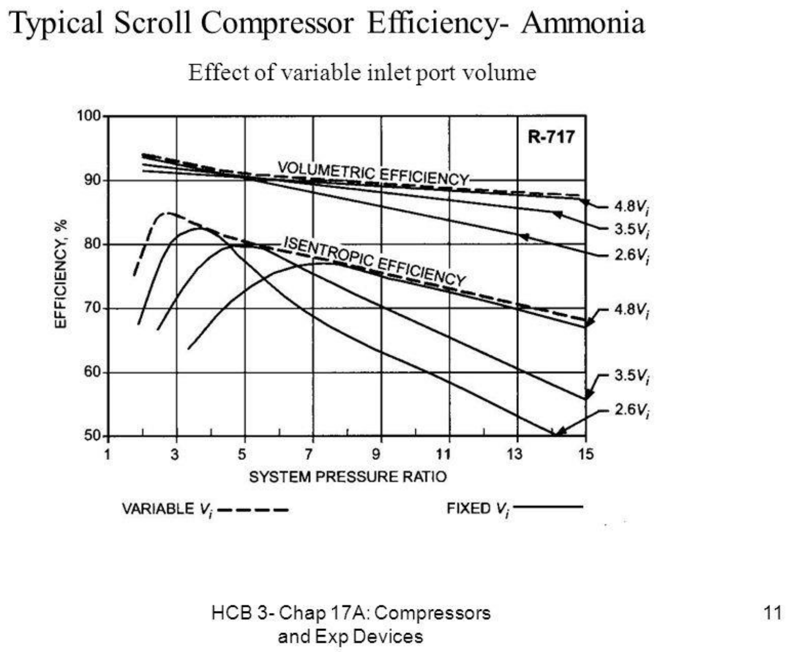


FIGURE 17 - SCROLL COMPRESSOR EFFICIENCY FOR AMMONIA **[33]**

4.1.4 Dynamic Compressors

Dynamic compressors operate with a different concept: while positive displacement works decreasing the volume, centrifugal increase pressure through a continuous exchange of angular momentum between the rotating part and the fluid. Where these compressor types find application is large scale application with power size ranging from 300 kW up to 20 MW. Especially with multi-stage cycle centrifugal compressors have very good performance and it is also possible to have two-stage compression using the same turbine. Furthermore, they are recommended when only low-pressure ratios are required; in those case, isentropic efficiency can reach around 80%.

When using ammonia as working fluid these are not very suited since corrosion phenomena are likely to occur with copper. They often work with limited pressure difference and their efficiency is commonly low when compared to other types. For these reasons, in this project, the cycle will be evaluated for screw and piston compressors only [12].

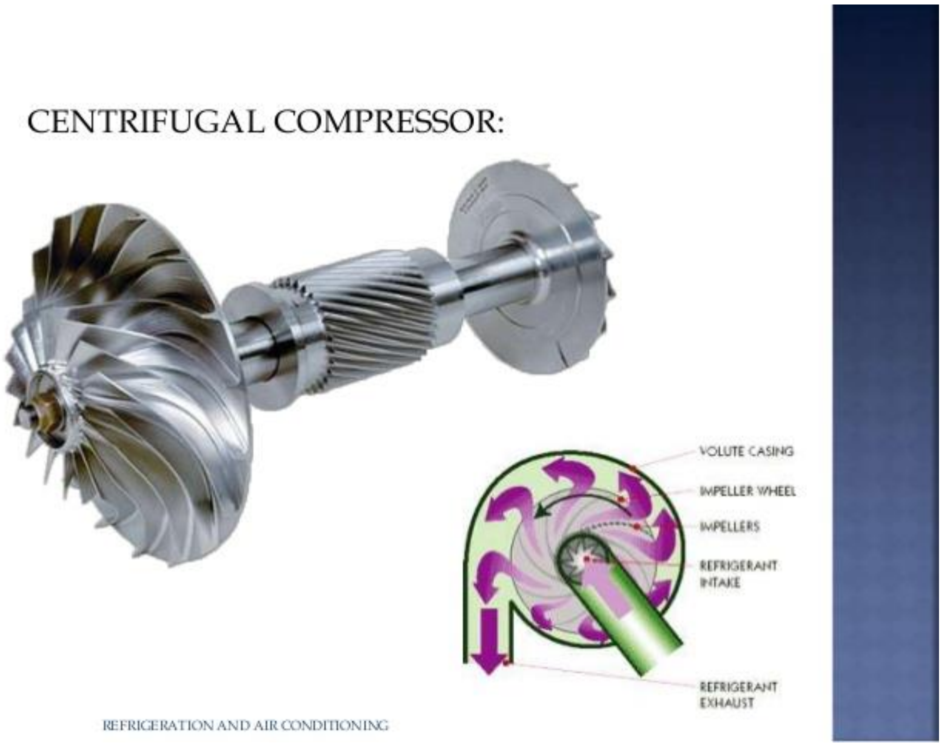


FIGURE 18 - CENTRIFUGAL COMPRESSOR OPERATION **[34]**

4.2 Capacity Control Systems

When a heat pump system is designed, the maximum load is considered in order to size the various components. However, when it is operated full load hours are usually a small fraction of the total amount. Throughout the year the heat pump will have to deal with changing parameters such as outside temperature or customers' needs. It should be obvious that in winter the heating load will be close to the maximum, while in other seasons it can decrease consistently. Therefore, it is crucial to implement some kind of control that can actually meet the load requirements during different periods [35].

That is exactly the purpose of capacity control systems. When the system is operated at part-load, the compressor should be able to vary the mass flow, possibly without affecting the overall efficiency of the system. Several systems are available nowadays; some of them will be presented below with a particular focus on the performance in different conditions.

As reduced thermal power may allow lower forward temperature, the temperature difference between evaporation and condensation will decrease as well. This leads to an improvement in efficiency due to the lower electric power required. Apart from this, there are several more advantages that come with modulating compressor capacity: some of them are fewer starts, longer operating times, possibility to *overrev* (i.e. operating above design speed) the compressor and thus increase its capacity beyond its limits for short periods [36]. Depending on the compressor type different methods are available and an overview is shown below in Figure 19. However, some of them can be adopted regardless of the compressor type.

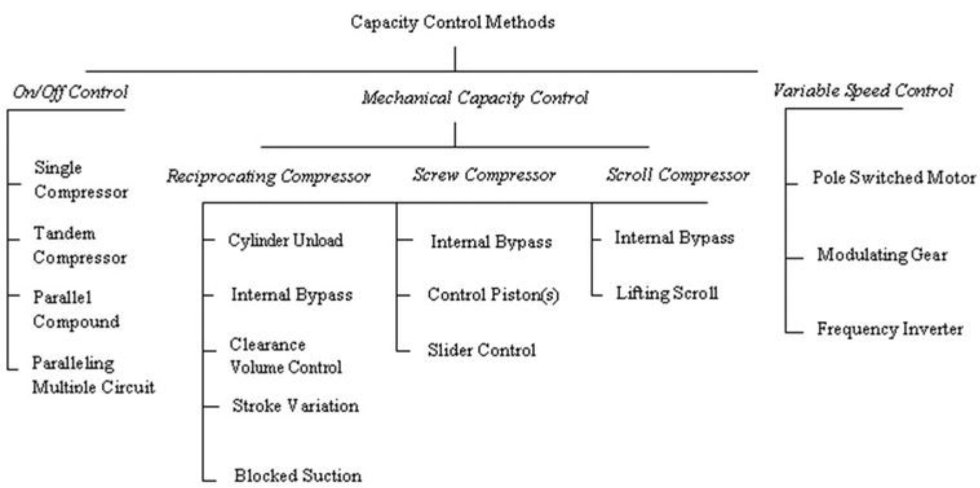


FIGURE 19 - OVERVIEW OF CAPACITY CONTROL METHODS FOR DIFFERENT COMPRESSOR TYPES **[6]**

4.2.1 ON/OFF System

This system is the simplest one; it just requires to turn on and off the compressor alternatingly. When the temperature reaches the set point, the compressor stops, so does the mass flow in the cycle. It is turned back on when the lower temperature level is reached [6]. Bagarella et al. [37] compared the on/off modulation with the variable speed system and found out that the optimum size, i.e. to obtain the maximum Seasonal Performance Factor (SPF), would be between 59% and 72% of the peak load. Doing so, the system would be able to meet the requirements for up to 98% working hours in the year.

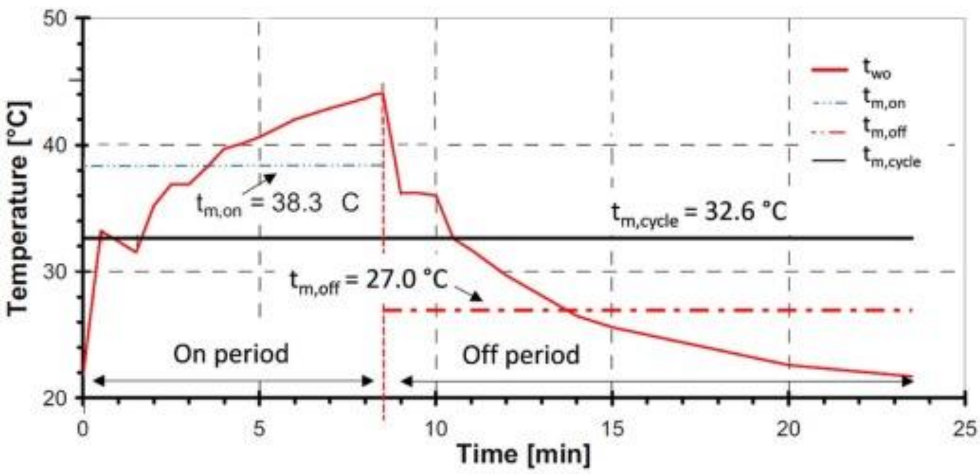


FIGURE 20 - EXAMPLE OF ON/OFF MODULATION **[37]**

4.2.2 Multiple Compressor

One smart way to modulate capacity is to install multiple compressors and operate them matching the variable heat load. At full load all of them are working, while at part load some can be excluded; as a result, some compressors will continue to work at full capacity with high efficiency and only one will operate at part load. This method is quite inexpensive and also reliable if one of the compressors should fail [6].

4.2.3 Variable Speed Drive

Speed control is widely used and it works regulating the speed of the compressor through an inverter. When speed decreases, volumetric displacement varies, so the circulating mass flow can be adjusted meeting the new heating load. The trend for isentropic efficiencies is very peculiar: when part load occurs, IE slowly increases thanks to the inverter and its control and it starts to diminish only when speed modulation is no more possible. From that point and below, capacity control can be achieved only by unloading cylinders (this method is explained later).

Due to the lower demand, COP increases as isentropic efficiency rises and the evaporation and condensation temperatures are closer [35]. This system can be performed continuously or step-wise and can achieve significant improvements. Both these approaches are quite expensive, but they can reduce speed down to 50% with no problems [12]. Since this system is very efficient, it is one of the most commonly used and will be applied in the case study later in paragraph 4.2.7.

4.2.4 Clearance Volume Control

The cylinder head contains an additional chamber connected to the main one thanks to a valve. With this mechanism it is possible to open the valve and obtain a larger clearance volume; doing so the cylinder capacity is reduced making possible to adapt it to the load. Re-expansion losses due to this larger dead space can be quite high affecting consistently part-load efficiency [38]. Of course, this is only available when employing reciprocating compressors.

4.2.5 Slide Valve Modulation

When operating conditions are those of the design point, compression efficiency reaches the highest value because discharge pressure matches the requirements; it can happen that required discharge pressure is lower than the one achieved by the screw compressor, leading to energy losses due to unnecessary compression (case c); the same losses can occur when required pressure is higher compared to design point: an additional amount of work is necessary to reach the wanted pressure value (case b). These situations are clearly displayed below in Figure 21. When the pressure required by the system does not match the discharge pressure, an extra work is in fact wasted, lowering isentropic efficiency.

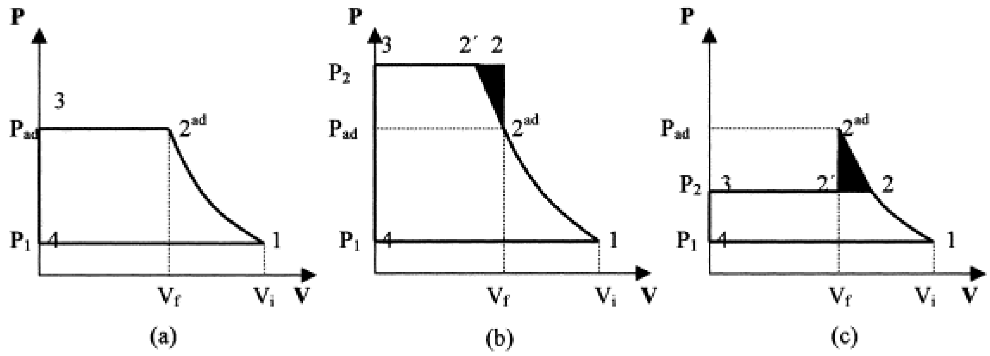


FIGURE 21 - ADDITIONAL COMPRESSION WORK WHEN REQUIRED PRESSURE VARIES

A) OPTIMAL CONDITION, B) UNDER COMPRESSION, C) OVER COMPRESSION **[39]**

This capacity control method is based on slide valves that are able to adapt the compressor displacement to power requirements by shifting the start of the compression. This way, the process is delayed and it results in a lower compression ratio. Also, the outlet window must be adapted when operating this control. Inlet valve modulation is mostly used in screw-type compressors and it can reduce the capacity between 25% and 100% by 25% steps for example [6].

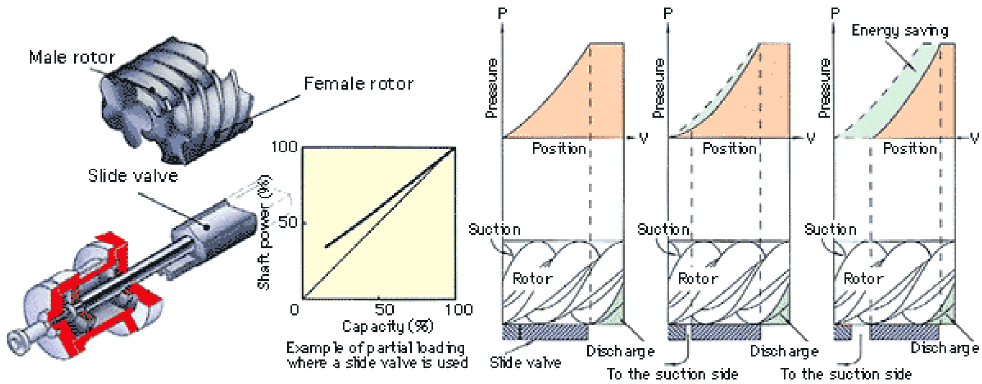


FIGURE 22 - EXAMPLE OF SLIDE VALVE OPERATION **[40]**

The potential of this system will be shown in the results when analyzing screw compressor performance at part load.

4.2.6 Hot Gas Bypass

This method consists in allowing the hot gas discharged from the compressor to return back to low-pressure zone. This can be done in two ways: the gas can be injected into the evaporator again without any benefits or back into the suction line. The bypass meets the changed heating demand but this results in a lower COP, therefore this method is not the best one [35]. Usually, this method is only used for safety in case other systems fail [41]. For scroll compressors this method can be quite effective: Wang et al. [42] found out that in heat pump systems it can decrease the heating capacity at 20% and improve energy efficiency up to 42%.

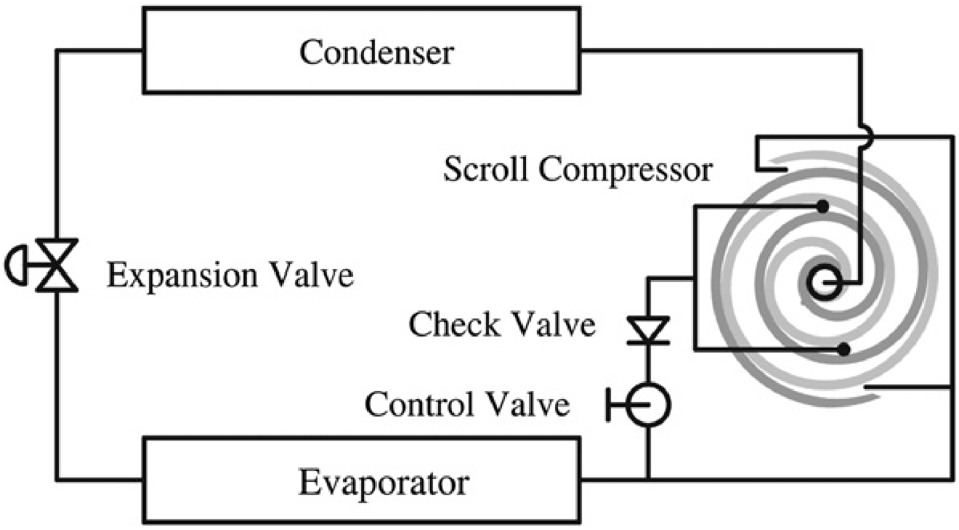


FIGURE 23 - HOT GAS BYPASS SCHEME WITH SCROLL COMPRESSOR **[42]**

4.2.7 Step Control

Another useful system for capacity control is unloading cylinders at part load. This is used with reciprocating compressors but there are similar systems for the screw and multi-compressor system. All these systems can also be referred to as "Step Control". What is done is open and close the suction ports for the screw and start and stop compressors in multi-compressor configuration (already seen in Paragraph 4.2.2). However, for reciprocating the system is quite useful and convenient [43]. For this type, capacity control is achieved by lifting the suction valves and holding them open. This allows the compressor to work also at 25% capacity. This system is often built-in and works automatically [44]. Moreover, it can be used together with variable speed and that will be examined in detail in the analysis in Chapter 6.

5 Method

To describe the system the first step is to build a basic model and develop it from there. The thermodynamic cycle described is an inverse Carnot cycle with electric power as input and heat as output. The efficiency of the system is given by the COP, which compares supplied heat with power input.

The software that will be used is *EES (Engineering Equations Solver)* which is very useful since many thermodynamic properties are already in its library. At first, components of the cycle should be designed and sized according to input data such as source and sink temperatures and power requirements. This is called the design point, where the available power is set at the maximum value. Then a part-load model is needed to assess the system performance throughout the year.

5.1 Basic Heat Pump Model

Before starting to build the model, some assumptions have to be made in advance and are presented next.

Assumptions:

1) No pressure losses and therefore no temperature variations in phase change. If pressure losses were considered, the temperature would change during the process of condensation and evaporation both, making the analysis more difficult; the cycle would look more like the one in Figure 24. These are still important to consider because overall COP can be affected in case they were consistent.

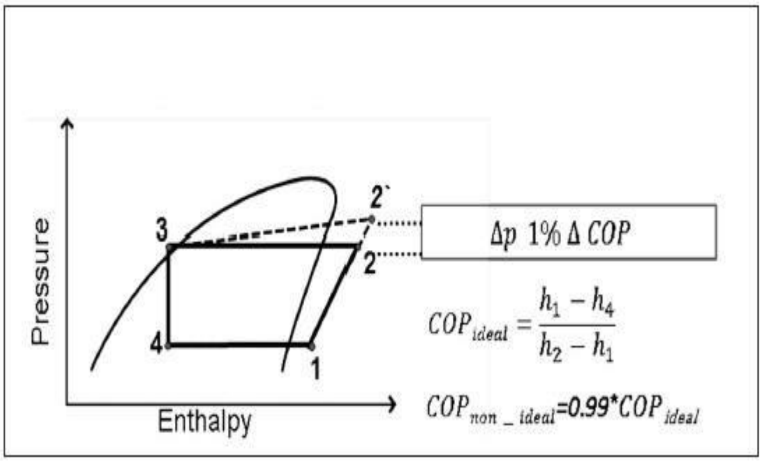


FIGURE 24 - PRESSURE LOSSES IN CONDENSATION **[45]**

2) U values: HTCs have been obtained from empiric relations showed in paragraphs 6.2.1 and 6.2.2 and they are considered as depending only on exchanged heat, as the HEX configuration has not been analysed. For different HEX types, U can vary much especially when working far from optimal conditions.

3) Pumping power and auxiliary neglected. Regarding electric power absorbed by pumps, some considerations are necessary. When the system operates at full load, the compressor absorbs the maximum power and that is usually much more than pumping power as may be agreed considering the following relation:

$$W \propto \frac{\Delta p}{\rho} \tag{5.1}$$

Unlike compressors, pumps work with liquids, which always have a much higher density than gases. Hereby, it is immediate to understand why pumping power can be neglected. When operating at part load, things change because compressor power starts to decrease and pumping power could assume more and more relevance, especially for very low loads. However, due to lack of time and layout knowledge, this has not been accounted when calculating COP. Therefore, it is important to keep that in mind when COP results will be displayed, since they could be higher than expected. This consideration is also valid for other auxiliary machines in the plant.

The *EES* model is presented in Appendix A for one-stage in *design point* and for two-stage in *off-design*. At first, a basic cycle was modeled with only 4 components and few equations. Then, it has been developed adding several more equations describing a two-stage cycle as well as the heat exchangers on water sides (groundwater and DH water).

In order to start the model, some input data are required such as the temperature of condensation and evaporation (pressure values are already determined inside the saturation curve, while the intermediate pressure value is needed), heat demand and efficiency values among others. When using EES, the order in which equations are written is not relevant, so it is possible to start at any point. In this case node 1 will be at the low-pressure compressor inlet. When IE is assigned the final state of the compression process can be detected using the simple equation displayed next for the low-pressure compression:

$$\eta_{is} = \frac{h_{2is} - h_1}{h_2 - h_1} \tag{5.2}$$

Given that h_1, η_{is} are known and h_2 has the same entropy of state 1 at an intermediate pressure, h_2 can be easily calculated and state 2 is finally determined. Of course, the same is valid for high-pressure compression.

Having said that, every node has been identified as can be seen next:

- Node 1 - *LP Compressor Inlet:* this point is found crossing evaporation temperature plus superheating with the correspondent saturation pressure.
- Node 2 - *LP Compressor Outlet:* first the compression is considered isentropic, so node 2is is determined by the intermediate pressure value. Then, node 2 can be found using isentropic efficiency.
- Node 3 - *HP Compressor Inlet:* intermediate pressure and temperature set as saturation temperature plus 5 degrees of superheating identify node 3.
- Node 4 - *HP Compressor Outlet:* same as node 2; isentropic efficiency value is the same and must be set in advance.
- Node 5 - *Condensation Start:* point 5 is determined by high-pressure value on saturation curve.
- Node 6 - *Condensation End:* when the fluid is all in a liquid state, vapor title is equal to zero at the condensation pressure.
- Node 7 - *Subcooling:* state 7 is determined if its temperature is fixed; pressure stays the same.
- Node 8 - *HP Expansion Valve:* at an intermediate pressure node 8 has the same enthalpy value as node 7.
- Node 9 - *Saturated liquid refrigerant at IP:* 9 is identified by the intermediate pressure value and zero vapor title.
- Node 10 - *LP Expansion Valve:* at low pressure, the state has the same enthalpy as 9.
- Node 11 - *Saturated liquid refrigerant at LP:* saturated fluid at low-pressure level.
- Node 12 - *Cycle End:* saturation curve at evaporation pressure (x=1).

Having defined the various nodes, it is possible to better understand all the processes in the cycle; some examples are the following: 1-2 LP compression, 3-4 HP compression, 4-5 *desuperheating*, 5-6 condensation, 6-7 *subcooling*, 7-8 HP expansion, 9-10 LP expansion.

In particular, when sizing the condenser it is very important to identify all the three different steps of the whole condensation process. This is due to the fact that refrigerant is superheated vapor at condenser inlet and subcooled liquid at condenser outlet. Therefore, heat transfer coefficient cannot be assumed constant for the entire process. This will be better explained later on.

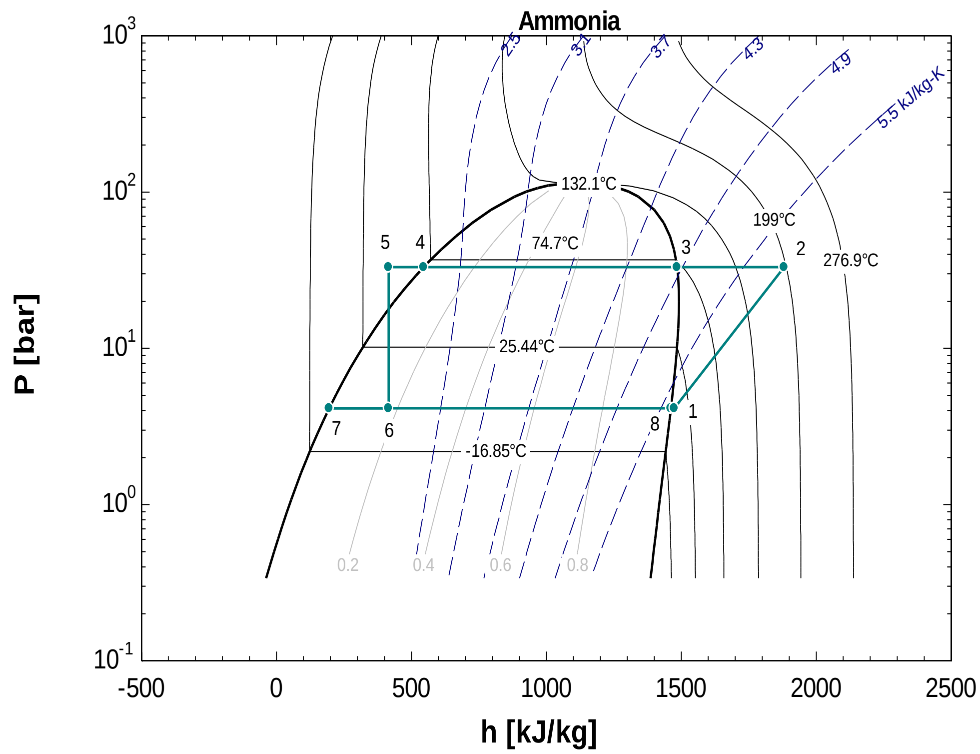


FIGURE 25 - ONE-STAGE P-H DIAGRAM

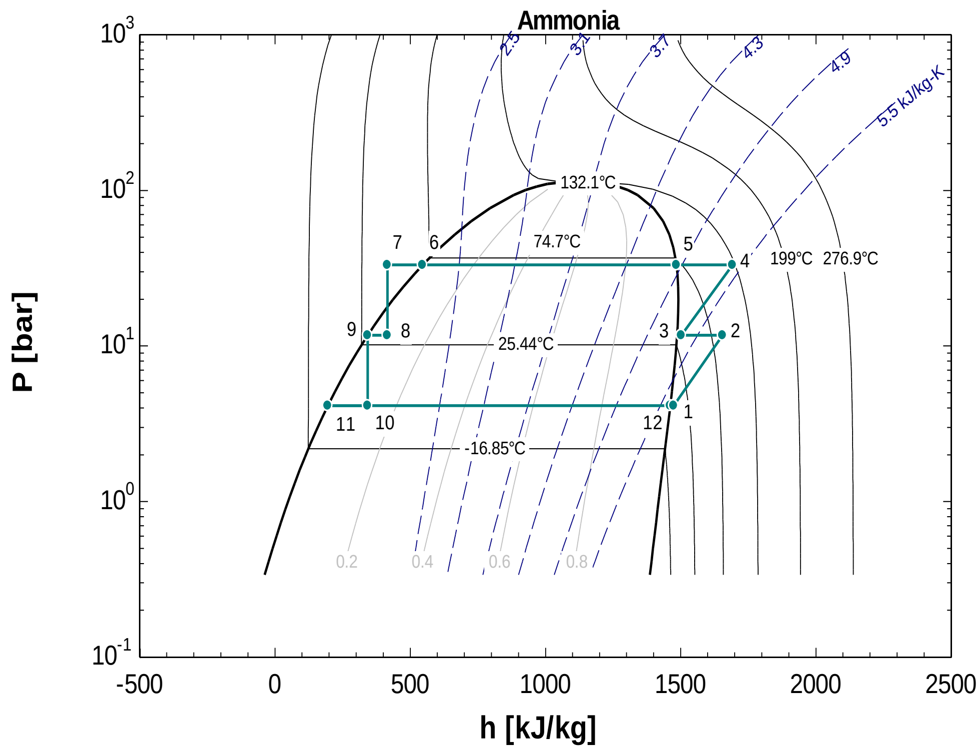


FIGURE 26 - TWO-STAGE P-H DIAGRAM

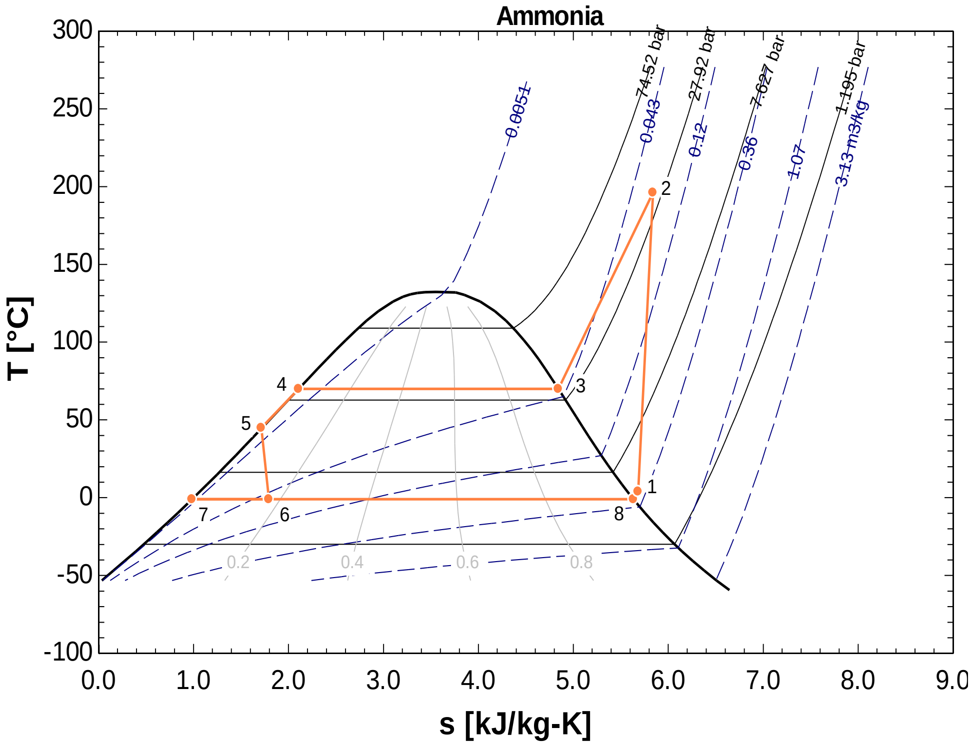


FIGURE 27 - ONE-STAGE T-S DIAGRAM

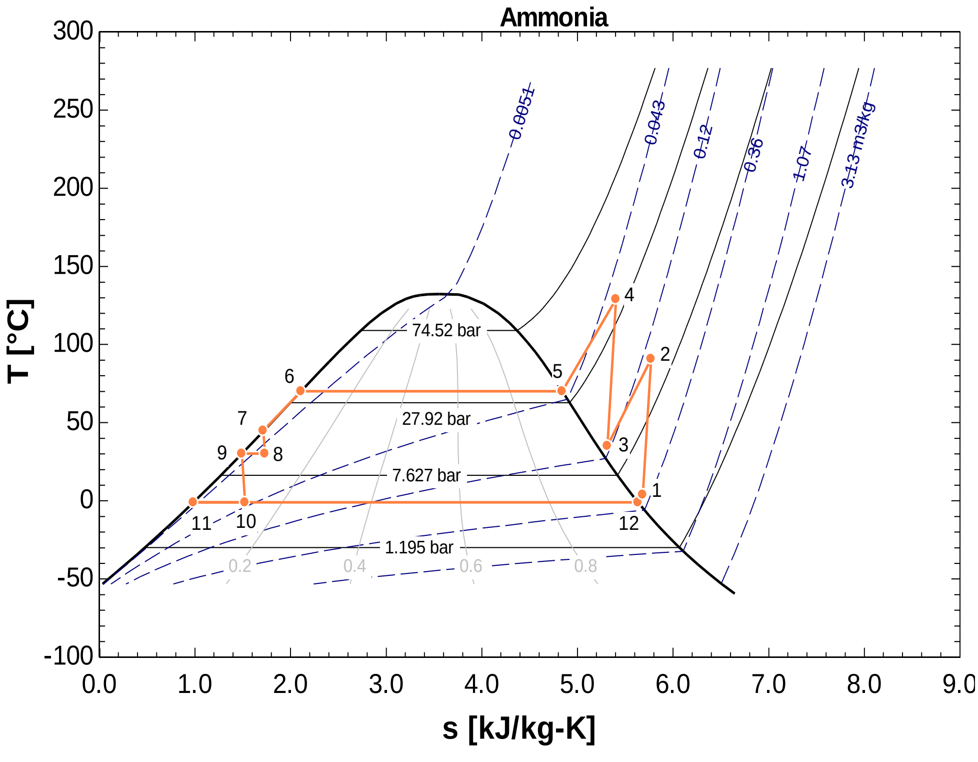


FIGURE 28 - TWO-STAGE T-S DIAGRAM

In Figures 25 and 26 one-stage and two-stage p-h diagrams are shown. Both have been obtained using ammonia. Please notice that it is in logarithmic scale. A T-s diagram is also shown in Figures 27 and 28 for both cases. The cycle is drawn using simple equations for the different thermodynamic states; the process is displayed below for the two-stage case.

5.2 *External Flows*

Apart from the working fluids, there are also secondary fluids that flow in condenser and evaporator. In this case, both are water: in the condenser, DH water absorbs heat, while in the evaporator water from the ground releases heat to the refrigerant. When sizing the system, it is very important to consider the interactions with external flows needed for a proper functioning. All the temperature values are shown in the table below for design point. To make calculation easier, some assumptions have been made like, for example, inlet groundwater temperature has been considered constant as it is sea water, i.e. high inertia source. This means that the mean temperature of groundwater does not decrease as heat is released.

	GROUND WATER	DH WATER
INLET TEMPERATURE [°C]	10.5	40
OUTLET TEMPERATURE [°C]	4	70
TEMPERATURE DIFFERENCE [°C]	6.5	30

TABLE 1 – TEMPERATURE VALUES FOR EXTERNAL FLOWS

5.2.1 *Evaporator Sizing*

The equations for the evaporator are fewer compared to the condenser case because the only process happening is the phase change (neglecting the superheating at the outlet). The evaporation occurs at a constant temperature; this value is usually fixed and it is the reference for nominal operating conditions. On the water side, the temperature decrease as the groundwater releases heat.

Input data for groundwater are inlet temperature, outlet temperature, and specific heat capacity. Overall exchanged heat is given by the cycle, so groundwater mass flow can be evaluated.

In order to size the evaporator, it is necessary to calculate the logarithmic mean temperature difference between the fluids as shown next:

$$\Delta T_{lm} = \frac{\Delta_1 - \Delta_2}{\ln(\frac{\Delta_1}{\Delta_2})} \ [°C] \tag{5.3}$$

Δ_1 and Δ_2 are the two temperature differences between the streams corresponding to inlet and outlet of the heat exchanger. Once this difference is known, an empiric relation should be used to obtain the heat transfer coefficient and then the evaporator is given by the well-known equation below:

$$A_{evap} = \frac{Q_{evap}}{(U_{evap} * \Delta T_{lm})} \ [m^2] \tag{5.4}$$

The heat balance in the evaporator for the groundwater side is:

$$Q_{evap} = m_{gw} * c_{p,w} * \left(T_{gw,in} - T_{gw,out}\right)[kW] \tag{5.5}$$

As for the U equation, it has been used one from a previous work [46] which analysed the same plant. This relation has been obtained evaluating a real heat exchanger performance and that resulted in the following:

$$U_{evap} = 5.5269 * Q_{evap} + 2103.9 \ [\frac{W}{m^2 * K}] \tag{5.6}$$

Instead of dividing the evaporator in two parts for evaporation and superheating it has been assumed that there will be the same temperature difference as if the ammonia would enter at -1°C and constantly increase its temperature until 4°C. This has been done to keep calculations as simple as possible. It has also been verified that the superheating area would have been less than 1% of the total evaporator area. Furthermore, with this guess the mean temperature difference would be underrated, therefore the corresponding area would be overestimated and thus it would be a safe approximation.

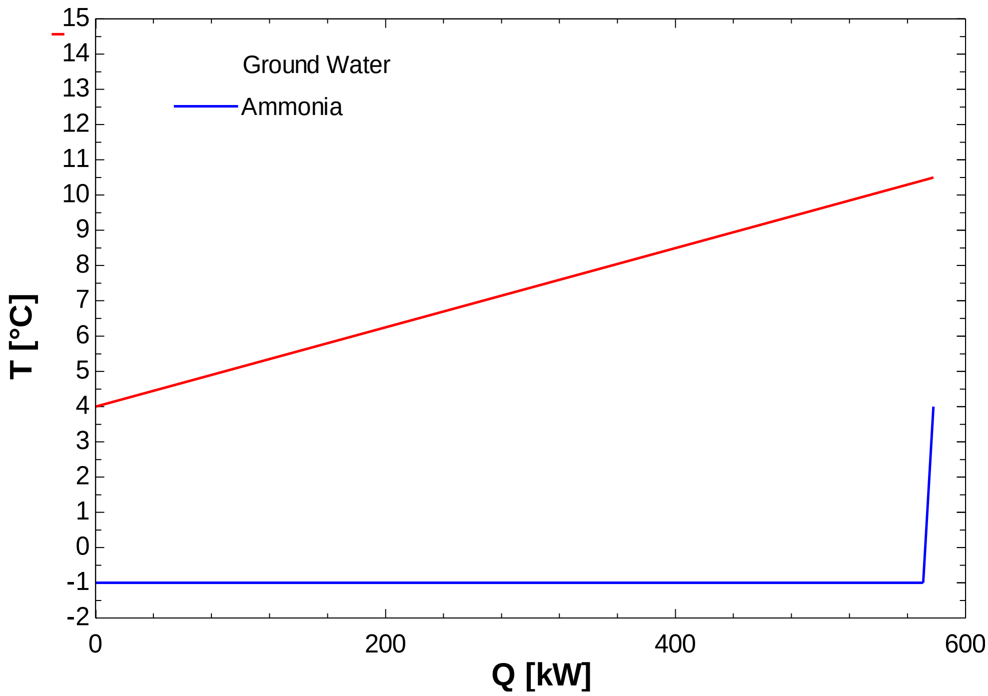


FIGURE 29 - TEMPERATURE PROFILE IN THE EVAPORATOR (GROUND WATER SIDE)

5.2.2 *Condenser Sizing*

In this case, more equations are needed since, as above mentioned, the whole condensation process is in fact divided into three steps: *desuperheating, condensation, subcooling*; each of the steps require its heat transfer coefficient as well as its own logarithmic mean temperature difference. The required mass flow can be obtained with the same procedure of the evaporator. Inlet and outlet temperature of the secondary fluid must be set in advance. Once the temperature differences are known, the procedure to follow is the same for the three steps, except for the fact that HTC varies. Usually, in *condensation,* it is obtained from empirical correlations, while in *desuperheating* and *subcooling* it has been assumed as a fraction of the one in condensation. As in evaporation, an empiric relation has been used to determine the U value. The equation is the following and was taken from the same study [46]:

$$U_{cond} = 0.5884 * Q_{cond} + 175.55 \ [\frac{W}{m^2 * K}] \tag{5.7}$$

$$U_{sc,ds} = 0.3 * U_{cond} \ [\frac{W}{m^2 * K}] \tag{5.8}$$

The heat exchanged in the condenser can be expressed using:

$$Q_{cond} = m_{DH} * c_{p,w} * (T_{DH,out} - T_{DH,in}) \; [kW] \tag{5.9}$$

The needed areas should be evaluated separately with the following equation for each phase:

$$A_{cond} = \frac{Q_{cond}}{(U_{cond} * \Delta T_{lm})} \; [m^2] \tag{5.10}$$

The last step is to sum the three areas obtained and get the total area for the condenser as follows:

$$A_{tot} = A_{ds} + A_{cond} + A_{sc} \; [m^2] \tag{5.11}$$

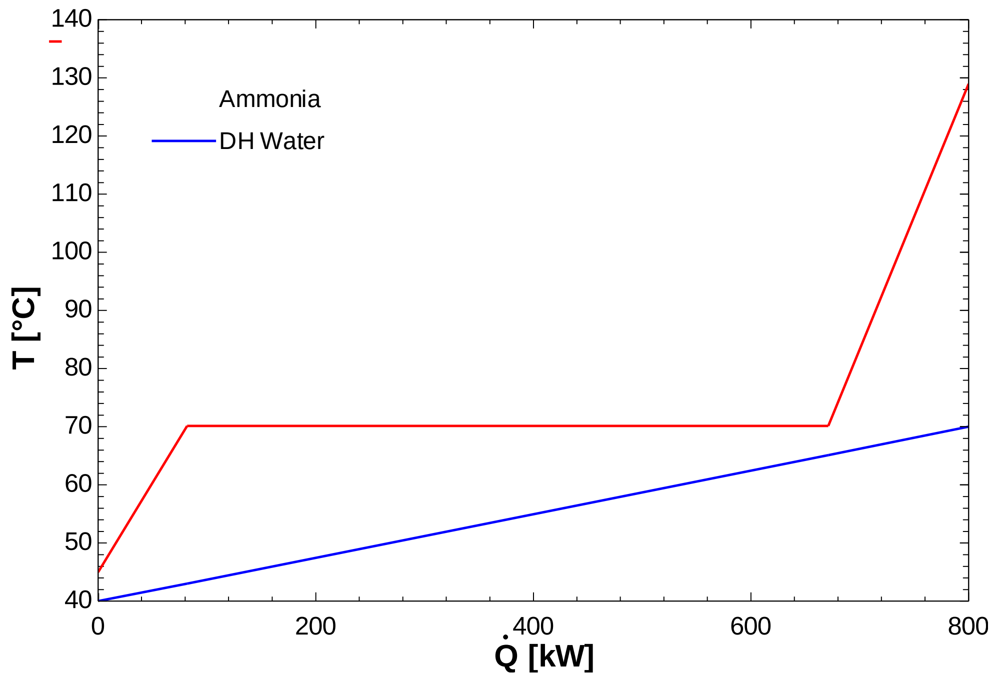


FIGURE 30 - TEMPERATURE PROFILES IN THE CONDENSER (DISTRICT HEATING WATER SIDE)

In Figure 30 it is possible to see the temperature trend of the two streams in the condenser: ammonia enters as superheated gas from the compressor and it releases heat to DH water. The HEX configuration is counter flow in order to achieve the best possible efficiency. The minimum temperature difference occurs when condensation starts and when ammonia exits the heat exchanger. This difference has been set for design point equal to 5°C, but it can get lower at part load. It was assumed that condensation starts when ammonia reaches the saturation curve when vapor quality is 1, but it is important to notice that the actual process might starts a bit earlier depending on the wall temperature.

5.3 Design Point Results

In order to design the system, some of the data have to be fixed. These data are presented below for both one stage and two stage for reciprocating compressors. They are necessary to have a preliminary idea on the size of each component of the cycle as well as other important parameters such as efficiency, electric power requirements and mass flows.

5.3.1 One-Stage Cycle

It is important to point out that one stage results are not to be considered as very accurate for various reasons later explained. However, the main purpose of presenting them is to show the process that has been followed in order to obtain an accurate model for two stage. Moreover, some of the data are highly approximate due to the fact that not many compressors operate in such a wide temperature range. In fact, efficiency values have been obtained for a lower temperature lift: 0-50, instead of 0-70. This approximation was necessary because the compressor data available were in that range. Since one stage will not be used for the final results this lack of accuracy should be forgiven.

In Table 2 below main output data are shown and they can provide a first idea of performance and necessary components. These results have been obtained with piston compressors. Later a comparison between those results and the ones for two stage will be made; doing so will demonstrate that two stage is highly recommended in these applications.

Input data choice:

- Ground Water Temperature: inlet and outlet values have been chosen using as reference the real heat pump system operating in Nordhavn district. These are assumed to be constant through the year and during the HP operation.
- District Heating Temperature: 70°C is the required temperature for domestic heating and is often used as a reference in this kind of application. Later on, two other values will be used in order to consider different scenarios (60°C and 80°C). The temperature of inlet water has been chosen arbitrarily.
- Condensation and Evaporation Temperature: these should be set according to the heat source and sink; in particular it was decided to set a minimum difference in the heat exchangers of 5°C, at least for design point. This is usually called "*pinch point*", i.e. the minimum temperature difference between the two streams. As an example, if 4°C is the groundwater outlet temperature and there has to be a minimum ΔT of 5°C, T_e has to be set at -1°C as it has been done. At

part load, it can happen that this difference goes below 5°C, but that is acceptable if the HEX areas are sufficiently extensive.

- Pinch Point: it is set as 5°C and it occurs in two different parts of the heat exchanger: the first when ammonia starts its condensation and then at the outlet when it is subcooled liquid.
- Compressor Efficiencies: for these parameters, some mean values have been considered; 0.8 for IE and 0.85 for VE (this process is referred to reciprocating compressors). These numbers represent a good compressor performance.
- Rotational speed: this has been taken from the compressor actually employed for the simulations. Reciprocating work at 1200 or 1500 rpm, whereas screw-type operate at 2950 rpm. These values come in handy to find the required swept volume for every compressor model.
- Power Size: 800 kW is a possible size for a large scale HP and happens to be the size of the real system taken as reference.

Input data		*UM*	*Output data*		*UM*
$T_{gw,in}$	10.5	°C	A_{sc}	28	m^2
$T_{gw,out}$	4	°C	A_{cond}	50	m^2
$T_{DH,in}$	40	°C	A_{ds}	26	m^2
$T_{DH,out}$	70	°C	A_{evap}	15	m^2
T_e	-1	°C	$\dot{m}_{ref}$	0.54	kg/s
T_c	70	°C	$\dot{m}_{gw}$	21.1	kg/s
p_e	4.1	bar	$\dot{m}_{DH}$	6.37	kg/s
p_c	33.2	bar	V	0.0043	m^3
$\dot{Q}_{cond}$	800	kW	$\dot{W}_{id}$	181	kW
η_{is}	0.8	-	$\dot{W}_{comp}$	228	kW
η_{vol}	0.8	-	$\dot{W}_{elm}$	240	kW
n	1200	rpm	COP	3.34	-

TABLE 2 - RESULTS FOR DESIGN POINT IN ONE-STAGE CYCLE (Q=800 KW)

In this case, the results for a piston compressor working at 1200 rpm has been reported, but of course, the results for other speed and types have been obtained in the same way. In particular, parameters that change between different models are IE, VE and rotational speed. Hence, the outputs that are affected are compression work and required compressor volume and capacity.

Another important task is to evaluate part load performance for both one-stage and two-stage cycles in order to point out the advantages of the two-stage. This comparison will be made in paragraph 5.4. Regarding full load operation, it is better to implement two-stage for the following reasons:

- Lower pressure ratio: in a two-stage system pressure is increased in two separate compressors, so each one of them works at a lower pressure ratio compared to a single compressor in the one-stage cycle. Compressor efficiency is usually higher at a low pressure ratio so overall cycle efficiency is improved.
- Electric power savings: in the two-stage cycle, hot gas at discharge can be *desuperheated* at an intermediate pressure (in this case this is accomplished thanks to the intercooler). This leads to the lower electric power required, due to the fact that the entropy increase is in fact reduced. This effect can be easily seen in the p-h diagram in Figure 31. Furthermore, the capacity of the system is higher because liquid refrigerant enters the evaporator at a lower enthalpy.
- Greater flexibility: with two-stage, it is easier to accommodate customer needs in different periods of the year [47].

Of course, the system is more complex and initial costs are higher but in most cases, two-stage has proven to be worthwhile [47].

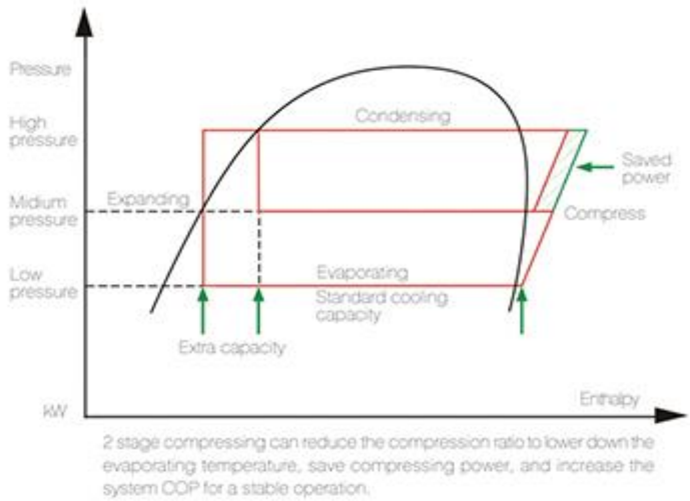


FIGURE 31 - SAVED COMPRESSION WORK AND EXTRA CAPACITY IN TWO-STAGE CYCLE **[48]**

5.3.2 Two-Stage Cycle

For two-stage, some changes are needed and will be presented next. These are choosing intermediate pressure, defining IE equations for both levels and designing an intercooler.

- *Optimal intermediate pressure:* in many cases, the approach to two-stage applications is to make the two compressors work with the same pressure ratio. Therefore the recommended intermediate pressure between evaporation and condensation pressure will be:

$$p_{int} = \sqrt{p_e * p_c} \tag{5.12}$$

  This is also what will be used for the model in the simulation. However, other approaches are possible: one is, for example, choosing the intermediate temperature as the arithmetic mean temperature of evaporation and condensation. This choice allows obtaining minimum throttling loss [28].
- *Low and high stage isentropic efficiency:* evidently two different IE equations must be implemented instead of only one. This choice is particularly important because overall efficiency is heavily affected by compressor efficiency. Usually, low-pressure compressors show lower efficiency because of a lower mean operating pressure and increased heat transfer to a much colder gas than in high stage [28]. These equations have been obtained with real data from *GEA RTSelect* software and will assume particular relevance in part load behavior analyzed in chapter 6.4. The whole procedure will be better explained when dealing with part-load behavior later on. For design point, IEs have been set at 0.8 for piston and for screw 0.8 in low pressure and 0.7 in high pressure (this is due to worse performance in high pressure for screw type).
- *Open Flash Intercooler:* between the two-stage, there is now an open intercooler that allows the energy exchange between low and high-pressure mass flow according to:

$$\dot{m}_{LP} * (h_2 - h_9) = \dot{m}_{HP} * (h_3 - h_8) \tag{5.13}$$

  High-pressure mass flow exits at x = 1 (or superheated), whereas low-pressure mass flow exits as saturated liquid (x = 0).

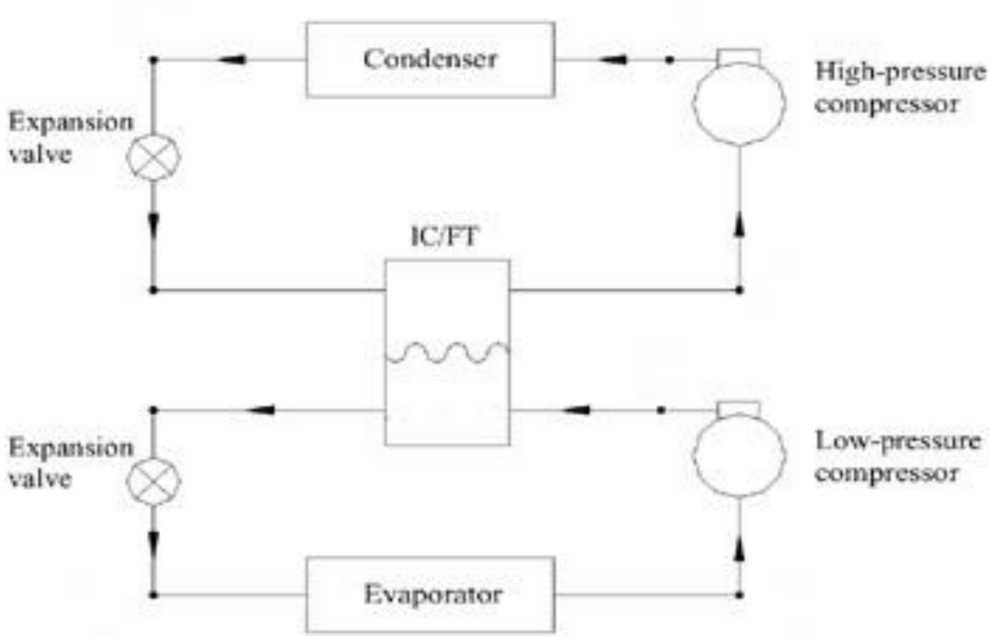


FIGURE 32 - LAYOUT SCHEME WITH OPEN FLASH INTERCOOLER **[49]**

Of course, the same procedure followed for one stage can be used for the two-stage cycle at design point: the results are displayed below in Table 3. As in one stage, these data refer to piston compressors.

Input data		*UM*	*Output data*		*UM*
$T_{gw,in}$	*10.5*	*°C*	A_{sc}	*32*	*m²*
$T_{gw,out}$	*4*	*°C*	A_{cond}	*70*	*m²*
$T_{DH,in}$	*40*	*°C*	A_{ds}	*31*	*m²*
$T_{DH,out}$	*70*	*°C*	A_{evap}	*20*	*m²*
T_e	*-1*	*°C*	$\dot{m}_{LP}$	*0.52*	*kg/s*
T_c	*70*	*°C*	$\dot{m}_{HP}$	*0.63*	*kg/s*
$\dot{Q}_{cond}$	*800*	*kW*	$\dot{m}_{gw}$	*21.5*	*kg/s*
$\eta_{is_{LP}}$	*0.8*	*-*	$\dot{m}_{DH}$	*6.37*	*kg/s*
$\eta_{is_{HP}}$	*0.8*	*-*	V_{LP}	*0.0093*	*m³*
$\eta_{vol_{LP}}$	*0.85*	*-*	V_{HP}	*0.0033*	*m³*
$\eta_{vol_{HP}}$	*0.85*	*-*	$\dot{W}_{elm,\ LP}$	*101*	*kW*
n_{LP}	*1200*	*rpm*	$\dot{W}_{elm,\ HP}$	*127*	*kW*
n_{HP}	*1500*	*rpm*	*COP*	*3.51*	*-*

TABLE 3 - RESULTS FOR DESIGN POINT IN TWO-STAGE CYCLE (Q=800 KW)

Isentropic efficiency values have been set referring to average compressor performance in full load operation and they were later compared with the ones actually calculated using the software, in order to verify that they resulted similar. However, they are only needed to have an idea of cycle efficiency. Regarding volumetric efficiency, the same values for low and high pressure have been used since pressure ratio is the same for both stages.

When analyzing COP of one and two-stage it is immediate to notice the increment in cycle efficiency. This compression power reduction increases as overall pressure ratio rises, so two stage cycle is highly recommended and almost mandatory when the temperature lift is quite high. On the other hand, if the sink and source temperature are quite close, benefits from two-stage might not be notable therefore one stage could be a most suitable solution [28]. Savings achieved by two stage can be calculated from equation 5.14 below. In the case study, total compression power is around 95% of one stage.

$$ES = \frac{\left(W_{elm,LP} + W_{elm,HP}\right)_{2stage}}{(W_{elm})_{1stage}} \tag{5.14}$$

Another major advantage is working with lower pressure ratio because compressors improve their performance when operating between close pressure levels. In one stage PR is around 8, while with two stage that is roughly 2.8. In this case, IE has been taken quite similar between the two cases, but in real applications that might not be true: with high pressure ratio, it can be very poor but here the purpose is to make a comparison trying to realize similar operating conditions. However, IE usually grows as pressure ratio diminishes so the benefits with two-stage should be evident. Furthermore, volumetric efficiency can be very high for low PR, whereas in one stage it cannot achieve the same value as in that case PR would be much higher. This happens because pressure ratio affects volumetric efficiency as was explained in Chapter 2.

5.4 Part Load Behavior

Once the data for design point are available from calculations it is possible to properly choose and size the main components: evaporator, condenser, and compressor. This is the start to build the off-design model and the input data are now the following: A_{cond} (desuperheating, condensation and subcooling), A_{evap}, V (for one particular speed). Moreover, other variables have to be set accordingly: mass flow is fixed once the speed is chosen (this will vary for different heat load and compressor types), η_{is} is calculated based on data from the manufacturers and depends on rotational speed, heat load, and pressure ratio, η_{vol} is obtained through relations found in literature and depending on pressure ratio only. As a matter of fact, a more thorough analysis would be recommended, especially considering that volumetric efficiency might heavily decrease when unloading cylinders (reciprocating compressors). Having established the needed input an off-design model can be built and thanks to that, results for the part load can be obtained. The main purpose is to evaluate COP for different heat load since isentropic and volumetric efficiency are affected at part load.

Before the part load analysis can start, it is important to present some of the assumptions made, so that the results and the choice of some input data would make more sense. These are displayed next:

- Minimum volumetric flow in pumps: it is well-known that centrifugal pumps usually operate close to the so-called BEP (Best Efficiency Point), i.e. with head and flow rate for which the highest efficiency is achieved. Anyway, when part load occurs, mass flow can decrease much. Hereby, some limits must be set in order to allow the pump to operate in safe conditions. This value has been fixed at around 8.5 kg/s (40%) for groundwater and at 1 kg/s for DH water. Anyway, this choice will appear clearer later when presenting the results. In practical applications, it is recommended not to go below 50% volumetric flow too often. However, it is also possible to implement a recirculation line. This adjustment allows the pump to keep low the mass flow towards the heat exchanger and still operate above the lower limit of 50%. With this system, a fraction of the pumped water would recirculate in the machine increasing the flow through the pump [50]. Another solution could be to employ several smaller units in parallel instead of a single pump. In this way, the mass flow could be better regulated and all the pumps would operate above their lower mass flow limit. The choice of two different mass flow limits for groundwater and DH water is explained next. Firstly, the two pumps are different because for groundwater a *submersible well pump* is employed, whereas in the DH side the pump model is a standard centrifugal. Secondly, for the DH network more flexibility is needed because the main goal is to be able to modulate the heat supplied to the customers in the best possible way. For this reason, mass flow limits should be less strict and it should be possible to operate the pumps properly with low loads.
- Pinch Point: in design point, this has been set equal to 5°C but at part load, it is assumed that it can be lower than that value as long as the areas are capable of allowing the heat exchange required.
- Isentropic Efficiencies: when gathering all the needed data for compressors, it has not been possible to run the software for the specified temperature that occurs in the model. Thereby, it is assumed that compressors would behave in the same way, even if the temperature range is not exactly the same. In addition, it is important to underline that some losses in the compression process may not have been considered by the software, so IE values might

not be achieved in the real system and thus the COP obtained through those efficiencies may be overrated.

- Electric Motor Efficiency: this value has been assumed equal to 0.95 and constant for every part load ratio. Some verifications have been made and when including the motor in the calculations, its efficiency has always been higher than 0.97.

5.4.1 One-Stage Cycle

In order to better understand the system performance, it is useful to start to analyze the cycle as if it was one-stage. These data displayed and discussed below do not represent the actual system, still, they can provide interesting information.

Since it is not the main purpose, only a few data are presented for one-stage; those are basically comparisons in performance between different reciprocating compressors models (V700 and V1100) and between fixed and variable speed. Regarding screw compressors two models have been taken into account: model LM and MM from GEA; first a comparison between these models is shown (with variable speed control), then model MM is evaluated both for variable speed and fixed speed. The compressor models have been chosen on the basis of similar power size (800 kW in condensation), but for some of them the maximum load was slightly lower, therefore, results can differ substantially when compared to actual performance. Anyway, for one stage what is important is to evaluate the trend of isentropic efficiency and its influence on COP. Some practical data are reported in Table 4 below. These data refer to the same conditions used to calculate isentropic efficiency.

Compressor Model	*Compressor Type*	*Mass Flow*	*Volume Flow (Suction)*	*Theoretical Displacement*
-	-	*[kg/h]*	*[m^3/h]*	*[m^3/h]*
V700	*Piston*	*1726*	*510*	*637*
V1100	*Piston*	*2589*	*766*	*955*
LM	*Screw*	*1955*	*581*	*666*
MM	*Screw*	*2570*	*763*	*867*

TABLE 4 – COMPRESSOR CHARACTERISTICS (ONE-STAGE)

Isentropic efficiency values have been obtained through simple calculations and the process will be quickly explained next. From *GEA RTSelect* software it is possible to gather data for every compressor at part load, for example, varying speed or number of cylinders running. Adapting the model in *EES* to the conditions provided by the manufacturer makes possible to obtain data as close as possible to the real ones. In particular, it is necessary to evaluate real compression work from the part load table and isentropic compression work from the solution table in *EES*. Comparing these two for different loads gives the isentropic efficiency in the wanted range of power according to:

$$\eta_{is} = \frac{\dot{m}_{ref} * \Delta h_{is}}{\dot{m}_{ref} * \Delta h_{real}} = \frac{W_{is}}{W_{real}} \tag{5.15}$$

This equation applies to every load, so the compressor efficiency can be estimated in the entire operating area. After that, a simple polynomial interpolation can be made to obtain a relation between isentropic efficiency and rotational speed or condensation heat. Some examples of those interpolating equations can be seen in the model presented in Appendix A. This whole process has been carried out using *RT Select*, *EES* and *Microsoft Excel®* .

5.4.2 Two-Stage Cycle

When running the model, some boundaries must be fixed in order to obtain consistent results. For example, rotational speed needs to have a lower limit of 500 rpm because none of the compressors can run below that speed value. Moreover, pumps work at its best only when the mass flow is above a certain value. In particular, in *EES* this has been done without setting constraints for the variables but implementing the min/max command. This allows the system to choose the minimum or maximum value between the one calculated from the equation and the one set as an upper/lower limit. A quick example is shown below:

$$n_{LP} = \max(\frac{V_{dot,LP}}{\eta_{vol} * V_{LP}} * 60;\ 500) \tag{5.15}$$

The real heat pump works on a two-stage cycle so it is more relevant to analyse the system as it is once some results for a basic one-stage cycle have already been obtained. What could be expected is to get a similar trend both in compressor efficiency and overall COP performance with improvements due to better modulation and lower electric power required.

When analysing the two-stage cycle it is important to evaluate compressor performance in their temperature range according to the results from the cycle. Unfortunately, it has not been possible to match the exact same range since compressor data are available for certain operating points only. Please notice that *RTSelect* software automatically assigns a cycle and some components, therefore, the results might not be exactly the same as in the *EES* model. Still, some validations of the results have been made by calculating the same data through two different procedures: besides the *EES* model, also the software *RefProp* has been used in order to confirm the data for the ideal compression work. Furthermore, the full load mass flow was available also in the *GEA* software, so it has been possible to compare it to the one from *EES* and in every case the deviations were below 2%. The same can be said for condensation heat, whose differences were even lower between the two outputs.

5.5 System Configurations

Before showing the various configurations that will be tested, it is important to display an expected trend of the results and this will be done, at least for the first case. This is useful especially to validate in the same way the developed model and to be sure to proceed in a proper way. As already discussed, using variable speed modulation improves piston compressor efficiency in a slow but steady way. All the IE trends analysed showed this along with the drop for low mass flow when cylinders were unloaded.

Thanks to other studies [51], it is possible to foresee a possible COP trend for different loads. This is shown in Figure 33 and considers the cooling mode as well as the heating mode. For the case study of this project, the operating area stops at 20% for practical reasons but still, it is interesting to see the entire curve. Moreover, in the figure the different reasons for COP variations are clearly detected; above 50% the inverter speed control improves efficiency, below that point, cylinders start to be unloaded causing a slow reduction and below 10% there is a sharp drop due to different losses such as oil return, parasitic power and system cycling.

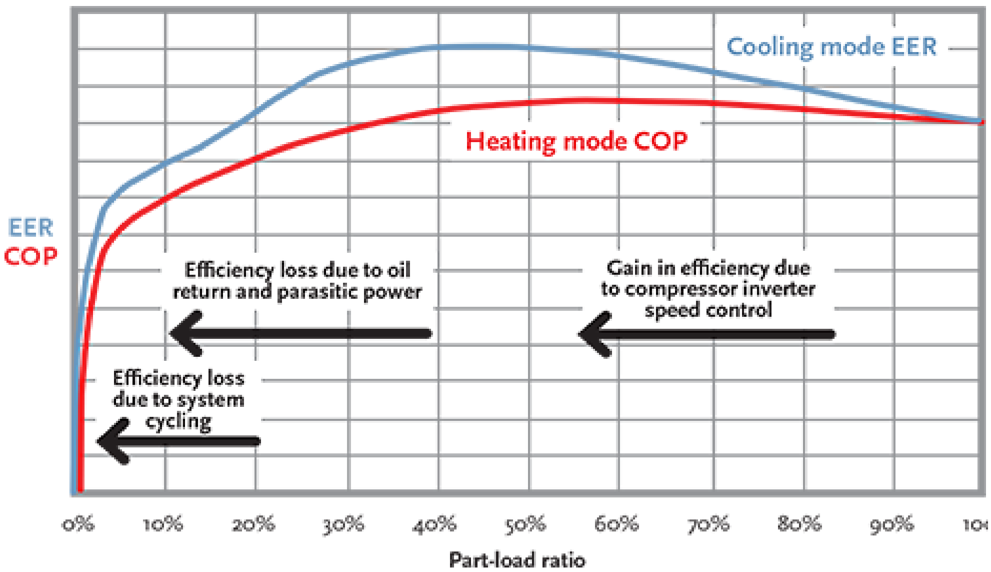


FIGURE 33 - EXPECTED COP TREND AT PART LOAD WHEN USING VARIABLE SPEED **[51]**

Shortly, the results obtained from the model will be presented and it will be possible to compare those to the ones in the figure above.

For the simulation four configurations have been chosen for different reasons: two cases regarding piston compressor with a focus on the possibility to use variable speed or not; two cases for screw compressors running at variable speed and employing slide valve. These cases are briefly presented next:

1) Piston compressors both with variable speed modulation:

   The first configuration that will be evaluated is using piston compressors with variable speed control for both stages. From previous analyses upon isentropic efficiency, this is expected to be one of the best choices since piston compressors usually perform better than the screw and variable speed improves efficiency when operating at part load at least above certain mass flow values.

2) Piston compressors; low pressure at fixed speed, high pressure at variable speed:

   The second case investigates the interaction between the two stages when one compressor works at variable speed and the other at fixed speed unloading cylinders. The intercooler balance fixes the relation between the two mass flows and the interaction between fixed speed and variable speed in the two stages could be relevant.

3) Screw compressors with variable speed and slide valve combined:

   The third scenario considers the employment of screw compressors; these are in general characterised by worse performance than piston types but still, they have interesting efficiency, especially in low pressure. Furthermore, screw compressors are characterised by few moving parts and longer service intervals; their size is usually bigger than reciprocating and they are able to work with a higher temperature lift between sink and source. Given that, choosing screw compressor could be a wise choice especially for very large application [26]. In particular, it is relevant to analyse the different capacity control methods, as they can guarantee consistent energy savings if they are employed properly.

4) Screw compressors comparison between methods:

   In the fourth analysis, the focus is upon capacity control methods available for screw compressors and in particular the relevance that variable speed and slide valve have for the various part load ratios and stages.

6 Results

In this chapter, the most relevant results are displayed and they are divided between IE values obtained as previously explained and the cycle analyses for the different configurations.

6.1 Compressor Efficiencies

6.1.1 One-stage Cycle

<u>Reciprocating compressors:</u>

The investigation on models V1100 and V700 has given the following results: they both have a slight and steady increase when the load diminishes, the maximum value is around 0.875 and it is reached at around 0.3 kg/s (400 kW or 50% load) for V1100 and at 0.2 kg/s (240 kW or 30% load) for V700; then they both start to drop quite quickly to 0.715 and 0.795 respectively (at 0.1 kg/s). Being V700 a smaller unit, it is most suited for low loads or for a device that often works below 50% load.

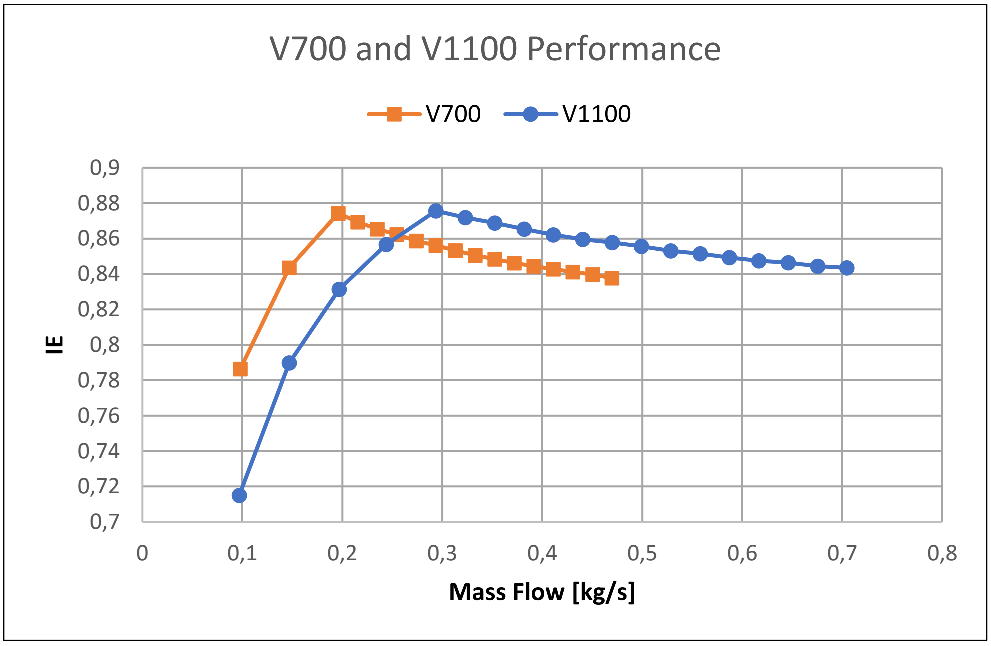


FIGURE 34 - V700 AND V1100 IE COMPARISON

Please notice that maximum power for this model is around 600 kW in condensation, so it would not be a valid choice anyway. However, V1100 shows a satisfactory behavior even at 30% of the load with IE above 0.8, therefore it could represent a good option for the case study, even if we are still in a preliminary stage. More results and charts regarding COP will be reported when presenting the different configurations and for two-stage only.

Model V1100 has been tested also running at fixed speed and some preliminary information is reported next. Variable speed allows the compressor to operate at high isentropic efficiency when power is way lower than nominal. On the other hand, with the fixed speed, the compressor can work with the same efficiency or slightly higher at full load but that value always decreases when part load occurs. Moreover, operating range at a fixed speed is way more narrow than with variable speed. This is due to the fact that without speed regulation the only way to modulate capacity is by unloading cylinders: in this particular case the compressor works with 6 cylinders and can reduce mass flow and still working until no less than 3 cylinders. Basically, the available steps for part load are only 4.

Design point efficiency at 800 kW is basically the same (around 0.85 for both) but at part load trends are quite different: for variable speed efficiency increases by around 3 percent before it starts to lower, while in the case of fixed speed it drops steadily to 0.765 at 400 kW. Besides, it is not advisable to use fixed speed below 400 kW since it is out of its working range.

In this case, it is clear that variable speed is to be preferred. In fact, as previously explained, HPs often operate at part load, therefore design point conditions are rarely achieved. This is the reason why it is more important to evaluate performance in different power range and also why design point efficiency is not that important. Having said that, it should be easy to understand why variable speed is better even if at design point it has a little worse performance.

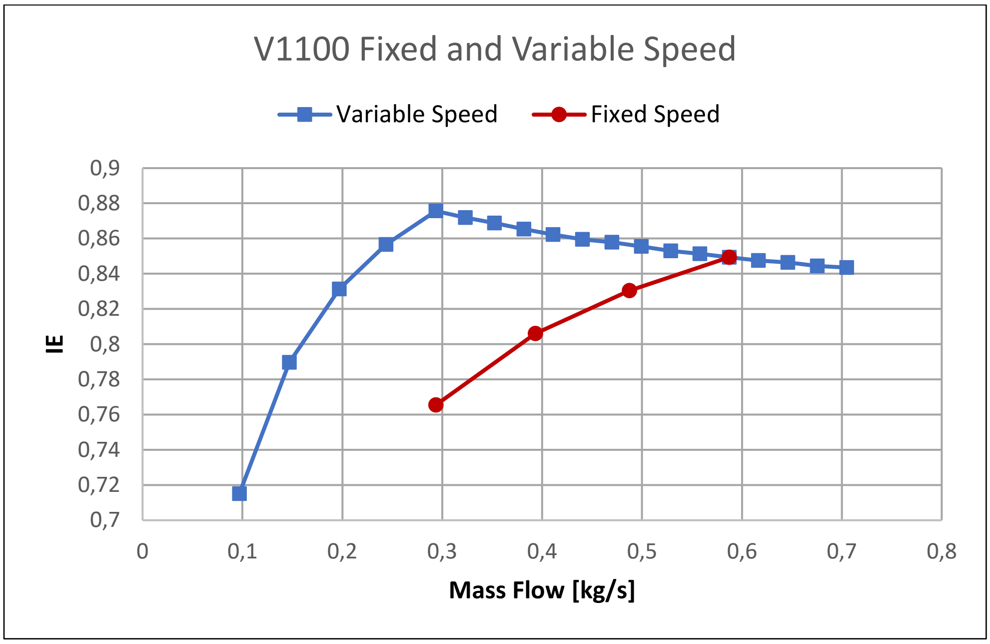


FIGURE 35 - PISTON MODEL V1100 COMPARISON BETWEEN FIXED AND VARIABLE SPEED

Screw Compressors:

As for screw compressors, the trend is very different: when operating at part load efficiency goes down in any case, instead of improving as in the piston compressor case. At first, two capacity control methods were compared separately and then together finding out that slide valve modulation achieves higher efficiency for high mass flow, whereas variable speed drive performs better when working with a low mass flow. The two systems combined seem to help each other when applied together: for high power, the trend is basically the same as in slide valve case, then it starts to decrease less than this last one thanks to the effect of variable speed. The result is that efficiency with both controls is higher in all the operating range. In the cases with slide valve included, built-in volume ratio control was available, meaning that at part load the actual volume ratio could be changed to meet the optimum operating conditions.

This has been done evaluating compressor model MM. When using slide valve the speed was fixed and for variable speed, the built-in volume ratio could not change. However, it is important to point out that when slide valve trend goes below variable speed case the isentropic efficiency is already quite low being around 0.55 for 0.25 kg/s. Therefore, this benefit from using also variable speed would only contain the drop. In other words, using this compressor model when mass flow often goes below a certain value would not be a wise choice in any of these cases. In Figure 36 the three trends are reported and it is easy to see that in the green line for high mass flow the curve resembles the one with slide valve, while for lower power it follows more or less the case with variable speed only.

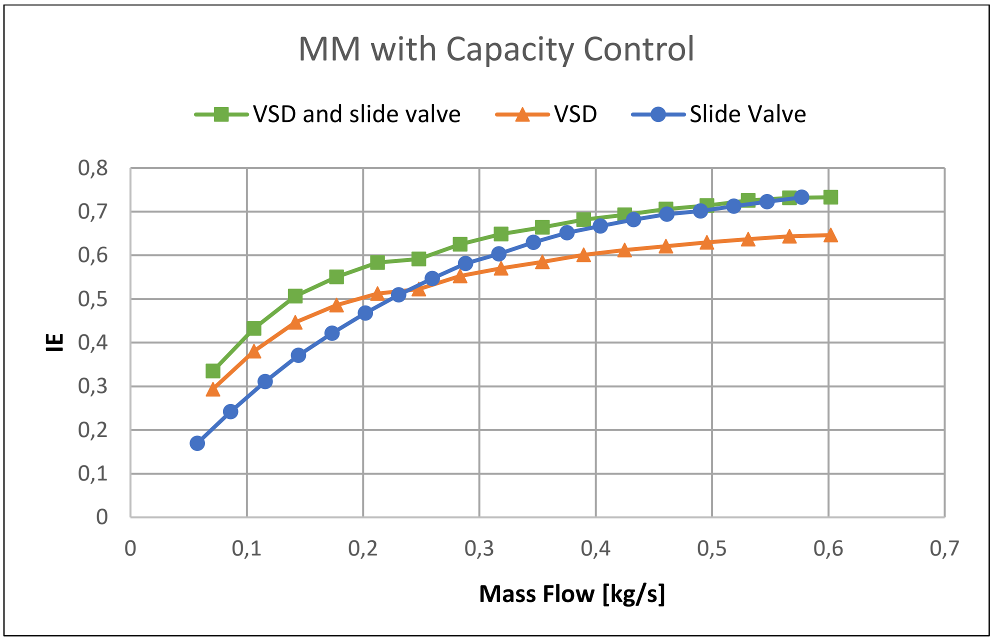


FIGURE 36 - MODEL MM PERFORMANCE WITH DIFFERENT CAPACITY CONTROL METHODS

In the next graph, there is a comparison between two different models of screw compressors: LM and MM with both controls available. The curve for MM is the same of the previous analysis and the one for LM has been calculated with the same procedure. For both these machines, variable speed and slide valve modulation are available and have been used while evaluating their performance.

Model LM has a maximum flow rate of around 0.54 kg/s and can achieve 722 kW as condensation heat in the given temperature range. Its trend is quite interesting since it has a very steady IE above 0.4 kg/s (60-70% load) and it starts to decrease consistently only below that point. In its best range, the efficiency value stands around 0.71.

When comparing the two models it can be noticed that LM has higher efficiency where they both can work properly, while MM achieves higher values with high mass flow, close to design point operations (800 kW).

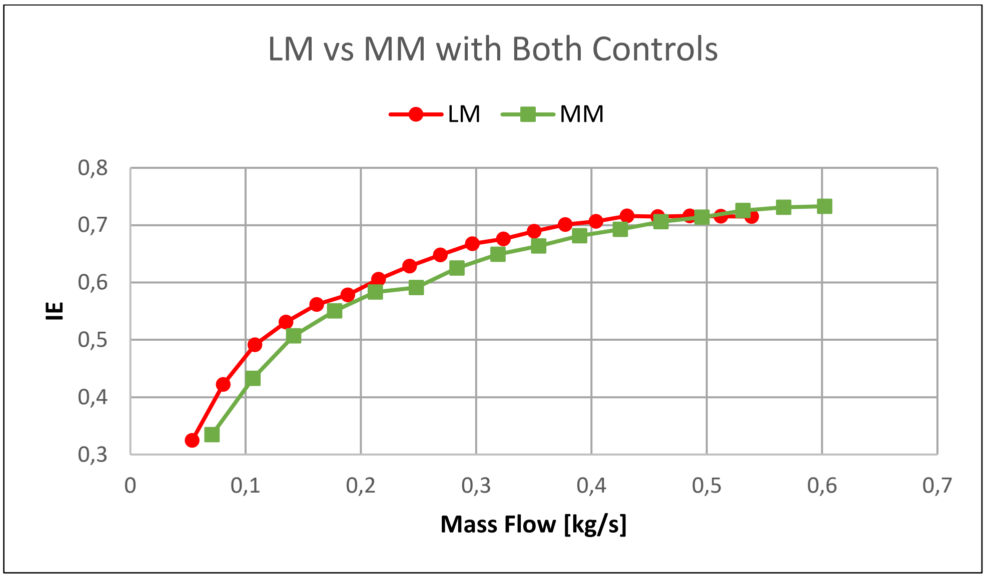


FIGURE 37 - PERFORMANCE COMPARISON BETWEEN MODEL LM AND MM (VARIABLE SPEED AND SLIDE VALVE TOGETHER)

Comparing piston with screw compressors is not very relevant since the first ones have consistently higher isentropic efficiency both in design point and part load. In fact, screw compressors become an interesting choice, when also volumetric efficiency, speed, and maintenance are considered. Higher speed and better volumetric efficiency are the reason why screw compressors can be smaller and occupy small room than reciprocating. Furthermore, due to the characteristics of their compression process and to fewer moving parts, they usually require less maintenance and don't make much noise.

6.1.2 Two-stage Cycle

<u>Reciprocating Compressors</u>:

Low Pressure

As expected the performance of model V1100 is basically the same as in one-stage cycle with some substantial differences: first, when operating at fixed speed it can reach lower power; secondly, in one-stage, it could work unloading up to 3 cylinders while for this range 4 cylinders can be unloaded and leave only 2 in operation. Furthermore, a consistent decrease in efficiency can be noticed for fixed speed, being IE always higher for one-stage conditions than in two-stage. These data have been obtained in the following temperature range: 0-50°C for one-stage and 0-30°C for two-stage; IE is clearly affected by pressure ratio but variations are more relevant when rotational speed does not vary. As for variable speed, some slight discrepancies can be detected but they are very limited. In the area between 0.5 and 0.7 kg/s one-stage case seems to give better performance even if the maximum difference does not exceed 2%. This happens because the operating point is in the area where IE increases with pressure ratio before it starts to lower. This tendency can be seen in Figure 39 for piston compressors. To summarize the two-stage case, variable speed has more or less the same trend, while fixed speed allows better modulation (wider range) but with slightly lower isentropic efficiency than in one-stage.

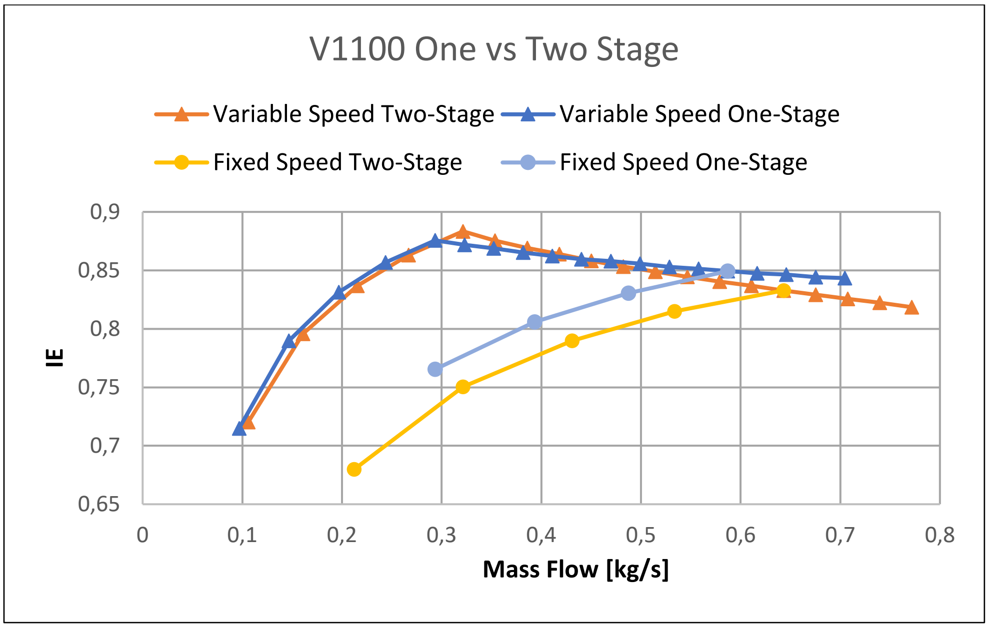


FIGURE 38 - PERFORMANCE COMPARISON BETWEEN ONE AND TWO-STAGE

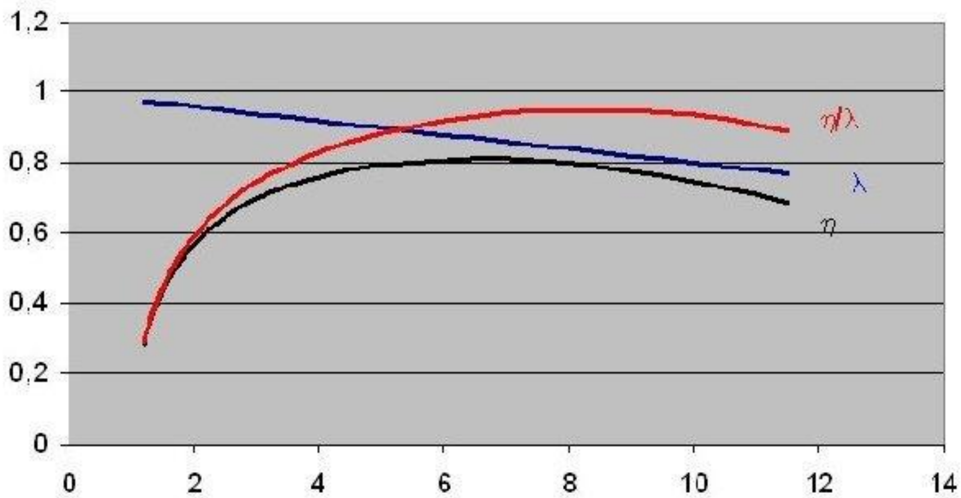


FIGURE 39 - IE, VE AND THEIR RATIO AS A FUNCTION OF PRESSURE RATIO (RECIPROCATING) **[52]**

When analysing in detail this piston compressor model, it can be seen that its operating range can be divided into two sections as visible in Figure 40: for a high mass flow, variable speed modulation is possible and IE grows, whereas for lower mass flow some cylinders have to be unloaded. When this occurs, a drop in efficiency is unavoidable.

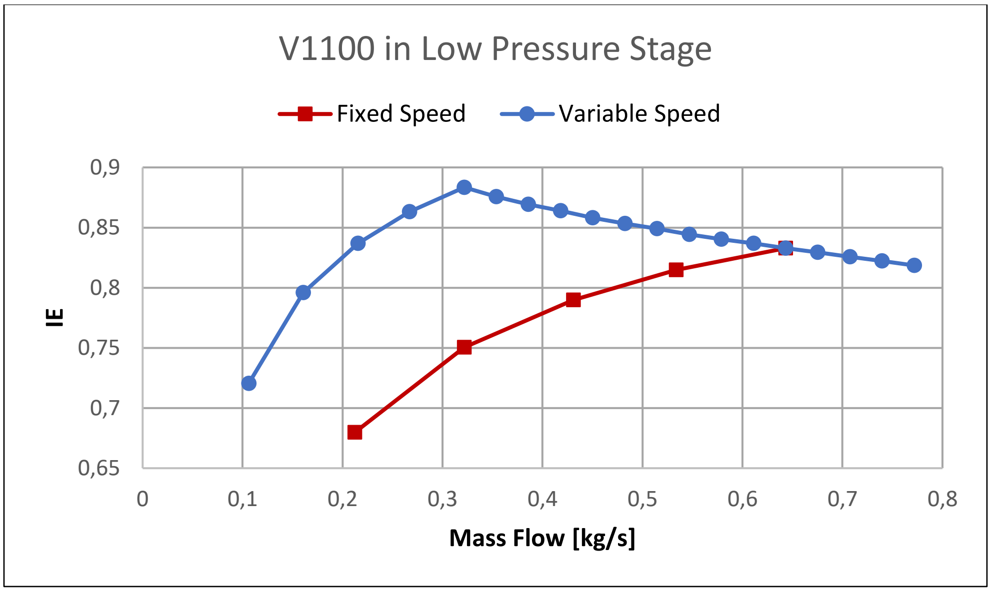


FIGURE 40 - MODEL V1100 LOW PRESSURE PERFORMANCE

Moreover, an analysis of different intermediate temperature has been carried out giving the results illustrated next. Of course, when forward temperature varies, operating conditions are affected: in particular intermediate pressure will vary giving a different pressure ratio. If this occurs, it is easy to understand why the following graph is important: with different forward temperature isentropic efficiency varies and all the cycle is affected. The influence of this deviation depends on how intermediate pressure is controlled when operating conditions vary. In this case, that is set to maintain the same pressure ratio between the two stages: for this reason, the intermediate pressure and thus the intermediate temperature does not suffer relevant variations. If other controls were employed, that could no longer be true and this performance verification would be quite important. In Figure 41 and 42 all curves are shown for different condensation temperature. When this rises efficiency grows; this improvement is more limited for high temperature: quite relevant from 25°C to 30°C and almost imperceptible from 35°C to 40°C. It also can be noticed that this effect is more important for higher mass flow, while in the area of cylinders unloading (low mass flow) it starts to be meaningful only for 25°C.

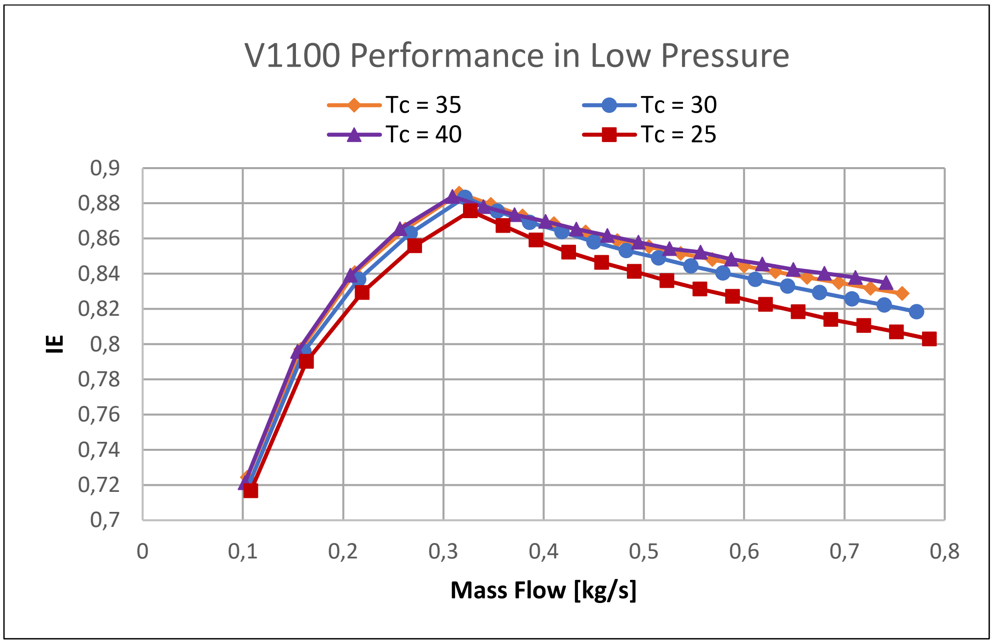


FIGURE 41 - IE FOR DIFFERENT FORWARD TEMPERATURE IN LOW PRESSURE

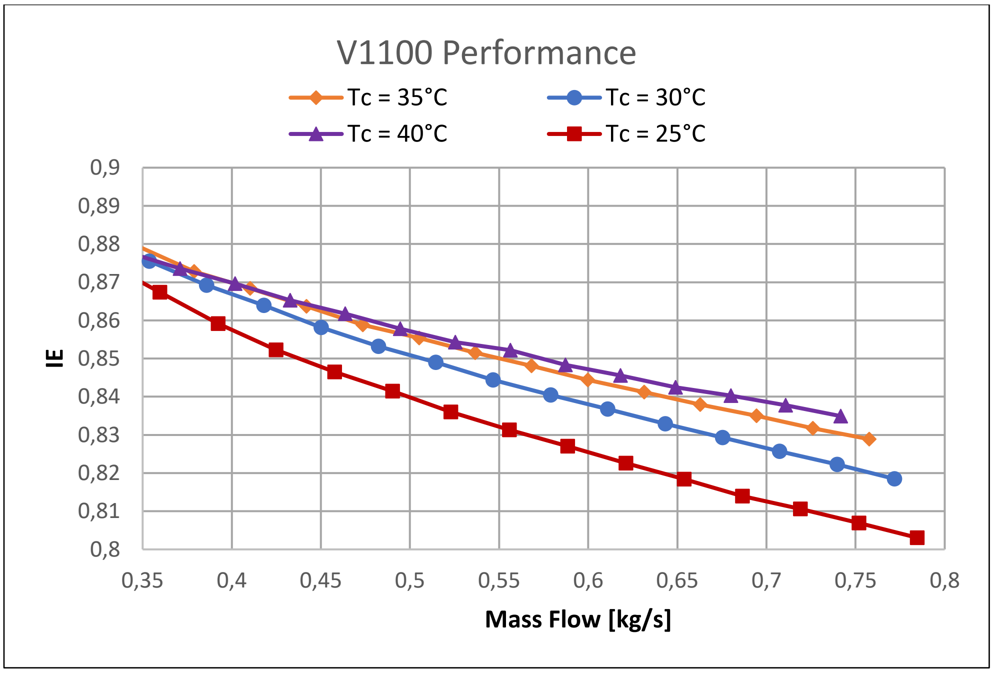


FIGURE 42 - ZOOM OF HIGH MASS FLOW AREA

High Pressure

The first model that has been considered is 65HP, which is meant to work in high temperature and pressure range. The inlet and outlet temperatures have been chosen equal to 30°C and 70°C, which are similar to those obtained in the EES model. Unfortunately, the required capacity cannot be achieved using this model. Its maximum mass flow is about 0.41 kg/s, corresponding to around 480 kW of condensation heat. Regardless, this model has been taken into account because it has a very interesting trend and can work in the exact same temperature range. The other model considered is V700, which can reach the design capacity but it is not able to work in that temperature range.

As said before, 65HP works with lower mass flows and this allows it to modulate capacity without unloading cylinders down to 15-20% load. It reaches a minimum working point of 0.137 kg/s with 85% isentropic efficiency. This regulation with variable speed only makes possible to obtain a slow but steady improvement in efficiency also for very low mass flow and no sudden drops in performance as seen in previous compressor models.

Although it does not reach the required capacity, 65HP would represent a valid choice in case of lower power size or if used in a parallel compound configuration (for example two of them elaborating the mass flow required for 400 kW each). However these configuration issues would fit better in a more detailed analysis than in this project work; for now only single compressor will be analysed, but further investigations on this subject would probably be appropriate.

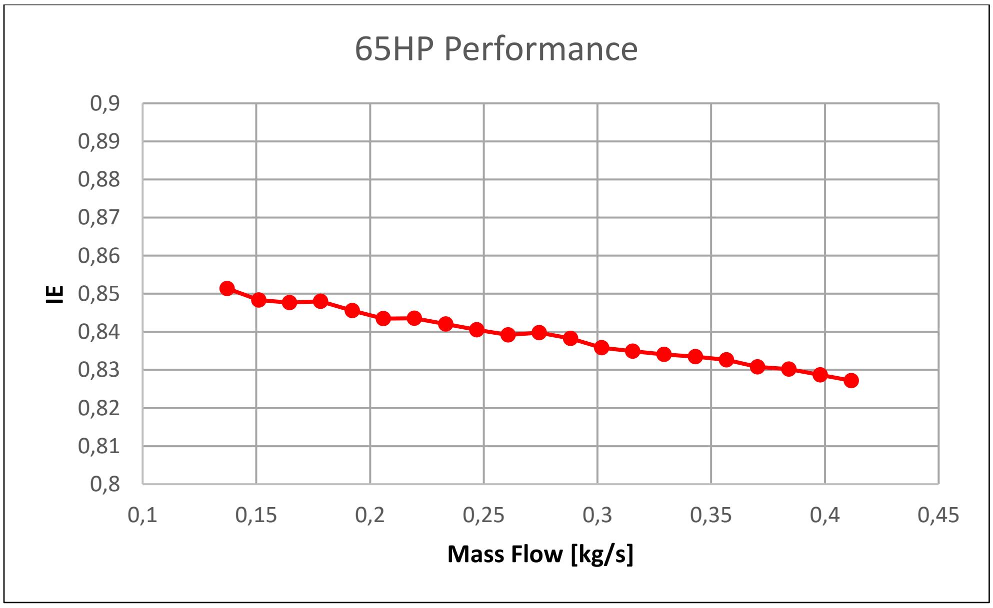


FIGURE 43 - MODEL 65HP ISENTROPIC EFFICIENCY

As for model V700, it has only been possible to evaluate its performance at a maximum of 50 °C as forward temperature. It shows a familiar trend as it slowly grows when the mass flow goes below design point conditions and then efficiency suddenly drops when cylinders start to be unloaded. The maximum value is slightly below 0.9 for around 0.3 kg/s, while for 800 kW it is around 0.845. The minimum value is reached for 100 kW but is below 15% load; what is interesting is the minimum value for 160 kW (20%) equal to 0.81. The trend is shown in Figure 44 and both these high-pressure models will be tried in the simulations since there is no univocal choice between one or the other. Please notice that IE is reported as function of mass flow.

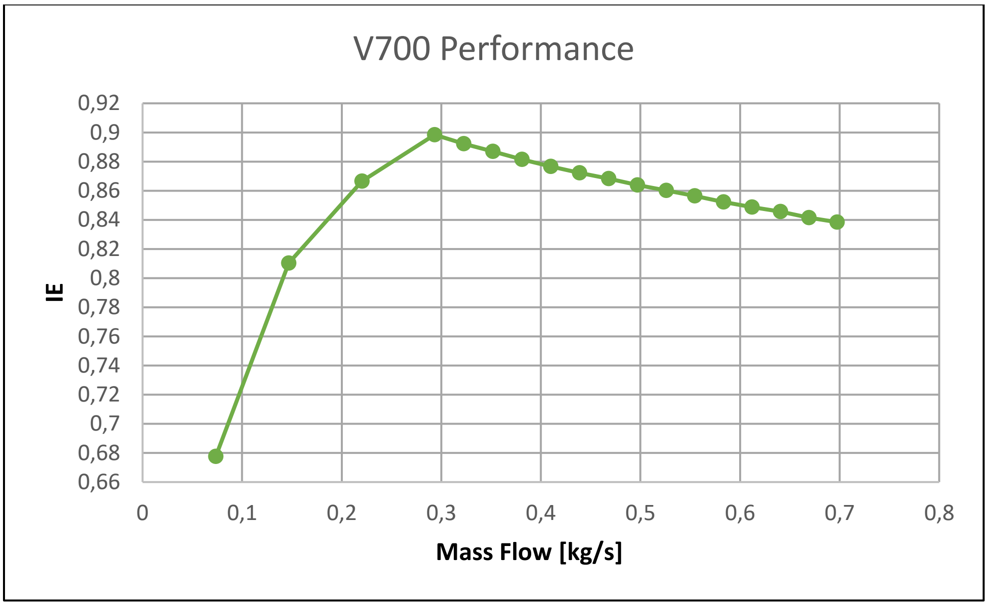


FIGURE 44 - MODEL V700 ISENTROPIC EFFICIENCY

For now, only isentropic efficiencies are shown but later all the results regarding COP and other parameters will be reported. Right next IE values are displayed for screw-type compressors.

<u>Screw Compressors</u>:

Low Pressure

When results for screw model MM have been calculated, three different control configurations have been considered as in one-stage case: variable speed drive alone, slide valve at fixed speed and variable speed along with slide valve. This analysis has shown that the improvement at part load given by variable speed is way more relevant than using slide valve only. In particular, as mass flow decreases, this difference becomes more and more significant. Near full load power, performance is basically the same for the various configurations, whereas they start to differ substantially at around 0.4-0.45 kg/s, where fixed speed becomes less competitive with the other two cases. These present very small variations, meaning that the impact of the slide valve while working at variable speed is quite limited. However, slide valve allows a small improvement, which goes from 0.8 to 2 points of IE maximum. This comparison can be seen in Figure 45 with a zoomed view of the upper zone in the following Figure. The only operating point for which fixed speed is better than variable speed (without slide valve) is above 0.6 kg/s, that is corresponding to more than 800 kW. For the two variable speed cases, in the graph, there is a sudden drop in efficiency before it increases again at around 0.25 kg/s. This is most likely due to an internal error in the software used for calculation, so that particular point should not be considered.

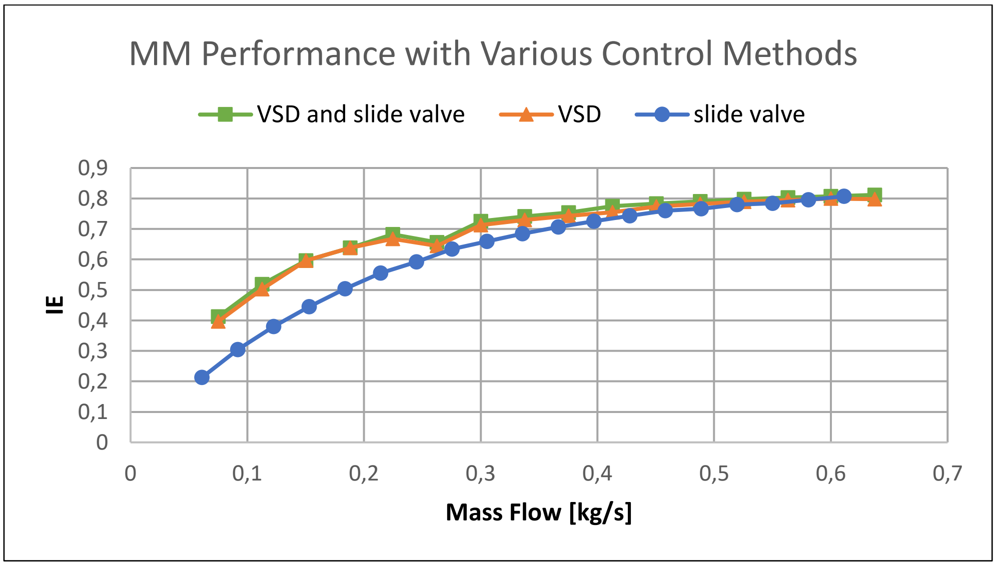


FIGURE 45 - MODEL MM ISENTROPIC EFFICIENCY

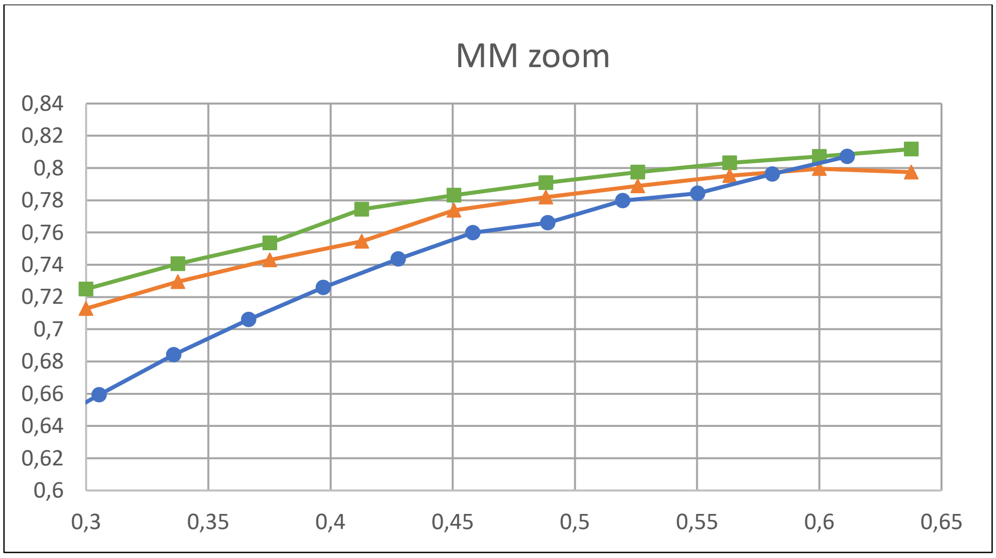


FIGURE 46 - ZOOM ON HIGH MASS FLOW AREA

The next graph displays compressor power required for different loads in comparison to the ideal case and for two different methods. The results shown are consistent with other obtained by several different studies on this subject. Please notice that the anomaly visible between 40% and 50% load is probably due to a computational error within the software itself. In fact, this singularity has been always found regardless of the screw model considered but still, it has been reported for completeness.

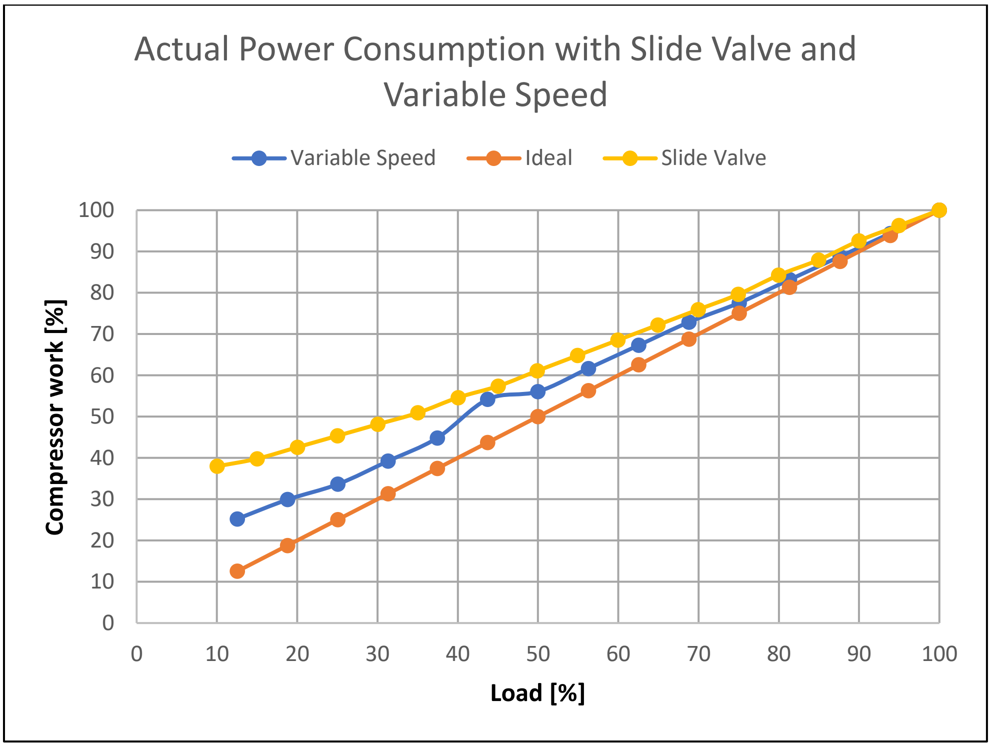


FIGURE 47 - PERCENTAGE OF COMPRESSOR POWER VERSUS LOAD FOR DIFFERENT CASES (MODEL MM)

High Pressure

For the high-pressure stage, all the analysis have shown that screw compressors would not be a very wise choice, since their efficiency when working with high temperature is quite low regardless of the capacity control method. In fact, among all the possible modulation systems considered, none achieves efficiency higher than 0.7. Full load IE is not satisfactory; this leads to thinking that screw models are more suited for lower temperature. The optimal range in the software was between -33°C and 33°C. As the results show, these model should find application in low pressure rather than in high pressure. However, sometimes it can be necessary to employ screw compressors for example due to very high mass flows or when compact and silent compressors are needed (reciprocating types usually needs more room and can be quite noisy). Hereby, the case with screw compressors will be considered anyway. Furthermore, besides the IE value, it is important to evaluate the trend and the differences between methods and configurations.

Figure 48 shows the efficiency curves for compressor model DM, which has the power size required by this application. As was done before, three cases have been considered at first: fixed speed with the possibility of changing the built-in volume ratio, variable speed with fixed volume ratio and the two combined. When analysing the first results, it was clear that the presence of a slide valve for high pressure made no difference either for fixed speed nor for variable speed and therefore from now on the slide valve will not be considered in high pressure. This has been verified testing also model CM and obtaining the same outcome. For low values, variable speed performs better but that is an area where neither of them should be used as the efficiency is already below 0.5. For different compressor models better suited for the temperature range, this trend should be verified in order to confirm these considerations. Unfortunately, for screw-types, not many models with these specifics are available and therefore it was not possible to validate the results. Anyway, the best choice with these data seems to be using variable speed, even though the two options are very close above 0.3 kg/s. However, since the difference is so little, both options will be considered next.

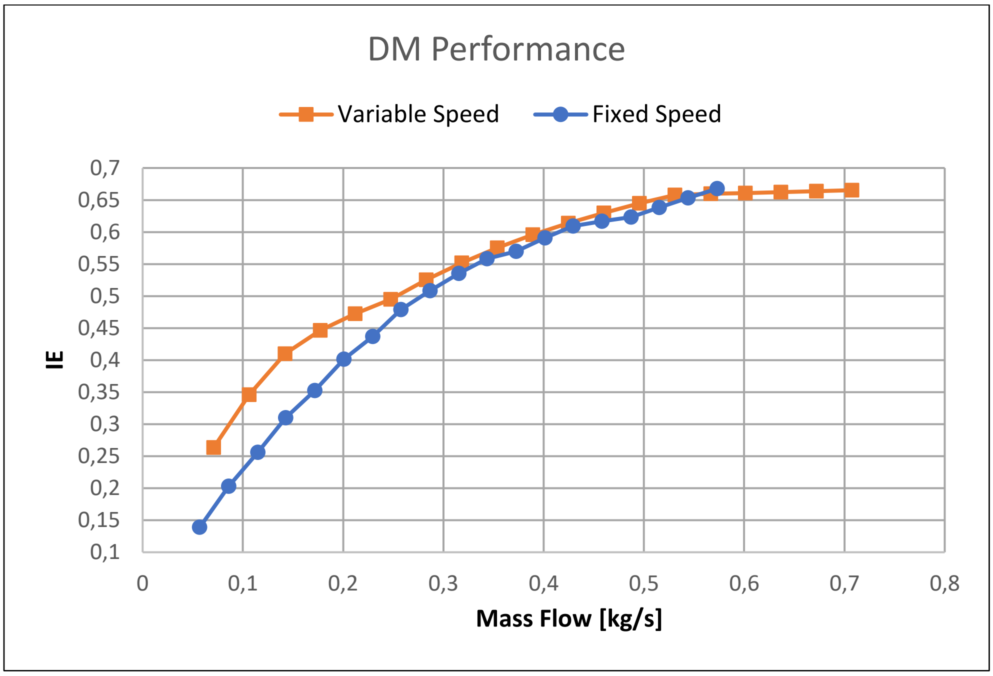


FIGURE 48 - IE OF MODEL DM IN HIGH PRESSURE

6.2 Cycle Analysis

6.2.1 First Case

In the first case, variable speed piston compressors are employed in both stages. Their IEs have already been shown in paragraph 6.1.2. These values were quite high so it is most likely to obtain a high COP as well. The two compressor models are V1100 for low pressure and V700 for high pressure. They have a similar trend of isentropic efficiencies, except that V1100 starts to unload cylinders sooner due to its larger capacity and therefore its inability to modulate mass flow with speed control in low range.

In Figure 49 below the cycle is reported in the T-s diagram for full load and 20% load (minimum) with details on the compression and condensation processes. In particular, it is clear that at part load condensation and evaporation temperature get closer leading to an improvement in cycle efficiency. Unfortunately, this has to face the decrease of isentropic efficiency, so COP will not be much higher than in full load operation. However, for medium loads IE is quite high and also condensation and evaporation are closer than for 100%. Therefore, in that range, COP can achieve a value of around 3.93, with the maximum reached for 40%. When the lowest groundwater mass flow is reached, the outlet groundwater temperature begins to rise and evaporation increases more than it does for higher loads. This leads also to an increased intermediate pressure because evaporation rises more than condensation lowers. It is important to underline that this is a consequence of how the intermediate pressure is set. In this case, it is always maintained such as to obtain the same pressure ratio for both stages. With other control methods, its trend would probably be different.

The lower rotational speed limit is reached at 40% for the low-pressure compressor and at 30% for the high stage. When this occurs, capacity modulation is obtained unloading cylinders and therefore isentropic efficiency is affected negatively. As a matter of fact, the efficiency drops a little before 500 rpm and that is due to the fact that there is no exact match between the volume of the real compressor and the one obtained from the model. However, it is clear to see that IE has a sharp decrement when the mass flow goes below 0.27 kg/s in low stage and 0.25 kg/s for the high stage.

In this case, the best area to operate the HP would be between 30% and 70% because COP is high and also quite stable. It is important to remember that its value could be overrated as many other losses are not accounted for and also considering that pumps and auxiliary are not included in the overall performance evaluation. Anyway, the lowest COP is 3.66 for full load operation, which are usually quite rare. Even the 20% efficiency is higher than full load, being above 3.7. As will be seen in the sensitivity analysis, that is not always the case if the forward temperature needed by DH network is higher than 70°C.

For these results, model V700 has been employed in high pressure but it is not entirely sure that it can work properly in that temperature range, as was previously explained. In the next analysis, another compressor will be considered along with V700 and that is 65HP particularly suited for those temperatures. Since it has a slightly lower efficiency, COP values with that compressor in high stage would probably be reduced. Anyway, this comparison will be clearer in the second case, when both will be operated along with a fixed speed compressor at low pressure.

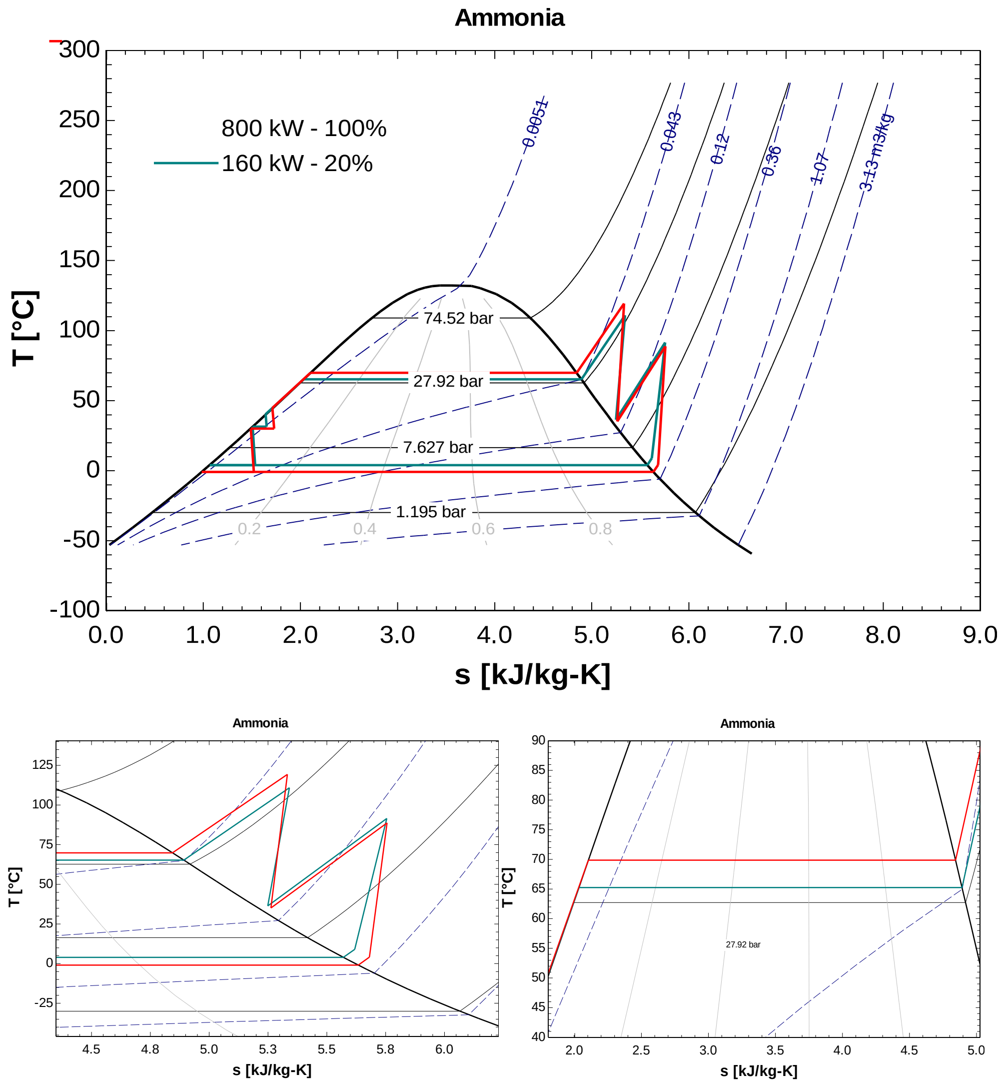


FIGURE 49 - CYCLE FOR 100% AND 20% LOAD AND ZOOM ON COMPRESSION AND CONDENSATION

Q_{cond}	p_{int}	COP	T_c	T_e	η_{is} LP	η_{is} HP
[kW]	[bar]	[-]	[°C]	[°C]	[-]	[-]
800	11.82	3.658	70.7	-0.9	0.848	0.846
720	11.84	3.706	70.5	-0.7	0.856	0.853
640	11.86	3.760	70.2	-0.4	0.865	0.860
560	11.89	3.821	69.9	-0.1	0.872	0.871
480	11.92	3.881	69.6	0.2	0.873	0.883
400	11.94	3.925	69.1	0.6	0.866	0.891
320	11.97	3.930	68.5	1.1	0.843	0.886
240	12.17	3.902	67.8	2.4	0.798	0.856
160	12.38	3.741	66.9	4.0	0.722	0.784
n_{LP}	n_{HP}	m_{LP}	m_{HP}	m_{DH}	m_{gw}	$T_{gw, out}$
[rpm]	[rpm]	[kg/s]	[kg/s]	[kg/s]	[kg/s]	[°C]
1192	1483	0.526	0.643	6.37	21.8	4
1068	1333	0.476	0.580	5.80	19.7	4
945	1183	0.425	0.516	5.10	17.7	4
822	1033	0.374	0.453	4.46	15.5	4
699	884	0.323	0.389	3.82	13.4	4
575	735	0.270	0.325	3.18	11.2	4
500	585	0.217	0.260	2.55	9.0	4
500	500	0.163	0.194	1.91	8.5	5.3
500	500	0.108	0.129	1.27	8.5	7.1

TABLE 5 - FIRST CASE RESULTS VARYING HEAT LOAD

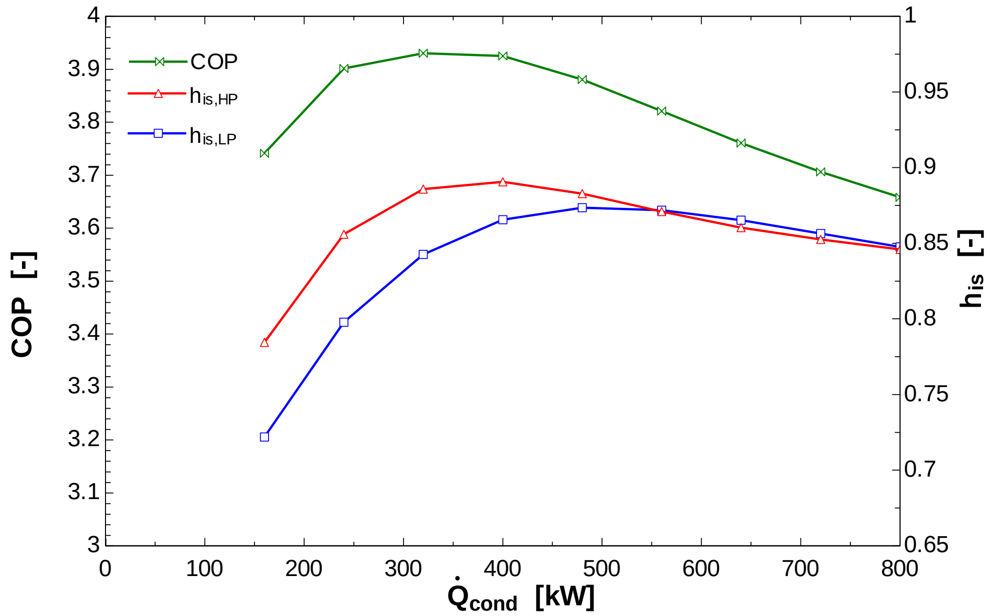


FIGURE 50 - COP AND IE AT PART LOAD
(VARIABLE SPEED)

6.2.2 Second Case

The second configuration considers the possibility of using a fixed speed compressor in low stage. For high pressure, two models are taken into account and their characteristics have been displayed in paragraph 6.1. Model 65HP is most suited for high pressure but does not reach the required capacity whereas V700 is bigger but is not recommended to work until 70°C in condensation. However, both have been tested in the model and the results are presented next. To simulate the fixed speed compressor, a new parameter was defined as *c* or fraction of loaded volume (and *C* as number of working cylinders): in full load that is 1 and at part load capacity is modulated with discrete steps. Condensation heat is calculated based on the mass flow elaborated in the low-pressure compressor.

With this configuration there is less possibility of controlling capacity, in fact, the minimum possible heat load is about 290 kW and the operating points are fewer than in the previous case. Moreover, also the variable speed high-pressure compressor is forced to work accordingly to the low stage compressor because its mass flow is fixed by the intercooler balance already seen in equation 5.13. It is clear that all these considerations lead to a more strict operational strategy. In this scenario, IE has opposite trends for the two stages: in low pressure, the efficiency always decreases; it starts at 0.816 for full load and goes down to 0.67 for 290 kW. The other compressor improves its performance from 0.81 to 0.84 (65 HP) when the heat load lowers.

Q_{cond}	p_{int}	COP	$V_{dot}LP$	c (C)	T_c	T_e	$T_{gw,\,out}$
[kW]	[bar]	[-]	[m^3/s]	[-]	[°C]	[°C]	[°C]
800	11.80	3.553	0.1592	1.000(6)	70.5	-0.9	4
676	11.83	3.563	0.1329	0.833(5)	70.1	-0.5	4
551	11.87	3.564	0.1067	0.667(4)	69.7	0.0	4
423	11.91	3.555	0.0802	0.500(3)	69.0	0.6	4
293	12.02	3.545	0.0537	0.333(2)	68.0	1.6	4.4
n_{LP}	n_{HP}	m_{LP}	m_{HP}	η_{is} LP	η_{is} HP	m_{DH}	m_{gw}
[rpm]	[rpm]	[kg/s]	[kg/s]	[-]	[-]	[kg/s]	[kg/s]
1000	1477	0.521	0.640	0.816	0.811	6.37	21.6
1000	1243	0.440	0.541	0.794	0.818	5.38	18.3
1000	1010	0.360	0.442	0.763	0.825	4.39	15.0
1000	772	0.276	0.340	0.721	0.833	3.37	11.5
1000	529	0.192	0.237	0.669	0.842	2.33	8.5

TABLE 6 - SECOND CASE RESULTS VARYING NUMBER OF LOADED CYLINDERS IN LP (65HP IN HIGH PRESSURE)

The results in Table 6 refer to compressor model 65HP for high pressure, but as was said, some simulations have been made also employing model V700. This last model has a higher efficiency in the entire range, but it decreases below 420 kW. However, COP stands higher in any case and achieves more than 3.6. In both situations, the overall performance is quite steady due to the fact that isentropic efficiency has opposite tendency for the two stages. Furthermore, it can be noticed that the IE drop in low pressure is much more relevant than the increase in high pressure. Nevertheless, the effect of the two temperatures getting closer compensates this difference. As a result COP trend is almost constant in the whole operating area. In both cases, the maximum performance difference is below 0.05.

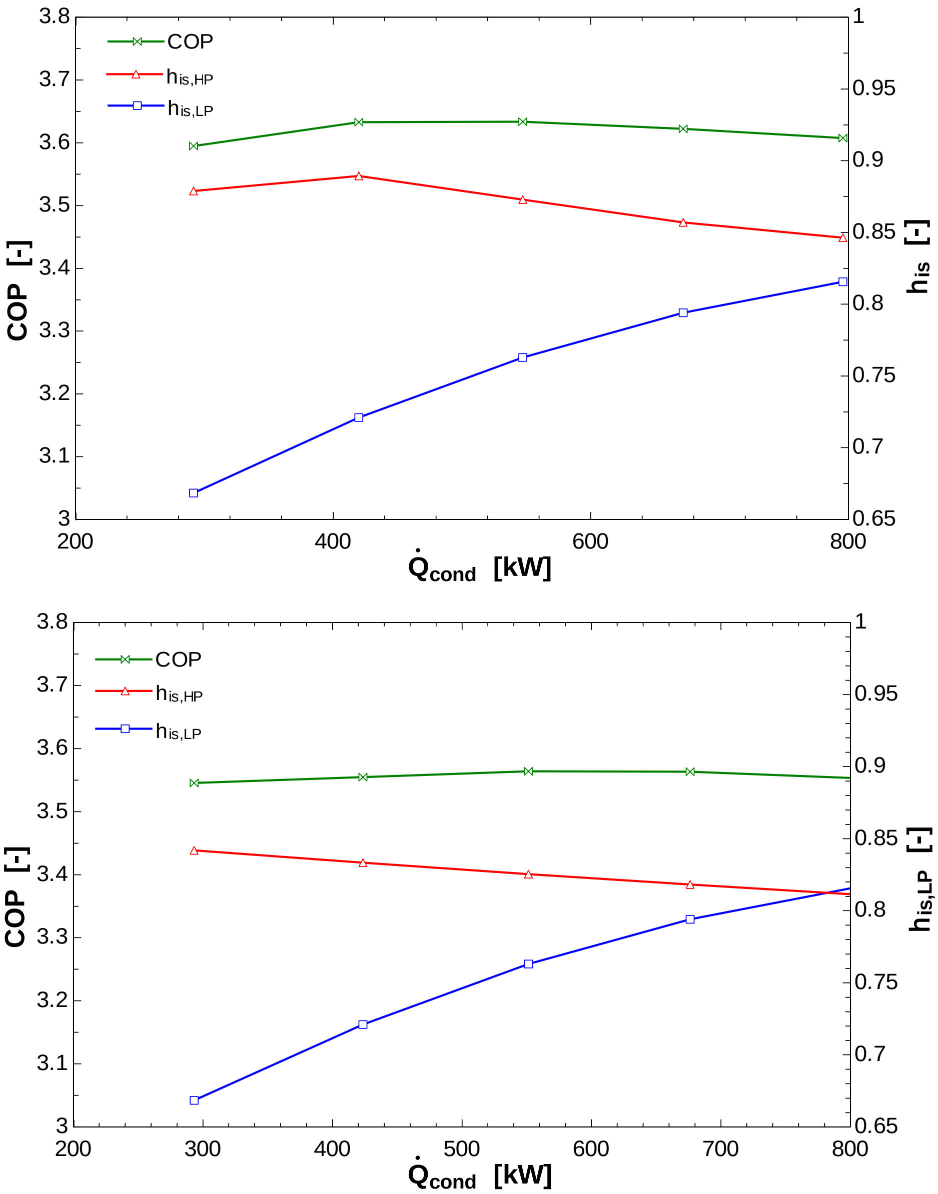


FIGURE 51 - COP AND IE TREND USING V700 (ABOVE) AND 65HP (BELOW) RESPECTIVELY

6.2.3 *Third Case*

In this third option, screw compressors are implemented in the cycle. Their efficiencies show quite different trends when compared to reciprocating. In fact, IE always lowers at part load. Usually, this drop is very slow for high loads and then it becomes sharp for very low heat demand. The compressors used in the model are MM for low pressure and DM for high pressure. For both these, the modulation is accomplished using variable speed along with a slide valve. In paragraph 6.2.4 the focus will be particularly on the advantages of using both capacity control methods together.

Through the previous analyses regarding isentropic efficiency, it is possible to notice that screw-type perform better with low pressure and temperature. As a further proof, the two models have a quite similar trend but the low stage compressor has always higher efficiency. Whereas one goes from 0.8 to 0.56, the other starts at 0.66 in full load to decrease below 0.43. From the results in Table 7, it would be recommended not to work below 400 kW very often because both efficiencies are very poor and COP is below 3. Therefore, screw compressors seem to be most suited to work with high loads, i.e. with high mass flows. In that range, the maximum COP is achieved for 800 kW and it is equal to 3.26. Please notice that these results have been obtained with a slightly different model thus condensation temperature might be a little different. Anyway, this deviation is not particularly relevant thus the outcomes are not substantially affected.

In this case, the modulation steps only reach 240 kW (30%). That happens because it is not possible to operate the system at 20% using only variable speed control; in that case, low-stage speed would be below 500 rpm. As a matter of fact, the system is actually able to operate for a lower heat load than the last shown in the table. That would be achieved running at 500 rpm and supplying around 185 kW. In case a lower heat load is required, modulation would only be possible using valves that open and close suction and discharge.

As for the external flows, it is important to notice that since the limits have been set to different values, one reaches the lowest limit and the other does not. An interesting thing to investigate is what would happen when the lower mass flow limit for groundwater was reached, so that value has been set accordingly. For the DH side, mass flow modulation is possible down to the last heat load, whereas for groundwater it stops at 8.5 kg/s in the two last rows. Given that the energy balance must be verified, outlet temperature increases and this allows evaporation temperature to grow faster.

Q_{cond}	p_{int}	COP	T_c	T_e	η_{is} LP	η_{is} HP
[kW]	[bar]	[-]	[°C]	[°C]	[-]	[-]
800	11.74	3.260	70.0	-0.8	0.795	0.662
720	11.72	3.248	69.4	-0.6	0.778	0.653
640	11.71	3.227	68.9	-0.3	0.760	0.639
560	11.69	3.191	68.3	0.0	0.741	0.618
480	11.68	3.132	67.6	0.4	0.718	0.587
400	11.68	3.038	66.9	0.8	0.686	0.546
320	11.75	2.898	66.1	1.6	0.639	0.493
240	11.98	2.703	65.5	3.1	0.566	0.426
n_{LP}	n_{HP}	m_{LP}	m_{HP}	m_{DH}	m_{gw}	$T_{gw, out}$
[rpm]	[rpm]	[kg/s]	[kg/s]	[kg/s]	[kg/s]	[°C]
2942	2938	0.503	0.618	6.37	20.9	4
2620	2642	0.452	0.555	5.73	18.8	4
2299	2342	0.400	0.492	5.10	16.7	4
1979	2040	0.349	0.428	4.46	14.5	4
1661	1735	0.297	0.364	3.82	12.4	4
1345	1428	0.244	0.299	3.19	10.2	4
1027	1116	0.191	0.236	2.55	8.5	4.4
708	801	0.139	0.173	1.91	8.5	6.1

TABLE 7 - THIRD CASE RESULTS VARYING HEAT LOAD

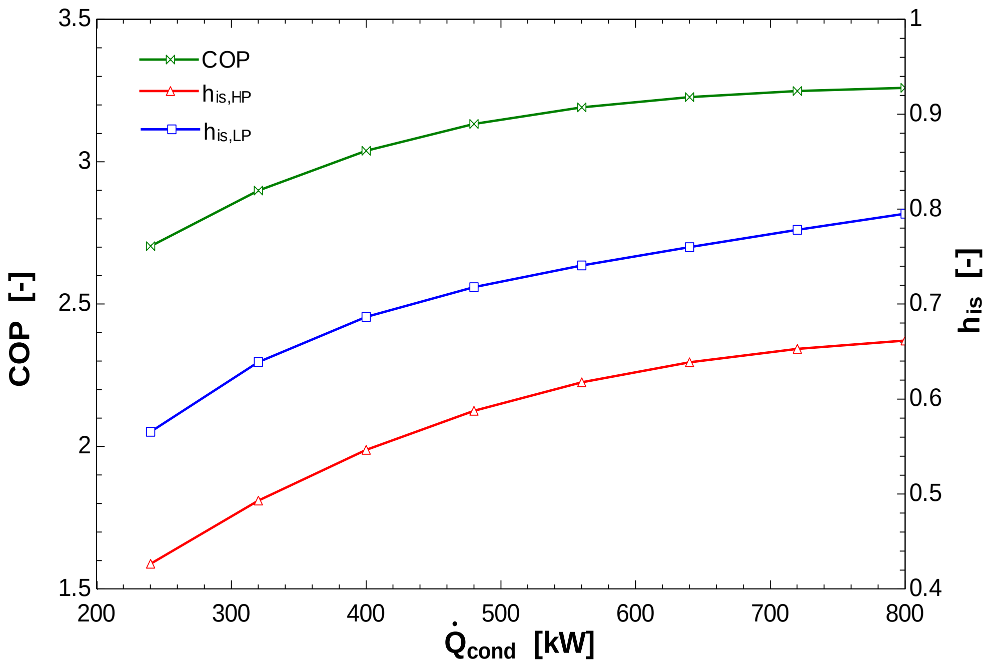


FIGURE 52 - COP AND IE AT PART LOAD
(VARIABLE SPEED AND SLIDE VALVE)

6.2.4 Fourth Case

In the last configuration, the focus is on detecting the improvements in the performance guaranteed by using slide valve and/or variable speed. In order to do so, the model has been run using fixed speed compressors with a slide valve and the results are reported in Table 8. Then, other data have been gathered using variable speed without slide valve and also the ones from the third simulation are available for comparison. For now, only the data from the table below will be commented but a deeper comparison will be made in the discussion chapter.

When comparing these results with the previous case, it can be stated that the full load COP is basically the same and it starts to differ more and more when the load lowers. The highest difference is reached for the lowest load and it is around 0.5. Anyway, the main deviations are registered below 50-60%, while above those loads COP is not very different. In fact, IEs are quite similar for high mass flows: low-pressure efficiency is a bit lower but high pressure seems to be a little higher. On the other hand, for low flows, variable speed perform much better especially in low stage and thus in that range COP is consistently higher than in this case.

Hence, fixed speed seems to perform well near the design point conditions and, in particular, employing slide valve is much more relevant at low pressure as can be seen analysing IE results in paragraph 6.1. In fact, in the high stage, there is no significant improvement using slide valve neither with variable speed nor with fixed speed. Therefore, the only influence that slide valve has is to improve efficiency in the low-pressure compressor. This will be better discussed in Chapter 8. However, COP ranges from 3.26 to 2.23 in the lowest load. It stands under 3 below 560 kW and therefore this configuration should be used when the load is quite steady and always above 60-70% because the efficiency for low loads would not be satisfactory.

Q_{cond}	p_{int}	COP	T_c	T_e	η_{is} LP	η_{is} HP
[kW]	[bar]	[-]	[°C]	[°C]	[-]	[-]
800	11.74	3.257	70.0	-0.8	0.771	0.678
720	11.73	3.217	69.5	-0.6	0.752	0.656
640	11.71	3.167	68.9	-0.3	0.727	0.633
560	11.70	3.094	68.2	0.0	0.694	0.604
480	11.69	2.980	67.6	0.4	0.647	0.566
400	11.69	2.807	66.8	0.9	0.582	0.513
320	11.81	2.569	66.1	1.9	0.495	0.442
240	12.07	2.231	65.4	3.5	0.381	0.349
n_{LP}	n_{HP}	m_{LP}	m_{HP}	m_{DH}	m_{gw}	$T_{gw, out}$
[rpm]	[rpm]	[kg/s]	[kg/s]	[kg/s]	[kg/s]	[°C]
2950	2950	0.503	0.621	6.37	20.9	4
2950	2950	0.450	0.556	5.73	18.7	4
2950	2950	0.397	0.491	5.09	16.5	4
2950	2950	0.344	0.426	4.46	14.3	4
2950	2950	0.290	0.361	3.82	12.1	4
2950	2950	0.235	0.296	3.18	9.8	4
2950	2950	0.180	0.231	2.55	8.5	4.8
2950	2950	0.124	0.165	1.91	8.5	6.6

TABLE 8

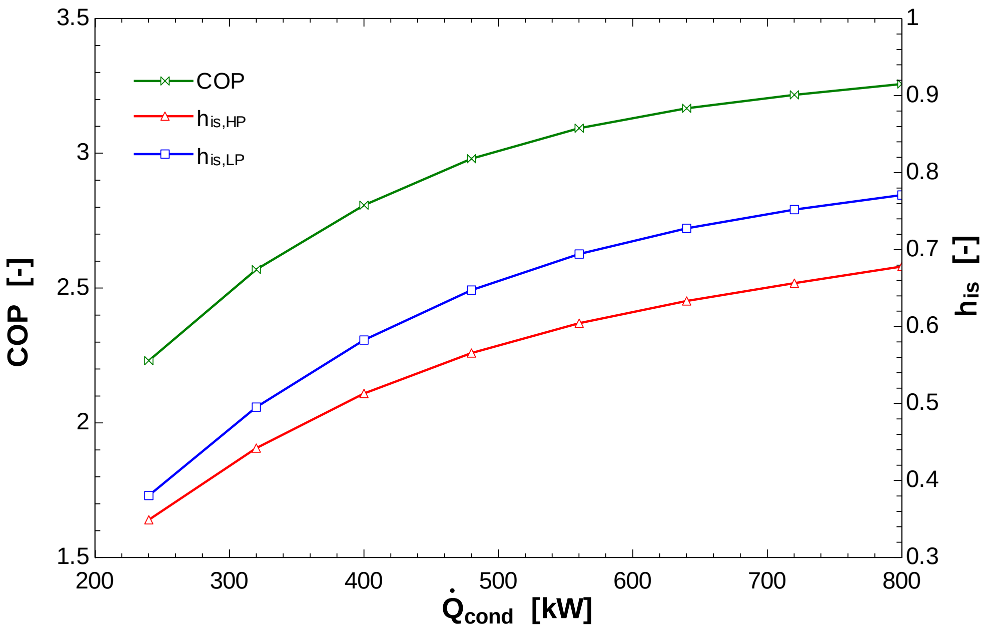


FIGURE 53 - COP AND IE AT PART LOAD
(FIXED SPEED AND SLIDE VALVE)

7 *Sensitivity Analysis*

The sensitivity analysis considers different scenarios. By definition, it means to evaluate new results obtained varying the initial assumptions. In this case, those assumptions are related to sink and source temperature. The most relevant analysis for the considered system would involve the forward temperature, that can consistently vary during the year. To accommodate the varying demand, different values of that temperature should be considered. On the other hand, source temperature in that particular site is almost constant throughout the year. For this reason, groundwater temperature will be fixed at the same value, whereas several temperatures for district heating will be considered. The chosen range goes from 60°C to 80°C with an increment of 4°C. The results are shown next and they refer to the first configuration, i.e. piston compressors running at variable speed.

In Figure 54 it is possible to see the heat load when DH mass flow varies and for different forward temperature. It is clearly visible that the mass flow varies more when the required temperature is lower. This is obviously to a lower temperature lift for DH water. Indeed, in order to supply lower power mass flow has to decrease much if the temperature lift is low. That can be easily understood with the following energy balance:

$$Q_{DH} = m_{DH} * c_{p,w} * \Delta T_{DH} \; [kW] \qquad (7.1)$$

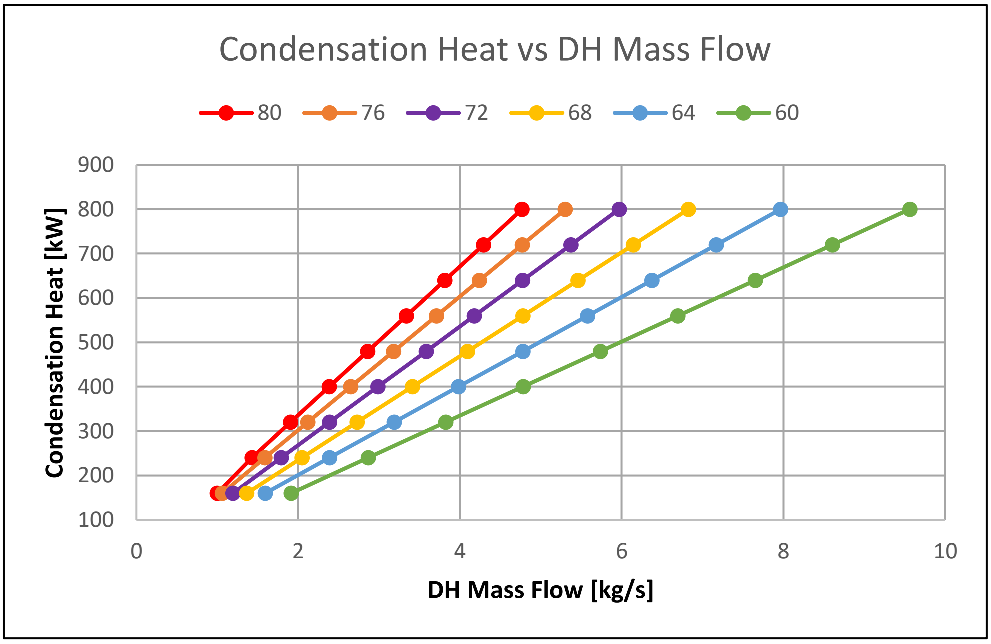


FIGURE 54 - HEAT LOAD FOR DIFFERENT MASS FLOWS AND FORWARD TEMPERATURE

In the next figure, COP is displayed for the different temperatures. The highest value is achieved when T_{DH} is 60°C and stands at around 4.3. It can be noticed that the higher the temperature gets, the lower is the increase in efficiency at part load (before cylinders are unloaded). For 80°C, this difference is roughly 0.2, whereas for 60°C that is almost 0.4. Furthermore, the highest COP is achieved for different heat loads. While for the low forward temperature that is between 30% and 40%, for higher values maximum COP shifts to 50% (400 kW). The explanation will be provided when analysing Figure 56. In addition, it can be seen that the case for 80°C is the only one in which full load COP (3.4) is not the lowest. In fact, for 160 kW, that is below 3.4. This happens because condensation and evaporation temperature does not get close as in the other cases, lowering the overall enthalpy lift and thus the work required. This effect is not consistent enough to compensate the drop in isentropic efficiency and therefore a lower COP value is reached.

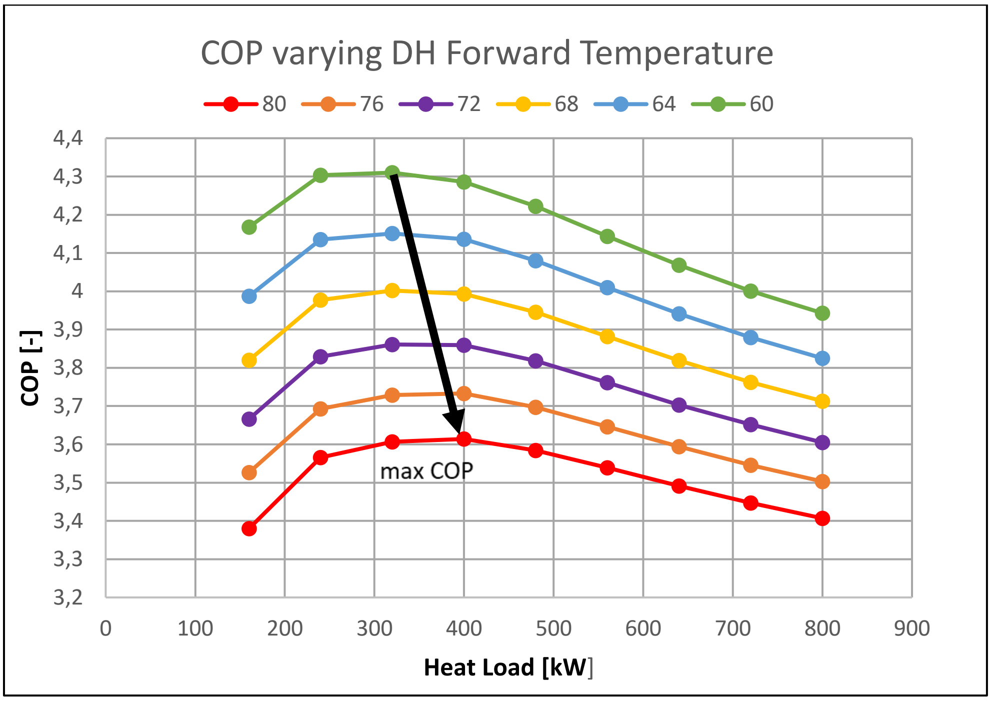


FIGURE 55 – COP FOR DIFFERENT FORWARD TEMPERATURE

In Figure 56, the condensation temperature is shown for every load. When the temperature requested by the DH network is lower, the condensation temperature is free to vary much. The same thing does not happen for higher forward temperature. Since the DH mass flow decreases as heat lowers, the temperature where the pinch point is set, i.e. the point where condensation starts, tends to increase a bit. Given that there must be a minimum temperature difference in order to transfer heat, it is not possible to go below a certain value. For 80°C, this constraint is more rigid because condensation temperature is actually lower than DH forward temperature. Moreover, the outlet temperature from the compressor depends only on IE (given that inlet and outlet source temperature are constant) and that affect both *desuperheating heat* and condensation temperature. These are bonded together because the fraction of *desuperheating heat* must be such as that condensation is above DH temperature when the phase change starts. Hereby, condensation temperature variation is contained compared to the other cases. Consequently, COP increases less because condensation is not much reduced at part load. On the other hand, for lower forward temperature, condensation can vary more and thus COP benefits from this. The highest improvement is in fact achieved for 60°C in DH outlet. In that case, condensation is able to vary from 65°C to less than 60°C.

Given that the heat demand can be very variable throughout the year, the sensitivity analysis is very useful since it considers both heat load variations and forward temperature variations. With these results, it is possible to couple supply and demand in the best possible way and the system can be evaluated more completely. Once the demand is known from previous data, some kind of strategy can be developed. In particular, considering the tendency of demand and the graphs from the sensitivity analysis, many considerations can be made regarding how often the HP should work for every load or how much the forward temperature should vary during the year. With one of these parameters known throughout the heating season, it is possible to obtain the system COP and thus detecting some ways to improve it. For example, if the forward temperature happens to be very high and quite constant in the operating period, the heat load should be between 40% and 60% for most hours. Instead, for lower forward temperature, the heat load could often go at 30%-40% since in those points COP achieves the highest value. On the other hand, if the heat load is known, the system efficiency can be evaluated depending on the forward temperature variations in the analysed period. In Figure 57, the COP trend is reported when forward temperature and DH mass flow vary.

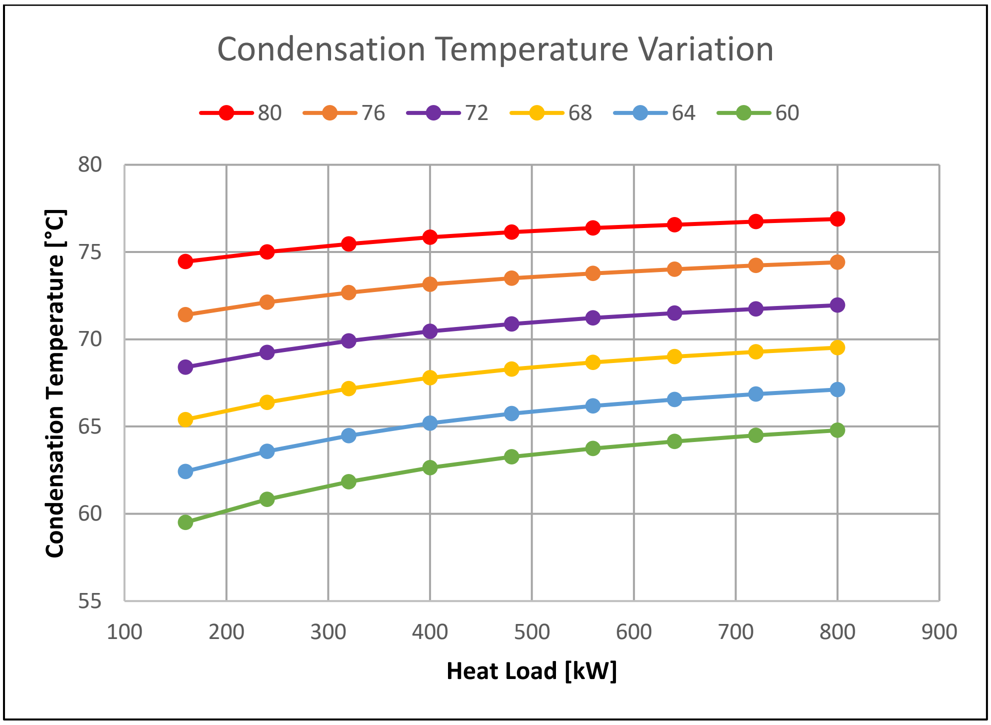


FIGURE 56 – CONDENSATION TEMPERATURE AT PART LOAD VARYING FORWARD TEMPERATURE

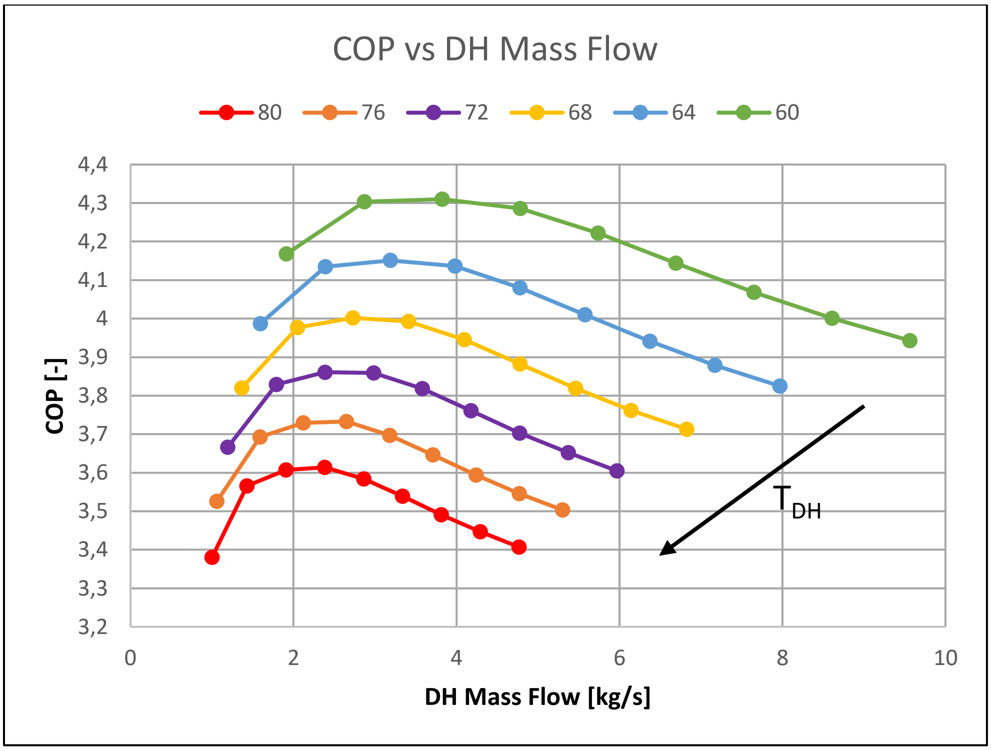


FIGURE 57 – COP FOR DIFFERENT MASS FLOWS AND FORWARD TEMPERATURE

8 Discussion

Isentropic Efficiency Results

Reciprocating Compressors

The first results have been obtained for the isentropic efficiency of the various piston compressors. These have shown the important improvement that variable speed can produce, especially if part load operations happen quite often. At first, compressors have been evaluated for the one-stage cycle and then also for two-stage. Between the two no major differences were found but it was noticed that compressors performed slightly better in one stage due to a higher pressure ratio. This fact does not lead to a higher COP in one-stage because the benefits that the two-stage offers are way more relevant. Anyhow, in both cases, the major advantage is that IE grows as the mass flow decreases. This happens only above a certain value because when cylinders start to be unloaded the efficiency drops. With fixed speed instead, the compressor efficiency always lowers and also the possibility of capacity control is narrower since it is performed by unloading cylinders. Furthermore, the data have confirmed that in general IE is higher for high stage compressors and that is due to a higher mean operating pressure. However, they show exactly the same trend but with different values. The only model that behaved differently was model 65HP because it is able to operate in a wide area with all cylinders loaded and modulating its displacement through variable speed only.

In case intermediate pressure would vary much, an analysis with different intermediate temperatures have been performed testing the performance of variable speed model V1100 in low pressure. This has shown that the isentropic efficiency varies mostly for high mass flows; the deviation is not very relevant anyway being 0.01 at the maximum and it is less important if the temperature increases rather than if it lowers. Besides, with the chosen control for intermediate pressure, the corresponding temperature never varies more than 1-2°C, while the range for the analysis is +/- 5°C. For this reason, the system is not consistently affected by these variations.

Screw Compressors

Regarding screw compressors, it has been found that IE trend is always decreasing at part load but with some differences when using different control methods. In fact, one of the objectives was to find out whether it is better to choose variable speed or a slide valve. In all the models that have been considered, variable speed resulted to be preferred between the two. The performance is similar only for very high mass flow, at least for low pressure; in high pressure, the deviations assume particular relevance for lower mass flows than in low pressure. Another interesting analysis concerns the use of a slide valve along with variable speed. The results vary from low stage to high stage: in the low-pressure compressor the improvement in efficiency goes between 0.04 and 0.08 in one-stage and between 0.01 and 0.02 in two-stage. This deviation between these two cases is due to the different pressure ratio (higher in one-stage). Hereby, the use of a slide valve would be strongly recommended in a one-stage system, while for two-stage, the choice would not be univocal. Instead, in the high-pressure compressor, the presence of a slide valve is not necessary at all along with variable speed because for two different models (DM and CM) the electric requirements are the same. However, it is always recommended in case of a fixed speed compressor. In fact, this allows important savings as it has been shown in Figure 47 back in the results chapter. At last, it should be noticed that for one-stage, fixed speed with slide valve performs better than variable speed without a valve for high mass flows and the combination of the two systems achieve the highest efficiency following the best trend depending on the mass flow value. On the other hand, in two-stage variable speed should be always adopted since its efficiency is higher in the entire range.

COP Results

As could be expected from the IE values, the best choice for the cycle in terms of COP is using reciprocating compressors. Furthermore, those would have to be running with variable speed in both stages. If COP was not the only factor to consider, using fixed speed compressor would still be a valid choice as their lower efficiency stands above 0.65 even at the minimum load. In fact, variable speed drive improves the efficiency with no doubt but it would require much more maintenance, system complexity and in general higher capital and O&M costs. It is also important to underline that employing fixed speed compressors would narrow the possibility of modulation. That means that the lowest achievable load would be around 300 kW and also the operating points would be less due to the fact that the low-pressure compressor control can be performed only by unloading cylinders. For this reason, only a discrete modulation is possible and this will affect also the mass flow in high pressure, given that it is bounded by the intercooler balance.

A summary of the system performance with reciprocating compressors is shown in Figure 58. COP is higher using variable speed compressors (average of 3.81) but also the other configurations show quite good performance with an average value of 3.56 when considering 65HP and 3.62 with V700. Moreover, these last two have quite steady overall efficiency, so the system is able to operate properly in the entire area. Unfortunately, modulation with a fixed speed compressor cannot go below 290 kW as it is the minimum reachable heat load by unloading the maximum number of cylinders. However, if the demand characteristics fit these requirements, employing a fixed speed could be considered as a possible smart choice. On the other hand, if demand is very variable and often goes below that limit, it would be recommended to use variable speed for both stages.

Between the two compressors used in high pressure, the best efficiency is achieved with model V700 but, as previously explained, it is not sure that it can actually work until 70°C in condensation. Model 65 HP instead has a smaller capacity but it is most suited for high-pressure values, therefore employing this compressor should not be a problem; this could be done for example using a parallel compound or in smaller plants.

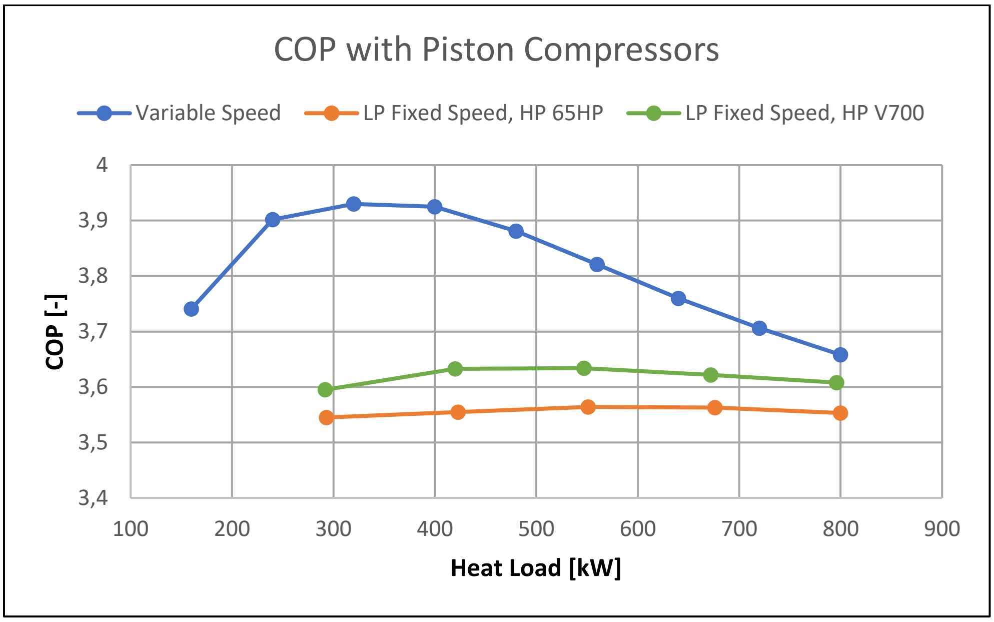
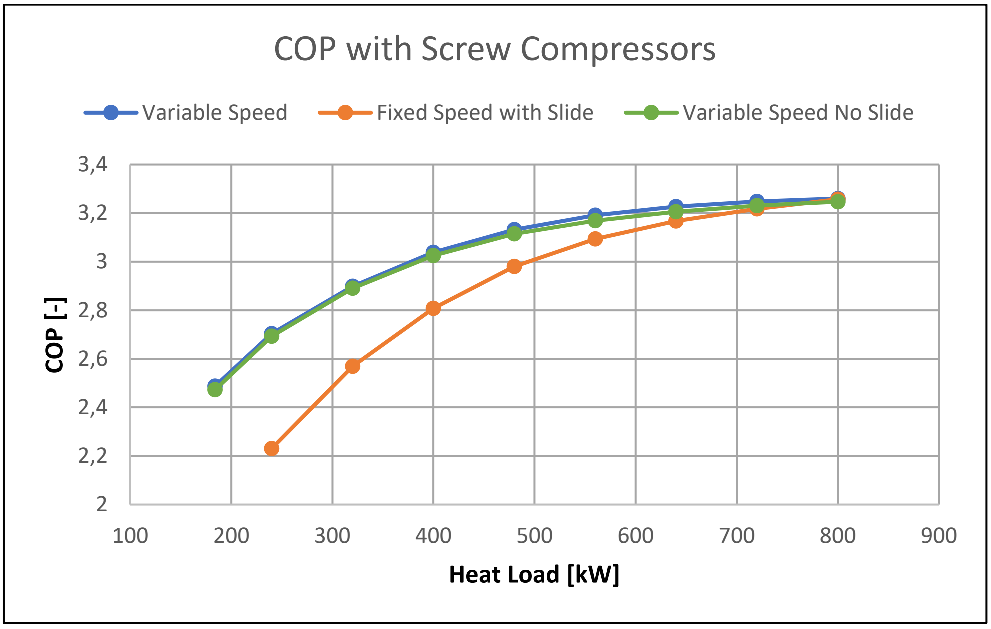


FIGURE 58 - COP USING RECIPROCATING COMPRESSORS

As for screw compressors, COP is quite lower but still satisfactory being above 3 from 50% to 100% using variable speed drive. Remembering that the employment of a slide valve is not very relevant, it is easy to explain why the two curves for variable speed are so close. In fact, the only improvement in COP is registered for high loads but the maximum deviation is about 0.02, so not very important. The maximum COP is 3.26 and is achieved at full load with model MM and DM both running at variable speed and with slide valves. At part load, the COP trend follows the efficiencies and therefore it is always decreasing. At first, it lowers very slowly but then it drops quite fast. There is no improvement at part load like it happened with reciprocating.

Having said that, the main focus can be put on the differences between variable and fixed speed. In this comparison, variable speed performs way better than fixed speed. When choosing the configuration for screw compressors, the main question was whether to choose variable speed or a slide valve in order to modulate capacity. Given these results, the answer should be evident: the choice between variable speed or fixed speed is much more relevant than implementing or not a slide valve. Using this valve is only interesting in low pressure and even so, it does not produce important increments in overall performance.



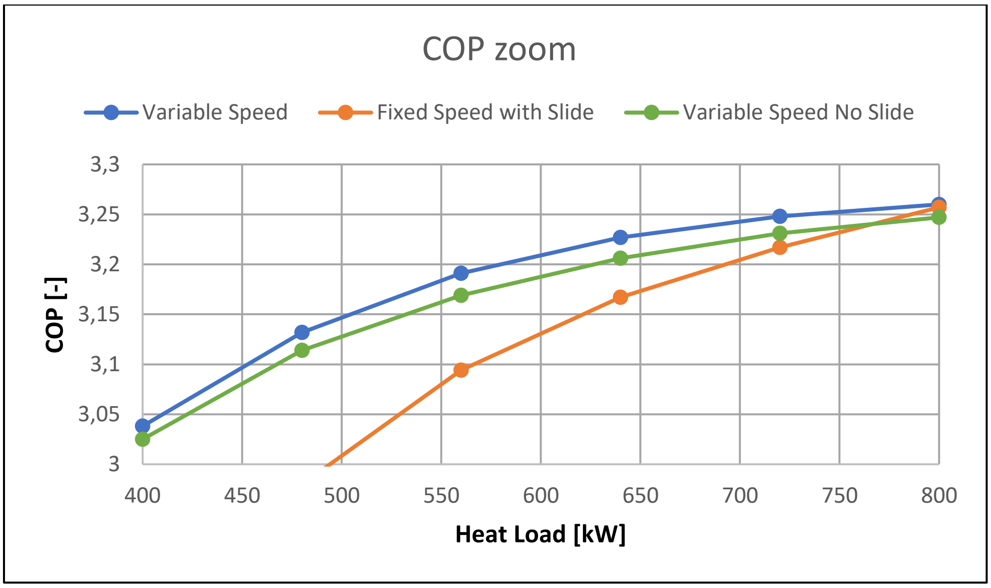


FIGURE 59 - COP USING SCREW COMPRESSORS AND ZOOMED VIEW

9 Conclusion

The purpose of this project was to build a model that can simulate the performance of a large ammonia heat pump. Some of the input data were taken from a plant that is actually running in the harbor district of Copenhagen. Of course, the layout of the real plant is much more complex than the one implemented in the model, therefore the results might be different. However, the main goal was to gather efficiency data for real compressors, implement them into the model and test the system with various configurations, compressor types and capacity control methods.

All the considered compressors have been tested as close as possible to the operating conditions of the cycle. For reciprocating compressors, variable speed and fixed speed were taken into account, with the first solution having always better performance. In particular, model V1100 starts with an isentropic efficiency of 0.82 (for more than 800 kW) and reaches 0.88. With cylinders unloading, the lower load is supplied with IE of 0.72. On the other hand, when it is running at fixed speed its modulation is possible for 5 points only (6 cylinders) and with IE ranging from 0.83 to 0.68. Model V700 has shown more or less the same behavior, while model 65HP manages to modulate capacity only varying speed and thus its efficiency always grows. This is possible because its maximum load is lower.

For screw compressors, the focus has been put on using slide valve, variable speed or the two combined. The outcomes have resulted quite interesting and different between cases. In low pressure, using a slide valve instead of variable speed is not recommended but employing it along with variable speed can produce an improvement of 0.01-0.02 in IE. Instead, in high pressure, the presence of a slide valve seems to be superfluous since it does not produce significant savings either with variable speed or fixed speed. However, in high pressure screw compressor performance is worse than in low pressure; variable and fixed speed displayed similar efficiency above 0.3 kg/s with an average deviation of around 0.01 in favor of variable speed. Below that mass flow, variable speed performs way better but quite poorly anyway.

The results suggest that for this system reciprocating compressors running at variable speed are the best possible choice as they can consistently improve their performance at part load thanks to the inverter. The maximum achievable COP has been found to be 3.93 for 320 kW (40% load), whereas the minimum for that configuration is 3.66 in full load operations. As for the second reciprocating configuration, COP resulted satisfactory (above 3.5 in average) and very steady but with a lower possibility of capacity modulation due to the fixed speed compressor adopted in low pressure.

When analysing the system performance with screw compressor, it has been noticed that it is clearly lower than with reciprocating and moreover the trend is always decreasing at part load. Their best fit would probably be when the heat load is often high and quite constant during the year. In average, COP is 3 for variable speed and 2.9 for fixed speed. The possibility of capacity control is narrower respect to piston compressors since the minimum heat load of 20% cannot be reached with the considered methods. The employment of a slide valve has not resulted decisive while using variable speed. In fact, it seems to improve a bit only low-stage isentropic efficiency but in high stage its effect is neglectable. When selecting the capacity control method, it has much more relevance whether to choose variable or fixed speed rather than the presence of a slide valve. The performance deviation is particularly important for low loads. In general, fixed speed for screw compressors is not to be preferred because it shows a similar efficiency only near full load operations.

The sensitivity analysis showed a much more rigid scenario in case the required forward temperature is higher. This involves both condensation temperature and DH mass flow variations, which are quite contained for example for 80°C as required temperature. Comparing the different curves it can be stated that the load for which the maximum COP is achieved shifts from 40% to 50% when T_{DH} increases. The maximum of all COPs is equal to 4.3 and is obtained for 60°C. With 80°C, the system efficiency range from 3.4 to 3.6.

9.1 Future Work

Due to lack of time, some issues have not been fully addressed, therefore there is an opportunity to improve the study in the future. Some of the possible subjects are suggested next. First, compressor data could be gathered on a larger scale in order to verify whether the trend stays the same or suffers some kind of variations depending on the operating conditions. In particular, the effects of the slide valve in high pressure should be studied more and on a wider range of screw-types. Furthermore, a deeper analysis would be needed for the other components of the cycle such as pumps and heat exchangers. Lastly, an interesting examination would involve the possibility of using HFO fluids in this system. These fluids could avoid the problem of excessive pressure and temperature at the outlet of compressors due to the shape of their saturation curve.

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Appendix A – EES Models

```
$reference ammonia IIR

"ONE STAGE DESIGN POINT model"

"exchange surfaces"
"A_cond=104 [m2]"
"A_sc=28"
"A_c=50"
"A_ds=26"
"A_evap=14 [m2]"

sh=5
sc=25 "[°C]"

"ground water"
Tg_in=10.5 [°C]
Tg_out=4 [°C]

"water for heating"
T_DH_out=70
T_DH_in=40

Q_dot_cond=800 [kW]
Q_dot_cond+Q_dot_oil=Q_dot_ev+W_comp

"input"
R$='ammonia'
T_e=-1 [°C]
T_c=70 [°C]
eta_is=0.8
eta_vol=0.8
"eta_vol=1.02*exp(-0.063*PR)" "piston fucntion of PR Pierre"
"eta_vol=-0.01*PR+0.94" "screw"
eta_elm=0.95
Q_dot_oil=1 [kW]

p_e=pressure(R$,T=T_e,x=0)
p_c=pressure(R$,T=T_c,x=0)
PR=p_c/p_e

"CYCLE"   "finds m_dot=0.5416 kg/s"
"node1 - start compression"
T[1]=T_e+sh
p[1]=p_e
h[1]=enthalpy(R$,p=p[1],T=T[1])
s[1]=entropy(R$,p=p[1],h=h[1])
v[1]=volume(R$,p=p[1],h=h[1])
V_dot_inlet=m_dot*v[1]

V_dot_inlet=n/60*V*eta_vol "finds V compressor for a certain speed"
vi=v[1]/v[2]
```

```
"n=1000" "fixed speed piston"
n=1200 "piston slow"
"n=1500" "piston fast, not used [rpm]"
"n=2950" "screw"

"node 2 - end compression"
sis[2]=s[1]
p[2]=p_c
his[2]=enthalpy(R$,p=p_c,s=sis[2])
Tis[2]=temperature(R$,p=p_c,h=his[2])
eta_is=(h[1]-his[2])/(h[1]-h[2])
s[2]=entropy(R$,p=p_c,h=h[2])
T[2]=temperature(R$,p=p_c,h=h[2])
v[2]=volume(R$,p=p_c,h=h[2])
V_dot_discharge=m_dot*v[2]

W_id=m_dot*(his[2]-h[1])
W_comp=m_dot*(h[2]-h[1]) "[kW]"
W_elm=W_comp/eta_elm
Q_dot_oil=m_dot*(h[2]-h_cool[2])
T_cool[2]=temperature(R$,p=p[2],h=h_cool[2])

"node 3"
p[3]=p_c
T[3]=T_c
h[3]=enthalpy(R$,p=p[3],x=1)
s[3]=entropy(R$,p=p[3],x=1)

"node 4 - end condensation"
p[4]=p[3]
T[4]=T_c
h[4]=enthalpy(R$,p=p[4],x=0)
s[4]=entropy(R$,p=p[4],x=0)

"node 5 - subcooling"
p[5]=p[4]
T[5]=T[4]-sc "temperature(R$;p=p[5];h=h[5])"
h[5]=enthalpy(R$,p=p[5],T=T[5])
s[5]=entropy(R$,p=p[5],T=T[5])

"node 6 - expansion valve"
h[6]=h[5]
p[6]=p_e
T[6]=temperature(R$,p=p[6],h=h[6])
s[6]=entropy(R$,p=p[6],h=h[6])
x[6]=quality(R$,p=p_e,h=h[6])

"node 7"
p[7]=p[6]
h[7]=enthalpy(R$,p=p[7],x=0)
T[7]=temperature(R$,p=p[7],x=0)
s[7]=entropy(R$,p=p[7],x=0)
```

```
"node 8"
p[8]=p_e
T[8]=T_e
h[8]=enthalpy(R$,p=p[8],x=1)
s[8]=entropy(R$,p=p[8],x=1)
v[8]=volume(R$,p=p[8],x=1)


"node 9"
T[9]=T_e+sh
p[9]=p_e
h[9]=enthalpy(R$,p=p[9],T=T[9])
s[9]=entropy(R$,p=p[9],h=h[9])

Q_dot_cond=m_dot*(h[2]-h[5])

COP=Q_dot_cond/W_elm

"ground water"
c_pw=cp(water,p=1,T=Tg_in)
Q_dot_ev=m_dot_gw*c_pw*(Tg_in-Tg_out)   "finds m_dot_gw=26 kg/s"
Q_dot_ev=m_dot*(h[1]-h[6])

"evaporator"
DELTA3=Tg_out-T_e
DELTA2=Tg_int-T_e
DELTA1=Tg_in-T[1]
DELTATml_sh=(DELTA1-DELTA2)/ln(DELTA1/DELTA2)
DELTATml_evap=(DELTA2-DELTA3)/ln(DELTA2/DELTA3)

A_ev_sh=Q_dot_ev_sh/(U_ev*DELTATml_sh)
A_evap=Q_dot_evap/(U_ev*DELTATml_evap)
A_ev=A_ev_sh+A_evap

Q_dot_ev_sh=m_dot*(h[1]-h[8])
"Q_dot_evap=m_dot*(h[8]-h[6])"
Q_dot_ev=Q_dot_evap+Q_dot_ev_sh "verification that superheating area is not
relevant"

Q_dot_ev_sh=Q_dot_gw_sh
Q_dot_evap=Q_dot_gw_evap

Q_dot_gw_sh=m_dot_gw*c_pw_gw*(Tg_in-Tg_int)
Q_dot_gw_evap=m_dot_gw*c_pw_gw*(Tg_int-Tg_out)


U_ev=(5.326*Q_dot_ev+2103.9)/1000 "empirical"

"condenser"
c_pw_DH=cp(water,p=1,T=T_DH_out)
m_dot_DH=Q_dot_cond/(c_pw_DH*(T_DH_out-T_DH_in))  "finds m_dot_DH=6.4
kg/s"

Q_dot_ds=m_dot*(h[2]-h[3])
```

```
Q_dot_c=m_dot*(h[3]-h[4])
Q_dot_sc=m_dot*(h[4]-h[5])

Q_dot_ds=Q_dot_ds_DH
Q_dot_c=Q_dot_c_DH
Q_dot_sc=Q_dot_sc_DH

Q_dot_ds_DH=m_dot_DH*c_pw_DH*(T[20]-T[21]) "find T21"
Q_dot_c_DH=m_dot_DH*c_pw_DH*(T[21]-T[22]) "find T22"

T[20]=T_DH_out
T[23]=T_DH_in
DELTAT_ds2=T_c-T[21]
DELTAT_ds1=T[2]-T[20]
DELTAT_ds_ml=(DELTAT_ds1-DELTAT_ds2)/ln(DELTAT_ds1/DELTAT_ds2)

DELTAT_c2=T_c-T[22]
DELTAT_c1=T_c-T[21]
DELTAT_c_ml=(DELTAT_c1-DELTAT_c2)/ln(DELTAT_c1/DELTAT_c2)

DELTAT_sc2=T[5]-T[23]
DELTAT_sc1=T_c-T[22]
DELTAT_sc_ml=(DELTAT_sc1-DELTAT_sc2)/ln(DELTAT_sc1/DELTAT_sc2)

"HTC"
U_ds=U*0.3
U_c=U
U_sc=U*0.3
U=(0.5884*Q_dot_cond+175.55)/1000

"Condenser areas"
Q_dot_ds_DH=U_ds*A_ds*DELTAT_ds_ml
Q_dot_c_DH=U_c*A_c*DELTAT_c_ml
Q_dot_sc_DH=U_sc*A_sc*DELTAT_sc_ml

A_cond=A_ds+A_c+A_sc
```

```
$reference ammonia IIR

"TWO STAGE OFF DESIGN model"

"Vary Qcond in parametric table to obtain part load results"

"exchange surfaces found from design point model: subcooling, condensation,
desuperheating, evaporation"
"A_cond0=133 [m2]"
A_sc=32
A_c=70
A_ds=31
A_evap0=20 [m2]

sh=5
“sc=25 [°C]”
"T[7]=T_DH_in+DELTAT_pinch"

"ground water inlet and outlet temperature"
Tg_in=10.5 [°C]
Tg_out_set=4 [°C]

"water for heating, outlet is fixed at 70"
T_DH_out=70
T_DH_in=40

"input data"
R$='ammonia'
"T_e=-1 [°C]"
"T_e=Tg_out-DELTAT_pinch"
DELTAT_pinch=5

"T_c=70 [°C]"
"eta_is_LP=0.8" "design point piston, screw 0.8"
"eta_is_HP=0.8" "screw 0.7"
eta_elm=0.95 "electric motor efficiency fixed at 95, from data was higher than 98%"
Q_dot_oil1=1 [kW]
Q_dot_oil2=1.2 [kW]

"Q_dot_cond=800" "design point"

"Q_dot_cond+Q_dot_oil1+Q_dot_oil2=Q_dot_ev+W_LP+W_HP" "cycle energy
balance, it is used to verify but it cannot be in the equations because conflicts with
compressor power"

"compressor data"
gamma=1.310
esp=(gamma-1)/gamma

"c_LP=1" "fraction of loaded cylinders"
V_LP=0.0095 [m^3] "piston 1200 0,009264, screw 2950 0,003391 m3"
"compressor displacement in low pressure"
m_dot_LP=eta_vol_LP*c_LP*V_LP*n_LP/60/v[1] "mass flow at low pressure"
```

```
"c_HP=1" "fraction of loaded cylinders"
V_HP=0.00334 [m^3] "piston 1500 0,003328, screw 2950 0,001544 m3" "must be fixed, and rpm LP and HP are related through intercooler balance"
m_dot_HP=eta_vol_HP*c_HP*V_HP*n_HP/60/v[3] "mass flow at high pressure"
"RECIPROCATING compressor"
n_LP=max(V_dot_LP/(eta_vol_LP*V_LP)*60, 500)
"n_LP=1000 [rpm]" "speed for fixed speed case"
"n_LP=1200 [rpm]"
n_HP=max(V_dot_HP/(eta_vol_HP*V_HP)*60, 500)
"n_HP=1500 [rpm]" "speed for variable speed"

"% mass flow LP" x=m_dot_LP/0.5238
"% mass flow HP" y=m_dot_HP/0.6412

"ISENTROPIC EFFICIENCY PISTON"
"LP"
"V1100 varspeed" eta_is_LP=-5.1355*m_dot_LP^4 + 11.903*m_dot_LP^3 - 10.017*m_dot_LP^2 + 3.4805*m_dot_LP + 0.4491
"V1100 varspeed vs %mass flow" "eta_is_LP=-1.8222*x^4+5.4721*x^3-5.9667*x^2+2.6862*x+0.4491"
"V1100 fixspeed 1000 rpm" "eta_is_LP=-0.718*m_dot_LP^2 + 0.9593*m_dot_LP + 0.5109"
"HP"
"65HP" "eta_is_HP = 0.0225*m_dot_HP^2 - 0.0954*m_dot_HP + 0.8632"
"V700" eta_is_HP = -8.2198*m_dot_HP^4 + 17.04*m_dot_HP^3 - 12.868*m_dot_HP^2 + 4.0403*m_dot_HP + 0.4434
"V700 vs % mass flow" "eta_is_HP = -1.94553020*y^4 + 5.78220414*y^3 - 6.26050849*y^2 + 2.81812256*y + 0.443432624"

"VOLUMETRIC EFFICIENCY PISTON"
eta_vol_LP=1.02*exp(-0.063*PR_LP) "Pierre"
eta_vol_HP=1.02*exp(-0.063*PR_HP)

"SCREW compressor"
"n_LP=2950" "speed for screw compressor"
"n_HP=n_LP"

v_i_LP=v[1]/v[2]  "built in volume ratio low pressure ,v_i around 2"
pi_i_LP=v_i_LP^gamma "*"

v_i_HP=v[3]/v[4]
pi_i_HP=v_i_HP^gamma

"ISENTROPIC EFFICIENCY SCREW"
"LP"
"MM fixed with slide" "eta_is_LP=4.0185*m_dot_LP^3 - 6.374*m_dot_LP^2 + 3.6948*m_dot_LP + 0.0136"
"MM variable no slide" "eta_is_LP= -20.489*m_dot_LP^4 + 33.214*m_dot_LP^3 - 19.95*m_dot_LP^2 + 5.648*m_dot_LP + 0.0775"
"MM variable with slide" "eta_is_LP= -14.97*m_dot_LP^4 + 25.101*m_dot_LP^3 - 15.898*m_dot_LP^2 + 4.8658*m_dot_LP + 0.134"
"HP"
"EM fixed with slide" "eta_is_HP= -1.2257*m_dot_HP^2 + 1.5756*m_dot_HP + 0.0444"
```

```
"EM variable no slide" "eta_is_HP= 0.6397*m_dot_HP^3 - 1.4608*m_dot_HP^2 +
1.2922*m_dot_HP + 0.1353"
"EM variable with slide" "eta_is_HP= 0.9473*m_dot_HP^3 - 1.8783*m_dot_HP^2 +
1.483*m_dot_HP + 0.1299"

"VOLUMETRIC EFFICIENCY SCREW"
"eta_vol_LP=-0.01*PR_LP+0.94"
"eta_vol_HP=-0.01*PR_HP+0.94"
"pressure levels"
p_e=pressure(R$,T=T_e,x=0)
p_c=pressure(R$,T=T_c,x=0)
p_int=sqrt(p_e*p_c) "this is the formula to calculate the optimal intermediate
pressure"

"CYCLE"
"node1 - start LP compression"
T[1]=T_e+sh
p[1]=p_e
h[1]=enthalpy(R$,p=p[1],T=T[1])
s[1]=entropy(R$,p=p[1],h=h[1])
v[1]=volume(R$,p=p[1],h=h[1])
V_dot_LP=m_dot_LP*v[1] "low pressure volumetric flow"

"node 2 - end LP compression"
sis[2]=s[1]
p[2]=p_int
his[2]=enthalpy(R$,p=p_int,s=sis[2])
Tis[2]=temperature(R$,p=p_int,h=his[2])
eta_is_LP=(h[1]-his[2])/(h[1]-h[2])
s[2]=entropy(R$,p=p_int,h=h[2])
T[2]=temperature(R$,p=p_int,h=h[2])
v[2]=volume(R$,p=p[2],h=h[2])

PR_LP=p_int/p_e
W_LP_elm=W_LP/eta_elm "electric power absorbed by motor"
W_LP=m_dot_LP*(h[2]-h[1]) "compression power required"
Q_dot_oil1=m_dot_LP*(h[2]-h_cool[2])
T_cool[2]=temperature(R$,p=p[2],h=h_cool[2])

"node 3 - start HP compression"
p[2]=p[3]
T[3]=temperature(R$,p=p[3],x=1)+5
h[3]=enthalpy(R$,p=p[3],x=1)
s[3]=entropy(R$,p=p[3],x=1)
v[3]=volume(R$,p=p[3],h=h[3])
V_dot_HP=m_dot_HP*v[3] "volumetric flow high pressure"

"node 4 - end HP compression"
p[4]=p_c
sis[4]=s[3]
his[4]=enthalpy(R$,p=p[4],s=sis[4])
Tis[4]=temperature(R$,p=p[4],s=sis[4])
eta_is_HP=(his[4]-h[3])/(h[4]-h[3])
T[4]=temperature(R$,p=p[4],h=h[4])
```

```
s[4]=entropy(R$,P=p_c,h=h[4])
v[4]=volume(R$,p=p[4],h=h[4])

PR_HP=p_c/p_int "pressure ratio high pressure"
W_HP_elm=W_HP/eta_elm "electric power high pressure"
W_HP=m_dot_HP*(h[4]-h[3])  "[kW]" "compression power"
Q_dot_oil2=m_dot_HP*(h[4]-h_cool[4])
T_cool[4]=temperature(R$,p=p[4],h=h_cool[4])

"node 5 - start condensation"
p[5]=p_c
T[5]=T_c
h[5]=enthalpy(R$,p=p[5],x=1)
s[5]=entropy(R$,p=p[5],x=1)

"node 6 - end condensation"
p[6]=p[5]
T[6]=T_c
h[6]=enthalpy(R$,p=p[6],x=0)
s[6]=entropy(R$,p=p[6],x=0)

"node 7 - subcooling"
p[7]=p[6]
T[7]=T[6]-sc
h[7]=enthalpy(R$,p=p[7],T=T[7])
s[7]=entropy(R$,p=p[7],T=T[7])

"node 8 - expansion valve HP"
h[8]=h[7]
p[8]=p_int
T[8]=temperature(R$,p=p[8],h=h[8])
s[8]=entropy(R$,p=p[8],h=h[8])
x[8]=quality(R$,p=p_int,h=h[8])

"node 9 - saturated liquid at p_int"
p[9]=p[8]
h[9]=enthalpy(R$,p=p[9],x=0)
T[9]=temperature(R$,p=p[9],x=0)
s[9]=entropy(R$,p=p[9],x=0)

"node 10 - expansion valve LP"
h[10]=h[9]
p[10]=p_e
T[10]=T_e
s[10]=entropy(R$,p=p_e,h=h[10])
x[10]=quality(R$,p=p_e,h=h[10])

"node 11 - saturated liquid at p_c"
p[11]=p[10]
h[11]=enthalpy(R$,p=p[11],x=0)
T[11]=T_e
s[11]=entropy(R$,p=p[11],x=0)

"node 12 - end cycle"
```

```
p[12]=p_e
T[12]=T_e
h[12]=enthalpy(R$,p=p[12],x=1)
s[12]=entropy(R$,p=p[12],x=1)
v[12]=volume(R$,p=p[12],x=1)

"node 13 - same as 1"
T[13]=T_e+sh
p[13]=p_e
h[13]=enthalpy(R$,p=p[13],T=T[13])
s[13]=entropy(R$,p=p[13],h=h[13])

"open intercooler"
m_dot_LP*(h_cool[2]-h[9])=m_dot_HP*(h[3]-h[8])  "at p_int energy exchange
between high and low pressure mass flow"

Q_dot_cond=m_dot_HP*(h_cool[4]-h[7])

"Cycle efficiency"
COP=Q_dot_cond/(W_HP_elm+W_LP_elm)

"EXTERNAL FLOWS"
"ground water"
c_pw=cp(water,p=1,T=Tg_in)
Q_dot_ev=m_dot_gw*c_pw*(Tg_in-Tg_out) "energy balance ground water in and
out"
m_dot_gw=max(Q_dot_ev/(c_pw*(Tg_in-Tg_out_set)),m_dot_gw_min)
m_dot_gw_min=8.5 "40% volumetric flow in design point"
rho_w=1000 [kg/m3]
V_dot_gw=m_dot_gw/rho_w

Q_dot_ev=m_dot_LP*(h[1]-h[10]) "energy balance of absorbed heat from ground
water"

"evaporator"
DELTAT2=Tg_out-T_e
DELTAT1=Tg_in-T[1]
DELTATml=(DELTAT1-DELTAT2)/ln(DELTAT1/DELTAT2)

A_evap0=Q_dot_ev/(U_ev*DELTATml)

U_ev=(5.326*Q_dot_ev+2103.9)/1000 "empirical"

"condenser"
c_pw_DH=cp(water,p=1,T=T_DH_out)
"Q_dot_cond=m_dot_DH*c_pw_DH*(T_DH_out-T_DH_in)" "energy balance in and
out district heating water"
m_dot_DH=max(Q_dot_cond/(c_pw_DH*(T_DH_out-T_DH_in)),m_dot_DH_min)
m_dot_DH_min=1 [kg/s] "less than 40% but recirculation or smaller units, more
flexibility needed"
V_dot_DH=m_dot_DH/rho_w

"heat balance in condenser areas"
Q_dot_ds=m_dot_HP*(h_cool[4]-h[5])
```

```
Q_dot_c=m_dot_HP*(h[5]-h[6])
Q_dot_sc=m_dot_HP*(h[6]-h[7])

Q_dot_ds=Q_dot_ds_DH
Q_dot_c=Q_dot_c_DH
Q_dot_sc=Q_dot_sc_DH

"Q_dot_sc_DH=m_dot_DH*c_pw_DH*(T[22]-T[23])"
"Q_dot_ds_DH=m_dot_DH*c_pw_DH*(T[20]-T[21])" "find T21"
"Q_dot_c_DH=m_dot_DH*c_pw_DH*(T[21]-T[22])" "find T22"

T[20]=T_DH_out
T[23]=T_DH_in

DELTAT_ds1=T[4]-T[20]
DELTAT_ds2=T[5]-T[21]
DELTAT_ds_ml=(DELTAT_ds1-DELTAT_ds2)/ln(DELTAT_ds1/DELTAT_ds2)

DELTAT_c1=T[5]-T[21]
DELTAT_c2=T[6]-T[22]
DELTAT_c_ml=(DELTAT_c1-DELTAT_c2)/ln(DELTAT_c1/DELTAT_c2)

DELTAT_sc1=T[6]-T[22]
DELTAT_sc2=T[7]-T[23]
DELTAT_sc_ml=(DELTAT_sc1-DELTAT_sc2)/ln(DELTAT_sc1/DELTAT_sc2)

"HTC"
U_ds=U*0.3
U_c=U
U_sc=U*0.3
U=(0.5884*Q_dot_cond+175.55)/1000

"Condenser areas"
Q_dot_ds_DH=U_ds*A_ds*DELTAT_ds_ml
Q_dot_c_DH=U_c*A_c*DELTAT_c_ml
Q_dot_sc_DH=U_sc*A_sc*DELTAT_sc_ml

A_cond0=A_ds+A_c+A_sc "[m2]"

"Temp profile"
"Q_dot[7]=0"
"Q_dot[6]=Q_dot_sc"
"Q_dot[5]=Q_dot_sc+Q_dot_c"
"Q_dot[4]=Q_dot_cond"

"Q_dot[7]=Q_dot[23]"
"Q_dot[6]=Q_dot[22]"
"Q_dot[5]=Q_dot[21]"
"Q_dot[4]=Q_dot[20]"
```

Appendix B – GEA Compressors Data

pistone Valutazione

modello V 1100 R-717

I DATI OPERATIVI - condizione 1					
EVAPORATORE			**CONDENSATORE**		
EVAP temperatura	0	*°C*	Cond / Inter temperatura	30	*°C*
EVAP pressione	4,3	*bara*	Cond / Pressione Inter	11,7	*bara*
Surriscaldamento (Utile)	0	*K*	Sottoraffreddamento A Cond	0	*K*
Surriscaldamento (Non-Utile)	5	*K*	condensatore HOR	1016,7	*kW*
ASPIRAZIONE			**SCARICO**		
Perdita di aspirazione Linea	0	*bar*	Linea di scarico Perdita	0	*bar*
Pacchetto di aspirazione Perdita	0	*bar*	Pacchetto di scarico Perdita	0	*bar*
Pressione di aspirazione satura	4,3	*bara*	di scarico di pressione	11,7	*bara*
temperatura di aspirazione	5	*°C*	Temperatura di scarico	86,3	*°C*
Mass Flow	2835,1	*kg/h*	Mass Flow	2835,1	*kg/h*
Volume di flusso	838,3	*m³/h*	Volume di flusso	399,7	*m³/h*
Teorica cilindrata	954,8	*m³/h*			
PERFORMANCE DATA					
COMPRESSORE			**MOTORE**		
capacità	882,4	*kW*	tensione	400/3/50	*V/PH/Hz*
potenza	134,4	*kW*	Motor Size	177	*kW*
Prestazioni Factor	6,6	*COP*	Grandezza	280L	
velocità	1200	*RPM*	recinto	IP23	
Percentuale a pieno carico	100	*%*	IE Classificazione	IE2	
Min a carico parziale	33	*%*	Ampere di pieno carico	434	*A*
Tipo di azionamento	direct Drive		Numero di poli	4	
			Frequenza regolamentato	sì	
OLIO DI GESTIONE DEI DATI					
Separatore olio Tipo	Not Included		Olio Lancia Prima	29,1	*cc/h*
Separatore olio Diametro	-	*mm*	Olio Lancia Prima	8,7	*ppm*
Separatore olio Efficienza	-	*%*	Olio Gettare Dopo	9,3	*cc/h*
Temperatura ambiente	40	*°C*	Olio Gettare Dopo	2,8	*ppm*
Tipo di olio	Mineral - ISO VG 68				
AVVERTENZE					
nessuno					

Parte del carico Riepilogo - a velocità variabile

modello V 1100 R-717

Condizioni di funzionamento #1

velocità (RPM)	parte del carico (%)	cilindro Quantità	capacità (kW)	potenza (kW)	Prestazioni Factor (COP)
0,0 / 30,0 °C					
1200	100	6	882,4	134,4	6,57
1150	100	6	845,6	128,3	6,59
1100	100	6	808,8	122,2	6,62
1050	100	6	772,1	116,1	6,65
1000	100	6	735,3	110,1	6,68
950	100	6	698,5	104,1	6,71
900	100	6	661,8	98,2	6,74
850	100	6	625,0	92,3	6,77
800	100	6	588,2	86,4	6,81
750	100	6	551,5	80,6	6,84
700	100	6	514,7	74,8	6,88
650	100	6	478,0	69,0	6,92
600	100	6	441,2	63,3	6,97
550	100	6	404,4	57,6	7,02
500	100	6	367,7	51,9	7,08
500	83	5	305,2	44,1	6,93
500	67	4	246,3	36,7	6,72
500	50	3	183,8	28,8	6,38
500	33	2	121,3	21,0	5,79

GEA si riserva il diritto di verifica finale di tutte le valutazioni dei risultati.

Vite Valutazione

Modello MMR-M20S-28 R-717 Fase Alta
Thermosiphon 2,0 Vi

engineering for a better world

DEI DATI DI FUNZIONAMENTO - Condizione 1					
EVAPORATORE			**CONDENSATORE**		
Temperatura evaporatore	0,0	*°C*	Cond / Inter temperatura	30,0	*°C*
Pressione evaporatore	4,29	*bara*	Cond / Pressione Inter	11,66	*bara*
Surriscaldamento (Utile)	5,0	*K*	Sottoraffreddamento A Cond	0,0	*K*
Surriscaldamento (Non-Utile)	0,0	*K*	Condensatore HOR	808,2	*kW*
ASPIRAZIONE			**SCARICO**		
Perdita di aspirazione Linea	0,0	*bara*	Linea di scarico Perdita	0,0	*bara*
Pacchetto di aspirazione Perdita	-		Pacchetto di scarico Perdita	-	
Pressione di aspirazione satura	4,3	*bara*	Di scarico di pressione	11,7	*bara*
Dimensione della valvola di aspirazione di arresto	-		Valvola di scarico Dimensioni di arresto	-	
Temperatura di aspirazione	5,0	*°C*	Disch. Temp (100% / min)	71,0/72,5	*°C*
Flusso di massa	2216,5	*kg/h*	Flusso di massa	2216,5	*kg/h*
Volume di flusso	658,0	*m³/h*	Volume di flusso	295,9	*m³/h*
Teorica cilindrata	710,4	*m³/h*	Min Temp Cond	-	*°C*

PERFORMANCE DATA					
COMPRESSORE			**MOTORE**		
Capacità	698,9	*kW*	Tensione	-	*V / Ph / Hz*
Potenza	109,2	*kW*	Dimensioni del motore	-	*kW*
EER	6,4		Grandezza	-	
Velocità	2950	*RPM*	Efficienza	-	
Percentuale di pieno carico	100	*%*			
Vi - Fisso	2,0				

OLIO DI RAFFREDDAMENTO DEI DATI					
Olio di raffreddamento Tipo	Thermosiphon		Olio di raffreddamento media	R-717	
Olio di alimentazione Temp	50,0	*°C*	Liquid Temp (ingresso / uscita)	30,0/-	*°C*
Portata olio funzionale	31,5	*l/min*	OCHR (100%)	30,0	*kW*
Iniezione Portata olio	15,7	*l/min*	OCHR (min)	32,6	*kW*
Portata olio totale	47,2		Pompa olio Dimensione	-/-	*l/min/kW*

AVVERTENZE

1. Summer: Required discharge temperature 80 °C is higher than discharge temperature at full load 71 °C
Global: Operation with or without oil pump allowed, maximal oil pressure = 15,16 BarA

DATI DI PROGETTO			
Nome del progetto:		riferimento:	
Nome cliente:	New Contact	Preparato da:	Andrea Sacco
Proposta numero:		Date:	22/01/2018

Parte del carico Sommario - a velocità fissa

modello Bare Model MM

capacità (%)	Capacità di raffreddamento (kW)	Pe albero (kW)	evaporatore Temperatura (°C)	condensatore Temperatura (°C)	COP [albero] (COP)	EER [albero] (EER)
100	699	109	0,0	30,0	7,40	6,40
95	664	105	0,0	30,0	7,29	6,29
90	629	102	0,0	30,0	7,18	6,18
85	594	97	0,0	30,0	7,13	6,13
80	559	92	0,0	30,0	7,06	6,06
75	524	88	0,0	30,0	6,97	5,97
70	489	83	0,0	30,0	6,88	5,88
65	454	79	0,0	30,0	6,75	5,75
60	419	75	0,0	30,0	6,60	5,60
55	384	71	0,0	30,0	6,42	5,42
50	349	67	0,0	30,0	6,22	5,22
45	315	63	0,0	30,0	5,97	4,97
40	280	60	0,0	30,0	5,68	4,68
35	245	56	0,0	30,0	5,34	4,34
30	210	53	0,0	30,0	4,96	3,96
25	175	50	0,0	30,0	4,51	3,51
20	140	47	0,0	30,0	3,99	2,99
15	105	44	0,0	30,0	3,39	2,39
10	70	42	0,0	30,0	2,68	1,68

Appendix C – Sensitivity Analysis

Sensitivity

COP	p_int	Q_dot_cc	T_DH_ou	T_c	T[21]	T_e	eta_is_LI	eta_is_H	n_LP	n_HP	m_dot_LI	m_dot_HP	m_dot_DH	m_dot_g	Tg_out	c_LP	c_HP
[-]	[bar]	[kW]	[°C]	[°C]		[°C]	[-]	[-]	[rpm]	[rpm]	[kg/s]	[kg/s]	[kg/s]	[kg/s]	[°C]		
3,407	12,68	800	80	76,89	73,5	-0,8647	0,849	0,845	1186	1420	0,5175	0,6519	4,769	21,28	4	1	1
3,447	12,72	720	80	76,74	73,59	-0,6367	0,8577	0,8517	1063	1276	0,4681	0,5877	4,292	19,24	4	1	1
3,491	12,75	640	80	76,56	73,68	-0,3778	0,8662	0,8593	940,1	1131	0,4184	0,5234	3,815	17,2	4	1	1
3,539	12,8	560	80	76,37	73,8	-0,08094	0,8724	0,8698	817,6	987,5	0,3682	0,459	3,338	15,13	4	1	1
3,584	12,85	480	80	76,14	73,94	0,2646	0,8732	0,8817	695,1	844,1	0,3174	0,3946	2,861	13,04	4	1	1
3,614	12,9	400	80	75,85	74,08	0,674	0,8644	0,8903	572,3	700,5	0,2657	0,3295	2,384	10,91	4	1	1
3,607	12,97	320	80	75,46	74,17	1,168	0,8403	0,8867	500	556,6	0,2129	0,2638	1,907	8,741	4	0,8989	1
3,566	13,24	240	80	75	74,24	2,576	0,7947	0,8583	500	500	0,1598	0,1977	1,431	8,5	5,491	0,6388	0,8128
3,38	13,55	160	80	74,65	74,35	4,09	0,718	0,7876	500	500	0,106	0,1309	1	8,5	7,203	0,4	0,5231
3,503	12,33	800	76	74,41	70,42	-0,8883	0,8484	0,8454	1189	1445	0,5208	0,6483	5,303	21,5	4	1	1
3,546	12,36	720	76	74,23	70,5	-0,6608	0,8572	0,852	1065	1298	0,4711	0,5844	4,772	19,45	4	1	1
3,594	12,39	640	76	74,02	70,59	-0,4024	0,8658	0,8598	942	1151	0,421	0,5204	4,242	17,38	4	1	1
3,646	12,43	560	76	73,78	70,7	-0,1059	0,8722	0,8703	819,2	1005	0,3706	0,4564	3,712	15,3	4	1	1
3,697	12,47	480	76	73,5	70,83	0,2393	0,8733	0,8822	696,5	859,5	0,3195	0,3922	3,182	13,19	4	1	1
3,733	12,51	400	76	73,15	70,95	0,6486	0,8649	0,8904	573,5	713,7	0,2675	0,3275	2,651	11,04	4	1	1
3,729	12,56	320	76	72,68	71,04	1,143	0,8413	0,8864	500	567,4	0,2143	0,2621	2,121	8,843	4	0,9009	1
3,693	12,8	240	76	72,12	71,1	2,521	0,796	0,8574	500	500	0,1609	0,1963	1,591	8,5	5,43	0,6411	0,8298
3,526	13,06	160	76	71,4	71,01	4,048	0,7196	0,7862	500	500	0,1064	0,13	1,061	8,5	7,154	0,3995	0,5355
3,605	11,99	800	72	71,95	67,27	-0,9117	0,8478	0,8458	1191	1470	0,5241	0,6449	5,969	21,73	4	1	1
3,652	12,01	720	72	71,74	67,34	-0,6846	0,8567	0,8524	1067	1321	0,474	0,5813	5,372	19,65	4	1	1
3,703	12,04	640	72	71,5	67,42	-0,4267	0,8654	0,8602	943,9	1172	0,4237	0,5175	4,776	17,56	4	1	1
3,761	12,07	560	72	71,22	67,52	-0,1306	0,872	0,8708	820,9	1024	0,3729	0,4538	4,179	15,46	4	1	1
3,818	12,1	480	72	70,88	67,64	0,2143	0,8734	0,8826	697,9	875,7	0,3215	0,3899	3,582	13,33	4	1	1
3,859	12,13	400	72	70,46	67,75	0,6235	0,8654	0,8905	574,7	727,5	0,2692	0,3255	2,985	11,16	4	1	1
3,861	12,17	320	72	69,91	67,83	1,118	0,8422	0,886	500	578,8	0,2158	0,2604	2,388	8,945	4	0,903	1
3,829	12,38	240	72	69,24	67,9	2,465	0,7972	0,8564	500	500	0,1621	0,195	1,791	8,5	5,368	0,6435	0,8479
3,666	12,61	160	72	68,4	67,83	4,009	0,7212	0,785	500	500	0,1073	0,1291	1,194	8,5	7,107	0,4013	0,5479

3,713	11,66	800	68	69,52	64,05	-0,9346	0,8473	0,8462	1194	1496	0,5274	0,6416	6,826	21,95	4	1	1
3,762	11,67	720	68	69,28	64,12	-0,708	0,8561	0,8527	1070	1345	0,477	0,5782	6,143	19,86	4	1	1
3,819	11,69	640	68	69	64,19	-0,4505	0,865	0,8606	945,8	1194	0,4263	0,5148	5,461	17,75	4	1	1
3,882	11,71	560	68	68,68	64,28	-0,1549	0,8718	0,8712	822,5	1043	0,3752	0,4513	4,778	15,62	4	1	1
3,945	11,74	480	68	68,29	64,38	0,1897	0,8735	0,8829	699,3	892,6	0,3235	0,3877	4,096	13,47	4	1	1
3,993	11,76	400	68	67,8	64,48	0,5987	0,8658	0,8906	575,9	741,9	0,2709	0,3236	3,413	11,28	4	1	1
4,002	11,78	320	68	67,17	64,56	1,093	0,8431	0,8857	500	590,7	0,2172	0,2588	2,73	9,047	4	0,905	1
3,977	11,96	240	68	66,38	64,62	2,409	0,7985	0,8556	500	500	0,1632	0,1938	2,048	8,5	5,306	0,6459	0,8669
3,82	12,16	160	68	65,4	64,57	3,969	0,7228	0,7838	500	500	0,1082	0,1284	1,365	8,5	7,061	0,4031	0,5611
3,825	11,33	800	64	67,12	60,78	-0,957	0,8468	0,8465	1196	1500	0,5306	0,6385	7,967	22,17	4	1	1,016
3,879	11,34	720	64	66,86	60,83	-0,7309	0,8556	0,853	1072	1370	0,4798	0,5754	7,17	20,06	4	1	1
3,941	11,36	640	64	66,55	60,89	-0,4739	0,8646	0,8609	947,7	1216	0,4289	0,5122	6,374	17,93	4	1	1
4,01	11,37	560	64	66,18	60,97	-0,1787	0,8715	0,8717	824,2	1063	0,3775	0,449	5,577	15,78	4	1	1
4,08	11,39	480	64	65,74	61,06	0,1656	0,8735	0,8833	700,7	910	0,3255	0,3856	4,78	13,61	4	1	1
4,136	11,4	400	64	65,19	61,14	0,5744	0,8663	0,8907	577,1	756,8	0,2726	0,3218	3,984	11,4	4	1	1
4,151	11,41	320	64	64,47	61,21	1,069	0,8439	0,8853	500	603,1	0,2186	0,2573	3,187	9,148	4	0,907	1
4,135	11,56	240	64	63,57	61,27	2,353	0,7997	0,8547	500	500	0,1644	0,1927	2,39	8,5	5,244	0,6484	0,8867
3,987	11,73	160	64	62,43	61,25	3,929	0,7243	0,7827	500	500	0,1091	0,1276	1,593	8,5	7,014	0,4049	0,575
3,943	11,02	800	60	64,78	57,44	-0,9789	0,8463	0,8468	1198	1500	0,5337	0,6355	9,563	22,39	4	1	1,034
4,001	11,03	720	60	64,49	57,48	-0,7531	0,8551	0,8533	1074	1395	0,4826	0,5726	8,607	20,25	4	1	1
4,068	11,04	640	60	64,15	57,54	-0,4966	0,8641	0,8613	949,6	1239	0,4314	0,5097	7,651	18,1	4	1	1
4,144	11,04	560	60	63,74	57,6	-0,2018	0,8713	0,8721	825,8	1083	0,3797	0,4468	6,694	15,94	4	1	1
4,222	11,05	480	60	63,26	57,67	0,1422	0,8736	0,8836	702,1	927,9	0,3274	0,3836	5,738	13,75	4	1	1
4,286	11,05	400	60	62,64	57,75	0,5508	0,8667	0,8908	578,2	772,1	0,2743	0,3201	4,782	11,52	4	1	1
4,31	11,05	320	60	61,84	57,81	1,045	0,8447	0,885	500	615,8	0,22	0,2559	3,825	9,247	4	0,909	1
4,303	11,17	240	60	60,82	57,86	2,299	0,8008	0,8539	500	500	0,1655	0,1916	2,869	8,5	5,184	0,6508	0,9072
4,168	11,32	160	60	59,51	57,85	3,888	0,7258	0,7816	500	500	0,11	0,1269	1,913	8,5	6,967	0,4067	0,5895