HEAT PUMP SIZING CALCULATOR DEVELOPMENT IN HEAT RECOVERY FROM MOIST PROCESS EXHAUST AIR

Lappeenranta–Lahti University of Technology LUT

Master's Programme in Biorefineries 2023

Pasi Kolehmainen

Examiners: Assistant Professor Kristian Melin

Professor Tuomas Koiranen

Jaakko Kuusisto, M.Sc. (Tech)

ABSTRACT

Lappeenranta–Lahti University of Technology LUT

LUT School of Engineering Science

Biorefineries

Pasi Kolehmainen

Heat pump sizing calculator development in heat recovery from moist process exhaust air

Master's thesis

2023

68 pages, 22 figures, 12 tables and 8 appendices

Examiners: Assistant Professor Kristian Melin, Professor Tuomas Koiranen, Jaakko Kuusisto, M.Sc. (Tech)

Keywords: Heat pump, waste heat recovery, calculation tool.

The thesis studies the feasibility of waste heat recovery from moist process exhaust airs, and the development of a calculator tool for estimating the waste heat potential. The tool is used for estimating the potential sales cases for energy as a service solutions provider, Adven Oy

The thesis focused on calculating the energy recovery potential of moist air and the heat pump process in these applications. These findings were combined into a MS Excel tool to develop a calculation model to estimate heat recovery potential and the operating conditions of a heat pump. The calculator was then tested with two case studies and the results were compared to the simulation tools of equipment manufacturers.

The tool estimates the heat recovery potential reliably up to +45 °C entering waste air temperatures. The calculator is also able to estimate the COP of the heat pump with adequate accuracy, especially when the intended application of the tool is considered. The waste heat recovery processes have significant amount of process variables. The accuracy of the calculator can be improved by introducing the tool to a more advanced calculation environment. This allows calculation of broader datasets, for example hourly data from longer time periods.

TIIVISTELMÄ

Lappeenrannan–Lahden teknillinen yliopisto LUT

LUT Teknis-luonnontieteellinen

Biorefineries

Pasi Kolehmainen

Mitoitustyökalun kehittäminen lämpöpumpuille kosteiden prosessihönkien lämmön talteenottoon.

Kemiantekniikan diplomityö

2023

68 sivua, 22 kuvaa, 12 taulukkoa ja 8 liitettä

Tarkastajat: TkT, dosentti Kristian Melin, TkT, professori Tuomas Koiranen, DI Jaakko Kuusisto

Avainsanat: Lämpöpumppu, Lämmön talteenotto, Mitoitustyökalu

Diplomityössä tutkittiin lämmön talteenottomahdollisuuksia kosteista prosessien poistoilmoista, sekä kehitettiin mitoitustyökalu talteenottopotentiaalin kartoittamiseksi. Työkalu tulee käyttöön energiapalveluratkaisuja tarjoavan Adven Oy:n hankkeiden alkuvaiheen arviointiin.

Työssä tutustuttiin kostean ilman lämmön talteenottopotentiaalin laskentaan ja lämpöpumppujen toimintaan sovelluksessa. Näiden pohjalta kehitettiin laskentamalli lämpöpumpun ja lämmöntalteenottopotentiaalin arvioimiseksi MS Excel-pohjaisessa työkalussa. Kehitettyä työkalua testattiin kahteen todelliseen lämmöntalteenottohankkeeseen ja verrattiin sen tuottamia tuloksia laitevalmistajien simulointityökaluihin.

Laskentatyökalun todettiin ennustavan kostean ilman talteenottopotentiaalia hyvin +45 °C lämpötilaan saakka. Mitoitustyökalu pystyy ennustamaan lämmön talteenottoprosessiin liitetyn lämpöpumpun COP:n riittävällä tarkkuudella laskentatyökalun käyttötarkoitus huomioiden. Lämmön talteenottoprosessissa on runsaasti prosessimuuttujia. Työkalun tarkkuutta voidaan kehittää siirtämällä se kehittyneempään laskentaympäristöön ja muokata se laskemaan suurempia datakokonaisuuksia, esimerkiksi tuntikohtaista prosessidataa pitemmältä jaksolta.

ACKNOWLEDGEMENTS

I would like to express my deepest gratitude to Professor Kristian Melin and Professor Tuomas Koiranen for their invaluable support and feedback during the thesis process. I'm also extremely grateful to Mr Jaakko Kuusisto for his professional support, expertise and especially for workload management during the thesis project.

Special thanks also to Mr Antti Salonen and his colleagues at LU-VE for their assistance with the dimensioning of the air heat exchangers. I would also like to thank my colleagues at work and friends off work who have given me pleasant distraction from the thesis when it has been needed.

Last but not least this endeavor would not have been even remotely possible without the undying support of my family and close ones.

SYMBOLS AND ABBREVIATIONS

A	Area	[m^2]
c_p	specific heat	[J/kgK]
p	pressure	[bar, Pa]
ρ	density	[kg/m^3]
ρ_i	partial density of component i	[kg/m^3]
H	Enthalpy of mixture	[J]
h	specific enthalpy	[J/kg]
l_{ho}	heat of evaporation	[kJ/kg]
M	molar concentration	[mol/m^l]
$\dot{m}$	mass flow rate	[kg/s]
m	mass	[kg]
N	condensating density	[$kg/s\ m^2$]
$\dot{Q}$	heat flow	[kW]
Q	Heat	[kJ]
T	temperature	[K]
t	temperature	[°C]
U	overall heat transfer coefficient	[$W/(m^2K)$]
v	specific volume	[m^3/kg]
V	volume	[m^3]
W	Work	[kJ]

Constants

R	gas constant	8.314 J/mol K

Dimensionless quantities

x	vapor content
η	efficiency

Subscripts

i	partial component
A	partial component, water
B	partial component, dry air
C	cold
h	humidity
H	hot
k	relative to dry air
LM	logarithmic mean
u	outer wall
1	inflow
2	outflow

Abbreviations

CIP	Cleaning in Place
COP	Coefficient of Performance
GWP	Global Warming Potential
HFO	Hydrofluoroolefin
ODP	Ozone Depleting Potential
REACH	Registration, evaluation and authorization of chemicals
TFA	Trifluoroacetic acid

Table of contents

Appendices

1 Introduction

Adven provides off-balance sheet financed energy, steam and cooling solutions, district heating and chemical processing solutions in Northern Europe. It has more than 350 sites of operations providing over 5 TWh of energy annually (Adven oy, 2023). The purpose of this thesis is to develop a quick feasibility tool for the organization to use to calculate the energy savings potential of customers' process waste heat sources.

Global megatrends require companies to reduce their CO_2-emissions and to seek savings in energy costs and primary energy consumption (European Comission, 2023). Since high temperature waste heat sources are mostly harvested, the focus is increasingly turning to lower temperatures. This thesis focuses on the heat recovery of low temperature waste heat from industrial exhaust air, which is often overlooked as the temperature levels are not feasible to use with air-to-air or air-to-water heat exchangers.

Heat pumps are widely used in fluid-to-fluid heat recovery when the required temperature of the heat consumer is slightly higher than the temperature of the heat source. Industrial drying processes consume large quantities of energy. The temperature levels in such drying processes are moderate. Especially in drying of sensitive products such as food or grain, the required temperature is usually within the operative feasibility window of heat pumps.

European Union energy consumption regulations push companies to energy savings (European Comission, 2023). Heat pumps are one means of saving energy as they can reduce the required energy consumption of processes by between 50% and 80% when compared to boilers producing heat with combustion or electricity (Marina, et al., 2021). Central Europe in particular is looking for solutions to reduce dependence on oil and natural gas, and heat pumps have been seen as one of the key solutions in the transition.

The calculation tool developed in this thesis focuses on calculation of energy balance around the heat pump and also touches on financial feasibility by estimating the fuel or electricity costs of the heat pump solution. Fuel and energy costs form only part of total energy costs. The present calculation tool does not include estimates of, for example, investment cost, maintenance costs, other consumables, or other operation costs.

2 Waste heat potential

As illustrated in Figure 1. below, total final energy consumption in the EU28 countries in 2019 was approximately 44335 petajoules. Industry consumed approximately 10900 PJ which is approximately 24.6% of the total energy consumption. The largest consumers of energy in Europe are households and the transport sector. (Eurostat, 2023).

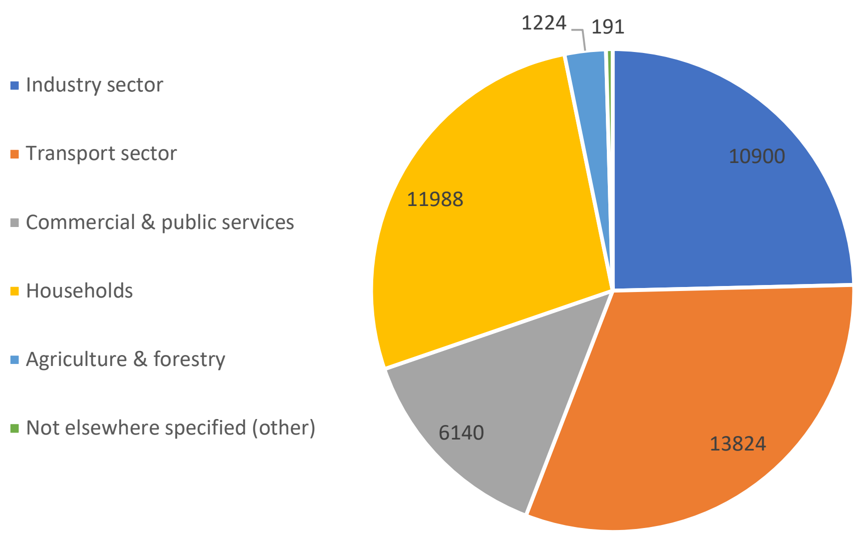


Figure 1. Total final energy consumption in EU28 in Petajoules (2019) (Eurostat, 2023).

Within the industry sector, the highest consumers of energy are chemical and petrochemical industry (20.7 % of the sector total), non-metallic minerals industry (13.8 %), paper, pulp and printing industry (13,1 %), food, beverages and tobacco industry (11,7 %) and iron and steel industry (9,8 %) (Eurostat, 2023) (see figure 2.). Iron and steel industry, minerals industry, and metal and machinery industries are commonly more dependent on electricity, mechanical work, and very high temperatures. In most cases, therefore, they are not suitable for waste heat recovery solutions, especially when heat is designed to be reused in the industrial process.

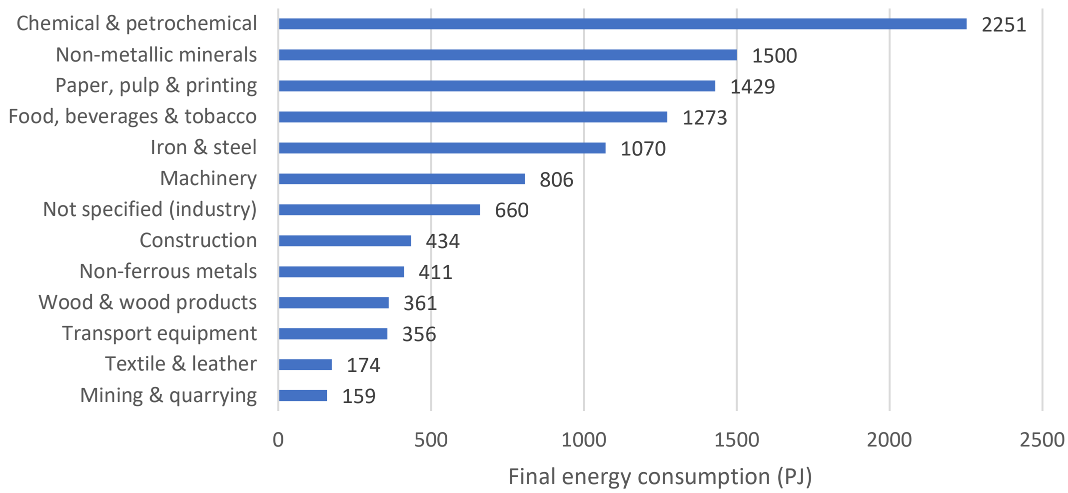


Figure 2. Final energy consumption of industrial sectors in EU28 (2019) (Eurostat, 2023).

Marina, Spoelstra, Zondag and Wemmers researched the industrial heat pump market potential in EU28 countries and identified three potential industries for waste heat recovery: Food, beverage and tobacco industry; paper, pulp and printing industry; chemical and petrochemical industry. Waste heat recovery feasibility is heavily dependent on the heat carrier and on the temperature levels of the heat sink and the waste heat. Marina et. al. determined that with current heat pump solutions the feasible limits for temperatures are a maximum outlet temperature of +200 °C and maximum lift temperature of 100 K for the heat pump. Industrial heat pump manufacturers do not openly offer heat pumps to temperatures over 150 °C, but there are demonstration projects and pioneering work ongoing in heat sink temperature range of +150…200 °C. (Marina, et al., 2021).

Marina et. al. identified approximately 1150 PJ/a of waste heat opportunities (Figure 3.) and a similar amount of heat needs in processes (Figure 4). As the Figures show, the waste heat availability and heat need in temperatures below 200 °C are not in equilibrium; for example, refinery industry has more waste heat available than it would need in processes at temperatures below 200 °C. (Marina, et al., 2021).

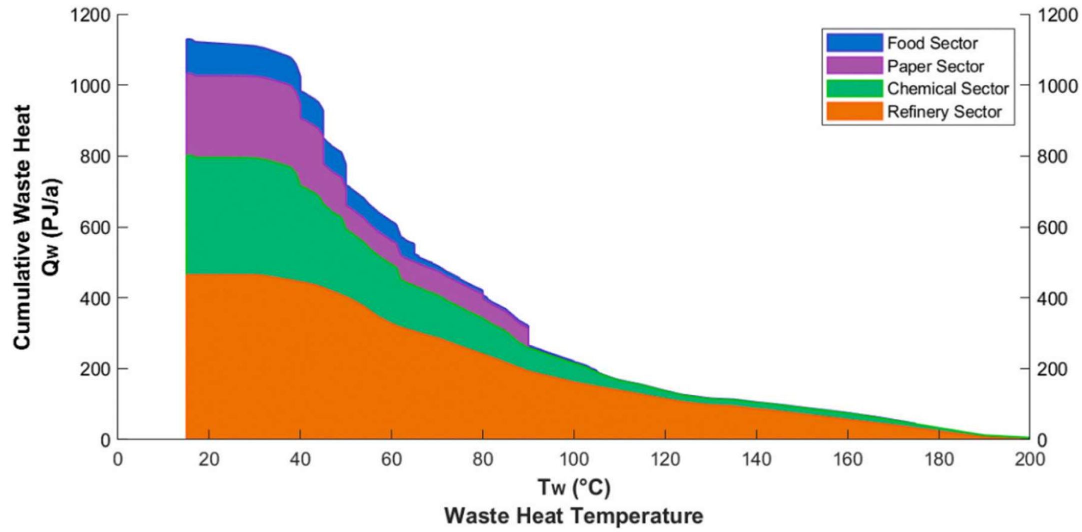


Figure 3. Cumulative available waste heat below 200 °C in EU28 (Marina, et al., 2021).

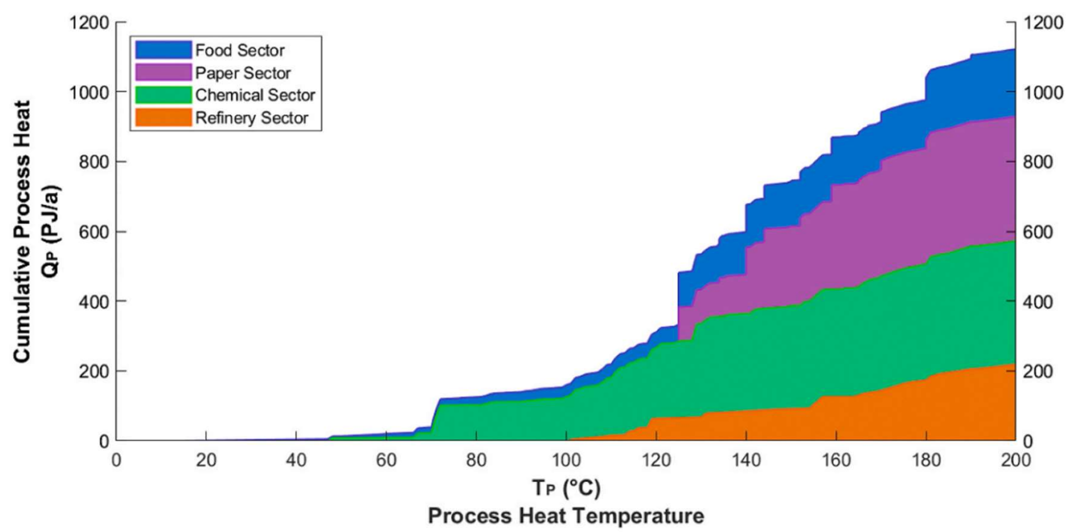


Figure 4. Cumulative process heat need below 200 °C in EU28 (Marina, et al., 2021).

Industrial processes are typically continuous and are less dependent on weather variations than, for example, energy needs in district heating. Process variations or variations caused by batch processes can be compensated with heat accumulators. Water is commonly used as heat accumulator fluid when the temperature of the process is below the boiling point of water. Pressurized heat accumulators with water are available, but large pressurized vessels have high investment costs and they are usually used for balancing process variations that last from minutes to hours. (Guelpa & Verda, 2019).

Higher temperature applications which use phase change materials, sand or heat transfer fluids designed for high temperatures are emerging on the markets. The scientific community is actively seeking new phase change materials suitable for energy storage applications. (Chuang, et al., 2022).

3 Air properties and energy recovery calculations

Several industrial drying processes are based on heating outside air and using this heated air to evaporate water from the dried product. Most industrial plants have already been fitted with heat recovery heat exchangers to recover waste heat at high temperature levels, but the usability of the heat reduces greatly as the temperature level decreases. Air flows from drying processes with high relative humidity still contain significant amounts of energy and significant primary energy consumption savings can be achieved.

Physical air properties vary in all air handling cases. Air pressure is dependent on outdoor air pressure and air pressure variation from fans or compressors. Air temperature can vary based on process variations, outdoor temperature or process properties. The water content of air is also a significant factor in air properties because of energy in latent heat of water. Maximum water content for air is determined by air pressure and temperature. (Sandberg, 2016).

3.1 Moist air humidity calculation

For the purposes of energy recovery calculations, moist air can be regarded as a mixture of two components: dry air and water vapor. The enthalpy of moist air depends heavily on the proportions of dry air and water vapor in the mixture. Dry air properties are close to ideal gas properties. Adding water vapor to dry air causes deviation from ideal gas properties. In the resultant moist air, the water vapor starts to condensate as the partial pressure of the vapor exceeds the saturation vapor pressure of the temperature of the vapor. This results in condensation of water when moist air is cooled; the latent heat of water can be recovered. (Aula, 2022).

Seppälä and Lampinen suggest that for humidity calculations it is useful to assume moist air is a mixture of water A and air B, where component A can have phase change in the process depending on temperature and pressure, and component B remains in gas phase. The relation of components A and B is essential for the calculation of moist air properties. When calculating properties of moist air, from ideal gas law it follows that component *i* partial density is

$$\rho_i = \frac{p_i M_i}{RT}, \tag{1}$$

Where p_i is the partial pressure of component i [Pa], M_i is the molar concentration [mol /m^3] of component i, T is the temperature in Kelvin, and R is the universal gas constant 8.314 J/mol K. The density of the mixture and the pressure of the mixture are the sums of the densities of the components and the pressures of the components, respectively. (Seppälä & Lampinen, 2017).

$$\rho = \sum_i \rho_i \tag{2}$$

$$p = \sum_i p_i \tag{3}$$

When mass of component A in volume V (m^3) is m_A (kg) and mass of component B in same volume is m_B (kg), the partial densities can be presented as

$$\rho_A = \frac{m_A}{V} \text{ and} \tag{4}$$

$$\rho_B = \frac{m_B}{V} \tag{5}$$

And the moisture is presented as a dimensionless value x, which is the relation of the masses of A and B, partial densities of A and B, or relation of molar masses multiplied by relation of partial pressures of A and B. (Seppälä & Lampinen, 2017).

$$x = \frac{m_A}{m_B} = \frac{\rho_A}{\rho_B} = \frac{M_A}{M_B}\frac{p_A}{p_B} = 0.6220\frac{p_A}{p_B} = 0.6220\frac{p_A}{p - p_A} \tag{6}$$

3.2 Enthalpy of moist air

Dry air is considered to be composed of the gases present in air, excluding water vapor, (chiefly nitrogen, oxygen, carbon dioxide, argon and other gases). When moist air is considered as an ideal mixture of two gases, air and water vapor, the enthalpy H [kJ/kg] of the mixture can be presented as

$$H = m_A h_A + m_B h_B \tag{7}$$

where m is the mass [kg] and h is the specific enthalpy of water vapor A and dry air B [J/kg]]. (Lampinen, 2015).

When calculating air flows, usually the calculations are based on mass flow of dry air as the moisture content in air varies. Therefore, if dry air is assumed to be component B, enthalpy h_k is defined as

$$h_k = \frac{H}{m_B}, \tag{8}$$

which is the enthalpy of the moist gas divided by the mass of the dry air (Lampinen, 2015).

When equations (6), (7) and (8) are combined, enthalpy h_k can be presented as

$$h_k = h_B + x h_A. \tag{9}$$

Reference points for enthalpy calculations of vapor and dry air can be chosen to be 0 °C and then the enthalpies can be obtained from the following equations (Sandberg, et al., 2016)

$$h_A = l_{hoA} + \int_{273.15K}^{T} c_{pA}(T)dT \tag{10}$$

$$h_B = \int_{273.15K}^{T} c_{pB}(T)dT \tag{11}$$

Where l_{hoA} is the heat of evaporation of water at chosen reference temperature 0 °C, c_{pA} is the specific heat [J/kgK] of water and c_{pB} is the specific heat of dry air. Evaporation heat of water at 0 °C is 2501 kJ/kg (Huhtinen, et al., 2016). The specific heats of dry air and water vapor depend on the temperature. If the temperature is between -10 °C and +40 °C the following average specific heat capacities can be used. (Sandberg, 2016).

c_{pA} =1.85 kJ/kg°C

c_{pB} = 1.006 kJ/kg°C

When equations (9), (10) and (11) are combined, an equation for calculating enthalpy of moist air in the temperature range -10 °C …+40 °C can be written as

$$h_k = 1.006t + x(2501 + 1.85t) \tag{12}$$

where t is the temperature of moist air in Celsius.

3.3 Heat and mass transfer in moist air

When a moist air stream meets a heat transfer surface that has a temperature below the dew point of the air, moisture in the air starts to condensate on the heat transfer surface. The air is at the same time cooled and dried. The energy flow transferred to the surface is the reduction of enthalpy of the air multiplied by the mass flow of the dry air $\dot{m}_B$ [kg/s]. (Seppälä & Lampinen, 2017).

$$\dot{Q} = -\dot{m}_B \Delta h_k \tag{13}$$

Where

$$\Delta h_k = h_{k1} - h_{k2} \tag{14}$$

Where h_{k1} is the enthalpy of air entering the heat transfer unit and h_{k2} is the enthalpy of air exiting the heat transfer unit (Seppälä & Lampinen, 2017).

The change in humidity $\dot{m}_h$ [kg/s] of air can be calculated with the equation

$$\dot{m}_h = N_h A_u = -\dot{m}_B \Delta x \tag{15}$$

Where

$$\Delta x = x_2 - x_1 \tag{16}$$

and x_2 is the humidity after the heat transfer unit [kg/kg], x_1 is the humidity before the heat transfer unit [kg/kg], N_h is the density of vapor mass flow condensating on the exterior wall of the heat exchanger and A_u is the area of the outer wall of the heat exchanger. (Seppälä & Lampinen, 2017).

3.4 Heat exchanger for heat recovery from air

Heat exchangers are used to transfer heat from one flow (for example a flue gas flow) to another flow, for example a brine flow or a heat pump refrigerant. According to the first law of thermodynamics, the heat flows into and out from the heat exchanger must be in equilibrium. It follows that the heat flow can be calculated from either side of the heat exchanger. If the heat is transferred to a heat transfer fluid flow, the heat flow can be calculated with the equation

$$\dot{Q} = \dot{m} c_p \Delta T \tag{17}$$

Where $\dot{Q}$ is the heat flow [kW], $\dot{m}$ is the mass flow [kg/s], c_p is the specific heat capacity of the liquid [kJ/(kg*K)] and ΔT is the temperature difference of entering fluid temperature and exiting fluid temperature [K]. (Hakala & Kaappola, 2019).

Heat flow in a heat exchanger can also be estimated with the equation

$$\dot{Q}_{hx} = UA\Delta T_{LM} \tag{18}$$

Where $\dot{Q}_{hx}$ is the heat transfer rate from flow A to flow B [W], U is the overall heat transfer coefficient[W/(m^{2}K)], A is the area of the heat exchange surface in m^2 and ΔT_{LM} is the logarithmic mean temperature difference of the flows [K]. (Aittomäki, et al., 2012).

$$\Delta T_{LM} = \frac{\Delta T_1 - \Delta T_2}{ln\left(\frac{\Delta T_1}{\Delta T_2}\right)} \tag{19}$$

Where ΔT_1 and ΔT_2 are the temperature differences at the ends of the heat exchanger.

According to Aittomäki and colleagues (Aittomäki, et al., 2012), the overall heat transfer values are slightly higher in condensers and especially high in, for example, plate heat exchangers functioning as ammonia condensers. General indications of overall heat transfer coefficient values are presented in Table 1.

Table 1. Values for heat exchanger overall heat transfer coefficient in refrigeration and heat pump solutions. (Aittomäki, et al., 2012).

Type of heat exchanger	**Cooled or heated media**	**Overall heat transfer coefficient [W/(m2*K)]**
Plate heat exchanger (cooling)	Water	800…2000
Plate heat exchanger (heating)	Water	1500…4500
Forced draft lamella air cooler	Air	12…25
Natural draft lamella air cooler	Air	5…10
Forced draft tube air cooler without lamellas	Air	20…30
Natural draft tube air cooler without lamellas	Air	10…20
Forced draft lamella air heater	Air	20…40

In industrial refrigeration and heat pumps the logarithmic temperature difference in the condenser is calculated by using the condensation temperature as the reference temperature on the refrigerant side. Heat transfer in subcooling and desuperheating is usually significantly less than in condensation of refrigerant. Subcooling and desuperheating are often neglected unless they are calculated as a subcooling section and a desuperheating section of the heat exchanger. If desuperheating or subcooling sections are considered significant in heat exchanger units, they usually are installed as independent equipment to ensure proper dimensioning of heat transfer surfaces (Aittomäki, et al., 2012).

In general, the heat transfer method is a complex series of events on both sides of the heat transfer surface of the heat exchanger. On the cooled air flow side, the condensation on the cold surface and mixing of cooled air in the layers of air close to the heat transfer surface affect the heat transfer efficiency. On the refrigerant side of the heat transfer surface, the viscosity and heat transfer properties of the refrigerant and possible lubricant mixed in the refrigerant affect the heat transfer. There are several equations developed for estimating the overall heat transfer coefficient; increased demand for accuracy of the estimate increases the complexity of the equation. Monograms for graphical solving of overall heat transfer coefficient have been widely used when manually dimensioning the heat exchangers. (Aittomäki, et al., 2012).

In air cooling solutions the temperature difference between air exiting the heat exchanger and evaporation temperature is chosen to be in the range 6…15 K, depending on the application and the investment cost of the heat exchanger. For flooded plate heat exchangers in ammonia evaporators, a value in the range 3…6 K is commonly used as high superheating of suction gas is not required. If a heat transfer fluid circuit is used for cooling of air, the temperature difference between heat transfer fluid and air is recommended to be 6…10 K. This temperature difference needs to be taken into account when evaluating the evaporation temperature of the heat pump if waste heat is collected with an intermediate heat transfer fluid network. (Hakala & Kaappola, 2019).

Equations 18 and 19 give only estimates of the heat transfer rate in the heat exchanger. This is because the overall heat transfer coefficient is dependent on the heat exchanger type, flow media properties (for example viscosity, enthalpy of vaporization), and flow media, heat characteristics (for example Reynolds number and Nusselt number). Heat transfer rate can also be increased by increasing the temperature difference or the heat transfer area of the heat exchanger.

3.5 Heat exchangers commonly in use for waste heat recovery

Glass tube heat exchangers are used in heat recovery when heat is transferred from gas to gas and when flows contain corrosive or otherwise problematic components. Glass tube heat exchangers have been used for example in malting process heat recovery and effluent gas heat recovery in circuit board manufacturing. They are also used in heat recovery from natatorium ventilation as vapors from swimming pools can be corrosive on metal heat exchangers. Glass tube heat exchanger manufacturers state efficiencies from 55% up to 94%, depending on, for example, air flow temperatures and moisture contents. If glass tube heat exchangers are contaminated and start developing fouling or scaling on the glass tubes, the cleaning process can be more difficult than with materials that are not as fragile. Usually cleaning must be done chemically and not mechanically. (Air Frölich, 2023).

Fin and tube heat exchangers are common in heat recovery solutions. Fins increase the heat surface area of the heat exchanger and increase the efficiency of heat transfer. Fin and tube heat exchangers are manufactured with several possible material combinations and surface treatments. Most common are copper tube with aluminum fins, but for higher temperatures or more corrosive atmospheres hot dip galvanized carbon steel or stainless steel are possible variants. High pressure variants are also produced, for example for CO_2 or steam. (Kelvion , 2023).

In Figure 5 is an installed heat recovery heat exchanger for moist process air. The heat exchanger is made of stainless steel, and it has been divided into several separately controlled sections. The heat exchanger is a pump circulated ammonia evaporator and the evaporation temperature is approximately +4 °C when in operation.

Figure 5. Stainless steel fin and tube heat exchangers for waste heat recovery with R717 refrigerant. (Adven Oy, 2023).

Tube heat exchangers without fins can be used in heat recovery, for example in applications where heavy fouling can be expected. Tube heat exchangers without fins can be cleaned easily and the cleaning process can be automatized with cleaning-in-place (CIP) systems. Tube heat exchangers without fins are similar to glass tube heat exchangers, but the tube material is, for example, duplex stainless steel that can withstand very corrosive environments. These heat exchangers are significantly more expensive than fin and tube heat exchangers. An example of a tube heat exchanger is seen in Figure 6 below. The presented heat exchanger weighs approximately 3 times the weight of a finned heat exchanger with similar heat transfer capacity. (Kelvion, 2022).

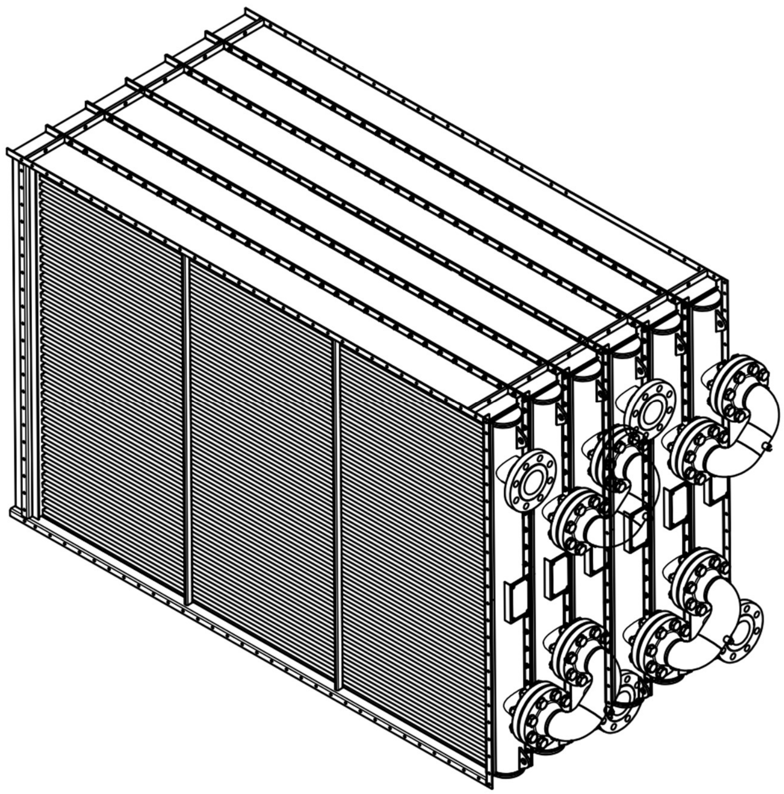


Figure 6. Tube heat exchanger with six blocks and 24 passes. (Kelvion, 2022).

Scrubbers are widely used with bio boilers to recover heat from the flue gases and to meet the requirements of environmental permits. Scrubbers are efficient in recovering heat to district heating networks if the water temperature of the return flow from the district heating network is low. Scrubbers are feasible when the moisture content of the gas flow is high and the flue gas flow can be cooled to levels where the moisture condensates in the scrubber. (Caligo, 2019).

Scrubbers are also efficient in removing particles and soluble chemicals from the gas flow, which can be an advantage or a disadvantage. Particles can form sludge and fouling can cause poor heat transfer and malfunction of equipment if not addressed properly. Dissolved chemicals such as sulfuric acid can cause corrosion of the scrubber and need to be neutralized, removed, or diluted. Depending on the composition of the effluent, the processing or waste management of scrubber effluents can incur significant costs. (Caligo, 2019).

4 Heat pumps in industrial heat recovery

Heat pumps are heat transfer equipment that can transfer energy from a lower temperature source to a higher temperature sink by using an external energy source. The cooled flow is called 'heat source' and the heated flow is called 'heat sink', as is shown in Figure 7. When simplified, the energy transferred to the heat sink is the sum of heat recovered from the heat source and work W required for operation of the heat pump (Hakala & Kaappola, 2019).

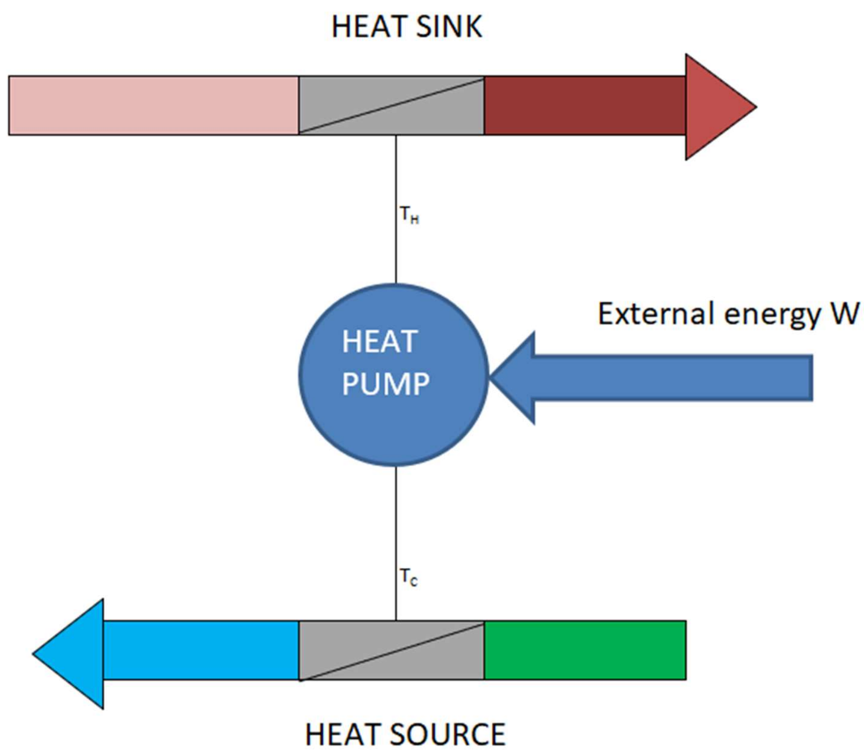


Figure 7. General diagram of a heat pump. T_H is the condensing temperature of the heat pump; T_C is the evaporating temperature of the heat pump and W is the used work to operate the heat pump. W can be high temperature heat in absorption heat pumps or electricity in compression heat pumps.

Compression heat pumps use the equilibrium of temperature and vapor pressure of the refrigerant to transfer heat. As presented in Figure 8, the refrigerant evaporates at low temperature and low pressure and the gas is compressed with a compressor to higher pressure. Compression and friction heat up the refrigerant when compressed. This superheated refrigerant is then cooled below the gas-liquid equilibrium line, the refrigerant condensates to liquid, and the heat is transferred to the heat sink. The liquid refrigerant can be subcooled to lower pressure and as the pressure is reduced with an expansion device, part of the refrigerant boils to cool the refrigerant to the equilibrium state of the evaporation pressure. (Kaappola, 2020).

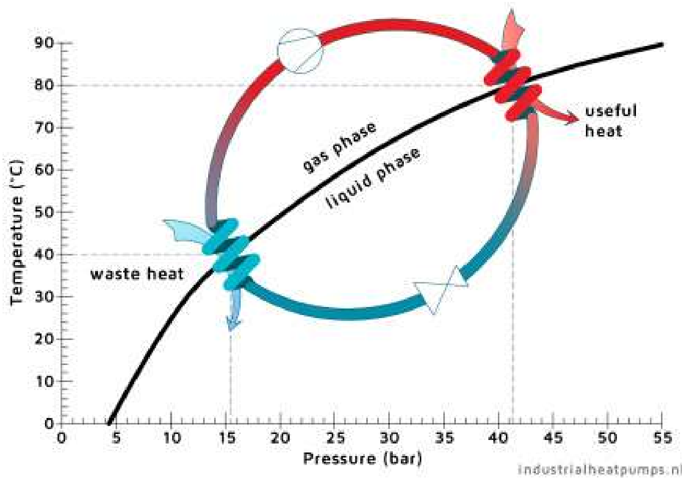


Figure 8. Gas liquid equilibrium graph of anhydrous ammonia with the heat pump process explained. (De Klejin, 2023).

The efficiency of this energy transfer process is often evaluated with Coefficient of Performance, COP. Coefficient of Performance is a dimensionless number that can be calculated from energy or from power with the following equation:

$$COP = \frac{Q_H}{W} = \frac{\dot{Q}_H}{P}, \tag{20}$$

where Q_H is heat produced to heat sink [kJ], W is external work [kJ], $\dot{Q}_H$ is the thermal power to heat sink [kW] and P is the external power introduced [kW], for example electricity or steam flow. (Aittomäki, et al., 2012).

Ideal heat pump process is defined by maximum theoretical COP, which is often referred to as COP_{Carnot} or COP_{max}. Maximum theoretical coefficient of performance was defined by French physicist Sadi Carnot as

$$COP_{Carnot} = \frac{T_H}{T_H - T_C}, \tag{21}$$

where T_H is the temperature of heat sink in Kelvin and T_C is the temperature of the heat source in Kelvin (as in Fig. 5). (Mujumdar, 2015).

The efficiency of any given heat pump can be evaluated by comparing its actual achievable *COP* to COP_{Carnot}. That is, by calculating η_{Carnot} as in equation (22):

$$\eta_{Carnot} = \frac{COP}{COP_{Carnot}} \tag{22}$$

Aittomäki and colleagues (Aittomäki, et al., 2012) state that modern compression heat pumps operate with Carnot efficiencies between 35% and 55% and Arpagaus and colleagues (Arpagaus, et al., 2018) have come to the same conclusion in their research, as can be seen from Figure 9.

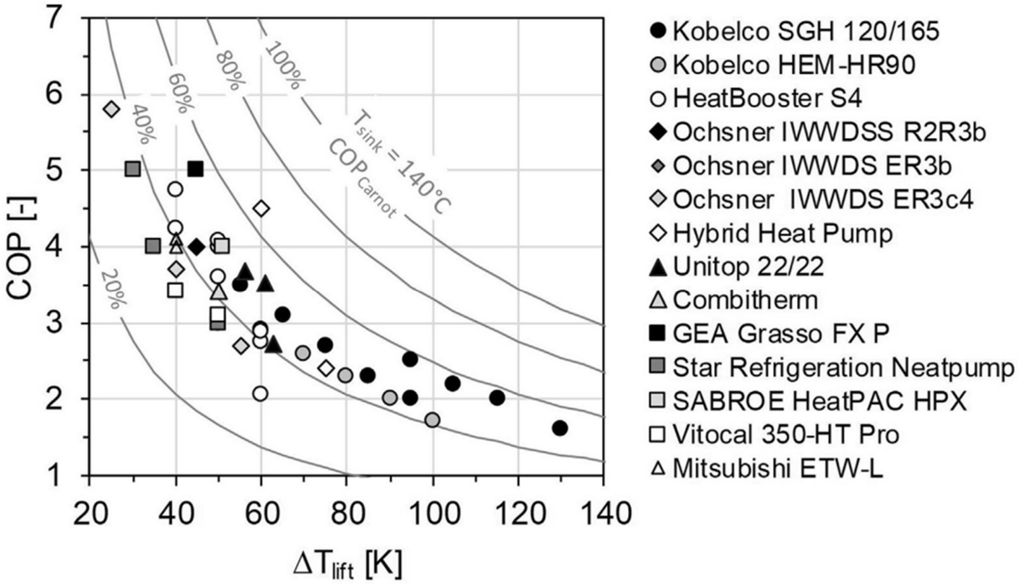


Figure 9. COP of industrial heat pumps compared to COP_{Carnot} with different lift temperatures (Arpagaus, et al., 2018).

Dr. Arpagaus recommends a 45% fit curve for Figure 9. in his webinar presentation, which results in estimation equation 23. (Arpagaus, 2023).

$$COP_{est} = 68.455 * \Delta T_{Lift}^{-0.76} \tag{23}$$

Where

$$\Delta T_{Lift} = T_{Sink,out} - T_{Source,in} \tag{24}$$

4.1 Heat pumps in general

Industrial heat pump solutions are typically built around compression heat pumps or absorption heat pumps. The choice of heat pump technology is linked to the availability of external energy source and energy price.

When low-cost high temperature water or low-pressure steam is abundant, absorption heat pumps can be a cost-efficient solution. Absorption heat pumps use heat as the driving energy source and electricity is required only for pumping refrigerant solutions. Refrigerant solutions are for example lithium bromide or ammonia-water solutions. The possibility of using waste heat as driving energy makes absorption pumps an interesting solution for power plants, industrial solutions with high exhaust gas temperatures, or pulp mills with surplus steam. The COP of absorption heat pumps is significantly lower than that of compression heat pumps, usually between 1.5…1.8 and a maximum of 2.0. Absorption heat pumps also require less mechanical maintenance than compression heat pumps as they have close to zero wearing mechanical parts. Corrosion, fouling of heat exchangers and accumulation of non-condensable gases can cause serious malfunctions if proper maintenance and operational professionalism is not followed. (Aittomäki, et al., 2012).

Compression heat pumps use volatile gas as refrigerant and heat carrier from heat source to heat sink. As refrigerant evaporates in the heat source heat exchanger (evaporator), it absorbs heat from the heat source and cools the heat source flow. Refrigerant vapor is then compressed with a compressor to high pressure. Compressors in industrial heat pumps include, for example, screw compressors, reciprocating compressors and turbocompressors. The first two are positive displacement compressors (gas volume is compressed mechanically) while turbocompressors are centrifugal compressors. High pressure refrigerant condenses in the heat sink heat exchanger (condenser) and heat is transferred to the heat sink. Compression of refrigerant requires mechanical energy, which is usually introduced with an electric motor. As refrigerant has condensed to liquid in the condenser, it is passed through a pressure reduction valve, where the pressure of the refrigerant drops to the pressure level at the evaporator and part of the refrigerant evaporates to reach the temperature/pressure equilibrium. (Aittomäki, et al., 2012).

The temperature that corresponds to the pressure where the refrigerant condenses is the condensing temperature T_H and the temperature that corresponds to the temperature at which the refrigerant evaporates in the evaporator is the evaporation temperature T_C.

Compression heat pumps have better COP than absorption heat pumps, but they require mechanical maintenance and mechanical energy for driving force. Usual COP for a mechanical heat pump is between 2 and 5, but in special cases COP it is possible to achieve COP of more than 10. (Hakala & Kaappola, 2019).

Norwegian Hybrid Energy AS has combined absorption and compression heat pumps and they have reached higher COP values than can be reached with absorption heat pumps or compression heat pumps alone. This can be seen in Figure 9, where the heat pump by Hybrid Energy AS is the only heat pump with COP_{Carnot} over 60 %. The heat pump process of the Hybrid heat pump is also presented in Table 3.

The process of hybrid heat pumps is based on the Osenbrück cycle, where evaporation and condensation processes are replaced with absorption and desorption processes. Working fluids are typically zeotropic mixtures, typically of ammonia and water. (Jensen, et al., 2015).

4.2 Refrigeration cycle in compression heat pumps

Industrial heat pumps with ammonia as refrigerant are commonly flooded evaporator processes as direct expansion evaporators are usually used for CO_2 and synthetic refrigerants. Figure 10 presents the refrigerant cycle for a flooded evaporator ammonia heat pump with pressure on the y-axis in logarithmic scale and enthalpy on the x-axis. Figure 11 shows the process diagram for a similar heat pump (IPU, 2023). At state 1 the refrigerant enters the compressor. As the refrigerant is compressed, it is also superheated. State 2 is the point at which the refrigerant is discharged from the compressor and at state 3 the refrigerant has passed through the discharge piping and enters the condenser. The condenser can be divided into three sections to improve the efficiency of heat transfer: desuperheater, condenser and subcooler. At state 4 the refrigerant has subcooled to 10 K below the condensing temperature, exits the condenser and enters the expansion valve. At state 5 a fraction (approximately 20%) of the refrigerant has evaporated to cool the remaining liquid refrigerant. Liquid refrigerant is then circulated to the intake of the flooded evaporator at state 6. The refrigerant flows out from the evaporator and is discharged to the liquid separator at state 7 as the flow includes droplets that are harmful for the compressor.

The liquid separator can be incorporated into the evaporator heat exchanger; AlfaLaval, Vahterus and Johnson Controls, for example, have their own solutions for how this is mechanically done. As refrigerant exits from the liquid separator (state 8.) it travels through the suction line to the intake of the compressor. The suction line and discharge line are presented separately from other components as, depending on process design, they have pressure losses that effect the total efficiency of the heat pump. Pressure losses are usually presented as temperature change in Kelvin, because the actual numerical pressure loss in Pascals is an insignificant process value compared to the temperature potential loss because of temperature/pressure equilibrium. In Figure 8. the losses are set at 0.5 K for both discharge and suction lines.

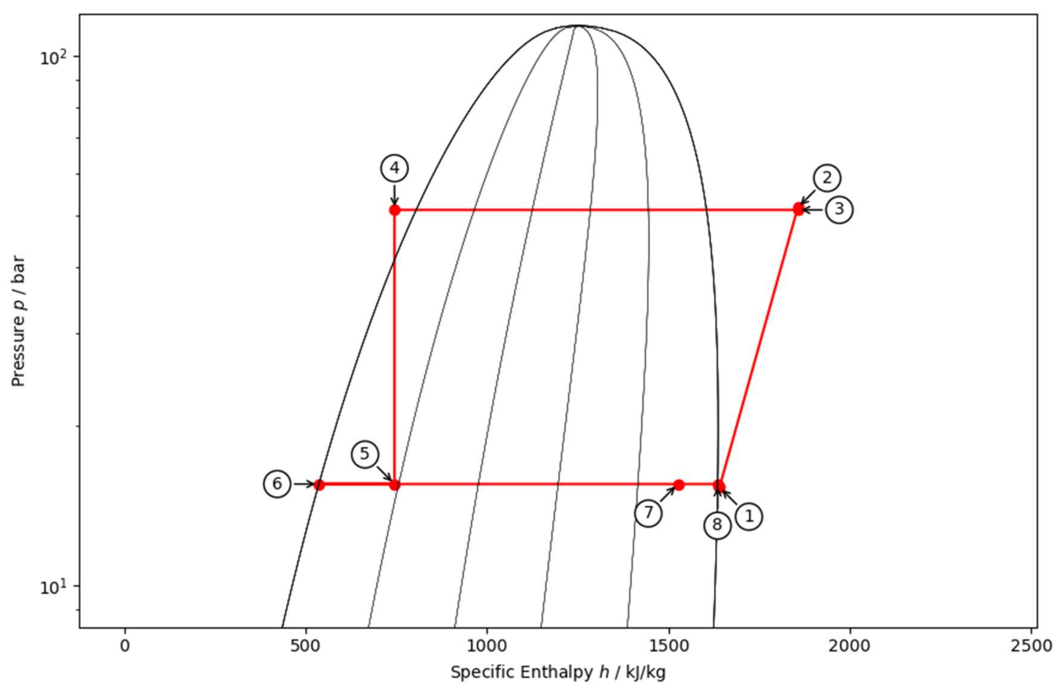


Figure 10. Log ph diagram of a flooded evaporator heat pump. Refrigerant R717, t_C= +40,0 °C, t_H= +90.0 °C, subcooling 10 K, isentropic efficiency of compression 75 % (IPU, 2023).

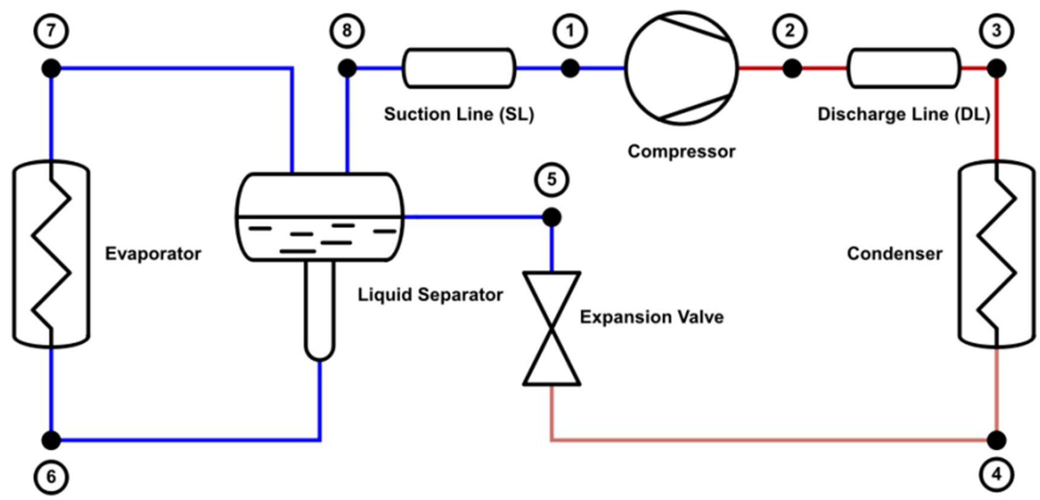


Figure 11. Process diagram of flooded evaporator heat pump and process measurement locations presented in Figure 10 (IPU, 2023).

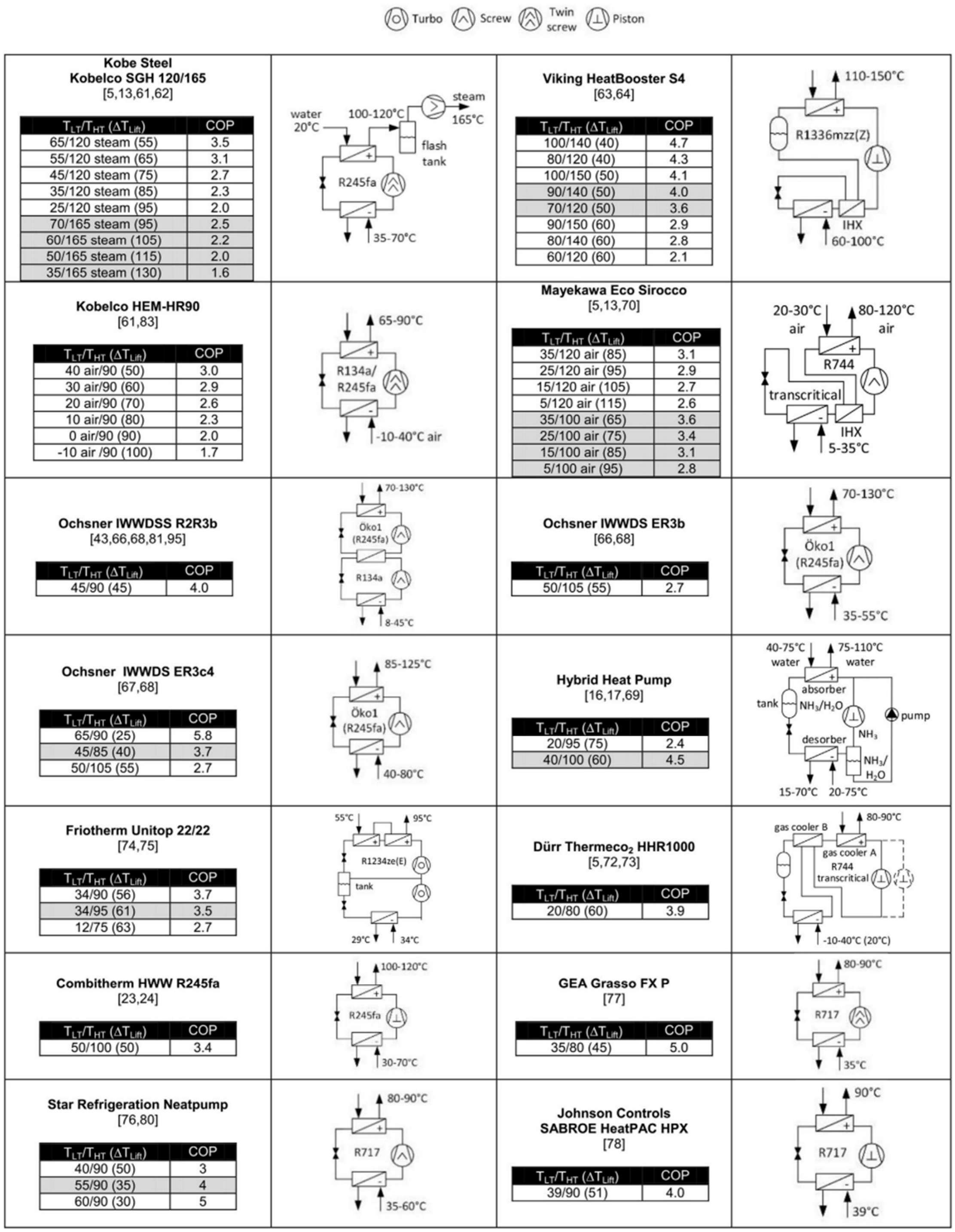
With the CoolTools program the process values for the state points can be calculated; these are presented in Table 2. The mass flow is calculated for a heat pump with heating capacity of 1000kW. The CoolTools simulation estimates a COP of 3.7 and η_{Carnot} of 59.6 % (IPU, 2023).

Table 2. Process values for state points in Figure 9.

	Temperature °C	Pressure bar	Specific Enthalpy kJ/kg	mass flow kg/s	Fraction of gas in flow %
State 1:	41.00	15.331	1640.85	0.8993	100 %
State 2:	151.87	51.693	1856.77	0.8993	100 %
State 3:	151.43	51.164	1856.77	0.8993	100 %
State 4:	80.00	51.164	744.80	0.8993	0 %
State 5:	40.00	15.545	744.80	0.8993	19 %
State 6:	40.00	15.545	536.12	0.9992	0 %
State 7:	40.00	15.545	1525.81	0.9992	90 %
State 8:	40.00	15.545	1635.77	0.8993	100 %

In Table 3 are presented different heat pumps, with COP, refrigerant and internal connection methods as well as the chosen compressor model for each heat pump presented also in Figure 9 (Arpagaus, et al., 2018). It should be noted that since 2018 the Global Warming Potential (GWP) requirements for heat pump refrigerants have been re-evaluated. Refrigerants R245fa and R134a, for example, are at risk of phase-out because of tightening F-gas regulation. The degradation of HFO-refrigerants such as R1234ze(E) to TFA (Trifluoroacetic acid) is being investigated for toxicity by the European commission under REACH-regulation (Voigt, 2022).

Table 3. Heat pump specifications for heat pumps presented in Figure 9. (Arpagaus, et al., 2018).



Kobe Steel Kobelco SGH 120/165
[5,13,61,62]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
65/120 steam (55)	3.5
55/120 steam (65)	3.1
45/120 steam (75)	2.7
35/120 steam (85)	2.3
25/120 steam (95)	2.0
70/165 steam (95)	2.5
60/165 steam (105)	2.2
50/165 steam (115)	2.0
35/165 steam (130)	1.6

Viking HeatBooster S4
[63,64]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
100/140 (40)	4.7
80/120 (40)	4.3
100/150 (50)	4.1
90/140 (50)	4.0
70/120 (50)	3.6
90/150 (60)	2.9
80/140 (60)	2.8
60/120 (60)	2.1

Kobelco HEM-HR90
[61,83]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
40 air/90 (50)	3.0
30 air/90 (60)	2.9
20 air/90 (70)	2.6
10 air/90 (80)	2.3
0 air/90 (90)	2.0
-10 air /90 (100)	1.7

Mayekawa Eco Sirocco
[5,13,70]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
35/120 air (85)	3.1
25/120 air (95)	2.9
15/120 air (105)	2.7
5/120 air (115)	2.6
35/100 air (65)	3.6
25/100 air (75)	3.4
15/100 air (85)	3.1
5/100 air (95)	2.8

Ochsner IWWDSS R2R3b
[43,66,68,81,95]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
45/90 (45)	4.0

Ochsner IWWDS ER3b
[66,68]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
50/105 (55)	2.7

Ochsner IWWDS ER3c4
[67,68]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
65/90 (25)	5.8
45/85 (40)	3.7
50/105 (55)	2.7

Hybrid Heat Pump
[16,17,69]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
20/95 (75)	2.4
40/100 (60)	4.5

Friotherm Unitop 22/22
[74,75]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
34/90 (56)	3.7
34/95 (61)	3.5
12/75 (63)	2.7

Dürr Thermeco$_2$ HHR1000
[5,72,73]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
20/80 (60)	3.9

Combitherm HWW R245fa
[23,24]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
50/100 (50)	3.4

GEA Grasso FX P
[77]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
35/80 (45)	5.0

Star Refrigeration Neatpump
[76,80]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
40/90 (50)	3
55/90 (35)	4
60/90 (30)	5

Johnson Controls SABROE HeatPAC HPX
[78]

T_{LT}/T_{HT} (ΔT_{Lift})	COP
39/90 (51)	4.0

An ideal refrigerant for heat pump applications should be non-toxic, non-flammable, have high efficiency, have a low GWP value, zero Ozone Depleting Potential (ODP) value, and have a short atmospheric lifetime. GWP value expresses what is the Global Warming Potential of a refrigerant compared to CO_2; ODP value describes the Ozone Depleting Potential compared to extremely harmful R11 refrigerant. Unfortunately, many of the ideal refrigerant properties are contradicting; industry is searching for the optimal solution for the refrigerant. It is certain that natural refrigerants such as R744 (CO_2), R717 (ammonia) and R718 (water) are allowed for use in the future, but they also have restricting elements in their use. For example, ammonia is very toxic and lethal in small dosages when leaked; CO_2 has low critical temperature; and water has poor volumetric capacity below 100 °C as the operating pressure is below atmospheric pressure. In Table 4 are different properties for commonly used refrigerants in heat pumps. Safety group classification of refrigerants is according to DIN EN 378-1 and ASHRAE 34. The first letter indicates the toxicity, where A is less toxic and B is highly toxic, and the second digit indicates the flammability of the refrigerant. Class 1 has no flame propagation, 2L is difficult to ignite and sustain flame and 2 is low flammability. (Arpagaus, 2020; Arpagaus, 2023).

In general, the heat pump unit is designed for a limited selection of refrigerants and commonly for only one, as the properties of the refrigerant stipulate the design dimensions and materials used in the heat pump. For this reason, the heat pump manufacturer selects the refrigerant that they allow for use with the heat pump. As a result, the end customer or process designer can only select the heat pump from a limited selection of refrigerants or suppliers. For the majority of end users, the toxicity, flammability and environmental friendliness of the refrigerant are major selection criteria used to eliminate heat pump candidates.

Table 4. Properties of refrigerants (Danfoss A/S, 2023; Arpagaus, 2023).

Refrigerant	Refrigerant group	$T_{critical}$ [°C]	$p_{critical}$ [bar]	ODP	GWP	Safety group
R245fa	HFC	154.0	36.5	0	804	A2
R134a	HFC	101.1	40.6	0	1430	A1
R1336mzz(Z)	HFO	171.3	29.0	0	2	A1
R1233zd(E)	HCFO	165.6	35.7	0.00034	<5	A1
R1234ze(E)	HFO	109.4	36.4	0	7	A2L
R717	Ammonia	132.4	113.6	0	0	B2L
R744	CO_2	31.0	73.8	0	1	A1
R718	Water	373.9	220.6	0	0	A1

Duclos, Gosselin and Buchet have studied at GDF Suez research center the theoretical mechanical COP of refrigerants in single stage compression. In their research they found out that the best COP with a refrigerant is commonly reached approximately 30 K below the critical temperature of the refrigerant (Duclos, et al., 2014). Figure 12. presents the COP of a heat pump utilizing different refrigerants when compression conditions are standardized. They selected 45 K as $\Delta T_{Lift,}$ subcooling of 5 K, superheating of 10 K and 75 % isentropic efficiency of compression.

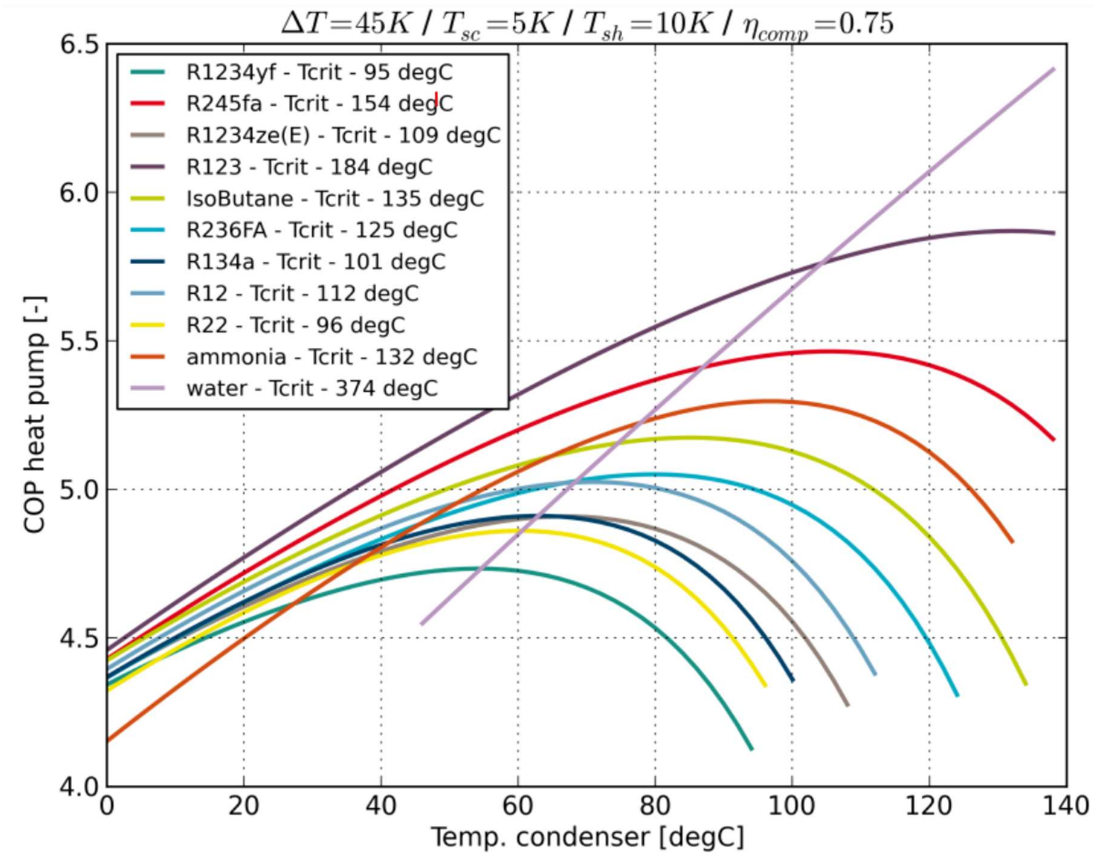


Figure 12. Comparison of efficiency of refrigerants in single stage compression (Duclos, et al., 2014).

4.3 Heat pump controls

Heat pump capacity control in modern heat pumps is covered with compressor variable speed drives. Frequency converters allow better efficiency than mechanical capacity control methods such as capacity slide control with screw compressors or cylinder unloading systems with reciprocating compressors. Mechanical capacity control methods can be used for achieving lower minimum capacity for the heat pump, but this has a significant effect on the efficiency of the heat pump in low capacities. (Johnson Controls, 2016).

As compressor capacity control can affect only the flow and pressure of the refrigerant, the heat pump can only adjust the capacity of the heat pump depending on the heat flux in the evaporator and condenser. One result of this is that most heat pumps are operated with unit controllers such as Unisab or Omni which are supplied by the manufacturer of the heat pump. This unit controller protects the heat pump from operating in undesired conditions, for example providing shutdowns due to lubrication problems, excess vibration, too high or low operating pressures, or insufficient flow of heat transfer media. Unit controllers are connected to the plant automation system with hardwired connections for safety shutdowns or operating permissions and bus connections to transfer operational data to and from the plant automation system. Examples of the data transferred include number of operating hours; alarms and shutdown messages; electricity consumption; and operating temperatures and pressures. (GEA, 2019).

Capacity control of heat pumps can be based on the condensation pressure of the heat pump or the temperature of the leaving heat sink flow. If the latter is used as control method, the unit controller is still set with both a ‘soft limit’ and a ‘hard limit’ on discharge pressure. When pressure rises to the soft limit, the unit controller starts unloading the compressor regardless of the temperature of the leaving heat sink flow. If the hard limit is reached, the unit controller initiates shutdown. Additional safety devices are hardwired pressure switches that kill the power from the motor via a safety shutdown relay of the frequency converter. (Johnson Controls, 2016).

The internal safety valve releases pressure from the discharge side of the compressor to the suction side if the discharge pressure is too high. Safety relief valves are also required by pressure equipment legislation and refrigeration equipment standard EN 378 (Metsta, 2017). Safety relief valves which discharge to open air are set to open at an interval of 2,0 bar above the safety pressure switch (Johnson Controls Denmark ApS, 2018).

Industrial heat pumps do not have internal energy buffers. This is significant as all the energy transferred in the evaporator is pumped through the compressor and discharged with added energy of operation to the heat sink flow through the condenser. Process variations (temperature, flow, humidity) in the entering temperature of the heat carrier to the evaporator or condenser can cause a shutdown, refrigerant leak via the safety relief valve or even catastrophic failure of pressurized components if the step of the variation is too large. For example, Johnson Controls recommends that the gradient of temperature change for heat pumps should not be more than 0,5 K /minute as heat transfer fluid flow remains constant. (Johnson Controls Denmark ApS, 2018).

4.4 Industrial solutions and connections

Industrial heat pumps are widely used in heat recovery from, for example, waste water, industrial refrigeration plants, flue gases from boilers and different process cooling circuits. Common practice is to use water or heat transfer fluid as the heat carrier to the evaporator heat exchanger. The heat sink is commonly a heating circuit that has medium or moderate temperature, usually between +50…+90C. This temperature range is often found in district heating networks. For example Helen and Turku Energia use heat pump technology to recover heat from the district cooling network and from waste water. COP in district cooling to district heating heat pumps is 3…5 when it is considered that the cooling and heating are both valuable products and are sold to customers. (Valor Partners Oy, 2016).

High temperature heat pumps are entering the market for steam production also. For instance, Siemens Energy is replacing steam produced by a gas fired boiler with an 11MW heat pump unit on a Finnish paper mill. Waste heat is recovered from the waste water, process cooling and exhaust air at temperature levels +15 °C from the heat pump and +23,6 °C to the heat pump evaporator. The heat pump produces 3,3 bar (abs)/ 143 °C steam which is then superheated to approximately 157 °C with an electric superheater. This results in COP of approximately 1,83. Siemens Energy has similar sized projects ongoing in Germany, for example an 8 MW heat pump system to produce heat to a district heating network at Potsdamer Platz in Berlin, where the heat source is cooling water from chiller systems. (Hüttl, et al., 2022).

MAN Energy systems has developed an transcritical CO_2 heat pump for very large industrial or district heating customers. The MAN heat pump is based on a magnetic bearing compressor designed for oil and gas industry and the collaboration MAN Energy Systems had with ABB for large scale electro-thermal energy storage (ETES). The system is fitted with an expander device to recover the energy of the pressure drop from the gas cooler to the heat recovery heat exchanger. The expander can also be used for electricity production in the ETES system. The heat pumps have thermal heating capacities of 10 to 50 MW and can produce heating up to 150 °C with maximum lift of approximately 170 K. (Decorvet & Jacquemoud, 2019).

Transcritical CO_2 systems are widely in use in commercial refrigeration solutions, for example supermarkets and small to medium sized logistic centers. In transcritical systems, the heat sink operates above the critical point of the refrigerant and the refrigerant does not condensate, thus the system does not include a condenser but a gas cooler. This requires very high pressures, often over 100 bar. High pressure allows smaller diameter piping and the heat transfer properties of supercritical CO_2 are significantly better than closer to critical point. Transcritical CO_2 heat pumps are especially energy efficient when the CO_2 gas temperature leaving the gas cooler is low and the required temperature of the heat sink is high. The differences between subcritical and transcritical processes are presented in p-h-diagrams in Figure 13. (Austin & Sumathy, 2011).

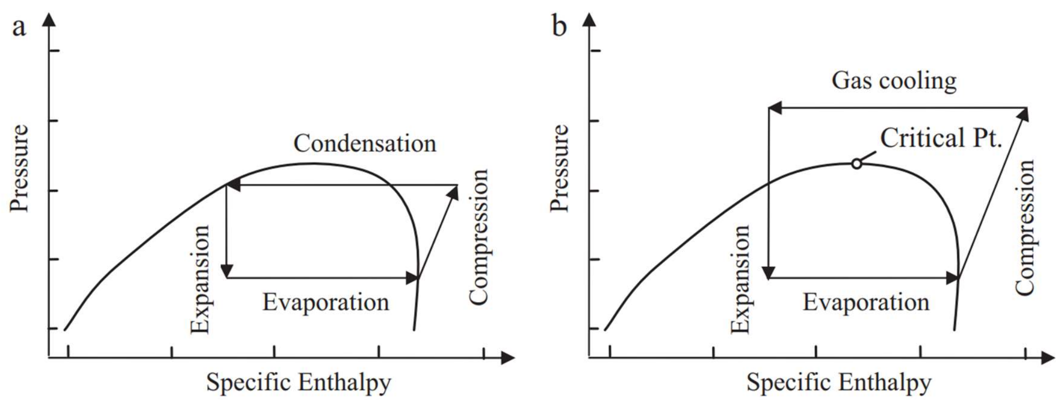


Figure 13. p-h diagrams of subcritical refrigeration cycle (a) and transcritical refrigeration cycle (b). (Austin & Sumathy, 2011).

5 The calculation tool

The aim of this thesis is to develop a tool for quick evaluation of heat pump potential in customer cases in the early sales phase. The tool was constructed in MS Excel and it also utilizes the X Steam Tables by Magnus Holmgren for, for example, water partial pressure calculations.

5.1 Calculation model

In the model the properties of the waste heat flow and heated flow are assumed to be known and the process is assumed to take place in normal atmospheric conditions at sea level. If the elevation of the location is significantly high or the heat recovery process is pressurized, the error in the results of the tool increases. The waste heat flow is assumed to be a mixture of dry air and water vapor. These conditions are typically encountered in drying processes and ventilation systems. The tool is not currently suitable for calculations of other gas mixtures, such as flue gases from boilers.

The calculation flow of the tool is presented in Figure 14. Required known properties of waste air flow are temperature before heat recovery, available mass flow and water content in air. Water content is entered into the tool as the mass fraction of water in dry air. The tool also includes a conversion tool as often only relative humidity is known in addition to mass flow and temperature.

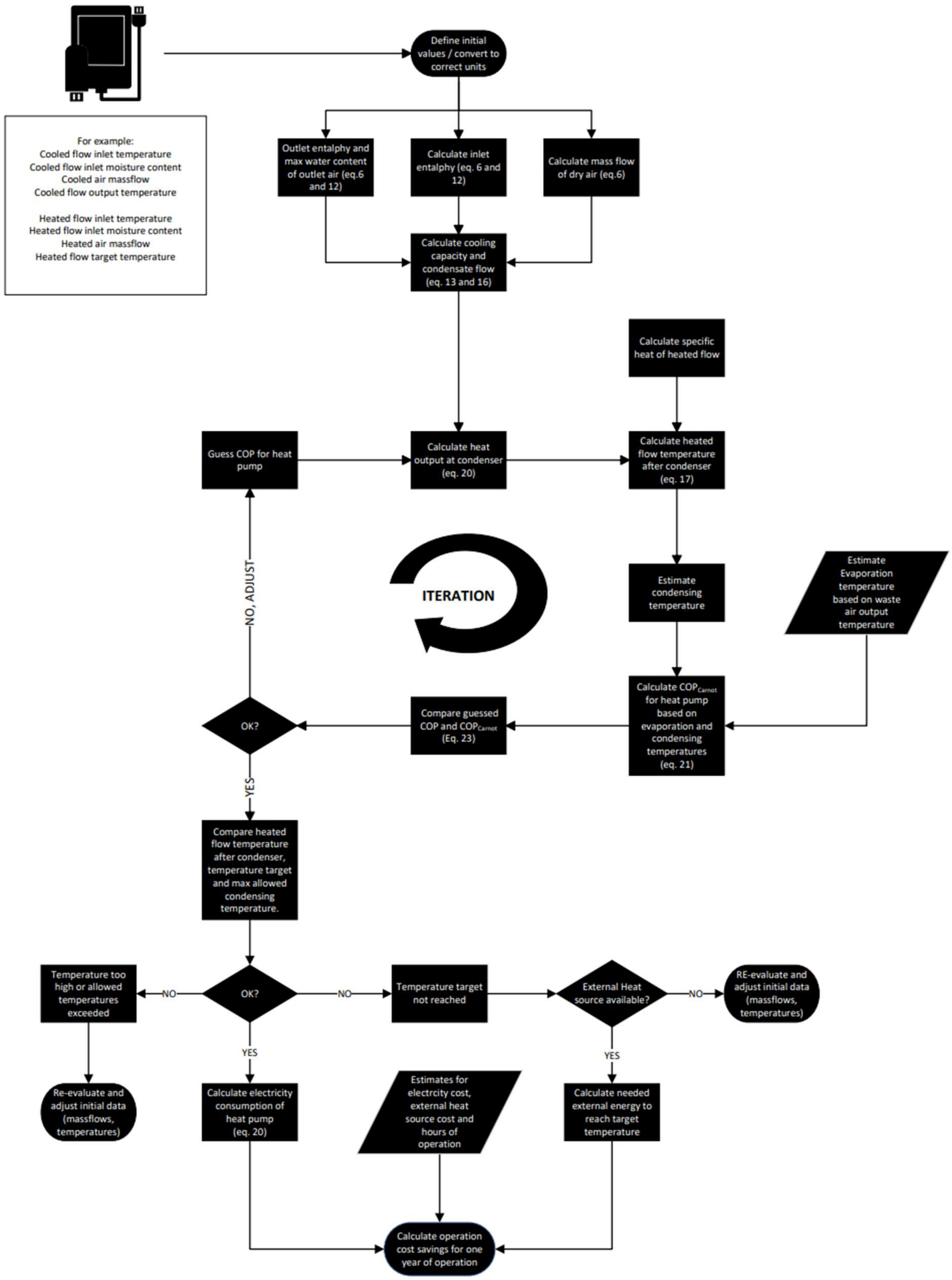


Figure 14. Calculation flow of the tool.

The user must enter the estimated temperature of the waste heat flow after the heat recovery. This input affects heavily the amount of recovered heat and the amount of condensated water from the heat recovery heat exchanger as the latent heat in water vapor is a significant source of heat in waste heat recovery. The tool calculates the amount of condensate based on the moisture content of air entering the heat recovery exchanger and the temperature of the air flow from the heat exchanger. The relative humidity of air flow after the recovery heat exchanger is calculated and capped at 100%. The rest of the humidity is considered to condensate. The temperature of the waste heat flow after the heat recovery also impacts the estimated evaporation temperature of the heat pump and hence also the COP estimate of the heat pump. The calculation tool assumes that the evaporation temperature of the heat pump is 8 K below the temperature of the air flow after the heat recovery heat exchanger. The temperature difference between the evaporation temperature and the air temperature after the heat exchanger affects on the condensation and relative humidity of the air flow. This is not taken into consideration in the tool.

The tool has separate calculation sheets for cases in which the heat can be utilized directly to air and for cases in which heat is transferred to a brine flow. The required parameters for heated flow are inlet temperature, mass flow and target temperature of the flow. If the heated flow is air, then the specific heat capacity is calculated based on the humidity of the inlet air. If the heated flow is other brine than water, the specific heat capacity must be looked up from the literature and entered into the tool.

The tool calculates the cooling power of the heat recovery heat exchanger and based on the cooling power and electricity consumption of the heat pump, the heating power of the heat pump is calculated. This heating power is used for calculating the temperature of the heated flow after the heat pump condenser. The condensing temperature of the heat pump is selected to be 6 K above temperature of the flow from the condenser in brine heat exchangers and 10 K in air heat exchangers.

As the evaporation temperature and the condensing temperature are the prime factors in predicting the COP of the heat pump, these are used for estimating the COP. The COP of the heat pump is iterated by Excel by using equation 23 (see page 27, above) and it is compared to the ideal COP with efficiency of 50%.

If the user enters also the estimated operation hours in a year at the given process values and costs for alternative fuel and electricity, the tool calculates an estimation of cost savings per year.

The accuracy of the calculation tool decreases if the temperature of the waste air flow is below 0 or above 50 °C as the used enthalpy equation for moist air is linear for this temperature range. Also, if the temperature of the waste air flow after the heat exchanger drops so low that the evaporation temperature of the heat pump is below zero, the heat exchanger starts to accumulate ice and the results are not valid. The tool can estimate the feasibility of solutions with a high temperature heat sink, but it alarms the user if the condensation temperature is above 95 °C as the availability of industrial solutions is restricted and the case should be evaluated more carefully before advancing.

5.2 Testing calculation models on actual cases

5.2.1 Data collection in existing plants

Data for feasibility study and process design is gathered from plant automation systems, special big data servers or by installing data collection equipment into the process if data is not otherwise available. In the case of green field projects, the data must be simulated based on available process information, historical data, (for example weather data) and data from similar process plants.

Data quality must be evaluated for decision making. Very few plants have stable process conditions throughout the day, week or year. Hourly data for one year is considered reasonable and in addition to hourly data, minute based or 10sec data is collected to evaluate rapid changes in the process conditions that can destabilize the heat recovery process.

For heat recovery feasibility evaluation, the capacity of waste heat is evaluated based on the temperature of the flow and other qualities of the flow. When recovering heat from low temperature moist air flows, the temperature and the relative humidity of the air is of the essence. Latent heat from moisture condensation is the main source of heat as the specific heat of air is very low. This means that large variance in flow humidity will cause large variance in recovered heat. If the air flow is a mixture of process air and air from heating, ventilation or air conditioning systems, outdoor temperature and outdoor humidity will be significant factors in the relative humidity variation through the year.

5.2.1 Case 1

Case 1 is a medium sized food ingredients processing plant situated in northern Europe. The elevation of the location is less than 200m above sea level. The customer uses fossil fuels for heat source and is looking for a solution to reduce the CO_2-footprint of their product. Most of the used energy is used in a drying process. The customer has conventional heat recovery with a glass tube heat exchanger and the drying air is heated with steam to air heat exchangers. The process is presented in Figure 15.

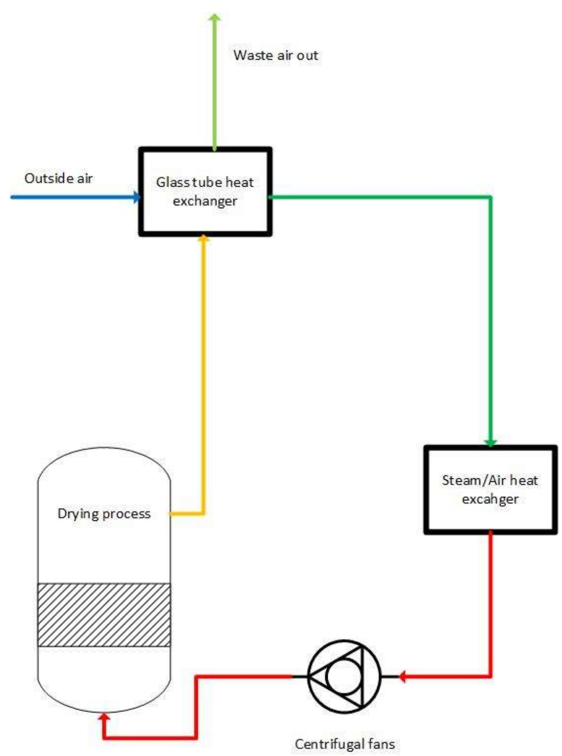


Figure 15. Simplified flow diagram of the drying process in Case 1 before heat recovery.

It is assumed that when the drying process is in its normal operation state, the dry fresh air mass flow is equal to the dry air mass flow out through the heat recovery heat exchanger, i.e. the process does not have significant leakages or losses. The operation time for the drying process is approximately 7400 hours per year and based on drying recipe of the product, 85% of the operating time is suitable for direct waste heat recovery. This results in an estimate of 6300 hours of operation per year. The process values for the case are presented below in Table 5.

Table 5. Process values for Case 1 before heat recovery.

Case 1. **Process values**	**cooled or heated media**		
Fresh air temperature after glass tube heat exchanger	Air	22	°C
Fresh air relative humidity after glass tube heat exchanger	Air	30	%
Waste air temperature after glass tube heat exchanger	Air	25	°C
Waste air relative humidity after glass tube heat exchanger	Air	100	%
Waste air temperature after heat recovery heat exchanger	Air	20	°C
Needed air temperature for drying process	Air	75	°C
Fresh air mass flow	Air	900 000	kg/h
Hours of operation		6 300	h/a

A heat pump can be installed to recover heat from the waste air leaving from the glass tube heat exchanger and the heat can be used for heating the fresh air flow from the glass tube heat exchanger before entering the steam/air heat exchanger. With this connection the glass tube heat exchanger can be maximally utilized for preheating the outside air, the heat recovery heat pump has minimal lift and the steam/air heat exchanger can always be used for assuring the right temperature of air into the drying process. The simplified flow diagram of the process with a heat pump is presented in Figure 16 below.

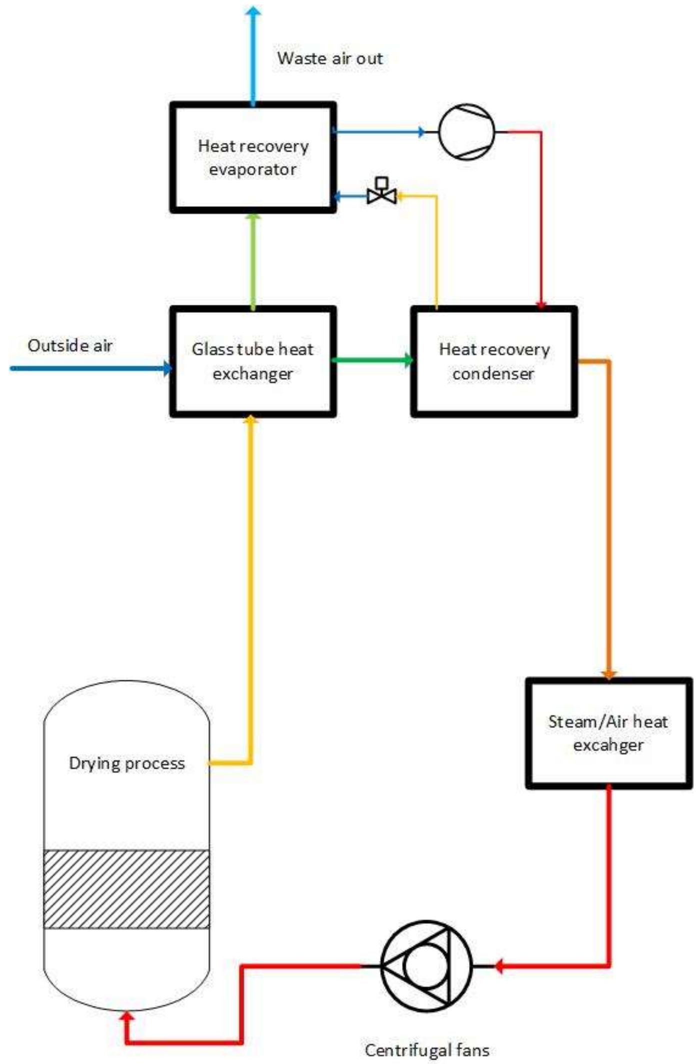


Figure 16. Simplified air flow diagram of the Case 1 drying process with heat pump incorporated.

When the values of Table 6 are entered into the calculation tool, the following data is returned:

Table 6. Results for Case 1.

Case 1. Results **Process values**	**cooled or heated media**		
Total energy needed for heating fresh air		13 500	kW
Evaporation temperature	Refrigerant	12	°C
Condensing temperature	Refrigerant	57.7	°C
Fresh air temperature after condenser	Air	47.7	°C
Cooling power of heat pump		4 790	kW
Heating power of heat pump		6 530	kW
Electric power of heat pump		1 740	kW
COP of heat pump		3.7	
Mass flow of condensate from evaporator	water	1.37	kg/s

The results in Table 6 show that approximately 50 % of the energy needed for the drying process can be provided cost efficiently with a heat pump.

Fuel costs with emissions costs are estimated to be 90 €/MWh and the cost of electricity with transfer costs is estimated to be 220 €/MWh. Based on these calculations the tool estimates that the savings in fuel costs would be approximately 1 287 000 € per year when the added electricity consumption is also considered. This cost savings estimation does not include other cost involved, for example investment or maintenance costs. The calculation was also executed with different temperatures for air leaving the heat recovery heat exchanger. As can be seen in Figure 17, optimal savings are achieved with a leaving temperature of +20 °C.

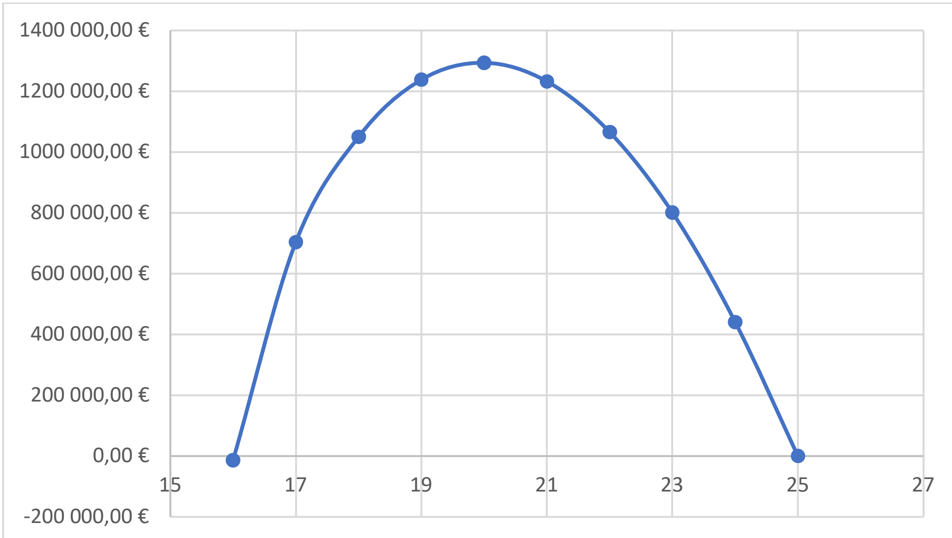


Figure 17, Fuel cost savings in euros versus temperature of the air from the heat recovery heat exchanger in Celsius.

5.2.2 Case 2

Case 2 is a processing plant located in central Europe at an elevation of approximately 100m. The plant dries a product with drum dryers. Drums are heated with steam and the product film on the top of the drum is heated and the water in the product slurry evaporates. The evaporated water is drawn into a ventilation hood and blown to outside air. The hood surface is heated to prevent the moisture from condensing on colder surfaces. Currently no heat recovery is installed on the ventilation system. The process is presented in Figure 18.

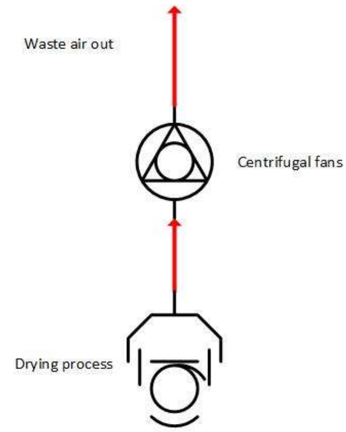


Figure 18. Case 2 drying process air flow, without heat recovery.

The plant requires hot water in preheating of process waters, heating of process liquids, building heating and can also use it in preheating air for pneumatic dryers. The hot water network leaving temperature is +75 °C and the returning water temperature is +40 °C. Consumption of hot water in the hot water network is predicted to be minimum 4 MW throughout the year. This hot water is currently produced with steam boilers using natural gas as fuel.

Waste heat recovery is installed to recover heat from the waste air of the drying process. Due to plant layout and local legislation the refrigerant charge needs to be minimized. This results in using an intermediate heat transfer fluid circuit between the air flow and the heat pump. In the calculations, the heat transfer fluid is assumed to be water, but due to possible freezing conditions, alternatives can be considered, for example water-propylene glycol mixture.

The customer has measured the waste air properties and these values are used for the input values for the simulation. Waste air from the dryers is very humid, with relative humidity approximately 92 % which results in air water content of approximately 0.06 kg of water in one kilogram of dry air. Air temperature is 45 °C and the air flow is 20 kg/s. The drying process runs approximately 6000 hours per year. The process values used in simulation are presented in Table 7. For the simulation, the temperature difference between the leaving air temperature from the heat recovery heat exchanger and the evaporation temperature of the heat pump is set as 10 K and it is estimated that the temperature difference is divided in to 6 K for the heat recovery heat exchanger (air/water) and 4 K for the heat pump evaporator (water/refrigerant). The returning water from the heat recovery heat exchanger to the evaporator is estimated to be +30 °C, but it is insignificant for the calculation. This temperature is fixed for calculation purposes for simulations with the heat exchanger manufacturer's and heat pump manufacturer's simulation tools.

Table 7. Process values for the Tool for Case 2.

Case 2. Process values	**cooled or heated media**		
Air from the dryer	Air	45	°C
Air water content	Air	0.06	kg/kg da
Air mass flow	Air	60 000	kg/h
Air temperature after heat recovery heat exchanger	Air	20	°C
Brine circuit to heat recovery heat exchanger	water	14	°C
Brine circuit out from heat recovery heat exchanger	water	30	°C
Hours of operation		6 000	h/a
Hot water network leaving temperature	water	75	°C
Hot water network returning temperature	water	40	°C

The simplified process diagram of the heat recovery with a heat pump and heat transfer fluid intermediate circuit is presented in Figure 18. Figure 18 also presents the estimated or calculated process temperatures when the leaving waste air temperature is set to +20 °C. The simulation results are presented in Table 8.

Table 8. Results for Case 2.

Case 2. Results Process values	**cooled or heated media**		
Evaporation temperature	Refrigerant	10	°C
Condensing temperature	Refrigerant	78	°C
Cooling power of heat pump		2 250	kW
Heating power of heat pump		3 520	kW
Electric power of heat pump		1 270	kW
COP of heat pump		2.8	
Mass flow of condensate from heat recovery	water	0.71	kg/s

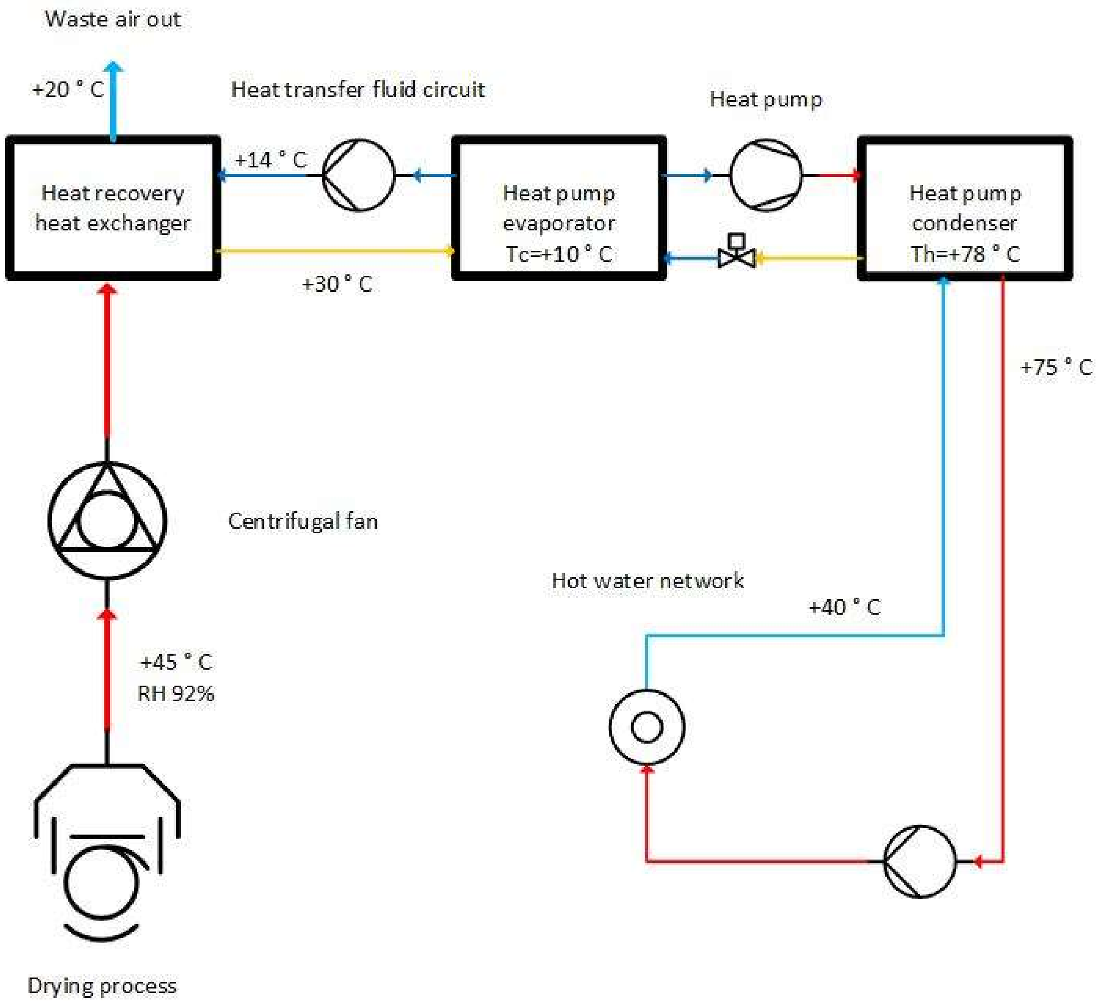


Figure 18. Connection method for the heat recovery from the process in Case 2.

Fuel costs with emissions costs are estimated to be 100 €/MWh and the cost of electricity is estimated to be 150 €/MWh. Based on these calculations the tool estimates that the savings in fuel costs would be approximately 970 004 € per year when the added electricity consumption of the heat pump is also considered. The cost savings estimation does not include other costs, for example investment cost and the electricity cost for the pumping of heat transfer fluids for the hot water network or intermediate circuit. The calculation was also executed with different temperatures for air leaving the heat recovery heat exchanger;

these results are presented in Figure 19. When the leaving waste air temperature is set to +20 °C, the heat pump can produce the 3,5 MW of heat needed in the hot water network.

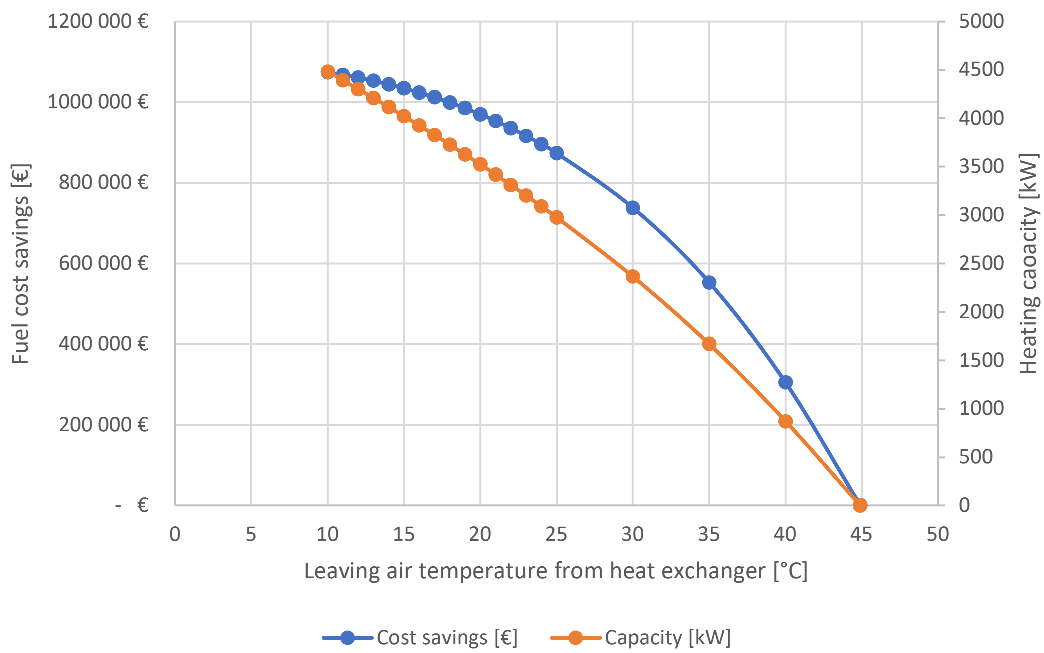


Figure 19. Case 2. Potential fuel cost savings and recovered capacity [kW] depends on the temperature of the waste air leaving from the heat recovery heat exchanger [°C].

5.3 Comparison of calculation model to other sources

The tool was developed as no similar tool was available. The validation of the tool was done by comparing the values calculated by the tool with those taken from the simulation software available from heat pump manufacturers, CoolPack and heat exchanger manufacturer LU-VE.

5.3.1 Comparison to air heat exchanger tools available

IPU calculation software CoolPack has an auxiliary built in tool for calculating air coolers for cooling and de-humidification of moist air. CoolPack is a legacy calculation software which is available free online. It is developed by the Department of Mechanical Engineering at the Technical University of Denmark and it is no longer maintained or developed. (IPU, 2023).

Simplifying somewhat, the waste heat recovery heat exchanger is an air cooler and dehumidifier. The CoolPack calculator for air coolers was used for validation of the results. The comparison with error as percentage for Case 1 is seen in Table 9 and for Case 2 in Table 10. The input values for the calculators are marked with asterisks (*).

Table 9. Comparison of calculated air properties for Case 1.

		Case 1		
The value	**Units**	**Tool**	**CoolPack**	**Difference**
Volumetric air flow to heat exchanger	m^3/h	56196,4	56196*	0.00 %
Air massflow to heat exchanger	kg/s	16.7*	16.744	-0.26 %
Dry air mass flow to heat exchanger	kg/s	15.698	15.795	-0.61 %
Entering temperature of air	°C	45*	45*	
Leaving temperature of air	°C	20*	20*	
Moisture content of air entering the heat exchanger	kg/kg dry air	0.06*	0.06006	-0.10 %
Relative humidity of air entering the heat exchanger	%		93*	
Capacity of heat exchanger	kW	2252	2259	-0.31 %
Enthalpy of entering air to heat exchanger	kJ/kg	200.36	200.41	0.02 %
Enthalpy of leaving air from heat exchanger	kJ/kg	56.87	57.42	0.97 %
Condensate out from heat exchanger	kg/h	2572	2580	0.31 %
Moisture content of air leaving the heat exchanger	kg/kg dry air	0.0145	0.0147	1.38 %

Table 10. Comparison of calculated air properties for Case 2.

The value	Units	Case 2 Tool	 CoolPack	 Difference
Volumetric air flow to heat exchanger	m^3/h	782486	782486*	0.00 %
Air massflow to heat exchanger	kg/s	254.3*	254.31	0.00 %
Dry air mass flow to heat exchanger	kg/s	249.214	249.303	0.04 %
Entering temperature of air	°C	25*	25*	
Leaving temperature of air	°C	20*	20*	
Moisture content of air entering the heat exchanger	kg/kg dry air	0.02*	0.02009	-0.45 %
Relative humidity of air entering the heat exchanger	%		100*	0.00 %
Capacity of heat exchanger	kW	4788	4708	-1.67 %
Enthalpy of entering air to heat exchanger	kJ/kg	76.085	76.31	0.30 %
Enthalpy of leaving air from heat exchanger	kJ/kg	56.87	57.42	0.97 %
Condensate out from heat exchanger	kg/h	4947	4835	-2.26 %
Moisture content of air leaving the heat exchanger	kg/kg dry air	0.0145	0.0147	1.38 %

The calculation of available waste heat was also calculated by LU-VE Group with their calculation tools for internal use. Kuoppala, Holanti and Salonen calculated one version of the Case 2 heat recovery heat exchanger as a reference using the values in Table 11 for air and heat transfer fluid. This resulted in 2841kW available waste heat with air output temperature of +19.9C and relative humidity of 99%. Heat transfer fluid temperature in was +14 °C in and +31.9 C °out. The dimensions for this heat exchanger would be 8.00m x 2.52m x 0.30 m. The calculation result is presented and compared to similar values from CoolPack and the developed calculation tool in Table 11. The difference of capacity in the developed calculator tool and LU-VE internal calculating program is approximately 5% or 143kW. (Kuoppala, et al., 2023).

Table 11. Manufacturer dimensioning of Case 2 heat recovery heat exchanger (Kuoppala, et al., 2023) (IPU, 2023).

		Case 2		
The value	**Units**	**Tool**	**CoolPack**	**LU-VE**
Volumetric air flow to heat exchanger	m^3/h	67301	67150*	60134
Air massflow to heat exchanger	kg/s	20*	20.015	20.0
Dry air mass flow to heat exchanger	kg/s	18.8	18.9	
Entering temperature of air	°C	45*	45*	45
Leaving temperature of air	°C	20*	20*	19.9
Moisture content of air entering the heat exchanger	kg/kg dry air	0.06*	0.0594	
Relative humidity of air entering the heat exchanger	%		92*	92
Capacity of heat exchanger	kW	2698	2667	2841
Enthalpy of entering air to heat exchanger	kJ/kg	200.36	198.59	
Enthalpy of leaving air from heat exchanger	kJ/kg	56.87	57.42	
Condensate out from heat exchanger	kg/h	3080	3040	
Moisture content of air leaving the heat exchanger	kg/kg dry air	0.0145	0.0147	

5.3.2 Comparison to heat pump tools of manufacturers

The COP of Case 1 calculated with the tool is 3.7. With evaporation temperature of +12 °C and condensing temperature of +58.5 °C Bitzer Software calculates 1765 kW condenser capacity with 383 kW of shaft power, resulting in COP of 4.6 with ammonia (R717) as refrigerant. Shaft power consumption does not include losses of electric motor or frequency converter. Figure 17 presents the system flow diagram. The Bitzer online software was requested for 2000 kW of cooling capacity to ensure that the calculations were made with the largest available compressors.

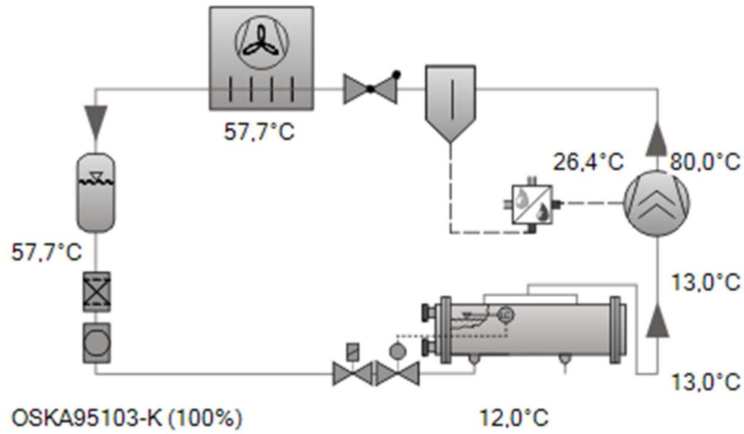


Figure 17. Bitzer compressor calculation tool process temperatures with R717 refrigerant and OSKA95103 compressor (Bitzer, 2023).

When similar conditions were calculated with R1234ze compatible compressor, the condenser capacity was 892 kW and the power input 201 kW. This results in COP of 4.4. The process and process values are presented in Figure 18.

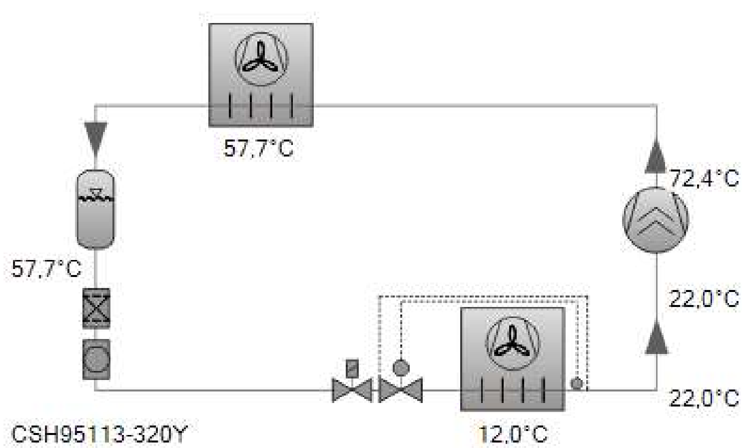


Figure 18. Bitzer compressor calculation tool process temperatures with R1234ze refrigerant and CSH95113 Compressor (Bitzer, 2023).

Case 1 heat recovery could be designed with a transcritical CO_2 system as the required temperature lift of air is high. With evaporation temperature of +12 °C and gas cooler output temperature of +40 °C the COP of the system would be 4.0. CO_2 systems require very high pressures for operation. The evaporation pressure for this system would be approximately 47.5 bar (a) and condenser pressure would be approximately 99 bar (a). (Bock GmbH, 2023).

In case 2 the tool estimates the COP of the heat pump to be 2.8 with chosen temperatures of +20 °C leaving waste air temperature, +14 °C/+30 °C heat transfer fluid temperatures and +40 °C/+75 °C hot water network temperatures. Using these values with heat pump manufacturers' simulation tools yields the following results.

The Oilon Selection tool estimates the heat pump COP to be 2.72 as presented in Figure 19. This is very close to the simulated value of 2.8.

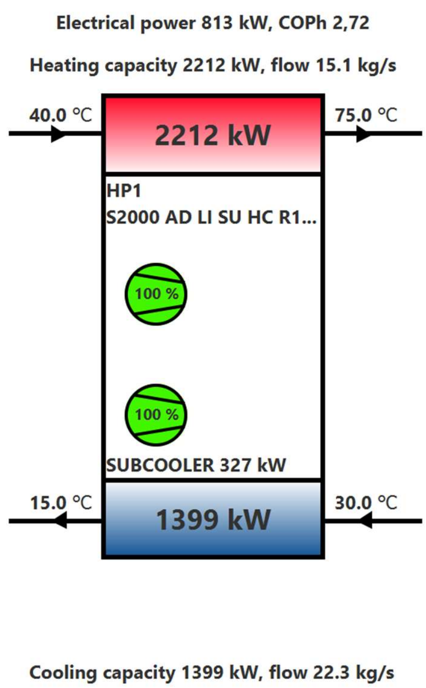


Figure 19. Case 2 heat pump simulation with Oilon heat pump selection tool (Oilon, 2023).

With the Sabroe simulation tool the COP is slightly higher at 2.99 (Figure 20), but the tool presents only shaft power consumption and does not include the losses of the electric motor and frequency converter. The temperature of leaving hot water from the condenser restricts the options for selecting larger units than HeatPAC 716V and this results in small unit size of heat pump with heating capacity of only 604 kW per unit. (Sabroe Technical Computing, 2023).

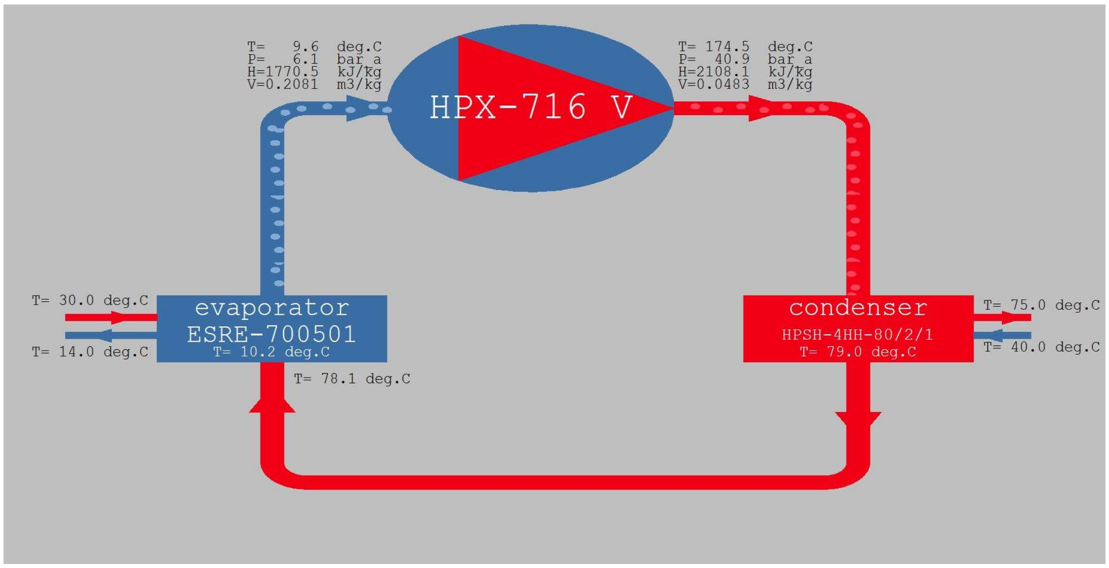


Figure 20. Case 2 with R717 heat pump (Sabroe Technical Computing, 2023).

The GEA ammonia heat pump and compressor simulation tool restricts the temperature difference at the evaporator to 10 K and automatically incorporates subcooler and oil cooling heat exchanger automatically in to the heat pump efficiency calculation. The heat pump system setup by GEA is presented in Figure 21. GEA simulation tool calculates a COP of 3.55 for the system, which includes the losses of the electric drive. (GEA, 2023).

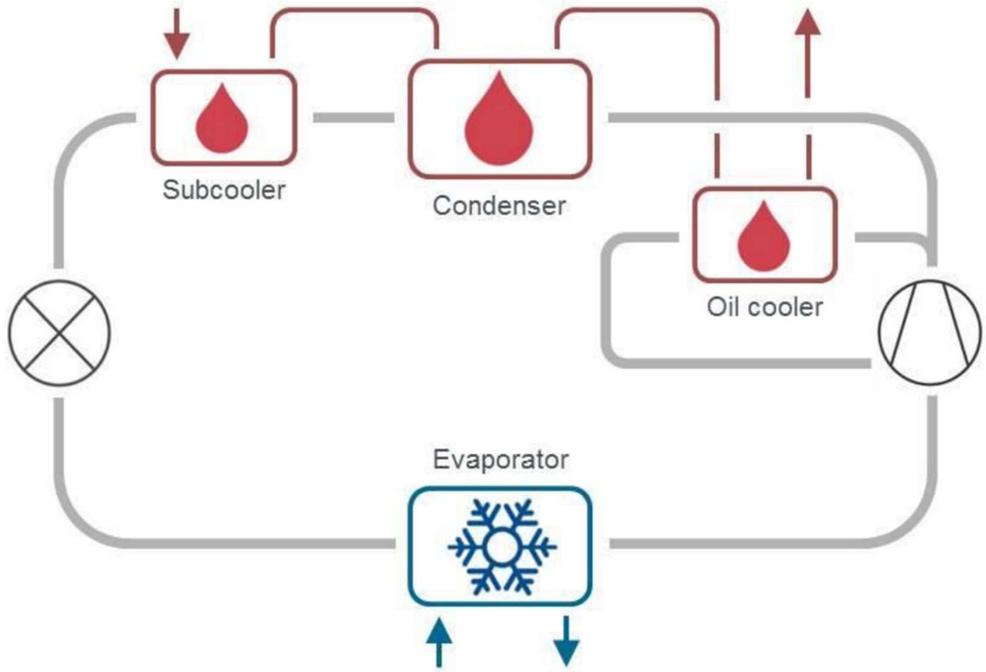


Figure 21. Heat pump system setup by GEA selection tool (GEA, 2023).

In Case 2, the temperatures of heat transfer fluid would allow a more advanced process layout. Selecting two slightly smaller heat pumps with the Oilon selecting tool would result in higher COP of 3.66 and is probably a more feasible solution. The simulation with two heat pumps in series is presented in Figure 22.

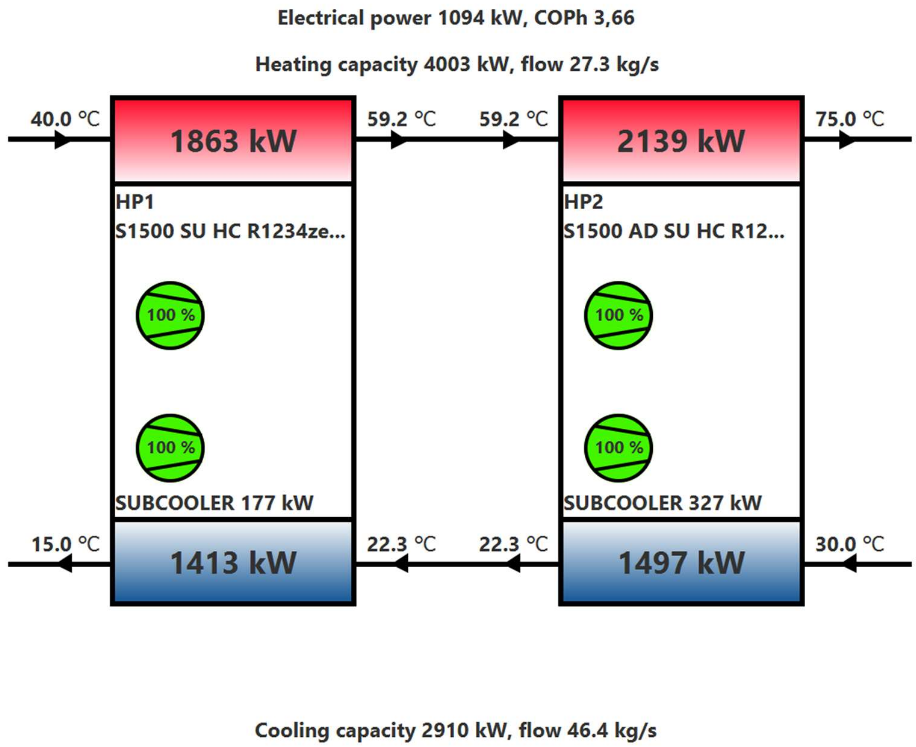


Figure 22. Case 2 heat pump simulation with two heat pumps in series.

The estimated COP from different sources are assembled in table 12. The losses of electric motors and frequency converters are not presented in the simulation tools by Johnson Controls and Bitzer and this results in higher COP estimate. 20 percent error in COP would generally be considered significant, but since in the present case the losses of electric drives are considered the accuracy of the COP estimate can be considered satisfactory.

It should be noted that the tool is purposed for an early phase in the sales process. In this context, input values such as temperatures, humidity of waste air, process operation hours and fuel prices must be regarded as estimates, unless they are backed up by long term measurement data from plant automation systems. For case 2 it is also important to note that neither the tool nor the simulation tools of manufacturers includes the electricity required for pumping of heat transfer fluids.

Table 12. COP estimations and difference between the compressor manufacturer simulation tool and the estimation tool. COP results marked with * does not include the losses of electric motors and frequency converters.

Comparison of COP estimations	**COP**	**Error**
Case 1 Tool COP	3.7	
R717 Bitzer* (Fig 17.)	4.6	24 %
R1234ze Bitzer* (Fig 18.)	4.4	19 %
R744 Bock*	4.0	8 %
Case 2 Tool COP	2.8	
R717 Johnson Controls* (Fig 20.)	3.0	7 %
R717 GEA (Fig 21.)	3.6	29 %
R1234ze Oilon (Fig 19.)	2.7	-4 %

5.3.3 Evaluation of savings calculated by the calculator

The tool calculates fuel savings based on energy consumption, cost of electricity and cost of an alternative heat source as input by the tool operator. Three limitations of the tool's fuel savings calculations are apparent. First, that the cost per MWh of the alternative heat source and the number of operating hours are input by the tool operator, introducing the possibility of operator estimation error. Second that related costs such as maintenance costs, personnel costs and investment costs are not taken into account. And third the volatilities of process values and heat loads are not included in calculations.

That said, the tool's fuel savings calculations can nevertheless be used to determine the feasibility of the investment by providing an upper limit on maximum allowed investment. For example if the customer has an allowed payback time of 3 years, the investment in Case 2 should not exceed 2 800 000… 3 000 000 euros.

With respect to investment costs in particular, it should be noted that these are significantly affected by the complexity of industrial sites. For example, the investment costs for refrigerant piping and steel structures required for heat recovery heat exchangers can vary from 20 000€ to 600 000€. On many sites, new buildings for heat pump machinery are required and electricity connection to the plant must be inspected for additional electricity consumption. The fuel savings calculations produced by the tool should be situated in the specific contextual parameters of each industrial site where the tool is used.

6 Conclusions

Industrial waste heat recovery systems are becoming increasingly complex as attention shifts to lower temperature waste heat sources. Currently commercially available evaluation tools provided by manufacturers of heat pumps, compressors, and heat exchangers exhibit limitations. In particular, such tools often require registration and deep knowledge of the process and calculation software to produce useful results. This limits the userbase for these tools to engineering consultants and process designers. Often more complex calculations are not available to anyone other than the representatives of the respective manufacturer. Hence the usefulness of such tools in sales phase evaluations of waste heat recovery is severely limited. The tool presented here is intended to provide a simpler, easier to use alternative with particular application in preliminary feasibility assessment of heat recovery solutions.

The accuracy of the tool presented in this thesis reduces as the temperature of the waste air flow increases. For example equation 12 is recommended only for air temperatures between -10 °C to +40 °C. In Case 2 it is seen that the tool's reliability is good with inlet temperature of +45 °C. It is not recommended to use the tool to sink temperatures above +90 °C or heat source temperatures above +45 °C before more thorough testing and validation.

The dynamic calculation model and iterative COP calculation allows the operator to input process values freely. This allows the operator to test several operation conditions, such as temperatures and moisture contents with the conditions of other parts of the process remaining constant.

The calculator tool is designed to take shortcuts in calculation to allow dynamic calculation of COP and heating capacity depending on temperatures and mass flows. For example, the heat recovery heat exchanger mean logarithmic temperature difference is not properly dimensioned and it is replaced by estimating the evaporation temperature difference to the air temperature exiting from the heat recovery heat exchanger. Were the heat exchanger designed in in the calculator in more detail, the calculator could estimate the needed heat exchanger surface area and potentially the pressure loss of airflow through the heat exchanger. This would complicate the calculation significantly and require more initial values. This would steer the calculator further away from the original idea of simplicity and ease of use.

Heat recovery projects are always complex and require more detailed engineering as there are several factors that influence the financial feasibility of the project. As presented in Case 2 by replacing one larger heat pump with two heat pumps in series, the COP of each solution can vary depending on more than just temperatures of the heat source and heat sink. However it is necessary to simplify the calculation in order to enable quick evaluation of the feasibility of heat recovery.

The calculation tool developed here for the evaluation of heat recovery from moist process exhaust air has proven to give reasonably consistent results. The tool enables the salesperson to do quick evaluation of the heat recovery case even on customer site in a few minutes by asking the general process values from the customer. Previously these initial evaluations were done by concept managers on request, which may have resulted in overlooking possibilities for heat recovery.

When compared to CoolPack simulation tool, the error of results was within a few percentage points. This is more than adequate, as the accuracy of process information in the early sales phase is often questionable. The errors of air properties calculations are a result of the equations used, since these are approximations. Should these errors be deemed excessive, future versions of the tool could reduce these errors by substituting more sophisticated equations for calculating moist air properties.

The tool is not designed to replace any simulating tools provided by the heat pump manufacturers, but to provide quick estimates based on customer process information to assist preliminary feasibility assessment of heat recovery solutions. This tool can also be used as an eliminating tool to allow the sales team to focus resources on the most financially viable cases. If the tool shows no profitability in the heat recovery case, other means of heat production should be primarily considered.

The error of the calculation tool can cause the tool to present optimistic or pessimistic results. In the cases tested in this thesis, the calculator tool presents COP values that are pessimistic when compared to the calculation tools of heat pump manufacturers. The average of errors in COP calculation is 16 %. In Case 2 the heat pump manufacturers calculators are expected to give reliable results but in Case 1 the manufacturers’ tools do not calculate the pressure losses of the refrigerant piping, droplet separation, control/shut-off valves, or losses of electric motors and frequency converters. These losses can be several Kelvins or kilowatts, depending on the complexity of the process, layout of the plant, chosen technology and compressor model. For these reasons the pessimistic estimate by the tool is acceptable and these errors are minimized in further design steps before the freezing of concept and committing to certain COP guarantees or investment decisions.

The calculator tool calculated approximately 10 % lower capacity for heat recovery than the manufacturers' design tools. The error cannot be completely explained by minor differences in calculation parameters, but as the calculation method of the LU-VE design tool is not available this error can be accepted. It should also be noted that air heat exchangers tend to collect dust on wet heat transfer surfaces. This results in reduction of heat transfer properties of the heat exchanger and therefore the heat exchanger should be dimensioned with more heat transfer surface and capacity to compensate the fouling of the heat exchanger.

Simplicity, ease of use and availability for all sales and concept personnel was one of the key reasons for opting to use Microsoft Excel as the platform for the calculation tool. Excel is widely used and available on all corporate computers, so no additional applications are needed and the threshold to start utilizing the tool is lower even for personnel without engineering background.

The calculation tool will be presented to the sales and concept development teams and training in use of the tool will be provided. The user interface of the tool should be streamlined before full presentation and training and documentation shall be included in the tool to allow future development steps.

The accuracy of the tool could be developed by a using more sophisticated platform for the calculations, for example Python or Matlab. This would allow the tool to calculate the efficiency, energy consumption and energy production on an hourly basis for a yearly production period. This would also allow more sophisticated calculations for moist air enthalpy and could take into account for example the thermodynamic properties of refrigerants.

When the tool is transferred to Matlab or Python, the structure should be considered carefully. The current excel-based solution is rigid and if the calculation needs to be approached from the condenser side, the complete excel sheet must be reassembled. With, for example, Python the structure should allow easy assembly of the required subcalculations. This is again not a task for the front end sales persons, but could be easily adopted by the concept team.

References

Adven Oy, 2023. *Photography.* Vantaa: Adven Oy.

Adven oy, 2023. *Tietoa Meistä.* [Online]
Available at: https://adven.com/fi/tietoa-meista/
[Accessed 12 May 2023].

Air Frölich, 2023. *Air Fröhlich, lasiputkilämmönsiirtimet,* Helsinki: Taniplan Oy.

Aittomäki, A. et al., 2012. *Kylmätekniikka.* 4th ed. Helsinki: Suomen Kylmäyhdistys ry.

Arpagaus, C., 2020. *Industrial Heat Pumps, Supplier update, suitable refrigerants and application examples in food & steam generation,* Online Webinar: Australian alliance for energy produtivity.

Arpagaus, C., 2023. *2023 high-temperature heat pumps update webinar,* s.l.: Australian Alliance for Energy Productivity.

Arpagaus, C., Bless, F., Uhlmann, M. & Schiffmann, J., 2018. High temperature heat pumps: Market overview, state of the art,. *Energy,* Volume 152, pp. 985-1010.

Aula, A., 2022. *Kylmätekniikan koulutuspäivät, Kostean ilman tilapiirros käytännössä.* Helsinki, Suomen Kylmäyhdistys ry.

Austin, B. T. & Sumathy, K., 2011. Transcritical carbon dioxide heat pump systems: A review. *Renewable and Sustainable Energy Reviews ,* 15(8), pp. 4013-4029.

Bitzer, 2023. *BITZER Software v6.18.0 rev2812.* [Online]
Available at: https://www.bitzer.de/websoftware/Calculate.aspx?cid=1681584468699&mod=CS
[Accessed 17 April 2023].

Bock GmbH, 2023. *VAP for stationary applications.* [Online]
Available at: https://vap.bock.de/stationaryapplication/Pages/Index.aspx
[Accessed 21 April 2023].

Caligo, 2019. *Caligo flue gas scrubbers.* [Online]
Available at: https://www.caligoindustria.com/en/our-solutions/heat-pump-connectible-scrubber-csx-hpc/
[Accessed 21 April 2023].

Chuang, W. et al., 2022. Research progress and performance improvement of phase change heat accumulators. *Journal of Energy Storage,* Volume 56, p. 105884.

Danfoss A/S, 2023. *Danfoss Ref Tools, Refrigerant slider,* Denmark: Danfoss A/S.

De Klejin, 2023. *Operating Principle, Industrial heat pumps.* [Online]
Available at: https://industrialheatpumps.nl/english/operating_principle/
[Accessed 09 May 2023].

Decorvet, R. & Jacquemoud, E., 2019. Supply of high Temperature Heat and Cooling. In: B. Zühlsdorf, M. Bantle & B. Elmegaard, eds. *Book of presentations of the 2nd Symposium on High-Temperature Heat Pumps.* Copenhagen: SINTEF, pp. 205-221.

Duclos, J., Gosselin, D. & Buchet, P., 2014. *High temperature gas heat pumps to recover industrial waste heat.* Saint Denis La Plaine, International Gas Union.

European Comission, 2023. *EU agrees stronger rules to boost energy efficiency.* [Online]
Available at: https://ec.europa.eu/commission/presscorner/detail/en/IP_23_1581
[Accessed 12 May 2023].

Eurostat, 2023. *Energy Balances 2019.* [Online]
Available at: https://ec.europa.eu/eurostat/web/energy/data/energy-balances

GEA, 2019. *Control, GEA Omni.* E_801011_6 ed. s.l.:GEA.

GEA, 2023. *RT Select, version 13,0,7,* Berlin: GEA.

Green, D. W. & Southard, M. Z., 2019. *Perry's chemical engineers' handbook.* 9th ed. New York: McGraw-Hill Education.

Guelpa, E. & Verda, V., 2019. Thermal energy storage in district heating and cooling systems: A review. *Applied Energy,* Volume 252, p. 113474.

Hakala, P. & Kaappola, E., 2019. *Kylmälaitoksen suunnittelu.* 4th ed. Tampere: Opetushallitus.

Holmgren, M., 2023. *X Steam, Thermodynamic properties of water and steam, ,* s.l.: MATLAB Central File Exchange.

Huhtinen, M., Korhonen, R., Pimiä, T. & Urpalainen, S., 2016. *Voimalaitostekniikka.* 3rd ed. Helsinki: Opetushallitus.

Hüttl, C., Wenn, N., Voss, J. & Reissner, F., 2022. *New perspectives for industrial heat pumps, Chillventa 2022.* Nürnberg, Siemens Energy Industrial Heat Pump Solutions.

IPU, 2023. *CoolPack - IPU.* [Online]
Available at: https://www.ipu.dk/products/coolpack/
[Accessed 10 May 2023].

IPU, 2023. *CoolTools v1.1.0,* Kongens Lyngby: IPU.

Jensen, J., Markussen, W., Reinholdt, L. & Elmegaard, B., 2015. On the development of high temperature ammonia–water hybrid absorption–compression heat pumps. *International journal of refrigeration,* Volume 58, pp. 79-89.

Johnson Controls Denmark ApS, 2018. *Engineering manual, SMC/TSMC/HPC/HPX Mk 4 LL & Mk 5 Reciprocating compressor units.* 5th ed. Højbjerg: Johnson Controls Denmark ApS.

Johnson Controls, 2016. *Engineering Manual, Unisab III Control.* Version 1.10.7 ed. Højbjerg: Johnson Controls Denmark ApS.

Kaappola, E., 2020. *Kylmäprosessi.* Helsinki, Suomen LVI-Liitto SuLVI ry.

Kelvion , 2023. *Compact fin and tube coils.* [Online]
Available at: https://www.kelvion.com/products/product/coils/
[Accessed 21 April 2023].

Kelvion, 2022. *Heat recovery heat exchangers,* Vantaa: Kelvion.

Kuoppala, J.-M., Salonen, A. & Holanti, M., 2023. *Sähköpostikeskustelu* [Interview] (5 May 2023).

Lampinen, M., 2015. *Thermodynamics of humid air,* Espoo: Department of Energy Technology, Aalto University.

Marina, A., Spoelstra, S., Zondag, . H. & Wemmers, A., 2021. An estimation of the European industrial heat pump market potential. *Renewable and Sustainable Energy Reviews,* 21 January, Volume 139, pp. 1-14.

Metsta, 2017. *SFS-käsikirja 65-1 Kylmälaitteet.* 4th ed. Helsinki: Suomen Standardoimisliitto SFS ry.

Mujumdar, A. S., 2015. *Handbook of Industrial Drying.* 4th ed. Boca Raton: CRC Press.

Oilon, 2023. *Oilon Selection tool, Version 2023.03.02-1762,* Kokkola: Oilon.

Sabroe Technical Computing, 2023. *COMP1 MatchMaster, Version 32.52,* Aarhus: Johnson Controls.

Sandberg, E., 2016. *Ilmastointilaitoksen mitoitus, Ilmastointitekniikka osa 2.* 2nd ed. Forssa: Talotekniikka-Julkaisut Oy.

Sandberg, E. et al., 2016. Ilmankäsittelyprosessit ja koneiden mitoitus. In: E. Sandberg, ed. *Ilmastointitekniikka 2, Ilmastointilaitoksen mitoitus.* Forssa: Talotekniikka-Julkaisut Oy, pp. 123-145.

Seppälä, A. & Lampinen, M. J., 2017. *Aineensiirto-oppi.* s.l.:Yliopistokustannus.

Valor Partners Oy, 2016. *Suuret lämpöpumput kaukolämpöjärjestelmässä, Loppuraportti,* Helsinki: Energiateollisuus ry.

Voigt, A., 2022. *Summary Asercom – EPEE Symposium.* Nûrnberg, Chillventa.

Appendices

Appendix 1. Simulation result for Case 1 heat pump, Johnson Controls

4/10/23, 12:38 PM Sabroe MatchMaster Program COMP1

SABROE BY JOHNSON CONTROLS

Sabroe

Refrigeration Plant Computation

Version 32.52

File	:	Ref	: JDOE	Page :	1
Date	: 2023/04/10	Time	: 12.37.46		
User	: GENERIC USER LEVEL 4				
Prog	: COMP1/409901	Print	: def. not found		

SINGLE STAGE COMPRESSOR

compressor type	HPC 116 S VSD	refrigerant	R 717
number of compressors	1.00	evaporating temperature	18.0 deg.C
compressor load	100.0 %	condensing temperature	58.5 deg.C
drive shaft speed	1800.0 RPM (list)	total suction superheat	0.0 K
no. of working cylinders:	16	suction line superheat	0.0 K
drive type	direct	total liquid subcooling	2.0 K
suction line loss	1.0 K	condenser liquid subcooling	2.0 K
discharge line loss	1.0 K		
total cooling capacity	1591.1 kW	total shaft power req.	331.1 kW
		drive shaft torque	1757. Nm
total heating capacity	1908. kW	cooling cap./shaft power ratio	4.81
equipment for head cooling	by WHC-system		
equipment for oil cooling	by WHC-system		

oil separator: OHUR 5014 (96)

operating conditions:

suction pressure	7.78 bar_a	discharge pressure	25.83 bar_a
suction temperature	17.00 deg.C	discharge temperature	116.39 deg.C
suction specific volume	0.1640 m3/kg	discharge specific volume	0.0662 m3/kg
enthalpy difference (ref.)	1009.85 kJ/kg	condenser subcooled liquid density	551.5 kg/m3
suction side mass flow	1.5756 kg/s	evaporator saturated liquid density	613.2 kg/m3
swept volume	1085.7 m3/h	pressure ratio (p2/p1)	3.32

errors and warnings:
NB: No weight found for compressor
NB: design limits check OK

The tolerance of all performance data issued for theproducts is according to EN-12900.
Measurement tolerances according to EN13771.
Data subject to change without notice.

SABROE
BY JOHNSON CONTROLS

Sabroe

Refrigerant Plant Computation

Version 32.52

File	:	Ref	: JDOE	Page :	2
Date	: 2023/04/10	Time	: 12.37.46		
User	: GENERIC USER LEVEL 4				
Prog	: COMP1/409901	Print	: def. not found		

SABROE base refrigerant composition

refrigerant designation	R-717				
number of elements in mixture	1				(1)
mixture critical temperature	132.25 deg.C	evaporating line temperature glide	0.00 K		
mixture critical pressure	113.33 bar_a	condensing line temperature glide	0.00 K		
mixture average mol. weight	17.03 kg/kmol	mixture surface tension at TE	.2664E-01 N/m		

number		name	mol-%	weight-%
1	R717	ammonia	100.000	100.000

errors and warnings:

errors and warnings:

The tolerance of all performance data issued for theproducts is according to EN-12900.
Measurement tolerances according to EN13771.
Data subject to change without notice.

Appendix 2. Simulation result for Case 1 heat pump, Bitzer

2.1 R1234ze refrigerant

****
BITZER Software v6.18.0 rev2812

Donnerstag, 11. Mai 2023 12:34:31
****

Compressor Selection: Compact Screw Compressors CS // CSV

Input Values:

Cooling capacity	2000 kW	
Refrigerant	R1234ze	
Reference temperature	Dew point temp.	
Evaporating SST	12,00 °C	
Condensing SDT	57,7 °C	
Liq. subc. (in condenser)	0 K	
Suct. gas superheat	10,00 K	
Useful superheat		100 %
Operating mode	Standard	
Power supply	460V-3-60Hz	
Capacity control		100 %
Additional cooling	Automatic	
Max. discharge gas temp.	110,0 °C	

Result

Compressor	CSH95113-320Y-40D		
Capacity steps		100 %	
Cooling capacity		691	kW
Cooling capacity *		691	kW
Evaporator capacity		691	kW
Power input		201	kW
Current (460V)		285	A
Voltage range	440-480V		
Condenser capacity		892	kW
COP/EER		3,43	
COP/EER *		3,43	

Mass flow LP		20725	kg/h
Mass flow HP		20725	kg/h
Operating mode	Standard		
Liquid temp.		57,7	°C
Oil volume flow		1,34	m^3/h
Cooling method	--		
Discharge gas temp. w/o cooling		72,4	°C

Largest compressor type - partition in several units required
Discharge gas temperature at least 20K (36°F) above condensing temperature
Consider national standards for the use of flammable refrigerants.
*According to EN12900 (10K suction gas superheat, 0K liquid subcooling, see tech. data/ notes)

2.2 R717 Refrigerant

****
BITZER Software v6.18.0 rev2812

Donnerstag, 11. Mai 2023 12:29:21
****

Compressor Selection: Open Screw Compressors OS

Input Values:

Compressor model	OSKA95103-K	
Refrigerant	R717	
Reference temperature	Dew point temp.	
Evaporating SST	12,00 °C	
Condensing SDT	57,7 °C	
Liq. subc. (in condenser)	0 K	
Suct. gas superheat	1,00 K	
Operating mode	Standard	
Speed	3500 /min	
Useful superheat		100 %
Additional cooling	Automatic	
Max. discharge gas temp.	80,0 °C	
Cooling capacity		100 %

Result

Compressor	OSKA95103-K		
Cooling capacity		100 %	
Cooling capacity		1616	kW
Cooling capacity *		1602	kW
Evaporator capacity		1616	kW
Shaft power		383	kW
Condenser capacity		1765	kW
COP/EER		4,22	
COP/EER *		4,18	
Mass flow LP		5822	kg/h
Mass flow HP		5822	kg/h
Operating mode	Standard		
Liquid temp.		57,7	°C
Oil volume flow		9,15	m³/h
Cooling method	External		
Oil injection temp. comp.		26,4	°C
Oil cooler load		234	kW
Recommended driving motor		448	kW
Discharge gas temp. with cooling		80	°C
Discharge gas temp. w/o cooling		128,2	°C

Tentative Data.Additional
cooling/ Limitations (see
Limits)!
Additional cooling/ Limitations (see Limits)!Starting point for motor selection see T. Data/ Notes
Starting point for motor selection see T. Data/ Notes only valid for flooded systems
Selection only valid for flooded systems*According to
EN12900 (5K suction gas superheat, 0K liquid subcooling)
*According to EN12900 (5K suction gas superheat, 0K liquid subcooling)

Appendix 3. Simulation results for Case 1, Bock compressors

HGX46/345-4 SH CO2 T

Engine: 440-480V Y/YY -3- 60Hz PW
Refrigerant: R744
Subject:

BOCK colour the world of tomorrow

Performance data

Application: Heat Pump

Refrigerant	R744	Heating capacity (incl. subcooling)	191.00 kW
Reference temperature	Dew point	Evaporator refrigeration capacity	144.00 kW
Supply frequency	60 Hz	Power consumption	47.70 kW
Power supply	60 Hz, 460 V	Current draw (460 V)	73.80 A [1)]
Evaporating temperature	12.0 °C	Coefficient of performance (COP)	4.00
Evaporating pressure (abs.)	*47.30 bar*	Gas cooler heat rejection	191.00 kW
High pressure (abs.)	98.82 bar	Mass flow	1.121 kg/s
Gas cooler outlet temperature	40.0 °C	Discharge end temperature	86.3 °C [2)]
Suction gas superheat	10 K		
Subcooling (outside cond.)	-- K		
Usable superheat	100%		

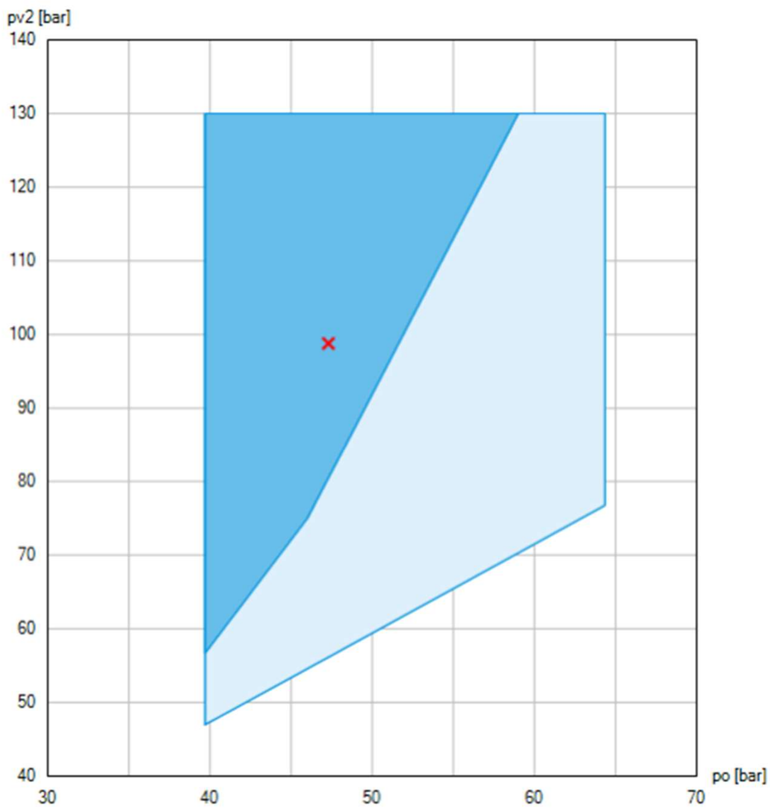
Evaporation temperatures < 5°C (40 bar) with the compressor type SH on request!

1) Value calculated. Deviation to reality possible.

2) The stated value of the discharge end temperature is a mere calculated value. Additional cooling and heat dissipation are not considered. Deviations (particularly in deep freezing applications) from the real measured discharge temperature during operation are possible.

HGX46/345-4 SH CO2 T
Engine: 440-480V Y/YY -3- 60Hz PW
Refrigerant: R744
Subject:

Operating limits



Unlimited application range

Unlimited application range (compressor with DCR22 CO2 flexxCO2NTROL not permitted - range preliminary)

Compressor operation is possible within the limits shown on the diagrams of application. Compressor application limits should not be chosen for design purposes or continuous operation. Evaporation temperatures < 5°C (40 bar) with the compressor type SH on request!

HGX46/345-4 SH CO2 T

Engine: 440-480V Y/YY -3- 60Hz PW

Refrigerant: R744

Subject:

Technical data

Number of cylinders / Bore / Stroke	6 / 40 mm / 46 mm
Displacement 50/60 Hz (1450/1740 ¹/min)	30,20 / 36,20 m³/h
Voltage [1]	380-420V Y/YY -3- 50Hz PW
	440-480V Y/YY -3- 60Hz PW
Winding divided into	50% / 50%
Max. working current [2]	92.3 A
Max. power consumption [2]	65.5 kW
Starting current (rotor blocked) [2]	222.0 / 361.0 A
Motor protection	INT69 G
Protection terminal box	IP 66
Weight	242 kg
Frequency range [3]	20 - 70 Hz
Max. permissible overpressure (g) (LP/HP) [4]	100 / 150 bar
Connection suction line SV	35 mm - 1 3/8 "
Connection discharge line DV	28 mm - 1 1/8 "
Lubrication	Oil pump
Oil type R744	Bock C 170 E
Oil charge	2,6 Ltr.
Dimensions Length / Width / Height	774 / 466 / 403 mm
Sound power level L_{WA} [5]	86 dB(A) @ +5 °C / 100 bar / 10 K
Sound pressure level L_{pA} [5]	73 dB(A) @ +5 °C / 100 bar / 10 K

1) Tolerance (± 10%) relates to the mean value of the voltage range. Other voltages and current types on request

All data are based on voltage rms values

2) - The stated value for the max. power consumption is valid for the adjusted power supply.

- Starting current (rotor blocked):
  - Part winding (PW) motors: Winding 1 / Winding 1+2
  - Delta/Star (Δ/Y) motors: Δ / Y

- Take account of the max. operating current / max. power consumption for designing motor contractors, feed lines, fuses and motor protection switches. Motor contractors: Consumption category AC3.

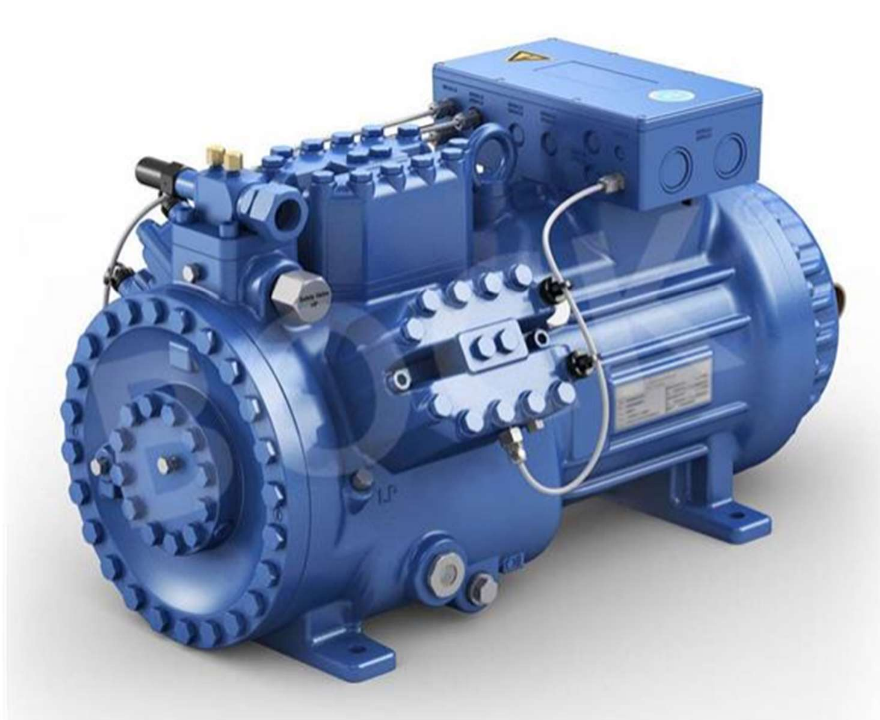
3) The maximum permissible working current of the compressor (Imax) must not be exceeded. Take account of the guidelines for use of frequency inverter (see compressor assembly instruction or selection software).

4) LP = Low pressure
HP = High pressure

HGX46/345-4 SH CO2 T
Engine: 440-480V Y/YY -3- 60Hz PW
Refrigerant: R744
Subject:

Product photo

Picture similar and/or with accessories.



Appendix 4. Simulation results for Case 2, Oilon Heat pumps

4.1 S2000 heat pump with R1234ze refrigerant

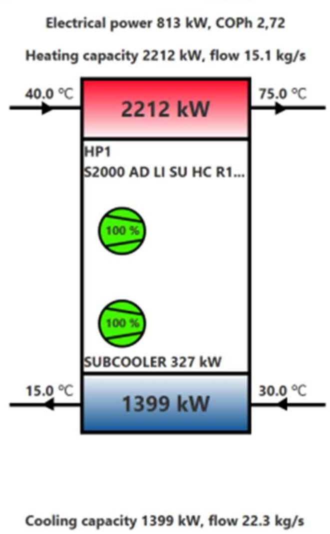
oilon

Quotation
2023-04-17

Oilon selection tool Case 2 +14_+75
1/5

PERFORMANCE (±5 % ACCURACY)

Type	
Heat pumps	1
Heating capacity	2212 kW
Refrigeration capacity acc. to EN 12900	1399 kW
Power consumption	813 kW
COP	2.72
Heat sink (condenser)	
Type of heating medium	water
Heat sink inlet temperature	40.0 °C
Heat sink outlet temperature	75.0 °C
Flow	15.4 l/s
Pressure loss in heat exchanger	21 kPa
Heat source (evaporator)	
Type of coolant	water
Heat source inlet temperature	30.0 °C
Heat source outlet temperature	15.0 °C
Flow	22.4 l/s
Pressure loss in heat exchanger	3 kPa

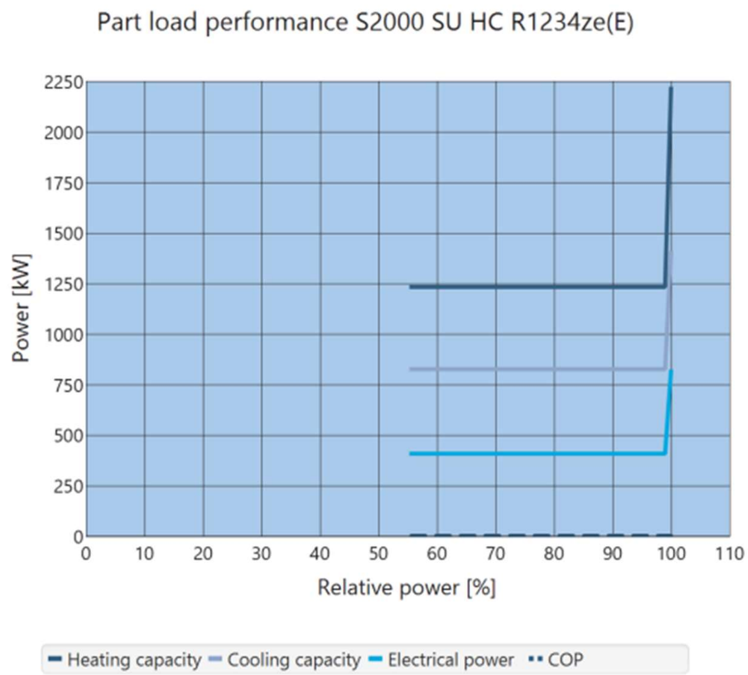


ERRORS AND WARNINGS

[HP1 S2000] Contact Oilon for a verification of the selection.
[HP1 S2000] Heat exchanger calculation should be enabled for improved calculation accuracy

PERFORMANCE - HP1 - S2000 AD LI SU HC R1234ZE(E)

CAPACITY CONTROL



CONDENSER

Flow	15.1 kg/s
Pressure drop	1 kPa

SUBCOOLER

Flow	2.3 kg/s
Pressure drop	20 kPa

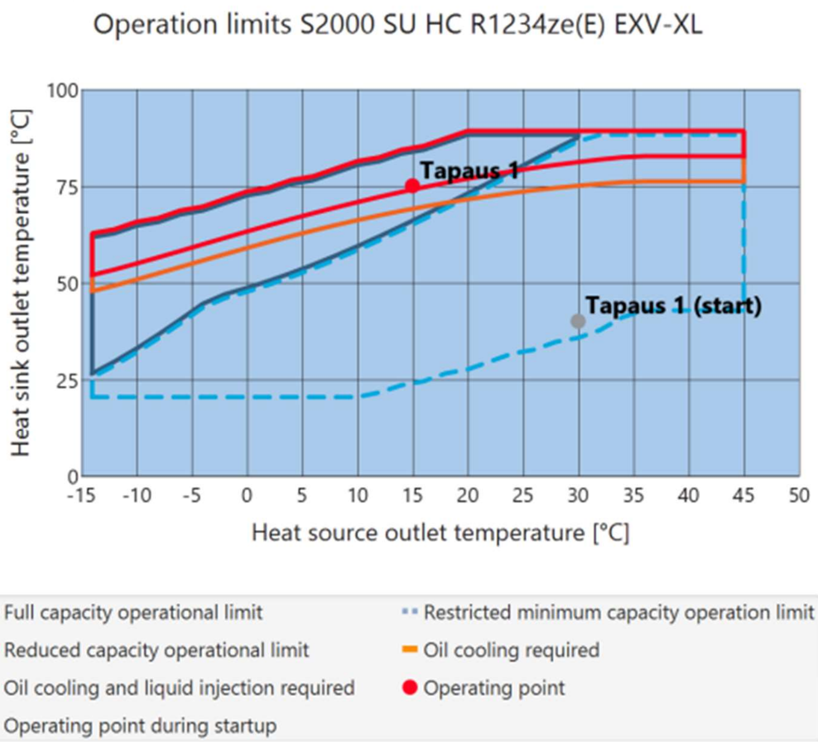
EVAPORATOR

Flow	22.3 kg/s
Pressure drop	3 kPa

OPERATION LIMITS

Maximum allowed heat source (evaporator) inlet temperature	50.0 °C
Maximum allowed heat source (evaporator) outlet temperature	45.0 °C
Minimum allowed heat source (evaporator) inlet temperature	-9.0 °C
Minimum allowed heat source (evaporator) outlet temperature	-14.0 °C

The chart below illustrates the allowed operation limits for the heat pump. The allowed operation limits are dependent on many factors, such as heat pump type, refrigerant, and component selections. Operation outside these limits is not allowed. Operation within the reduced capacity area will lead to a reduction in the performance. Operation in the restricted minimum capacity area is allowed; however, the allowed minimum capacity within this area cannot be determined beforehand. Suitable heat transfer medium with low enough a freezing point must be used as the heat source (evaporator).

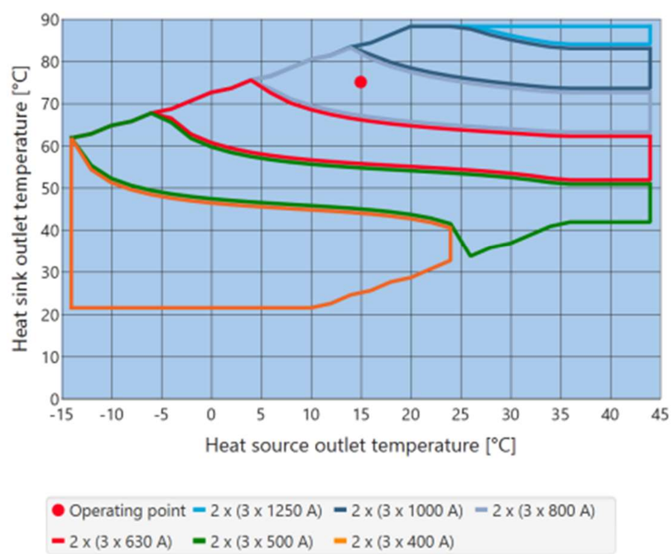


ELECTRICAL DETAILS

Actual power at design point	813 kW
Reactive power at design point	418 kvar
Apparent power at design point	914 kVA
Power factor at design point	0.889
Current at design point	1318 A
Starting current at design point	2637 A
Current at the most demanding operating point	1997 A
Starting current at the most demanding operating point	3993 A
Recommended fuse size for a maximum range of operation	2 x (3 x 1250 A)

A smaller fuse size can be selected from the chart below. If a smaller than recommended fuse size is selected, the customer must ensure that the operating conditions stay within the fuse size limits or that the maximum electrical power draw of the heat pump is actively limited. The customer must ensure that the selected fuse complies with local regulations.

Fuse size selection S2000 SU HC R1234ze(E)



4.2 2 x S1500 Heat pumps in series with R1234ze refrigerant

Quotation

2023-04-17

Oilon selection tool Case 2 +14_+75, improved

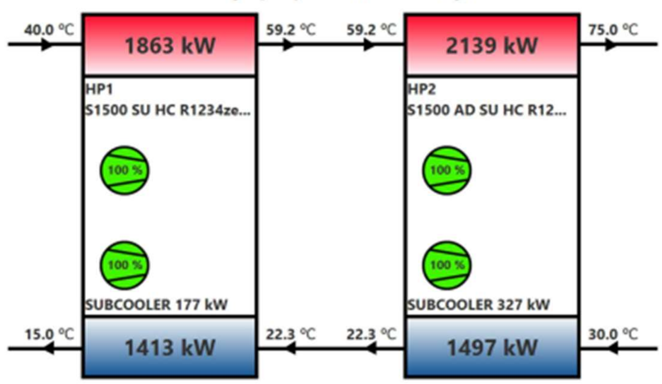
1/8

PERFORMANCE (±5 % ACCURACY)

Type	
Heat pumps	2
Heating capacity	4003 kW
Refrigeration capacity acc. to EN 12900	2910 kW
Power consumption	1094 kW
COP	3.66
Heat sink (condenser)	
Type of heating medium	water
Heat sink inlet temperature	40.0 °C
Heat sink outlet temperature	75.0 °C
Flow	27.8 l/s
Pressure loss in heat exchanger	29 kPa
Heat source (evaporator)	
Type of coolant	water
Heat source inlet temperature	30.0 °C
Heat source outlet temperature	15.0 °C
Flow	46.5 l/s
Pressure loss in heat exchanger	42 kPa

Electrical power 1094 kW, COPh 3,66

Heating capacity 4003 kW, flow 27.3 kg/s



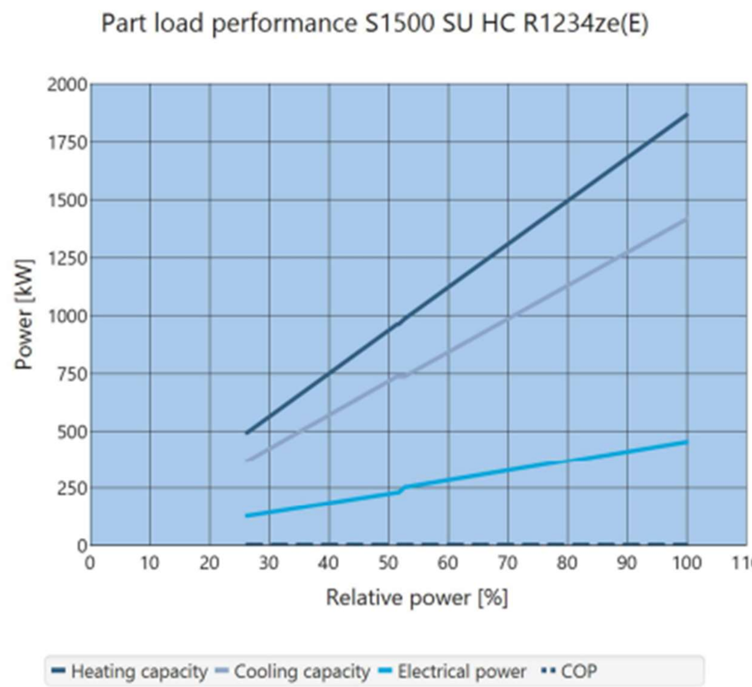
Cooling capacity 2910 kW, flow 46.4 kg/s

ERRORS AND WARNINGS

[HP1 S1500] Contact Oilon for a verification of the selection.
[HP1 S1500] Heat exchanger calculation should be enabled for improved calculation accuracy
[HP2 S1500] Contact Oilon for a verification of the selection.
[HP2 S1500] Heat exchanger calculation should be enabled for improved calculation accuracy

PERFORMANCE - HP1 - S1500 SU HC R1234ZE(E)

CAPACITY CONTROL



CONDENSER

Flow	27.3 kg/s
Pressure drop	4 kPa

SUBCOOLER

Flow	2.3 kg/s
Pressure drop	20 kPa

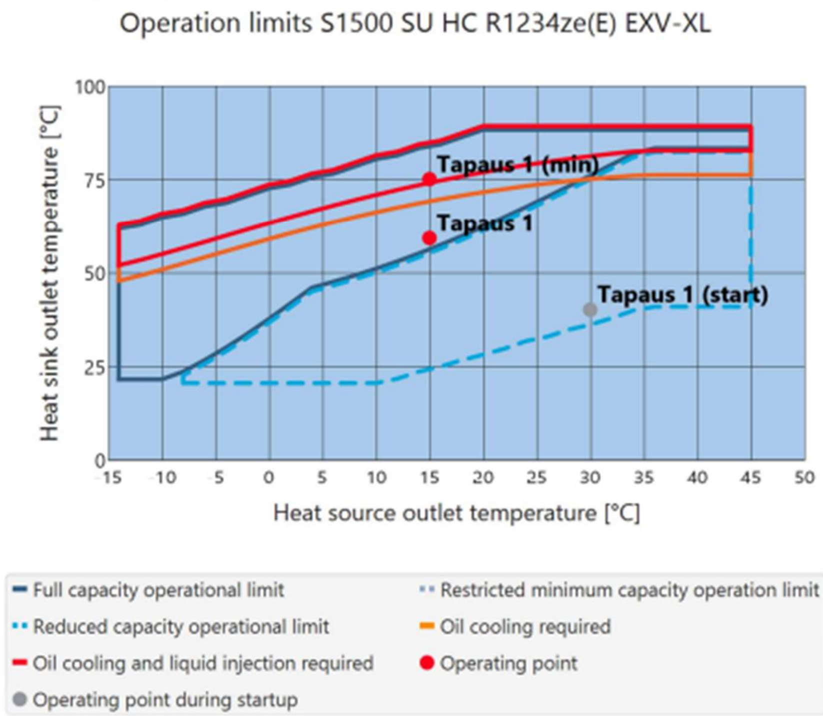
EVAPORATOR

Flow	46.4 kg/s
Pressure drop	21 kPa

OPERATION LIMITS

Maximum allowed heat source (evaporator) inlet temperature	50.0 °C
Maximum allowed heat source (evaporator) outlet temperature	45.0 °C
Minimum allowed heat source (evaporator) inlet temperature	-9.0 °C
Minimum allowed heat source (evaporator) outlet temperature	-14.0 °C

The chart below illustrates the allowed operation limits for the heat pump. The allowed operation limits are dependent on many factors, such as heat pump type, refrigerant, and component selections. Operation outside these limits is not allowed. Operation within the reduced capacity area will lead to a reduction in the performance. Operation in the restricted minimum capacity area is allowed; however, the allowed minimum capacity within this area cannot be determined beforehand. Suitable heat transfer medium with low enough a freezing point must be used as the heat source (evaporator).

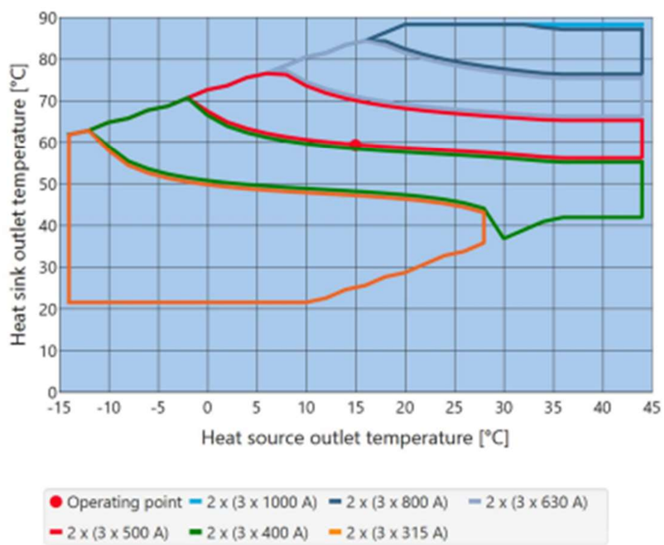


ELECTRICAL DETAILS

Actual power at design point	451 kW
Reactive power at design point	233 kvar
Apparent power at design point	508 kVA
Power factor at design point	0.888
Current at design point	733 A
Starting current at design point	1465 A
Current at the most demanding operating point	1478 A
Starting current at the most demanding operating point	2956 A
Recommended fuse size for a maximum range of operation	2 x (3 x 1000 A)

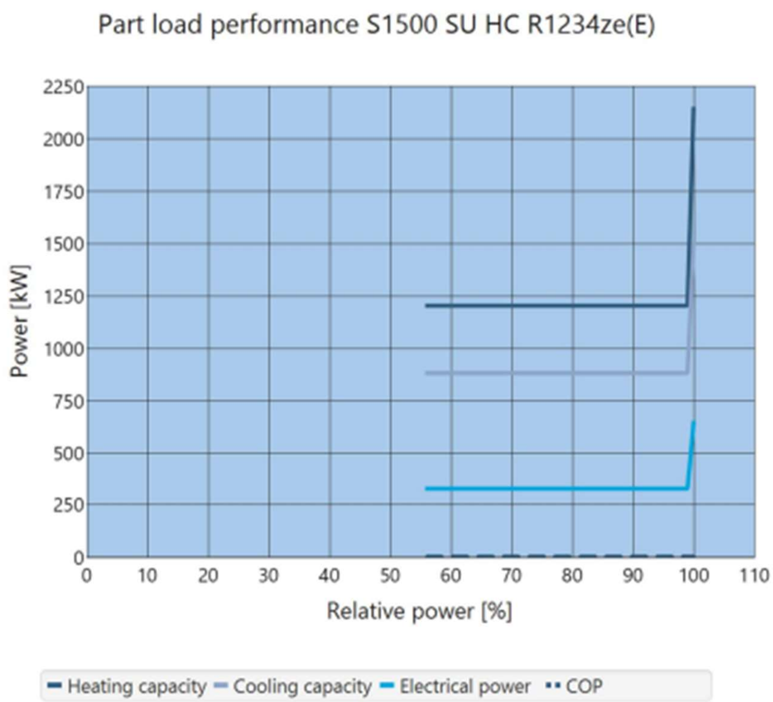
A smaller fuse size can be selected from the chart below. If a smaller than recommended fuse size is selected, the customer must ensure that the operating conditions stay within the fuse size limits or that the maximum electrical power draw of the heat pump is actively limited. The customer must ensure that the selected fuse complies with local regulations.

Fuse size selection S1500 SU HC R1234ze(E)



PERFORMANCE - HP2 - S1500 AD SU HC R1234ZE(E)

CAPACITY CONTROL



CONDENSER

Flow	27.3 kg/s
Pressure drop	4 kPa

SUBCOOLER

Flow	2.3 kg/s
Pressure drop	20 kPa

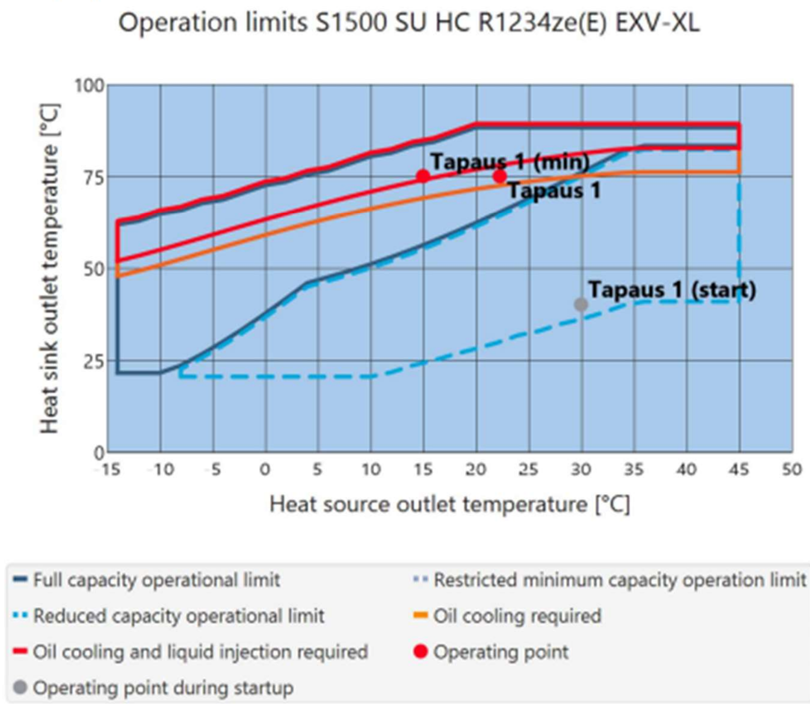
EVAPORATOR

Flow	46.4 kg/s
Pressure drop	21 kPa

OPERATION LIMITS

Maximum allowed heat source (evaporator) inlet temperature	50.0 °C
Maximum allowed heat source (evaporator) outlet temperature	45.0 °C
Minimum allowed heat source (evaporator) inlet temperature	-9.0 °C
Minimum allowed heat source (evaporator) outlet temperature	-14.0 °C

The chart below illustrates the allowed operation limits for the heat pump. The allowed operation limits are dependent on many factors, such as heat pump type, refrigerant, and component selections. Operation outside these limits is not allowed. Operation within the reduced capacity area will lead to a reduction in the performance. Operation in the restricted minimum capacity area is allowed; however, the allowed minimum capacity within this area cannot be determined beforehand. Suitable heat transfer medium with low enough a freezing point must be used as the heat source (evaporator).

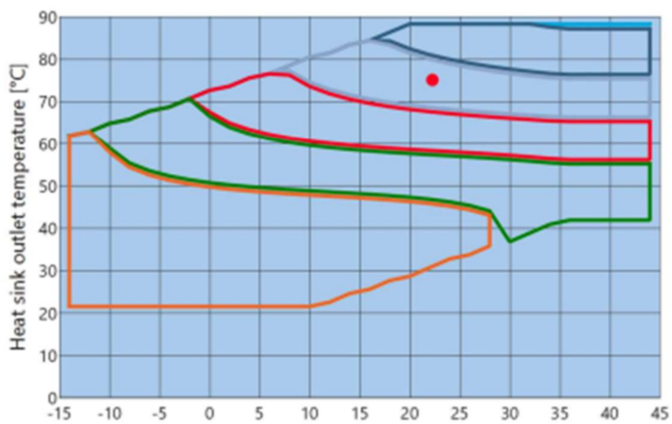


ELECTRICAL DETAILS

Actual power at design point	643 kW
Reactive power at design point	333 kvar
Apparent power at design point	724 kVA
Power factor at design point	0.888
Current at design point	1044 A
Starting current at design point	2088 A
Current at the most demanding operating point	1478 A
Starting current at the most demanding operating point	2956 A
Recommended fuse size for a maximum range of operation	2 x (3 x 1000 A)

A smaller fuse size can be selected from the chart below. If a smaller than recommended fuse size is selected, the customer must ensure that the operating conditions stay within the fuse size limits or that the maximum electrical power draw of the heat pump is actively limited. The customer must ensure that the selected fuse complies with local regulations.

Fuse size selection S1500 SU HC R1234ze(E)



Appendix 5. Simulation results for Case 2, Johnson Controls Heat pumps

4/17/23, 10:37 AM Sabroe MatchMaster Program COMP1

SABROE BY JOHNSON CONTROLS

Sabroe

Refrigeration Plant Computation

Version 32.52

File : Moilas_oy_nykyinen	Ref : PKO		Page : 1
Date : 2023/04/17	Time : 10.36.38		
User : GENERIC USER LEVEL 4			
Prog : COMP1/409901	Print : MIE ver. 9.11.19041.0		

SINGLE STAGE COMPRESSOR

compressor type	HPX 716 VSD	refrigerant	R 717
number of compressors	1.00	evaporating temperature	10.2 deg.C
compressor load	100.0 %	condensing temperature	79.0 deg.C
drive shaft speed	1800.0 RPM (list)	total suction superheat	0.0 K
no. of working cylinders:	16	suction line superheat	0.0 K
drive type	direct	total liquid subcooling	0.9 K
suction line loss	0.5 K		
discharge line loss	0.5 K		
total cooling capacity	438.4 kW	total shaft power req.	202.4 kW
		drive shaft torque	1074. Nm
total heating capacity	604. kW	cooling cap./shaft power ratio	2.17
equipment for head cooling	by WHC-system		
equipment for oil cooling	by WHC-system		

oil separator: OVUR 3209 (77)

operating conditions:

suction pressure	6.07 bar_a	discharge pressure	40.95 bar_a
suction temperature	9.60 deg.C	discharge temperature	174.51 deg.C
suction specific volume	0.2081 m3/kg	discharge specific volume	0.0483 m3/kg
enthalpy difference (ref.)	891.62 kJ/kg	condenser subcooled liquid density	509.7 kg/m3
suction side mass flow	0.4917 kg/s	evaporator saturated liquid density	624.5 kg/m3
swept volume	532.0 m3/h	pressure ratio (p2/p1)	6.75

errors and warnings:
NB: Please use the online drawing generator to get exact measures.
NB: sound level computation - no data available for this compressor
NB: design limits violation - please run Design Limits Check !

Full load performance data for chillers and other refrigeration systems are according to ISO-R916.
Measurement tolerances according to EN13771.
Data subject to change without notice.

Sabroe

Refrigerant Plant Computation

Version 32.52

File	: Moilas_oy_nykyinen	Ref	: PKO	Page :	2
Date	: 2023/04/17	Time	: 10.36.38		
User	: GENERIC USER LEVEL 4				
Prog	: COMP1/409901	Print	: def. not found		

SABROE base refrigerant composition

refrigerant designation R-717
number of elements in mixture 1 (1)

mixture critical temperature	132.25 deg.C	evaporating line temperature glide	0.00 K
mixture critical pressure	113.33 bar_a	condensing line temperature glide	0.00 K
mixture average mol. weight	17.03 kg/kmol	mixture surface tension at TE	.2870E-01 N/m

number		name	mol-%	weight-%
1	R717	ammonia	100.000	100.000

errors and warnings:

errors and warnings:

Full load performance data for chillers and other refrigeration systems are according to ISO-R916.
Measurement tolerances according to EN13771.
Data subject to change without notice.

SABROE
BY JOHNSON CONTROLS

Sabroe

Refrigeration Plant Computation

Version 32.52

File : Moilas_oy_nykyinen	Ref : PKO	Page : 3	
Date : 2023/04/17	Time : 10.36.38		
User : GENERIC USER LEVEL 4			
Prog : COMP1/409901	Print : def. not found		

EVAPORATOR

evaporator type	ESRE 700501	number of evaporators	1.00
primary side:			
primary refrigerant	R-717	total capacity	438.4 kW
evaporating temperature	10.2 deg.C	mean temperature diff.	9.65 K
		fouling factor	0.000035 m2.K/W
inlet velocity - prim. side	5.84 m/s	outlet velocity - prim. side	7.33 m/s
secondary side:			
secondary refrigerant (200) WATER			
inlet temperature	30.0 deg.C		
outlet temperature	14.0 deg.C	total flow	23.6 m3/h
pressure loss	10.7 kPa		
velocity	0.35 m/s		
density	997.5 kg/m3	specific heat capacity	4.184 kJ/kg.K
dynamic viscosity	0.984 Cpoise	thermal conductivity	0.602 W/m.K
inlet velocity - sec. side	0.37 m/s	outlet velocity - sec. side	0.37 m/s
min. wall temperature	12.1 deg.C		
		secondary side pass number	1
built-in liquid separator performance:		separator speed	0.56 m/s
separator pressure loss	0.0 K	velocity ratio (cmax/cgas)	0.52
special PHE output:			
no. of cassettes and type (hot/cold side)	1*49 MW/1*50 MG	service transfer coefficient	2357.7 W/m2K
design/rating mode	design	clean transfer coefficient	2699.9 W/m2K
plate material	AISI-316	refrigerant pressure loss	0.32 mbg
plate thickness	0.6 mm	margin	5.00 %
max. pressure loss sec. side	10.00 mbg	available liquid head	0.40 mbg
primary side connection - in/out	1/1	quality of vapour	0.85
secondary side connection - in/out	2/2	excessive area	0.00 %
hot side channel pressure loss	10.6 kPa		
cold side channel pressure loss	0.25 mbg		
PAC liquid separator type			

errors and warnings:
NB: suitable for closed systems only (dp-channel < 3.0 mbg.)
NB: add demister!

Sabroe

Refrigeration Plant Computation

Version 32.52

File	:	Moilas_oy_nykyinen	Ref	:	PKO	Page :	4
Date	:	2023/04/17	Time	:	10.36.38		
User	:	GENERIC USER LEVEL 4					
Prog	:	COMP1/409901	Print	:	def. not found		

CONDENSER

condenser type	HPSH 4HH-80/2/1		number of condensers	1.00	
primary side:					
primary refrigerant	R-717		total capacity	604.0	kW
condensing temperature	79.0	deg.C	mean temperature diff.	27.32	K
condenser liquid subcooling	0.9	K			
			fouling factor	0.000020	m2.K/W
primary side pass number	1				
inlet velocity - prim. side	3.89	m/s	outlet velocity - prim. side	0.11	m/s
secondary side:					
secondary refrigerant (200) WATER					
inlet temperature	40.0	deg.C			
outlet temperature	75.0	deg.C	total flow	15.1	m3/h
pressure loss	27.0	kPa			
velocity	0.78	m/s			
density	984.5	kg/m3	specific heat capacity	4.183	kJ/kg.K
dynamic viscosity	0.484	Cpoise	thermal conductivity	0.648	W/m.K
inlet velocity - sec. side	0.78	m/s	outlet velocity - sec. side	0.78	m/s
			secondary side pass number	2	
special PHE output:					
no. of cassettes and type (hot/cold side)	1*40 HH/1*39 HH		service transfer coefficient	1767.0	W/m2K
design/rating mode	design		clean transfer coefficient	2032.0	W/m2K
plate material	AISI-316		refrigerant pressure loss	0.05	mbg
plate thickness	0.7	mm			
max. pressure loss sec. side	10.00	mbg			
primary side connection - in/out	1/1		superheated vapour temp.	174.51	deg.C
secondary side connection - in/out	1/1		excessive area	5.00	%
errors and warnings:					

Sabroe

Refrigeration Plant Computation

Version 32.52

File	: Moilas_oy_nykyinen	Ref	: PKO	Page :	5
Date	: 2023/04/17	Time	: 10.36.38		
User	: GENERIC USER LEVEL 4				
Prog	: COMP1/409901	Print	: def. not found		

PLANT DATA - HeatPAC716V-A

plant load percentage	100.0	%
plant cooling capacity	438.4	kW
plant heating capacity	604.4	kW
totalt shaft power consumption	202.4	kW
total line power consumption	0.0	kW
shaft cooling power ratio	2.17	
shaft heating power ratio	2.99	
line cooling power ratio	0.00	
line heating power ratio	0.00	

unit approx. refrigerant charge	18.	kg
unit approx. evaporator brine/water charge	20.	kg
unit approx. condensor water charge	16.	kg
number of vibration dampers	4	
chiller unit expansion valve	1 x HFI-060FD	
unit expansion valve load	27.0	%
Sound level (max)	*	dB(A)

errors and warnings:
NB: Please use the online drawing generator to get exact measures.

Appendix 6. Simulation results for Case 2, GEA Heat pumps

Scope of Supply

GEA Engineering for a better world.

Item 1) 1 x GEA RedAstrum RR (W)

Ammonia-Liquid Heat Pump Type GEA RedAstrum RR (W)

High-efficient ammonia liquid heat pump with GEA Grasso screw compressor, evaporator and condenser in execution as shell & plate type heat exchanger, evaporator as combined device (evaporator/liquid separator) as a compact, factory packaged unit, ready for connection on site. The GEA RedAstrum series is equipped with VSD (variable speed drive) by default. The base tub is executed as a drip tray.

1. Technical data

	Cond. 1	
Refrigerant	R717	
Rated cooling capacity	1485	kW
Rated heating capacity	1998	kW
EER - Energy Efficiency Ratio (line)	2,64	
EER - Energy Efficiency Ratio (shaft)	2,89	
COP - Coefficient of Performance (line)	3,55	
COP - Coefficient of Performance (shaft)	3,89	
Total electrical consumption (line)	563	kW

Evaporator / liquid separator

Plate heat-exchanger of fully welded design with integrated liquid separator; stainless steel secondary refrigerant ports, completed with flanges and counter flanges.

Medium	Water (non-corrosive)	
Secondary refrigerant inlet temperature	24,0	°C
Secondary refrigerant outlet temperature	14,0	°C
Secondary refrigerant volume flow	126,1	m³/h
Secondary refrigerant NB in/out	100	DN
Design pressure liquid side	16	bar(g)
Design pressure refrigerant side	16	bar(g)
Secondary refrigerant min. circuit pressure	1	bar(g)
Plate material	AISI 316L	
Pressure drop	< 1,0	bar

Heat carrier circuit with condenser, oil cooler and subcooler

Plate heat-exchanger of fully welded design; stainless steel media ports, completed with flanges and counter flanges.

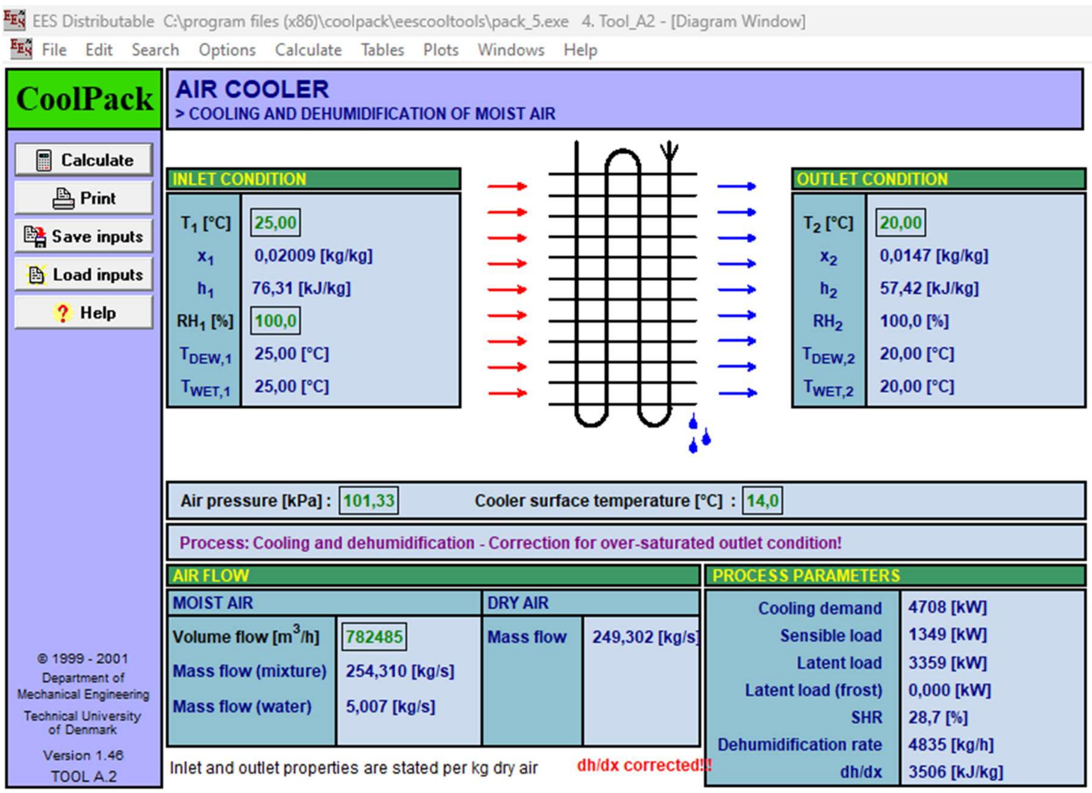
Medium	Water (non-corrosive)	
Heat carrier inlet temperature	40,0	°C
Heat carrier outlet temperature	75,0	°C
Heat carrier volume flow	49,4	m³/h
Heat carrier NB in/out	100	DN
Design pressure liquid side	16	bar(g)
Design pressure refrigerant side	40	bar(g)
Heat carrier min. circuit pressure	1	bar(g)
Cassette material	AISI 316L	
Pressure drop	< 1,2	bar

Please notice that in case of water as secondary refrigerant or cooling medium the selection of the heat exchanger plate material is based on the assumption of a water quality comparable to VDI rule 3803 and a maximum concentration of chloride ions of 200 ppm (evaporator) and 150 ppm (condenser / oil cooler). Also required are pH values of both – secondary refrigerant and cooling media – between 7 and 9.

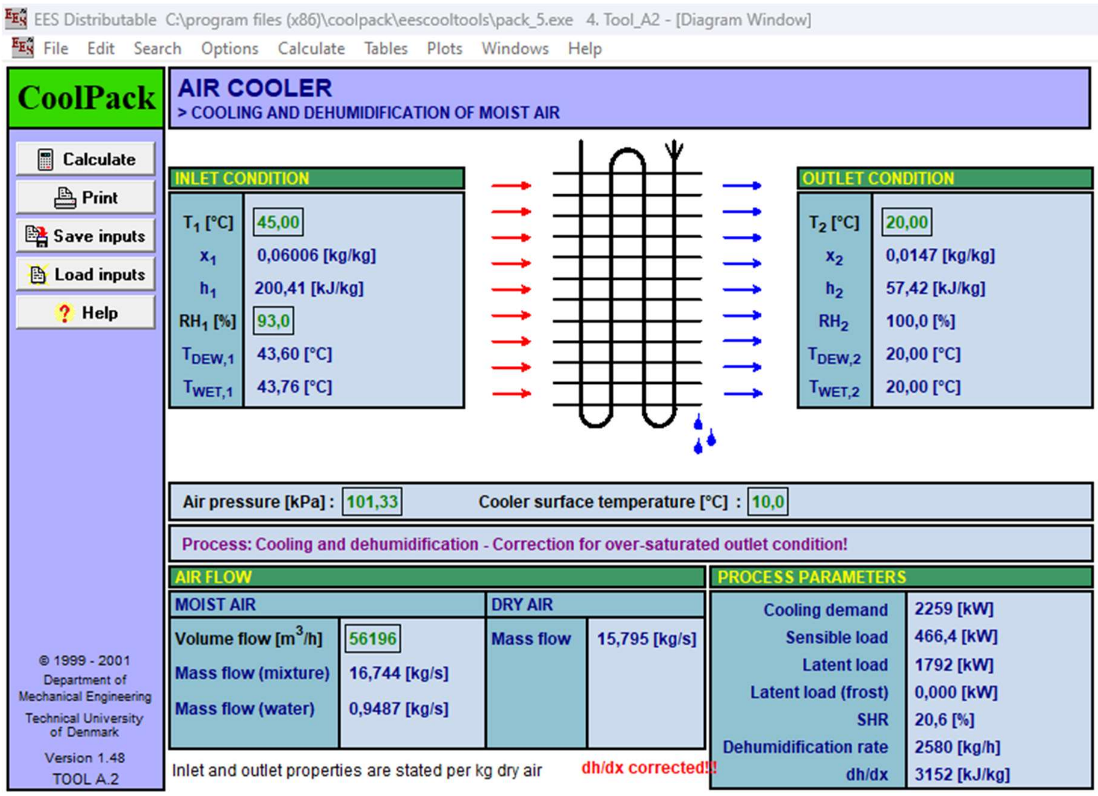
Project Name:		Prepared By:	Pasi Kolehmainen
Proposal Number:		Company:	Adven Oy
Customer Name:	New Contact	Date:	20.4.2023

Appendix 7. Simulation results for Case 1 and Case 2, CoolPack

Case 1



Case 2


EES Distributable C:\program files (x86)\coolpack\eescooltools\pack_5.exe 4. Tool_A2 - [Diagram Window]
File Edit Search Options Calculate Tables Plots Windows Help
CoolPack
AIR COOLER
> COOLING AND DEHUMIDIFICATION OF MOIST AIR
Calculate
Print
Save inputs
Load inputs
Help
INLET CONDITION
T1 [°C] 45,00
x1 0,06006 [kg/kg]
h1 200,41 [kJ/kg]
RH1 [%] 93,0
TDEW,1 43,60 [°C]
TWET,1 43,76 [°C]
OUTLET CONDITION
T2 [°C] 20,00
x2 0,0147 [kg/kg]
h2 57,42 [kJ/kg]
RH2 100,0 [%]
TDEW,2 20,00 [°C]
TWET,2 20,00 [°C]
Air pressure [kPa] : 101,33
Cooler surface temperature [°C] : 10,0
Process: Cooling and dehumidification - Correction for over-saturated outlet condition!
AIR FLOW
MOIST AIR
Volume flow [m3/h] 56196
Mass flow (mixture) 16,744 [kg/s]
Mass flow (water) 0,9487 [kg/s]
DRY AIR
Mass flow 15,795 [kg/s]
PROCESS PARAMETERS
Cooling demand 2259 [kW]
Sensible load 466,4 [kW]
Latent load 1792 [kW]
Latent load (frost) 0,000 [kW]
SHR 20,6 [%]
Dehumidification rate 2580 [kg/h]
dh/dx 3152 [kJ/kg]
© 1999 - 2001
Department of Mechanical Engineering
Technical University of Denmark
Version 1.48
TOOL A.2
Inlet and outlet properties are stated per kg dry air
dh/dx corrected!!!

Appendix 8. Simulation results for Case 2, LU-VE

F5-CA-8000-2520-10/10-2.5; S=180; N=4; X=0; VV

Performance / Air side

Variable:	Actual:
Total capacity:	2841 kW
Sensible cap.:	506 kW
SHR:	0,18
Air in:	45,0 °C / 92 %
Air out:	19,9°C / 99 %

Process (inside)

Variable:	Calculated:
Liquid in:	14,0 °C
Liquid out:	31,87 °C
Liquid flow:	38,2 l/s, 38,1 kg/s
Velocity:	1,67 m/s (Re=23132)
Pressure drop:	106 kPa
Liquid:	Water

Fans

No. of fans:	–
Diameter:	–
Speed:	–
Nom. inp. power:	–
Power input:	–
Max. current:	–
Voltage:	–
Face velocity:	0,88 m/s

Air flow (total):
16,7 m^3/s
60134 m^3/h
20,0 kg/s

dp* (relat. dp):
0%

dp (act./nom.):
27/29 Pa

Coil block

Area:	4510 m^2	Conn.(in):	1xDN 125
Internal vol.:	809 dm^3	Conn.(out):	1xDN 125
Weight, fins:	1212 kg	xx:	...
Weight, tubes:	771 kg		
Weight:	2039 kg		

F5-CA-6500-2520-12/12-2.5; S=216; N=4; X=0; VV

Performance / Air side

Variable:	Actual:
Total capacity:	2841 kW
Sensible cap.:	505 kW
SHR:	0,18
Air in:	45,0 °C / 92 %
Air out:	19,9°C / 99 %

Process (inside)

Variable:	Calculated:
Liquid in:	14,0 °C
Liquid out:	31,87 °C
Liquid flow:	38,2 l/s, 38,1 kg/s
Velocity:	1,39 m/s (Re=19275)
Pressure drop:	67,1 kPa
Liquid:	Water

Fans

No. of fans:	–
Diameter:	–
Speed:	–
Nom. inp. power:	–
Power input:	–
Max. current:	–
Voltage:	–
Face velocity:	1,08 m/s

Air flow (total):
16,7 m^3/s
60135 m^3/h
20,0 kg/s

dp* (relat. dp):
0%

dp (act./nom.):
45/49 Pa

Coil block

Area:	4397 m^2	Conn.(in):	1xDN 125
Internal vol.:	794 dm^3	Conn.(out):	1xDN 125
Weight, fins:	1181 kg	xx:	...
Weight, tubes:	755 kg		
Weight:	1993 kg		