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Research report 2

Thermal hydraulic functioning of unconventional systems using renewable resources.
Experimental study

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Abbreviations

COP – Coefficient of performance,

ASHP – Air source heat pump,

WSHP – Water source heat pump,

RCTCs – Refrigerant cycle temperatures coefficients,

CFD – Computational Fluid Dynamics,

w/w – with or without,

NIST – National Institute of Standard and Technology,

GWP – Global Warming Potential,

ODP – Ozone Depletion Potential,

CO_2 – Carbon Dioxide,

R2016a – MATLAB based 2016 SW version,

SW – Software,

AC – Alternative Current,

DHW – Daily Hot Water,

RC – Remote Control,

RMS – Root Mean Square value of sine wave measurements,

Nomenclature

COP_{CD} – Coefficient of Performance in heating, -,

COP_{VP} – Coefficient of Performance in cooling, -,

η_c – compressor efficiency, -

η_{vp} – evaporator efficiency, -

P_{CD} – condenser Power, kW,

P_{VP} – evaporator Power, kW,

P_{EL} – electrical power consumption, kW,

P_{EL_sym} – electrical power for symmetrical loads, kW,

Q_{CD} – liquid flow at condenser, l/min,

Δ_{CD} – temperature difference between condenser and condenser environment, K

Δ_{VP} – temperature difference between evaporator and evaporator environment, K

t_{ml_cond} – secondary line condenser medium logarithmic temperature, K

Δ_{VP_drop} – evaporator temperature drop between inlet and outlet of air heat exchanger, K

c_{tl}, c_{th} – low, respectively high refrigerant temperatures coefficient, -

θ_{CD} – condenser temperature, ˚C

θ_{VP} – evaporator temperature, ˚C

t_C – hot water temperature, ˚C,

t_R – cold water temperature, ˚C,

t_{LL} – liquid line temperature, ˚C,

t_{GL} – gas line temperature, ˚C,

t_{SC} – sub cooling temperature, ˚C,

t_C – overheating temperature, ˚C,

t_{11} - heat exchanger primary input temperature, ˚C,

t_{12} - heat exchanger primary output temperature, ˚C,

t_{evap_man} – evaporator temperature on manifold measurement, ˚C,

t_{cond_man} – condenser temperature on manifold measurement, ˚C,

t_{out_vp} – evaporator air output temperature, ˚C,

ρ_w – water density, kg/m^3,

c_w – water specific heat, kJ/(kg·K),

U_1 – voltage phase 1, V,

U_2 – voltage phase 2, V,

U_3 – voltage phase 3, V,

I_1 – phase 1 current, A,

I_2 – phase 2 current, A,

I_3 – phase 3 current, A,

cos(Φ) – electrical power factor, -,

E_{el} – electrical energy for nonsymmetrical loads, kW·h,

E_{sim} – electrical energy for symmetrical loads, kWh,

τ – sampling time, h,

Introduction

Earth is, and will remain an exploitation land where humans discover nature laws and were they do their own activities, evaluates and explore earth behavior. Using natural laws, humans activities are continuously changing, defined and bringing together, in his great and amazing religion and science immensity.

Man walks without a way of return on the road of understanding the space-temporal Universe, with the unsatisfactory impulse of inner determination in defining the behavior of nature. When energy, as the impulse of life, changes the habitat or system size, man has a role, assumed, to understand what is happening around him, and how a change leads to propagation, both inside and outside, of complexes coordinated elements. On the one hand, the energy leads to the balance of the systems at a given moment, but also to the modification of their parameters once the energy exchange beyond the imagined own borders.

The cause, due to energy exchange, leads to an effect in natural systems, in human habitats and in the planet behavior. Imagining the connection between cause and effect, understanding it and then experimenting with the understanding of the date, man is able to achieve an improvement in the sense of his own comfort and protection of the surrounding environment.

Humans will never stop inspiring themselves, will copy all known natural laws, for eternity will be inspired by nature, which is an infinite ocean of governing laws of surrounding substances natural behavior. It seems that energy, have many types of transmission and consequences like heat, due to atomic agitation, light, due to electronic movements etc. Energy, in a context, create or change system parameters such temperature, wind or precipitation, change material molecular link, creating or disrupting substances by accumulation respectively release of captured energy.

The context of energy, together with natural materials, helps humans defining and keeps their self-comfort, thermal comfort being one sine qua non condition in all leaving spaces.

Historically, the thermal comfort was kept by releasing captured energy from molecular link between Carbon and Hydrogen in natural substances like coal, petrol or gas. But this type of maintaining thermal comfort conducts to climatic changes, in which more and more global temperature is rising. For summer periods, today is a real challenge to create thermal comfort in conjunction with saving energy to reduce the climatic changes. Because of the most used cooling technology is electricity based, and because the world biggest electricity production is coal based, the context of cooling and heating can be a way to save the planet if entire or partially used primary energy is changed with renewable energy in such a way of keeping the same thermal comfort in leaving spaces.

A way to decrease the primary energy demand is to use reusable and renewable energy, such air, water or ground heat respectively solar energy transmitted to earth by solar irradiation. Unconventional systems to come are developed today as a multilateral technologies definition, where material properties, thermodynamic behavior of heating or cooling systems or electronic units, being linked together in equipment to drive for a better efficiency and cost-effective purpose. A lot of hybrid system were developed until today, such absorption heat pumps together with classical systems (1), solar systems together with classical heating units that burn natural gas or wood residues but also combined obtained energy like heat and power generation in big cities district thermal systems (2). All those today are costly effective analyzed and for improving thermal comfort inside buildings (3). Humans put together their knowledge and share the best practices, take the best configuration as a result of science research for a final design of equipment that drives the air and thermal comfort management in a way satisfying heat or cooling demands but, in same time, not forgetting to preserve environment. As a general rule, we tried over the past years to give all our best comply with the needs and reduce fossils consumption decreasing carbon emissions.

Doing the right is not always known from the beginning; studying and search for the best choice in technology or dynamic algorithm control mix, will always conduct for it. For sure the final decision assures at least people needs, but not only. A real testing will conduct us to all design oversight, will tell us which are the right configuration and if we need a new calibration of algorithm parameters or material changes.

A step in design is testing. An important step of testing is taking the right decision about parameters, with conclusion on results obtained. Assuming a value of a parameter or another will conduct to the real value which not always is the imagined one. Testing in such way, for system parameters search is common known as an experimental study. Putting system elements together, keeping as much as possible, with an accepted and assumed gap, the known parameters in their specification, will give us the right interval of searched parameter/s. Not always what we found is the imagined value, what we see after the experiment conducting to decide if the kind of design is feasible or not.

In research area a good tool for designing is simulation software. Today researchers, engineering, technicians have computational environment to simulate a system, to simulate a theoretical method to write equipment mathematical low, to simulate also management algorithm and find an appropriate value for the minima/maxima of system description function as optimal account. Hear, experiment will confirm or give us a new value of searched parameter to calibrate the real equipment comparing simulation.

The experiment is used, in context described here, for two purposes. One is validation of the system parameters and the second, and most important, calibration of mathematical model describing equipment.

In complex heating systems today developed, the mixing of technologies from different manufacturers just keeping the customers cost-effect needs, building facilities management design engineering offices will always work with equation and national norms, to find the optimal solution. Not always is an easy way. Not always mixing different manufacturers equipment with equations is possible. Approximation for system mathematical laws is it common used today. Find and validate them is a know-how that companies are interested for, being their value-up. The feedback after implantation of build system, and registration of energy consumption and working parameters after system functioning equilibrium, is the experiment of design offices which for sure will calibrate their design tool.

Experiments always will give us a natural behavior eyesight of thermal comfort equipment functioning. Experiments will always be a right choice when we search for calibration parameters levels. For those purposes, experiments will remain the countered less tool in researchers hands. Therefore, imagine the right experiment will be the biggest challenge.

1. Mathematical modeling of heat pump thermal systems

Mathematical modeling of thermal system will always be a challenging engagement, diversity of systems and mixing technologies being a real thread in majority of equipment. Hybrid systems are most frequent subjected to high complexity. Global energy consumption is increasing yearly and climate change being a consequence (4) (5) (6) (7).

Engineering use plurality of tools to estimate energy demand, simulation of the system and customer thermal comfort satisfaction based on mathematical algorithms, physical laws and together with right software. Based on that, complex algorithm or mathematical equations are describing the physical behavior of materials but also their dynamic action on weather parameters or internal buildings air change, heat demand or temperature set-up.

Writing the right equation of system physical description is the most challenging requirement once due to system complexity but not the last one the high number of parameters and interaction between them. Coming with a solution to this dilemma, reducing the system mathematical and physical description complexity is the key of the scope. Taking into consideration the right and most important parameters, globally we can precisely described, into acceptable limits of uncertainty, the dynamic behavior of system when some parameters are change, or to keep into the limits the internal temperature of heated/cooled building.

Internal thermal comfort is also experimental studied by using manikins, in specialized laboratory with essential equation in system description (8). Polynomial end exponential equation approach are the most common used technics. A plurality of experimental methods generate polynomial equation with a direct polynomial regression of measured data. Due to this, this type of application is common used because his simplicity, direct applied on systems but also due to mathematical regression approach which define a lot of today database software or tables statistical analyze. For right system evaluation, polynomial regression can be the best evaluation method in case monitoring of temperature and important parameters is possible and collected data represent a significant period of time. When statistical data are analyzed, the equation obtained, as a polynomial expression, can be used in newer system efficiency evaluation by system parameters variations or simply reading of known diagram, obtained on real equipment's. Therefor the statistical analyze and system equation obtain by this method, give a plus to use it in energetic evaluation of old systems, with decreased efficiency just imposing a comparison between old and new measurements in same conditions or for the same atmospheric parameters such wind, temperature, humidity or solar radiation.

The biggest disadvantage of this type of method make it impossible to be used in case of new types of mixed technologies like hybrid thermal systems. Once installed, those systems are hardly to be changed, due to financial impact and customer satisfaction. A good estimation of functionality and dynamic behavior, being the method need in engineering. Just for this reason and for major disadvantage, the polynomial regression is not a good choice. We can conclude here that for evaluation and confirmation of the system estimation from development stage, when feasibility studies are performed, polynomial regression will help. Therefore we need additional techniques when systems are selected to serve for thermal system of a building or other areas of heating equipment.

Physical model description (9) of thermal systems is an old technique, used from the beginning of computational evaluation of heating equipment or physical systems. Used as a link correlation between real environment and mathematical equation, physical model approach will be kept long time from now on. The necessity of system description in a mathematical converted equation, create a big database of system models, used by engineering, researchers and technicians over the world in a variety of software tools or evaluation techniques. Will remain the graphical understandable representation of natural behavior of matter.

For thermal systems, some basic models with application together with thermodynamic systems laws, were developed last century to help in equation writing for mathematical function optimization but also to drive the cost-effective characteristics of systems. Thermodynamic laws and material parameters such thermal capacity, phase change temperature, thermal resistance, etc. are engaged as equation constants or system dynamic behavior representation. Volume, mass, pressure and temperature are state variable to be used in dynamic behavior by imposing parameters change or energy added. Common used technique to describe dynamic behavior is to fix the parameters in interest. For that, all other parameters changes can be fixed or approximate as a constant value during research. It is right, that in fact, we have different methods to describe and resolve systems of equations using differential equation resolvers or exponential matrices resolvers as for thermal system but for beginning it is helpful to simplify the algorithm, to understood how system work in a dynamic point of view, were energy is added and physical change happened in it. It was demonstrate over the last century decades that staring from the simplest equation and mathematical system description, we are able to find, in the end, a complex equation description, being able to obtain same characteristics or even graphs comparable with experiments results. At least with same trends.

Thermal system for cost reduction are coming today as a mix between different types, with final scope to deliver costly effective heat for building thermal comfort energy demand. Somehow, used in the beginning as refrigerators, the capacity of compression cycle systems to heat the space is, surprising, bigger comparable with the purpose of initial scope. Called heat pump, this equipment is commonly used in building space air conditioning (10) also for cold (11) but even in hot season (12). A lot of studies, theoretical or experimental, were led to understand dynamic behavior, efficiency, heat capacity or customer needs dimensioning.

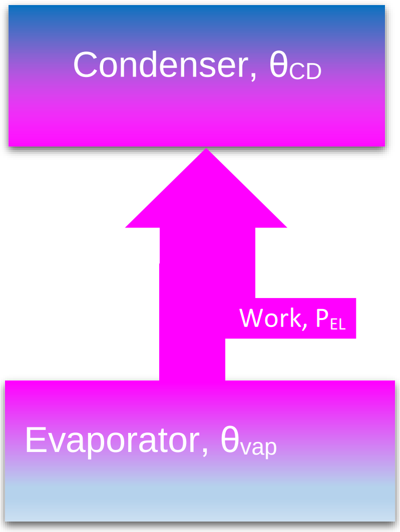

Figure 1 Carnot cycles model

Mixed system such combination of heat-pumps with solar systems, heat accumulation, classical heat generation by fossil burning or wood, is in the early beginning of model description or capacity dimensioning. Old buildings are subjected today to different purpose utilization comparing initial scope or are changed and rehabilitated for cost reduction with thermal comfort heat needs. For this purpose, using entire or a portion from the old heating systems can be a wise decision in terms of cost reduction with implementation but also an efficiency increasing with no big investments. So, just adding heat-pumps to old building heating or cooling systems make it today wanted for investments in industrial or residential areas.

Carnot cycle was historically used as physical model for work to heat and reverse conversion cycle (13) for heat pumps thermal evaluation. Is the starting point. Just writing the right cycle between upper and lower Carnot cycle temperatures, make the difference. Describing the realistic Carnot cycle is a matter of knowing heat pump working principle, the refrigerant cycle used for specific application (14), conduct to confident analyze data after simulation with good result after implementation. A classical schematic representation is commonly known and exposed into Figure 1. This physical model is commonly used to describe a turbine electrical power generator or heat pump functioning description. In left variant of Carnot cycle from Figure 1 is drawn a generator physical model, which can be understood as heat drop into condenser and electrical work generation as consequence of rotating the turbine through the steam comes out. In the right, which is our interest of this paper, the electrical work is used to drive the heat from evaporator to the condenser. From physical model describing the inverse cycle, is directly written the heat balance into a mathematical [Equation 1].

Equation 1

$$P_{CD} = P_{VP} + \eta \cdot P_{EL}$$

In the inverse Carnot cycle it is commonly used the representation of coefficient of performance, COP (15) and given by equation [Equation 2] (16). This term is used prior to efficiency due to his property being higher than one not only sub-unitary, comparing efficiency in Carnot cycle which is by definition a sub unitary term. This coefficient, due to Carnot cycle properties, can be written as a representation between intervals limits of temperatures between which the cycle is working [Equation 3] (17).

Equation 2

$$COP_{CD} = \frac{P_{CD}}{P_{EL}}$$

The transfer of heat from condenser and to evaporator, to the environment of condenser respectively from the environment of evaporator, is realized by the heat pump which should work in different temperatures compared with those environments. The heat pump condenser temperature should be higher than his environment and also the temperature of evaporator of the heat pump is always below the temperature were evaporator is immersed or exposed (18) (19) (20).

Equation 3

$$COP_{CD} = \eta_c \cdot \frac{T_{CD}}{T_{CD} - T_{VP}} = \eta_c \cdot \frac{\theta_{CD} + \Delta_{CD} + 273.15}{\theta_{CD} - \theta_{VP} + \Delta_{CD} + \Delta_{VP}}$$

Equation 4

$$P_{CD} = \eta_c \cdot P_{EL} \cdot \frac{\theta_{CD} + \Delta_{CD} + 273.15}{\theta_{CD} - \theta_{VP} + \Delta_{CD} + \Delta_{VP}}$$

The evaluation of COP is done from the condenser point of view, considering that for heating this performance is a term of interest. As a rapport between condenser and compressor electrical power consumption, and with [Equation 3], we can easily describe the correlation between electrical power consumption, compressor efficiency and heat pump refrigerant cycle specification resulting [Equation 4]. In [Equation 4] the electrical power consumption should be noted that for the heat pump contain all electrical consumers as evaporator ventilator, if ASHP (air source heat pump), or circulating pump, in case WSHP (water source Heat pump), all circulating pumps in downstream of heat pump, automation devices and last but not least compressor power which generally is the biggest consumption element from the heat pump electrical power demand. This is a consequence of the high pressures refrigerant cycles, and was historically noted that is a direct influence in electrical power consumption of compressor comparing working pressure of gas but also pressure differences between inlet and outlet. Volume and temperature it is also a consequence which conduct to this behavior but is a result of gas physical behavior as an equation between pressure, temperature and volume together with specific gas constant.

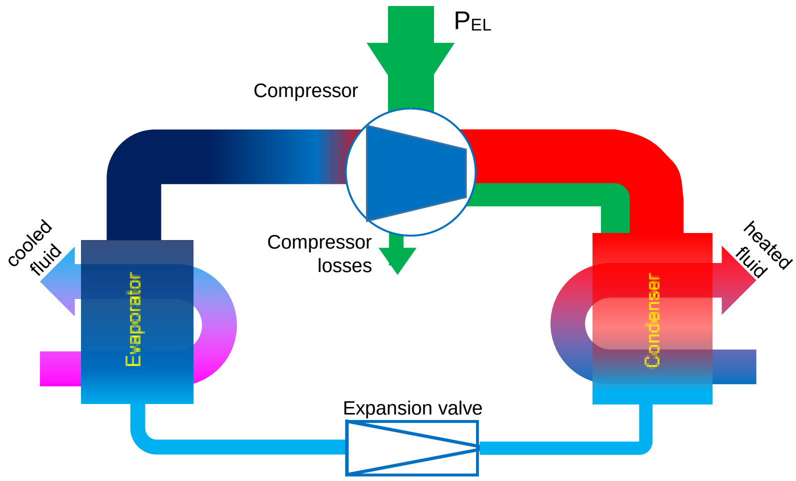

Figure 2 Heat pump scheme

A generic scheme of a heat pump is presented in Figure 2, were can be seen that a fluid is present at evaporator and cooled down, energy being carried to the condenser by the compressor. At the condenser, the fluid from his environment, liquid or air, is heated from an initial state to a new temperature value. The real compressor add heat to entire volume transported to condenser, but a part of it being lost caused by his inefficiency or isentropic efficiency.

For an ASHP or WSHP, for condenser, in case water/liquid transport thermal fluid, the thermal power can be written as thermal balance between entry and exit states of the working fluid, which flow through condenser.

For a heat pump which produce hot water at condenser we can draw a physical schematic diagram, Figure 3, to be able to write the thermal balance which handle the equipment from a mathematical point of view. Diagram present only the compressor, the condenser, schematically and theoretically as a heat exchanger, were the refrigerant condensate and the heat transported by the fluid which is carried into the heating unit by flow pump.

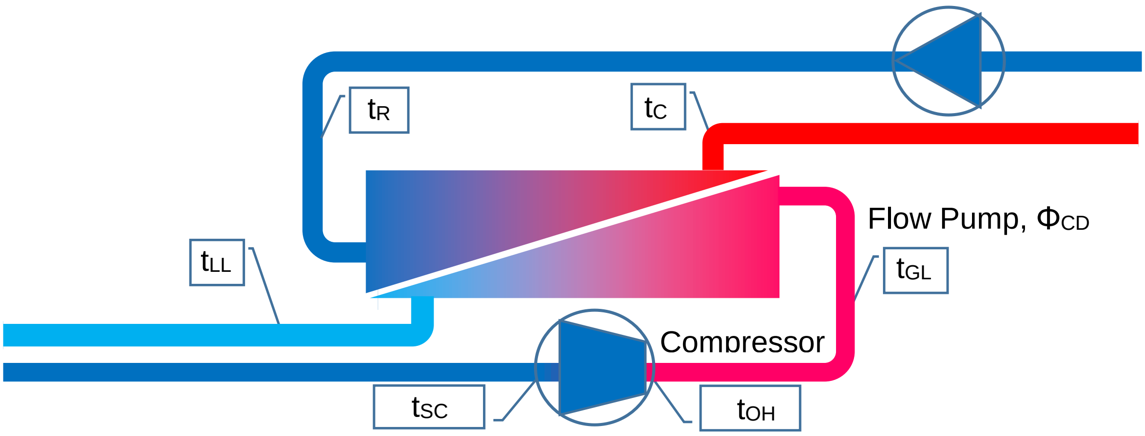

Figure 3 Heat Pump Condenser physical model

All together form the heat pump condenser were is take place two process, one into the primary unit of heat exchanger once with condensing of refrigerant at condensing temperature and secondly, inside secondary unit of the exchanger, were the fluid is heated until reach the hot temperature, schematically noted t_C. Into compressor, the gas is transported by low pressure generation at inlet plug of it or pressure drop; commonly used temperature notation is sub cooling temperature (21) (22), noted schematically tsc which represent the temperature of refrigerant state corresponding pressure at the inlet of compressor. At the inlet plug of condensing primary unit, the temperature of gas compressed start to fall down, reaching the condensing temperature inside heat exchanger. This temperature called gas line temperature and noted schematically t_{GL}, is lower comparing exit compressor gas temperature, which correspond to a higher pressure at compressor outlet boundary (23), a temperature noted t_{OH} and called over heating temperature.

For the purpose of this paper, what happen up and downstream of unit described in Figure 3, is not so important, the mathematical model being written only to describe an equation for condenser power. It is clear that evaporator power and the consumer behavior impose to entire system a dynamic functionality, due to thermal inertia and heating system dimension.

For a given consumer, the power at condenser outlet should be well dimensioned to satisfy the heating/cooling needs for specific period of time but also, together with accumulation system to face all possible edge worst weather conditions always present and unpredictable in cold or hot seasons. Condenser power, when inlet and outlet temperatures from secondary unit of heat exchanger are available, together with fluid mass/volume flow, can be easily written as a product of temperature difference, flow, and fluid characteristics, described by [Equation 5].

It is necessary to give some precisions for this formula. Temperatures are noted as previously paragraph description as inlet and outlet of heat exchanger secondary unit, were the heat from refrigerator is discharged into carrier fluid; his velocity inside exchanger are directly described by flow Q_{CD}, measured in cubic meter per hour. Fluid parameters is described by specific heat c_W respectively fluid density ρ_W. All those together provide a clear dynamic behavior or, in case constant functionality a static state values, being able to create a correlation between heat pump refrigerant cycle statements and consumer thermal power necessity. The constant value 0.016 coming from parameters unit correlation.

Equation 5

$$P_{CD} = 0.016 \cdot Q_{CD} \cdot \rho_w \cdot c_W \cdot (t_C - t_R) \cdot 10^{-3}$$

By equalization, [Equation 4] and [Equation 5] become together the expression of thermal balance between evaporator and condenser environment. This state is mathematically noted as [Equation 6].

Equation 6

$$0.016 \cdot Q_{CD} \cdot \rho_w \cdot c_W \cdot (t_C - t_R) \cdot 10^{-3} = \eta_c \cdot P_{EL} \cdot \frac{\theta_{CD} + \Delta_{CD} + 273.15}{\theta_{CD} - \theta_{VP} + \Delta_{CD} + \Delta_{VP}}$$

For known data, such as temperatures of refrigerant state, condensing/evaporating temperatures, we can simulate the behavior of system, we can extract whatever parameters are in interest but also to define some parameters which cannot be measured.

If we rewrite [Equation 6] into [Equation 7], were parameters c_{th} and c_{tl} are described by [Equation 8], noted RCTCs (refrigerant cycle temperatures coefficients), we are able to estimate electrical power, on compressor inlet. This is possible taking into consideration compressor type efficiency, known RCTs, but, more important condenser power.

Equation 7

$$P_{EL} = {\eta_c}^{-1} \cdot P_{CD} \cdot (c_{th} - c_{tl})$$

Equation 8

$$c_{tl} = \frac{\theta_{VP} - \Delta_{VP}}{\theta_{CD} + \Delta_{CD} + 273.15}$$

$$c_{th} = \frac{\theta_{CD} + \Delta_{CD}}{\theta_{CD} + \Delta_{CD} + 273.15}$$

This is a direct approach of electrical power based on some system characteristics such as compressor construction type and refrigerant properties. High and low temperature coefficients give through a mathematical expression, how much influence have temperature differences in electrical power consumption, correlated with condenser power. That's as minimum as possible is temperature differences between condensing and evaporator states, therefore power consumption decrease for same condenser power. Also, as much as possible compressor efficiency is bigger the electrical demand will decrease. Those heat pump properties are generally known and this paper approach comes with a specific form to provide an equation which stand to mathematical model for heat pump to be used in numerical simulation.

Evaporator power is generally written by difference from condenser power and a part of electrical power through compressor efficiency. This is revealed by [Equation 9] generally valid for a constant evaporator temperature.

Equation 9

$$P_{vp} = P_{CD} - \eta_c \cdot P_{EL}$$

Due to evaporator overheating behavior and pressure losses with refrigerant transportation into pipes refrigerant vapors are overheated and evaporator power described by [Equation 9] is decreased. Real temperature of liquid and vapor inside equipment is different than ideal condition. Compressor maintain the pressure inside evaporator equipment but cause of overheating the temperature is higher comparing with corresponding temperature at refrigerant pressure state. Rapport between real evaporator power and ideal condition evaporating power described by [Equation 9] is noted as evaporator state efficiency, and defined by [Equation 10].

Equation 10

$$\eta_{vp} = \frac{P_{vp\_real}}{P_{vp}}$$

By equalization, [Equation 9] and [Equation 10] become the expression of real evaporator power, with losses and overheating effect taken into consideration. [Equation 11] reveal the real behavior of evaporator equipment, when by designing the parameters states, the overheating and pressure losses are a real constrain which merge to a evaporator power lower than ideal equipment. Vapor boiling inside equipment inevitably will be overheated by equipment wall which are at a temperature above the liquid temperature.

Equation 11

$$P_{vp\_real} = \eta_{vp} \cdot (P_{CD} - \eta_c \cdot P_{EL})$$

2. Thermal system simulation program

System simulation, a powerful tool in design optimization, cost effective analyze and energetic demand estimation, is widely used today in research and energy exploitation industries. Opportunity given by advantages of computational environment to give result of simulation in almost high speed and instant calculation, make this tool being a mandatory need in laboratories and design offices.

Historically, for thermal systems, was developed a series of equipment and software tools with specific physical built-in models. Deep in the model are actually mathematical equation which helps in getting required results. Software like TRNSyS (24) (25), an academically used tool for thermal simulation, ANSYS (26) for CFD simulation but even mathematical based software like MATLAB Simulink (27) or DUPREX for refrigerant cycles and performance studies.

Generally, heat pumps are electrically operated by inverters or on/off systems. Working period of heat pump electric motor is given by the time until the pressures are established, in evaporator and condenser also, and also due to discharge time needed in condenser for refrigerant to condensate. If we work on intervals were this behavior work, and were system parameters are constant, we can easily indicate the power consumption and energy at input of system and the amount of energy produced at the end. For those periods, we can conclude about the interdependency of system parameters, correlated [Equation 6], and saying that for a specific temperature, how much electrical power is needed at heat pump input. To be able simulate such a system, tables, graphs and weather data should be available. Having in hand outside temperature, as evaporator environment state, condensing temperature and condenser power, with [Equation 7] we can easily detail the interdependency between RCTs and electrical power consumption. Noted above, one of the plurality software used in refrigerants study which easily computes refrigerant cycle parameter, compressor power estimation based on a specific isentropic efficiency, condenser and evaporator thermal power specific refrigerant enthalpy based, give us a representation of system COP and working conditions. Heat pump manufacturers normally select cycle parameters such sub-cooling and superheated environments for their applications mostly due to marketing constrains and overall system performance, in correlation with compressor and system type and refrigerant properties.

In Figure 4 a DUPREX based cycle graphic description show the cycle properties and state parameters for refrigerant R-410A used in an ASHP system (28), into a simple system configuration were evaporator and condenser are drive the heat from and respectively into the air environment. Normally used in ASHP systems, R-410A offer a variety of advantages but most important, which make it suitable for heat pump usage, is the capacity of deliver high COPs comparing classical refrigerants with ozone layer impact, like R-11 and R-12 or with higher GWP like R-404A or R-502 (29).

Typical cycle description consist of two main stages. First stage, were the heat is absorbed into the system is built from evaporator, for ASHP generally formed by a force convection ventilated array of coils and thin parallel foils for heat transfer and surface enlargement. In Figure 5 mostly used evaporator or even condenser in reversible heat pumps, is schematically drawn.

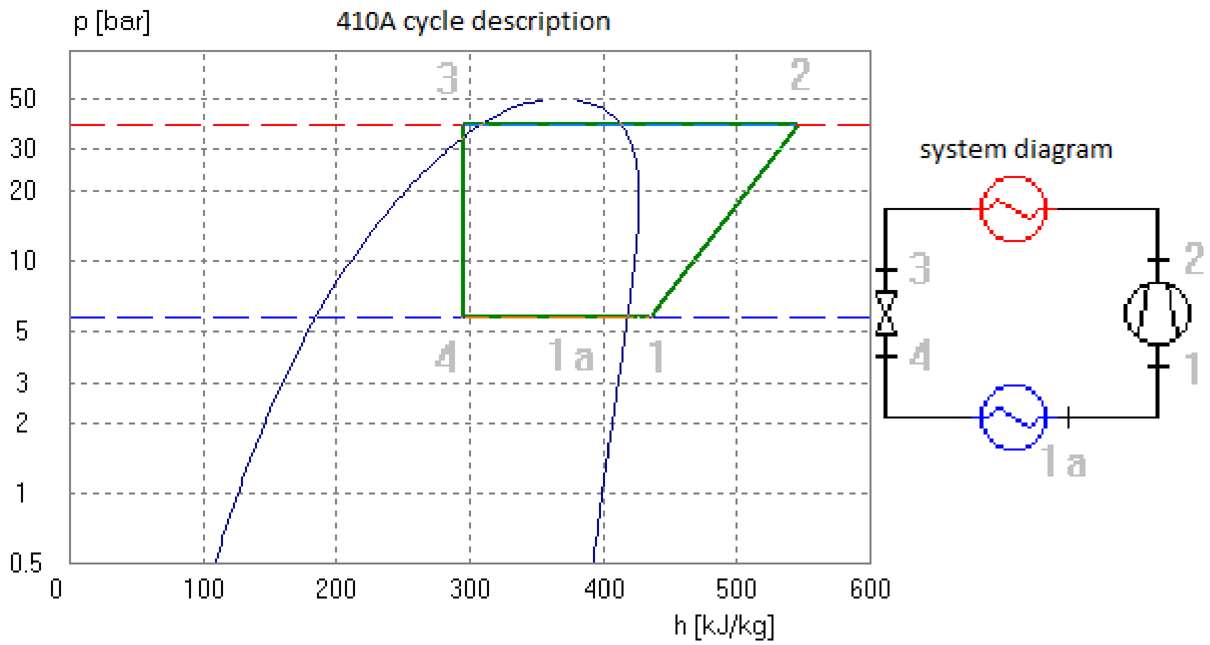

Figure 4 DUPREX Refrigerant cycle and system schematic

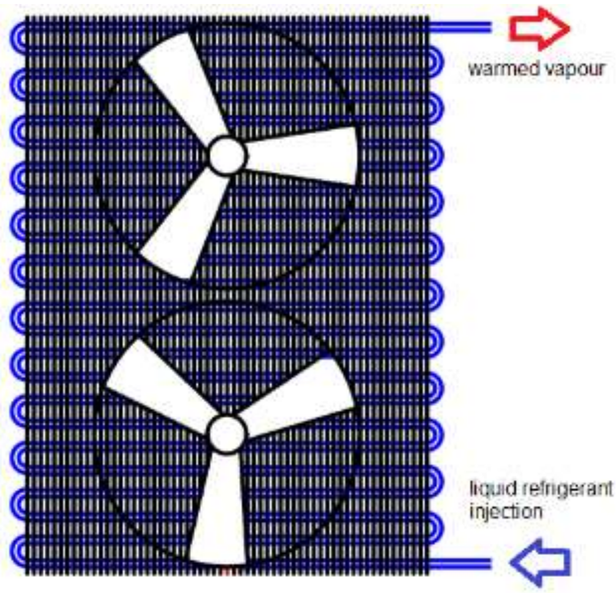

Figure 5 Air source evaporator

This type of equipment is capable to transfer the heat into the liquid injected by the lower drawing inlet, from the air set in motion, by force of ventilators. Injected air, due to ventilator movement, is cooled down and the heat extracted from it by boiling effect of refrigerant at the right pressure, at which the phase change from liquid to vapor is happening. Knowing the outside temperature by temperature sensors, heat pumps are able to drive the compressor until evaporator gas pressure is reached.

Regarding the second stage of refrigerant cycle is referring to condenser statement and pressure/temperature parameters. The condenser state, in this study is considered a two phase heat exchanger, as described into Figure 4, were a contraflow heat exchange is realized.

A DUPREX, Dupont Refrigerant Expert, implementation of the refrigerant state and system configuration is launched. This instrument largely used in refrigerant study and engineering field for equipment design or system functional simulation, consist in 3 main stages in preparation of a simulation.

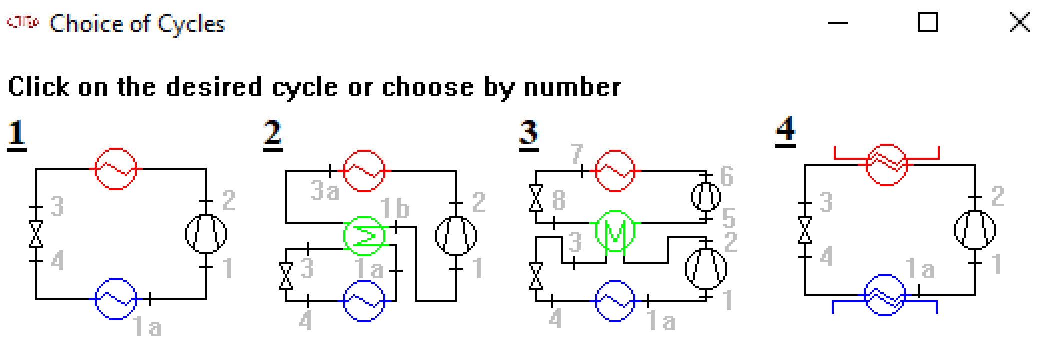

Figure 6 DUPREX cycles specification

First stage in software usage, is the starting window which allowed users to choose the cycle wanted for simulation. From the starting window, user can select one of the 4 available system types, with one or two compression level, with water or air source or w/w internal heat exchanger. In Figure 6 it is drawn all four possibilities with a simple way by clicking on needed stage, this is selected and the next window will be available to introduce the system parameters.

Regarding the system parameters, it is interesting that a lot of specific requirements for cycle definition can be set. Starting from stage temperatures, evaporator and condensing specification are possible to be selected in dew point condition, and considering average value in a stationary point of view.

The refrigerant capacity and compressor displacement are dependent one to each other and cannot both to be defined. If capacity is defined, it will remain constant during cycle calculation or if displacement will be defined, the capacity will vary in calculation step.

In Figure 7 can be seen, in the upper right sign, a list tab were refrigerant type is selected. Also, temperatures for adjusting system details, like sub cooling, super-heated, in evaporator or suction line can be set. Pressure loss in evaporator and suction line together with isentropic and volumetric efficiencies can be introduced inside system definition. Refrigeration capacity is introduced in btu/h but this is not an inconvenient, because results of the simulation are available also in table format and easily can be updated.

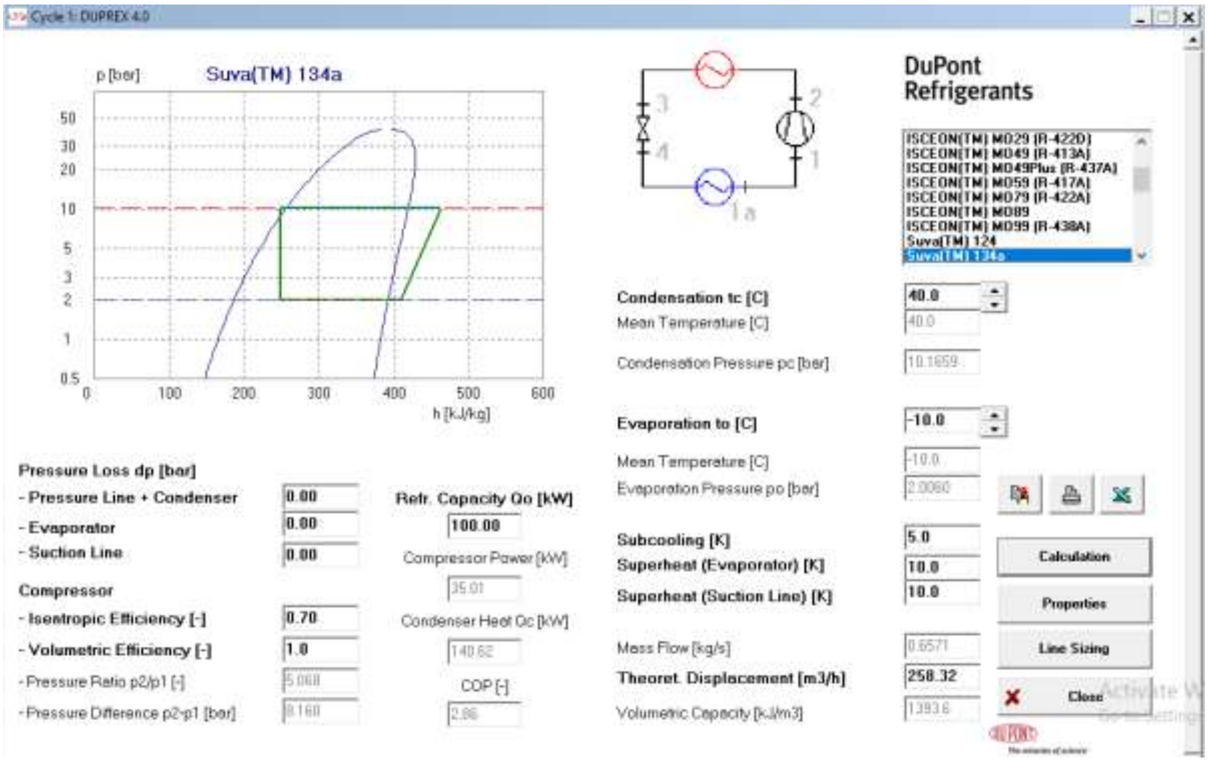

Figure 7 DUPREX Cycle parameters definition window

From an experimental point of view this useful tool is a desired alternative for understanding the refrigerant systems behavior and also to estimate the parameters which are not known or cannot be measured at the right time. By calculation button push in, the entire refrigerant thermodynamic cycle, according input data, is operated and calculation performed, after which the results of simulation are available in the memory and can be released by pressing the button properties. Temperatures, pressures, enthalpy, entropy, etc. are shown on a specific window or are available in an editable table through excel, copied in buffer memory by click on “X” tab of the introduction window. Same as tabular format, a picture format file is available for drop it in a specific document or folder together with the print facility given by pushing the button print.

For specific systems, were lines size are known, DUPREX easily can adjust pressure/temperature drop inside coils and pipes, by using the pressure/temperature drop coefficient of different materials standardized by values memories inside program, or by values introduced inside the specific line sizing tab, available in the software and described in Figure 1Figure 8.

The standard values for temperature drop coefficients are 0.04K/m in suction line and 0.02K/m for the liquid line. This value is also adjustable inside line sizing tab. Suction, liquid and discharge lines properties and physical dimensions are available inside same tab, their values being editable in a table or dropped into a document or folder in a picture format.

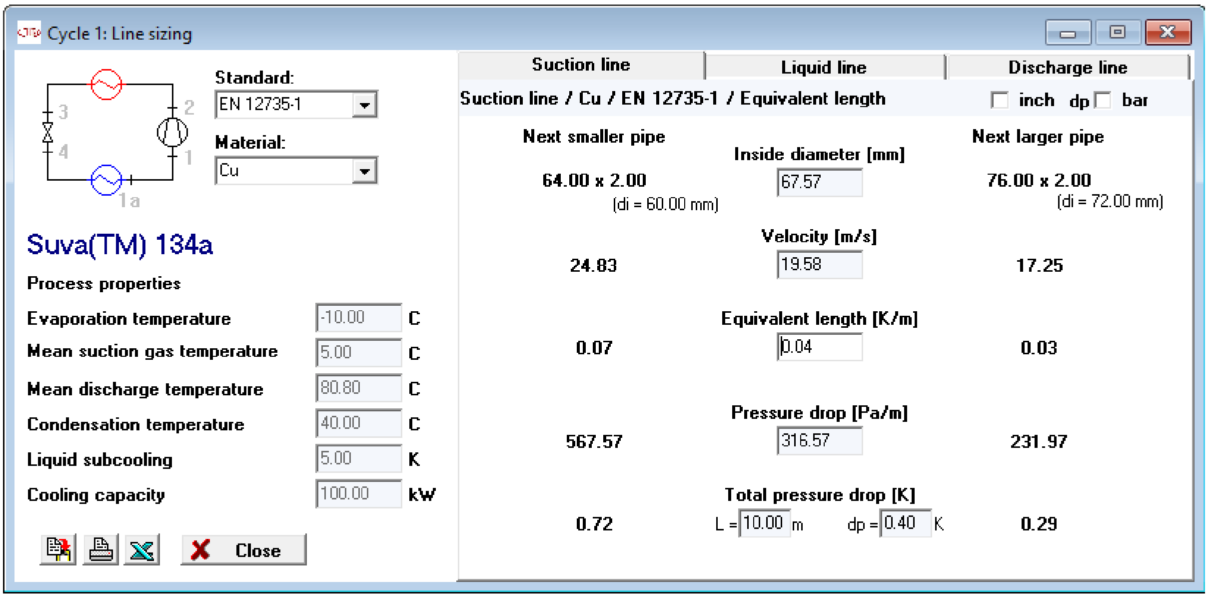

Figure 8 DUPREX Line sizing properties and calibration window

Single state, tables and units are 3 main editable properties of the program. Selection of single state option button will open a window that allows the calculation of pressure, volume, enthalpy, entropy, thermal conductivity, viscosity, kinematic viscosity and thermal diffusivity for the liquid and vapor phases in the saturated, superheated or subcooled regions for the refrigerant chosen from list. Table option allows the calculation of thermodynamic or transport properties tables for refrigerant selected from the list in the upper left hand side of the window.

For each selected refrigerant and selected temperature or pressure as the defined parameter, can be seen the others cycle characteristics as enthalpy, entropy or specific volume in kind of state saturated, subcooled or superheated. All those can be set or accessed inside single state window, as shown in Figure 9.

For deeper investigation related heat pumps cycle definition, thermal conductivity, viscosity, thermal diffusivity and kinematic viscosity are also available in the same window and correlated for each kind of state. This approach is available only in readable format of the program data base, properties aren’t modifiable. For precisely measurements on an experimental study, those available data are a gold mine for researchers in physical model calibration and mathematical approach, for engineers in system design and for technicians which can verify the equipment set up or make software calibration for heat pump controllers.

An important feature is available for refrigerant state parameters. In case complex developed software tools for equipment functional controllers, are needed big databases with refrigerant properties for passing complex calculation algorithm for cycle parameters data, the speed of controller increase once with latency diminution of data results calculation. Having all state parameters as a database, the only operation, in software point of view, will be picking the corresponding value of needed parameter and reduce the time need for an eventual algorithm calculation.

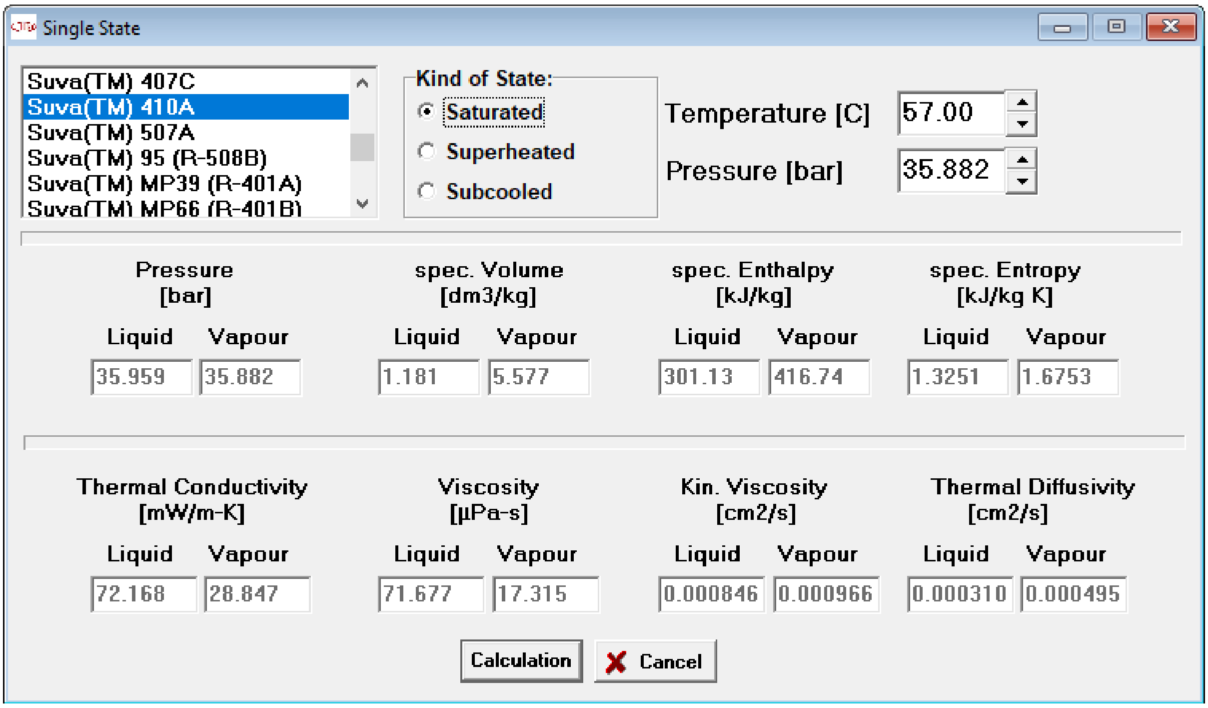

Figure 9 DUPREX Single State characteristics window

In Figure 10 an example of table with cycle parameters between -3ºC to 70ºC is presented. For each kind of state and for each incremented value from the selected interval, state variable or transport properties are released by program and available in excel or picture format to be dropped into a file or folder. A detailed table is present for each state, bubble (boiling) and dew (condensing) properties are available, both coming in help for designers of such systems. Bubble point is the state of refrigerant when boiling of liquid is considered, are useful information in case evaporators are designed, physical dimensions and boiling temperatures/characteristics being necessary. For condensers, the philosophy is almost on an inverse phase change process, when the vapor will condensate and loose energy to the environment through condenser metallic walls.

Units allows the calculation to be done in user convenient measurements units from SI to English format or vice-versa like bar and ºC and respectively psi and ºF. Changing measurements units will make unavailable data in all the others field, a new calculations being a must.

Also, some additional properties are available like user preferences for languages, printer properties or some additional information.

In DUPREX the refrigerant characteristics are evaluated according REFPROP, an acronym for REFeference fluid PROPerties and developed by NIST. All refrigerants are categorized by ODP and GWP levels. For GWP the values define the equivalent mass level of CO_2 in atmosphere for one kg of refrigerant, based on a century. An example for R410A, one kg of this substance released into atmosphere, will have an impact on global warming equivalent to 1924 kg of CO_2 in 100 years based on Du Pont refrigerant tables and without any impact in ozone atmospheric layer with ODP equal to zero.

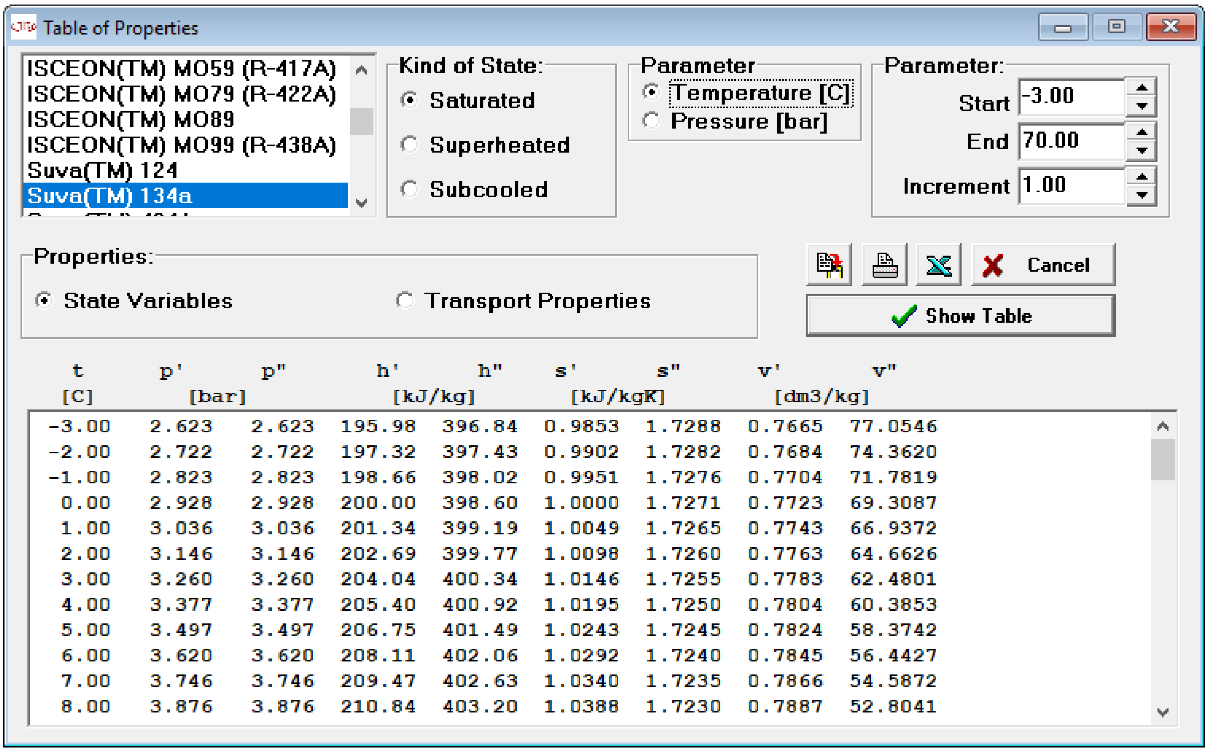

Figure 10 DUPREX Table of Properties window

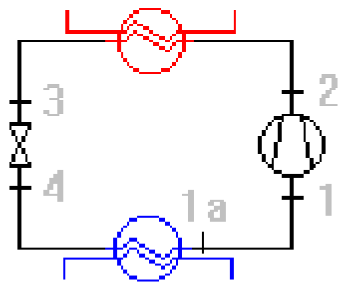

Figure 11 DUPREX cycle 1 WSHP Heat Pump

To see how DUPREX work, a simulation program was started. Imagined heat pump WSHP with schematic diagram given by Figure 11 and system description given by Table 1. Schematic diagram is given by DUPREX environment, which make possible to see the hydraulic network of the system considered. The water source heat pump from this software is imagined being constructed from 2 heat exchangers, one at evaporator and one at the condenser.

Water source heat pump considered have few specificities. Firstly, the circulated water through evaporator is a special mix which contain antifreeze agent, able to drive liquid in the evaporator below -20ºC. In Figure 11 different points are considered. 1 is entry point for compressor and 2 is the state point of compressor output. 3 and 4 represent liquid line, respectively evaporator states, once with vapor properties in point 1a.

Table 1 DUPREX cycle 1 parameters

Refrigerant:	R 410A	
Pressure Line + Condenser	0	[bar]
Evaporator	0	[bar]
Suction Line	0	[bar]
Condenser Temperature Difference	5	[K]
Evaporator		
Evaporator Temperature Difference	5	[K]
Subcooling	5	[K]
Superheat (Evaporator)	5	[K]
Superheat (Suction Line)	10	[K]
Compressor		
Volumetric Efficiency	1	[-]
Isentropic Efficiency	0.61	[-]
Condenser Heat Qc	10	[kW]

The simulation in DUPREX environment is a stationary state, calculation being made considering a constant parameters behavior inside system. Also the refrigerant driven by compressor has standard lubrication oil utilized in R410A systems and taken into account by DUPREX.

Losses are not considered for this simulation; Condenser and pressure line, evaporator and suction lines, are realistic imagined, but without pressure drop during transport into pipes of equipment. Same water differences are considered inside evaporator and condenser between their inputs and outputs connectors. Sub cooling temperature and superheated vapor in evaporator are considered equal for each-other cancellation. A 10kW condenser power heat pump is imagined with a high performance compressor with 0.61 efficiency level. No volumetric losses are considered for this simulation for a maximum yield of the system.

Running simulation reveal all necessary data of energetic, efficiencies, refrigerant state cycles and point enthalpies useful for future calculations and for make all conclusions about system working cycle. Full data obtained for 3 different condensing temperatures and equipment losses are present in annexes.

For two of condensing temperature and no losses in equipment, below are figured, noted and commented data obtained by DUPREX simulation.

Pressure ratio, report between suction and condensing line pressures, is not a linear curve, approximated according polynomial regression displayed on Figure 12, which show that pressure ratio is a function of the operating conditions, between condenser and evaporator environment.

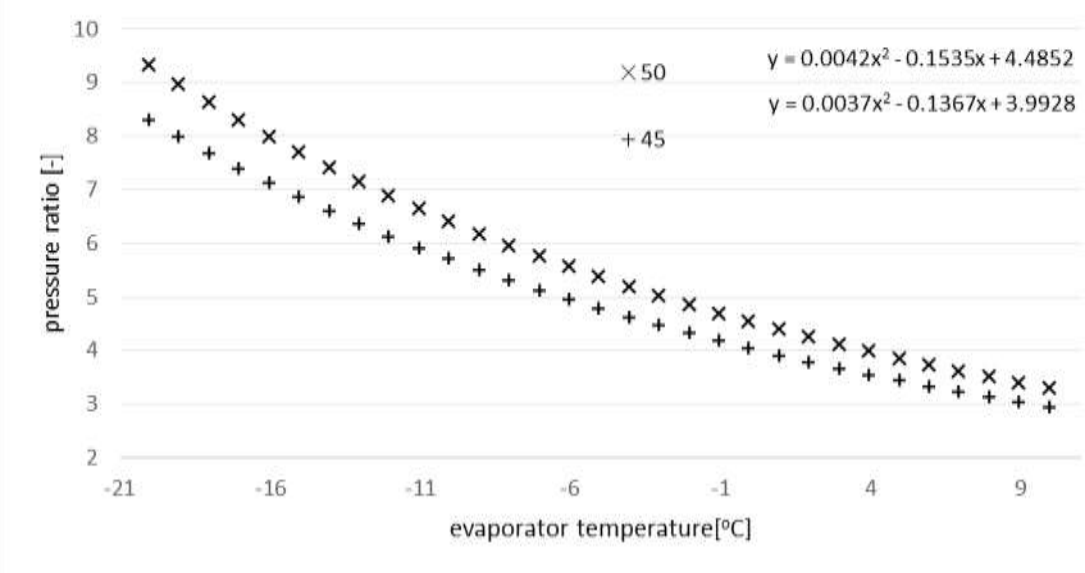

Figure 12 Pressure ratio between suction and condenser lines at 50 and 45 condensing temperature

Pressure ratio in refrigerant compressors are intently to be used for minimize electrical power consumption. It was studied that for a given pressure ratio exist a resonance frequency which, if driven by electronic controller will decrease the total electrical consumption of compressor (30) (31). Knowing pressure ratios for specific evaporator and condenser state, together with resonant frequencies for a given compressor, will provide to final consumer a total cost gain if the frequency inverter have implemented an algorithm which count it. This kind of approach should be very well studied because resonant frequency depends much on physical construction of compressor and less on pressure ratio but always will be coherence between them.

Polynomial regression return the equations to be used in software to determine specific pressure ratios for given evaporator and condenser temperatures. In Figure 12 are presented grade 2 polynomial function of pressure differences for 45 and 50ºC in simulated conditions through DUPREX input specification. For lower temperature differences, respectively lower pressure ratios, the curvature of the function decrease once with it second coefficient.

Theoretical displacement, rated m^3/h, inversely proportional to mass flow rate of compressor, in kg/s are two coefficients used by the simulation software to calculate other parameters like enthalpy or electrical power. In Figure 13 theoretical displacement and mass flow rate obtained for R410A refrigerant show an interesting behavior regarding compressor volumetric efficiency. From 50˚C to 45˚C or 10% drop in condenser temperature and negative -20˚C evaporating temperature, exist a

1.44% decrease theoretical displacement. Together with mass flow rate increased for higher evaporating temperature, this behavior reveal that for a small volume of vapor, is driven by compressor a much more gas mass, transferred between evaporator and condensing line.

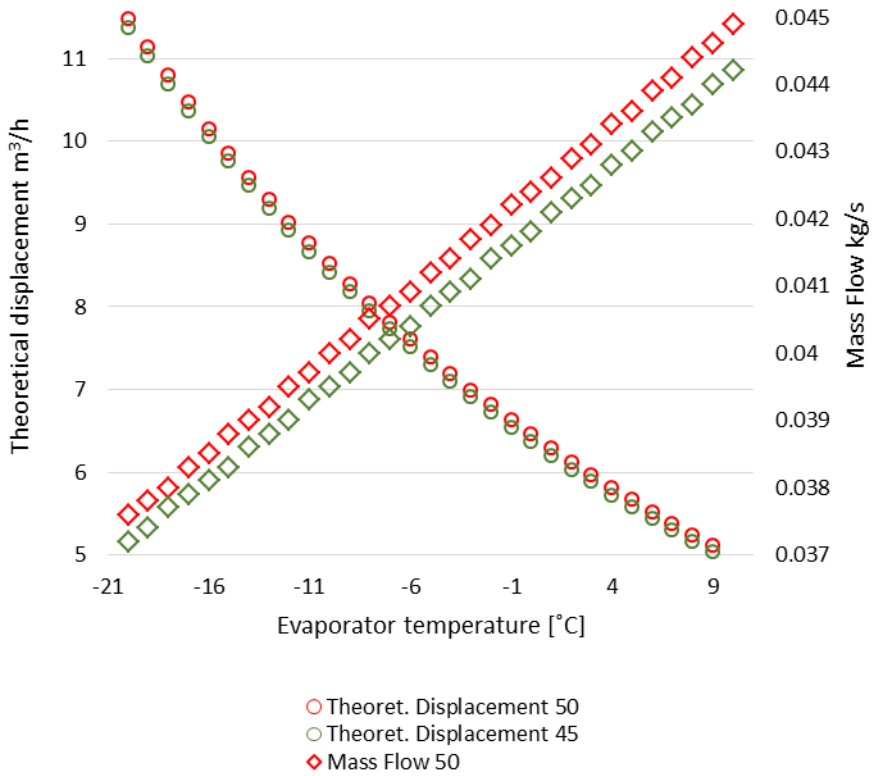

Figure 13 Theoretical displacement and mass flow obtained by DUPREX for R410A and respectively 45 ˚C and 50 ˚C condensing temperature.

Driving a lot more refrigerant into condenser comparing low and high evaporator temperatures, will reduce significantly the electrical power consumption. Once is reduced the volume by 58% as theoretical displacement between -20˚C to 10˚C and wile is increased the flow mass with 15%, for the same temperature difference, is obtained into condenser a lot more refrigerant mass which increase condensing power.

Compressor electrical power in refrigerant compression equipment decrease when temperature differences between evaporator and condenser drop. For R410A electrical power consumption in Figure 14 are presented the power consumption of electrical compressor for 3 condensing temperatures and evaporating interval of -20˚C to +10˚C.

As can be seen, in this graph the compressor electrical power consumption for a 10kW condensing power, have his minimum 2,34kW with corresponding COP 3.78 in heating conditions and a refrigeration COP of 2.6. Those values are calculated for no pressure losses in condensing line, evaporator and suction line, and 0.61 isentropic efficiency of compressor. A closer look on the graph reveal a drop with 42.36% in electrical consumption, from -20˚C to 10˚C evaporating temperature for a constant condensing temperature of 45˚C. A lower reduction can be seen in case of higher condensing temperature with a 39.31% electrical power reduction at 50˚C. Comparing both percentages, can be noted a gain of 3.04% in electrical power differences from -20˚C to 10˚C when the condensing temperature is decreased with 5˚C.

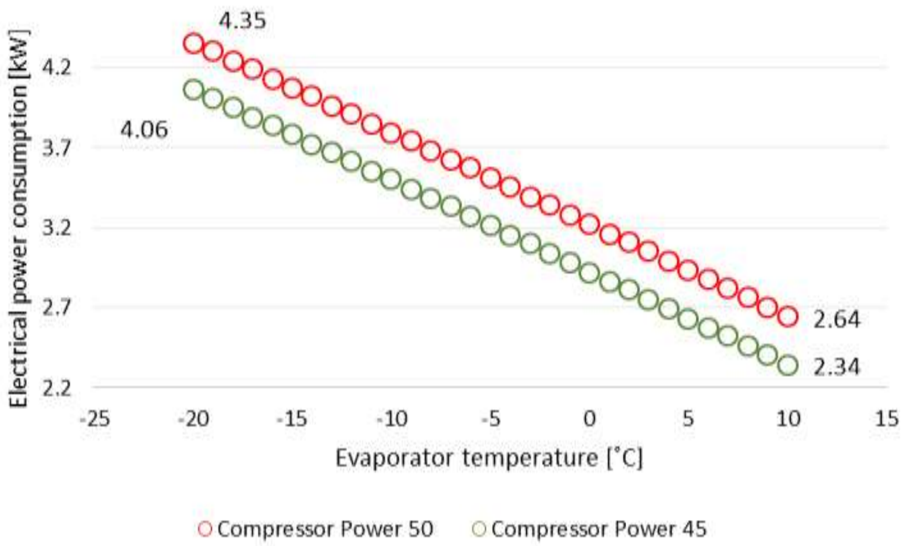

Figure 14 Compressor power consumption for 50˚C and 45˚C condensing temperature for refrigerant R410A

COP for evaporator and condenser are also outputs of DUPREX software environment. For given data, the algorithm is able to reveal the stationary behavior of equipment when constant parameters are involved. Comparing power consumption, at -20˚C evaporating temperature, for 50˚C versus 45˚C condensing temperature level is reached an increase of electrical demand with 7.25%, instead of 12.8% for same condensing conditions but when evaporator temperature s -10˚C. This reveals a known behavior that for lower condensing and evaporating temperature differences, the efficiency of the equipment drop. For heating coefficient of performance, rated in Figure 15 the values obtained are situated between 2.3 for -20˚C evaporating temperature respectively 50˚C condensing temperature and a maximum of 4.27 heating COP at 10˚C evaporating temperature and 45˚C condensing temperature level. For positive evaporator temperature the minimum COP obtained reach values between 3.1 to 3.78 which means at triple condensing power is obtained above compressor electrical consumption. Should be noted again that those values are reached for conditions regarding equipment construction without specific pressure losses into system respectively high efficiency compressor.

Heating purpose using compression machines encounter higher efficiency equipment comparing the ones for cooling needs. Due to condensing power higher than refrigerant power, the COP for cooling obtained are shown in Figure 16 were a minimum of 1.22 and maximum 3.07 are calculated in same equipment conditions listed for cooling needs. Here can be noted that the evaporating power is lower comparing heating demand with maximum value around triple of electrical power consumption.

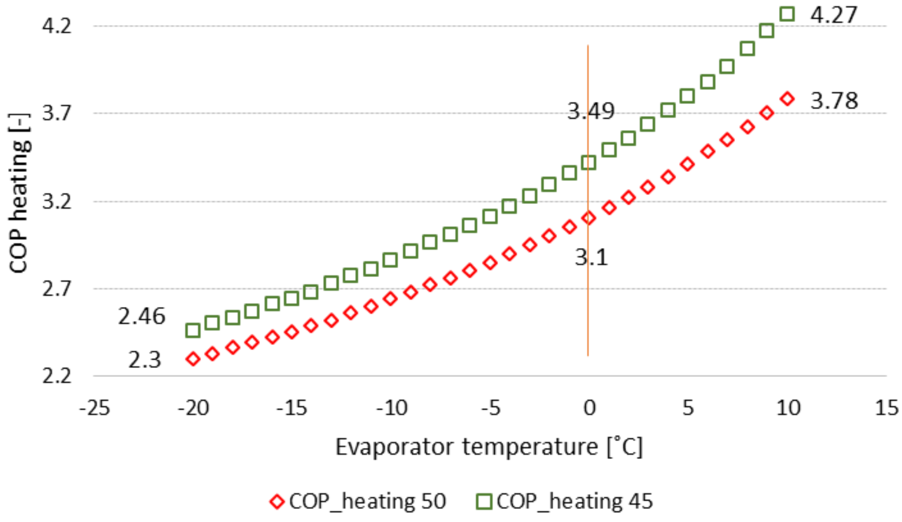

Figure 15 R410A cooling COP for 45˚C respectively 50˚C condensing temperature for each unitary evaporating temperature between -20˚C to 10˚C

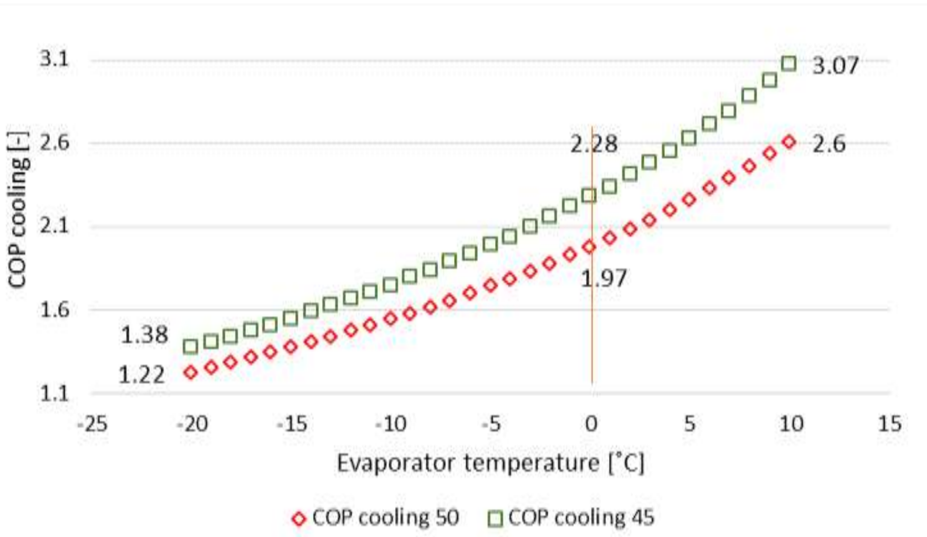

Figure 16 R410A heating COP for 45˚C respectively 50˚C condensing temperature for each unitary evaporating temperature between -20˚C to 10˚C

In positive evaporator temperatures obtained COP for cooling show that for positive evaporating temperature the thermal power at cooling is more than twice electrical demand.

Evaporator and condensing powers are graphically shown into Figure 17 with a minimum of 5.53, half of condenser power in 50°C condensing, respectively -20°C evaporator temperatures. The maximum refrigerator power is 37.43% increase over the minimum value, for 40ºC condensing respectively 10ºC evaporator temperatures. That's why a much less electrical power is added to obtain the same needed condenser power for heating demand.

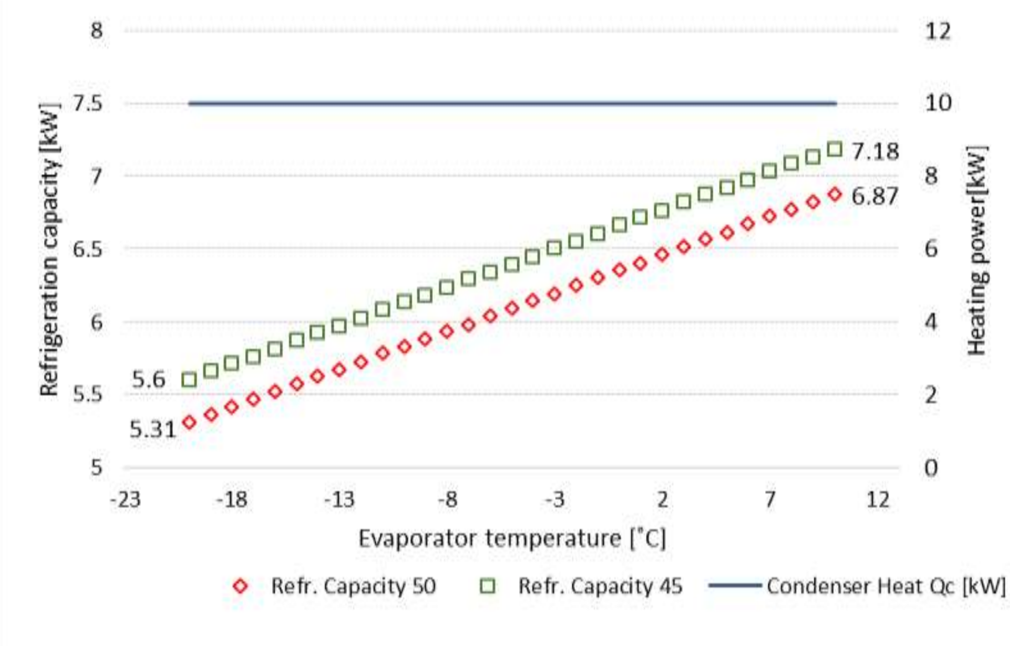

Figure 17 R410A refrigeration capacity for 50˚C respectively 45˚C condensing temperatures

The refrigerant cycles for the 3 condensing temperatures imagined and for lowest to highest evaporator refrigerant temperatures are shown in Figure 18 were 6 cycle drawings are present for corresponding cycle parameters. Red lines denote the condensing states together with blue lines for evaporation states. Red lines moving from 25bar to 30bar indicate the pressures to which refrigerant will condensate, with higher pressure for higher condensing temperature. Working pressures are relatively high but are correlated with evaporator pressures, pressure ratio not being bigger than 10 as sown in Figure 12 of this chapter. Blue lines have a lot more wide pressure interval were is moving but the chosen evaporator temperatures have a bigger interval compering condensing imagined states. Evaporation pressures are found in the interval of minimum 3.2941 bar for -20ºC and higher pressure 9.3315 bar corresponding 10ºC evaporating temperature. Over the entire evaporator temperatures interval, for a fixed condensing temperature the only pressure change is in evaporator, condensing pressure remaining constant. This takes us to think that in a real situation, when internal comfort temperature is constant for different sourced water temperature. The heat pump should only change the pressure inside evaporator fact that is important in construction of such equipment. Because the internal temperature can be set by user, is not needed inside system to have a temperature sensor and a specific algorithm built in equipment supervisory controller for condenser ambient. The only permanent change conditions remain in the evaporator, were temperature can rapidly even significantly change making it necessary to have complex equipment at evaporator comparing condenser.

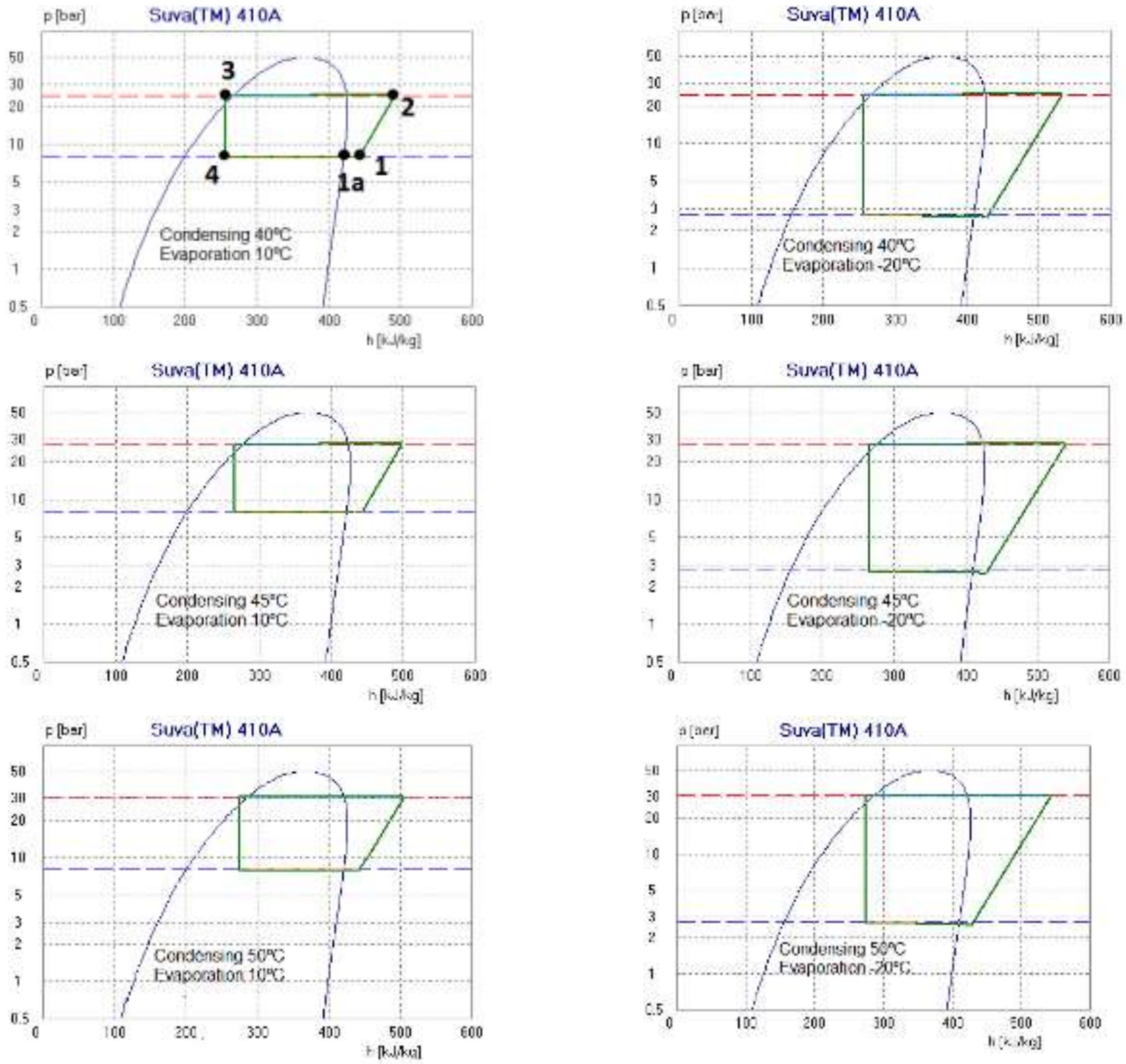

Figure 18 R410 refrigerant cycles for 50ºC, 45ºC respectively 40ºC condensing temperature and for minimum and maximum evaporator temperature

In case heat pump is also used in hot season for air conditioning or to prepare cold water at a given state, this mean heat pump is reversible and have the same complexity implemented at evaporator and condenser together. The cycles presented in Figure 18 are corresponding to a WSHP schematically draw in Figure 11 picture front left upper side, with point 1a corresponding evaporator pressure state, point 1 is the entry point in compressor, state number 2 is the vapor characteristics at the outlet of compressor, the state 3 is the liquid parameters founded in condenser liquid line, point 4 corresponding to final state of refrigerant, after the lamination of refrigerant.

For a similitude analyze of method described in chapter Mathematical modeling of heat pump thermal systems and DUPREX results, a MATLAB program was started, were algorithm from list below was applied.

```
% refrigerant cycle parameters-----------------------------------
teta_cd = 50;% [Celsius]
teta_vp = xlsread('matlab_simulation.xlsx','Date_50','B1:AF1');
delta_cd = 5;% [Celsius]
delta_vp = 5;% [Celsius]
%-----------------------------------------------------------------

%sistem parameters------------------------------------------------
eta_c = 0.61;% compressor efficiency [-]
P_CD = 10;% condenser power [kW]
eta_vp = 0.785;
%-----------------------------------------------------------------

%RCTC-------------------------------------------------------------
ctl = (teta_vp-delta_vp)/(teta_cd+delta_cd+273.15);% [-]
    xlswrite('matlab_simulation.xlsx',ctl,'Date_50','B2:AF2')  ;
cth = (teta_cd+delta_cd)/(teta_cd+delta_cd+273.15);% [-]
    xlswrite('matlab_simulation.xlsx',cth,'Date_50','B3:AF3')  ;
%-----------------------------------------------------------------

%results----------------------------------------------------------

%compressor power-------------------------------------------------
P_EL = eta_c^(-1)*P_CD*(cth-ctl);% [kW]
    xlswrite('matlab_simulation.xlsx',P_EL,'Date_50','B4:AF4')  ;

%heating COP------------------------------------------------------
COP_CD = P_CD./P_EL;% [-]
    xlswrite('matlab_simulation.xlsx',COP_CD,'Date_50','B6:AF6')  ;
%-----------------------------------------------------------------

%cooling capacity-------------------------------------------------
P_VP = eta_vp*(P_CD-(eta_c*P_EL));
    xlswrite('matlab_simulation.xlsx',P_VP,'Date_50','B8:AF8');
%end--------------------------------------------------------------
```

This program was wrote around an excel file were evaporator temperatures and results were saved. For a fixed condensing temperature, each evaporator temperature was taken into account for calculation. Condenser power and compressor efficiency are intended to be constant during evaluation with values of 10kW for condenser respectively 0.61 for compressor. In a small loop, the RCTCs parameters are calculated and data stored inside the excel file as table for each step of the redundant equation. MATLAB environment will always work with matrices and if the vector have more than one element than a matrices or element by element mathematical operation is applied. In the final loop of the program, the intended results for analyze are calculated one by one and data stored in the same way as RTCTs loop. All results are available in table format at the end of report in Annex 4. Data are presented as DUPREX versus MATLAB, with 2 subunit digits and parameters measurement units are listed.

The curves, comparing both results DUPREX respectively mathematical model from MATLAB are plotted in Figure 19 and Figure 20. COP from both simulation have similar results with a small gap between values with an error below 5.00% of elevated COP differences.

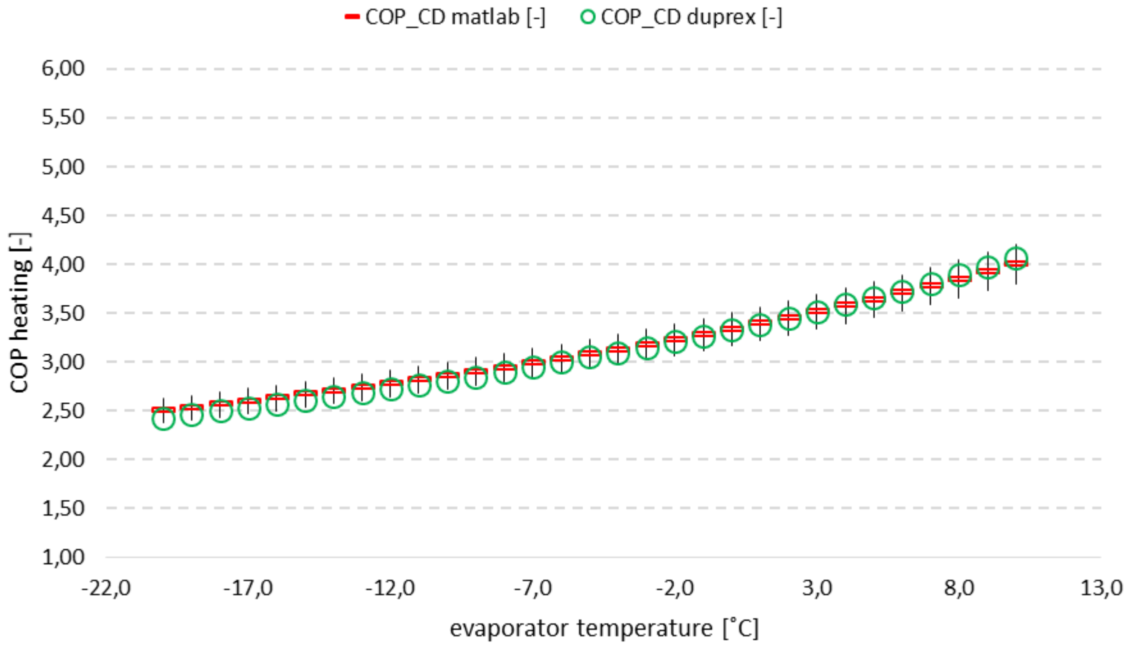

Figure 19 DUPREX and MATLAB simulation comparison for heating COP

For heating purpose, the minimum COP obtained reveal a value of 2.4 times more than electrical power consumed by compressor to drive the refrigerant through system to obtain a 10kW thermal power at condenser outlet. For maximum value, more than four times electrical power is obtained at condenser.

The compressor electrical power consumption 10˚C evaporator temperature is reduced with 60% comparing the value at -20˚C for MATLAB obtained data, with a minimum value 2.49kW respectively maximum 4.99kW. Data DUPREX and MATLAB are more appropriate inside interval, between values of -5˚C to 5˚C evaporating temperature, outside those limits, the error increase. This can reveal the behavior of equipment related compressor efficiency which depends on temperature difference or pressure ratio between condenser and evaporator states. The small vertical lines presented in Figure 19 represent error interval between the two evaluations with a value equal 5%.

Compressor power in Figure 20 reveal the behavior of equipment with value of 10kW condenser power, no losses in equipment and 0.61 compressor efficiency with values inside interval limits of +/-5.00% error between both MATLAB and DUPREX simulated data.

Evaporator real thermal power is drawn on Figure 21. Differences between MATLAB and DUPREX evaluation are close to +/-5.00% error limits. Both data values are plotted inside graph with centering of curvature in the middle of evaporator temperature. The simulation was lunched and optimization of evaporator efficiency was taken for a minimum data difference value at -5˚C evaporator temperature. The value of evaporator efficiency was found being 0.785 for an optimized match of both MATLAB and DUPREX data simulation.

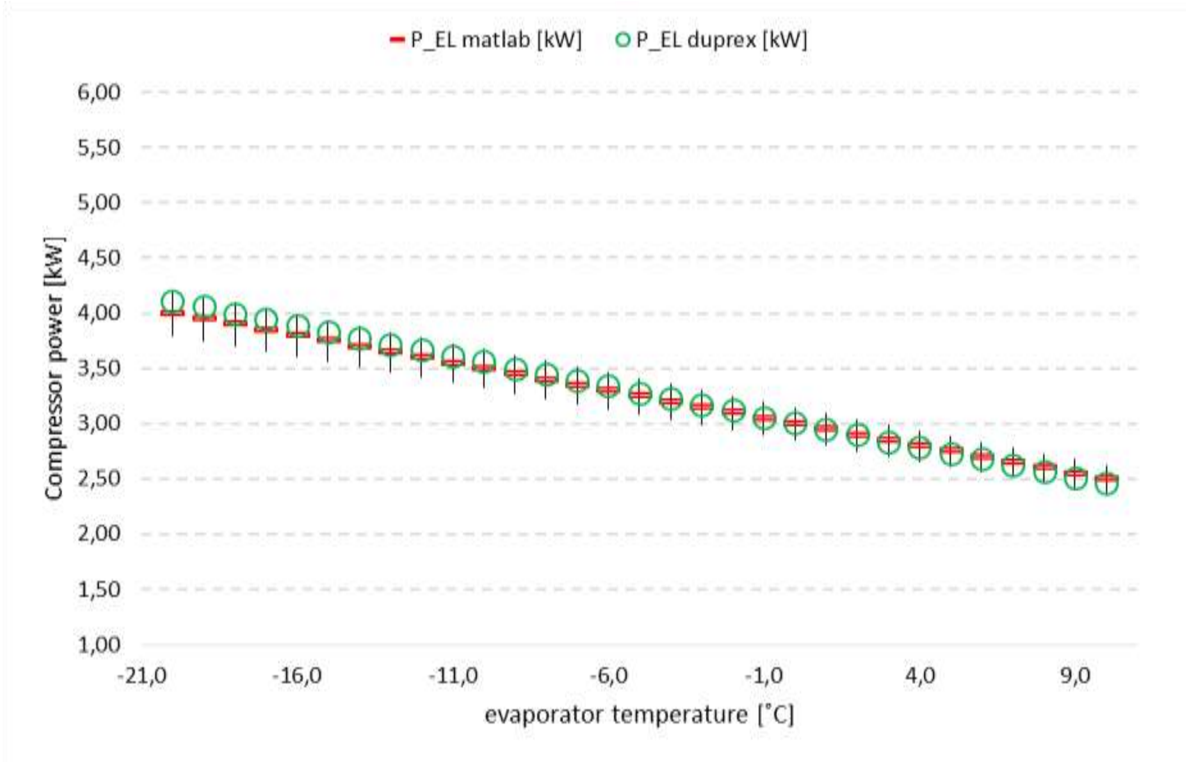

Figure 20 R410A Compressor power consumption MATLAB and DUPREX comparison

DUPREX simulation reveal a biggest evaporator power elevation for the entire evaporator temperature interval comparing MATLAB evaluation with a slower ramp of values in same conditions. Figure 19, Figure 20 and Figure 21 are obtained coefficient of performance, compressor power and respectively evaporator capacity for a condensing temperature value equal to 50˚C. All others parameters are correlated being 5˚C for both evaporator and condenser delta temperature differences and no losses in equipment functioning.

This simulation were lunched also for 45˚C condensing temperature with available data in Annex 4.

Next to this step is chose MATLAB software environment for all future analyzes. The error are taken into account for final data simulation analyze; definition of state parameters are same as DUPREX and MATLAB initial calibration of equipment. MATLAB software used for evaluation is R2016a version.

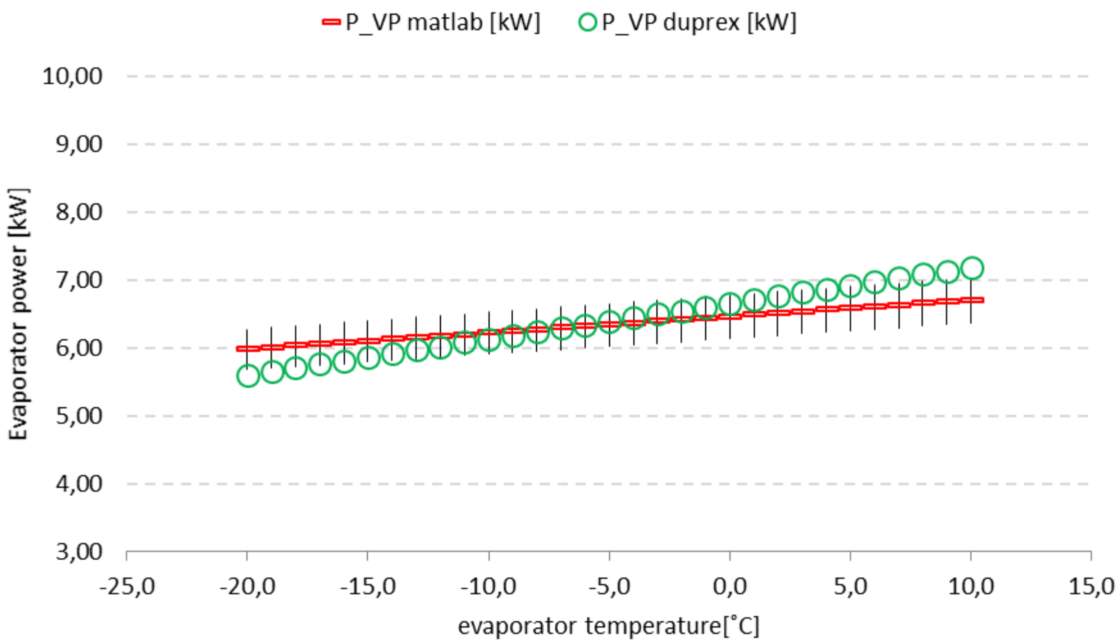

Figure 21 R410A evaporator power simulation DUPREX and MATLAB data comparison

A 3D plot evaluation were lunched inside MATLAB. Comparing DUPREX software environment, MATLAB use matrices mathematical operation which reduce significantly the time to simulate. Surf or 3Dplot functions are commonly used to generate surfaces which visually present the correlation of output for two parameters evolution. Below algorithm represent the data used as cycle states and the interval used for evaporating with vector X respectively vector Y for condensing temperature to plot COP of heat pump:

```
%cycle states------------------------------------
delta_cd = 5;
delta_vp = 5;
eta_compressor = 0.61;
P_CD = 10;
%------------------------------------------------

%3D plot-----------------------------------------
[X, Y] = meshgrid(-20:1:10,35:1:50);
ctl = (X-delta_vp)./(Y+delta_cd+273.15);
cth = (Y+delta_cd)./(Y+delta_cd+273.15);
P_EL = eta_compressor^(-1)*P_CD*(cth-ctl);
Z = P_CD./P_EL;
surf(X,Y,Z)
%------------------------------------------------
```

For entire interval, -20˚C to 10 ˚C evaporator respectively 35 ˚C to 50 ˚C condenser temperatures, was obtained the 3D plots in Figure 23, with electrical power used by compressor evaluated and plot with below algorithm and presented in Figure 22. Data used for plotting is same as for COP plotting were constant 10kW condenser power respectively 0.61 compressor isentropic efficiency. Both algorithm are MATLAB based matrices operation, point operand making possible the element by element matrices calculation. Plotting is performed using surf function which need a mesh grid opening entry data. The mesh grid is easily written as an element vector, with X and Y as evaporating temperature respectively condensing temperature were increment of element interval is unitary.

```
%cycle states------------------------------------
delta_cd = 5;
delta_vp = 5;
eta_compressor = 0.61;
eta_vp = 0.785;
P_CD = 10;
%-----------------------------------------------

%3D plot----------------------------------------
[X, Y] = meshgrid(-20:1:10,35:1:50);
ctl = (X-delta_vp)./(Y+delta_cd+273.15);
cth = (Y+delta_cd)./(Y+delta_cd+273.15);
W = eta_compressor^(-1)*P_CD*(cth-ctl);
W = P_EL ;
V = P_CD./P_EL ;
Z = eta_vp*(P_CD - (eta_compressor*P_EL))
surf(X,Y,Z)
%-----------------------------------------------
```

Differences between both algorithms are introduction of evaporating power and electrical compressor demand in same script, plotting the required data as user want, by select corresponding vector.

For compressor power Figure 22 were obtained. Axes represents evaporating temperatures in interval -20˚C to 10˚C, with 1˚C steps, respectively condensing temperature in interval 35 ˚C – 50 ˚C with 1 ˚C step. Colors selected represent user opinion about compressor electrical demand comparing condensing temperature. For this evaluation, the power over 3kW electrical demand, were considered being over a desired limit. Above this value, COP decrease below 3 which is the imposed limit for analyze and comments. Keeping this limit for a specific user, can be considered the indicator of how to select a right heat pump for heating purpose. This limit can be change however final heat pump user or design engineer consider as a compromise between cost of equipment, return benefits comparing classical heating, and exploitation conditions like water or air temperature at installation place. For this study, to be under the red color, a minimum -5 ˚C for evaporator temperature is the low limit of heat pump condition for a maximum condensing temperature equal to 50 ˚C.

If the limit imposed for condensing temperature not passing 40 ˚C can be concluded that for all evaporator temperature interval, the heat pump is obviously a right choice. With 40˚C condition at condenser and -20˚C at evaporator, the limit is minimal overpassed. This drive to conclusion that the heating equipment inside building is designed to work with maximum entry liquid temperature equal to 40 ˚C, the heat pump is able to deliver heating agent for all evaporating temperatures, with electrical consumption below 3.6kW for a condensing power equal to 10kW.

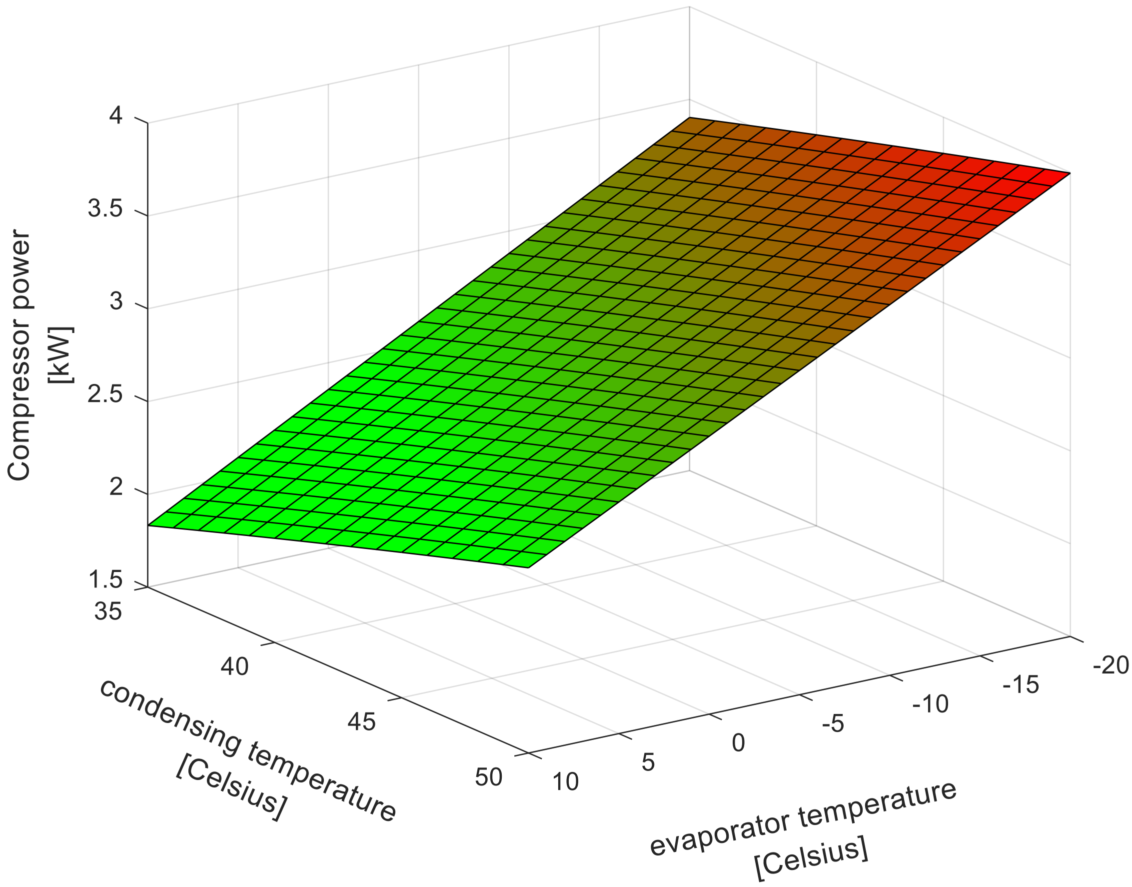

Figure 22 MATLAB based electrical power demand of system evaluation

COP graph reveal an exponential behavior when condenser temperature and evaporating temperatures are swept within same interval limits. In this case, the limits of color distribution is selected COP equal to 4, indicator which shows on graph that for 35˚C condensing temperature the evaporator temperature should overpassed -5˚C. In this situation, for a designed heating unit that use 50˚C entry level of temperature is not recommended to be used heat pump with evaporating temperatures below 5˚C, COP of system decreasing under value of 3.8. Graph obtained are referring to environment were evaporator and condenser are immersed. Refrigerant state reaches other values, higher in case of condenser and smaller in case of evaporator.

Condenser and evaporator refrigerant temperatures are depending of equipment setup. In this situation, the limits of temperatures differences are considered being 5˚C, but this selection can be different in case of a specific equipment and refrigerant. Designers or end users should properly document their necessities, study very carefully the equipment datasheet and setup equations for desired feasibility of heating equipment. In case equipment should be evaluated for performance, a measurement can also be a right choice to register temperature differences or to evaluate experimentally evaporator or compressor efficiencies.

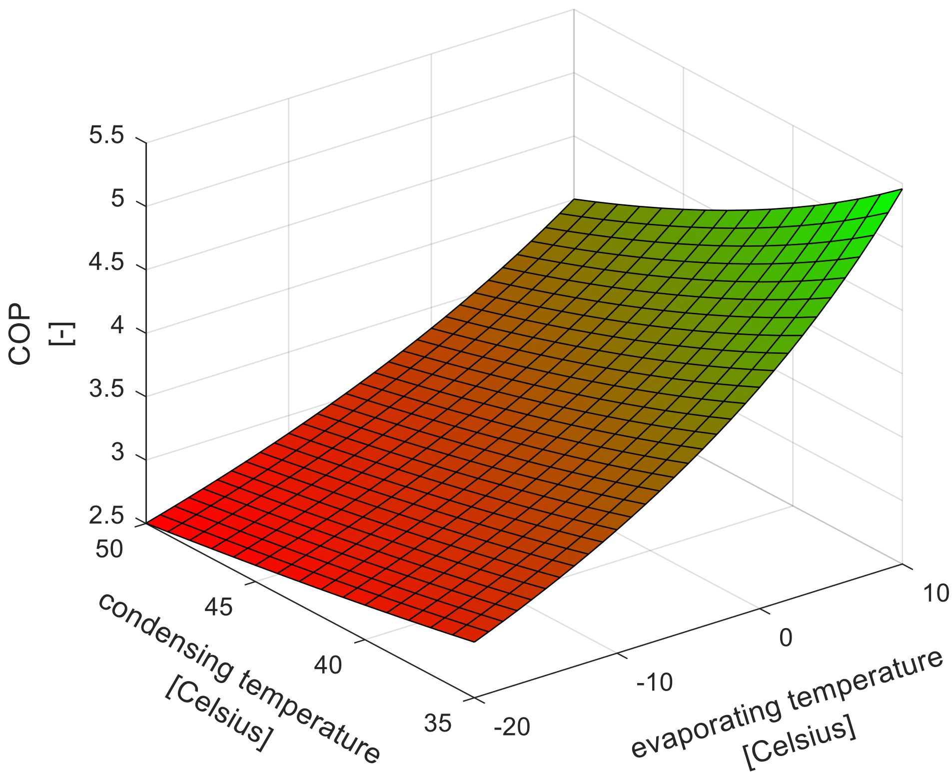

Figure 23 MATLAB based COP system evaluation

COP obtained can be different than exposed graph from Figure 23, this being only an indicator of the equipment which work in the environment with limits of -20˚C to 10 ˚C for evaporator and for heating entry liquid temperature above 35 ˚C and below 50 ˚C. Should be noted that in case of heat exchangers are selected for transferring the thermal energy from refrigerant to a liquid at condenser and respectively from liquid to refrigerant at evaporator, additional temperature differences should be tacked into account. To conclude, refrigerant evaporating and condensing temperatures differences can overpass the temperature difference of 5 ˚C selected for simulation. Heat exchangers temperature limits between input and output additionally to refrigerant temperature difference required for heat transfer, represent the total temperature differences. For simulation made in this paper, temperature differences were selected only for refrigerant to boil inside evaporator respectively to condensate and drop the heat in condenser.

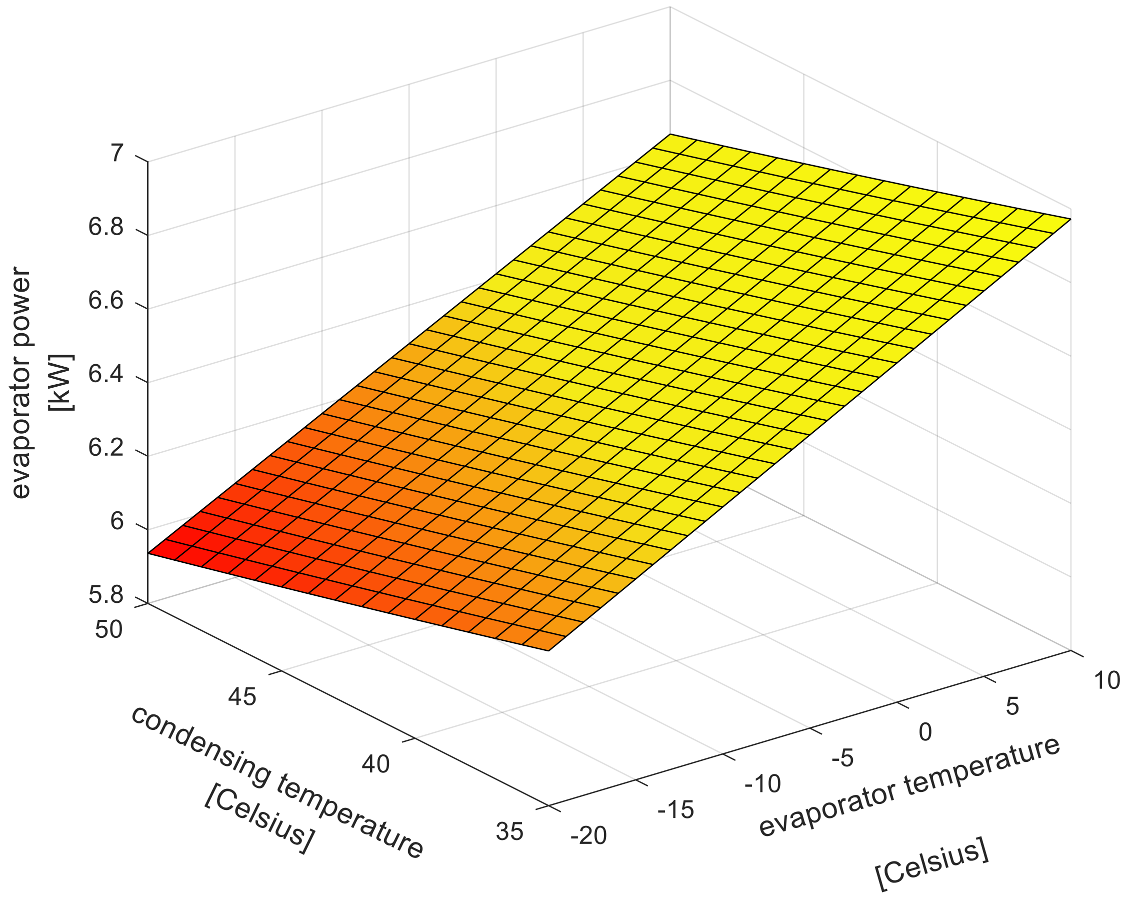

Figure 24 MATLAB based evaporator power

Evaporator power is graphically presented in surface elevated of Figure 24. The evaporator power is comprised in a small interval. From the worst condition at -20 ˚C evaporator temperature and 50 ˚C condensing temperature to maximum obtained value at 35 ˚C condensing, respectively 10 ˚C evaporator temperatures, evaporator power increase with only 16.80% for a gain in compressor electrical demand equal to 55%. This reveal the fact that doubling electrical demand in compressor power the evaporator capacity increase only with 16.80%.

All parameters simulated in the MATLAB software are intently to be analyzed in future simulations or experimental studies. Fact that constant condensing power, constant temperature difference for entire analyzed interval, constant evaporator and compressor isentropic efficiency, merge to a uncertain result in a desired error limitation. To observe deeply those parameters behavior and values, an evaluation should be done. In experimental study chapter of this paper, a heat pump were supposed to be measured. In experimental study plan, all measurement points and data acquisition systems are defined. All uncertainty parameters are identified and named in a specific chapter of the paper. A correlation between model and experiment will be lunched in a simulation program calibration chapter. Also, some simulation related model calibrated after experimental measurement are lunched to show energetic gain for some examples of equipment states.

3. Uncertainty parameters identification

Parameters identification, is an important step in experiment design. Knowing what are the needs to create an experimental study and necessity of identification of certain parameters, is an important objective. Priory to experimental plan, a SW based simulation is necessary to established what is known in equipment functionality, what can be measured directly on the experiment bench and what data cannot be measured directly and required a mathematical calculation.

Some of this paper suppositions are related evaporator efficiency comparing ideal condition, isentropic compressor efficiency, comparing ideal compressor without losses. Refrigerant states parameters, such evaporating and condensing pressure corresponding temperatures, differences between environment and refrigerant temperatures are parameters which can be measured directly on bench. Electrical power consumption can be measured directly using a data acquisition station for currents and voltages. For compressor efficiency [Equation 6] is re-drawn into [Equation 12].

Equation 12

$$\eta_c = \left(P_{EL} \cdot \frac{\theta_{CD} + \Delta_{CD} + 273.15}{\theta_{CD} - \theta_{VP} + \Delta_{CD} + \Delta_{VP}}\right)^{-1} \cdot\ 0.016 \cdot Q_{CD} \cdot \rho_w \cdot c_W \cdot (t_C - t_R) \cdot 10^{-3}$$

For a specific equipment, condensing temperature can be directly measured with temperature resistor or thermocouple sensors, or condensing pressure can be measured by refrigerant corresponding manifold.

For evaporator state, method is the same as for condenser. The differences is in select the right manifold indicator in case cold or hot equipment is evaluated. The condensing and evaporator temperatures can be stored in specific data tables or can be measured by manifold and manually the pressures and temperatures being noted in the table.

Condenser power is evaluated by thermal balance at condenser equipment. For [Equation 12] a heat exchanger were imagined, thermal balance being wrote as an equation between flow rate inside secondary of heat exchanger and temperature of water at inlet and exist pipes of it. The liquid have a specific heat and a specific density noted c_w respectively ρ_w in equation and Q_{cd} as flow rate in $l \cdot min^{-1}$. Constant 0.016 represent te correlation between minutes (from flow rate) and seconds from Joule to Watt conversion. All measured parameters together with equation will return after mathematical calculation the value of isentropic efficiency of compressor.

4. Measured data list

Measured data, as previously analyzed for compressor efficiency identification are given by [Equation 12]. Electrical power can be registered using a data acquisition station, a power meter or energy registration unit. Electrical power for three phase AC current, is a product between current, voltage and power factor, and is given by [Equation 13].

Equation 13

$$P_{EL\_sym} = 3^{0.5} \cdot U \cdot I \cdot \cos(\Phi) \cdot 10^{-3}$$

Equation 14

$$P_{EL\_sym} = \cos(\Phi) \cdot 10^{-3} \cdot \sum_{i} U_i \cdot I_i$$

Electrical energy comprise entire electrical power consumption during a period of time. Because heat pumps didn't work continuously and his power can be different from time to time, due to inverter technology and refrigerant states, total energy can be written as a sum of all energy measurements in constant periods of time, specially set for constant values of current and voltage.

Equation 15

$$E_{sym} = 1.73 \cdot \cos(\Phi) \cdot 10^{-3} \cdot \sum_{i} U_i \cdot I_i \cdot \tau_i$$

The [Equation 15] is true when three phase symmetrical load is applied on power consumption. This represent an equal current value on each phase.

For data acquisition station which provide effective value of alternative current, it is necessary summing of each individual line power. This approach is more realistic when line currents are not equal, and the loads are not symmetrical. The effective value of current is commonly used in electrical power and energy calculation.

Equation 16

$$E_{el} = \cos(\Phi) \cdot 10^{-3} \cdot \sum_{i=1}^{3} \sum_{j} U_{ij} \cdot I_{ij} \cdot \tau_i$$

Power factor is an electrical compressor parameter and depend on each technology of compressors. For this paper, the power factor parameter value, is selected from equipment datasheet provided by manufacturer. This parameter is used for correction of apparent power, given by product between current and voltage measurements. Due to reactive power of inductive like compressor coils, real power is smaller compering direct measurements. Generally this correction can be made by data acquisition station, through an initial setup of power facto value, or can be introduced after measurements, when a dedicated energy calculation is done. In [Equation 16] is presented the mathematical relation to be used in unsymmetrically three phase electrical power systems. Sampling rate should be equal to all three phases, and can be also a constant sampling rate, depends on data acquisition tool.

Table 2 main parameters list

Main parameters		
param	description	units
I_1	current phase 1	A
I_2	current phase 2	A
I_3	current phase 3	A
t_{CD}	condensing temperature	˚C
t_{VP}	evaporator temperature	˚C
t_C	heat exchanger secondary out temperature	˚C
t_R	heat exchanger secondary input temperature	˚C
Q_{CD}	heat exchanger secondary liquid flow rate	l/min
U_1	voltage phase 1	V
U_2	voltage phase 2	V
U_3	voltage phase 3	V
p_{VP}	evaporator pressure	bar
p_{CD}	condensing pressure	bar
τ	sampling rate	hour

Table 3 secondary parameters list

Secondary parameters		
t_{11}	heat exchanger primary input temperature	˚C
t_{12}	heat exchanger primary output temperature	˚C
t_{OH}	overheated evaporator vapour temperature	˚C
t_{SC}	subcooled condenser refrigerant liquid temperature	˚C

In Table 2 is presented the list of parameters to be measured and registered during experimental data registration. Are proposed two types of measurements; main parameters being used in mathematical modelling calibration presented in above methodology, respectively some secondary measurements parameter list, being helpful for additional evaluations of equipment.

Generally, temperature measurements are subjected to an evaluation of acquisition sensor calibration. Over time, resistive sensors and thermocouples register degradation from their initial values, when return data were calibrated. A representative measurement should take into account a new calibration, or used the last calibration in warrant period of time. Registered curves of sensors types are directly dependent with measured temperature value, sometimes the difference between positive and negative temperatures or big temperature intervals not being a constant value. For different values of temperatures, same sensor can reveal unequal values of calibration differences.

Table 4 thermocouple calibration results

Calibration parameters results				
	measured value	reference value	calibration value	units
	m_v	r_v	m_v-r_v	
thermocuple 1				˚C
thermocuple 2				˚C
thermocuple 3				˚C
thermocuple 4				˚C
thermocuple 5				˚C
thermocuple 6				˚C
thermocuple 7				˚C
thermocuple 8				˚C

Calibration results **Table 4** should be multiplied for all temperatures considered as relevant or which make case of new calibration value. If in entire temperature interval, values are constant, one table can be sufficient to be used in calibration of measurements and future calculations.

5. Experimental study plan

Experimental study, main purpose of this paper, to be used in model calibration, require a deeper analyze of studied equipment, measurements acquisition stations and calibration necessities for used sensors.

Starting from model description, the experimental study were performed on an ASHP for building heating purpose and DHW production. Model description of this study targets the isentropic efficiency and evaporator respectively condenser temperature differences comparing their environments of a heat pump. Those values are used after in the mathematical model for energy evaluation of such systems, in a specific period of time and for a specific customer.

The equipment is a Mitsubishi Electric ZUBADAN PUHZ-SHW112YHA R410A refrigerant based ASHP system. A three phase compressor power related electrical consumption is around 2.5kW, and come with two fan speed for forced air convection purpose, with electrical consumption equal 0.074kW. Fan electrical power consumption value, together with automation unit and condenser heat exchanger circulating pump should be part of entire electrical demand to be used into COP evaluation and equipment specificity. Heat pump present a defrosting mechanism, by reverse cycle method, with a compressed vapor into evaporator, for temperature increase purpose, to be able to reach positive temperature values in evaporator coils when defrosting is needed.

Table 5 Experimental Heat Pump datasheet specification

Service Ref.					PUHZ-SHW112YHA PUHZ-SHW140YHA
OUTDOOR UNIT	Power supply (phase, cycle, voltage)				3phase, 50Hz, 400V
		Max. current		A	14
	External finish				Munsell 3Y 7.8/1.1
	Refrigerant control				Linear Expansion Valve
	Compressor				Hermetic
		Model			ANB33FJLMT
		Motor output		kW	2.5
		Starter type			Inverter
		Protection devices			HP switch, LP switch Discharge thermo, Comp. surface thermo
	Crankcase heater			W	—
	Heat exchanger				Plate fin coil
	Fan	Fan(drive) × No.			Propeller fan × 2
		Fan motor output		kW	0.074+0.074
		Airflow		m³/min(CFM)	100(3,530)
	Defrost method				Reverse cycle
	Noise level		Cooling	dB	51
			Heating	dB	52
	Dimensions		W	mm(in.)	950(37-3/8)
			D	mm(in.)	330+30(13+1-3/16)
			H	mm(in.)	1,350(53-1/8)
	Weight			kg(lbs)	134(295)
	Refrigerant				R410A
		Charge		kg(lbs)	5.5(12.1)
		Oil (Model)		L	1.40(FV50S)
REFRIGERANT PIPING	Pipe size O.D.		Liquid	mm(in.)	9.52(3/8)
			Gas	mm(in.)	15.88(5/8)
	Connection method		Indoor side		Flared
			Outdoor side		Flared
	Between the indoor & outdoor unit		Height difference		Max. 30m
			Piping length		Max. 75m

External unit presented in Figure 25 contain the most critical elements of the heat pump. In case of heating necessity evaporator, compressor, expansion valve and reversing unit are comprise by the unit. Also, automation oh heat pump and inverter for driving the compressor are included. Evaporator in case of heating purpose or condenser in case of cooling needs, is built from refrigerant cooper coils with thin aluminum plates register for heat transfer.

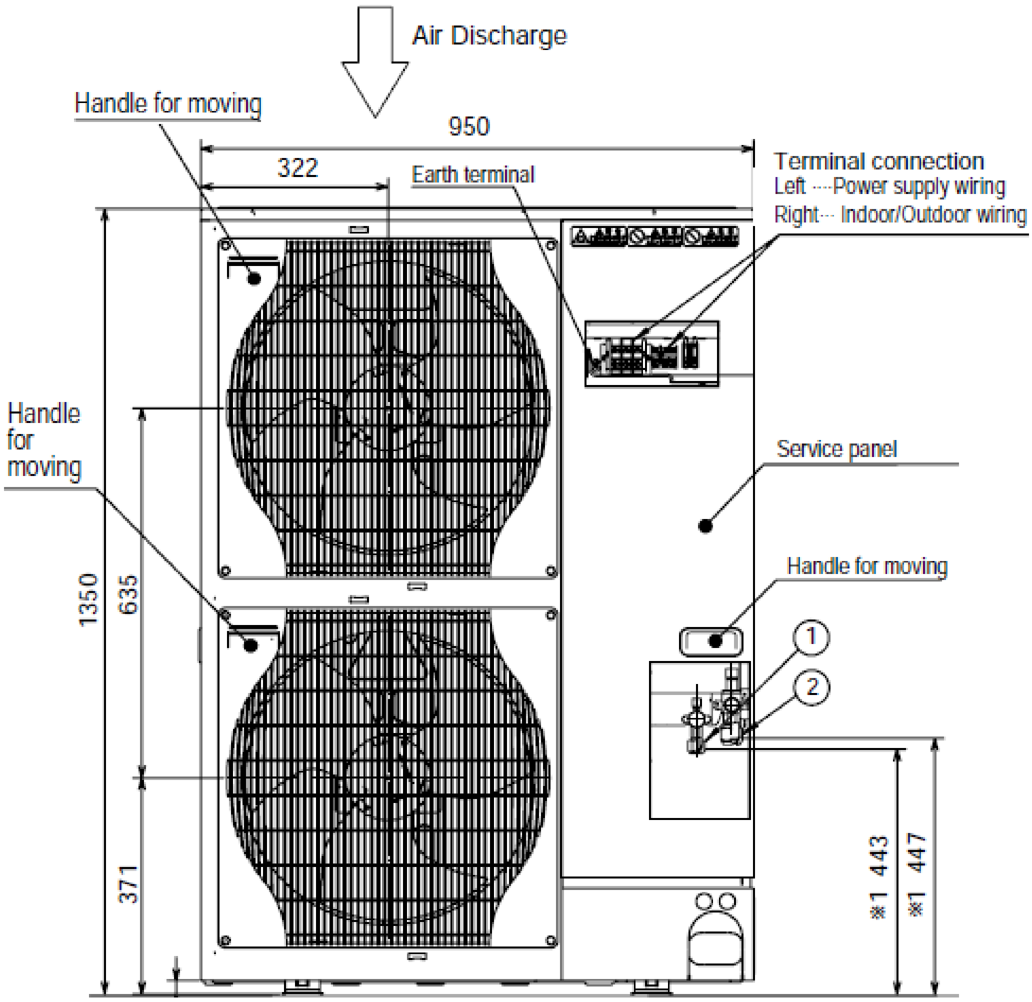

Figure 25 Experimental heat pump external unit

Terminal plugs for refrigerant gas (point 1 in the picture) and liquid (point 2 in the picture) lines, are available to be handled from exterior, for maintenance or installation purpose. Also a removable service panel make possible electrical connections for power supply and also internal unit cable connections. The two fans useful for air source equipment increased heat transfer efficiency, are electrically operated by the same inverter unit, being able to permanently sweep their rotating speed.

Heat pump external unit have implemented a high efficient heat pump technology, with 8 thermistors for temperature measurements, mounted in key positions like liquid and discharge line, or evaporator and suction lines. Three linear expansion valves are included, two as primary and secondary refrigerant expansion valves, for liquid expansion into power receiver respectively into evaporator(in heating) or condenser(in cooling); the third expansion valve is used in parallel connection for heat interchange circuit when used.

An important element of external unit is the four way valve, component which make the entire system reversible to be used in heating or cooling, depends on necessity. In power receiver, the refrigerant is accumulated, entire system being coupled to this; liquid line for refrigerant receiving from internal unit, compressor refrigerant input, through a coil immersed for preheating/precooling of evaporated gas, and expansion valve lines for immersion of refrigerant into evaporator/condenser. A complete hydraulic diagram is presented in Figure 39 part of Annex 5.

Internal unit is comprise of a heat plate exchanger, a circulating pump for secondary liquid flow, the water temperature controller, DHW accumulator and heating water accumulator. Hydraulic diagram for DHW and heating unit are presented in Annex 6 with element description inside Table 14. Heating accumulator is a 200 liters insulated water tank, with two entries for hot water and return water of heating equipment respectively two outputs, one for cooled water which entry inside plate heat exchanger and the last one going to heating equipment of the building.

Building heating equipment is formed by two low temperature floor units, with their specific automation and agent preparation units. A main circuit for floor heating units is present. Water alimentation circuit is possible due to an additional circulating pump, able to drive the water through the system at the required flow rate. Circulating pump for secondary unit of heat exchanger, is an WILO Yonos PICO 25/1-6-(EU3), with maximum power consumption equal to 40W able to deliver a flow rate above value of $15 l \cdot min^{-1}$prezented in Figure 46 of Annex 7.

Selection between DHW and heating water tanks is made with a three way hydraulic valve, in priority being DHW tank instead of heating unit which is considered having a sufficient thermal inertia until hot water demand is delivered at required temperature value. DHW water tank have an immersed coil for heating agent, specific clean water being delivered through comparing with daily heated water which have a based ground water source. Water tank for DHW and three way valve connection are presented inside Annex 7, Figure 42. Three way controlling valve is an ON/OFF valve, with 3/2inch standard connection plugs. It works controlled by main automation unit, priority state A selecting the DHW tank circuit. On main display of controlling RC, is displayed the circuit in work, with “water heating” respectively “heating” for heating accumulation tank. An example of displaying in active heating mode is presented in Annex 7, Figure 44. For heating accumulation tank, the installed system is based on 200litres tank equipped with an expansion tank to keep the required pressure into the system, presented in Annex 7, Figure 43.

For heating and DHW preparation, the system condenser consist of a heat plate exchanger with 500mm height and 120mm depth, built from cooper foils with 2 plugs for each primary respectively secondary circuit. Secondary circuit plugs are standard 3/4inch connection and primary plugs specially designed for hard soldering, where condensate liquid respectively hot vapors lines are connected. Water lines, from secondary heat exchanger unit are equipped with standard 3/4inch water taps for maintenance purpose, and an overpressure protection valve, right to it, presented in Figure 45.

Liquid and heated vapor lines, well insulated, have a short traces between external unit and condensing heat plate exchanger, causing a minimal heat losses in circuit. Plugs are standard 3/8inch for liquid respectively 5/8inch for gas line refrigerant connections presented in Figure 47 of Annex 7.

Experimental study bench is equipped with data acquisition stations, for temperatures, water flowmeter and electrical current consumptions units.

For phase currents measurements a PicoLog CM3 acquisition data logger from Pico technology is used. Current clamp accuracy +/- 2% with a 0.1 to 200 Amperes AC RMS range with a scaling factor equal to 1mV/A. Data logger accuracy is +/- 1% for measurements under 20mV and +/-2.5 for currents up to 200mV. For this paper, the accuracy of interest regarding CM3 voltage, is kept +/- 1% due to currant range which not pass 15A. System components can be followed into Annex 8 Figure 48. All specification for current clamp and CM3 station are presented in Figure 49. Main power connection of heat pump is directly measured by current acquisition station data logger, and information registered into PC through an USB connection.

Temperatures and flowmeter are sampled and registered with Ahlborn Almemo 5690-2 data acquisition station and saved into PC through a USB connection. Data logger in working progress is presented in Annex 8 Figure 50. System consist in dedicated measurements plugs connections, for each type of sensors connectors having internally built in analog voltage scaling devices. This system is high performance and high resolution with increased speed of sampling, being able to register over 50 data per second. Data input plug modules are automatically detected and measurements conversion are directly displayed into their specific units. The format of information is text based and saved through Ahlborn SW delivered by same manufacturer.

In Table 6 is presented the measurement points for heat-pump temperatures, flowmeter and current clamps with correlation pictures in Annex 8 and input locations for dedicated data acquisition unit.

Table 6 Experimental study bench Ahlborn and PicoLog measurements sensors location description

Measurement points				
abbreviation	description	Almemo/CM3 input	picture	units
I_1	phase 1 power line	CM3 in 1	-	A
I_2	phase 2 power line	CM3 in 2	-	A
I_3	phase 2 power line	CM3 in 3	-	A
t_{11}	heat exchanger primary input temperature	Almemo in 94	Figure 38	˚C
t_{12}	heat exchanger primary output temperature	Almemo in 93	Figure 38	˚C
t_C	heat exchanger secondary out temperature	Almemo in 90	Figure 38	˚C
t_R	heat exchanger secondary input temperature	Almemo in 91	Figure 38	˚C
Q_{CD}	heat exchanger secondary liquid flow rate	Almemo in 92	Figure 39	l/min
t_{in_c}	compressor input temperature	Almemo in 97	Figure 40	˚C
t_{out_c}	compressor output temperature	Almemo in 96	Figure 40	˚C
t_{in}	interior temperature	Almemo in 95	-	˚C
$t_{in\_air\_ev}$	evaporator input air temperature	Almemo in 99	-	˚C
$t_{out\_air\_ev}$	evaporator output air temperature	Almemo in 98	Figure 41	˚C
τ	sampling rate	-	-	hour

Evaporator temperature sensor return a dry bulb value and all the other temperatures are type K thermocouple sensors with corresponding calibration values inside Table 7 and graphically exposed into Figure 26. Calibration were conducted to see the differences related to a reference sensor and measurements accuracy. Calibration station were done inside laboratories of Technical University of Civil Engineering - Faculty of Building Services and presented in Figure 55 and Figure 56. Calibration station for reference sensor is LAUDA DigiCal DGS 2 and together with LAUDA calibration bath with standard distilled water liquid used to immerge the sensors. Data were manually registered into a text document and future used for calibration results values calculation.

Inside Table 7 multiple temperatures are presented. Calibration reference sensor are listed in the left of the table with red color cell fill type, representing the displayed value of LAUDA DigiCal thermometer at the set value of LAUDA bath listed in the first column cells of table. Calibration time is also listed, time delay between measurements being enough to reach set temperature. In the table are listed in cells with white fill color, the values measured with Almemo data acquisition station. The grey and light orange color represent the calibration values, being the difference between reference sensor and thermocouple measured values. Light orange values in a column were used to calculate the average level and displayed in dark orange inside average row. As can be seen, the selected values for averaging are selected only to satisfy the range of temperatures were thermocouples work in experimental study. As an example, for input temperature of gas at suction line of compressor, temperature values didn't overpassed 20˚C comparing output compressor temperatures values in working process which didn't fall below values of 40˚C. Graphically, calibration temperature results are plotted into Figure 26, were on primary axes are distributed calibration set points.

Table 7 Experimental study bench thermocouples measurements calibration results and reference sensor values for 6 temperatures

set value[°C]	Ref sensor		time	t_C	t_R	t_{12}	t_{11}	t_{out_c}	t_{in_c}	
1 ˚C	0,91	°C	11:33:54	1,7	3,4	1,7	1,7	1,8	1,6	°C
			calibration value	-0,79	-2,49	-0,79	-0,79	-0,89	-0,69	
10 ˚C	10,26	°C	11:41:06	10,5	11,1	10,6	10,5	10,6	10,4	°C
			calibration value	-0,24	-0,84	-0,34	-0,24	-0,34	-0,14	
20 ˚C	20,19	°C	11:50:39	20,5	19,2	20,5	20,5	20,5	20,5	°C
			calibration value	-0,31	0,99	-0,31	-0,31	-0,31	-0,31	
30 ˚C	30,02	°C	11:59:05	30,5	27,1	30,4	30,5	30,5	30,4	°C
			calibration value	-0,48	2,92	-0,38	-0,48	-0,48	-0,38	
40 ˚C	39,97	°C	12:05:39	40,5	36,1	40,3	40,4	40,4	40,4	°C
			calibration value	-0,53	3,87	-0,33	-0,43	-0,43	-0,43	
50 ˚C	49,85	°C	12:12:55	50,6	46,3	50,4	50,5	50,6	50,5	°C
			calibration value	-0,75	3,55	-0,55	-0,65	-0,75	-0,65	
Average				-0,52	2,83	-0,39	-0,52	-0,59	-0,38	°C
legend					values used for mediation					
					average calibration value					
					table fill					
					calibration value					
					reference value					

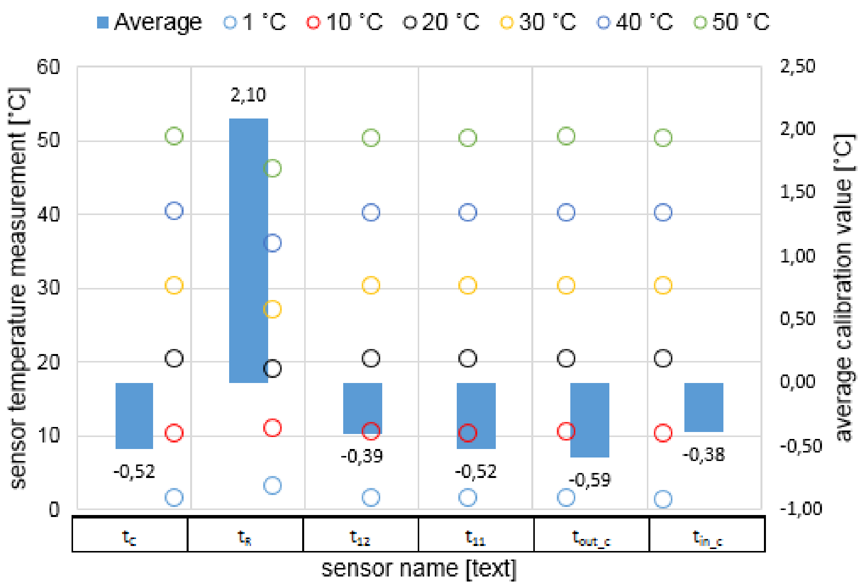

Figure 26 Experimental study bench thermocouples calibration measurements results

Related sensor measurements can be observed in circle points and average values plotted in bars with values displayed on below end point correlated in secondary axe.

Interesting values are displayed on graph. Measurement thermocouples present closer calibration results with one exception related with return water temperature from heating accumulation tank. This high deviation, comparing with the other thermocouples results, is caused by thermocouple possible defects like contact resistance between the two metals and changed over time due to ageing effect. Calibration results obtained by this measurement are used in data processing chapter for parameters identification.

Experimental study bench is installed into a production factory in Ciocanesti town Dambovita County. Placement location of heat pump evaporator is at the North of the building, with all day working in a shady environment, solar irradiation never touching the external unit. Experimental measurements were conducted in a cold period of winter during 21 – 24 February 2018.

Table 8 Experimental study bench evaporator and condenser temperature difference evaluation

time		$t_{exterior}$			$t_{out\_evaporator}$			t_{12} exchanger secondary output			t_{11} exchanger secondary input		
28.03.2018	19:35	average value	9.17	°C	average value	9.85	°C	average value	39.25	°C	average value	26.88	°C
								t_{ml_cond}	32.68	°C			
Manifold data		t_{evap_man}	2	˚C	t_{cond_man}	46	˚C	evaporator temperature dfference				7.17	°C
		p_{evap_man}	7.8	Bar	p_{cond_man}	26.5	Bar	condenser temperature dfference				13.32	°C
28.03.2018	19:40	average value	8.25	°C	average value	9.80	°C	average value	40.37	°C	average value	26.8	°C
								t_{ml}	33.13	°C			
Manfold data		t_{evap_man}	1	˚C	t_{cond_man}	54	˚C	evaporator temperature dfference				7.25	°C
		p_{evap_man}	7.5	Bar	p_{cond_man}	32.4	Bar	condenser temperature dfference				20.87	°C
28.03.2018	19.46	average value	8.80	°C	average value	11.13	°C	average value	45.07	°C	average value	29.67	°C
								t_{ml}	36.83	°C			
Manfold data		t_{evap_man}	1	˚C	t_{cond_man}	57	˚C	evaporator temperature dfference				7.80	°C
		p_{evap_man}	7.4	Bar	p_{cond_man}	35	Bar	condenser temperature dfference				20.17	°C

Condensing and evaporation temperatures used in mathematical model description were manually read on a specific R410a refrigerant manifold, with pressures plugs presented in Annex 8 Figure 57 and a read example in Figure 58. Cold and hot gas lines plugs are R410a standards refrigerant connectors with red color for high pressure line respectively blue color for low pressure line. Example presented in Table 8 shows the values of 7,8Bar corresponding with 2˚C evaporating respectively 35 bar corresponding 57˚C condensing temperatures. Evaporator temperature differences show an appropriate value in all 3 measurements of system functioning, condenser temperature difference showing only an equivalence at 32.4

bar respectively 35 bar condensing pressures, the value from 26.5 bar not being taken into consideration due to high difference. This value can come from a measurement error or the heat pump weren't in equilibrium state when pressure were registered. The read data and evaluation is given in Table 8

Temperature differences were evaluated with Equation 17 and Equation 18 were logarithmic medium temperature is given by Equation 19. At the condenser, the condensing temperature is considered as being a value of temperature inside interval of condensing secondary circuit temperature difference, by evaluation with logarithmic medium technique.

Equation 17

$$\Delta_{VP} = t_{ext} - t_{VP\_man}$$

Equation 18

$$\Delta_{CD} = t_{ml\_cond} - t_{CD\_man}$$

Equation 19

$$t_{ml\_cond} = \frac{t_{12} - t_{11}}{ln\frac{t_{12}}{t_{11}}}$$

Temperatures differences measurements with manifold show for constant evaporator environment temperature the compressor driver is able to maintain a constant pressure inside evaporator for a good control of evaporator heat transfer rate. The pressure in condenser correspond to a condensing temperature setup which were varied during measurement between 35°C and 45°C with a 5K temperature steps. For those 3 condenser temperature setup, the heat pump only change the condensing temperature varying compressor working frequency or change the pressure stop threshold.

6. Registered data processing and parameters identification

Registered data, is a result of the experimental study during 4 days from 21.02.2018 until 24.02.2018, time interval were more than 25000 samples for each of 13 measurement parameters. A big data base for the system were realized to be used in the future processing.

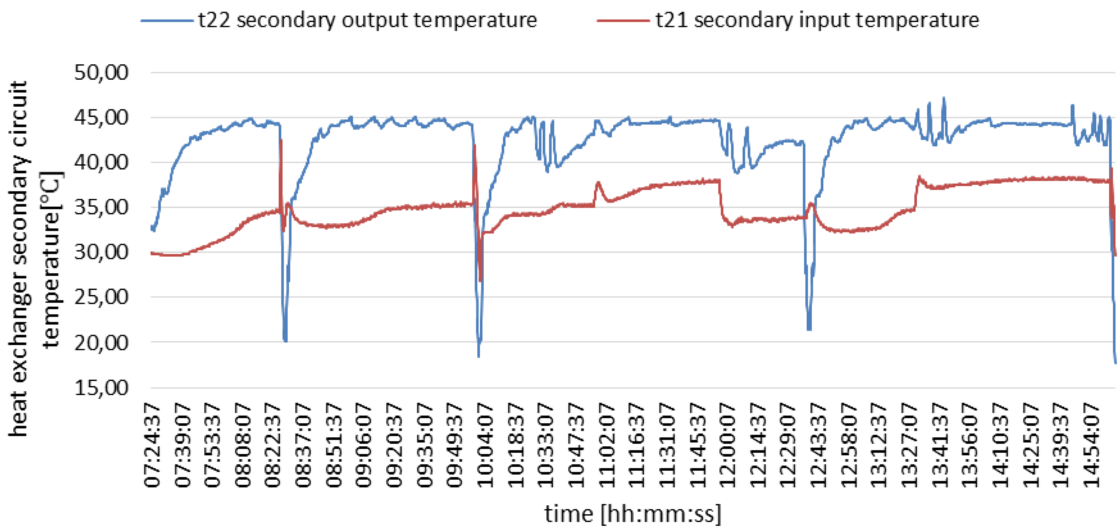

Figure 27 Experimental study bench measurements of condenser secondary circuit temperatures from 21.02.2018

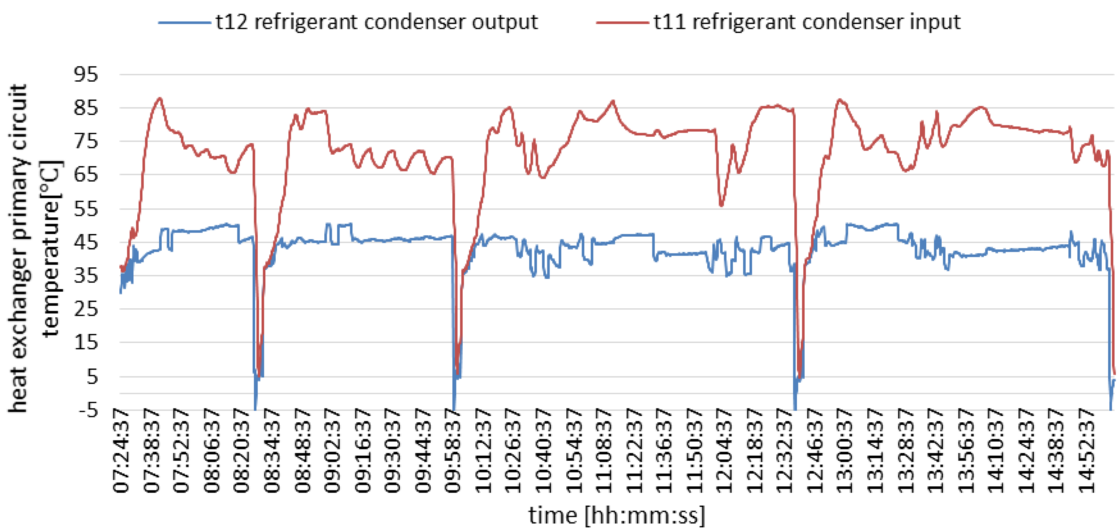

Figure 28 Experimental study bench measurements of condenser primary circuit temperatures from 21.02.2018

Sampling rate were set at 10s, enough to take relevant data in an inertia point of view, due to big dimensions of system comparing latency. Data measurements were divided in 23 intervals, some of them bigger than others, without a specific requirement but more to be able displaying of them in a readable format.

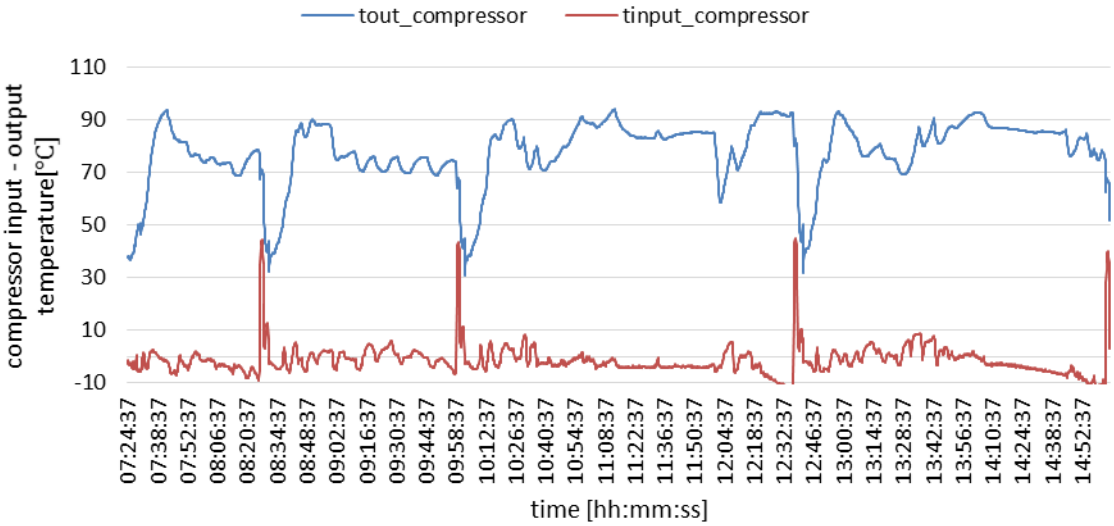

Figure 29 Experimental study bench measurements of input and output compressor plugs temperatures from 21.02.2018

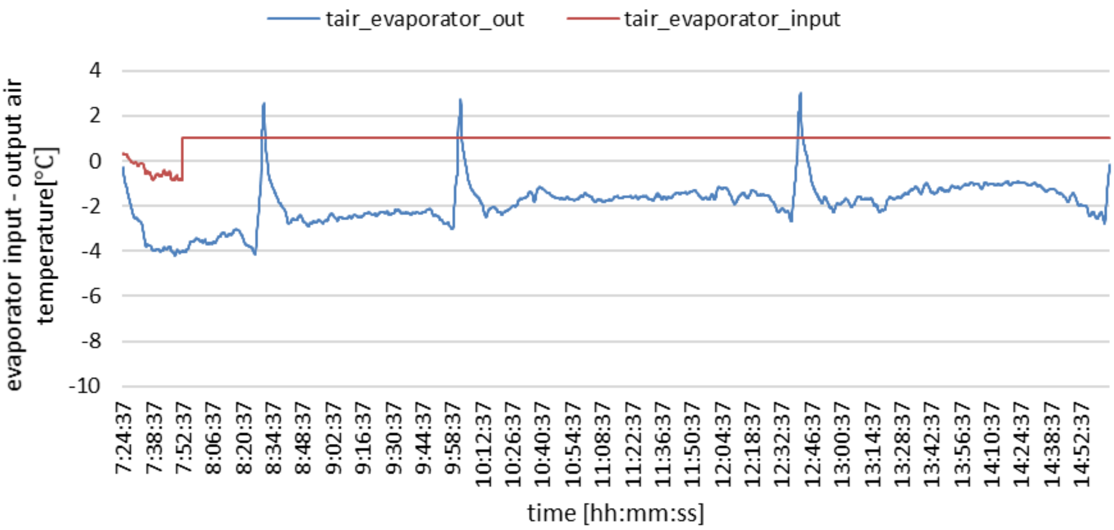

Figure 30 Experimental study bench measurements of input and output evaporator air temperatures from 21.02.2018

For evaporator environment temperatures, input and output, temperatures registered shows a sampling error for external temperature or inlet air temperature measured with a dry bulb temperature sensor. This error is a consequence not investigated but the inlet air temperature is obtained by calculation with medium temperature differences, obtained at the second experiment. This difference can be seen in Figure 33.

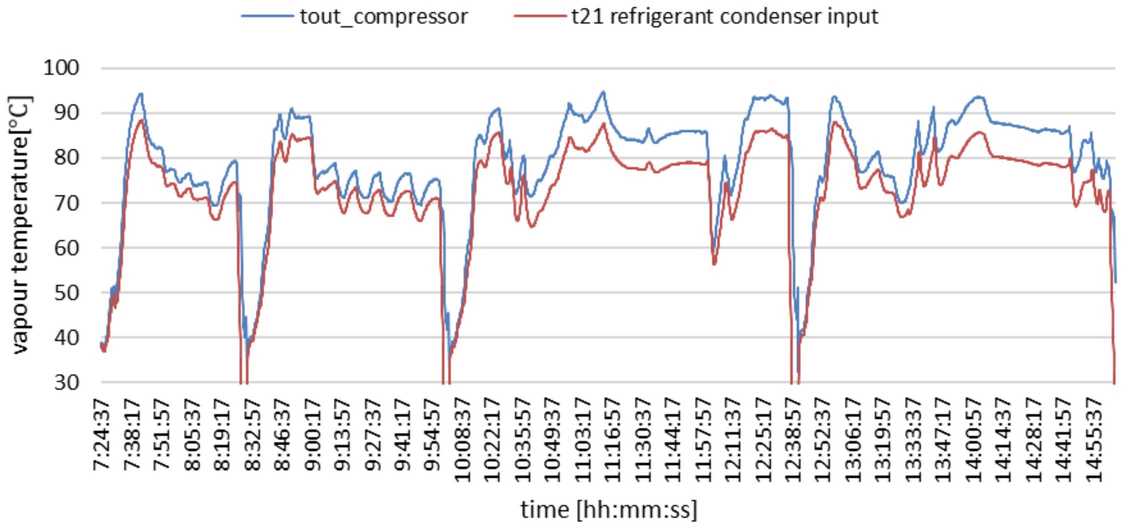

Figure 31 Experimental study bench measurements vapor air temperature drop between compressor output and condenser input from 21.02.2018

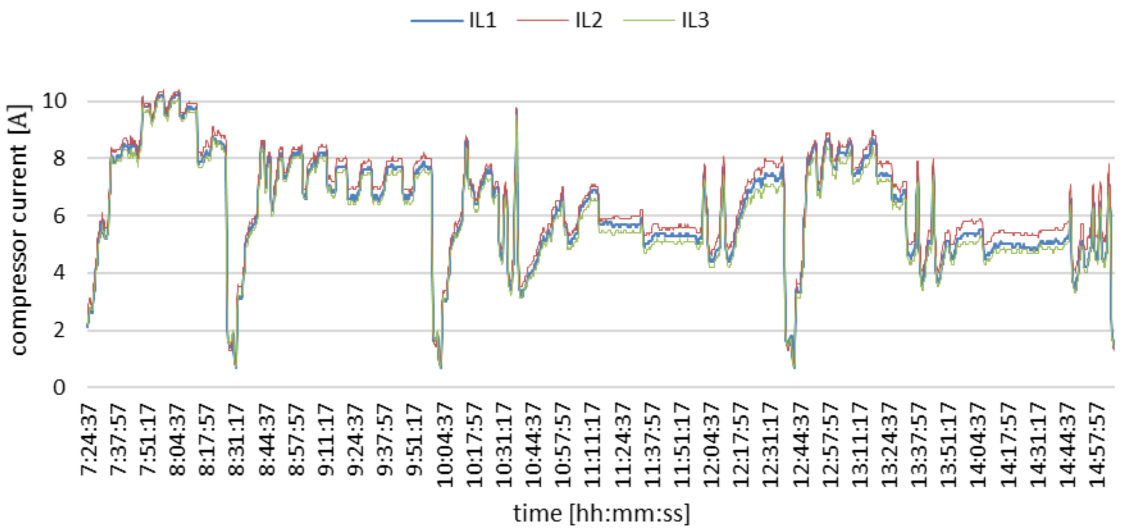

Figure 32 Experimental study bench measurements of compressor input current from 21.02.2018

Average value of temperature drop, graphically shown on Figure 33 of air inside evaporator have value of 1.19K. Because the measurements of evaporator inlet temperature were stopped during the experimental study in 21.02.2018, the exterior temperature drop in evaporator air were conducted again in 28.03.2018. The average value obtained were used to establish the right external temperature of the environment from which heat is extracted by refrigerant.

The evaluation of compressor efficiency is now easy to be done, sing condenser temperature as a logarithmic medium value between inlet and outlet of condenser heat exchanger. For evaporator is used the evaporator outlet air temperature with difference drop added to it value. For its value, efficiency is evaluated with Equation 20 using Equation 5 for condenser power and Equation 13 for electrical power.

Equation 20

$$\eta_c = \left(\frac{t_{ml\_cond} + \Delta_{CD} + 273.15}{t_{ml\_cond} - \theta_{VP} + \Delta_{CD} + \Delta_{VP}}\right)^{-1} \cdot \frac{P_{CD}}{P_{EL}}$$

Condensing temperature is obtained using inlet and output plugs temperatures of secondary heat exchanger circuit and temperature difference at condenser and evaporator being before established. Both parameters used in this paper have 20.17K for condensing and respectively 7.80K for evaporator. Rewriting Equation 20 we obtain Equation 21 were only condenser and evaporator temperatures and condenser and electrical power are used.

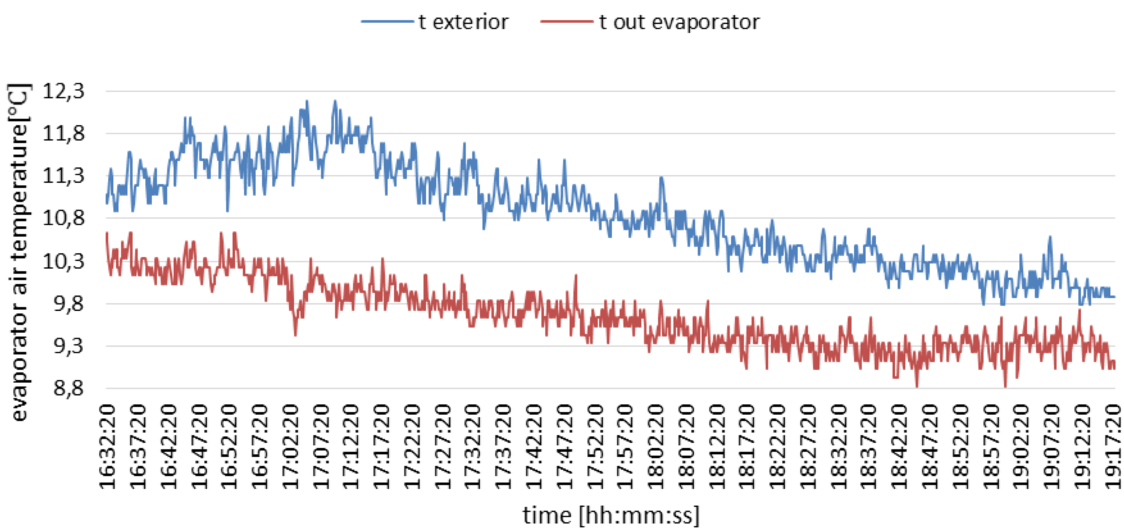

Figure 33 Experimental study bench measurements of temperature drop in evaporator input and output 28.03.2018

Equation 21

$$\eta_c = \left(\frac{t_{ml\_cond} + 293.32}{t_{ml\_cond} - \theta_{VP} + 27.97}\right)^{-1} \cdot \frac{P_{CD}}{P_{EL}}$$

Electrical power is obtained after summing of power on all three lines, using a power factor coefficient equal 0.9.

In Figure 34 are presented model evaporator and condensing temperatures. Those temperature represent real condensing and evaporator temperatures for which are added or subtraction temperatures difference for condenser respectively evaporator.

By calculation using condenser and electrical power equations, in Figure 35 are presented condenser and electrical power obtained at condenser heat exchanger output respectively electrical power consumption at heat pump voltage plugs. Some of condenser power are negative; this is a normal behavior in heat pump functioning, because circulating pump is always on, and the temperature inside heat exchanger is transferred from secondary circuit to primary.

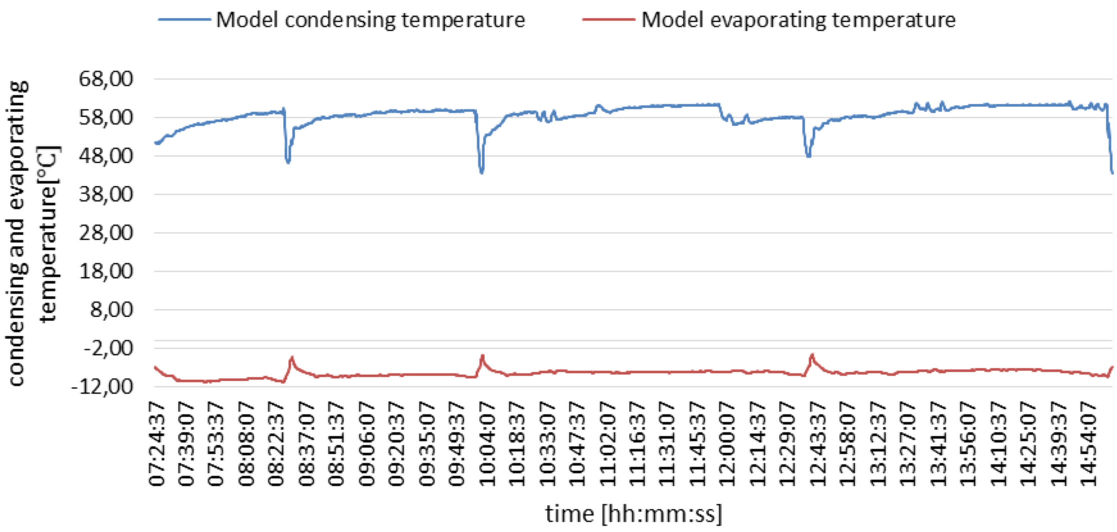

Figure 34 Condensing and evaporator model temperatures for 21.02.2018

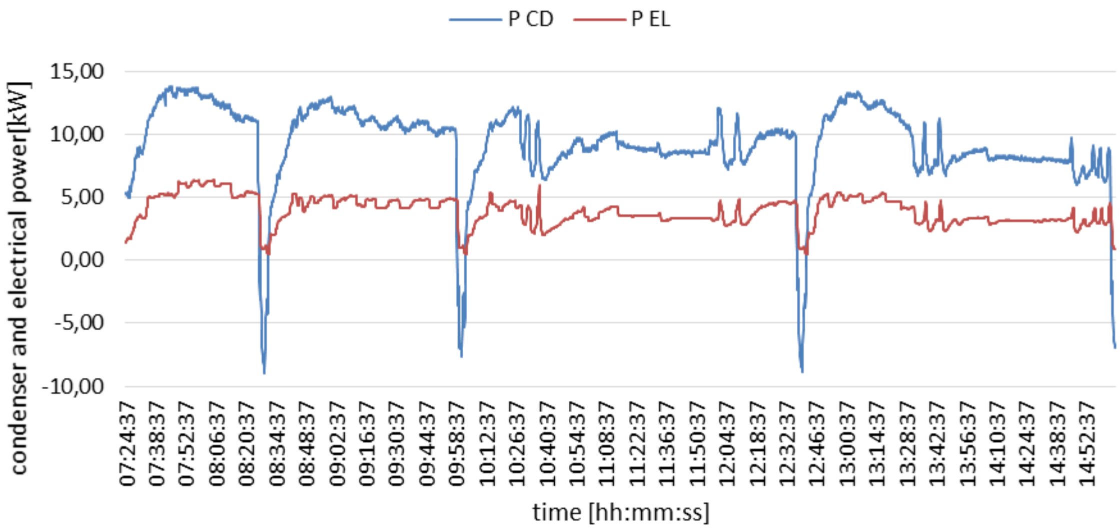

Figure 35 Condensing power and electrical power consumption for the experiment of 21.02.2018

For electrical power consumption versus compressor efficiency, representing lines are plotted in Figure 36. Again, some data seems to not be realistic. The compressor efficiency cannot be negative. Normally the model evaluation should be stopped when compressor didn't work. Model try to estimate the efficiency based on a negative condenser power which is wrong. For this cause, the interval of interest is again reduced starting with 07:20:00 until 08:35:00, data length were compressor power is not equal to 0, which means that heat pump entirely work in this time.

A mean value of 0.5038 were obtained for compressor efficiency in this time interval. Registration and calculation show that compressor efficiency is also a dynamic parameter which is depending of system calibration or temperature setup. This value will be kept in next chapter for program calibration.

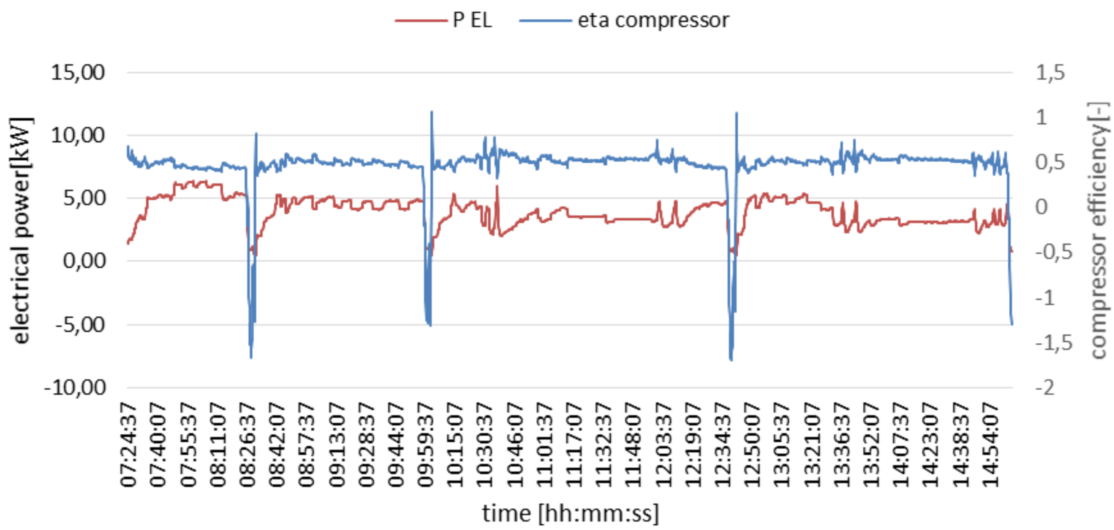

Figure 36 Electrical power consumption and compressor efficiency for data from 21.02.2018

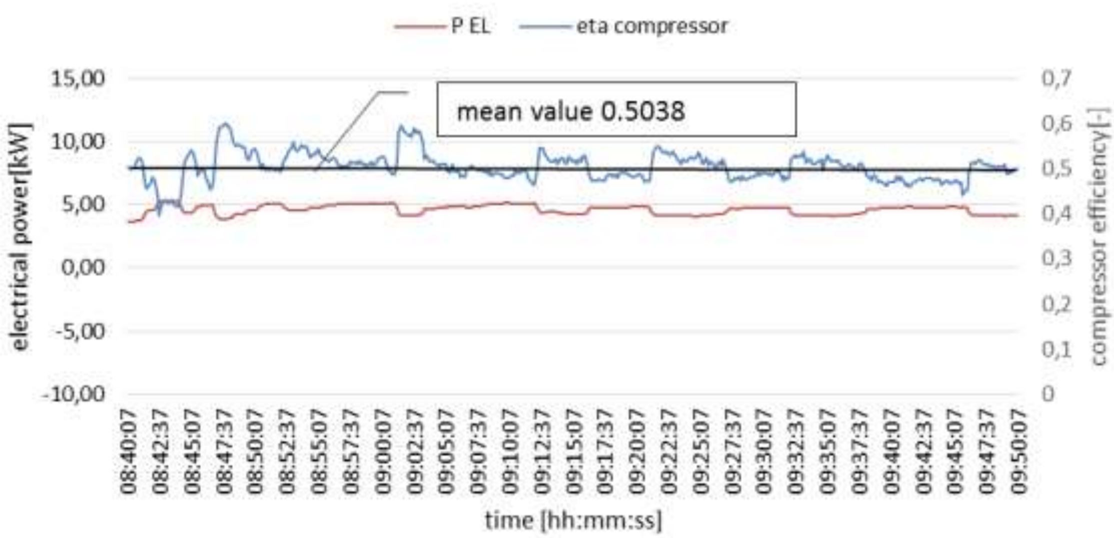

Figure 37 Electrical power efficiency and efficiency mean value for 07:20 to 08:35 data of 21.02.2018

7. Simulation program calibration

Simulation program calibration is conducted to setup the simulation program for evaluation of equivalence between the mathematical model and experimental registration. For comparison the measured electrical power directly on heat pump input plugs and calculated electrical power using model described by Equation 7.

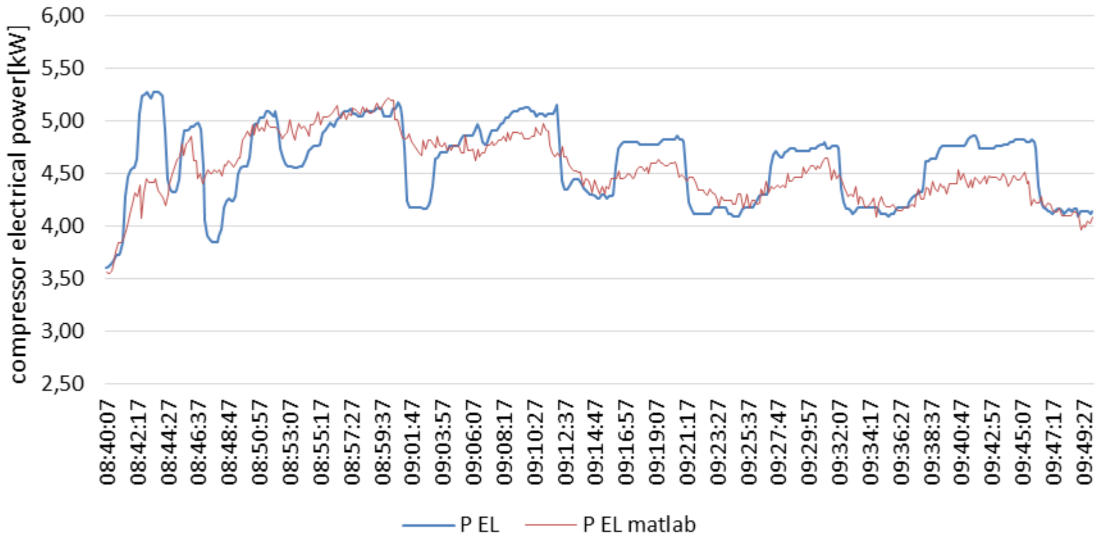

Figure 38 Compressor electrical power from 21.02.2018 measurement and MATLAB based model

For MATLAB based calculation model were used the evaporator inlet temperature measured in experiment together with logarithmic medium temperature given by Equation 19. Compressor efficiency were set at 0.5083 value obtained as a mean value of measured data. The comparison shows similar results for compressor electrical demand when a specific condenser power is took by experimental measurement and evaluated as a thermal balance of water in heat exchanger secondary circuit.

Comparing both curves, in electrical power measurements can be observed a thermal inertia which smooth the line. For calculated values, the model used a sampled data for air temperature at evaporator, and condenser water temperature at the condenser, both behavior of equipment having less inertial typo comparing the entire system.

MATLAB used software can be seen below, were input temperatures for condenser, evaporator, condenser and evaporator temperature differences, and condenser power were read from a specific location. All values are registered value or calculated values based on this paper methodology.

```
% refrigerant cycle parameters-------------------------------------
teta_cd      = xlsread('date_prelucrate.xlsx','matlab','H2:H422');%
[Celsius]
```

```
teta_vp      = xlsread('date_prelucrate.xlsx','matlab','U2:U422');%
[Celsius]
delta_cd     = 20.17;% [Celsius]
delta_vp     = 7.1;% [Celsius]
%-------------------------------------------------------------------

%sistem parameters--------------------------------------------------
eta_c        = 0.5038;
P_CD         = xlsread('date_prelucrate.xlsx','matlab','AB2:AB422');
%-------------------------------------------------------------------

%RCTC---------------------------------------------------------------
ctl          = (teta_vp-delta_vp)./(teta_cd+delta_cd+273.15);% [-]
cth          = (teta_cd+delta_cd)./(teta_cd+delta_cd+273.15);% [-]
%-------------------------------------------------------------------

%results------------------------------------------------------------
%compressor power---------------------------------------------------
P_EL = eta_c.^(-1).*P_CD.*(cth-ctl);% [kW]
    xlswrite('date_prelucrate.xlsx',P_EL,'matlab','AF2:AF422')   ;

%heating COP--------------------------------------------------------
COP_CD = P_CD./P_EL;% [-]
    xlswrite('date_prelucrate.xlsx',COP_CD,'matlab','AG2:AG422')   ;
%end----------------------------------------------------------------
```

As a short conclusion, this paper give a methodology for experimental evaluation of temperature differences of heat pump systems at their evaporator and condenser. Mathematical model used for compressor efficiency calculation is also described and its mathematical evaluation show similar results for experiment and calculation data.

Regarding the methodology used for condenser power evaluation and electrical power calculation, this paper show similar results by graphically exposure of both experimental and calculated curves.

Future modelling research and parameters investigation can be done. Some simplifying techniques were used, like no thermal inertia considered in the system or averaging the measurements to extract a value of interest to be used after in the simulation. Also, in the heat pump, a more realistic evaluation can be done on pressure based measurements of refrigerant condensing and evaporating temperatures with electronic data acquisition station.

Regarding condensing real temperature obtained in heat exchanger, for this material were used the logarithmic mean values between entry and exist plugs of heat exchanger. Different technique can be useful to be done to extract condensing temperature value.

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Annex 1 Simulation Data for 50˚C

Table 9 DUPREX calculated values at 50ºC Condensing temperature

Cycle 4: DUPREX 4.0								
Single-stage Heat Pump Cycle								
Refrigerant:	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A
Pressure Line + Condenser	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95
Evaporator	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09
Suction Line	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
Return Temperature tR [C]	45	45	45	45	45	45	45	45
Temperature Difference DT1c [K]	5	5	5	5	5	5	5	5
Condensation tc [C]	50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1
Mean Temperature [C]	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5
Condensation Pressure pc [bar]	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056
Heat Source	-20	-19	-18	-17	-16	-15	-14	-13
Temperature Difference DT1o [K]	5	5	5	5	5	5	5	5
Evaporation to [C]	-25	-24	-23	-22	-21	-20	-19	-18
Mean Temperature [C]	-25.4	-24.4	-23.3	-22.3	-21.3	-20.3	-19.3	-18.3
Evaporation Pressure po [bar]	3.2941	3.4257	3.5613	3.701	3.8448	3.993	4.1454	4.3023
Subcooling [K]	5	5	5	5	5	5	5	5
Superheat (Evaporator) [K]	5	5	5	5	5	5	5	5
Superheat (Suction Line) [K]	10	10	10	10	10	10	10	10
Volumetric Efficiency [-]	1	1	1	1	1	1	1	1
Isentropic Efficiency [-]	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
Pressure Ratio p2/p1 [-]	10.036	9.634	9.253	8.89	8.544	8.216	7.903	7.605
Pressure Difference p2-p1 [bar]	28.501	28.37	28.234	28.095	27.951	27.803	27.65	27.493
Theoret. Displacement [m3/h]	12.52	12.11	11.72	11.34	10.98	10.64	10.31	9.99
Mass Flow [kg/s]	0.0392	0.0395	0.0397	0.0399	0.0401	0.0404	0.0406	0.0408
Volumetric Capacity [kJ/m3]	1591.2	1660.1	1731.3	1804.8	1880.7	1959	2039.8	2123.1
Condenser Heat Qc [kW]	10	10	10	10	10	10	10	10
Compressor Power [kW]	4.11	4.06	4	3.95	3.89	3.83	3.78	3.72
Refr. Capacity Qo [kW]	5.53	5.58	5.64	5.69	5.74	5.79	5.84	5.89
COP_h [-]	2.43	2.47	2.5	2.53	2.57	2.61	2.65	2.69
COP_c [-]	1.35	1.38	1.41	1.44	1.48	1.51	1.55	1.58
	t	t	t	t	t	t	t	t
	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]
< 1a >	-20.7	-19.68	-18.66	-17.64	-16.62	-15.6	-14.59	-13.57
< 1 >	-10.7	-9.68	-8.66	-7.64	-6.62	-5.6	-4.59	-3.57
< 2 >	130.81	129.51	128.22	126.96	125.72	124.49	123.29	122.1
< 3 >	45	45	45	45	45	45	45	45
< 4 >	-25.05	-24.05	-23.05	-22.05	-21.05	-20.05	-19.05	-18.05
	h	h	h	h	h	h	h	h
	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]
< 1a >	416.3	416.78	417.25	417.71	418.17	418.63	419.08	419.52
< 1 >	425.37	425.89	426.41	426.93	427.44	427.94	428.45	428.94
< 2 >	530.19	528.69	527.21	525.76	524.33	522.91	521.52	520.15
< 3 >	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24
< 4 >	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24
< 1-2 >	104.81	102.8	100.8	98.83	96.89	94.97	93.07	91.2
< 1-2s >	73.37	71.96	70.56	69.18	67.82	66.48	65.15	63.84

R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A
0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95
0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09
0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
45	45	45	45	45	45	45	45	45	45	45	45
5	5	5	5	5	5	5	5	5	5	5	5
50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1
50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5
30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056
-12	-11	-10	-9	-8	-7	-6	-5	-4	-3	-2	-1
5	5	5	5	5	5	5	5	5	5	5	5
-17	-16	-15	-14	-13	-12	-11	-10	-9	-8	-7	-6
-17.3	-16.3	-15.3	-14.3	-13.3	-12.3	-11.3	-10.3	-9.2	-8.2	-7.2	-6.2
4.4637	4.6296	4.8002	4.9756	5.1559	5.341	5.5312	5.7265	5.927	6.1328	6.344	6.5606
5	5	5	5	5	5	5	5	5	5	5	5
5	5	5	5	5	5	5	5	5	5	5	5
10	10	10	10	10	10	10	10	10	10	10	10
1	1	1	1	1	1	1	1	1	1	1	1
0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
7.321	7.051	6.793	6.546	6.311	6.086	5.872	5.666	5.47	5.282	5.102	4.93
27.332	27.166	26.995	26.82	26.64	26.455	26.264	26.069	25.869	25.663	25.452	25.235
9.68	9.39	9.11	8.84	8.58	8.32	8.08	7.85	7.62	7.41	7.2	7
0.0411	0.0413	0.0415	0.0417	0.042	0.0422	0.0424	0.0426	0.0429	0.0431	0.0433	0.0435
2209	2297.6	2388.9	2483	2579.9	2679.7	2782.4	2888.2	2997.1	3109.2	3224.5	3343.1
10	10	10	10	10	10	10	10	10	10	10	10
3.67	3.61	3.56	3.5	3.45	3.39	3.34	3.28	3.23	3.17	3.12	3.06
5.94	5.99	6.04	6.09	6.15	6.2	6.25	6.3	6.35	6.4	6.45	6.5
2.73	2.77	2.81	2.85	2.9	2.95	3	3.05	3.1	3.15	3.21	3.27
1.62	1.66	1.7	1.74	1.78	1.83	1.87	1.92	1.97	2.02	2.07	2.12
t	t	t	t	t	t	t	t	t	t	t	t
[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]
-12.55	-11.54	-10.52	-9.51	-8.5	-7.48	-6.47	-5.46	-4.45	-3.43	-2.42	-1.41
-2.55	-1.54	-0.52	0.49	1.5	2.52	3.53	4.54	5.55	6.57	7.58	8.59
120.93	119.78	118.64	117.52	116.42	115.33	114.26	113.2	112.16	111.13	110.11	109.11
45	45	45	45	45	45	45	45	45	45	45	45
-17.05	-16.05	-15.06	-14.06	-13.06	-12.06	-11.06	-10.06	-9.06	-8.06	-7.06	-6.06
h	h	h	h	h	h	h	h	h	h	h	h
[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]
419.97	420.4	420.84	421.26	421.69	422.1	422.52	422.92	423.33	423.72	424.12	424.5
429.44	429.93	430.42	430.9	431.38	431.85	432.32	432.78	433.25	433.7	434.15	434.6
518.79	517.45	516.13	514.82	513.53	512.26	511	509.76	508.53	507.32	506.12	504.93
275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24
275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24
89.35	87.52	85.71	83.92	82.16	80.41	78.68	76.98	75.29	73.62	71.97	70.33
62.54	61.26	60	58.75	57.51	56.29	55.08	53.88	52.7	51.53	50.38	49.23

R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	
0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	
0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	
0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	
45	45	45	45	45	45	45	45	45	45	45	
5	5	5	5	5	5	5	5	5	5	5	
50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1	50.1	
50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	50.5	
30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	30.7056	
0	1	2	3	4	5	6	7	8	9	10	
5	5	5	5	5	5	5	5	5	5	5	
-5	-4	-3	-2	-1	0	1	2	3	4	5	
-5.2	-4.2	-3.2	-2.2	-1.2	-0.2	0.8	1.8	2.8	3.8	4.8	
6.7828	7.0107	7.2443	7.4837	7.7291	7.9805	8.2381	8.5019	8.772	9.0485	9.3315	
5	5	5	5	5	5	5	5	5	5	5	
5	5	5	5	5	5	5	5	5	5	5	
10	10	10	10	10	10	10	10	10	10	10	
1	1	1	1	1	1	1	1	1	1	1	
0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	
4.765	4.607	4.456	4.311	4.171	4.037	3.909	3.786	3.667	3.553	3.444	
25.013	24.785	24.551	24.312	24.066	23.815	23.557	23.294	23.024	22.747	22.464	
6.8	6.62	6.43	6.26	6.09	5.93	5.77	5.62	5.47	5.33	5.19	
0.0438	0.044	0.0442	0.0444	0.0447	0.0449	0.0451	0.0453	0.0455	0.0458	0.046	
3465	3590.4	3719.3	3851.8	3987.9	4127.8	4271.5	4419.1	4570.6	4726.2	4885.9	
10	10	10	10	10	10	10	10	10	10	10	
3.01	2.95	2.9	2.84	2.79	2.73	2.68	2.62	2.57	2.51	2.46	
6.55	6.6	6.65	6.7	6.75	6.8	6.85	6.9	6.94	6.99	7.04	
3.33	3.39	3.45	3.52	3.59	3.66	3.73	3.81	3.89	3.98	4.07	
2.18	2.24	2.29	2.36	2.42	2.49	2.56	2.63	2.7	2.78	2.86	
t	t	t	t	t	t	t	t	t	t	t	
[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	
-0.4	0.61	1.62	2.63	3.64	4.64	5.65	6.66	7.67	8.68	9.68	
9.6	10.61	11.62	12.63	13.64	14.64	15.65	16.66	17.67	18.68	19.68	
108.12	107.14	106.18	105.23	104.29	103.36	102.44	101.54	100.64	99.76	98.89	
45	45	45	45	45	45	45	45	45	45	45	
-5.06	-4.06	-3.07	-2.07	-1.07	-0.07	0.93	1.93	2.93	3.93	4.93	
h	h	h	h	h	h	h	h	h	h	h	
[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	
424.88	425.26	425.63	425.99	426.35	426.7	427.05	427.38	427.72	428.04	428.36	
435.04	435.48	435.91	436.34	436.76	437.18	437.6	438	438.4	438.8	439.19	
503.76	502.6	501.45	500.31	499.19	498.07	496.97	495.88	494.79	493.72	492.66	
275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	
275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	275.24	
68.72	67.12	65.54	63.97	62.42	60.89	59.37	57.87	56.39	54.92	53.46	
48.1	46.98	45.87	44.78	43.7	42.62	41.56	40.51	39.47	38.44	37.43	

Annex 2 Simulation Data for 45˚C

Table 10 DUPREX calculated values at 45oC Condensing temperature

Cycle 4: DUPREX 4.0								
Single-stage Heat Pump Cycle								
Refrigerant:	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A
Pressure Line + Condenser	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95
Evaporator	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09
Suction Line	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
Return Temperature tR [C]	40	40	40	40	40	40	40	40
Temperature Difference DT1c [K]	5	5	5	5	5	5	5	5
Condensation tc [C]	45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1
Mean Temperature [C]	45.5	45.5	45.5	45.5	45.5	45.5	45.5	45.5
Condensation Pressure pc [bar]	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354
Heat Source to1 [C]	-20	-19	-18	-17	-16	-15	-14	-13
Temperature Difference DT1o [K]	5	5	5	5	5	5	5	5
Evaporation to [C]	-25	-24	-23	-22	-21	-20	-19	-18
Mean Temperature [C]	-25.4	-24.4	-23.3	-22.3	-21.3	-20.3	-19.3	-18.3
Evaporation Pressure po [bar]	3.2941	3.4257	3.5613	3.701	3.8448	3.993	4.1454	4.3023
Subcooling [K]	5	5	5	5	5	5	5	5
Superheat (Evaporator) [K]	5	5	5	5	5	5	5	5
Superheat (Suction Line) [K]	10	10	10	10	10	10	10	10
Volumetric Efficiency [-]	1	1	1	1	1	1	1	1
Isentropic Efficiency [-]	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
Pressure Ratio p2/p1 [-]	8.968	8.609	8.267	7.943	7.635	7.341	7.062	6.796
Pressure Difference p2-p1 [bar]	25.131	25	24.864	24.724	24.581	24.432	24.28	24.123
Theoret. Displacement [m3/h]	12.34	11.94	11.55	11.18	10.82	10.48	10.15	9.84
Mass Flow [kg/s]	0.0387	0.0389	0.0391	0.0393	0.0395	0.0398	0.04	0.0402
Volumetric Capacity [kJ/m3]	1696.1	1769.2	1844.7	1922.6	2003.1	2086.1	2171.7	2260
Condenser Heat Qc [kW]	10	10	10	10	10	10	10	10
Compressor Power [kW]	3.84	3.78	3.72	3.67	3.61	3.56	3.5	3.45
Refr. Capacity Qo [kW]	5.81	5.87	5.92	5.97	6.02	6.07	6.12	6.17
COP_h [-]	2.61	2.65	2.69	2.73	2.77	2.81	2.86	2.9
COP_c [-]	1.52	1.55	1.59	1.63	1.67	1.71	1.75	1.79
	t	t	t	t	t	t	t	t
	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]
< 1a >	-20.7	-19.68	-18.66	-17.64	-16.62	-15.6	-14.59	-13.57
< 1 >	-10.7	-9.68	-8.66	-7.64	-6.62	-5.6	-4.59	-3.57
< 2 >	123.25	121.95	120.66	119.39	118.15	116.92	115.71	114.52
< 3 >	40	40	40	40	40	40	40	40
< 4 >	-25.05	-24.05	-23.05	-22.05	-21.05	-20.06	-19.06	-18.06
	h	h	h	h	h	h	h	h
	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]
< 1a >	416.3	416.78	417.25	417.71	418.17	418.63	419.08	419.52
< 1 >	425.37	425.89	426.41	426.93	427.44	427.94	428.45	428.94
< 2 >	524.57	523.1	521.65	520.21	518.8	517.41	516.04	514.68
< 3 >	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94
< 4 >	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94
< 1-2 >	99.2	97.21	95.23	93.29	91.37	89.47	87.59	85.74
< 1-2s >	69.44	68.04	66.66	65.3	63.96	62.63	61.31	60.02

R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A
0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95
0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09
0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
40	40	40	40	40	40	40	40	40	40	40	40
5	5	5	5	5	5	5	5	5	5	5	5
45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1
45.6	45.6	45.6	45.6	45.6	45.6	45.6	45.6	45.6	45.6	45.6	45.6
27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354
-12	-11	-10	-9	-8	-7	-6	-5	-4	-3	-2	-1
5	5	5	5	5	5	5	5	5	5	5	5
-17	-16	-15	-14	-13	-12	-11	-10	-9	-8	-7	-6
-17.3	-16.3	-15.3	-14.3	-13.3	-12.3	-11.3	-10.3	-9.2	-8.2	-7.2	-6.2
4.4637	4.6296	4.8002	4.9756	5.1559	5.341	5.5312	5.7265	5.927	6.1328	6.344	6.5606
5	5	5	5	5	5	5	5	5	5	5	5
5	5	5	5	5	5	5	5	5	5	5	5
10	10	10	10	10	10	10	10	10	10	10	10
1	1	1	1	1	1	1	1	1	1	1	1
0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
6.542	6.3	6.07	5.849	5.639	5.438	5.247	5.063	4.888	4.72	4.559	4.405
23.962	23.796	23.625	23.45	23.27	23.084	22.894	22.699	22.498	22.293	22.081	21.865
9.53	9.24	8.96	8.69	8.44	8.19	7.95	7.72	7.5	7.28	7.08	6.88
0.0404	0.0406	0.0409	0.0411	0.0413	0.0415	0.0417	0.0419	0.0421	0.0424	0.0426	0.0428
2351	2444.8	2541.5	2641.1	2743.7	2849.4	2958.2	3070.1	3185.4	3304	3425.9	3551.4
10	10	10	10	10	10	10	10	10	10	10	10
3.39	3.34	3.28	3.23	3.17	3.12	3.06	3	2.95	2.89	2.84	2.78
6.23	6.28	6.33	6.38	6.43	6.48	6.53	6.58	6.63	6.68	6.73	6.78
2.95	3	3.05	3.1	3.15	3.21	3.27	3.33	3.39	3.45	3.52	3.59
1.84	1.88	1.93	1.98	2.03	2.08	2.13	2.19	2.25	2.31	2.37	2.44
t	t	t	t	t	t	t	t	t	t	t	t
[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]
-12.55	-11.54	-10.52	-9.51	-8.5	-7.48	-6.47	-5.46	-4.45	-3.43	-2.42	-1.41
-2.55	-1.54	-0.52	0.49	1.5	2.52	3.53	4.54	5.55	6.57	7.58	8.59
113.35	112.19	111.06	109.93	108.83	107.73	106.66	105.6	104.55	103.52	102.5	101.49
40	40	40	40	40	40	40	40	40	40	40	40
-17.06	-16.06	-15.06	-14.06	-13.06	-12.06	-11.06	-10.06	-9.06	-8.07	-7.07	-6.07
h	h	h	h	h	h	h	h	h	h	h	h
[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]
419.97	420.4	420.84	421.26	421.69	422.1	422.52	422.92	423.33	423.72	424.12	424.5
429.44	429.93	430.42	430.9	431.38	431.85	432.32	432.78	433.25	433.7	434.15	434.6
513.35	512.03	510.73	509.44	508.17	506.92	505.68	504.46	503.25	502.05	500.87	499.7
265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94
265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94
83.91	82.1	80.31	78.54	76.79	75.07	73.36	71.67	70	68.35	66.72	65.1
58.74	57.47	56.22	54.98	53.76	52.55	51.35	50.17	49	47.84	46.7	45.57

R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A
0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95
0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09
0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
40	40	40	40	40	40	40	40	40	40	40
5	5	5	5	5	5	5	5	5	5	5
45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1	45.1
45.6	45.6	45.6	45.6	45.6	45.6	45.6	45.6	45.6	45.6	45.6
27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354	27.3354
0	1	2	3	4	5	6	7	8	9	10
5	5	5	5	5	5	5	5	5	5	5
-5	-4	-3	-2	-1	0	1	2	3	4	5
-5.2	-4.2	-3.2	-2.2	-1.2	-0.2	0.8	1.8	2.8	3.8	4.8
6.7828	7.0107	7.2443	7.4837	7.7291	7.9805	8.2381	8.5019	8.772	9.0485	9.3315
5	5	5	5	5	5	5	5	5	5	5
5	5	5	5	5	5	5	5	5	5	5
10	10	10	10	10	10	10	10	10	10	10
1	1	1	1	1	1	1	1	1	1	1
0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
4.258	4.117	3.981	3.852	3.727	3.608	3.493	3.383	3.277	3.175	3.077
21.643	21.415	21.181	20.942	20.696	20.445	20.187	19.924	19.653	19.377	19.094
6.68	6.5	6.32	6.15	5.98	5.82	5.66	5.51	5.37	5.23	5.09
0.043	0.0432	0.0434	0.0436	0.0438	0.0441	0.0443	0.0445	0.0447	0.0449	0.0451
3680.4	3813	3949.4	4089.5	4233.4	4381.3	4533.2	4689.2	4849.4	5013.9	5182.7
10	10	10	10	10	10	10	10	10	10	10
2.73	2.68	2.62	2.57	2.51	2.46	2.4	2.35	2.29	2.24	2.18
6.83	6.88	6.93	6.98	7.03	7.08	7.13	7.18	7.23	7.28	7.33
3.66	3.74	3.82	3.9	3.98	4.07	4.16	4.26	4.36	4.47	4.58
2.5	2.57	2.65	2.72	2.8	2.88	2.97	3.06	3.15	3.25	3.35
t	t	t	t	t	t	t	t	t	t	t
[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]
-0.4	0.61	1.62	2.63	3.64	4.64	5.65	6.66	7.67	8.68	9.68
9.6	10.61	11.62	12.63	13.64	14.64	15.65	16.66	17.67	18.68	19.68
100.5	99.52	98.56	97.6	96.66	95.73	94.81	93.9	93	92.12	91.24
40	40	40	40	40	40	40	40	40	40	40
-5.07	-4.07	-3.07	-2.07	-1.07	-0.07	0.93	1.93	2.92	3.92	4.92
h	h	h	h	h	h	h	h	h	h	h
[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]
424.88	425.26	425.63	425.99	426.35	426.7	427.05	427.38	427.72	428.04	428.36
435.04	435.48	435.91	436.34	436.76	437.18	437.6	438	438.4	438.8	439.19
498.54	497.4	496.27	495.15	494.04	492.95	491.86	490.78	489.72	488.66	487.62
265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94
265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94	265.94
63.5	61.92	60.36	58.81	57.28	55.76	54.26	52.78	51.31	49.86	48.42
44.45	43.34	42.25	41.17	40.09	39.03	37.98	36.95	35.92	34.9	33.9

Annex 3 Simulation Data for 40˚C

Table 11 DUPREX calculated values at 40ºC Condensing temperature

Cycle 4: DUPREX 4.0								
Single-stage Heat Pump Cycle								
Refrigerant:	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A
Pressure Line + Condenser	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95
Evaporator	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09
Suction Line	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
Return Temperature tR [C]	35	35	35	35	35	35	35	35
Temperature Difference DT1c [K]	5	5	5	5	5	5	5	5
Condensation tc [C]	40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1
Mean Temperature [C]	40.6	40.6	40.6	40.6	40.6	40.6	40.6	40.6
Condensation Pressure pc [bar]	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565
Heat Source to1 [C]	-20	-19	-18	-17	-16	-15	-14	-13
Temperature Difference DT1o [K]	5	5	5	5	5	5	5	5
Evaporation to [C]	-25	-24	-23	-22	-21	-20	-19	-18
Mean Temperature [C]	-25.4	-24.4	-23.3	-22.3	-21.3	-20.3	-19.3	-18.3
Evaporation Pressure po [bar]	3.2941	3.4257	3.5613	3.701	3.8448	3.993	4.1454	4.3023
Subcooling [K]	5	5	5	5	5	5	5	5
Superheat (Evaporator) [K]	5	5	5	5	5	5	5	5
Superheat (Suction Line) [K]	10	10	10	10	10	10	10	10
Volumetric Efficiency [-]	1	1	1	1	1	1	1	1
Isentropic Efficiency [-]	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
Pressure Ratio p2/p1 [-]	7.992	7.672	7.368	7.079	6.804	6.542	6.293	6.056
Pressure Difference p2-p1 [bar]	22.052	21.921	21.785	21.646	21.502	21.354	21.201	21.044
Theoret. Displacement [m3/h]	12.19	11.78	11.4	11.03	10.68	10.34	10.01	9.7
Mass Flow [kg/s]	0.0382	0.0384	0.0386	0.0388	0.039	0.0392	0.0394	0.0397
Volumetric Capacity [kJ/m3]	1797.4	1874.6	1954.3	2036.5	2121.3	2208.9	2299.1	2392.2
Condenser Heat Qc [kW]	10	10	10	10	10	10	10	10
Compressor Power [kW]	3.57	3.51	3.46	3.4	3.35	3.29	3.24	3.18
Refr. Capacity Qo [kW]	6.08	6.14	6.19	6.24	6.29	6.34	6.4	6.45
COP_h [-]	2.8	2.85	2.89	2.94	2.99	3.04	3.09	3.14
COP_c [-]	1.7	1.75	1.79	1.83	1.88	1.93	1.98	2.03
	t	t	t	t	t	t	t	t
	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]
< 1a >	-20.7	-19.68	-18.66	-17.64	-16.62	-15.6	-14.59	-13.57
< 1 >	-10.7	-9.68	-8.66	-7.64	-6.62	-5.6	-4.59	-3.57
< 2 >	115.6	114.29	113	111.73	110.49	109.26	108.04	106.85
< 3 >	35	35	35	35	35	35	35	35
< 4 >	-25.05	-24.06	-23.06	-22.06	-21.06	-20.06	-19.06	-18.06
	h	h	h	h	h	h	h	h
	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]
< 1a >	416.3	416.78	417.25	417.71	418.17	418.63	419.08	419.52
< 1 >	425.37	425.89	426.41	426.93	427.44	427.94	428.45	428.94
< 2 >	518.87	517.42	515.99	514.58	513.18	511.81	510.46	509.13
< 3 >	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95
< 4 >	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95
< 1-2 >	93.5	91.52	89.57	87.65	85.75	83.87	82.02	80.18
< 1-2s >	65.45	64.07	62.7	61.35	60.02	58.71	57.41	56.13

R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A
0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95
0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09
0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
35	35	35	35	35	35	35	35	35	35	35	35
5	5	5	5	5	5	5	5	5	5	5	5
40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1
40.6	40.6	40.6	40.6	40.6	40.6	40.6	40.6	40.6	40.7	40.7	40.7
24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565
-12	-11	-10	-9	-8	-7	-6	-5	-4	-3	-2	-1
5	5	5	5	5	5	5	5	5	5	5	5
-17	-16	-15	-14	-13	-12	-11	-10	-9	-8	-7	-6
-17.3	-16.3	-15.3	-14.3	-13.3	-12.3	-11.3	-10.3	-9.2	-8.2	-7.2	-6.2
4.4637	4.6296	4.8002	4.9756	5.1559	5.341	5.5312	5.7265	5.927	6.1328	6.344	6.5606
5	5	5	5	5	5	5	5	5	5	5	5
5	5	5	5	5	5	5	5	5	5	5	5
10	10	10	10	10	10	10	10	10	10	10	10
1	1	1	1	1	1	1	1	1	1	1	1
0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
5.83	5.614	5.409	5.213	5.025	4.846	4.675	4.512	4.356	4.206	4.063	3.926
20.883	20.717	20.546	20.371	20.191	20.005	19.815	19.62	19.419	19.214	19.002	18.786
9.4	9.11	8.84	8.57	8.32	8.07	7.83	7.6	7.38	7.17	6.97	6.77
0.0399	0.0401	0.0403	0.0405	0.0407	0.0409	0.0411	0.0413	0.0415	0.0417	0.0419	0.0421
2488.2	2587.1	2689	2793.9	2902	3013.3	3127.9	3245.9	3367.3	3492.1	3620.6	3752.7
10	10	10	10	10	10	10	10	10	10	10	10
3.12	3.07	3.01	2.96	2.9	2.85	2.79	2.74	2.68	2.63	2.57	2.52
6.5	6.55	6.6	6.65	6.7	6.75	6.8	6.86	6.91	6.96	7.01	7.06
3.2	3.26	3.32	3.38	3.44	3.51	3.58	3.65	3.73	3.81	3.89	3.97
2.08	2.13	2.19	2.25	2.31	2.37	2.44	2.5	2.57	2.65	2.72	2.8
t	t	t	t	t	t	t	t	t	t	t	t
[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]
-12.55	-11.54	-10.52	-9.51	-8.5	-7.48	-6.47	-5.46	-4.45	-3.43	-2.42	-1.41
-2.55	-1.54	-0.52	0.49	1.5	2.52	3.53	4.54	5.55	6.57	7.58	8.59
105.68	104.52	103.38	102.25	101.14	100.05	98.97	97.9	96.86	95.82	94.8	93.79
35	35	35	35	35	35	35	35	35	35	35	35
-17.06	-16.06	-15.06	-14.06	-13.06	-12.07	-11.07	-10.07	-9.07	-8.07	-7.07	-6.07
h	h	h	h	h	h	h	h	h	h	h	h
[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]
419.97	420.4	420.84	421.26	421.69	422.1	422.52	422.92	423.33	423.72	424.12	424.5
429.44	429.93	430.42	430.9	431.38	431.85	432.32	432.78	433.25	433.7	434.15	434.6
507.81	506.51	505.23	503.97	502.72	501.48	500.26	499.06	497.87	496.69	495.53	494.38
256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95
256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95
78.37	76.58	74.82	73.07	71.34	69.63	67.94	66.27	64.62	62.99	61.37	59.78
54.86	53.61	52.37	51.15	49.94	48.74	47.56	46.39	45.24	44.09	42.96	41.84

R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A	R410A
0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95	0.95
0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09	0.09
0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05	0.05
35	35	35	35	35	35	35	35	35	35	35
5	5	5	5	5	5	5	5	5	5	5
40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1	40.1
40.7	40.7	40.7	40.7	40.7	40.7	40.7	40.7	40.7	40.7	40.7
24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565	24.2565
0	1	2	3	4	5	6	7	8	9	10
5	5	5	5	5	5	5	5	5	5	5
-5	-4	-3	-2	-1	0	1	2	3	4	5
-5.2	-4.2	-3.2	-2.2	-1.2	-0.2	0.8	1.8	2.8	3.8	4.8
6.7828	7.0107	7.2443	7.4837	7.7291	7.9805	8.2381	8.5019	8.772	9.0485	9.3315
5	5	5	5	5	5	5	5	5	5	5
5	5	5	5	5	5	5	5	5	5	5
10	10	10	10	10	10	10	10	10	10	10
1	1	1	1	1	1	1	1	1	1	1
0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7	0.7
3.795	3.669	3.548	3.432	3.321	3.215	3.113	3.014	2.92	2.829	2.742
18.564	18.336	18.102	17.863	17.617	17.366	17.108	16.845	16.575	16.298	16.015
6.58	6.4	6.22	6.05	5.88	5.72	5.57	5.42	5.28	5.14	5
0.0423	0.0425	0.0427	0.0429	0.0431	0.0433	0.0435	0.0437	0.0439	0.0441	0.0443
3888.5	4028.1	4171.6	4319.1	4470.6	4626.2	4786.1	4950.3	5118.8	5291.8	5469.4
10	10	10	10	10	10	10	10	10	10	10
2.46	2.41	2.35	2.3	2.24	2.19	2.14	2.08	2.03	1.97	1.92
7.11	7.16	7.21	7.26	7.31	7.36	7.4	7.45	7.5	7.55	7.6
4.06	4.15	4.25	4.35	4.46	4.57	4.68	4.8	4.93	5.07	5.21
2.89	2.97	3.06	3.16	3.25	3.36	3.47	3.58	3.7	3.83	3.96
t	t	t	t	t	t	t	t	t	t	t
[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]	[°C]
-0.4	0.61	1.62	2.63	3.64	4.64	5.65	6.66	7.67	8.68	9.68
9.6	10.61	11.62	12.63	13.64	14.64	15.65	16.66	17.67	18.68	19.68
92.8	91.81	90.85	89.89	88.94	88.01	87.09	86.18	85.28	84.39	83.51
35	35	35	35	35	35	35	35	35	35	35
-5.07	-4.07	-3.07	-2.07	-1.08	-0.08	0.92	1.92	2.92	3.92	4.92
h	h	h	h	h	h	h	h	h	h	h
[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]	[kJ/kg]
424.88	425.26	425.63	425.99	426.35	426.7	427.05	427.38	427.72	428.04	428.36
435.04	435.48	435.91	436.34	436.76	437.18	437.6	438	438.4	438.8	439.19
493.24	492.11	491	489.9	488.81	487.73	486.66	485.6	484.55	483.51	482.48
256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95
256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95	256.95
58.2	56.63	55.09	53.56	52.04	50.55	49.06	47.6	46.15	44.71	43.29
40.74	39.64	38.56	37.49	36.43	35.38	34.34	33.32	32.3	31.3	30.3

Annex 4 MATLAB results

Table 12 Simulation results comparison MATLAB and DUPREX for 50˚C condensing power

Evaporator temperature [˚C]	ct [-]	cth [-]	P_EL matlab [kW]	P_EL duprex [kW]	COP_CD matlab [-]	COP_CD duprex [-]	P_VP matlab [kW]	P_VP duprex [kW]
-20,0	-0,08	0,17	4,00	4,11	2,50	2,43	5,97	5,60
-19,0	-0,07	0,17	3,95	4,06	2,53	2,47	6,00	5,66
-18,0	-0,07	0,17	3,90	4,00	2,57	2,50	6,02	5,71
-17,0	-0,07	0,17	3,85	3,95	2,60	2,53	6,05	5,76
-16,0	-0,06	0,17	3,80	3,89	2,63	2,57	6,07	5,81
-15,0	-0,06	0,17	3,75	3,83	2,67	2,61	6,09	5,87
-14,0	-0,06	0,17	3,70	3,78	2,71	2,65	6,12	5,92
-13,0	-0,05	0,17	3,65	3,72	2,74	2,69	6,14	5,97
-12,0	-0,05	0,17	3,60	3,67	2,78	2,73	6,17	6,02
-11,0	-0,05	0,17	3,55	3,61	2,82	2,77	6,19	6,08
-10,0	-0,05	0,17	3,50	3,56	2,86	2,81	6,21	6,13
-9,0	-0,04	0,17	3,45	3,50	2,90	2,85	6,24	6,18
-8,0	-0,04	0,17	3,40	3,45	2,94	2,90	6,26	6,23
-7,0	-0,04	0,17	3,35	3,39	2,99	2,95	6,29	6,29
-6,0	-0,03	0,17	3,30	3,34	3,03	3,00	6,31	6,34
-5,0	-0,03	0,17	3,25	3,28	3,08	3,05	6,34	6,39
-4,0	-0,03	0,17	3,20	3,23	3,13	3,10	6,36	6,44
-3,0	-0,02	0,17	3,15	3,17	3,18	3,15	6,38	6,50
-2,0	-0,02	0,17	3,10	3,12	3,23	3,21	6,41	6,55
-1,0	-0,02	0,17	3,05	3,06	3,28	3,27	6,43	6,60
0,0	-0,02	0,17	3,00	3,01	3,34	3,33	6,46	6,66
1,0	-0,01	0,17	2,95	2,95	3,39	3,39	6,48	6,71
2,0	-0,01	0,17	2,90	2,90	3,45	3,45	6,50	6,76
3,0	-0,01	0,17	2,85	2,84	3,51	3,52	6,53	6,82
4,0	0,00	0,17	2,80	2,79	3,57	3,59	6,55	6,87
5,0	0,00	0,17	2,75	2,73	3,64	3,66	6,58	6,92
6,0	0,00	0,17	2,70	2,68	3,71	3,73	6,60	6,97
7,0	0,01	0,17	2,65	2,62	3,78	3,81	6,62	7,03
8,0	0,01	0,17	2,60	2,57	3,85	3,89	6,65	7,08
9,0	0,01	0,17	2,55	2,51	3,92	3,98	6,67	7,13
10,0	0,02	0,17	2,50	2,46	4,00	4,07	6,70	7,18

Table 13 Simulation results comparison MATLAB and DUPREX for 50˚C condensing power

Evaporator temperature [˚C]	-20,0	-19,0	-18,0	-17,0	-16,0	-15,0	-14,0	-13,0	-12,0	-11,0	-10,0	-9,0	-8,0	-7,0	-6,0	-5,0	-4,0	-3,0	-2,0	-1,0	0,0	1,0	2,0	3,0	4,0	5,0	6,0	7,0	8,0	9,0	10,0
ctl [-]	-0,08	-0,07	-0,07	-0,07	-0,06	-0,06	-0,06	-0,06	-0,05	-0,05	-0,05	-0,04	-0,04	-0,04	-0,03	-0,03	-0,03	-0,02	-0,02	-0,02	-0,02	-0,01	-0,01	-0,01	0,00	0,00	0,00	0,01	0,01	0,01	0,02
cth [-]	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15	0,15
P_EL matlab [kW]	3,80	3,75	3,70	3,65	3,60	3,55	3,50	3,45	3,40	3,35	3,30	3,25	3,20	3,15	3,09	3,04	2,99	2,94	2,89	2,84	2,79	2,74	2,69	2,64	2,59	2,54	2,49	2,44	2,38	2,33	2,28
P_EL duprex [kW]	4,06	4,01	3,95	3,89	3,84	3,78	3,72	3,67	3,61	3,55	3,50	3,44	3,38	3,33	3,27	3,21	3,15	3,10	3,04	2,98	2,92	2,86	2,81	2,75	2,69	2,63	2,57	2,52	2,46	2,40	2,34
COP_CD matlab [-]	2,63	2,66	2,70	2,74	2,78	2,82	2,86	2,90	2,94	2,99	3,03	3,08	3,13	3,18	3,23	3,29	3,34	3,40	3,46	3,52	3,58	3,65	3,72	3,79	3,87	3,94	4,02	4,11	4,19	4,29	4,38
COP_CD duprex [-]	2,46	2,50	2,53	2,57	2,61	2,64	2,68	2,73	2,77	2,81	2,86	2,91	2,96	3,01	3,06	3,11	3,17	3,23	3,29	3,36	3,42	3,49	3,56	3,64	3,72	3,80	3,88	3,97	4,07	4,17	4,27
P_VP matlab [kW]	6,03	6,05	6,08	6,10	6,13	6,15	6,17	6,20	6,22	6,25	6,27	6,30	6,32	6,34	6,37	6,39	6,42	6,44	6,47	6,49	6,51	6,54	6,56	6,59	6,61	6,64	6,66	6,68	6,71	6,73	6,76
P_VP duprex [kW]	5,60	5,66	5,71	5,76	5,81	5,87	5,92	5,97	6,02	6,08	6,13	6,18	6,23	6,29	6,34	6,39	6,44	6,50	6,55	6,60	6,66	6,71	6,76	6,82	6,87	6,92	6,97	7,03	7,08	7,13	7,18

Annex 5 ZUBADAN heat pump description

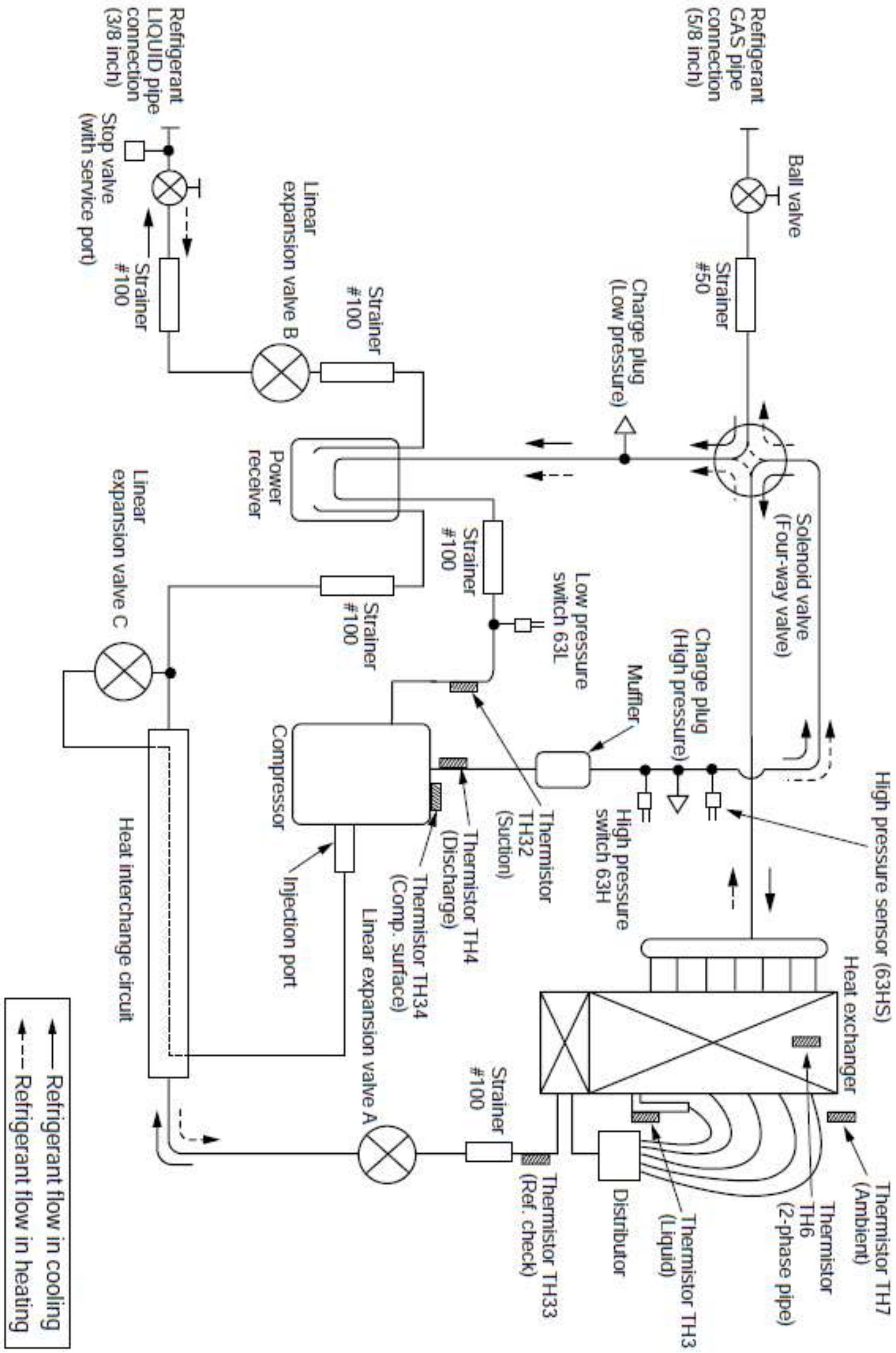

Figure 39 Experimental heat pump external unit hydraulic diagram

Annex 6 Experimental study bench

Table 14 Experimental heat pump unit system components description

System hydraulic diagram		
1.	Evaporator	
2.	Condenser	
3.	Compressor	
4.	Laminating valve	
5.	3 ways valve	
6.	Hot water tank	
7.	Electric heater	
8.	Immersed coil	
9.	Over-pressure valve	
10.	Pressure tank	
11.	Pressure equalization tank	
12.	Circulating pump 1	
13.	Flow meter 1	
14.	Flow meter 2	
15.	Circulating Pump 2	
16.	Floor heating adjustment system 1	
17.	Floor immersed coil 1	
18.	Floor heating adjustment system 2	
19.	Floor immersed coil 2	
20.		
21.		

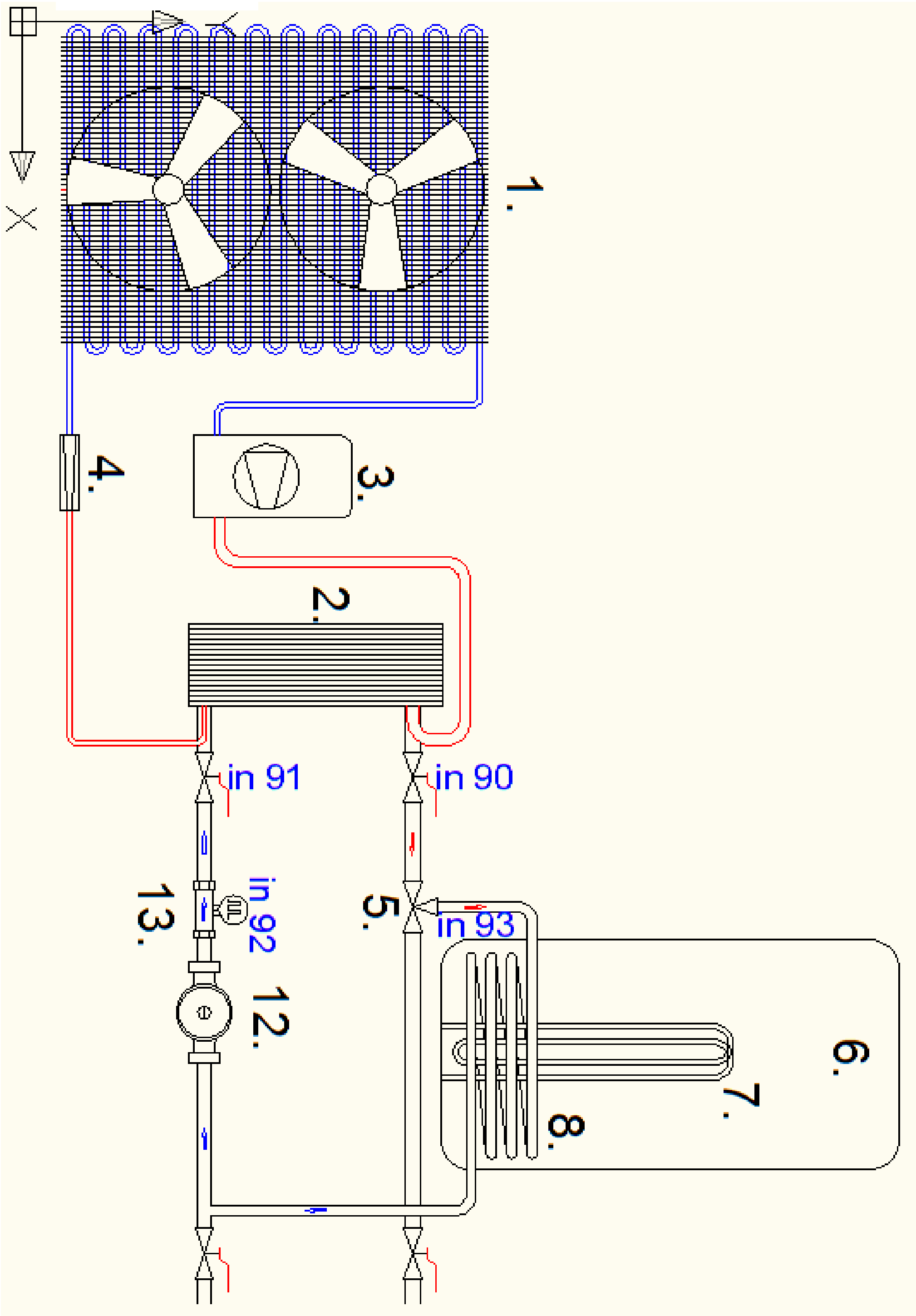

Figure 40 Experimental heat pump unit in DHW configuration

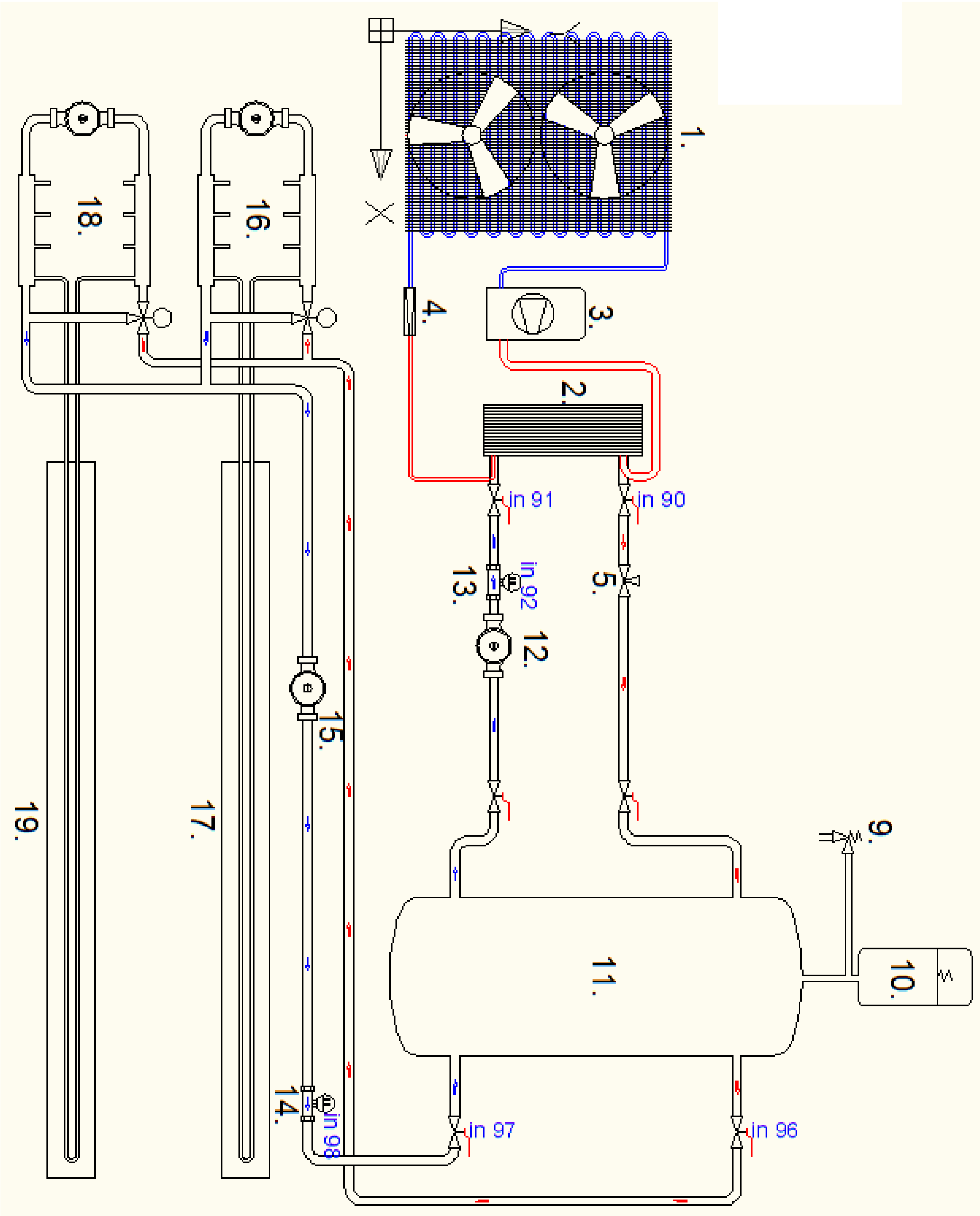

Figure 41 Experimental heat pump in building heating configuration

Annex 7 Experimental study bench presentation

Figure 42 Experimental study bench DHW water tank and three way selection valve.

Figure 43 Experimental study bench heating accumulator with expansion pressure tank

Figure 44 Experimental study bench heating temperature remote control

Figure 45 Experimental study bench heat plate exchanger condenser

Figure 46 Experimental study bench condenser secondary line circulating pump

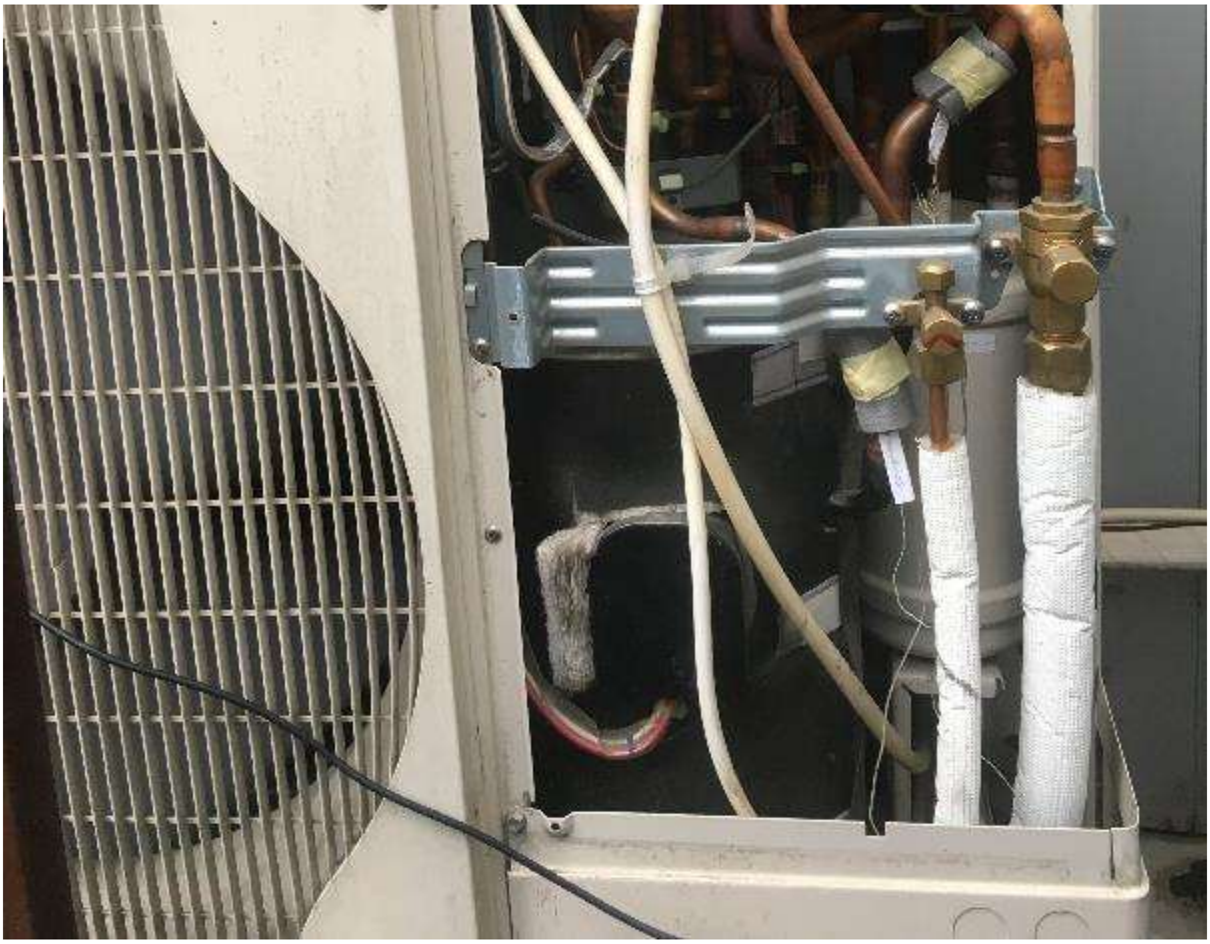

Figure 47 Experimental study bench refrigerant liquid and vapor connection lines

Annex 8 Experimental study bench measurement devices

Figure 48 Experimental study bench current clamps and data acquisition station

SPECIFICATIONS

Current Clamps

Not being powered by batteries these unpowered current clamps are ideally suited to applications where current consumption is being monitored over long periods of time.

Range (1 mV/A)	0.1 to 200 A AC RMS
Accuracy	±2 %, ±0.5 A
Output voltage	1 mV/A
Maximum output impedance	1 kΩ
Maximum conductor size	16 mm
Output connectors	4 mm banana plugs
Operating environment	0 °C to 50 °C
Storage environment	-30 °C to 60 °C
Size in mm (W x L x D)	111 x 50 x 33 mm
Weight	129 g
Standards	IEC1010-1 (1995); EN61010-1 (2001) Category II 600 V Category III 300 V

PicoLog CM3

The table below gives technical details for the PicoLog CM3 data logger.

Number of channels	3
Range (voltage input)	0 to 1 V AC RMS
Accuracy (voltage input)	±1 % (to 200 mV) ±2.5 % (to 1 V)
Overload protection	±30 V DC
Voltage input	1V RMS 20 Hz to 1 kHz
Resolution	24 bit ADC
Reading rate	<1 second per conversion
Input connectors	4 mm socket
PC connection	USB or Ethernet
Dimensions	184 x 135 x 36 mm

Figure 49 PicoLog CM3 data logger and current measurement clumps specification

Figure 50 Almemo 5690-2 system in working process

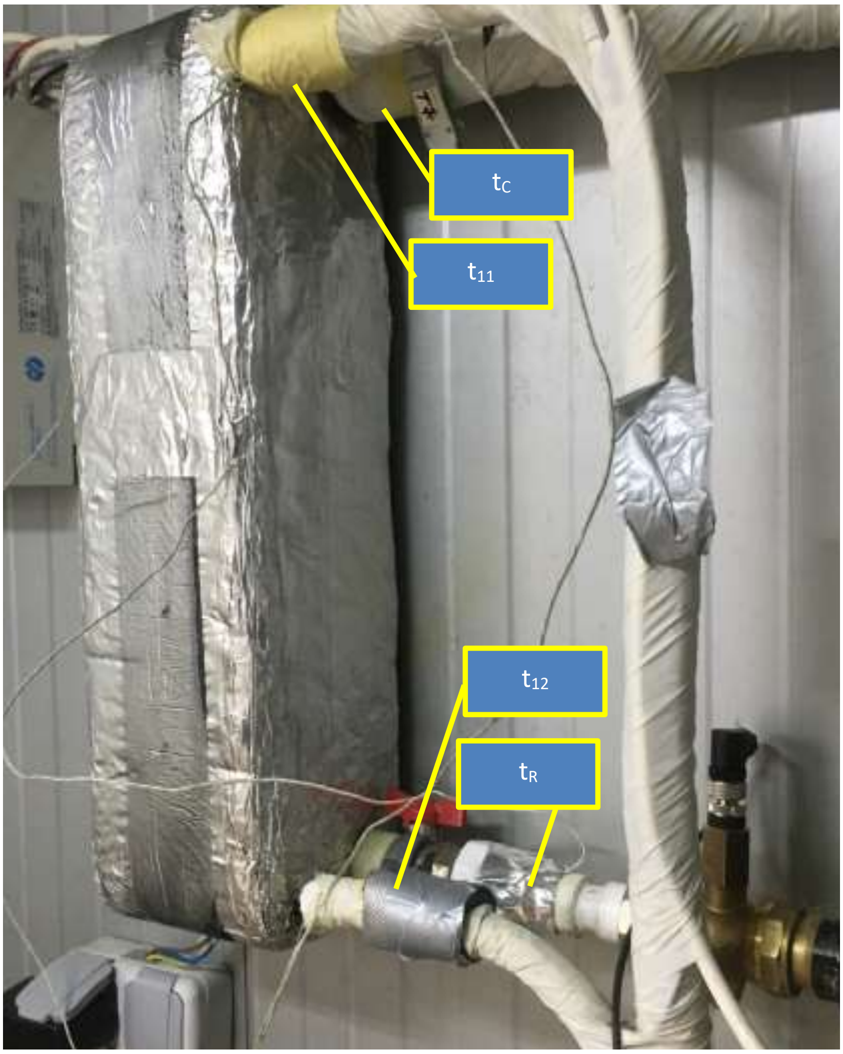

Figure 51 Experimental study bench heat exchanger measurements points

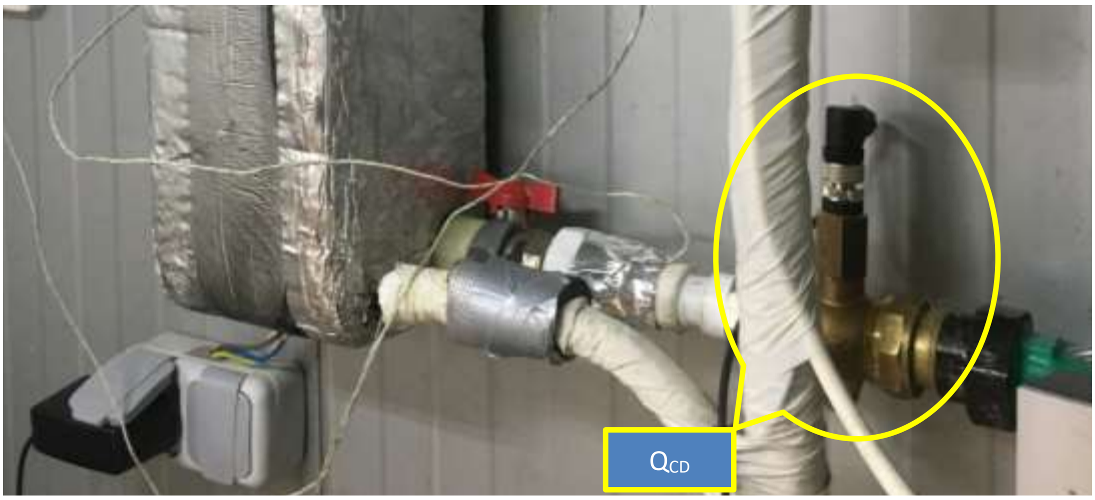

Figure 52 Experimental study bench flowmeter installation

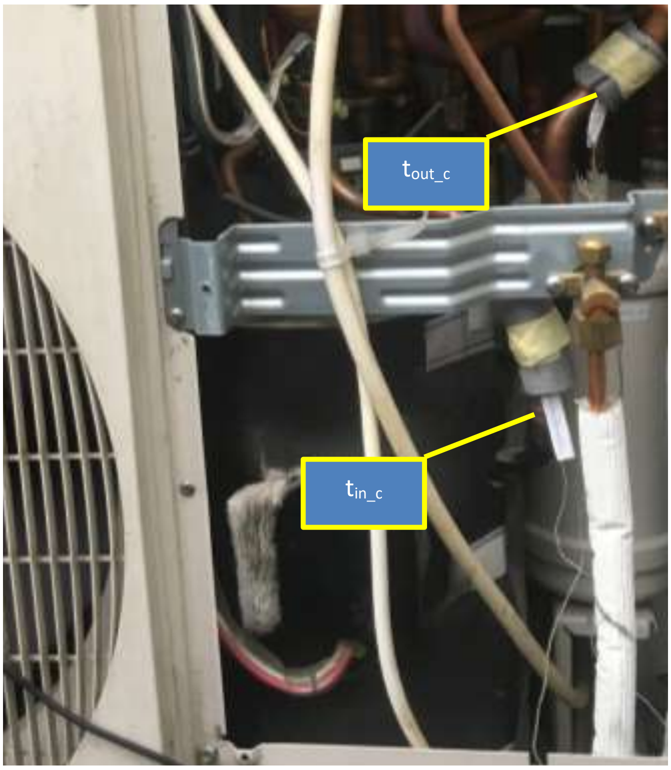

Figure 53 Experimental study bench heat pump compressor suction and discharge plugs temperature

Figure 54 Experimental study bench evaporator output air temperature sensor location

Figure 55 Experimental study calibration bench

Figure 56 Experimental study bench calibration station used for reference sensor

Figure 57 Experimental study bench cold and hot gas lines pressure measurement plugs

Figure 58 Experimental study bench cold and hot gas lines pressure measurement example

Annex 9 Temperature differences evaluation

Manifold pressure and temperature read data

	time	$t_{exterior}$ read data	$t_{exterior}$ calibrated value	Units	$t_{out\_evaporator}$ read data	$t_{out\_evaporator}$ calibrated value	Units	t_{12} exchanger primary output read data	t_{12} calibrated value	Units	t_{11} exchanger primary input read data	t_{11} calibrated value	Units
19:35	19:35:00	9.70	9.18	°C	6.80	9.63	°C	40.70	40.31	°C	61.30	60.78	°C
	19:35:10	9.70	9.18	°C	6.80	9.63	°C	40.70	40.31	°C	61.30	60.78	°C
	19:35:20	9.80	9.28	°C	6.90	9.73	°C	40.70	40.31	°C	61.30	60.78	°C
	19:35:30	9.60	9.08	°C	7.20	10.03	°C	40.50	40.11	°C	61.40	60.88	°C
	19:35:40	9.60	9.08	°C	7.50	10.33	°C	40.40	40.01	°C	61.40	60.88	°C
	19:35:50	9.70	9.18	°C	6.90	9.73	°C	40.20	39.81	°C	61.50	60.98	°C
Manifold data		t_{evap_man}	2	°C	t_{cond_man}	48	°C						
		p_{evap_man}	7.8	Bar	p_{cond_man}	26.5	Bar						
19:40	19:40:00	8.70	8.18	°C	6.80	9.63	°C	38.70	38.31	°C	72.10	71.58	°C
	19:40:10	8.60	8.08	°C	6.90	9.73	°C	42.60	42.21	°C	72.90	72.38	°C
	19:40:20	8.50	7.98	°C	7.20	10.03	°C	44.20	43.81	°C	73.80	73.28	°C
	19:40:30	8.80	8.28	°C	7.00	9.83	°C	44.70	44.31	°C	75.00	74.48	°C
	19:40:40	9.10	8.58	°C	7.00	9.83	°C	44.60	44.21	°C	76.00	75.48	°C
	19:40:50	8.90	8.38	°C	6.90	9.73	°C	44.60	44.21	°C	76.90	76.38	°C
Manfold data		t_{evap_man}	1	°C	t_{cond_man}	54	°C						
		p_{evap_man}	7.5	Bar	p_{cond_man}	32.4	Bar						
19:46	19:46:00	11.70	11.18	°C	8.20	11.03	°C	53.10	52.71	°C	85.90	85.38	°C
	19:46:10	11.30	10.78	°C	8.10	10.93	°C	50.70	50.31	°C	86.10	85.58	°C
	19:46:20	11.20	10.68	°C	8.00	10.83	°C	50.10	49.71	°C	86.20	85.68	°C
	19:46:30	11.30	10.78	°C	8.30	11.13	°C	49.90	49.51	°C	86.60	86.08	°C
	19:46:40	11.10	10.58	°C	8.50	11.33	°C	49.90	49.51	°C	86.90	86.38	°C
	19:46:50	11.30	10.78	°C	8.70	11.53	°C	50.00	49.61	°C	87.20	86.68	°C
Manfold data		t_{evap_man}	1	°C	t_{cond_man}	57	°C						
		p_{evap_man}	7.4	Bar	p_{cond_man}	35	Bar						

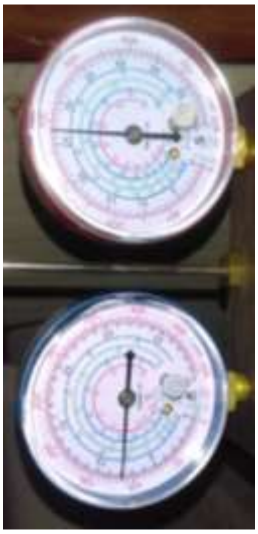

Figure 59 Experimental study bench evaporator and condenser temperature difference evaluation