KTH Industrial Engineering
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Experimental Evaluation Of Large Scale Propane Heat Pump For Space Heating Application

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To my nephews Francesco,
Beatrice and Elisa

KTH Industrial Engineering and Management	**Master of Science Thesis** **EGI 2015:014MSC** **Experimental Evaluation Of Large Scale Propane Heat Pump For Space Heating Application** Salvatore Piscopiello	
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Abstract

A significant part of the environmental impact of a heat pump is generally related to the direct emission of the refrigerant fluid during the life time of the machine. Although the Montreal Protocol has already largely secured its status as a success story by cutting the ozone deplentation refrigerants drastically, we still have to deal with the challenges resulting from climate change.

Natural refrigerants such as Propane permits to design more eco-friendly system without sacrificing the performance of the machine. The thesis work is part of a European project called Next Generation of Heat Pumps working with Natural fluids (NxtHPG) whose primary aim is the development of heat pumps working with Natural refrigerant that are safe, reliable, with high efficiency and high capacity. Royal Institute of technology (KTH) is one of the partner of the project and responsible for the experimental campaign of two large propane heat pumps: an air source heat pump (*Case 1*) and a ground source heat pump suitable for boreholes (*Case 2*). Few cases in literature report on studies about large capacity heat pumps using propane as refrigerant.

This thesis project focuses on the experimental evaluation of the *Case 2* from an energy point view. The safety issues about the use of

flammable refrigerant, i.e propane, are briefly reported; however they are not considered strictly part of the thesis work since they are covered in other phases of NxtHPG project. Experimental tests for *Case 1* were not conducted during this thesis work because of delays in the prototype delivery from the manufacturer. However, concerns about the hydraulic loop of the test rig of both the heat pumps was identified and a new design was suggested and implemented.

During the experimental campaign for *Case 2* a significant number of tests were performed according to a specific text matrix defined from the EN 14285 standard. A simple heat pump model has been used to evaluate the overall performance of the machine from the direct measurements. The heat pump components (compressors, condenser, evaporator, expansion valve) have been analysed in details by defining specific analysis model for each one. The results have been compared with the manufacture expectation.

The prototypes demonstrated to have potentially very good performance, since in the first set of tests the machine behaved as expected by the IMST-ART software. On the other hand, a drop of the unit efficiency and capacity have been registered during the experimental campaign for similar working condition. Two different explanations are investigated to clarify the strange phenomenon. The first hypothesis considers minor internal damage in the compressor, as check valve leakage; consequence of the use of the compressor in a tandem layout. It generates a back-flow in the non working compressor. The second explanation regards inert gas infiltration in the system. On the other hand they are not able to explain entirely the system issues and they need to be confirmed by the manufacturer analysis of the open compressor. The further improvements, proposed and discussed with the manufacture companies involved, can help to solve the question marks on the strange system behaviour during the future work. The next experimental campaign for *Case 1* that will start on April 2015.

The software IMST-ART, used to predict the performance of the heat pump, is demonstrated to be a fast and useful tool. The model of the software for propane as refrigerant and brazed plate heat exchanger predicts adequately the experimental measurement.

Keywords: Heat pumps, propane, tandem compressor, IMST-ART model, hydraulic system design

Sommario

Una parte significativa dell'impatto ambientale relativo a una pompa di calore è legato alle emissioni dirette di refrigerante durante il periodo di vita della macchina. Nonostante l'accordo di Montreal sia già stato considerato come un successo storico per il drastico taglio dei refrigeranti dannosi per lo strato di ozono, bisogna ancora fronteggiare le sfide relative al cambiamento climatico.

Refrigeranti naturali, come il Propano, permettono di realizzare macchine più ambientalmente compatibili senza però sacrificare le prestazioni della macchina stessa. Questo lavoro di tesi fa parte di un progetto Europeo chiamato Next Generation of Heat Pumps working with Natural fluids (NxtHPG), in cui obiettivo primario è quello di sviluppare pompe di calore che lavorano con fluidi Naturali che allo stesso tempo siano sicure, affidabili, con elevata efficienza e taglia.

Royal Institute of technology (KTH) è uno dei partner del progetto e anche responsabile della campagna sperimentale di due pompe di calore a propano di grossa taglia: una pompa ad aria *Case 1* e una geotermica*Case 2*. Sono pochi gli esempi in letteratura che riguardano lo studio di macchine di questo tipo di grande taglia, questo costituisce la novità del lavoro.

Il lavoro di tesi presentato è incentrato principalmente sulla valutazione sperimentale da un punto di vista energetico di *Case 2*. Le problematiche legate alla sicurezza legate all'utilizzo di refrigeranti infiammabili, quale è il propano, sono solo citate; d'altra parte non sono considerate strettamente facenti parte del lavoro di tesi in quando altre fasi del progetto NxtHPG sono incentrate su questo argomento.

Misure sperimentali per *Case 1* non sono state condotte durante il lavoro di tesi a causa di ritardi nella consegna dei prototipi da parte dell'azienda produttrice. Ad ogni modo sono stati individuate problematiche legate ai circuiti idraulici dell'installazione sperimentale di entrambe le pompe di calore e successivamente risolte con l'implementazione di una nuova proposta di circuito idraulico.

Durante la campagna sperimentale di *Case 2*, sono stati svolti un discreto numero di misurazioni secondo una ben specifica *test matrix* derivata dalla normativa europea EN 14285. Un semplice modello di pompa di calore è stato utilizzato per la valutazione delle prestazioni generali della macchina a partire dalle misurazioni dirette. I componenti della pompa di calore, quali compressore, condensatore, evaporatore, valvola di espansione, sono state analizzati separatamente definendo uno specifico modello per ognuno di essi e comparato con i risultati attesi dai produttori.

Il prototipo di *Case 2* dimostra avere ottime potenzialità come prestazioni, in quanto, durante il primo set di test, il comportamento della macchina era in linea con le previsioni del software IMST-ART. D'altra parte, durante la campagna sperimentale, si è evidenziato un forte calo delle prestazioni nonostante le condizioni di funzionamento fossero pressoché simili. Per spiegare questa anomalia, due differenti ipotesi sono state formulate. La prima assume la presenza di di deterioramenti minori nel compressore, come perdite nella valvola di non ritorno (check valve), conseguente ad l'utilizzo in configurazione tandem. Si genererebbe un flusso riverso di refrigerante nel compressore non funzionante. La seconda ipotesi riguarda infiltrazione di gas inerte all'interno della macchina. D'altra parte tali ipotesi non sono in grado di spiegare interamente le problematicità nel sistema e hanno necessità di essere confermate dall'analisi a compressore aperto da parte della azienda produttrice.

Successivi perfezionamento dei prototipi sono stati proposti e discussi con le aziende produttrici coinvolte, e questo permetterebbe di rispondere ai punti di domanda riguardo lo strano comportamento del sistema. La prossima campagna sperimentale per *Case 2* inizierà ad Aprile 2015. Il software IMST-ART, usato per predire le performance della pompa di calore, ha dimostrato essere uno strumento veloce e utile. Il modello implementato nel software per il propano e per gli scambiatori di calore a piatti predice adeguatamente le misure sperimentali.

Keywords: Pompe di calore, propano, compressori tandem, modello IMST-ART, design di sistemi idraulici

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I am using this opportunity to express my gratitude to everyone who supported me throughout my master thesis work performed at the Applied thermodynamics and refrigeration division in the department of Energy Technology, KTH Royal Institute of Technology in Sweden. I am thankful for their aspiring guidance, invaluably constructive criticism and friendly advice during the project work.

Firstly, I would like to express my warm thanks to my supervisor Samer Sawalha and coworker Mr. Willem Mazzotti, for their tireless help and support during all the duration of the work. Secondly, I would like to thank professors José Miguel Corberán and Primal Fernando, for their dedication and illuminating free recommendations
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Contents

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Nomenclature

ΔT_{sc}	Subcooling
ΔT_{sh}	Superheat
$\dot{m}_{ref,dsh}$	Refrigerant mass flow rate, evaluated with a desuperheater heat balance
$\dot{m}_{ref}$	Refrigerant mass flow rate, evaluated with a condenser heat balance
cp_w	Water heat capacity
Q_{con}, Q_{eva}	Heat flux transferred in the condenser and evaporator
Q_{dsh}	Heat flux transferred in the desuperheater
Δh_{is}	Isentropic enthalpy difference
$\Delta p, \Delta p_{nom}$	Generic and nominal pressure drop;
Δp_{con}	Pressure drop in the condenser
$\dot{V}, \dot{V}_{nom}$	Generic and nominal volumetric flow rate;
$\dot{V}_{w,con}, \dot{V}_{w,dsh}$	Volumetric water flow rate trough condenser and desuperheater
$\dot{V}_{displace}$	Total displace volume rate of the compressor

ρ	Fluid density
$cazzo$	Isentropic enthalpy difference
COP_h	Coefficient of performance of the heat pump machine
eta_{is}	Isentropic efficiency of the compressor
$eta_{overall}$	Overall efficiency of the compressor
eta_{vol}	Volumetric efficiency of the compressor
h_1	Refrigerant enthalpy in the common inlet of compressors
h_2	Refrigerant enthalpy in the common outlet of compressors
h_3	Refrigerant enthalpy in desuperheater outlet
h_4	Refrigerant enthalpy in expansion valve inlet
h_5	Refrigerant enthalpy in expansion valve outlet
k	Generic located pressure drop coefficient;
$LMTD$	Logarithmic mean temperature difference
MTD	Mean temperature difference in a condenser or evaporator
p_{con}	Condensing pressure, high pressure stage
p_{eva}	Evaporating pressure, low pressure stage
Q	Heat capacity
T_1	Refrigerant temperature in the common inlet of compressors
T_2	Refrigerant temperature in the common outlet of compressors

T_3	Refrigerant temperature in desuperheater outlet
T_4	Refrigerant temperature in expansion valve inlet
T_{1r}, T_{1l}	Refrigerant temperature inlet of right or left compressor
T_{2r}, T_{2l}	Refrigerant temperature outlet of right or left compressor
T_{3bis}	Refrigerant temperature in condenser inlet
T_{con}	Condensing temperature
T_{eva}	Evaporation temperature
$T_{in,sf}$	Secondary fluid temperature at the inlet of the evaporator
$T_{in,w,con}$	Water temperature at the inlet of the condenser
$T_{out,sf}$	Secondary fluid temperature at the outlet of the evaporator
$T_{out,w,con}$	Water temperature at the outlet of the condenser
u	Velocity of the fluid
UA	UA-value
W_{el}	Electricity consumption of compressors

Chapter 1

Introduction

1.1 Background

Climate change is an alarming, global environmental problem. Melting polar icecaps and glaciers, rising sea levels and entire coastlines under threat are just a few of the issues that we are facing. The atmosphere is responding unmistakeably to human-induced global warming, positive feedback tends to amplified the magnitude of the changes with disastrous damage to the economy and society as a result. The Montreal Protocol and the Kyoto Protocol are two global environmental agreements with very strong impact on the refrigeration industry. Although the Montreal Protocol has already largely secured its status as a success story by cutting the ozone deplentation refrigerants drastically, we still have to deal with the challenges resulting from climate change [1].

In fact part of the environmental impact of a heat pump or a refrigeration unit is related to the direct emission of the refrigerant fluid during the life time of the machine. For these reason, the use of technology with low-(or zero) GWP is necessary to reduce the effect the release of refrigerant will have on the environment.

In some country as Australia, Norway, Denmark, Sweden, Germany the choice of refrigerant to be used is becoming an important economical issue, because of the aggressive "GWP" taxes policy for HydroFluoroCarbon refrigerants (HFC). One solution is to use Natural refrigerant such as CO_2, ammonia or Hydrocarbon refrigerants (HCs). However authorities and enterprises as well as individual technicians and engineers find some barriers to their implementation. Many of these barriers are related to a lack of information and misperceptions about the flammability issue leading to fear and reluctance [2].

While everyone is already using HCs gas in everyday life, e.g. in the kitchen as cooking gas and in refrigerators, in vehicles as an environmentally friendly fuel replacing gasoline, as propellant gas in hygiene products such as deodorants and hairsprays, we do acknowledge that HCs refrigerants must be used safely in all applications including air conditioners considering the implementation of adequate safety measures [2].

In the European contest the project NxtHPG can give a significant contribution filling the gap between the manufacture company and the users in this misperception of the natural refrigerant solution. It can be achieved finding technical solutions, proving that the technology works efficiently and then promote the technology by discussing the safety standards and prove that the technology is safe to use.

1.2 The NxtHPG project

"Next Generation of Heat Pumps working with Natural fluids" (in short NxtHPG) is a European project is the European Union's *Seventh Programme for research, technological development and demonstration*. The primary aim is the development of heat pumps working with HCs and CO_2 that are safe, reliable, with high efficiency and high capacity [3].

The selection of NxtHPG case studies has been considering the possibility of a new fast commercial exploitation of the Natural refrigerants with a deployment of the technology to other sizes, ranges and applications. The idea is to have a practical solution to decrease the environmental impact of the heat pump technology, reaching higher efficiency (10 - 20% SPF improvement) and lower Carbon footprint (20% lower TEWI) compare to the traditional heat pump technologies (HFCs) [3]. In the same time the economic feasibility is taken in account keeping almost the same or just a bit higher (10%) in a way that the better environmental performance clearly compensates for the extra cost;

1.2.1 Partneship and organization

The project is a collaboration of six *RTD* partners which have worked widely in the field of natural refrigerant applied to heat pumps and refrigeration equipment together with six key European industrial partners, among the European leaders in this Sector, all with previous experience and interest in both HCs and CO_2. The industrial consortium includes: two compressor manufacturers: DANFOSS DCC and DORIN; two heat-exchangers manufacturers: LU-VE and ALFA-LAVAL, two heat pump manufacturers: CIAT and ENEX. The RTD working organization is based on three teams: HCs, CO2 and Heat Exchangers (HEs) with different team leaders: KTH, ENEA and EPFL [3]. The complete partner list is available in table 1.1.

Table 1.1: List of the partner involved in the NxtHPG project

RTD Partners		*Companies Partners*	
	Project coordinator UNIVERSITAT POLITECNICA DE VALENCIA | UPVLC	DANFOSS COMMERCIAL COMPRESSORS | DCC	
	ROYAL INSTITUTE OF TECHNOLOGY | KTH	COMPANIA INDUSTRIAL DE APLICACIONES TERMICAS SA | CIAT	
	ITALIAN NATIONAL AGENCY FOR NEW TECHNOLOGIES, ENERGY AND SUSTAINABLE ECONOMIC DEVELOPMENT | ENEA	OFFICINE MARIO DORIN SPA | DORIN	
	UNIVERSITY OF NAPLES FEDERICO II | UNINA	ENEX SRL | ENEX	
	FEDERAL POLYTECHNIC SCHOOL OF LAUSANNE | EPFL	LU-VE SPA	
	NORWEGIAN UNIVERSITY OF SCIENCE AND TECHNOLOGY | NTNU	ALFA LAVAL CORPORATE AB | ALFA- LAVAL	

1.2.2 NxtHPG case studies

During the first phase of the NxtHPG project five different study cases were selected and a prototype was developed for each of the case studies (see figure 1.1). Every case has some particular feature in the technology used, type of heat pump or application.

KTH is responsible for two experimental campaigns which use Propane as fluid: Cases 1 and 2. Both are designed for hot water production to feed space heating system with the possibility to cover a low demand of Domestic hot water (DHW). The principal differences are related to the source type and the nominal heat capacity. *Case 1* prototype is a 40kW air heat pump, so outdoor air is used as source where the heat is extracted from; while *Case 2* is related to a ground source heat pump that use a secondary fluid to extract heat from the ground source.

The prototype in *Case 3* is tested by UPVLC, in Spain, it recovers heat from sewage water or other waste heat with a temperature level around $15 - 25^oC$. It is used forDHW production. Finally ENEA is in charge of the two CO_2 air-to-water heat pumps: *Case 4* for production of DHW and *Case 5* for high temperature space heating with a particular target on the renovation market for the replacement of old gas boiler [4].

<table>
<tr><th>Case</th><th>Fluid</th><th>Source</th><th>T(ºC)</th><th>Sink</th><th>T(ºC)</th><th>Application</th><th>(kW)</th></tr>
<tr><td rowspan="2">1
(KTH)</td><td rowspan="2">HC
(Propane)</td><td rowspan="2">Air</td><td rowspan="2">-10 to 35
(outdoor air)</td><td rowspan="2">Water</td><td>40 to 50</td><td>Heating Water production</td><td rowspan="2">40</td></tr>
<tr><td>60</td><td>Low demand
of Domestic hot water</td></tr>
<tr><td colspan="8">CASE 1 is an air to water heat pump for the production of hot water for heating applications also covering a low demand of domestic hot water with the use of a de-superheater. The unit will be reversible on the refrigerant circuit, so providing heating and cooling. An extra water-air hydraulic loop is going to be considered in order to have an air-to-air heat solution.</td></tr>
<tr><td rowspan="2">2
(KTH)</td><td rowspan="2">HC
(Propane)</td><td rowspan="2">Water (brine) from a ground coupled heat exchanger</td><td rowspan="2">-5 to15</td><td rowspan="2">Water</td><td>40 to 50</td><td>Heating water production</td><td rowspan="2">60</td></tr>
<tr><td>60</td><td>Low demand
of Domestic hot water</td></tr>
<tr><td colspan="8">CASE 2 is a geothermal heat pump for the production of hot water for heating applications also covering a low demand of domestic hot water with the use of a de-superheater. It will be reversible on the refrigerant circuit, so providing heating and cooling.</td></tr>
<tr><td>3
(UPVLC)</td><td>HC
(Propane)</td><td>Water
(Neutral loop)</td><td>10 to 15 (Sewage water)
or
25 to 30 (Condensation loop)</td><td>Water</td><td>60</td><td>Domestic
hot water production</td><td>50</td></tr>
<tr><td colspan="8">CASE 3 consists of a heat pump booster from a neutral water loop, (10-30 ºC) (recovery of waste heat from condensation (25-30ºC) or sewage water (10-15ºC)) up to 60ºC for domestic hot water production.</td></tr>
<tr><td>4
(ENEA)</td><td>CO2</td><td>Air</td><td>-10 to 10 (winter)
20-35 (summer)
(outdoor air)</td><td>Water</td><td>60 (up to 80)</td><td>Domestic
hot water production</td><td>30</td></tr>
<tr><td colspan="8">CASE 4 is an air to water heat pump for hot water production at 60ºC or up to 80ºC for high temperature applications.</td></tr>
<tr><td>5
(ENEA)</td><td>CO2</td><td>Air</td><td>-10 to 35
(outdoor air)</td><td>Water</td><td>80
(return water 40)</td><td>Heating &
DH water production
(DHW in summer)</td><td>50</td></tr>
<tr><td colspan="8">CASE 5 is an air to water heat pump for heating applications. It targets the renovation market for the replacement of old gas boiler heating systems (5-6 family houses) with high temperature radiators as terminal units. Main role is hot water production for heating but it must also provide DHW all along the year. Therefore, the development will be targeted for winter operation, although the unit will be also used during summer for DHW production.</td></tr>
</table>

Figure 1.1: Specifications of heat pump case studies in NxtHPG project

1.3 Role of the Thesis Work in the NxtHPG project

The work presented in this Thesis is part of the overall NxtHPG project that includes different partners and different type of activities: from the modelling to the prototypes and design construction; from the experimental campaign to the management. The thesis focus on KTH tasks that belongs to the *1st experimental campaign* mainly on the ground source heat pump of Case 2 and in small part on the air heat pump of Case 1. The data has been processed and widely analysed producing useful information needed during the 6th and 7th stages of the project (see table 1.2): *analysis of the Seasonal performance factor (SPF)* and *prototypes improvement.*

Table 1.2: NxtHPG project phases

Project Stage	Description	
1	Analysis of applications and case studies definition	Before the thesis work
2	Modelling of heat pumps and systems	
3	HP prototypes design	
4	HP prototypes construction	
5	1st experimental campaign	Thesis work involved
6	Analysis of seasonal performance and initial assessment	
7	Improvement of HP prototypes design and construction	
8	2nd experimental campaign and control optimization	Thesis work not involved
9	Design Guidelines. Final assessment of results	
10	Dissemination and exploitation of results	
11	Management	

Chapter 2

Propane Heat Pump

The fundamental information about propane heat pump are presented in this chapter. It consist of three main part: the characteristic of propane as refrigerant, the application of propane in the refrigerant industry and the safety issued about hydrocarbon usage. Several literature sources are presented to give an overview of the topic.

2.1 Propane as refrigerant

The suitability of a refrigerant is related to different factors. They can be divided in two main categories: property-based and market factor [5]. The first group include:

- Global warming potential
- Ozone depleting potential
- Flammability
- Toxicity
- Chemical reactivity and material compatibility
- Potential efficiency (thermophysical properties)

The market issues are:

- Refrigerant cost
- Cost of components
- Access to refrigerant
- Availability of suitable components, oils, service equipment
- Adequate expertise and training

Propane (R290) and other HCs have been reconsider recently and exploited commercially as alternative refrigerants for refrigeration and heat pump and A/C applications thanks to their generally good suitability factor.
Compared with CFCs, HCFCs, and HFCs, HC refrigerants has zero ODP and greatly low GWP and the performance are at a similar level compared to the traditional refrigerant. In addition they offer high compatibility with the traditional refrigeration equipment and good miscibility with mineral oils. The latter aspect guarantees the oil return in the compressor, on the other hand since the propane solved in oil decrease the viscosity, higher viscosity oil needed to be used to guarantee the right level of lubrication in the compressor [6]. The refrigerant cost is generally not so high compare to synthesised freon, on the other hand the quality level required is much higher compared to other hydrocarbon application [7] and such extra cost for purification has to be considered.

The real issue about the application of HC refrigerants is related to the safety concerns in handling relatively large amounts of flammable fluids. Propane has “lower toxicity” and “higher flammability”, giving them a Group A3 classification in ISO 817 [5]. Nowadays the high environmental friendly fracture of hydrocarbon refrigerants determine a strong driver for the international standard in the refrigeration and heat pump equipment to provide the necessary additional safety measures for the design, repair and servicing of equipment using flammable refrigerants [6].

2.2 Application of propane and HCs

The application of propane as refrigerant has been studied widely in literature. Most of the application are related to retrofit of R22 system: window type air conditioners (AC), milk cooling units, split type AC, and a central chiller AC system [8] are few examples. On the other hand R-290 has been used for new system as well for very different temperature level of the heat source/sink that generally required a minimum charge quantity: bottle/beer cooler, ice-cream freezer, dehumidifier and small heat pump [**13**]. One of the most important market integration of the HCs technology are related to the refrigerators and freezers with isobutane, introduced in response to a market demand on the European market already in 1993. Nowadays most of domestic refrigerator are built using this technology.

In literature the field of relative large heat pump is generally not widely investigated, a 100 kW heat pump using low charge heat exchangers is discussed by A. Cavallini [**14**].

2.3 Safety using heat pump with hydrocarbon

The main concern about the use of hydrocarbon in heat pump are related to the flammability of the refrigerant since propane and other hydrocarbons are in the A3 group. The safety requirements for refrigerant charge are introduced in the safety standards *IEC 60335 2 40:2013* from the International Electrotechnical Commission and the European normative *EN 378 1:2008+A2:2012*.

They permit to define the maximum charge in the system. In case of solutions for medium charge heat pumps (to 40.0..60.0 kW), the increase of the heat capacity need an higher amount of refrigerant. According to the standard the maximum refrigerant charge can be increased to 4.94 kg for propane, when the mechanical ventilation is applied for room or separate enclosure in the room [9].

For a complete dissertation of the safety issue, equipment design, management and training about the use of hydrocarbon refrigerants, please refer to the literature about the topic [10].

Chapter 3

Experimental Setup

As mentioned in the description of the NxtHPG project in chapter 1, KTH is one of the chef R&D partner for the HCs team and it is the the responsible for building the experimental rigs and testing the propane heat pumps for cases 1 and 2. All the experimental setup has been built on a roof-top at the Energy technology department at KTH, this is due to safety reason involving flammability of propane as a refrigerant.
Despite some common features in the experimental configuration, the two prototypes have different layout, related to the different nature of the heat source; i.e. air or ground. Following sections details the features of experimental setup and the heat pump prototypes will be described.

3.1 Test rig description - CASE 1

CASE 1 is a 40 kW capacity air-to-water heat pump for the production of hot water for space heating applications and at the same time it covers a low demand of DHW. The unit has been designed reversible on the refrigerant circuit, to provide cooling in summer.

This section will give all the details about the experimental setup of the air HP in CASE 1, going through the general description of the HP prototype and the whole test rig. It includes the test rig modifications suggested during the thesis work, as consequence of the new hydraulic design whose a complete dissertation is presented in paragraph 4.4.1.

3.1.1 General scheme

Figure 3.1 presents a general schematic of the test rig for CASE 1, full-detailed drawing are available in the appendix D. The system consists of a different number of separated loops that permits to transfer the heat in the different portions of the system, the loops are clearly indicated in the figure with different colors. The core of the system the heat pump unit, is indicated with green. In addition to this, several types of system (ventilation and hydraulic) are needed to simulate and control the boundary conditions under which the heat pump has to operate. Particularly a room has been built in which the heat pump will be installed and where the temperature in the room itself can be controlled to meet the required operating conditions. The different loops, analysed separately in the next section, are indicated in the list below.

- Refrigerant loop;
- Climate chamber;
- Condenser hydraulic circuit;
- Desuperheater hydraulic circuit;
- HX (or dry cooler) hydraulic loop, supplying heat to climate chamber;
- Hydraulic loop for tank cooling

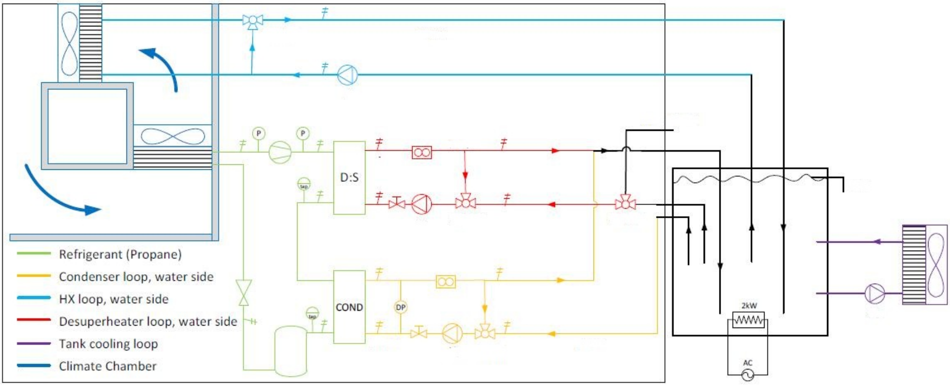


Figure 3.1: General and schematic diagram of test rig CASE 1

Refrigerant loop

The refrigerant loop; i.e. heat pump, uses a single stage prototype compressor, specially modified to be used with propane from Danfoss. During heating mode of the heat pump, the hot compressed refrigerant flows in the desuperheater (D:S) where it discharges its heat to water for the production of DHW, where higher temperature is required. The propane vapour, near to saturated condition, goes inside the condenser and all the condensation heat is transferred to the water in the condenser hydraulic loop, that simulated the condition required for heating application. The subcooled liquid propane going through the thermostatic expansion valve decreasing its temperature and pressure. The two-phase mixture absorbs the heat from the air in the climate chamber, at the expense of the evaporation of the liquid. The superheated vapour is compressed again by the compressor, starting again the cycle.

The 4-ways valve (not included in the figure) permits to the refrigerant circuit to let the machine working in cooling mode.

Climate chamber

The climate chamber is a closed volume where the air temperature and humidity can be controlled. As can be observed in 3.1 the climate chamber of CASE 1 works as a closed loop, where the air is recirculated. The heat pump's evaporator cools down the air, which is then heated up again to a desired temperature by a water/air heat exchanger connected with relative warm water from the tank. The structure of the climate chamber is made of insulating and light material. It permits the structure to collapse in case of ignited leakage of propane; so it will not lead to sever explosion.

Condenser circuit

The condenser loop facilitates simulating the temperature conditions of water supplied to and returning from the space heating system. Relative cold water is taken from the water tank and mixed with warm water coming from the exit of the con-denser using a three way valve. The loop has a differential pressure drop meter on the water side of the condenser and a magnetic flow meter measuring the water flow. During the start-up, the water in the circuit is quite cold and it needs to be heated up. The three-way valve remains fully closed, creating a recirculation of water through the condenser. In this way, all the heat rejected by the condenser is used to increase the water temperature until the desired level is reached and the three-way valve starts opening. The temperature at the outlet of the condenser is controlled with a proper mix of warm and cold water from the condenser itself and the tank, respectively. The temperature difference can be controlled by regulating the water flow by changing the speed of the water pump.

Desuperheater circuit

The desuperheater loop function is to control the conditions in-/out of the desuperheater for the production of DHW or other high temperature application. This loop is similar to the condenser one, but an additional three way valve is used to decrease the temperature of the water from the tank by mixing with tap water. The pump in the loop is activated only during desuperheater mode on of the heat pump.

HX circuit (dry cooler)

Since the air in the climate chamber is cooled down by the heat pump's evaporator, it needs to be heated up again to maintain certain room temperature. This energy is provided by the water-in-tank through a heat exchanger (HX loop). The warm water in the tank is pumped in the circuit into the water-air heat exchanger providing the needed heat to the room. A three way valve is installed to regulate the water temperature and the amount of heat rejected by the HX.

Hydraulic loop for tank cooling

The tank is the component in the test rig that helps stabilizing the working condition with its thermal inertia. It works as heat supplier and receiver for all the hydraulic loops in the test rig. Although the tank is simultaneously used as heat source and sink for the evaporator and condenser, respectively; there is a need to cool it down. Indeed, the condensation process rejects more heat than the evaporation absorbs; therefore, the water temperature in the tank will keep increasing if not cooled down. Therefore an additional water loop is needed to cool down the tank water through a water-air heat exchanger, the extra heat is transferred to the ambient air. A three way valve regulates the flow into this exchanger to match the tank temperature set point. Since the tank is located outdoor, it is equipped with an electrical heater to avoid the temperature of the water dropping too low during winter time.

3.1.2 Test rig updates

The experimental experience of the similar test rig, i.e. CASE 2, had shown some problems about the low flow rate in some of the hydraulic systems for the first installing scheme, *version A* (scheme reported in the appendix in figure D.1). Taking the advantage of CASE 1 hydraulic loop was not already installed, the test rig design has been modified and updated to *version B* to avoid facing similar experimental problems as in CASE 2. The components, whose changes are related to, are reported in red in figure D.2 in the appendix.

The test rig modification are connected to three main actions and they involved the hydraulic system:

- Sizing up the pipes from Ø 33mm (1”) to Ø 42mm (1¼”);
- Sizing up the three way valve from DN 25 to DN 40 and install it in mixing configuration;
- Sizing up the pump from MAGNA3 32-100 (3a) to MAGNA3 32-120F (1b).

None of the two *version A* and *B* where finished to be installed during the thesis project. HP prototype delivery was postponed compare to the initial plan of the NxtHPG project because of delay in the manufacturing stage.

3.2 Test rig description - CASE 2

CASE 2 is a geothermal heat pump for the production of hot water for heating applications also covering a low demand of DHW with the use of a desuperheater. Cooling in summer can be provided thanks to the possibility to let reverse the refrigerant circuit.

This section will explain the details about the experimental setup of the Ground source HP (GSHP) in CASE 2, going through the general design of the HP prototype, the initial configuration of the test rig (*version A*) and the further modifications implemented during the thesis work (*versions B* and C).

3.2.1 General scheme

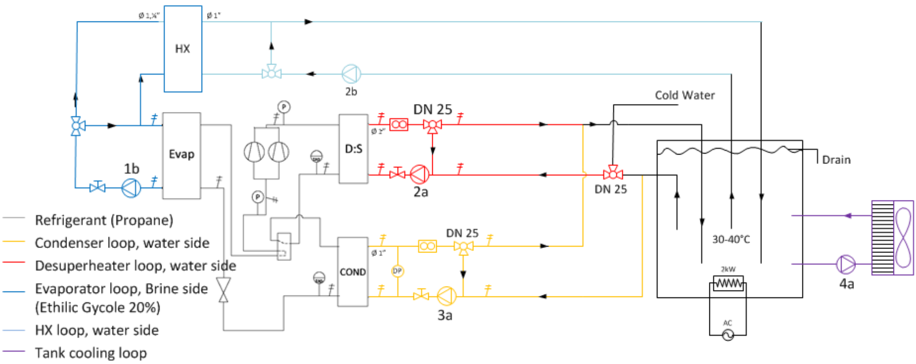


Figure 3.2: Schematic diagram of test rig CASE 2

The detailed drawing of the system are available in the appendix D, on the other hand the general schematic of the test rig for CASE 2 is shown in Figure 3.2 where the different separated circuit are clearly underlines with different colors.

The layout is similar to CASE1 (see figure 3.1). The main differences are related to the compressor installation and the heat source of the HP, air of the climate chamber in CASE 1 and the hydraulic secondary fluid circuit for CASE 2 (dark blue). The proper HP prototype is indicated in green and gather together all the components in which the refrigerant circulates. All the other hydraulic loops are needed to simulate and control the boundary conditions under which the heat pump has to operate. The different loops are listed and analysed separately in the next section. Here is a short list:

- Refrigerant loop;
- Condenser hydraulic circuit;
- Desuperheater hydraulic circuit;
- Evaporator hydraulic circuit;
- HX loop, supplying heat to evaporator loop from the tank;
- Hydraulic loop for tank cooling

Refrigerant loop

The refrigeration cycle in essentially the same of the one analysed for CASE 1 (paragraph 3.1.1). On the other hand, in this system, two scroll compressors work in tandem and can operate separately, reducing the capacity of the HP (half mode). It helps for a better match of the load compare to the use of single large unit. The presence of a 4-way valve permits to reverse the unit, switching functions of the evaporator and the condenser to produce chilled water for cooling.

Condenser hydraulic circuit

The condenser loop facilitates simulating the temperature conditions of water supplied to and returning from the space heating system. The system is similar to the corresponding hydraulic system in CASE 1, where a brief description can be found in section 3.1.1.

Desuperheater hydraulic circuit

Desuperheater hydraulic circuit simulates the management of a small demand of DHW, the full description of the loop, essentially similar to CASE 1, is presented in paragraph 3.1.1

Evaporator hydraulic circuit

The evaporator loop facilitates simulating the temperature condition on the cold side of the heat pump. A water solution with 20%-vol of Ethylene Glycol is used as secondary fluid. It acts as the secondary fluid used in the borehole of the GSHP that exchanges heat with the ground in a real system. In the test rig this heat exchange is simulated with a well-defined condition of the secondary fluid temperature and heat is provided by an additional HX using relative warm water coming from the tank (HX loop). A three way valve permits to regulate the exit brine temperature in the evaporator to a given set point.

HX loop

The HX loop for CASE 2 is similar to the one in CASE 1 where the secondary fluid to water heat exchanger is replaced with the water/air one. Since the secondary fluid is cooled down by the heat pump's evaporator, it needs to be heated up again to maintain the right temperature range to simulate the energy transfer with the ground. Relatively warm water from the tank is pumped in the circuit into the water/brine heat exchanger and rejected in the tank. A three way valve is installed to regulate the water temperature and avoid the risk of freezing in the heat exchanger. This is not a main control device but rather a tool providing more stability to the control system.

Hydraulic loop for tank cooling

The system is the same as presented for CASE 1 in section 3.1.1

3.2.2 Heat pump components

Compressor

Danfoss Commercial Compressors is the manufacturer in charge of providing compressor for the heat pump in CASE 2. They are two scroll compressor prototypes modified from the PSH038 model for R410A. The choice was based on its relatively high overall and volumetric efficiencies. The swept volume flow rate of this compressor is 29,6 m3/h at 2900 rpm, the capacity of this heat pump is about 47 kW and there will be two compressors working in tandem [11]. The heat pump prototypes and a detail on the compressors are shown in figure 3.3a.

Condenser and Evaporator

The condenser and the evaporator are provided by the manufacturing company ALFA-LAVAL, two brazed plate heat exchangers. Counter current flow permits to achieve a good value in the efficient heat transfer with a compact design. The specific model selected is AC220EQ-76AM-F for both evaporator and condenser, as final decision of the design campaign in the project [11]. The only difference is releated to the secondary fluid that passes through the heat exchanger: water as heat sink for the condenser and 20% Ethylene Glycol as heat source for the evaporator. When the heat pump works in cooling mode, the evaporation and the condensation happen in reversed location in the components. Both the heat exchanger are insulated (see figure 3.3b).

Expansion valve

The HP is equipped with a thermostatic expansion valve, model TGEX TR 15 manufactured by Danfoss. It can be used for several refrigerant, propane included. On the other hand the gas inside the bulb is R407C. CIAT, the heat pump manufacture responsible to assemble the prototype, planned to install an other thermostatic valve more suitable for propane, but is could not because of delays in the supply.

(a) front (compressors in the foreground)

(b) side (condenser on the left)

Figure 3.3: CIAT HP prototype of CASE 2

3.2.3 Instrumentation

The complete list of all the experimental measurement are reported in table 3.1. They include T-type thermocouples provided at necessary locations for temperature measurements; flow meters are installed in the warm hydraulic circuits; pressure transducers connected at the compressor inlet and outlet for pressure measurements. Other measurement equipments includes a differential pressure transducer to check the pressure drop in the water side of the condenser and a power meter for electricity consumption measurement of the compressors. All measurements are recorded on a computer via a data logger.

On the other hand several automatic control 3-way valves are provided at the hydraulic circuit and they arrange the flow distribution in the circuit to get a certain set point temperature. The most important are related to:

- Condenser circuit, controlling $T_{out,w,con}$
- Evaporator circuit, controlling $T_{in,sf}$
- Desuperheater circuit, controlling $T_{out,w,sh}$

Table 3.1: List of the measurement equipment and the parameters measured in CASE 2

Equipment	**Measured Parameter**	
	Symbol	*Description*
Pressure transduser x2	p_{con}	High pressure stage, compressor outlet
	p_{eva}	Low pressure stage, compressor suction
T-type Thermocouples x10 (x15) [a]	T_1	Compressors inlet
	T_2	Compressors outlet
	T_{1r}	Right compressor inlet[a]
	T_{1l}	Left compressor inlet[a]
	T_{2r}	Right compressor outlet[a]
	T_{2l}	Left compressor outlet[a]
	T_3	Desuperheater outlet
	T_{3bis}	Condenser inlet[a]
	T_4	Expansion valve inlet
	$T_{in,w,con}$	Inlet water in the condenser
	$T_{out,w,con}$	Outlet water in the condenser
	$T_{in,w,dsh}$	Inlet water in the desuperheater
	$T_{in,w,dsh}$	Outlet water in the desuperheater
	$T_{in,sf}$	Inlet of the secondary fluid in the evaporator
	$T_{out,sf}$	Outlet of the secondary fluid in the evaporator
Magnetic Flow meter x2	$\dot{V}_{w,con}$	Volumetric flow of the water thought the condenser
	$\dot{V}_{w,dsh}$	Volumetric flow of the water thought the desuperheater
Differential pressure transducer	$\Delta p_{con,w}$	Pressure drop in the condenser, water side
Power meter	W_{el}	Electricity consumption of the compressors

[a] Introduced in *version B* of the experimental setup

3.2.4 Test rig modification

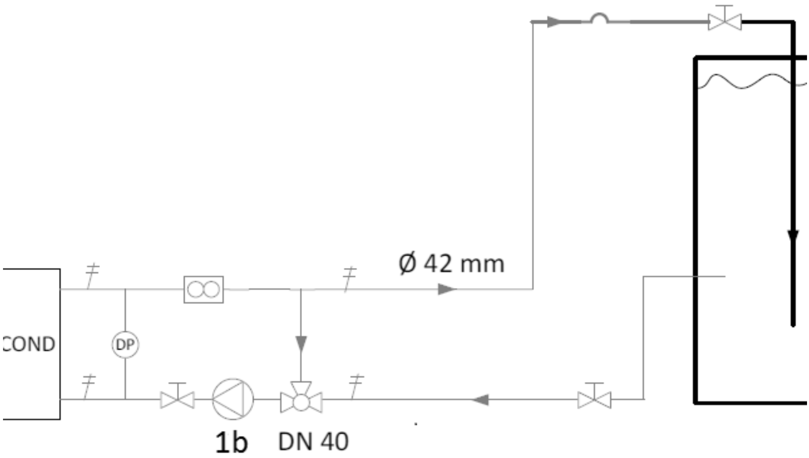
The preliminary test in the test rig, had shown some problems about the low flow rate in some of the hydraulic system for the first installing scheme, *version A*. It would determines the impossibility to get the condition in the test matrix of section 3.3 when both compressors work (full mode). Additional modifications, not releated to the latter problem, are the installation of new thermocouples in the refrigerant loop improving the quality of information about the system and other minor features. The sum of all these changes are implemented in the experimental setup and indicated as *version B*.

The components, whose changes are related to, are reported in red in figure D.6. This permits an easy comparison with the *version A* of the scheme D.5. Both the figures are reported in the appendix D. The test rig modifications connected to the condenser and evaporator hydraulic system are (see figure 3.4):

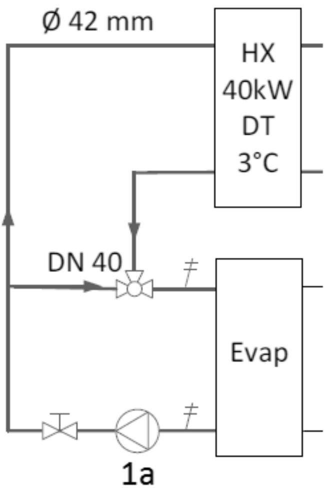
- Sizing up the pipes from Ø 33mm (1”) to Ø 42mm (1¼”);
- Sizing up the three way valve from DN 25 to DN 40 and install it in mixing configuration;
- Sizing up the pump from MAGNA3 32-100 (3a) to MAGNA3 32-120F (1b) (condenser circuit);
- Sizing up the pump from MAGNA3 32-120f (1b) to MAGNA3 40-180f (3b) (evaporator circuit).

In the refrigerant loop and in the tank:

- Installation of thermocouples on the suction and discharge of each compressor;
- Installation of thermocouple at the condenser inlet;
- Installation of the electric heater in the tank.



(a) condenser



(b) evaporator

Figure 3.4: Details of test rig changes in the hydraulic circuit

3.2.5 Installation details

Figures 3.5 and 3.6 are two panoramic photos reporting the actual installation. The heat pump and all the test rig are placed in a small green house that permit to create a protection for the system from the outdoor condition (figure 3.5, in the middle). On the left and on the right of the same figure it is possible to notice the drycooler and the tank respectively. They are part of the hydraulic loop for tank cooling.
The inside of the small green house is shown in figure 3.6, with a specific view on all the hydraulic system of the test rig. The heat pump is not included in the framing, on the other hand, on the left, it is possible to noticed the six different hydraulic pipes coming from the heat exchangers of the HP machine. The first and the second are part of the evaporator loop, the two pipes in between are related to the condenser and the last two are connected to the desuperheater.

Figure 3.5: Outdoor view of the experimental setup of CASE 2 installed on the roof

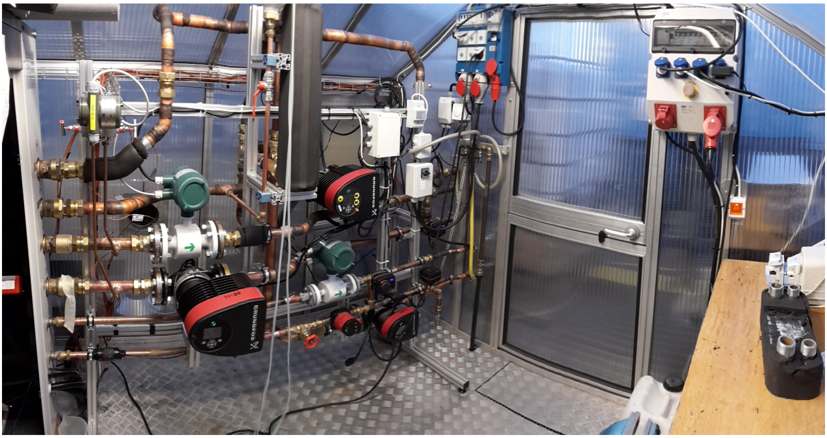


Figure 3.6: Outdoor view of the experimental setup of CASE 2, detail on the hydraulic loops

3.3 Test Matrix

All the tests of the experimental campaign has been performed according to specific boundary condition imposed on the HP. They were selected during an early stage on the NxtHPG project according to the the European Normative EN 14285 and EN 14511-2. On the other hand, some other tests have been selected according to EU Reg. 811/2013, but they were considered as secondary priority in the project. The same consideration for cooling mode tests. All the experimental campaign presents test according to the EN 14285 and EN 14511-2, only in cooling mode. Tables 3.2 and 3.3 show the temperature values with the desuperheater deactivated and activated respectively. They are related to the the secondary fluid in the heat exchangers. The mass flow obtained imposing the temperature of all inlet and outlet in the first test, need to be maintained constant for all the same type tests.

Table 3.2: Test Matrix for CASE 2, heating mode, desuperheater deactived

Evaporator		**Condenser**		**Normative**
Sec. Fluid Inlet $T_{sf,in}$ *[$^o C$]*	*Sec. Fluid Outlet* $T_{sf,out}$ *[$^o C$]*	*Water Inlet* $T_{w,in,con}$ *[$^o C$]*	*Water Outlet* $T_{w,out,con}$ *[$^o C$]*	
0	-3	40	45	EN 14285 (medium temperature)
0	fmf [a]	fmf	37	
0	fmf	fmf	33	
0	fmf	fmf	28	
-5	fmf	fmf	45	EN 14511-2 (medium temperature)
5	fmf	fmf	45	
-5	fmf	fmf	55	EN 14511-2 (high temperature)
5	fmf	fmf	55	
0	-3	47	55	EU Reg. 811/2013 [b]
0	fmf	fmf	52	
0	fmf	fmf	42	
0	fmf	fmf	36	

[a] fmf = fixed mass flow
[b] secondary tests, not tested

Table 3.3: Test Matrix for CASE 2, heating mode, desuperheater activated

Desuperh.	**Evaporator**		**Condenser**		**Normative**
Water Inlet $T_{w,in,dsh}$ *[^{o} C]*	*Sec. Fluid Inlet* $T_{sf,in}$ *[^{o} C]*	*Sec. Fluid Outlet* $T_{sf,out}$ *[^{o} C]*	*Water Inlet* $T_{w,in,con}$ *[^{o} C]*	*Water Outlet* $T_{w,out,con}$ *[^{o} C]*	
15 35 45	0	-3	40	45	EN 14285 (med. temp.)
15 35 45	0	-3	47	55	EU Reg. 811/2013

Chapter 4

Preliminary study

4.1 First tests

After the installation of the experimental setup, several preliminary tests were performed to study and optimize the performance of the system. The work was focused on three different aspects:

- Optimization of the amount of refrigerant in the system.
- Amount of superheating of the vapour at the inlet of the compressors, regulating manually the thermostatic valve;
- Test matrix conditions matching in the hydraulic loops with the references condition defined in section 3.3.

4.2 Optimization of the charge

The charge in the system affects the value of subcooling achieved in the condenser and as consequence the general performance of the system. Too high subcooling means that large part of the condenser is filled with liquid and since the heat transfer in two-phase flow is generally much higher that in single phase, it would decreases the the UA-value in the condenser. On the other hand, too low subcooling would increase the risk of vapour bubble going though the expansion valve. The optimal subcooling has been consider to be around $\Delta T_{SC} = 4 - 5K$, and since the system doesn't have a receiver it is expected that the amount of propane needed to get the same subcooling is different depending on the number of compressors working. A sight glass after the condenser has been installed to check the presence of vapour bubble in the system.

The experimental charge is $\sim 2,5kg$ and $\sim 2,7kg$ for full and half mode respectively. An explanation of this phenomenon was that accumulation of liquid propane could occur in the non-working compressor, the detailed hypotesis is presented in the compressor section in the componets analysis (7.2).

It is important to mention that the presence of bubble was registered during all the experimental campaign, with high value of subcooling as well (up to $\Delta T_{SC} = 30K$).

4.3 Superheat adjustment

The value of superheat can be changed through the regulation of the spring inside the thermostatic valve. The optimal superheat value has been consider to be at least $\Delta T_{SH} = 5K$. For this reason the regulation has been done in the condition where we expected the lowest superheat: low condenser and evaporation temperatures. The HP boundary condition that satisfied these condition are $T_{sf,in} = -5^oC$ and $T_{w,out,con} = 28^oC$ according to the test matrix in table 3.2.

4.4 Test matrix matching: the problem of the hydraulic loop

Match all the condition planned in the test matrix was consider one of the priority of the experimental campaign. Heating mode with desuperheater turned off was considered as reference to evaluate this match (see table 3.2). A first set of tests was performed for tandem and single compress runs adjusting the head of the variable speed pumps to get conditions closest as possible to the plan. The result of the measurement on the test are shown in table 4.1, it is a comparison between the temperature boundaries of the reference conditions and the values obtained in the these preliminary tests.

Table 4.1: Comparison of reference condition in the evaporator and condenser with the experimental result

Working condition	**Evaporator**		**Condenser**	
	Sec. Fluid Inlet $T_{sf,in}[^oC]$	*Sec. Fluid Outlet* $T_{sf,out}\ [^oC]$	*Water Inlet* $T_{w,in,con}[^oC]$	*Water Outlet* $T_{w,out,con}[^oC]$
Reference	0	-3	40	45
Full mode [a]	0,16	-5,82	37,93	44,87
Half mode [b]	0,04	-2,89	40,24	44,87

[a] Both tandem compressors working

[b] Only one compressor is active

First of all we noticed that the the reference value and the measured one are really close for $T_{sf,in}$ and $T_{w,out,con}$. It indicates that the three way valve is able to regulate as expected, since the valve sensor is sensible to these temperature and it regulates according to the setpoint. The real match is decided by the value of the other two temperatures in the heat exchangers. As can be observed from the values in Table 4.1, when the tandem compressors are working, the value of $T_{sf,out}$ and $T_{w,in,con}$ are quite different compare to the expected -3^oC and 40^oC: the match of the temperature conditions is not satisfied. For single compressor however, the reference boundary conditions are obtained.

During full load mode the difference in water and brine temperatures between inlet and outlet is higher than in the reference condition with the pump working at maximum capacity. High pressure drop on the water side compared to the expectation and undersized pumps in the hydraulic circuit have been considered the cause of the low flow. It assumes the pressure drop is similar inside the 3-way valve independently if it is fully closed or opened.

To solve the problem, a more detailed pressure drop analysis was performed using available experimental and theoretical data. The complete description is presented in section 4.4.1.

4.4.1 Hydraulic system analysis

Solving the problem about the low hydraulic mass flow in the system, the component that are responsible for the high pressure drop have to be identified. An experimental study was performed, evaluating the pressure drop by varying the capacity of the variable speed circulation pump for three different part of the hydraulic circuit involving the condenser. On the right part of figure 4.1 a graph indicates exactly these circuit, on the left the related measurement about:

- Condenser pressure drop (green line), measured by the differential pressure transducer;
- Circuit with 3-ways valve in full closed position (blue line);

- Circuit with 3-ways valve in full open position (red line).

The condenser was expected to have the largest source of pressure drop in the circuit, but the experimental data doesn´t match the expectation. As 4.1 shows, the condenser pressure drop is relative small (green line) compare to the total (blue and red lines), regardless of the 3-way valve opening. In addition the pressure drop with a 3-ways valve fully opened is not significant higher than the one in closed position. For this reason, it was concluded that the 3-way valve was responsible for the excessive pressure drop in the circuit.

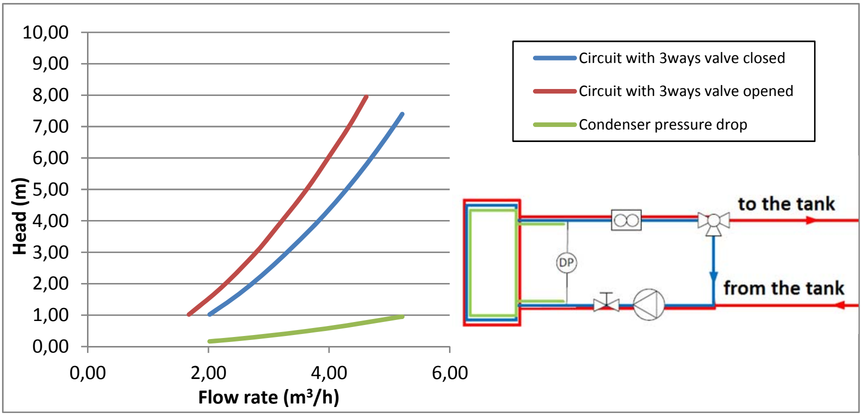


Figure 4.1: Experimental data of pressure drop for different sections of the condenser loop

Figure 4.2 shows a comparison between the theoretical pressure drop calculated by the 3-ways valve nominal data in the datasheet and the experimental pressure drop considering the entire loop with the valve in closed position. The theoretical values have been calculated using equation (4.1).

$$\Delta p = \Delta p_{nom} \left(\frac{\dot{V}}{\dot{V}_{nom}} \right)^2 \tag{4.1}$$

Where:

Δp_{nom}, $\dot{V}_{nom}$ are the nominal pressure drop and flow rate;

Δp, $\dot{V}$ are the new values of pressure drop and flow rate.

The theoretical curve in Figure 4.2 seems to fit the idea of a 3-ways valve has the main source of pressure drop.

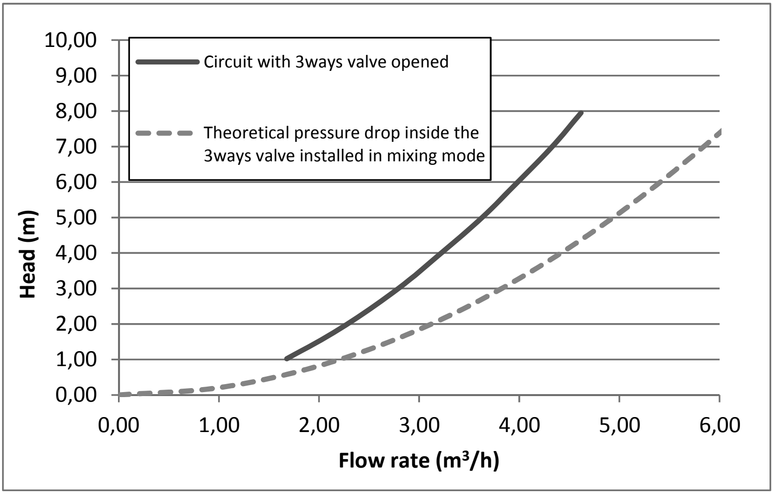


Figure 4.2: Comparison between the experimental total head in the condenser loop and the theoretical pressure drop inside the 3-ways valve (diverting mode)

4.4.2 Pressure drop calculation and new design

Calculation model

A new design of the condenser and evaporator loop has been performed with the following the following guidelines:

1. Size up the three way valve until it is not the main source of pressure drop anymore. Avoid to oversize the valve too much, not to compromise the time response of the valve itself during the control;
2. Install the 3-ways valve in mixing position (from the datasheet a lower pressure drop is expected compare to diverting mode)
3. Choose a new pump to fit the new requirement in the circuit (if necessary);
4. Choose pipe size to fit the inlet diameter of the pump to avoid disturbance in the flow and additional pressure drop in the inlet of the pump;

Point 3 requires implementing a model of pressure drop calculation for the hydraulic circuit that is a good compromise in calculation time and value estimation. For this reason, has been decided to focus on the following pressure drop source:

- Heat exchanger;
- 3 ways valve;
- "Other"

"Other" has been estimated from the experimental data in the condenser loop after subtracting the pressure drop from the 3 ways valve and the heat exchanger. It has been done using the definition of the located pressure drop coefficient in equation 4.2 for each component. The coefficient for "other" term is calculated from the one for all the circuit after subtracting the coefficient for the 3-ways valve and the heat exchanger as expressed in equation 4.3.

$$\Delta p = k \cdot \frac{\rho u^2}{2} \tag{4.2}$$

Where:

u: velocity of the fluid;

ρ: density

Δp: pressure drop.

$$k_{oth} = k_{cir} - k_{3wv} - k_{HX} \tag{4.3}$$

Where:

k_{oth}: located pressure drop coefficient for the remain part of the circuit;

k_{cir}: located pressure drop coefficient for the all circuit, calculated by the experimental values 3 way valve opened in the nominal condition;

k_{3wv}: located pressure drop coefficient for the all circuit, calculated by the experimental values condenser in the nominal condition;

k_{HX}: the theoretical located pressure drop coefficient, directly from the datasheet.

The estimation of k_{oth} permits to calculate the expected pressure drop curve for the new design. The pressure drop calculation can be done only for a specific flow rate, all the rest of the point on the curve can be drawn assuming a parabolic function relation beetween Δp and $\dot{V}$

$$\Delta p = \Delta p_{oth}(k_{oth}) + \Delta p_{HX} + \Delta p_{3wv} \tag{4.4}$$

Where:

Δp_{oth} is calculated by (k_{oth}) using equation 4.2;

Δp_{HX} is defined by the previous measurement or HX datasheet since the component is not replaced;

$\Delta p_{3wv,new}$ is calculated by the datasheet of the 3 way valve. Two different values can be obtain depending the type of installation of the valve: mixing or diverting mode.

Result for k-oth

The results are showed in Table 4.2. Linear pressure drop of the pipe could have been considered separately, but since it is really small compare to the other terms, it hasn´t been taken separately but included in "other". k_{oth} has been used to characterized the pressure drop in the new design assuming it remain constant. In reality it changes because it depends by the shape of the loop, size pipe and so on. On the other hand the calculation is referred to a quite severe situation with the small pipes of the current condenser loop therefore the calculation can overestimate it being in the safe range.

New design and related pressure drop curve

The details of the final design loop are already described in the test rig modification section (3.2.4). In addition, figure 4.3 presents the new expected pressure drop curves of the new condenser hydraulic system. They are two versions (in dash lines), related to the 3ways valve in opened position. The difference between the two curves is related to the value of the pressure drop in the

Table 4.2: Result of pressure drop characterization for "other" component from evaporator loop data

HX	Location	Flow rate $[m^3/h]$	Head [m]	k [-]
Condenser	*All circuit*	3,98	6	23,7
	Condenser	3,98	0,58	2,3
	3ways valve	-	-	14,8
	Other	-	-	**6,6**
Evaporator	*All circuit*	3,95	6	28,1
	Evaporator	10,1	2,9	2,0
	Water/Brine HX	10,5	4,4	2,9
	3 ways valve	-	-	14,8
	Other	-	-	**8,4**

3ways valve itself, depending if it is installed in mixing mode or in diverting mode. The comparison between the pump characteristic curve and the two pressure drop expectation shows the two curve to be below the new pump curve, in both cases, when the flow rate is the nominal flow $\dot{V}_{nom} = 8m^3/h$.

After the installation of the system, some tests have been performed to check the new condition in the hydraulic loop and compare with the prediction. Figure 4.4 shows the comparison between the experimental pressure drop measurement for the old system and the new one considering two positions of the 3-way valve: fully closed or fully opened. In both cases the curve are shifted on the right, it indicates a small value of pressure drop for the same flow rate, and the possibility to reach an higher flow rate. The intersection of the new measured pressure drop curve and the characteristic curves of the new pump is around $\dot{V} = 7 - 8m^3/h$, the maximum water flow rate needed in the system.

Figure 4.3 compares the expected value and the measured one for the same 3-way valve position (fully opened). The experimental curve (valve in mixing mode) is really near to the expected one in diverting mode, the pressure drops are really similar. A theory about can the the fact that for every 3-way valve installed in the opposite mode, there is a t-junction of the pipe that replace the position of the 3-way valve. It compensates the reduction of pressure drop inside the 3-way valve. On the other hand, in the case analysed the discrepancy in pressure drop was not so big to justified a deeper analysis of the problem.

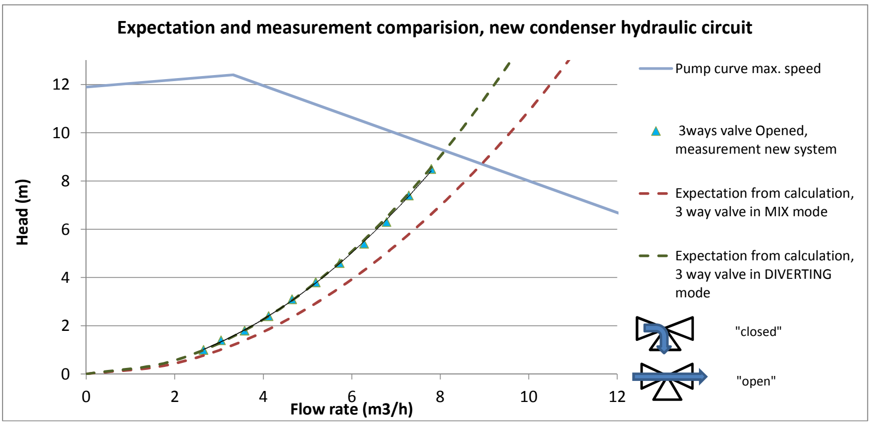


Figure 4.3: Comparison between the experimental total head in the condenser loop for the new configuration *(version B)* and the two expectation from the pressure drop calculation (3-ways valve in diverting or in mixing mode).

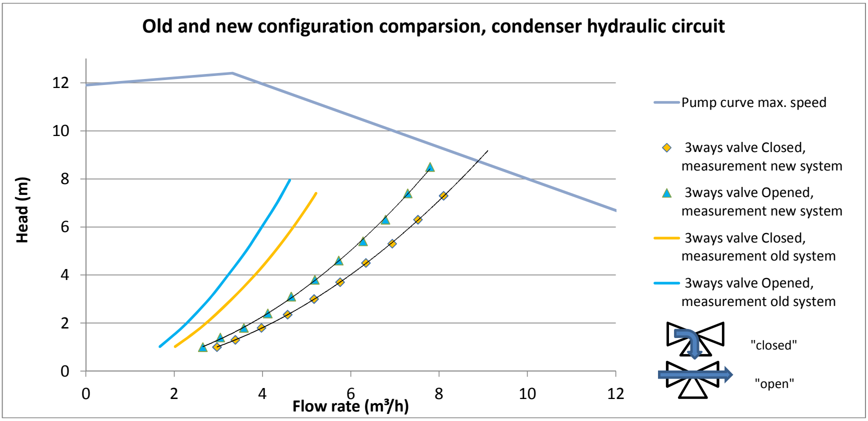


Figure 4.4: Comparison between the experimental total head in the condenser loop for the old configuration (*version A)* and the new one *(version B)*.

Chapter 5

Heat Pump model

In this chapter the calculation model for the heat pump is presented. All post-processing of the experimental data is implemented in Excel. Section 5.1 introduces the assumption in the model and the formulas implemented. Section 5.2 gives more detail about the software IMST-ART used for the COP calculation in section 6.3 and heat exchangers analysis in 7.3.

5.1 The Excel model

All the physical properties measured and presented in the tables 3.1 are used to perform the calculation and to characterized the performance of the machine using Excel software. The physical properties of propane and water are calculated using the software REFPROP 9.0 [12].
The calculation model is based on certain number of assumptions, the following are the key assumptions:

- Steady state condition is reached and no variation in refrigerant accumulation occurs in any part of the system;

- All the heat exchangers are considered adiabatic as regards the surroundings;
- Pressure drop in the condenser and evaporator in the refrigerant side are negligible;
- The additional heat losses in the pipes that link the main components of the heat pump are neglected;
- The additional electric work of the control system in the compressor is considered much smaller compare to the total power absorbed;

Selecting a time interval where the system is stable, the average value of all the measured physical properties are calculated and steady state indicators such as standard deviation are used to have a numerical check of the stability of the conditions. First of all, all the thermodynamic conditions of propane are calculated using the temperature and pressure measurement in the propane loops. The nomenclature used is reported in table 5.1.

Table 5.1: Nomenclature of the thermodynamic condition of the propane

Nomenclature	Description
1	Compressors inlet, Evaporator outlet
2	Compressors outlet, Desuperheater inlet
3	Condenser inlet, Desuperheater outlet
4	Condenser outlet, Expansion valve inlet
5	Expansion vale outlet, Evaporator inlet

The energy balance is performed in the condenser using data from the water side. The rejected heat by the condenser Q_{con} and the mass flow of propane m_{ref} are calculates using equations 5.1 and 5.2:

$$Q_{con} = \dot{V}_{w,con} \cdot \rho_w \cdot cp_w \cdot (T_{out,w,con} - T_{in,w,con}) \tag{5.1}$$

$$\dot{m}_{ref} = \frac{Q_{con}}{h_3 - h_4} \tag{5.2}$$

Where:

$\dot{V}_{w,con}$ is the volumetric water flow across the condenser;

ρ_w and cp_w are the density and the heat capacity of water, respectively;

$T_{out,w,con} - T_{in,w,con}$ is the difference in water temperature across the condenser;

$h_3 - h_4$ is the enthalpy difference of propane across the condenser;

Similar calculations are performed in the desuperheater side with equation 5.3 and 5.4. The mass flow check is done comparing $\dot{m}_{ref,dsh}$ with $\dot{m}_{ref}$.

$$Q_{dsh} = \dot{V}_{w,dsh} \cdot \rho_w \cdot cp_w \cdot (T_{out,w,dsh} - T_{in,w,dsh}) \tag{5.3}$$

$$\dot{m}_{ref,dsh} = \frac{Q_{dsh}}{h_2 - h_3} \tag{5.4}$$

Where:

$\dot{V}_{w,dsh}$ is the volumetric water flow across the desuperheater;

$T_{out,w,dsh} - T_{in,w,dsh}$ is the difference in water temperature across the desuperheater;

$h_2 - h_3$ is the enthalpy difference of propane across the desuperheater;

The energy balance in the evaporator permits to calculate the heat involved Q_{eva}, equation 5.5:

$$Q_{eva} = \dot{m}_{ref} \cdot (h_1 - h_5) \tag{5.5}$$

Where:

$h_1 - h_5$ is the enthalpy difference of propane across the evaporator;

Finally the energy performance of all the cycle is calculated using the definition of COP for heat pump (equation 5.6), together with the different type of compressor efficiencies (from equation 5.7 to 5.9).

$$COP_h = \frac{Q_{dsh} + Q_{con}}{W_{el}} \tag{5.6}$$

$$\eta_{vol} = 3600 \frac{\dot{m}_{ref}}{\rho_{ref,1} \cdot \dot{V}_{displace}} \tag{5.7}$$

$$\eta_{is} = \frac{\Delta h_{is}}{h_2 - h_1} \tag{5.8}$$

$$\eta_{overall} = \frac{\dot{m}_{ref} \cdot \Delta h_{is}}{W_{el}} \tag{5.9}$$

Where:
$\rho_{ref,1}$ is the density of the refrigerant at the compressors inlet;

$\dot{V}_{displace}$ is the total displace volume in $[m^3/h]$ of the working compressors. The value is coming from the datasheet of the compressor, $\dot{V}_{displace} = 29,61 m^3/h$ for each of the two twin compressors;

Δh_{is} is the isentropic difference in enthalpy for the same compressor inlet condition and outlet pressure;

$h_2 - h_1$ is the enthalpy difference of propane across the compressor;

5.2 IMST-ART software

IMST-ART is a performance simulation, computer-aided engineering design system for modelling vapour-compression refrigeration systems. Developed by the *Universidad Politécnica de Valencia*, it is based on their long experience on detailed modelling of refrigeration components. The software is used widely during the design phase of the NxtHPG project [13], [14],[15]. In this thesis work, it has been used to evaluated the performance of the condenser and evaporator in the analysis of the Heat pump components 7.3.

5.2.1 Software feature

IMST-ART is able to perform an accurate evaluation of the refrigeration unit performance including a quite detailed model for all the single component at the same time. It includes specific model for the most popular technologies in vapour compression system. The aim is to combine a high accuracy with a low CPU time in order to be a feasible tool to support the design of refrigeration equipment.

The main feature includes:

- Evaluation of fluid thermodynamics and transport properties of pure fluids and any mixture with thanks to the GENMAP software included in the main program.
- Calculation of the Theoretical cycle;
- Calculation of the Real working cycle, using models for the prescribed components: evaporator, condenser, piping and compressor. They are defines for specific boundary conditions given by temperature and flow rate of the heat source and sink;
- Parametric studies with single or combined input variables, for optimization studies;
- Study of stand-alone heat exchanger

The global model of the system is divided in submodels: compressor, heat exchangers, expansion valve, accessories, and piping. Here a short description of the heat exchanger model is given, further information are available on the literature references [16] and [17].

5.2.2 Heat Exchanger model

IMST-ART can simulated a heat exchanger using two different level of detail depending on the required accuracy. The simplest way consist of the characterization of the heat exchanger with total area and known Overall Heat Transfer Coefficient (OHTC). Then, $\epsilon - NTU$ method is used by the software to model the heat exchanger.

This simple model allows closing the refrigeration cycle taking into account the major effect of the HE on the operating cycle. OHTC can be imposed constant or as a function of the heat flux or the secondary fluid mass flow rate. This approach is usefull when it is interested in the a optimization study placed on another component.

If we are interested in an optimization of the heat exchanger, a very accurate estimation of the unit performance is required; and as consequence a detailed definition of the HE model must be used. The heat exchanger is discretized in cells along the refrigerant and secondary fluid paths, assuming one-dimensional flow. The model is able to take into account both heat transfer and pressure drop, with local evaluation of the heat transfer coefficient and friction factor, by built in correlations, as well as of the fluid properties.

The global solution method employed is called Semi Explicit method for Wall Temperature Linked Equations (SEWTLE). It is based on an iterative solution procedure where a first guess of the wall temperature is made and it permits the equations for the fluid to be solved explicitly, applying the finite volume methods on the discretized cells. A solution of the fluid properties are obtained at any fluid cell, then the wall temperature at every wall cell is estimated from the balance of the heat transferred across it. This procedure is repeated until convergence is reached. The short computational time is obtained thanks to the series of explicit calculation steps.

The method is applied for every type of heat exchanger, using the right correlation and model. For the compact HE, as the plates heat exchanger of the HP prototype under study, the flow along the channel has been considered to be 2-D and split into separated 1-D paths. Then, every 1-D flow path is discretized in as many elements as required, following the same SEWTLE strategy including also the calculation of longitudinal conduction. [18].

Chapter 6

Main experimental Campaign

The experimental campaign had as main objective to measure the energy performance of the heat pump prototype on certain specific boundary condition, defined in the test matrix in 3.3. On the other hand, other parameters can influence the performance of the test. It is important to keep track of the test condition to be able to analyse the result later. They are:

- Working compressor: 2C for two tandem compressors, LC and RC for only left and right compressor runs;
- Approximate charge in the system: usually optimized for one compressor operation (2.7 kg) or two compressors operation (2.5 kg);
- Water temperature at condenser exit $T_{w,out,con}$;
- Secondary fluid temperature at evaporator inlet $T_{sf,in}$;
- Desuperheater on or off: DS ON or DS OFF.

6.1 Measurement

Every test was performed according to this procedure:

1. Turn on all the pumps of the hydraulic loop, fluid start to circulate inside all the heat exchanger of the systems.
2. Turn on the HP prototypes, one of the two compressors starts followed by the other after few seconds (in full mode). The temperature inside the condenser increase and propane start to condensate rejecting heat to the water, in the evaporator the low pressure decrease the temperature letting possibile to absorb the heat from the secondary fluid.
3. The temperature of the water cooling the condenser increases. The 3 way valve in the circuit is usually fully closed at the moment, the same water is recirculated inside the condenser and it became hot quickly. Same concept, for the cooling of the secondary fluid in the evaporator.
4. The increase of the water temperature drives up the condenser pressure, the water temperature osculates around the set point, reducing the oscillation width until it get stability. The expansion valve keep the same aperture. Analogue phenomenon in the brine circuit.
5. When a certain operation stability is reached, the set point of the condenser (or evaporator) 3 ways valve is increase step by step until the desired value. It is important to wait for the stability of the system before change the set point of the valve, otherwise the oscillation in temperature can be too big to reach the temperature limits of $T_{w,out,con} > 50^oC$ in the condenser and $T_{sf,in} < -9^oC$. In this case safety system stops the compressor. The stability of the system is compromised
6. If the desired temperature condition are reached in the condenser and in the evaporator, the pumps speed is changed

to match the temperature difference in the HXs.

7. A couple of hours with all the boundary condition reached and stable are needed before starting to record the measurement from the experimental setup via computer.

8. The data are checked if steady state condition are reached and they are ready to be evaluated.

6.2 Experimental work chronology

The information about the experimental chronology is represented in figure 6.1. The two arrow shows the its evolution from two different point of view: Modification chronology and tests chronology. The first one gives indication of the physical changes in the test rig and it has been discusses in section 3.2.4. The second arrow is related to the main type of tests has been performed during the experimental campaign, three different period can be identified:

- *Tandem compressors 1 (TC1):* It is a period where most of the tests are performed with both twin compressors active with version A;
- *One compressor (OC):* Tests with mainly only one compressor is working are recorded.
- *Tandem compressors 2 (TC2): A new period with two compressors working testing the new hydraulic loops of the test rig (version B). One compressor tests was not available for problem with the control system of the machine in this working condition*

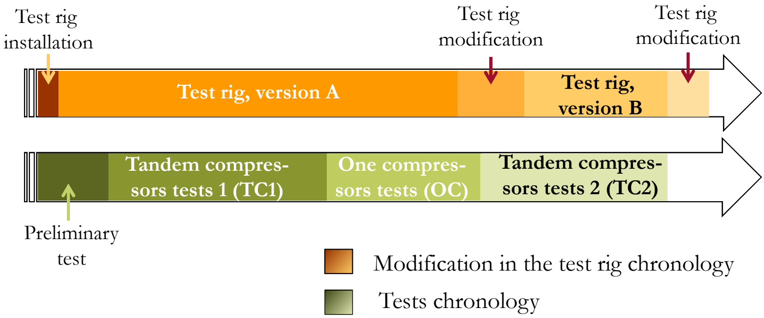


Figure 6.1: Experimental Chronology

6.3 COP analysis

During the experimental campaign, several tests on the unit has been performed. However, for the COP analysis, three tests have been considered representatives of the behaviour of the heat pump machine for all the experimental campaign. Every test is related to a different experimental periods that are indicated in previous section. All of theme have some features in common:

1. Temperature boundary according with the first condition in the test matrix achieving similar temperature level of the heat sink and source. (3.2):
   $T_{out,w,con} \sim 45^oC$ and $T_{in,sf} \sim 0^oC$;

2. Desuperheater off;

3. Superheating and subcooling "optimized", according to the preliminary tests discusses in section 4.2 and 4.3.

All the test has been compared with the equivalent IMST-ART simulation. The experimental measurement in table are the inputs for the simulation.

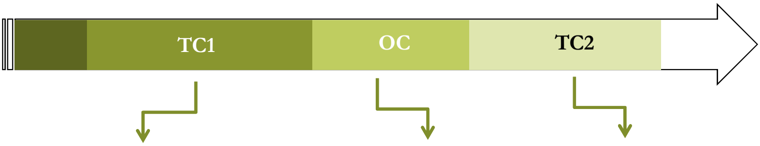
Table 6.1: List of the IMST-ART inputs for the equivalent experimental simulation

Symbol	*Description*
$T_{in,w,con}, T_{out,w,con}$	In and out water temperatures in the condenser
$T_{in,sf}, T_{out,sf}$	In and out water temperatures in the evaporator
ΔT_{sh}	Superheating
ΔT_{sc}	Subcooling

6.3.1 COP Result and discussion

The experimental result for each period are presented in figure 6.2 with the equivalent simulation in IMST-ART. It includes values for heat capacity (Q_{cond}), cooling capacity (Q_{eva}), compressor work (W_{ele}) and COP (heat pump). For the test in TC1 the match is less than 7 % for all the energy parameters and it decrease to less than 1% for COP. Apposite results for tests in OC and TC2 where the discrepancy in COP is 60& and 30% respectively.

All the other tests in the same category show similar results. It permits to conclude that the system working with both compressor in the beginning of the experimental campaign behaves according to the expectation. In all the test with one compressor and tests with two compressor in the second part of the experimental campaign, a drop of efficiency and capacity is registered. Something was going wrong in the system, for this reason a detailed component analysis is needed which is discussed in chapter 7.



	TC1		OC		TC2	
	Experimental test	*IMST – ART equivalent*	*Experimental test*	*IMST – ART equivalent*	*Experimental test*	*IMST – ART equivalent*
Qcond	43,0	40,3	11	23	31,8	41,9
Qeva	30,1	28,1	8,1	16,8	21,8	28,9
Wel	14,0	13,0	7,2	6,7	14,9	13,8
COP	**3,1**	**3,1**	**1,5**	**3,5**	**2,1**	**3,0**

Figure 6.2: Result of the Energy HP performance parameter evaluated experimentally and with equivalent IMST-ART simulations for tests belonging to different experimental period.

Chapter 7

HP components analysis

This chapter describes the analysis of the components of the heat pump: compressor, heat exchangers and expansion valve. They are discussed separately in each paragraph and at the end all the information are presented in the prospective of heat pump machine. Different hypothesis are presented to explain the unexpected result, the further prototypes improvements are also discussed.

7.1 Test experiment selection

During the experimental campaign, several tests on the unit have been performed, a selection is needed deciding which tests are considered more significant for the components analysis. First of all we need to have tests that cover different compressor configuration, it means to consider test in full mode, with both tandem compressors working, and in half mode with just one in operation. The second parameter in the selection is the time of record of the test; the aim is to understand if the HP components change their function with time. Six tests has been chosen in these criteria and they are reported in table 7.1. It includes two tests in partial mode (3, 4) and four in full mode; two of them are recorded during the first stage of the experimental campaign (1, 2) and the other two dated at the end of the experimental campaign (5, 6). The detailed result from the experimental measurement are reported in the appendix A.

Table 7.1: Detail of test selection to perform the HX analysis

N	File name	Date	Mode	$T_{w,out,cond}$ $[^oC]$	$T_{sf,in,eva}$ $[^oC]$
1	2.5.. kg - 45 0 - 2 comp - DS OFF - v31c"	19-09-2014	Full	45	0
2	2.7.. kg - 37 0 - 2 comp - DS OFF - v31c"	02-10-2014	Full	37	0
3	2.7...kg - left comp - DS OFF - v31c"	01-10-2014	Half	37	0
4	2.7...kg - left comp - 33 0 - DS OFF - v31c"	01-10-2014	Half	33	0
5	"2.5 kg 2 compr 45 0 v32v2"	28-11-2014	Full	45	0
6	"2.5 kg 2 compr 45 5 v32v2"	09-12-2014	Full	45	5

7.2 Compressor

7.2.1 Model of the analysis

The analysis of the compressor performance has been based on the comparison of series of compressor parameters. It permits to have an high information level about the behaviour of the compressor, according with the data available. The complete list is reported and for each parameter the origin of its value is mentioned in brackets.

1. Compressor electrical power (Direct measurement);
2. Refrigerant mass flow (Calculated with equations (5.2) and (5.1));
3. Discharge temperature (Direct measurement);
4. Volumetric efficiency (Calculated with equation (5.7));
5. Isentropic efficiency (Calculated with equation (5.8));
6. Overall efficiency (Calculated with equation (5.9)).

Every parameter, evaluated or directly measured experimentally, is compared with the expectation from the manufacture company Danfoss according to the same working condition. ARI correlation, provided by the manufacture, was used for this plane. They are polynomial correlation (see equation 7.1) that permits to predict a compressor parameter X using only two input parameters: suction and discharge dew point temperatures (S and D respectively) [19]. A set of correlation coefficients C_i for every parameter X are calculated by Danfoss with a best fit of experimental data.

$$\begin{aligned} X = C_1 + C_2 \cdot S + C_3 \cdot D + C_4 \cdot S^2 + C_5 \cdot S \cdot D + C_6 \cdot D^2 + \\ + C_7 \cdot S^3 + C_8 \cdot D \cdot S^2 + C_9 \cdot S \cdot D^2 + C_{10} \cdot D^3 \end{aligned} \tag{7.1}$$

The X parameter can be:

- Compressor power;
- Refrigerant flow rate;
- Cooling capacity

- Discharge temperature;

The refrigerant mass flow is adjusted in relation with the superheating using equation 7.2:

$$\frac{\dot{m_{new}}}{\dot{m_{ARI}}} = 1 + 0.75\left(\frac{\rho_{new}}{\rho_{ARI}} - 1\right) \tag{7.2}$$

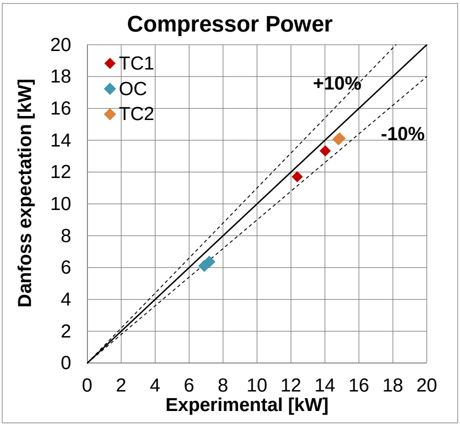
7.2.2 Result of the analysis

Figure 7.1 consist of six different plots, one for each compressor parameter. They show the direct comparison of the value of the parameter experimentally evaluated (x-axe) and using the ARI-correlation provided by Danfoss (y-axe). Every point in the graph is related to one of the six tests in table 7.1, they are gathered together in three color groups according to the similarity in the working condition or test date.

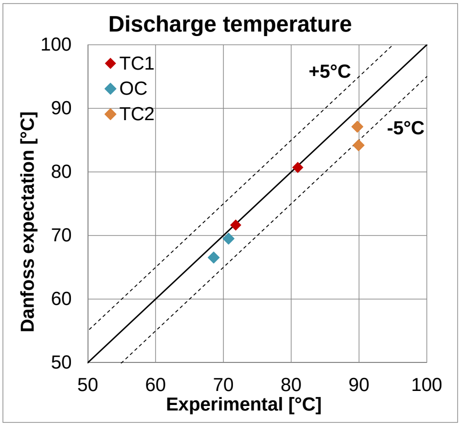
- *Tests 1 & 2*: the match is generally good with all the parameters. For discharge temperature the discrepancy is less than 1 K, most of the other show less than 5% error. Only the overall efficiency reaches a discrepancy around 10%.

- *Tests 3 & 4*: The performance are relatively positive for some of the compressor parameters: discrepancy around 10 % for compressor power and isentropic efficiency, 1-2 K for discharge temperature. On the other hand for refrigerant mass flow, volumetric and overall efficiencies the mismatch is huge (40-60%).

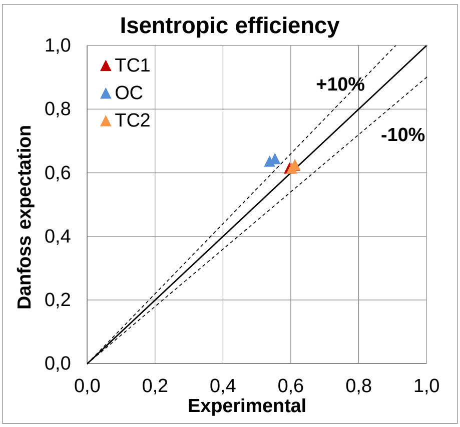
- *Tests 5 & 6*: Similar consideration of tests 3 & 4 but for the poor parameters the difference is less pronounced but still large (20-30%).



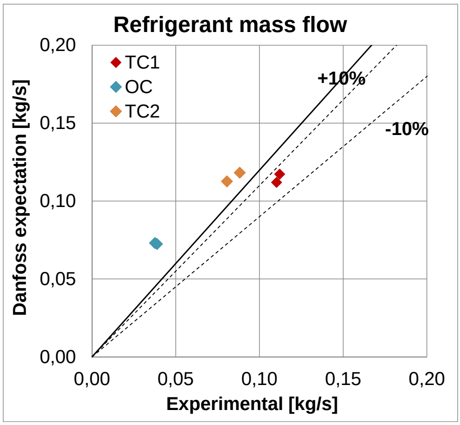
(a) Compressor power



(b) Discharge temperature

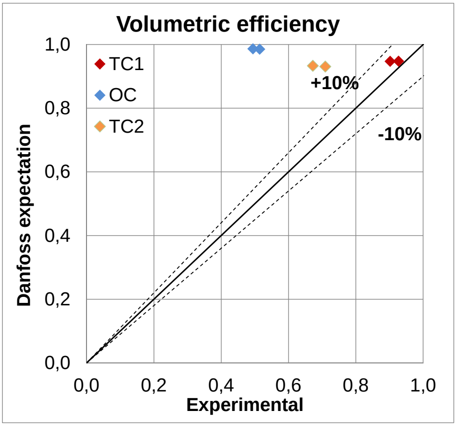


(c) Isentropic efficiency

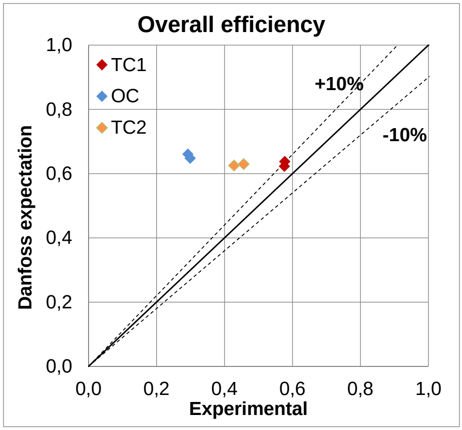


(d) Refrigerant mass flow

Figure 7.1: Compressor parameter comparisons between the experimental value and the manufacture Danfoss expectation (ARI correlation) *(1st figure of 2)*.



(a) Volumetric efficiency



(b) Overall efficiency

Figure 7.2: Compressor parameter comparisons between the experimental value and the manufacture Danfoss expectation (ARI correlation) *(2nd figure of 2)*.

7.2.3 Comments of the analysis

The six compressor parameters should be able to answer to the question about the state of the compressor. A drop is registered for refrigerant mass flow rate, overall and volumetric efficiencies during the experimental campaign for tandem compressors working. When one compressor is working, the performance registered were never fully satisfying. On the other hand the result seem to be in conflict for these tests if the parameters are analysed one by one. Few of them are truly near to the expectation, indicating a well-behaved compressor; others parameter denote unexpected and undesirable compressor performance. A special patter was noticed: the well-behaves parameter are independent by the refrigerant mass flow in their estimation contrariwise the poor ones. The compressor seems to work with a flow rate different compare to the estimation from the condenser heat balance.

7.3 Condenser and Evaporator

Two heat exchangers of the same model *AC220EQ-76AM-F* are used as evaporator and condenser [11]. The aim is to understand if these components are working properly or not, according to the expectation from the manufacture company Alfalaval.

The analysis of a heat exchanger, specifically of a condenser or an evaporator, contains intrinsically some difficulties that can be listed in the following points:

1. *Nominal condition*: The manufacture company gives usually details about one working condition, specific company software are usually needed to perform analysis in different and specific conditions. On the other hand the software was not available for this thesis work;
2. *Variable UA-value*: The heat transfer in a condenser/evaporator is strongly dependent by the vapour quality. The UA-value changes according to the real operation of the heat exchanger.
3. *Model for LMTD*: According to the analytical derivation of LMTD [20], it can not be used in an evaporator or condenser since the fluid properties changes drastically in the heat exchanger. However the heat exchanger can be divided in cells considering constant the properties locally and apply the definition of LMTD. On the other hand the type of discretisation used for the calculation can have a strong influence on the final result of the UA-value. The comparison of the experimental result with the nominal condition has to be done using the same calculation model.

Considering all the reasons mentioned before, an evaluation model has been defined using IMST-ART software and it will be described in 7.3.1 section.

7.3.1 Analysis model

The analysis model allows determining the temperature profiles, the UA-value, the LMTD of the heat exchanger using the inlet/outlet conditions experimentally evaluated as input information. The *standalone heat exchanger* tool, in IMST-ART, has been considered the best choice to implement a relatively quick calculation with the necessary level of accuracy. As mentioned before, no manufacture software were available for the analysis.
All the geometrical data of the heat exchanger *AC220EQ-76AM-F* product were already available in IMST-ART, since it has been used during the design phase of the NxtHPG project. Figure 7.3 shows a detail of the window of the software about the geometrical input that characterizes the heat exchanger. It is a plate HE in counter-current flow composed by different 76 plates.

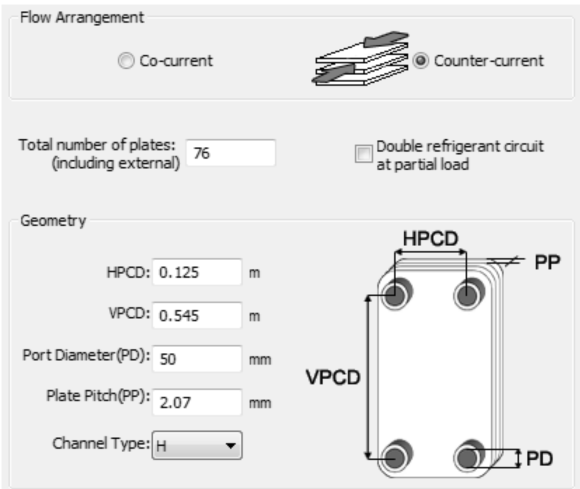


Figure 7.3: Geometrical input of the model *AC220EQ-76AM-F* in IMST-ART

According to figure 7.4 the model can be described as composed by four different steps: calibration, software calculation, software output, post-process calculation.

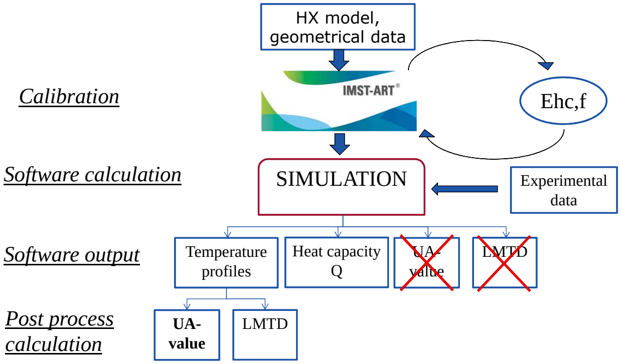
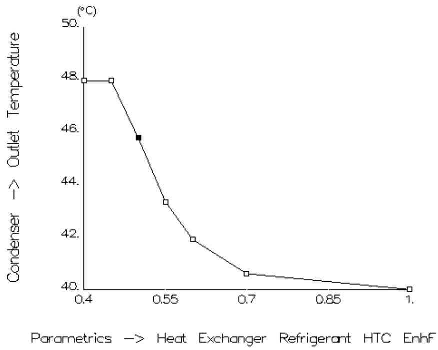


Figure 7.4: Scheme of the analysis model for the heat exchangers

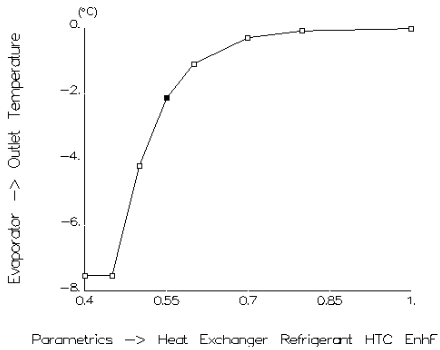
Calibration

The first phase includes a calibration of the model according to the datasheet of the HE. The aim is to define a simulation that is able to fit with all the boundary conditions (temperatures and flow rates) given in the datasheet. The parameter that permits to define the calibration is Ehc_f (Enchantment factor). It gives the possibility to amplify or decrease the heat transfer coefficient from the refrigerant side of an arbitrary value, until all the nominal conditions are reached in the HE. Two parametric studies have been performed, one for the condenser and one for the evaporator to achive the correct calibration for each one. The varying parameter is Ehc_f and the outlet temperature of the refrigerant is controlled to have the perfect match with the the value indicated in the datasheet for the *Nominal heating mode* (appendix

B).
Figure 7.5 shows the result of the calibration. The outlet temperature of the secondary fluid is reported on the y-axe versus the Ehc_f on the x-axe. The values $Ehc_{f,cond} = 0.50$ and $Ehc_{f,eva} = 0.55$ has be obtained to fit the refrigerant outlet temperatures of 45.5^oC and -2^oC in the condenser and evaporator respectively.



(a) Condenser



(b) Evaporator

Figure 7.5: Result of the parametric study during the calibration of the HX model in IMST-ART

Software calculation

With the calibrated model, it is possible to precede with the IMST-ART simulation according to the real experimental conditions. It can be done choosing a set of the direct experimental measurements (or calculations deduced from them) that can be used as input parameters for the simulation. In this way the working condition of the heat exchanger are completely defined and the simulation can now describes the behaviour inside the HX. Figure 7.6 displays an example of the software screen where the data are inserted in, while table 7.2 gives the detailed indication of the inputs selection for condenser and evaporator. The main difference in inputs is related to the way the inlet conditions of the refrigerant are provided. In the condenser, the superheated vapour at the inlet is directly defined with temperature and pressure (from the condensing temperature). In the evaporator the inlet condition of the two-phases fluid are provide with pressure and quality. The first one is defined indirectly with the evaporating temperature, the couple parameters condensing temperature & subcooling fixes the second. IMST-ART assumes an isenthalpic transformation to calculate the inlet quality.

Software output

Every simulation gives widely number of outputs, on the other hand just few of them are necessary for the HE analysis, here a short list and description:

- *Temperature profiles*: It is used as input for the post process calculation
- *Heat Capacity*: It is the parameter used to validate the model according to the real experimental result, the only one than can be compared directly with the experimental data.

Table 7.2: Inputs parameter used in the IMST-ART simulation of the AC220EQ-76AM-F heat exchanger used as condenser and evaporator

	Input parameters for the simulation
Condenser	$T_{w,in}$, Inlet water temperature $T_{w,out,con}$, Outlet water temperature p_{water}, Water pressure[a] T_3, Inlet refrigerant temperature T_{con}, Condensing temperature at the inlet m_{ref}, Refrigerant flow rate
Evaporator	$T_{sf,in}$, Inlet secondary fluid temperature $T_{sf,out}$, Outlet secondary fluid temperature p_{sf}, Secondary fluid pressure[a] T_{eva}, Evaporating temperature at the inlet m_{ref}, Refrigeration flow rate T_{con}, Condensing temperature at the outlet ΔT_{sc}, Subcooling

[a] Small changes does not affect the result, assumed to be equal to the atmosferic pressure

- *UA-value and LMTD*: It has been calculated by the temperature profile through the post process calculation, despite the fact they can be calculated directly by IMST-ART. The main reason is the empirical observation that the value of LMTD, given from the software, is sometimes incompatible with the temperature profiles, probably because of some minor bugs in the software.

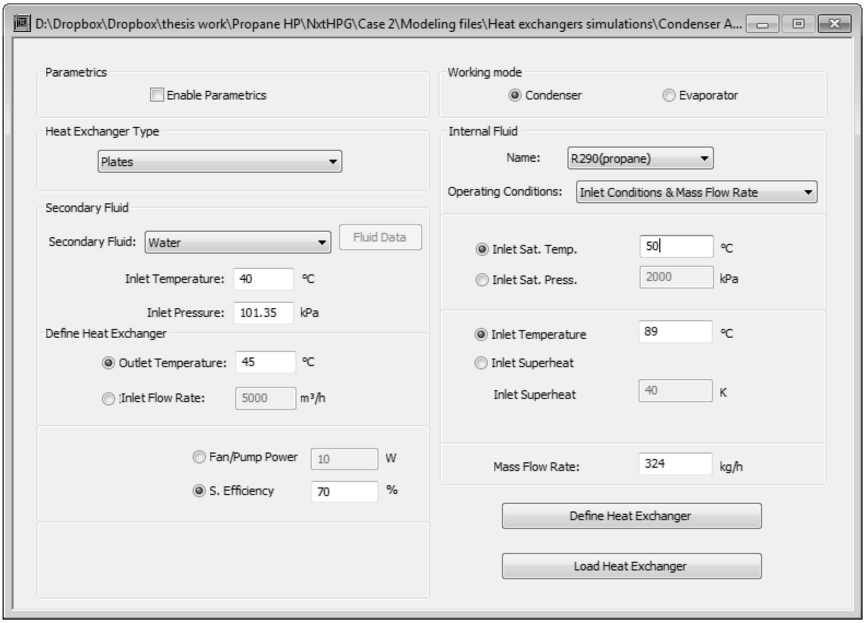


Figure 7.6: Input example of the thermal data of condenser in IMST-ART

Post process calculation

The post process calculation permits to determine the UA-value and LMTD from the 1D temperature profiles in the heat exchanger given by IMST-ART. The HE is divided in small cells and the fluid properties are considered constant in every cell. Two fluids (hot and cold) are exchanging the heat Q (see figure 7.7). Considering the i-th cell, the temperature of the hot fluid decreases from $T_{h,i}$ to $T_{h,i+1}$, the cold fluid temperature increases from $T_{c,i+1}$ to $T_{c,i}$ and Q_i is the amount of heat transferred through the cell. The latter can be calculated according to equation 7.3, derived by an heat balance in the secondary fluid (the hot fluid in the figure 7.7). The definition of LMTD can be applied to the i-th cell according to equation 7.4, since the properties are assumed constant.

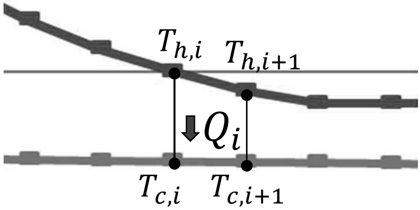


Figure 7.7: Detail of the 1D heat exchange model

$$Q_i = m_{sf} c_{p,sf} \cdot (T_{h,i} - T_{h,i+1}) \tag{7.3}$$

$$LMTD_i = \frac{(T_{h,i} - T_{c,i}) - (T_{h,i+1} - T_{c,i+1})}{\ln \dfrac{T_{h,i} - T_{c,i}}{T_{h,i+1} - T_{c,i+1}}} \tag{7.4}$$

Finally the overall values can be computed according to equations 7.5 and 7.6. Since in every cell the thermodynamic properties varies, we can not speak about LMTD. The name mean temperature difference (MTD) seems to be a more appropriate term instead. For the rest of the analysis MTD will be used instead of LMTD.

$$MTD_{overall} = \frac{\sum Q_i}{\sum \dfrac{Q_i}{LMTD_i}} \tag{7.5}$$

$$UA_{overall} = \sum UA_i = \sum \frac{Q_i}{LMTD_i} \tag{7.6}$$

7.3.2 Model validation

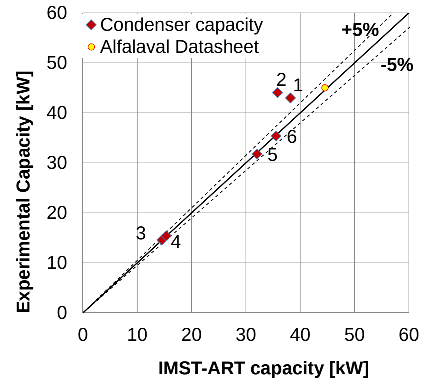
The model validation has been performed using the heat capacity of the heat exchanger as parameter. As mentioned before, it is the only parameter than can be calculated independently with a simple heat balance from the experimental data and with IMST-ART in the same time.

Model Result

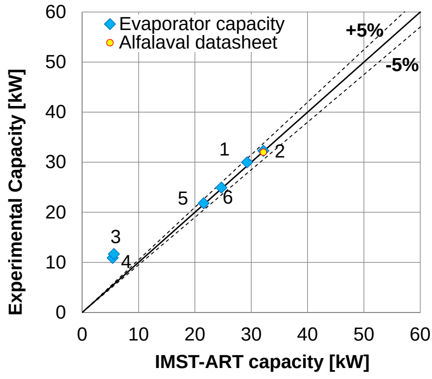
Figure 7.8 shows the result of the comparison between the experimental and the IMST-ART prediction for the UA-value, in the condenser and in the evaporator respectively. Every point is numbered from 1 to 6, according to the legend mentioned in table 7.1. A wide capacity range is covered in the analysis: from 15 kW to 45 kW in the condenser and from 11 kW to 32 kW in the evaporator. For most of the tests the discrepancy in the heat capacity is less than 5%, on the other hand the difference can be significant in some cases. In test 1 and 2 the measured heat capacity in the condenser is more than 15% higher than the IMST-ART prediction, in fact in these simulations not fully condensation is reached despite the fact subcooling is measured experimentally. For the evaporator the tests in partial mode (test 3 & 4) gives the large mismatch (-50%). All the other tests are in the range of $\pm$5%.

Conclusion

Generally the model defined in IMST-ART and calibrated on the datasheet is able to describe, with a sufficient level of detail, the heat transfer inside the heat exchanger. On the other hand for some tests the discrepancy can be significant, so the result from these simulation can not be considered valid (tests 3 and 4 in the evaporator). Despite the fact a deviation is present in the con-



(a) Condenser



(b) Evaporator

Figure 7.8: Comparison of the heat capacity evaluated experimentally and by software IMST-ART calculation for different experimental working condition, in the condenser and evaporator.

denser for test 1 and 2, it has been considered not so large to let impossible to get conclusion for the UA calculation from these simulations. For tests 1 and 2, the heat capacity is underestimated compared to the experimental value and, as consequence, the UA-value too.

7.3.3 Heat exchangers Result

UA-value

The result of the UA-value calculation are presented in figures 7.9 and 7.10 for the condenser and evaporator respectively. The first column is related to the calibrating nominal condition and it is used as comparison for all the tests. As consequence of the validation model discussion in the section above, tests 3 and 4 in the evaporator are not considered reliable in the UA-value calculation. However, to be thorough, they are reported in figure 7.10 with the indication *not valid*. The obtain results are generally lower in value compare to the expectation, on the other hand tests 1 and 2 result to be the nearest to the expected nominal in both heat exchangers with a discrepancy of 8-16%. For the condenser, the real outcome related to these tests will be probably better than the situation shown in the plot, since the UA-value is expected to be underestimated for these tests. The lowest values in the condenser are registered for test 3 and 4. The same tests, in the evaporator, are considered to be not valid. Despite tests 5 and 6 are related to tandem compressors, the result is lower compare to test 1 and 2. The difference is bigger in the condenser compare to the evaporator.

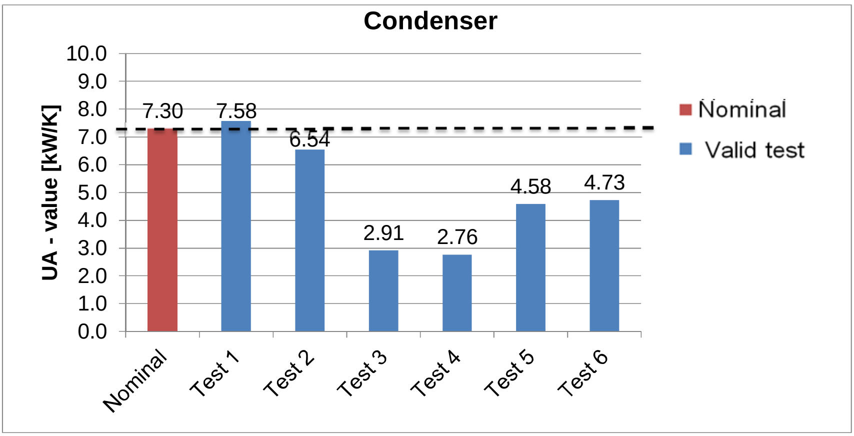


Figure 7.9: Comparison of the UA-value result for the selected tests and the Nominal condition for the condenser

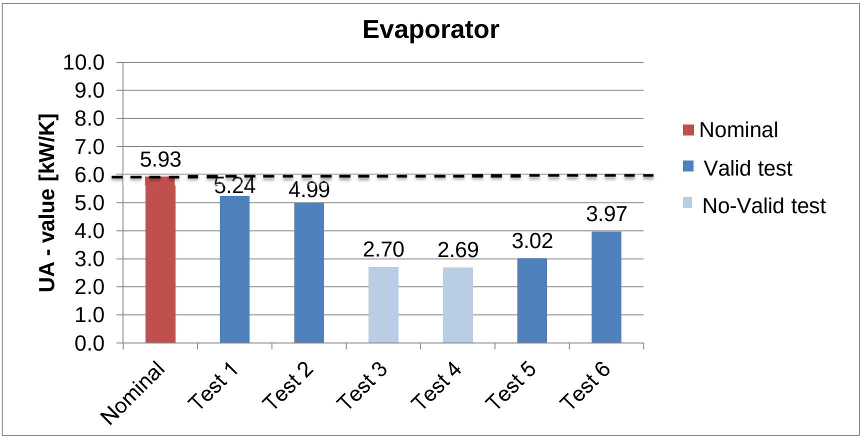


Figure 7.10: Comparison of the UA-value result for the selected tests and the Nominal condition for the evaporator

HX internal profile

The 1D model in IMST-ART permits to get a certain number of information related to the behaviour of the heat exchanger looking to the expected internal profile of same parameter such as: temperatures of the two fluids, refrigerant vapour quality and local UA-value.
All of them are reported in figures 7.11 and 7.12 for the calibrating nominal condition and in the appendix C for all the tests mentioned.
The area below the local UA-value curve gives the total UA-value, whose values have already been presented in section 7.3.3. In every plot, the real boundary condition are reported with green point. They are related to the data from the datasheet or experimentally evaluated. Thanks to the calibration, all the temperature profiles fit perfectly with the datasheet condition.

Figure 7.11 shows graphically the feature of a good condenser that behaves according to the expectation. Condensation occurs in most of the internal HX space, the pitch point is located at the beginning of the condensation with a reasonable value of 4 K; the subcooling is relatively low (2.5 K).
Similar aspects can be recognizes in figure 7.12 as symptom of good-behaves evaporator: evaporation occupies most of the heat transfer area, low pitch point and superheating ($2K$ and $5K$).

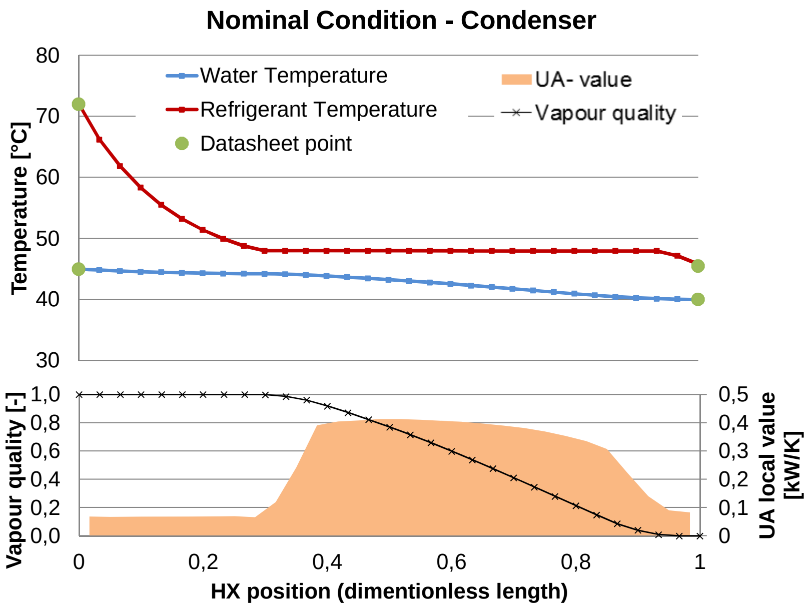


Figure 7.11: Model result applied in the nominal condition for the condenser

The results about the internal profile parameters for all the experimental selected tests are reported in the appendix C. It includes six plots for the condenser and six for the evaporator, one for each selected test. In the following list, the main feature are analysed.

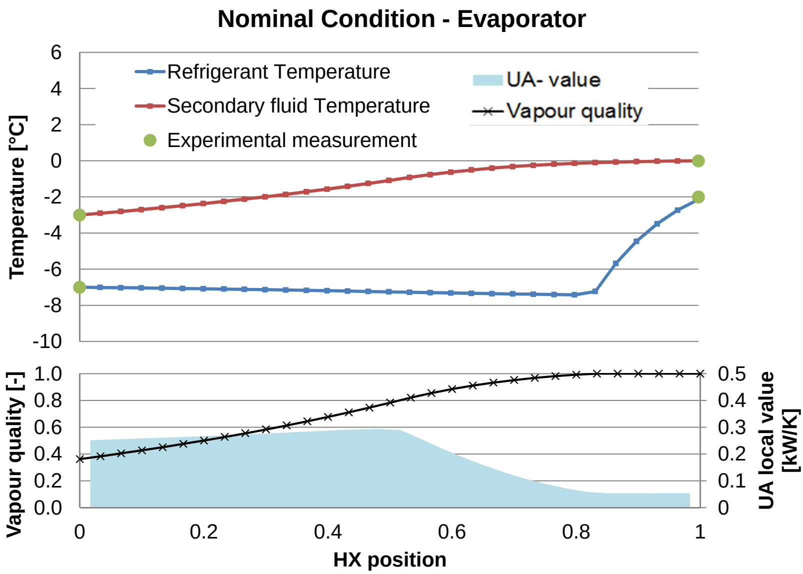


Figure 7.12: Model result applied in the nominal condition for the evaporator

- *Tests 1 & 2*:
  - Condenser & Evaporator: The profiles of temperature, quality, local UA-value are similar to the nominal case. It confirms the good performance predicted by the overall UA-value. Their behaviour agrees with the expectation (nominal condition);
- *Tests 3 & 4*:
  - Condenser: The condensation area in the condenser is much lower than the nominal condition, resulting in a low UA-value. The pinch point of the heat exchanger is near to zero and located at one of the fluid exit. The HE is oversized, but the low performance are compatible with the partial load the condenser working with (low

refrigerant flow).

  - Evaporator: The results are not able to represent the real situation in the heat exchanger with a calculated quality exit of 0.6 instead of 1.

- *Tests 5 & 6*:
  - Condenser: Similar consideration of tests 3 & 4, despite the fact they are tests in full mode as tests 1 & 2. The condensation pressure and subcooling are higher than the one registered for test 1 & 2 and the refrigerant charge is supposed to be similar. The charge influence the subcooling, but the increase of the latter is not linked with this aspect. The phenomenon can be compatible with inert gas in the system, however the control in the charge was not so accurate to get useful information on the variation in the subcooling;
  - Evaporator: The evaporation happens only in a small portion of the evaporator. Most of its area is used for superheating with a pitch point really close to zero. The heat transfer is not effective. Low UA-value, the temperature profile and the relative high superheat suggests that the expansion valve is not able to regulate the amount of superheat at the evaporator refrigerant outlet. Comments about it are reported in the 3 way valve section 7.4.

7.3.4 Heat exchangers Analysis

The heat exchangers seem to be designed property and it is confirmed by the good performance registered in the beginning of the experimental campaign with test 1 & 2 whose are condition really similar to the nominal one. The low performances for the other tests can be explain by the fact that lower refrigerant flow rate going determines an oversized heat exchanger for this condition. In the last tests with tandem compressors (5 & 6) some evidences are compatible with the theory of inert gas in the system. On the other hand they can not be consider as truly strong evidence.

7.4 Expansion valve

The analysis of the expansion valve component consists of a list of experimental observation related to its functionality instead of model comparison as it is for the compressor and heat exchangers. The expansion valve installed is a commercial thermostatic valve produced by Danfoss and it is not a prototype. However it is important to check if unexpected behaviour of the valve can be noticed with the use of propane as refrigerant.

7.4.1 Experimental observation

The experimental observations is divided by test groups:

- *tests 1 & 2*:
  - Superheating regulated by the expansion valve with regulation of the spring screw of the valve in a intermediate position;
  - Presence of bubble in the sight glass with the presence of subcooling;
- *tests 3 &4* :
  - Superheating regulated by the expansion valve. The valve is set to be more closed with the spring screw adapting to the lower refrigerant flow capacity;
  - Presence of bubble in the sight glass with the presence of subcooling;
- *tests 5 & 6* :
  - The valve is set to the maximum opening setting with the spring screw but it is not able to regulate the superheating any more. It is generally higher than the desirable value and it varies strongly with the temperature level of the heat source in the evaporator. It seems

the thermostatic control of the valve open it to the maximum, no more opening is available for the regulation.

– Presence of bubble in the sight glass with the presence of subcooling without the real possibility to see if they are more or not compare to the previous conditions;

7.4.2 Comments

During the first part of the experimental campaign no issue were registred with the expansion valve. The value behaves as expected and the design size was the right one to have a proper regulation of the superheating. On the other hand in the second phase of the tests, the regulation was not any more possible and the valve was acting as it is undersized. An explanation of the phenomena is related to the inert gas hypothesis that will be discussed in section 7.5.2. Since the specific volume of inert gas is much higher than the liquid propane, small amount of inert gas can take out large space in the valve, reducing drastically the available region for liquid to go through. In fact bubble has been always registered in the sigh glass after the condenser. However it is hard to evaluate changes in the amount of bubble during the different tests, because of the horizontal position of the sight glass. For test with one compressor working, the reduction of capacity avoid any issue for the valve.

7.5 Operation of the HP machine

In this section the functionality of the entire heat pump machine is considered summarizing all the information provided by the component by component analysis and connecting together in a global prospective. The analysis is performed dividing in tandem compressors and one compressor working.

- *Tandem compressors:* The performance of the heat pump varied during the experimental campaign. During the **first period** it as generally **satisfying** the expectation, in fact (reference tests 1 & 2):

  1. The COP measured is predicted by the software IMST-ART;
  2. The compressor analysis is positive for all the six compressor parameter;
  3. The UA-value, the temperature and quality profiles of the condenser and the evaporator are close to the respective nominal condition;
  4. No unexpected functionality of the expansion valve ;

  The same machine in the **second period** of the experimental campaign did not result in the same performance(reference tests 5 & 6). As result:

  1. The COP measured is significantly lower than IMST-ART calculation;
  2. The compressor parameter depending by the refrigerant mass flow are inadequate;
  3. The condenser and the evaporator are oversized for this working condition, low UA-value;
  4. The condenser seems to work with an higher subcooling and condensing pressure compare to previous tests;
  5. Expansion valve is not able to regulate the superheat in the evaporator;

- *One compressor:* **All the tests** performed show **unexpected result** compare to the expectation (reference tests 5 & 6). The most important features are:
    1. The COP measured is significantly lower than IMST-ART calculation, worst then all the tests with tandem compressor;
    2. The compressor parameter depending by the refrigerant mass flow are inadequate;
    3. The condenser and the evaporator are oversized for this working condition, low UA-value;
    4. Expansion valve regulates the superheating, no issued noticed;

Two different hypothesis has been formulated trying to give reasonable explanation for the system unexpected behaviour. They are:

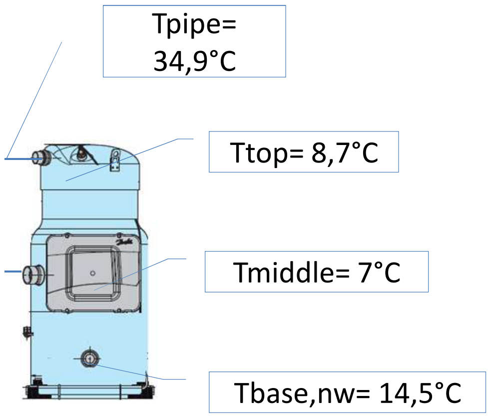
- Backflow in the no-working compressor
- Inert gas in the system

7.5.1 Back-flow in the no-working compressor

This hypothesis assumes that, liquid propane goes though the non working compressor when the other one is activate. In this way the working compressor is compressing an higher amount of refrigerant compared to the amount that reach the condense. In addition the eventual mixing of the propane in suction and discharge of the compressors modifies the real inlet/outlet condition of the working compressor. The compressor performance are calculated with measurement that are not the real boundary condition of the machine.
The back-flow of propane is assumed to be liquid, since it can explain an additional experimental observation related to a cooling effect noticed on the crankcase of the non working compressor. In

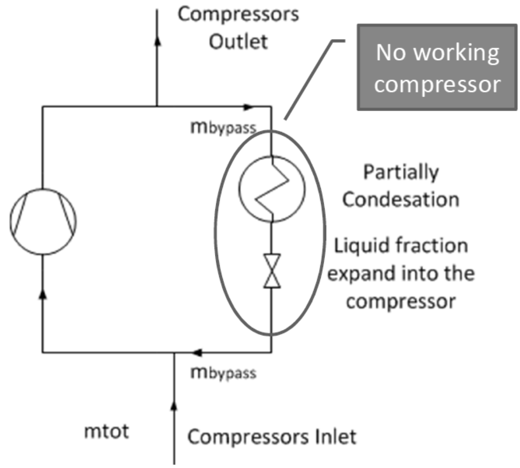
fact, after several hours of working time of the machine and steady state condition reached, the non working compressor crankcase gets sufficient cold to let external humid air condense on it (figure 7.13). The non working compressor acts as a condenser and expansion valve in series (figure 7.13a). The propane has to condensate and accumulate in the head of the compressor and then expand going through a not fully sealing check valve, expanding liquid in the compressor body. The temperature after the expansion is generally really low since it is the saturation temperature related to the pressure in the low pressure stage (figure 7.14b). If superheated vapour expand through the compressor, the final temperature is expected to be much higher (around 20^oC) without a visible cooling effect.

(a) Real picture

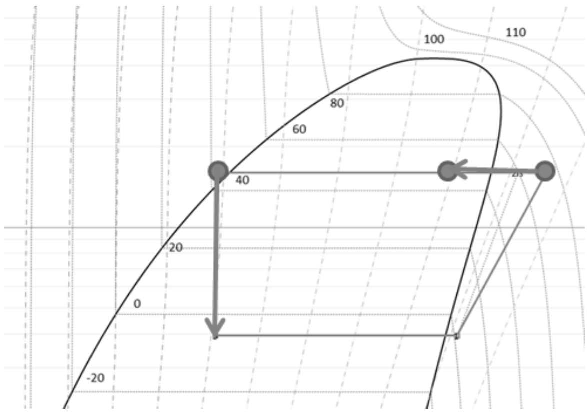


(b) Crunckcase Measurements

Figure 7.13: Formation of condensation on the compressor case, symptom of cooling effect takes place inside



(a) Equivalent thermodynamic model



(b) p-h diagram

Figure 7.14: Equivalent thermodynamic model for the non working compressor and p-h diagram showing the reasonable thermodynamic transformation happing in the system

7.5.2 Inert has in the system hypothesis

According to this hypothesis a significant amount of inert gas is present in the system. The gas goes through the system without creating any significant additional cooling/heating capacity since it can not evaporate or condense. It is possible that the compressor compress the right amount of volume flow rate, but since it is a mixture propane vapour / gas the performance measured are not high since referred only to the propane fraction.
The experimental observation that agree with the theory are:

- Poor performance of the compressor for parameter that includes refrigerant mass flow in the calculation;
- Presence of bubble in the sight gas after the condenser;
- Expansion valve not able to regulate the superheat, acting as undersized.

On the other hand this hypothesis has some limitation and it can to explain everything in the system. First of all it is not so clear how the inter come inside the system. It is not so easy for air penetrate in a pressured system and a suitable vacuum has been reached before filling the system. On the other hand several charge and discharge of the system have been taken place, in this way the most reasonable explain could be some impurity in the refrigerant used. Secondly, if we need to explain the drop in refrigerant mass flow only with this hypothesis, the amount has to be really high. However the presence of bubble in the glass or the problem in the expansion valve can be explain with not so high amount of gas in the system as well.

7.6 Further improvements

During the experience of the 1st experimental campaign several question marks are related to the results of the machine, for this reason the highest priority has been given to the proposal of experimental setup changes that can clarify the situation during the next phase of the experimental campaign. In addition the results of the performance of the machine and the single components were discuss with the manufacturers and they provided some raccomandation in the system improvements. All the terms in the list below are already confirm by partner of the NxtHPG project and they will be applied for the next phase of the experimental campaign.

1. *Compressor check valve:* Five manual check valves is planned to be installed in the compressor pipes refer to the figure 7.15: two in the suction pipe, two in the discharge and one in the oil equalizer. In this way is possible to isolate completely and without any doubt the non working compressor from the other and confirm or debunk the backflow hypothesis. The check valve in the oil equalizer is needed for problem related to propane solved in oil circulation.

2. *Desuperheater position*: Alfalaval propose to change the position of the desuperheater compare to the current situation. Schemes of both the configurations are presented in figure 7.16. Currently the position of the desuperheater is at the same level of the condenser and the company had concern about the instability of the operation if any condensed liquid is trapped in the bottom part of the heat exchanger. If it is installed to an higher level compare to the condenser, it helps to the liquid to be drained out more constantly. In addition the liquid trapped can be also be related with oil accumulation. This concert was not supported by any experimental observation, since only few test with the desuperheater ac-

tive has been performed during the experimental campaign. Since tests in this condition are considered of secondary importance, they are not presented during this thesis work.

3. *Expansion valve replacement*: The current expansion valve is going to be replaced with an other thermostatic one offered by CIAT, designed truly for propane. To avoid every type of the problem with the regulation of the superheating, a electronic expansion is going to be installed in parallel with the other. The first tests will be performed using the latter valve, giving the possibility to regulate very accurately the outlet condition of the evaporator. Whenever good performance will be registered, the system will be switched to the new thermostatic expansion valve to check if the machine reacts as expected

4. *Receiver installation*: A receiver is going to be installed in the system for the optimization of the charge in the system avoiding the problem of charge/discharge refrigerant with the possibility to switch between test in full and half mode. It does not mean a receiver suggestion n the final commercial prototype since it increase the amount of the flammable refrigerant in the system and the risk as well. It is an experimental requirement to improve the quality of the result obtain and semplified the management of the experimental campaign.

5. *Sight glasses disposal*: Moving the sigh glass right before the expansion valve. The current position of the expansion valve is right after the condenser, therefore, there is a filter dryer and a liquid line before the sight glass and the expansion valve. In this way is possible to have a better check about the presence of bubble on the liquid line.

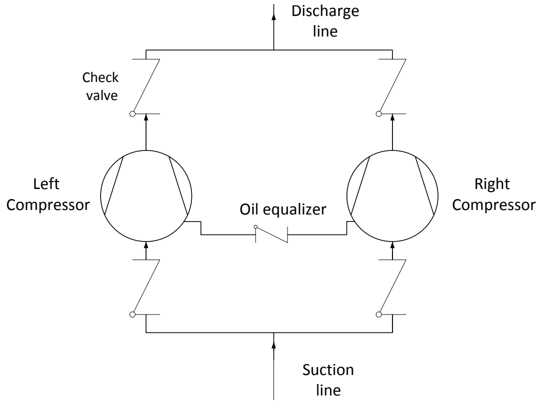


Figure 7.15: Proposal scheme for the installation of the check valves on the compressors pipe

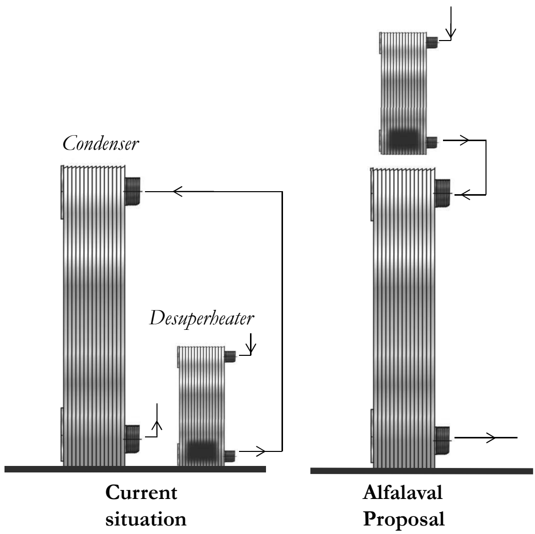


Figure 7.16: Comparison of the current installed position of the desuperheater and the proposal from the heat exchanger manufacturer Alfaval.

Chapter 8

Conclusions and recommendations

This thesis work has been conducted with the target of perform the experimental evaluation of large propane heat pumps tested in KTH for the European project NxtHPG. The work can be summarized in four different tasks:

- Test rig investigation and modification to fulfil the experimental requirements;
- Perform tests of the prototypes according to specific boundary condition;
- Performance analysis of the heat pump and its components;
- Suggest further improvements for the next experimental campaign.

The analysis of the functionality of the test ring manifests the requirement of design modification of some hydraulic loops in the system. After the implementation of the modification, the new tests shows the possibility to achieve the required flow rate and the right boundary condition as a consequence.

The prototypes demonstrated to have potentially the expected good performance with the two tandem compressors simultaneously running. On the other hand, a drop of the performance has been registered during the experimental campaign during the later stage in the project after the unit was run with single compressor. On the contrary, tests with one compressor working showed always poor results.
The analysis of the single HP components did not show clear signs of internal problematic related to the components design. In the beginning of the experimental campaign, all the components behaved according to the expectation, but in further tests some components exhibit ambiguous results:

1. *Compressor:* The compressor seems to work with a flow rate different compare to the estimation from the condenser heat balance. Some compressor parameters are truly near to the expectation, other parameters denote unexpected and undesirable compressor performance. Minor internal damage as check valve leakage can be a hypothesis, as consequence of the use of the compressor in a tandem layout.

2. *Condenser and Evaporator:* The heat exchangers seem to be designed property with very low approach temperature. On the other hand they are truly oversized for partial load and it is compatible for an experimental campaign. For a commercial exploitation of the prototypes the size can be reduced and optimized from an economical point view.

3. *3-way valve:* Problem in the regulation of the superheat only at the end of the experimental campaign. The valve was not design for propane from the beginning.

Two different explanations have been investigate to clarify the strange behaviour of the system: inert gas in the system and back-flow in the non working compressor. The first hypothesis assumes that, liquid propane goes though the non working compressor. According to the second hypothesis, a significant amount of inert gas is present in the system. They can explain some of the experimental observations but they can not provide a complete explaining analysis.
The result of the experimental campaign was presented during the annual meeting for the NxtHPG project with all the partners. A series of further improvements was proposed and discussed with the manufacture companies involved. The aim of the improvements is to confirm or deny the hypothesis above and add more flexibility in the system in reason of contrast the difficulties of the new phase of the experimental campaign.
The software IMST-ART, used to predict the performance of the heat pump, is demonstrated to be a fast and useful tool. The software predictions are really close to the experimental measurement with propane as refrigerant. The tool *"stand alone HX"*, applied for brazed plate heat exchanger, provides useful information to perform the analysis whenever a manufacturer software is not available.

Appendix A

Experimental result for tests 1 to 6

Summary table			1	2	3
File Description (Remarks in comments) (*) Direct measurement			2.5.. kg - 45 0 - 2 comp - DS OFF - v31c.xlsm	2.7.. kg - 37 0 - 2 comp - DS OFF - v31c.xlsm	2.7...kg - left comp - 37 0 - DS OFF - v31c
Pressures	HP (*)	bar	15,48	12,74	13,86
	LP (*)	bar	3,58	3,41	4,32
Saturated temperatures	T_{cond}	°C	45,40	36,90	40,53
	T_{evap}	°C	-8,88	-10,38	-3,00
Temperatures level water / secondary fluid side (tests conditions)	$T_{w,out,cond}$ (*)	°C	44,83	36,43	36,70
	$T_{w,in,cond}$ (*)	°C	37,94	28,31	34,32
	$T_{sf,in,evap}$ (*)	°C	-0,50	-0,75	0,06
	$T_{sf,out,evap}$ (*)	°C	-5,43	-6,09	-1,81
Flow rates water	Cond, flow rate (*)	m^3/h	5,41	4,69	5,32
	DS flow rate	m^3/h	0,00	0,00	0,00
Refrigerant cycle	Condenser capacity	kW	42,98	44,04	14,58
	Desuperheater capacity	kW	0,00	0,00	0,00
	Cooling capacity	kW	29,97	32,27	10,88
	Subcooling	K	3,38	4,76	6,53
	Superheating	K	9,20	8,64	4,62
	COP	-	3,07	3,56	2,03
	COP Carnot	-	5,87	6,56	7,21
	Carnot Efficiency	-	0,52	0,54	0,28
	R290 mass flow	kg/s	0,11	0,11	0,04

Summary table			4	5	6
File Description (Remarks in comments) (*) Direct measurement			2.7...kg - left comp - 33 0 - DS OFF - v31c.xlsm	2.5 kg 2 compr 45 0 v32v2.xlsm	2.5 kg 2 compr 45 5 v32v2.xlsm
Pressures	HP (*)	bar	12,97	16,94	16,67
	LP (*)	bar	4,29	3,49	3,68
Saturated temperatures	T_{cond}	°C	37,67	49,48	48,76
	T_{evap}	°C	-3,24	-9,70	-8,05
Temperatures level water / secondary fluid side (tests conditions)	$T_{w,out,cond}$ (*)	°C	33,09	45,31	44,91
	$T_{w,in,cond}$ (*)	°C	30,59	41,40	39,60
	$T_{sf,in,evap}$ (*)	°C	0,07	0,30	5,10
	$T_{sf,out,evap}$ (*)	°C	-2,06	-1,53	1,71
Flow rates water	Cond, flow rate (*)	m^3/h	5,33	7,05	5,80
	DS flow rate	m^3/h	0,00	0,00	0,00
Refrigerant cycle	Condenser capacity	kW	15,39	31,76	35,38
	Desuperheater capacity	kW	0,00	0,00	0,00
	Cooling capacity	kW	11,65	21,80	24,92
	Subcooling	K	7,48	7,54	8,50
	Superheating	K	4,86	11,45	14,39
	COP	-	2,23	2,14	2,39
	COP Carnot	-	7,60	5,45	5,67
	Carnot Efficiency	-	0,29	0,39	0,42
	R290 mass flow	kg/s	0,04	0,08	0,09

Appendix B

AlfaLaval HEs datasheet

The datasheet information for the heat exchanger model *AC220EQ-76AM-F* produced by Alfalaval are reported.
The same model is used as condenser and evaporator, so two versions of the datasheet are presented. They are related to the nominal condition for the heat pump in heating mode, one working as condenser and the other one working as evaporator.

Brazed Plate Heat Exchanger

Technical Specification

Model : AC220EQ-76AM-F(32871 1877 5)
ItemName : CASE2 Cond 45 kW Heat modeDate : 2015-01-21
Units : 1

		Hot Side Primary side(S4)	Cold side Secondary side
Fluid		Propane	Water
Mass flow rate	kg/s	0.1267	2.156
Fluid Condensed/Vapourized	kg/s	0.1267	0.000
Inlet temperature	°C	72.0	40.0
Dew point	°C	48.0	
Outlet temperature(vapor/liquid)	°C	48.0/45.5	45.0
Operating pressure(in/out)	bara	17.1/17.1	
Pressure drop	kPa	0.317	7.49
Velocity connection(in/out)	m/s	2.05/0.402	1.31/1.31
Heat exchanged	kW	45.00	
Heat transfer area	m²	8.29	
OHTC clean conditions	W/(m²*K)	1811	
OHTC service	W/(m²*K)	1029	
Fouling resistance*10000	m²*K/W	0.0	
Margin	%	76.1	
Mean Temperature Difference	K	5.3	
Relative directions of fluids		Countercurrent	
Number of passes		1	1
Materialplate/ brazing		Alloy 316 / Cu	
ConnectionS1 (Cold-out)		Threaded (External)/ 2" ISO 228/1-G (B23) Alloy 304	
ConnectionS2 (Cold-in)		Threaded (External)/ 2" ISO 228/1-G (B23) Alloy 304	
ConnectionS3 (Hot-out)		Soldering/ 1"1/8 (L55) Alloy 304	
ConnectionS4 (Hot-in)		Soldering/ 2 1/8" (D21) Alloy 304	
Pressure vessel code		PED	
Design pressure at 90.0 Celsius	Bar	37.0	37.0
Design pressure at 225.0 Celsius	Bar	30.0	30.0
Design temperature	°C	-196.0/225.0	
Overall length x width x height	mm	221 x 190 x 616	
Net weight, empty / operating	kg	34.6 / 48.0	
Package length x width x height	mm	365 x 210 x 700	
Package weight	kg	2.695	

Performance is conditioned on the accuracyof customers data and customers abilityto supply equipment and products in conformity therewith.

Brazed Plate Heat Exchanger
Technical Specification

Model : AC220EQ-76AM-F(32871 1877 5)
ItemName : CASE2 Evap 32 kW Heatmode Date : 2015-01-21
Units : 1

		Hot Side **Secondary side**	**Cold side** **Primary**
side(S4)			
Fluid		20.0% Eth.glycol	Propane
Mass flow rate	kg/s	2.753	0.1266
Fluid Condensed/Vapourized	kg/s	0.000	0.08078
Inlet temperature	°C	0.0	-6.6
Dew point	°C		-7.0
Outlet temperature(vapor/liquid)	°C	-3.0	-2.0
Operating pressure(in/out)	bara	/	4.19/3.81
Pressure drop	kPa	14.1	38.6
Velocity connection(in/out)	m/s	1.60/1.60	7.85/8.21
Heat exchanged	kW	32.00	
Heat transfer area	m²	8.29	
OHTC clean conditions	W/(m²*K)	1711	
OHTC service	W/(m²*K)	1068	
Fouling resistance*10000	m²*K/W	0.0	
Margin	%	60.3	
Mean Temperature Difference	K	3.6	
Relative directions of fluids		Countercurrent	
Number of passes		1	1
Materialplate/ brazing		Alloy 316 / Cu	
ConnectionS1 (Hot-in)		Threaded (External)/ 2" ISO 228/1-G (B23) Alloy 304	
ConnectionS2 (Hot-out)		Threaded (External)/ 2" ISO 228/1-G (B23) Alloy 304	
ConnectionS3 (Cold-in)		Soldering/ 1"1/8 (L55) Alloy 304	
ConnectionS4 (Cold-out)		Soldering/ 2 1/8" (D21) Alloy 304	
Pressure vessel code		PED	
Design pressure at 90.0 Celsius	Bar	37.0	37.0
Design pressure at 225.0 Celsius	Bar	30.0	30.0
Design temperature	°C	-196.0/225.0	
Overall length x width x height	mm	221 x 190 x 616	
Net weight, empty / operating	kg	34.6 / 48.4	
Package length x width x height	mm	365 x 210 x 700	
Package weight	kg	2.695	

Performance is conditioned on the accuracyof customers data and customers abilityto supply equipment and products in conformity therewith.

Appendix C

HEs analysis result

The complete results of the heat exchanger analysis in presented in this section. They are related to the condenser and evaporator of six different tests, twelve different figure in total.. Each figure consist of the 1D internal trend of the following heat exchanger parameters:

- Temperature profile of the hot fluid;
- Temperature profile of the cold fluid;
- Refrigerant vapour quality;
- Local UA-value.

The six tests are part of three experimental periods: Tandem compressor, first period (TC1); One compressor (OC); Tandem compressor, second period (TC2). For more detail, see the experimental chronology in chapter 6

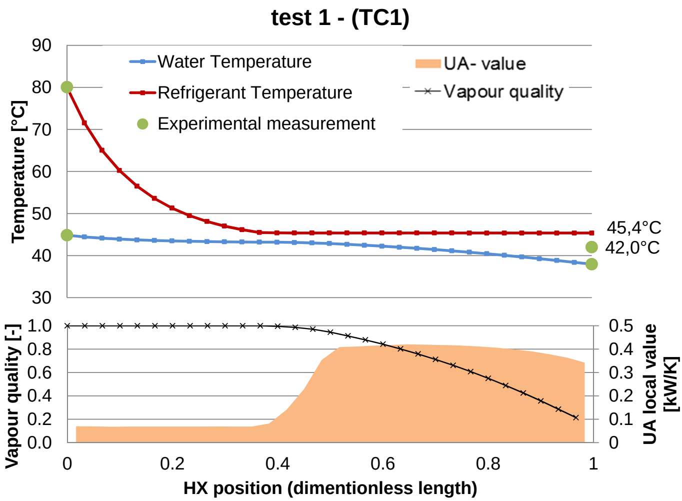


Figure C.1: Result for the condenser in test 1 (TC1 test group)

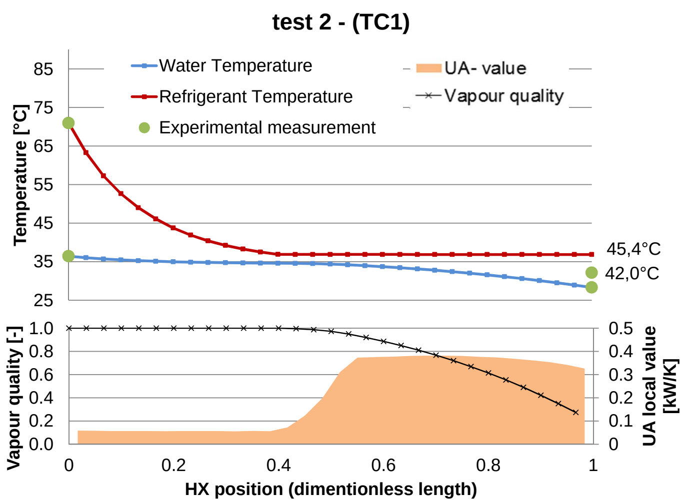


Figure C.2: Result for the condenser in test 2 (TC1 test group)

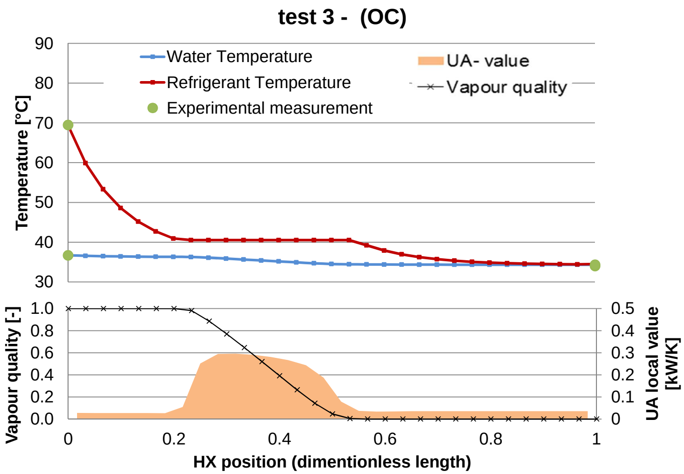


Figure C.3: Result for the condenser in test 3 (OC test group)

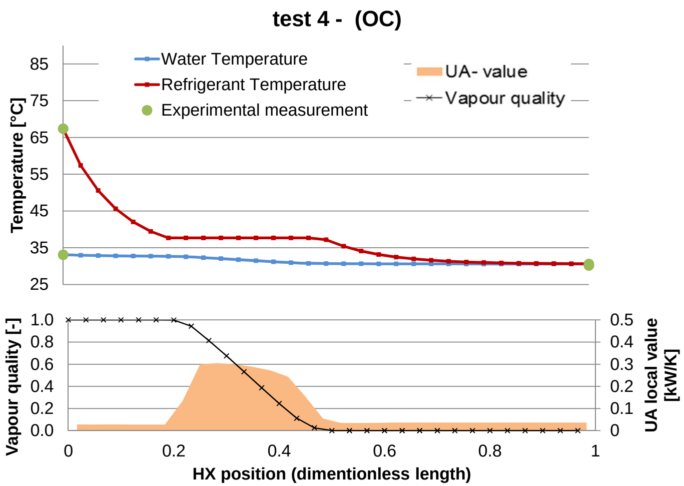


Figure C.4: Result for the condenser in test 4 (OC test group)

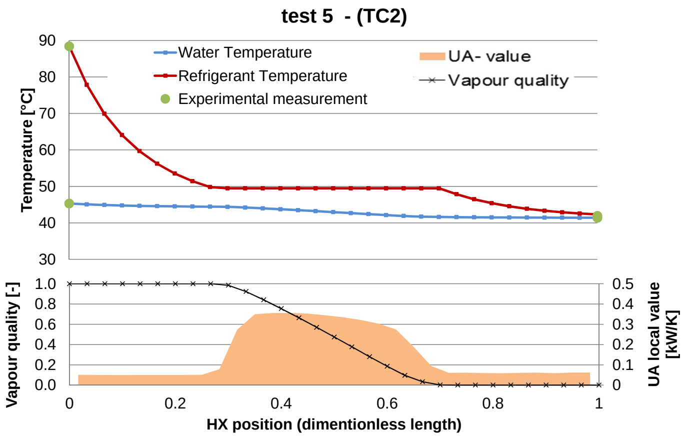


Figure C.5: Result for the condenser in test 5 (TC2 test group)

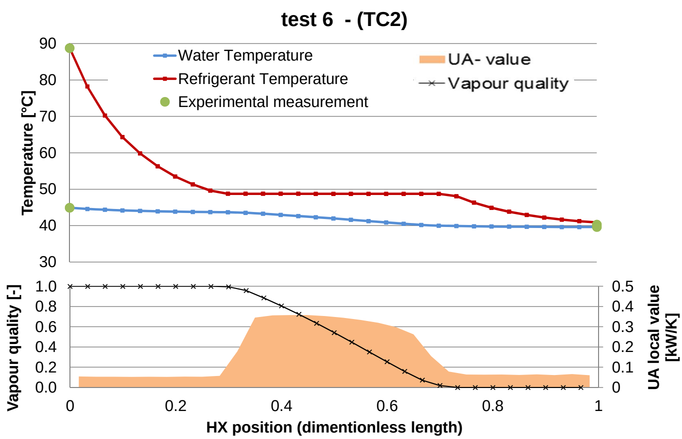


Figure C.6: Result for the condenser in test 6 (TC2 test group)

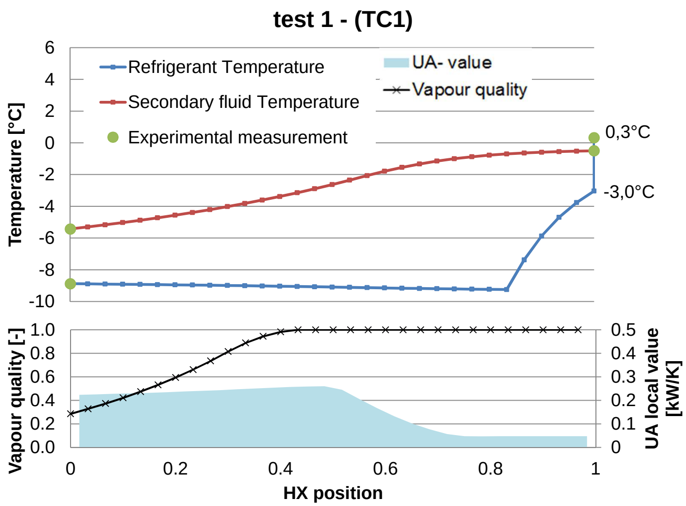


Figure C.7: Result for the evaporator in test 1 (TC1 test group)

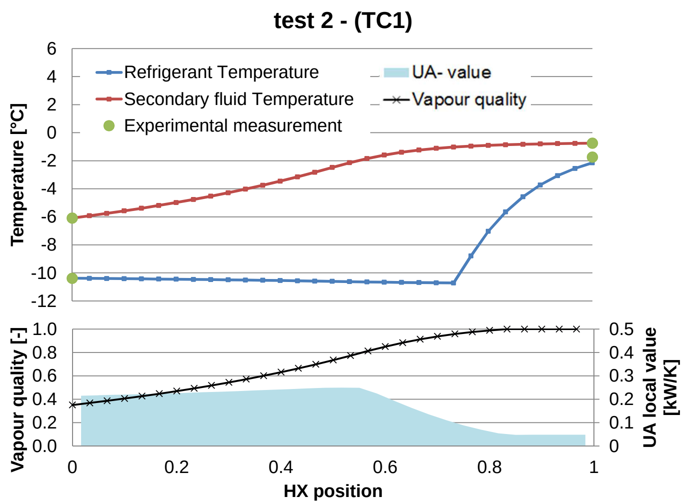


Figure C.8: Result for the evaporator in test 2 (TC1 test group)

Figure C.9: Result for the evaporator in test 3 (OC test group)

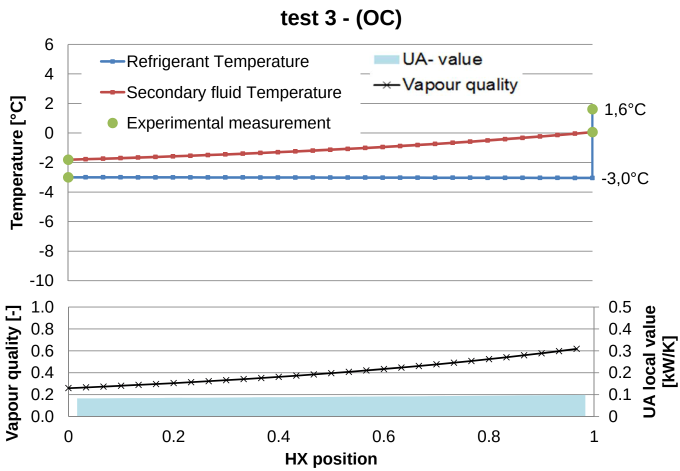
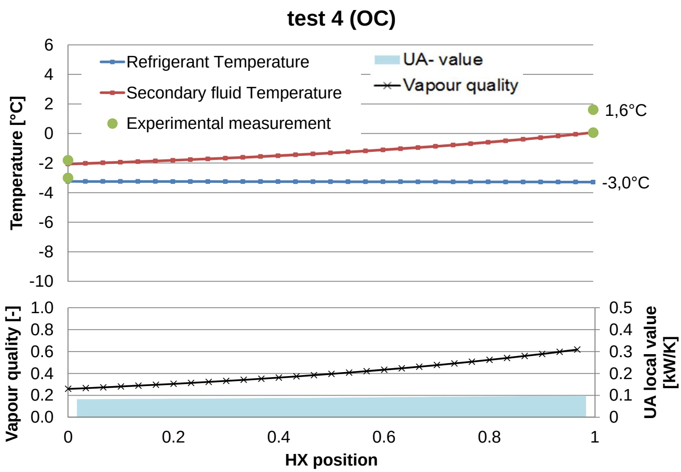


Figure C.10: Result for the evaporator in test 4 (OC test group)



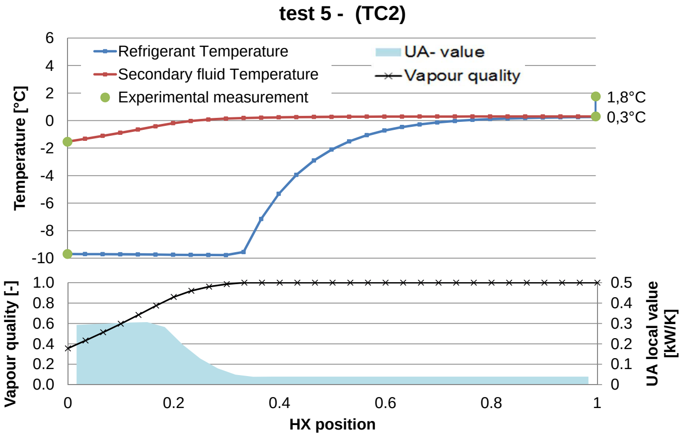


Figure C.11: Result for the evaporator in test 5 (TC2 test group)

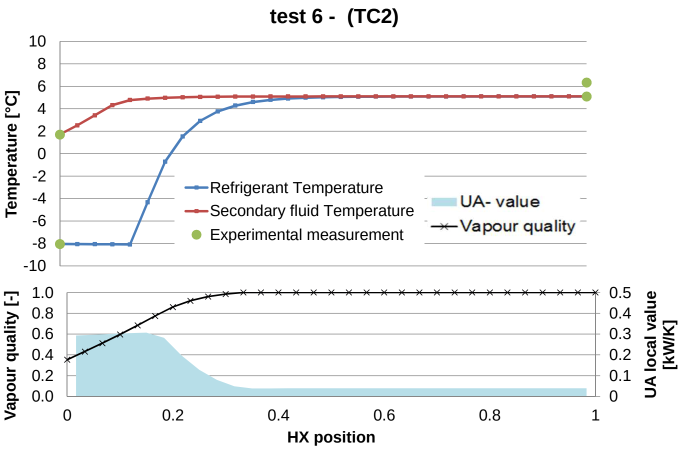


Figure C.12: Result for the evaporator in test 6 (TC2 test group)

Appendix D

Test rig drawings

The test rig drawings for the two heat pumps in Cases 1 and 2 are presented in this appendix. It includes:

- Schematic diagram of test rig CASE 1 in its initial version (version A, figure D.1) and after the hydraulic modification (version B, figure D.2);
- Schematic diagram of the climate chamber in CASE 1, a side view (figure D.3) and a top view (figure D.4);
- Schematic diagram of test rig CASE 2 in its initial version (version A, figure D.5) and after the hydraulic and system modification (version B, figure D.6)

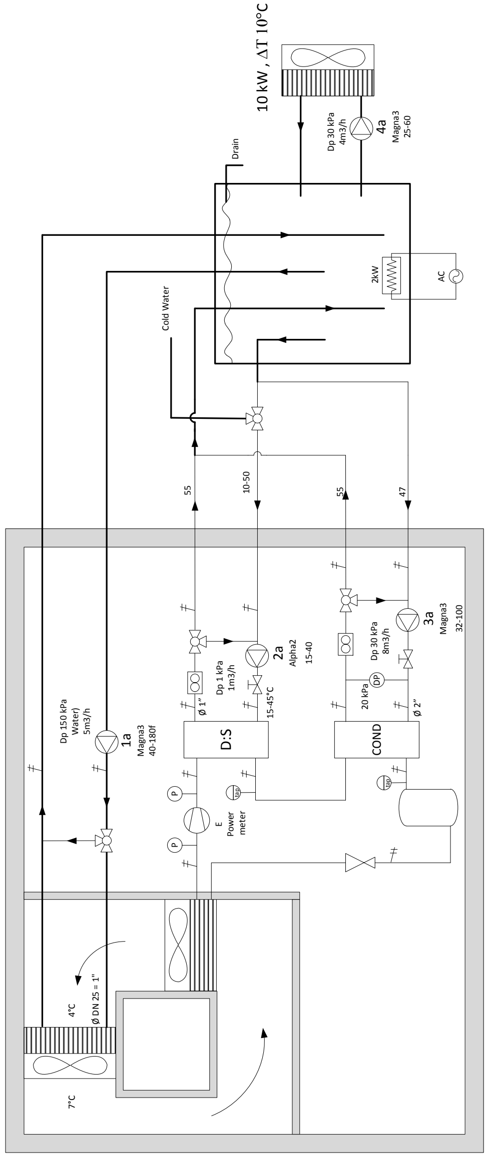


Figure D.1: Schematic diagram of test rig CASE 1, version A

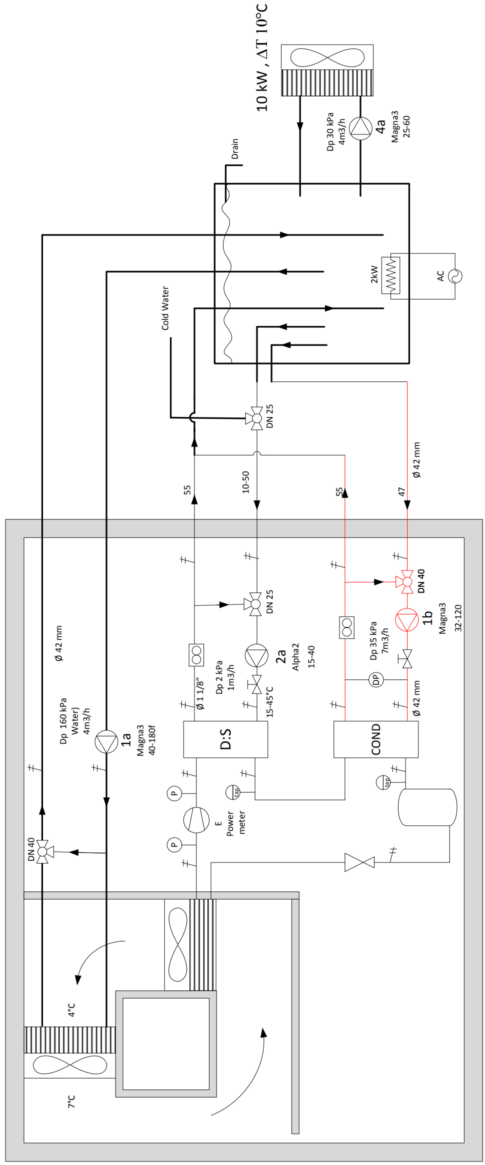


Figure D.2: Schematic diagram of test rig CASE 1, version B. In red the modification compare to versione A.

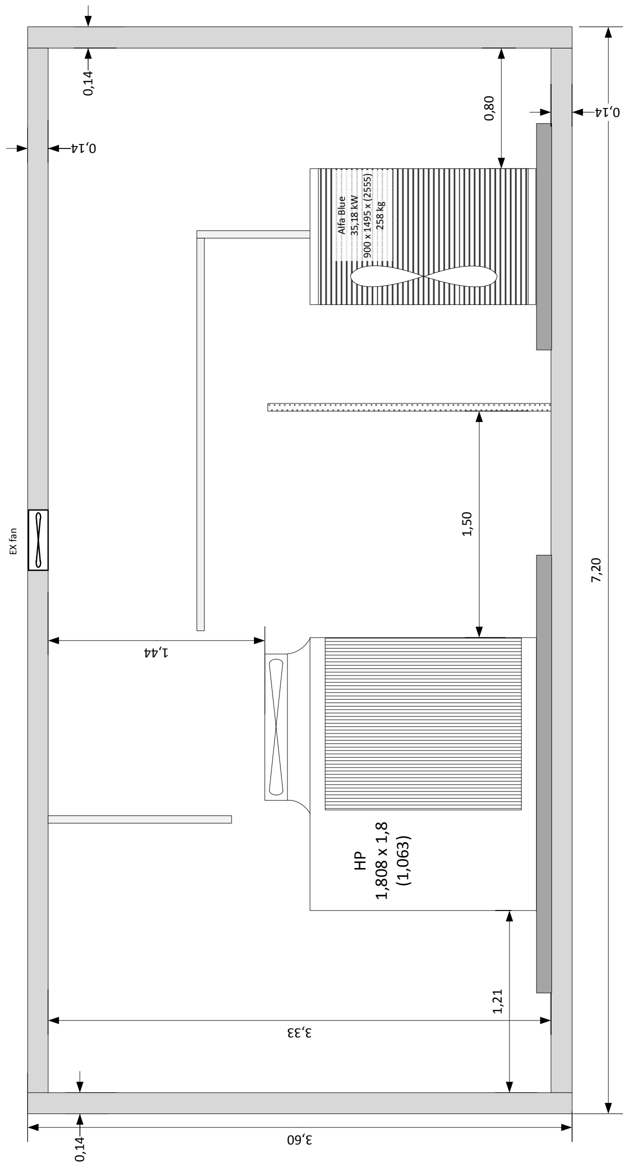


Figure D.3: Schematic diagram of the climate chamber in CASE 1, side

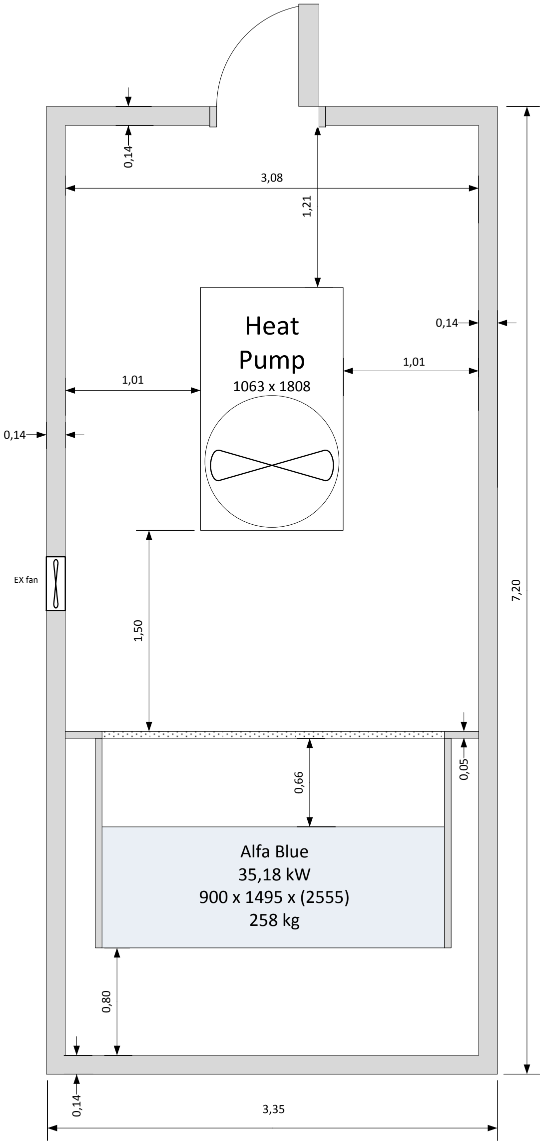


Figure D.4: Schematic diagram of the climate chamber in CASE 1, top

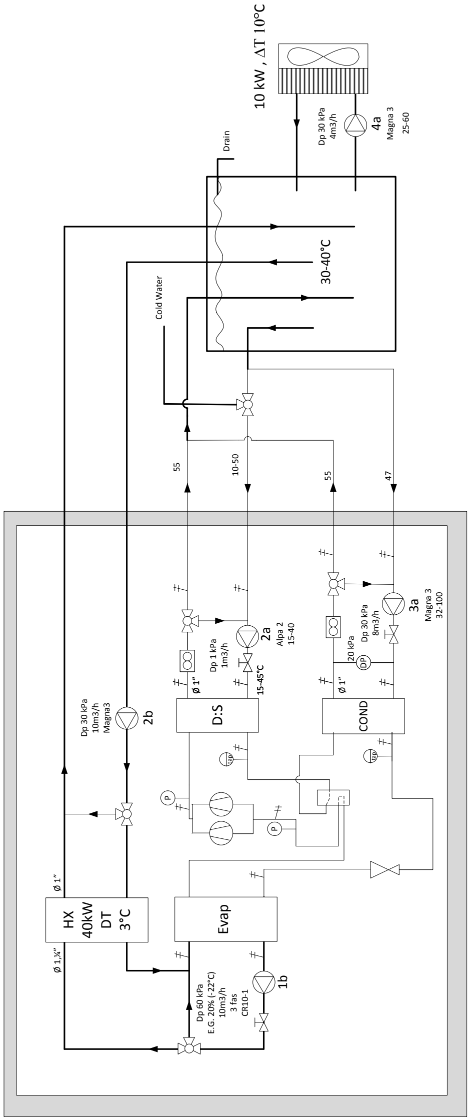


Figure D.5: Schematic diagram of test rig CASE 2, version A

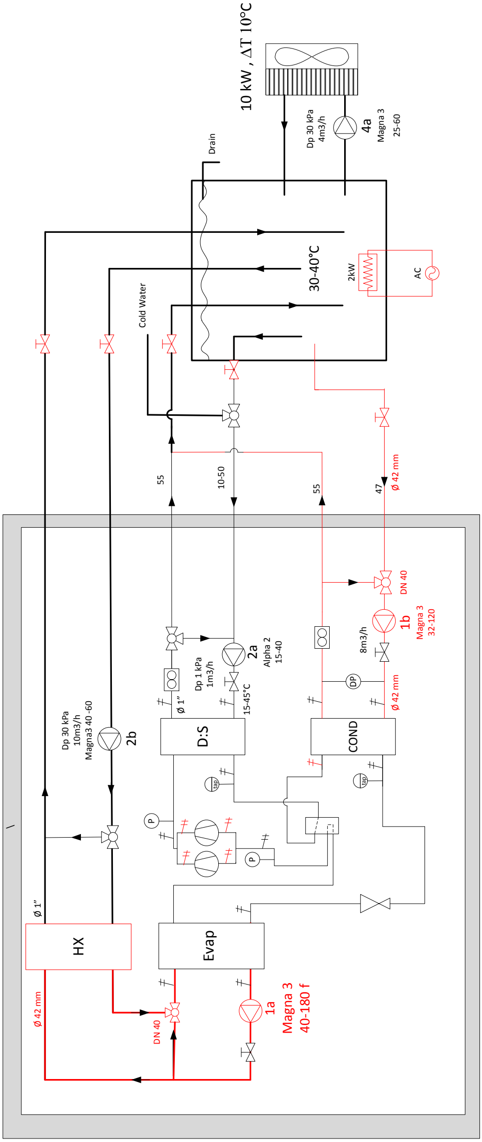


Figure D.6: Schematic diagram of test rig CASE 2, version B

Acronym

NxtHPG Next Generation of Heat Pumps working with Natural fluids

HP Heat Pump

HCs Hydrocarbon refrigerants

HFC HydroFluoroCarbon refrigerants

CFC ChloroFluoroCarbon refrigerants

HCFC HydroChloroFluoroCarbon refrigerants

HEs Heat Exchangers

TEWI Total equivalent warming impact

ODP Ozone depletion potential

GWP Global Warming potential

SPF Seasonal performance factor

DHW Domestic hot water

GSHP Ground source HP

SEWTLE Semi Explicit method for Wall Temperature Linked Equations

LMTD Logarithmic Mean temperature difference

MTD Mean temperature difference

TC1 Tandem compressors, period 1

TC2 Tandem compressors, period 2

OC One compressor

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