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TESI DI LAUREA

Dimensionamento di un banco di prova per la caratterizzazione di una pompa di calore acqua-acqua per la produzione di acqua calda sanitaria

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Alla mia famiglia,

Grazie di tutto

Index:

Summary:

The following thesis concerns the design, dimensioning and construction of a test rig for an innovative high capacity water to water heat pump, working with propane as refrigerant. The test rig has to be able to supply water to the evaporator at the desired temperature and to drain the produced heat flux in the most efficient way simulating the sanitary hot water consumption by a consumer.

The work is part of the European project NxtHPG: “Next Generation of Heat Pumps working with Natural fluids” presented in paragraph 1.1.

The dimensioning is focused in four secondary circuits: the first provides the heat source at the evaporator side while the other three, that are two hydraulic loops and one refrigerant cycle, simulate the sanitary hot water demand of the consumer. The goal is to build a system that can test different operative conditions and different heat pump configurations in order to further investigate the optimal working conditions of the heat pump.

The work is split in different parts. The first consists of a theoretical section in which the design of the circuits and the dimensioning of the different components are done; the used software are IMST-ART [1] to simulate the working of the two refrigerant cycles in the different operative points, and EES [2] to calculate the operative parameters in the secondary hydraulic circuits.

Further the assembly of the various circuits components is carried out at the Thermal Laboratory in the IIE Institute of the Polytechnic University of Valencia (Instituto de Ingeniería Energética de la Universitat Politècnica de València: UPV).

The thesis work has been carried out in the Polytechnic University of Valencia within the Erasmus European exchange program in the second semester of the academic year 2013-2014.

Sommario:

La seguente tesi riguarda la progettazione, il dimensionamento e la costruzione di un banco di prova per la caratterizzazione di un'innovativa pompa di calore acqua-acqua ad elevata capacità funzionante con propano come refrigerante. Il banco di prova deve essere in grado di fornire acqua all'evaporatore alla temperatura desiderata e di smaltire nel modo più efficiente il flusso di calore prodotto simulando il consumo di acqua calda sanitaria da parte di un'utenza.

Il lavoro è parte del progetto Europeo NxtHPG : "Next Generation of Heat Pumps working with Natural fluids" presentato al paragrafo 1.1.

Il dimensionamento è incentrato in quattro circuiti secondari: il primo si occupa di fornire la sorgente di calore lato evaporatore mentre gli altri tre, cioè due circuiti idraulici e un ciclo frigorifero, simulano la richiesta di acqua calda da parte dell'utenza. L'obiettivo è costruire un sistema che possa testare la pompa di calore in varie condizioni operative e nelle sue diverse configurazioni al fine di poter in seguito studiare le condizioni ottimali di funzionamento di tale pompa di calore.

Il lavoro è suddiviso in diverse parti. La prima consiste in una sezione teorica nella quale è studiata la progettazione dei circuiti e il dimensionamento dei diversi componenti; i software utilizzati sono IMST-ART [1] per simulare il funzionamento dei due circuiti refrigeranti nelle varie condizioni operative, ed EES [2] per calcolare i parametri nei circuiti idraulici secondari.

In seguito l'assemblaggio dei diversi componenti dei circuiti è svolto nel laboratorio termico nell'Istituto IIE dell'Università Politecnica di Valencia (Instituto de Ingeniería Energética de la Universitat Politècnica de València: UPV).

Il lavoro di tesi è stato svolto presso l'Università Politecnica di Valencia all'interno del progetto Europeo Erasmus nel secondo semestre dell'anno accademico 2013-2014.

Introduction

1.1 The NxtHPG project

The paragraph has as objective to introduce the NxtHPG[1] project, the following information are available in the official site [3] of the undertaking.

This thesis work is part of the European project NxtHPG that has as purpose to find and develop the most relevant cases in which the employment of a natural refrigerant in an innovative and high efficiency heat pump can lead to a fast commercial exploitation.

The general target of the project is to realize a heat pump with better characteristics than actual commercial heat pumps: from the point of view of the seasonal performance factor an increase of the 10÷20% is the goal, and from the point of view of the carbon footprint the objective is to reduce the total equivalent warming impact TEWI of the 20%. At the same time the new machine has to be competitive in the market so the cost does not have to be a lot higher (in the range of a 10%) than the actual commercial alternatives; also are preferable an efficient capacity of modulation in the generated heat load, and the possibility of integration with other renewable sources.

The choice of using natural refrigerants instead of the most used HCFs is because, as said in the NxtHPG's official site[2], the «*Development of equipment specifically designed for natural fluids could lead to even higher efficiencies than the ones obtained with synthetic refrigerants, contributing at the same time to the reduction of the energy consumption and to the practical elimination of direct emissions, resulting in a double reduction of the global environmental impact of heat pumps*».

The natural refrigerants taken into account are of two kinds so the project is split into two different paths: one studies the employment of hydrocarbons (HC's) and the other of carbon dioxyde (CO_2) as refrigerants.

In the past the large employ of these fluids as refrigerants used to be contained by some critical factors that the project intends to overcome.

Using CO_2 as refrigerant leads to expensive equipment but as the cost is strongly dependent on their production, this barrier could fall down if the developed solutions would be produced in an industrial scale.

Using HC's as refrigerant sometimes could be felt like dangerous by the equipment manufacturers because of the risk to guarantee a component that works with a flammable fluid; despite that the large penetration in all over the word of small domestic systems working with HC's, i.e. isobutane, demonstrates that these equipment have reached an adequate level of safeness. Furthermore in a commercial-industrial use the risks could also be lower because of the possibility to put these machines in open air and in restricted

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[1] NxtHPG: Next Heat Pumps Generation or Next Generation of Heat Pumps working with Natural fluids

[2] http://www.NxtHPG.eu/

areas. In NxtHPG project the study is concentrated in applications in which the extra cost due to safety measures can be compensated by the lower energy consumption.

As the project takes place in Europe and it is financed by the European Union, the target market of the developed heat pumps and the technologies used are dominated by EU industry and not by low price components coming from other countries; in this way the exploitation plan could rise up in EU market and then expand in the rest of the world. In fact the prospects for the future are that new restrictions on greenhouse gases and new regulations about the energy efficiency of the edifices will impose the substitution of obsolete equipment and the installation of low climate impact heating systems in new buildings like the ones taken into account in NxtHPG project.

The scientific coordinator of the project is the Polytechnic University of Valencia and the project structure is composed by three different teams that will work in three fields: the development of cycles working with CO_2 or hydrocarbons (HCs) as refrigerant and the development of heat exchangers (HEs). See Figure 1.

In every team there are European research institutions and European industrial companies specialized in the sector.

The firsts are UPV: Universitat Politècnica de Valencia; KTH: Royal Institute of Technology, EPFL: Ecole Polytechnique Fédérale de Lausanne, ENEA: Italian National Agency for New Technologies, Energy and Sustainable Economic Development; UNINA: Università degli Studi di Napoli Federico II; NTNU: Norwegian University of Science and Technology.

While the sector leaders companies are: Danfoss and Dorin as compressors manufacturers; LU-VE Group and Alfa Laval as heat exchangers manufacturers; Ciat and Enex as heat pumps manufacturers.

The project structure is explained in Figure 1.

The expected results are the identification of 4÷6 applications advantageous to EU industry that can satisfy the requirements about the feasible and safe use of natural refrigerants, about the possibility to join high efficiencies, about a fast commercial development thanks to a large potential market and a competitive cost solution.

Considering all of these purposes the project focuses in five promising cases from an energetic, technical, and economic point of view that will be studied and developed by the different research groups, all these cases are described in the Deliverable 1.3 of the project. The case that will be developed in UPV University and so the one that will be taken into account in this work is the number 3.

It focuses on a 50 kW water to water heat pump that uses as refrigerant propane; the heat source is a water loop at 10÷35 degrees depending on the kind of that source; two different possibilities are taken into account: sewage water loop at 10÷20 degrees or building condensation loop at 15÷35 degrees. The aim of case 3 is the production of sanitary hot water at 60 degrees.

Different return water temperature from 10 to 55 degrees will be studied to simulate different working conditions as explained in chapter 2.

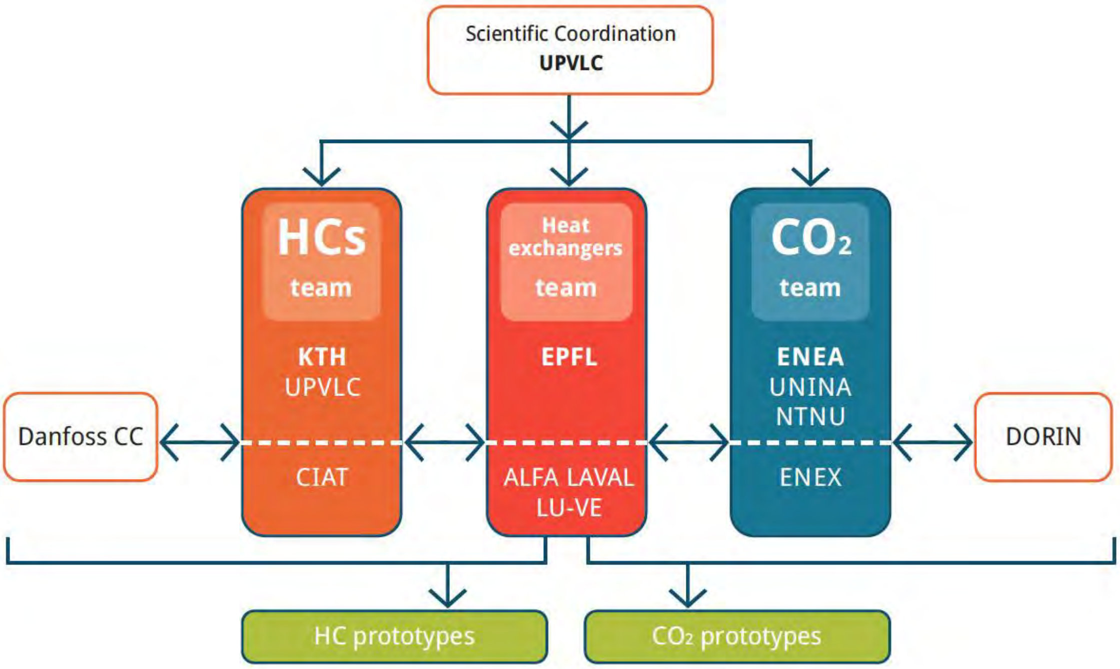


Figure 1: NxtHPG project structure

1.2 The working fluids for heat pumps

1.2.1 History

A heat pump is a vapour-compression refrigerating machine that works with an inverse cycle transferring heat from a colder source to a hotter sink through a working fluid that evaporates and condenses cyclically. The invention of the refrigerating machines is not a recent fact in effect the firsts were built in the middle of the 18th century and since that period a lot of different refrigerant fluids have been used and tested.

One of the first refrigerants used in large scale was ammonia but some improper characteristics like toxicity and corrosiveness drove to a search of a safer substitute that could provide for these disadvantages.

From the 1930s the synthetic fluids chlorofluorocarbons (CFCs) took the place of ammonia in the most part of applications thanks to their good thermodynamic properties, non-toxicity and stability.

More than forty years later, when CFCs like R11 and R12 were the most common refrigerant fluids, a publication of Molina and Rowland [4] connected their use as refrigerant to the destruction of ozone layer in the atmosphere.

They proved that the increased presence of bromine and chlorine in the atmosphere due to the production of the new refrigerants was the cause of the ozone depletion that was reducing the natural guard from the ultraviolet-B radiation.

To avoid the depletion of ozone in the atmosphere the use of CFCs and other high ODP gases has been gradually banned starting from 1987 with the international Montreal protocol and the following modifications; nowadays in the developed countries CFCs production is banned from 1996.

The CFCs were substituted by hydrochlorofluorocarbons (HCFC) like R22 that have a lower impact on the ozone and similar thermodynamic properties so they could be used in existing plants without big modifications on the circuits.

An even more improvement has been done using hydrofluorocarbons (HFCs) and other synthetic refrigerants in order to continue the path undertaken in Montreal Protocol.

Furthermore the global warming potential GWP of the refrigerants has been taken into account too because, also if the percentage quantity of these gases in the atmosphere is very low, their greenhouse effect is not negligible.

1.2.2 Environmental matter

The international agreements discussed in the previous chapter demonstrate the increasing attention of the western governments on environmental contents. The taken up choices are directed to solve primarily two global phenomena that nowadays are felt by the public opinion: the stratospheric ozone depletion and the increasing greenhouse effect.

In the following paragraphs will be introduced the two phenomena and the relative parameters internationally adopted to correlate the refrigerant gas emission to its environmental effects.

1.2.2.1 Stratospheric Ozone depletion

Ozone O_3 is a gas present at low concentration in the atmosphere, it is the triatomic form of the oxygen O_2 and much less stable than the latter; as the ozone layer in the atmosphere acts as barrier for a big part of the detrimental UV radiation emitted from the sun, its presence is fundamental for the life in the Earth.

In 1974 Mario Molina and F. Sherwood Rowland [4] associated the presence of fluocochemicals refrigerants in the atmosphere to the destruction of the ozone layer, they demonstrated that chlorine and bromine present in CFC can arrive in the atmosphere and react with ozone molecules destructing them.

The UV radiation can dissociate atoms of Cl from CFC giving rise to the following reactions in which the chlorine is not consumed and continues react with ozone.

$$Cl + O_3 \rightarrow ClO + O_2$$

$$ClO + O \rightarrow Cl + O_2$$

In 1985 a group of English scientists published the results of a series of atmosphere measurements over the Antarctic [5], showing that the thickness of the ozone layer in the upper atmosphere was clearly reducing from the starting of austral spring, when the Sun starts to shine again, after the long winter. This phenomenon had been observed for different years before the publication, and the data from the satellites in the previous years revealed an even more serious situation, in fact the ozone layer was decreasing year by year and a similar circumstance, also if in a smaller scale, had been observed in the Artic Pole too.

To compare the effect of different substances in the destruction of atmospheric ozone an indicator has been adopted, it is called ODP (Ozone Depletion Potential) and it takes into account the number of Cl and Br atoms present in the molecule, the lifetime of the substance in the atmosphere and the degradation mechanisms. The bigger is the number of these atoms and the bigger the lifetime, the higher will be the depletion impact; the value of the ODP indicator is referred to the R-11 impact so a value equal to one means the same impact of R-11 and a lower value, a lower impact. In other words the ODP value indicates the equivalent kilograms of R-11 that have to be released in the atmosphere to have the same impact as 1 kg of refrigerant.

In Figure 2 can be seen the values of ODP for some common refrigerant gases.

Nowadays, thanks to the international agreements about the gradual ban of ozone depleting gases, the ozone layer is reforming and the hole is decreasing [6].

1.2.2.2 The greenhouse effect

The average surface temperature of the Earth hails from equilibrium between the incoming solar energy and the heat radiated back to the Space; as the biggest part of this radiation is in the infrared field, some gases present in the atmosphere can absorb it warming the Earth and producing the greenhouse effect.

Several used refrigerant gases have high atmospheric lifetime and good absorption properties in infrared radiation so they are classified as greenhouse gases.

To compare the greenhouse effect of different substances in the atmosphere, the commonly used indicator is the global warming potential (GWP); it takes into account the absorption properties of the gas and also its lifetime in the atmosphere.

The GWP value is referred to the most present greenhouse gas in the atmosphere (CO_2). It compares the amount of heat absorbed by a certain mass of the gas in question, to the amount of heat absorbed by the same mass of CO_2. The GWP value for carbon dioxide is standardized to 1 and normally the GWP of another gas is calculated for an interval of 100 years, that is comparing the effect of the release in the atmosphere of 1 kg of CO_2 and 1 kg of the gas in question for a century.

In other words the GWP value indicates the equivalent kilograms of CO_2 that have to be released in the atmosphere to have the same impact in the considered range of time (normally 100 years).

In Figure 2 can be seen the values of GWP for some common refrigerant gases, it highlights that the release of some kilos of several of them has the same impact of the release of tons of CO_2.

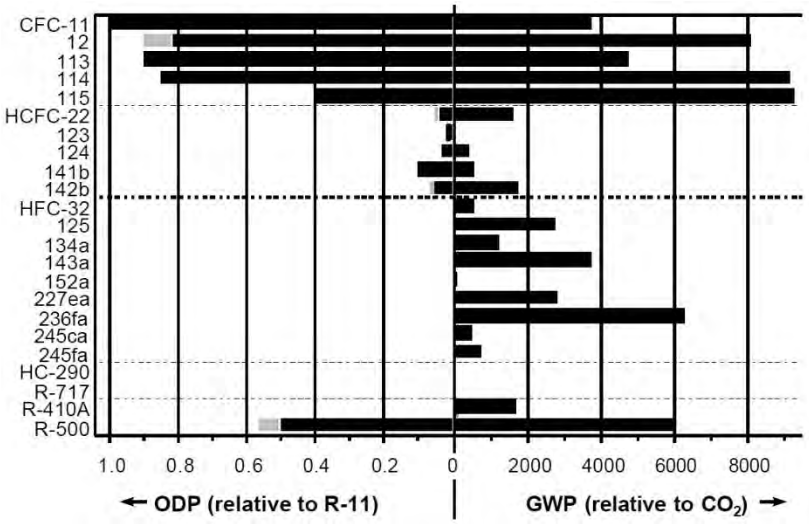


Figure 2: ODP and GWP values for some common refrigerant gases, from [7]. The grey bars are the highest values calculated by semi-empirical determination that contrast with modelled black bars from [8].

In the IPCC/TEAP Report of 2005 [9] is affirmed that the direct global warming impact of these manmade gases, calculated before the ban of CFCs, had a percentage contribution to the global release in CO_2 equivalent tons higher than 20%.

As heat pumps, air conditioners and refrigeration devices in general that work with refrigerant gases also use energy, in the calculation of the global warming impact we have to take into account both the refrigerant release and the emission of greenhouse gases like CO_2 in powering the devices.

That is possible using the TEWI indicator, i.e. Total Equivalent Warming Impact; it evaluates the direct effect due to the quantity of refrigerant used in the machine and scattered in the atmosphere, and also the indirect effect depending on the energy used to power the machine and the relative greenhouse gases emission.

$$TEWI = m * GWP + \propto_{CO_2} * \tau * e$$

Where:

m is the total mass of refrigerant released in the atmosphere during the whole life of the machine, considering installation, leakage, service and disposal losses,

GWP is referred to the used refrigerant gas;

$\propto_{CO_2}$ is the equivalent mass of CO_2 emitted in the atmosphere per electric energy unit, it depends on the energy production system of the country where the machine is installed;

τ is the lifetime of the machine;

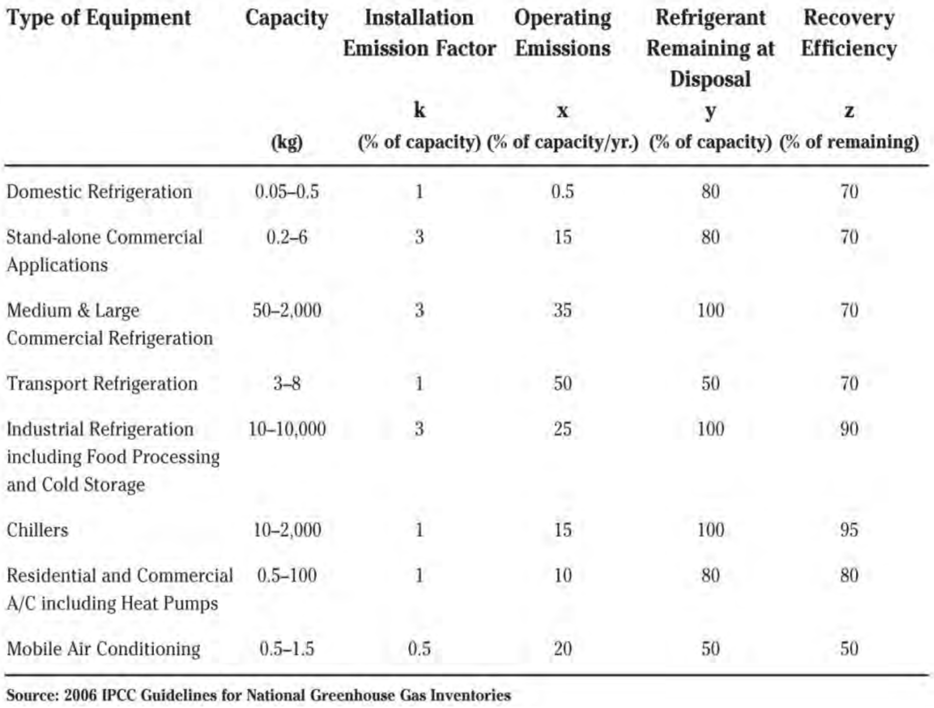
e is the average electric energy consumed in the time unit.

Considering this indicator, the direct effect of refrigerant emissions usually turns out to be much smaller than the indirect impact [10]; this statement is normally true but there are also exceptions like for example in mobile air conditioners and in commercial refrigeration where the refrigerant loss rates are on average higher.

Furthermore loss rates can be even reduced if the refrigerant is recovered for reuse or safe disposal reducing the production and so saving other resources.

In respect of TEWI perspective the gradual elimination of hydrofluorocarbons (HFCs) in chillers with low refrigerant releases provides only small gain, while an efficiency improvement could be more relevant for the greenhouse point of view [10]; contrary this is not true in applications where refrigerant losses are high.

In Table 1 colud be seen some default values relative to refrigerant emissions for refrigerating equipments.



Type of Equipment	Capacity (kg)	Installation Emission Factor k (% of capacity)	Operating Emissions x (% of capacity/yr.)	Refrigerant Remaining at Disposal y (% of capacity)	Recovery Efficiency z (% of remaining)
Domestic Refrigeration	0.05–0.5	1	0.5	80	70
Stand-alone Commercial Applications	0.2–6	3	15	80	70
Medium & Large Commercial Refrigeration	50–2,000	3	35	100	70
Transport Refrigeration	3–8	1	50	50	70
Industrial Refrigeration including Food Processing and Cold Storage	10–10,000	3	25	100	90
Chillers	10–2,000	1	15	100	95
Residential and Commercial A/C including Heat Pumps	0.5–100	1	10	80	80
Mobile Air Conditioning	0.5–1.5	0.5	20	50	50

Source: 2006 IPCC Guidelines for National Greenhouse Gas Inventories

Table 1: Default emission factors for refrigerator and air conditioning equipments, from [11].

Where: the installation emission factor is the average percentage of refrigerant emitted during the installation of the machine; the operating emissions are calculated during the working operation; the refrigerant remaining at disposal is the quantity of refrigerant that is possible to recover during the decommissioning; and the recovery efficiency indicates the quantity of disposal refrigerant that can be reused for other employments.

In conclusion also if the two explained phenomena of stratospheric ozone depletion and greenhouse effect are different, it's not possible to take care about the ones separately, and besides without considering that the choices made to solve one issue could have fallouts on the other.

In some cases the taken efforts for the reduction of one problem, lead to the lowering of also the other.

For example the stratospheric ozone depletion acts easily with a lower concentration of CO_2 in the atmosphere. An increase of greenhouse effect cools the stratosphere with a consequent increase of ice crystals formation; their presence is detrimental for the ozone layer because bromine and chlorine react easily with O_3 when ice crystals are present.

Contrary banning refrigerant gases with a low ODP to protect the ozone layer could worsen the global warming. E.g. the refrigerant R123 has a low ODP (see Figure 2) and permits to join high performances [7]; banning it precludes a cost effective option to reduce global warming.

Summarising only an open minded policy that comprehends the link between the different phenomena of ozone depletion and global warming will be useful to solve these problems.

1.2.3 Refrigerant properties

An ideal refrigerant would be non-toxic, non-flammable, stable and inert inside a system to assure safeness requirements also in case of leakage.

Some other thermal properties are desired like low vapour heat capacity, low viscosity, high thermal conductivity, low pressure ratio for the defined condensing and evaporating temperature, low density.

Furthermore other characteristics are preferable for practical reasons: good oil solubility and compatibility with common lubricants, compatibility with the materials normally used in the refrigeration systems, low freezing point and low cost production.

Other optimal characteristic could be found, but there are no refrigerants satisfying neither only all these partial requirements.

The future synthesis of new ideal refrigerants is doubtful [7] for the reason that desired properties for the molecule are in conflict.

Increasing the hydrogen content of a refrigerant molecule decreases its atmospheric lifetime (this expedient has been adopted to decrease the ODP of CFC gases obtaining the more environmental friendly HCFCs) but simultaneously increases the flammability. This can be noticed in Table 2 in fact CFCs lifetime is normally longer and they are low flammable because they do not contain hydrogen; contrary HCs are highly flammable and their life is shorter. HCFC and HFC properties are in the middle.

It should be noticed that if the atmospheric lifetime is short, the ODP and GWP are low because a big part of these gases will dissociate before arriving in the stratosphere where the ozone concentration is maximum; further the part of emitted gases that anyway arrives will have a shorter life and so a smaller impact on the global warming and ozone depletion.

Also the chlorine and fluorine content has a relevant impact on the molecule properties: increasing the chlorine involves an increase in the normal boiling temperature but above all it increases the ODP. In fact compounds that do not contain chlorine or bromine generally have an ODP near to zero.

Increasing the fluorine tends to reduce toxicity but on the other hand it generally increases GWP.

The previous considerations can be directly visualized in Figure 3 where can be seen the influence of hydrogen, chlorine and fluorine presence, towards to flammability, toxicity, lifetime, ODP and GWP.

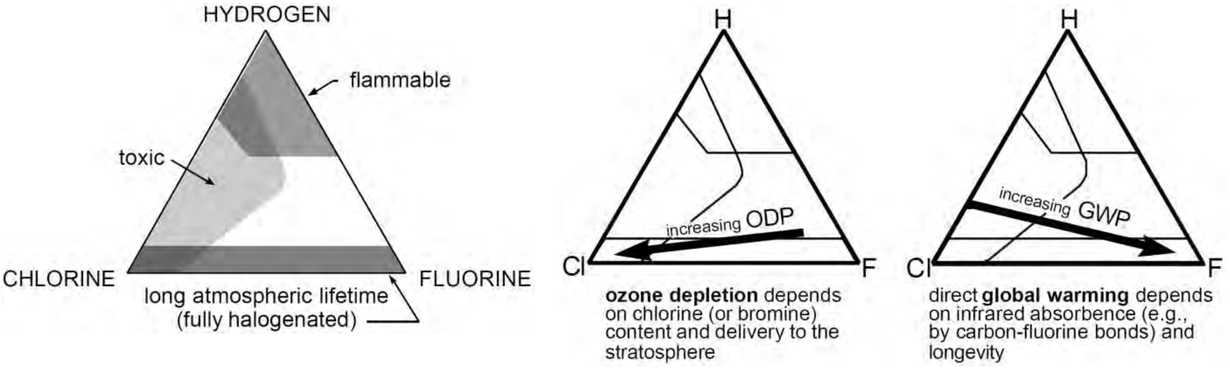


Figure 3: influence of hydrogen, chlorine and fluorine presence, towards to flammability, toxicity, lifetime, ODP and GWP. From [7].

1.2.3.1 Refrigerant classification

The refrigerants are designated by their chemical composition in order to clearly recognize them in an international way, the following classification rules are taken from the International Institute of Refrigeration [12].

An identifying number that consists of a prefix made up of letters and a suffix made up of digits is assigned to each refrigerant.

The prefix is every time composed by the letter R that means refrigerant, but in non-technical publications the letter C could be used to denote carbon, also preceded by B, C or F to indicate the presence of bromine, chlorine or fluorine. Compounds containing hydrogen must be preceded with letter H.

The suffix is different depending on the kind of gas:

- Hydrocarbons and derived, (example Rxyz)

x: number of carbons in the molecule reduced by one, if the value is zero it is omitted;

y: number of hydrogen atoms in the molecule plus one;

z: number of fluorine atoms.

Example: R290 for propane (C_3H_8)

For refrigerants containing bromine the letter B is added after the identification number followed by the number of atoms present.

The number of chlorine atoms could be found by subtracting the sum of fluorine, bromide and hydrogen atoms from the total number of atoms that can be connected to the carbon atoms.

In the case of isomers[3] each has the same number that can be followed by a letter indicating the grade of symmetry.

Example: R134 and R134a.

- Azeotropic[4] and zeotropic[5] mixtures:

Mixtures are designated by their respective refrigerant numbers and mass proportions. Refrigerants shall be named in order of increasing normal boiling points of the components.

Zeotropic mixtures shall be assigned an identifying number in the 400 series. This number designates which components are in the mixture but not the amount of each. To differentiate among zeotropes having the same components with different amounts (percent by mass), an uppercase letter shall be added as a suffix.

The numbers are in chronological order of the refrigerant's approval by ASHRAE.

Example: R410A (R32/R125 (50/50)) and R410B (R32/R125 (45/55))

- Miscellaneous organic compounds:

To miscellaneous organic compounds are assigned a number in the 600 series; numbers are given in numerical order.

Example: R600a, isobutene

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[3] Isomers: molecules that contain the same number of atoms of each element but have different arrangements of their atoms in the space.

[4] Azeotripic mixture: mixture of two or more liquids whose proportions cannot be altered by simple distillation.

[5] Zeotropic mixture: mixture that never has the same vapour phase and liquid phase composition at the vapour-liquid equilibrium state.

- Inorganic compounds:

To inorganic compounds shall be assigned a number in the 700 series, identification numbers are formed by adding the relative molecular mass of components to 700.

Example: R717 corresponds to ammonia which has a molecular mass of 17.

1.2.3.2 Safety group classifications

The safety classification, again from [12], consists of two characters: a capital letter corresponds to toxicity and a number to flammability.

Concerning the toxicity classification refrigerants are divided into two classes:

- Class A comprehends refrigerants for which toxicity has not been identified at concentrations less than or equal to 400 ppm;

- Class B comprehends refrigerants for which there is evidence of toxicity at concentrations below 400 ppm.

Concerning the flammability classification refrigerants are divided into three classes:

- Class 1 comprehends refrigerants that do not show flame propagation when tested in air at 21°C and 101 kPa;

- Class 2 comprehends refrigerants having a lower flammability limit of more than 0.10 kg/m3 at 21°C and 101 kPa and a heat of combustion of less than 19 kJ/kg;

- Class 3 indicates refrigerants that are highly flammable as defined by a lower flammability limit of less than or equal to 0.10 kg/m3 at 21°C and 101 kPa or a heat of combustion greater than or equal to 19 kJ/kg.

For zeotropic or azeotropic mixtures, characterized by possible changing on flammability and toxicity characteristics due to composition changes during fractionation, a safety group classification based on the worst case is adopted.

Example: R404A is classified A1

In the following Table 2 the characteristics of some refrigerants.

Refrigerant:	R12	R22	R410A	R407C	R290	R744
	CFC	HCFC	HFC	HFC	propane	Carbon dioxide
natural	no	no	no	no	yes	yes
Atmospheric lifetime (years)	102	13.3	17	15	<1	3*10^4
ODP	1	0.05	0	0	0	0
GWP	8500	1700	1725	1600	~3	1

(100 years)						
Safety classification	A1	A1	A1	A1	A3	A1

Table 2 : Characteristics of some refrigerants.

1.2.4 Properties of propane

As the aim of NxtHPG project is to develop an innovative and high power heat pump working with propane, in this paragraph will be analysed the propane properties comparing them to the characteristics of some other manmade and natural refrigerants.

Propane is a colourless and odourless gas that is not toxic but has an anaesthetic effect so to prevent dangers in the unlike event of leakage a trace gas is normally added to allow a faster detection of the drip.

The renovated interest on hydrocarbons in general as refrigerants, despite their drawback about flammability, is supported by different favourable characteristics, above all the similar properties to some refrigerant gases nowadays used in the refrigeration market.

That permits to switch refrigerant with minimum changes on the biggest part of components and system design and so without expensive rebuilds of all the single units for the refrigeration sector companies.

The most common hydrocarbons used in the refrigeration field are Isobutane (R600a), Propane (R290) and Propene (propylene, R1270).

The first is the most widely used and it dominates the market of domestic refrigerators and freezers in Europe. The last-named on the other hand are utilized (also in blends) especially from some heat pump manufacturers but their market share, until now, is not so ample.

In Table 3 Table 3: Refrigerant properties at 40ºC (saturation point) where applicable.can be seen some of the most important properties of the introduced hydrocarbons and of R22, R134a and Ammonia in order to compare their characteristics. R22 is out of market in EU from 2010 but it is present in the comparison for the wide use in the past.

	Isobutane	Propane	Propene	R134a	R22	Ammonia
Refrigerant name	R600a	R290	R1270	R134a	R22	R717
M (kg/kmol)	58.12	44.096	42.08	102.03	86.468	17.03
T_{nbp} (ºC)	-11.67	-42.09	-47.69	-26.074	-40.81	-33.327
h_{fg} (kJ/kg)	311.4	306.51	303.14	163.02	166.6	1099.31
ρ_l (kg/m3)	530.0	467.07	476.66	1146.7	1128.5	579.44
ρ_v (kg/m3)	13.667	30.202	35.708	50.085	66.193	12.034
T_{tr} (ºC)	-159.59	-187.67	-185.2	-103.3	-157.42	-77.655
T_{cr} (ºC)	134.67	96.675	92.42	101.06	96.145	132.25

Table 3: Refrigerant properties at 40ºC (saturation point) where applicable.

Where T_{nbp} is the normal boiling temperature, h_{fg} is the heat of vaporization, ρ_l and ρ_v the liquid and vapour density, T_{tr} and T_{cr} the triple point temperature and critical temperature.

As can be seen the three hydrocarbons have molar masses and so densities much smaller than R134a and R22. This is particularly relevant for the HCs liquid density that is about half of the manmade refrigerants ones; it is an advantage for the HCs because, other things being equal, the refrigerant charge will be lower.

Another property favourable for the HCs is the high heat of vaporization, near to the double of the manmade refrigerants one, that involves a lower refrigerant mass circulation rate for the same heat transport, anyway as the HCs densities are more or less the half, the volumetric thermal capacity will be the same as for the manmade refrigerants.

Normally the refrigerating cycles are designed to work in a pressure range between 1 and 25 bar, that advice is because higher pressure will entail extra costs for the components while a pressure lower than 1 bar could imply the entrance of the ambient air in the circuit in case of leakage.

As the minimum and maximum pressure in a refrigerant cycle is related to the evaporation and condensing temperature, it is useful to study the saturation pressure of the refrigerants at different temperatures. In Figure 4 are reported the saturation pressure-temperature curves of the most common hydrocarbons refrigerants and of the relative alternative refrigerants.

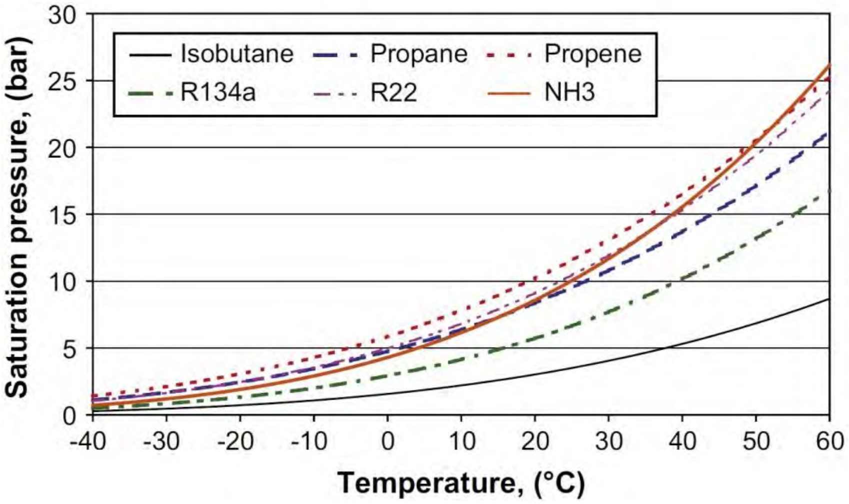


Figure 4: Saturation pressure in function of the temperature for isobutene, propane, propene, R134a, R22 and Ammonia from [23].

As can be seen in the diagram, the propane curve follows the trend of the R22 one, and so the applications areas of these two refrigerants are similar.

Furthermore the saturation pressure of propane at 5ºC is near to 6 bar while at 60ºC is near to 20 bar, so the previous pressure range advice is respected for a general propane heat pump working between these two temperatures; i.e. the conditions near which the propane prototype installation in the UPV laboratory will work.

From another point of view, the saturation pressure is similar at low temperatures while at higher temperatures it is lower for propane than for R22 (i.e. the high pressure section of a heat pump), that implies a lower pressure ratio for a cycle working with propane instead of R22 and so a general better working for the compressor.

To compare different refrigerants it is possible to note similarities and differences in their enthalpy-temperature diagrams. From Figure 5 it is immediate to see the different entities for the heat of evaporation; for the HCs (and also for ammonia) this value is bigger than for R134a and R22. That is a favourable property because the bigger is the heat of evaporation (or vaporization) the smaller will be the refrigerant mass to achieve the same cooling (heating) load.

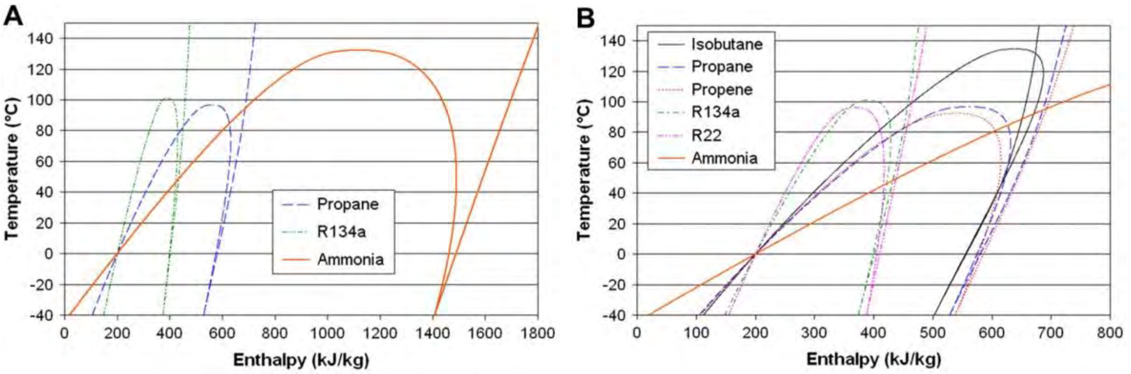


Figure 5: Temperature-enthalpy diagram for: (A) propane, R134a and ammonia; (B) isobutene, propane, propene, R134a, R22 and ammonia. From [23].

Anyway the refrigerant mass flow depends also on the density of the refrigerant; the refrigeration capacity of a vapour compressor cycle can be determined by the product between the swept volume from the compressor [m^3/s], the refrigerant vapour density at the entrance of the compressor [kg/m^3] and the heat of vaporization [kJ/kg]. Considering that the last two depend on the pressure, it is possible to draw in function of pressure a diagram that shows the capacity reachable for different refrigerants per unit swept volume.

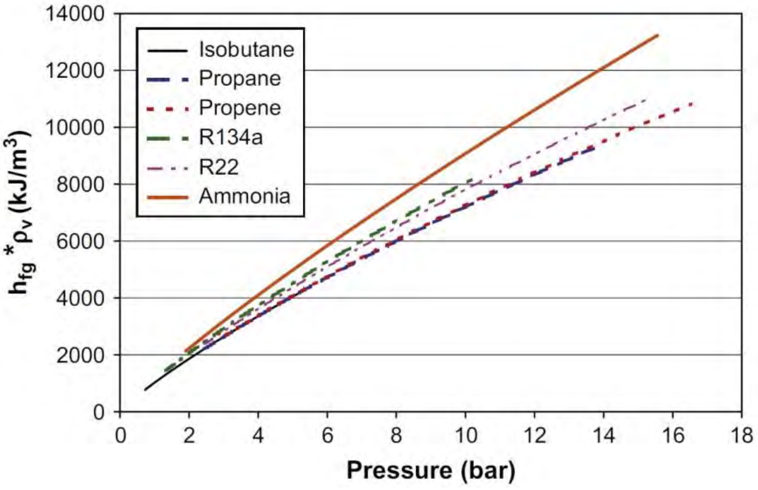


Figure 6: Refrigeration capacity reachable for different refrigerants per unit swept volume in function of the evaporation pressure. From [23].

Where h_{fg} is the heat of vaporization and ρ_v the refrigerant vapour density.

In Figure 6 can be seen that the characteristic curve for propane is a bit lower than the R22 one and around equal to the propene one, the reason is that also if for propane h_{fg} is bigger than for R22, its lower molecular mass bring back the product $h_{fg}*\rho_v$ to a value smaller than the one of R22. It means that to have the same refrigeration/heating capacity a cycle working with propane will have a bigger compressor.

The thermal conductivity of propane and hydrocarbons in general are higher than those of R22 and R134a and it entail a favourable heat transfer performance.

The viscosity of propane is lower than the one of the considered manmade refrigerants and it involves lower pressure drop.

There are no problems in the utilization of common used components in refrigeration field with propane and in general with HCs because they are compatible with the biggest part of metal alloy and polymers; furthermore hydrocarbons are compatible and highly soluble with the most common synthetic and mineral oils.

The fact that the latters are highly soluble is positive because the return oil to compressor is assured, but from the other part this can involve a reduced viscosity of the diluted oil and an excessive foaming at the start-up. As the reduced viscosity can entail problems for the compressor, oils with higher viscosity are used with hydrocarbons.

The hydrocarbons are highly flammable, for example the lower and upper flammability limit for propane in air are respectively LFL=2.1 and UFL=9.5 in volume percentage.

The risks due to that inconvenient can be reduced minimizing the refrigerant charge in the system, e.g. utilizing compact designs and utilizing as compact as possible heat exchangers.

Another expedient that can be taken into account [13] is the positioning of the refrigerating system in open air in a restricted area like it could be the roof of a building, that idea has been adopted for the NxtHPG project case 3.

In conclusion the possibility to use an alternative refrigerant in developed systems without requiring significant changes in materials or techniques is for sure a very attractive point, especially considering the good thermodynamic characteristics and the environmental-friendly properties.

2 Installation description

As said before the aim of the NxtHPG project in case 3 is to study a heat pump working with propane that will produce domestic hot water at 60ºC exploiting as source a neutral water loop with an available temperature of 10÷30º C.

The installation in the UPV laboratory that tests this prototype is composed by different circuits and secondary loops.

The main cycle is the heat pump working with propane; circuit 1 provides the warm water source and circuits 2, 3 and 4 simulate the heat consumption of a consumer that needs sanitary hot water.

The scheme of the installation is reported in the following Figure 7.

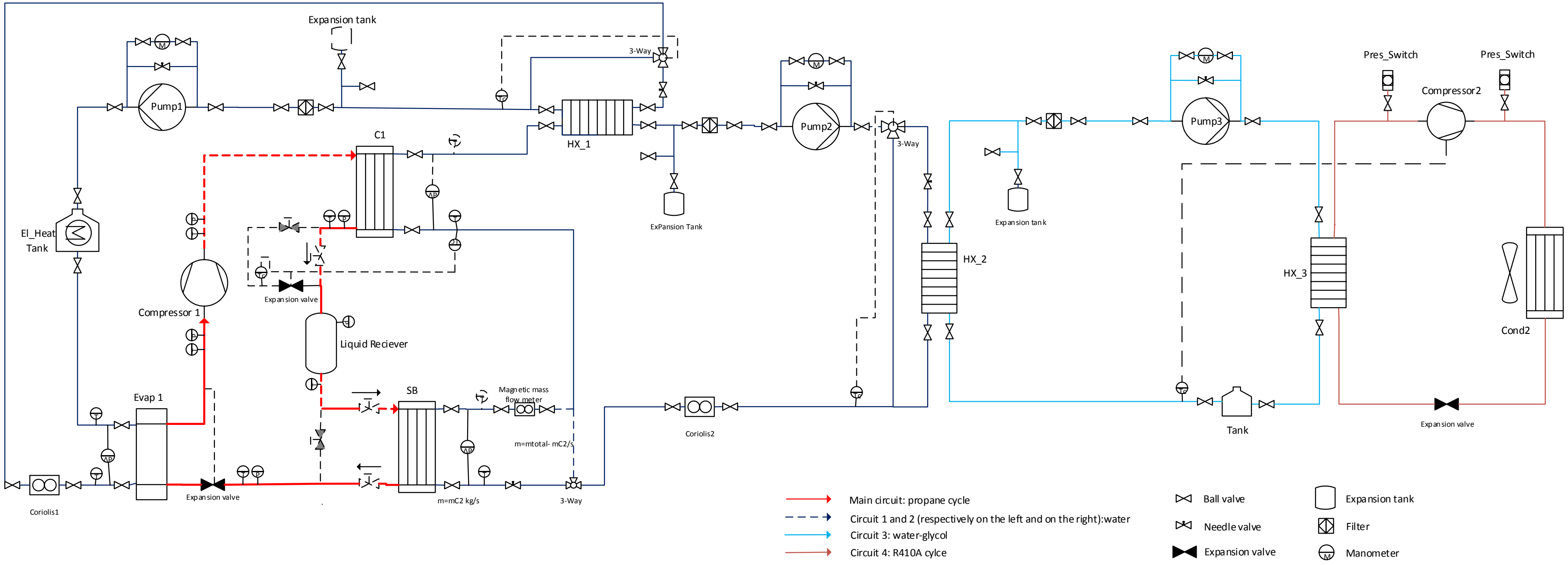


Figure 7: The scheme of the whole installation

2.1 Main circuit

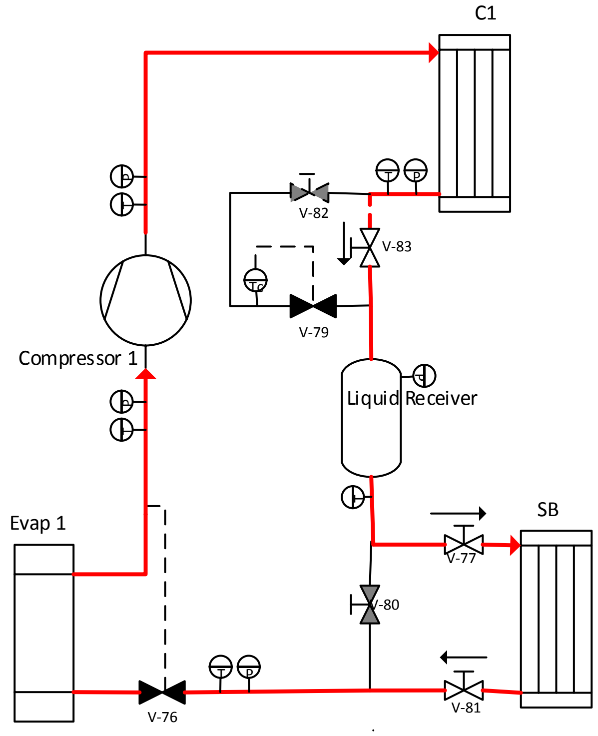
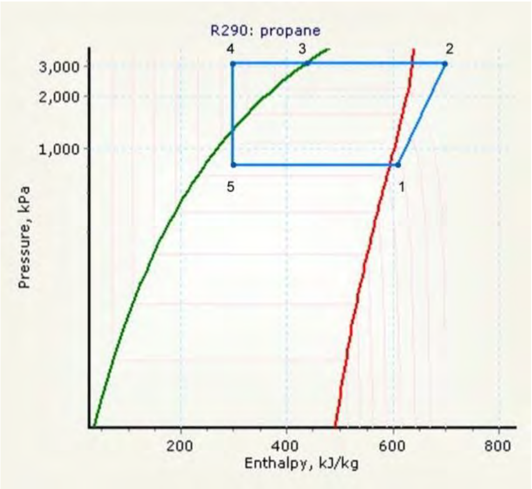
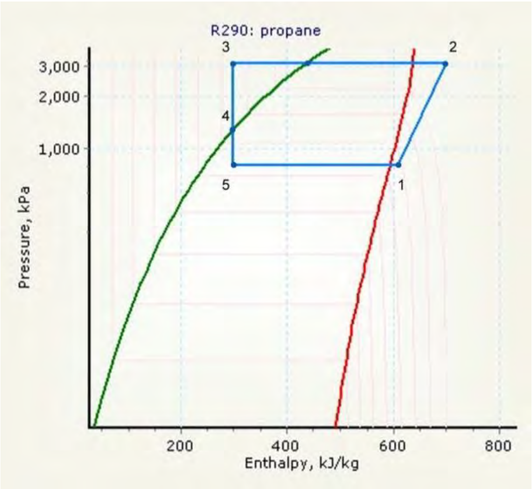


Figure 8: Main circuit

The main circuit is the heat pump working with propane as refrigerant that will be tested and developed as final aim of the NxtHPG project. It consists of a water to water heat pump that works using a neutral water loop at 10÷30ºC (recovery of waste heat from condensation or sewage water) and warming water up to 60ºC as domestic hot water production.

While the heat exchange with the water in the circuit one is done with a single evaporator: Evap1, the condensing side is split in two different possible configurations as in Figure 9.





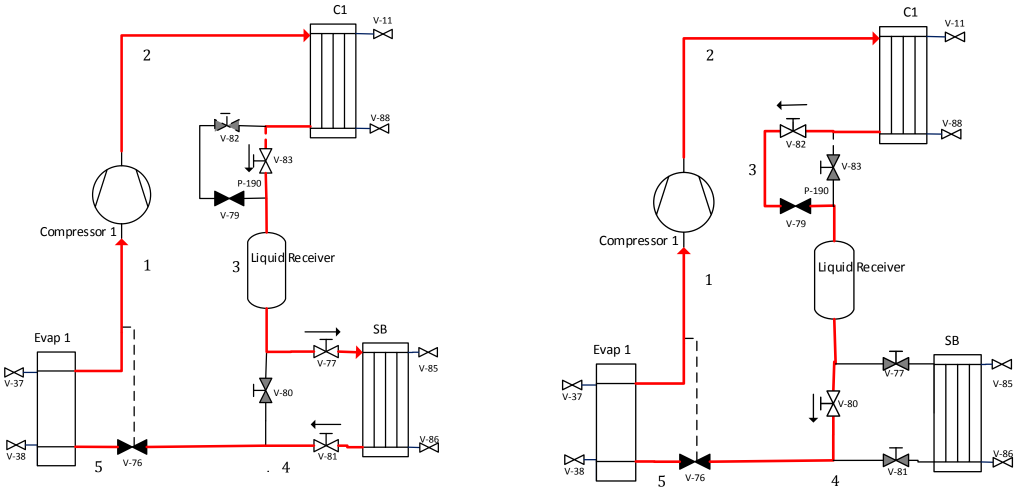


Figure 9: Configuration A on the left and B on the right and relative pressure-enthalpy diagrams, the diagrams are shown only to explain the two configurations and the pressure and enthalpy values are not the effective ones.

In the configuration A the heat exchanger C1 condensates the propane till the saturated liquid point (3), the liquid propane goes to a liquid receiver and then to a subcooler (SB) in wich it is subcooled before passing in the expansion valve V-76.

In case B the heat exchanger C1 is used both to condensate and to subcool the propane while the SB is bypassed; the liquid receiver takes care about the subcooling variating the quantity of refrigerant available in the whole circuit, that is possible thanks to an automatic expansion valve placed in the inlet of the liquid receiver.

It is possible to switch between the two configurations opening and closing parallel lines of the circuit as it can be seen in the Figure 9, to switch from configuration A to B is sufficient to close the valves v-83, v-77, v-81 and open the valves v-82 and v-80.

The compressor used in this circuit is a scroll compressor.

Before and after the compressor and every heat exchanger there is a measurement of pressure and temperature in order to supervise the fluid conditions and the entity of the heat transfer in the heat exchangers.

2.2 Circuit 1

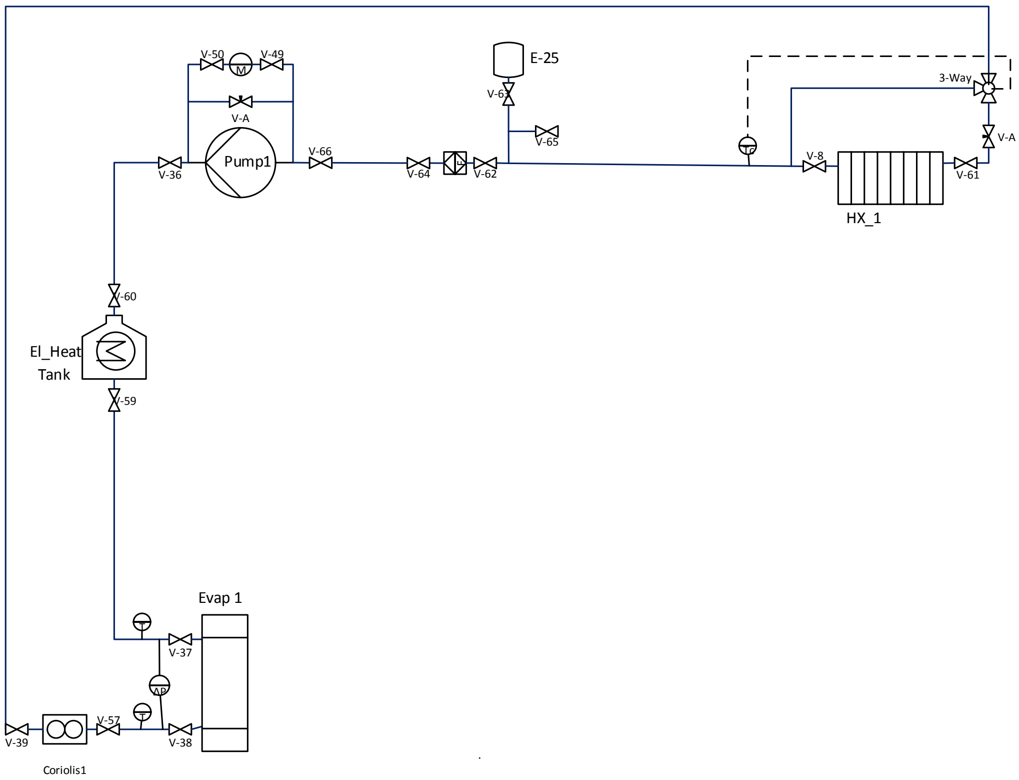


Figure 10: Circuit 1

Circuit 1 is built to assure the desired water temperature and mass flow rate at the evaporator inlet of the heat pump prototype; in the following Chapter 3 will be described how these variables are controlled.

The heat sources analysed for this prototype are water loops that exploit heat fluxes which otherwise will be wasted to the external ambient.

As the prototype installation is targeted for residential buildings or commercial buildings the necessity to waste heat to the external ambient (for example waste heat with fan coils is common for the big commercial refrigerator aisle) can be exploited to heat a water loop that works as heat source for other uses. This is the building condensation loop that is supposed to be at 15÷30ºC respectively in winter and summer.

The second possible loop hypothesized is heated with the sewage water discharged, in a big building its amount can be relevant, hence the possibility to obtain a water loop at 10÷15º C respectively in winter and summer.

As the water source really used in the laboratory is in a loop circuit and as the produced hot water does not have a real consumer but the heat flux has to be drained, the water loop of the source will be heated up with a recovery unit using the produced hot water at 60ºC.

The loop water circuit (from now called circuit 1) is made up of various components.

An electric heat tank can adjust in every condition the temperature of the water guaranteeing that the water inlet temperature to the evaporator is at the desired value in the range 10÷35º C. This component is necessary to maintain the wanted inlet temperature in the cases when the heat exchange between circuit 1 and circuit 2 through the heat exchanger HX_1 is not sufficient. Furthermore as the electric heat tank can set the temperature in a more precise way, when the heat exchange between the two circuits can be fulfilled by the HX_1 alone, the temperature of the water leaving HX_1 will be maintained around 0,6ºC under the target inlet temperature for Evap_1 taking into account that it will be warmed up by the following pump and then the water temperature will reach the set point thanks to the electric heat tank. The control of the resistance heating will be governed by the water temperature measured after the electric heat tank.

Pump1 takes in charge the pressure drop permitting the circulation of the water, its velocity is variable and moreover a needle valve allows to modify accurately the water flow also in border situations.

An heat exchanger (HX_1) warms up the cold water of circuit 1 cooling down the produced hot water of circuit 2; as we want to test the circuit in different conditions a three way valve is placed. It regulates the mass flow through HX_1 and through the bypass in order to control the heat exchange and consequently to obtain the desired water temperature downline of HX_1. The three way valve is controlled by the measured water temperature and as it has to control a flow variation in all the range between totally close and totally open position, a needle valve is used to obtain a more precise adjustment.

In the same circuit there are also:

- a coriolis mass flow meter for an accurate mass flow measurement,;
- an expansion tank to balance the dilatations of the fluid and of the pipes due to the different operative and ambient temperatures;
- a filter situated before the pump, this positioning is valid in every circuit to avoid the possible entrance of residuals present in the circulating liquid that could damage the pump ;
- temperature and pressure sensors to measure the inlet and outlet conditions of the water in the heat exchanger (Evap1) in order to determine the heat exchange through it.

2.3 Circuit 2

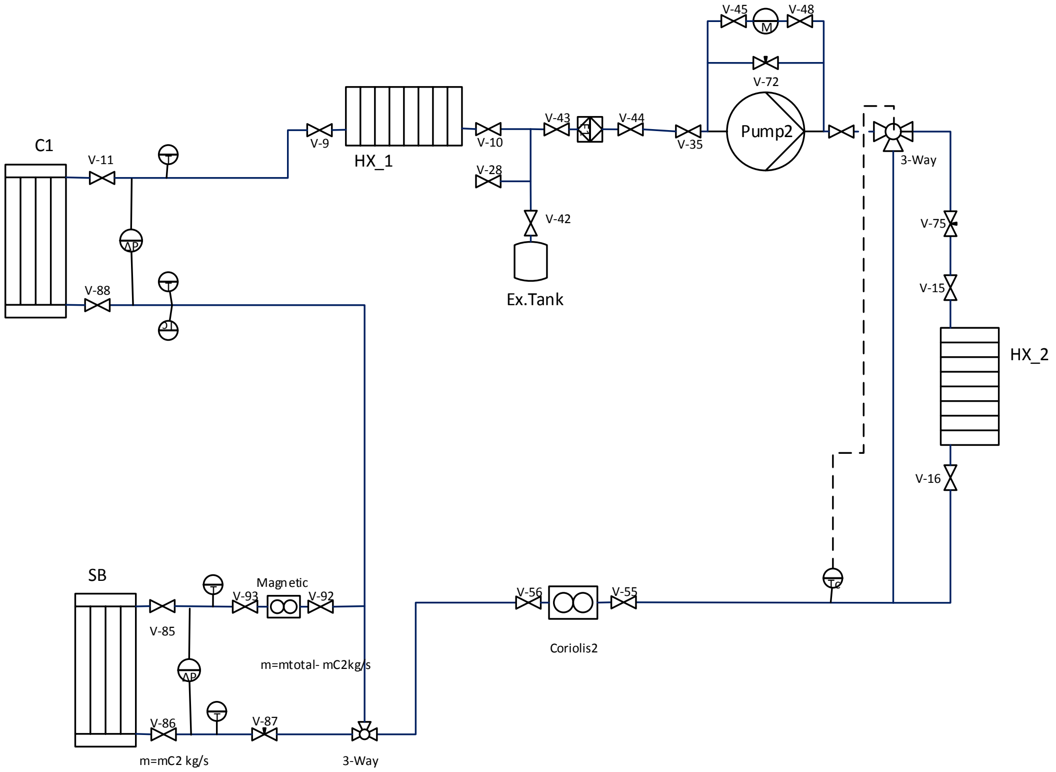


Figure 11: Circuit 2

To simulate the hot water consumption two secondary circuits are used: the produced hot water is cooled exchanging with circuit 1 and with the water-glycol loop that in turn exchanges with another refrigerant cycle.

The consumer water circuit (from now called circuit 2) reproduces the heat demand of a fictitious consumer cooling down the water from 60ºC to 10÷55ºC depending on the test conditions.

The connection between the heat pump unit and the consumer's heating system is foreseen as in Figure 12: a storage tank will be used as heat accumulator guaranteeing the availability in the periods of maximum sanitary hot water demand, the starting and stop of the heat pump compressor will be commanded by the water temperature in the tank.

The temperature we want to assure for the water delivered to the consumer is near to 60ºC, so the compressor will stop when the temperature in the tank will be near to 60ºC and start again when it will decrease below 55ºC. For this reason the working simulation of the installation is made with a maximum water return temperature of 55ºC.

Differently when the storage tank have to be filled with water for the first time or after having emptied it for maintenance, the water through the heat pump condenser will be at the aqueduct temperature so 10ºC, its temperature will raise heated by the heat pump until it joins the 60 ºC. The working simulation with the inlet water condenser temperature from 10º to 55ºC refers to this starting operation.

As the water in the tank will be directly used by the consumer, an electric resistance is used to cyclically warm the water till 75ºC to avoid the risk of legionella as the safety regulations indicate.

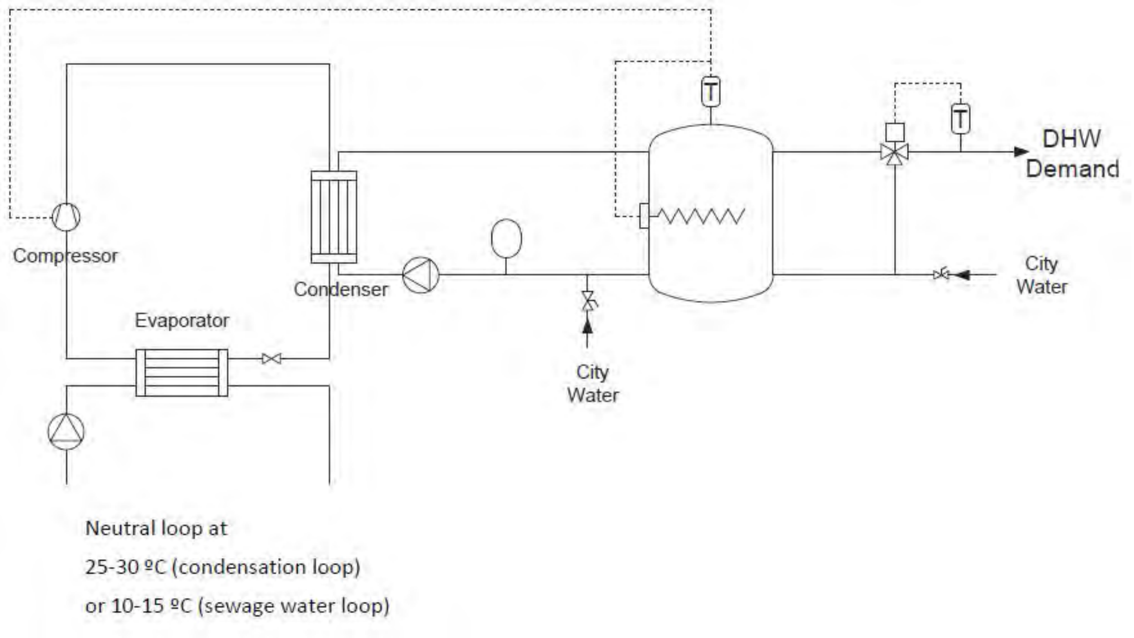


Figure 12: Connection of the heat pump unit to the heating system from NxtHPG Deliverable 1.3

To simulate the explained different conditions the water is cooled down in part exchanging with the source water loop through the HX_1 as said yet and in part with a water glycol circuit through another heat exchanger (HX_2), see Figure 11.

Like in the circuit 1 the water flow through HX_2 is regulated by a three way valve that controls the flow through a bypass pipe governed by a thermometer and a needle valve is used for more precise adjustments.

Once again the pump (pump 2) works at variable velocity and a needle valve allows to modify accurately the water flow, a filter is put before the pump and an expansion tank is needed to balance the dilatations of the fluid and of the pipes due by the different operative temperatures.

As we want to test different configurations for the heat pump, two heat exchangers (SB and C1) are used in different layouts to warm up the water till 60ºC, for both temperature and differential pressure sensors will measure water inlet and outlet conditions. To modify the water flow through SB a three way valve is adopted, this valve is used in ON-OFF way to switch between the two heat exchangers' configurations.

Furthermore, if it will be necessary, it will be possible also to reduce the water mass flow rate through SB when it is too high for the pressure drop; in this case after the three way valve a needle valve will permit an accurate control of the flow.

In order determine the heat exchange through each heat exchanger, also in the case when the water mass flow through the subcooler mSB is different from the one through the condenser $mC1$, it could be enough to measure the total mass flow $mtot$ and to solve a system equation on the heat exchangers:

$$qtot = mtot * cp * \Delta ttot$$

$$qSB = mSB * cp * \Delta tSB$$

$$qC1 = mC1 * cp * \Delta tC1$$

$$mtot = mC1$$

$$qtot = qSB + qC1$$

But another mass flow meter is put to have a direct measure of the mass passing through the SB taking into account that to solve the equations system we need to know the specific heat cp that changes at the different working temperatures.

So a coriolis mass flow meter measures $mtot$ and a magnetic mass flow meter put after the SB measures mSB; the choice of use a magnetic mass flow meter instead of another coriolis is because, as said, this measure could be also obtained solving an equation system and it is preferable to use a less expensive and less bulky device, furthermore the magnetic mass flow meter measurements will be compared with the coriolis ones when $mSB = mC1$ in order to know for the future installations in which conditions it works more accurately.

2.4 Circuit 3

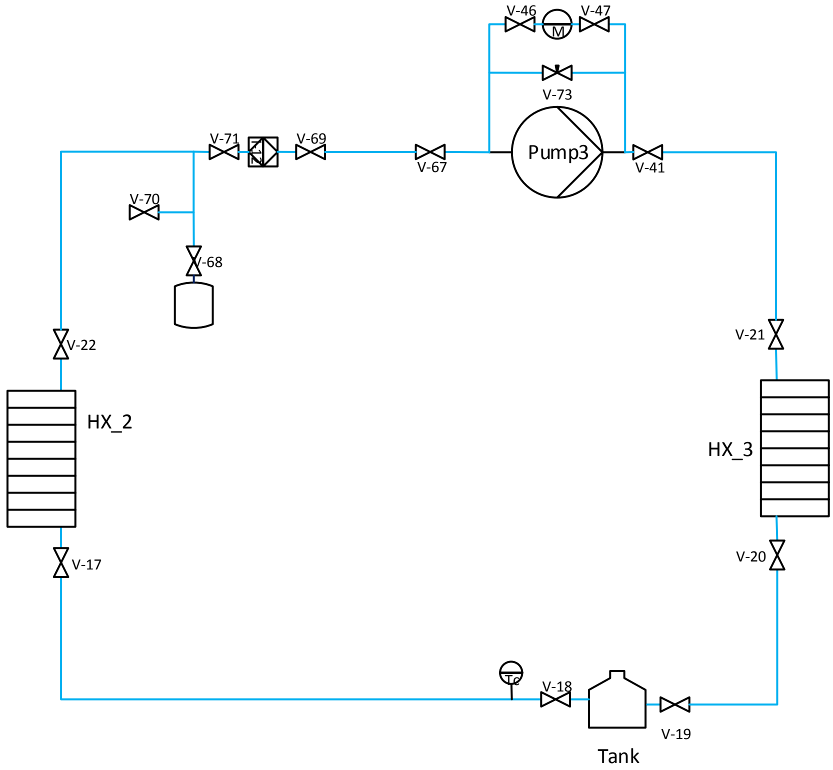


Figure 13: Circuit 3

To cool down the water of circuit 2 till the desired temperature 10÷55ºC we use a refrigerant cycle (circuit 4) connected via a water-glycol loop (circuit 3) to the second circuit. Circuit 3 is necessary to prevent the risk of freeze up the water in circuit 2 damaging the heat exchanger, that is because the refrigerant cycle can join dangerous low temperatures due to the fact that its compressor is on-off.

Water-glycol loop is connected with circuits 2 and 4 respectively by HX_2 and HX_3, pump3 works at constant velocity and a needle valve is placed in parallel to permit a degree of freedom.

A tank is placed after HX_3 to avoid the risk of freeze in circuit 2: the inertia of the water-glycol mass contained in it allows to maintain HX_2 inlet near to 0ºC also if the inlet of the tank falls down to lower temperatures.

The expansion tank of the water-glycol loop contrary to the other circuits has also to compensate the dilatation of the fluid contained in the tank so its required volume calculated in chapter 3.13 is the biggest.

2.5 Circuit 4

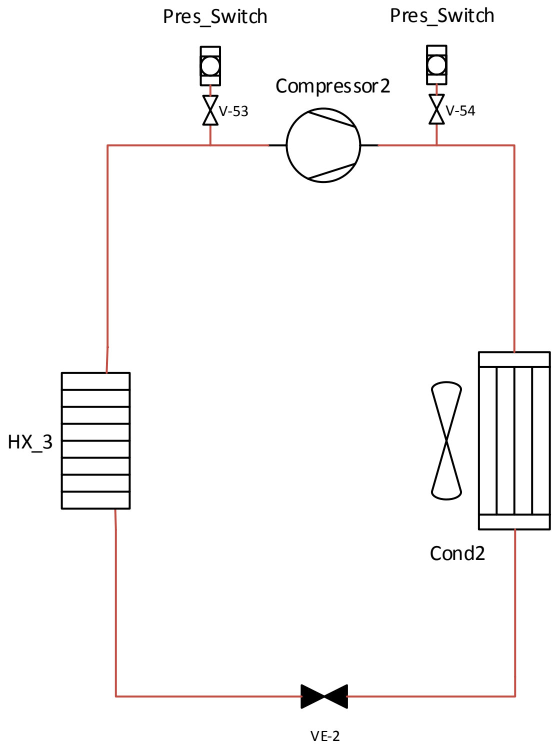


Figure 14: Circuit 4

Circuit 4 is a refrigerant cycle working with R410A as refrigerant; the heat flux coming from circuit 3 is exchanged through HX_3 and released to the external ambient with a fan coil heat exchanger (Cond2).

Compressor 2 is ON-OFF and it is commanded by the water-glycol temperature measured after the tank in circuit 3, the compressor turns OFF when the water-glycol temperature decreases under a certain value and turns ON when the temperature rises over another value.

The tank in circuit 2 allows to avoid continuous starts and stops of the compressor and a consequent fast damaging of it, for that reason the temperatures on HX_3 will vary growing up when the compressor is OFF and falling down when it is ON.

For circuit 4 components safety, two pressure switch are put in the position of lowest and highest pressure respectively after and before the compressor, they will be useful to stop the compressor in case of the pressure will overrun the minimum or maximum security value.

3 Dimensioning

The aim of the chapter is to present the most relevant criteria followed in the dimensioning of the different components of the installation.

As the objective of this thesis is not the study of the propane main cycle but of the four secondary circuits, in this chapter will not be taken into account the dimensioning of the components of the propane circuit. Despite that I found interesting the study made for the condenser part on the main cycle so hereafter in paragraphs 3.1 and 3.2 is present a brief description on the taken choices.

The procedure followed in the dimensioning of the components in the secondary circuits starts with the dimensioning of the condenser in circuit 4, this choice was made for different practical reasons. The first one is that the necessity to reject a heat flux that can exceed 50 kW could not be satisfied warming the water of the supply network as in other installations in the Thermal Laboratory is done; a heat exchange with external air was necessary and this new installation needed a certain amount of space and time to be set up. In effect the construction of the air conduct and the positioning of condenser4 took a considerable period because of the size of the conduct and the design in the limited space available; two openings (in Figure 28 can be seen one) in the external wall of the Thermal Laboratory were made to permit the aspiration and the discharge of the air, furthermore the electrical illumination system was changed to permit the positioning of the air conduct in the roof.

Furthermore circuit4 is the only refrigerant cycle in the secondary circuits so its components could take a longer time to arrive if not present in the seller’s stock.

The following dimensioned components are the heat exchangers because knowing their characteristics it is possible to know the real heat flux exchanged in the different situations, the needed mass flow and finally the pressure drops: once calculated the mass flow it is possible to fix a diameter of the pipes, estimate the pressure losses and choose the pumps that fulfil the required conditions.

At the end the other components like the electric heat tank, the buffer tank and the expansion tanks were designed.

3.1 Heat exchangers propane-water

To test the two different configurations A and B there are different possible system designs.

The first one is using three different heat exchangers: a condenser and a subcooler will work in configuration A and a different heat exchanger will work alone in configuration B providing both for condensing and subcooling.

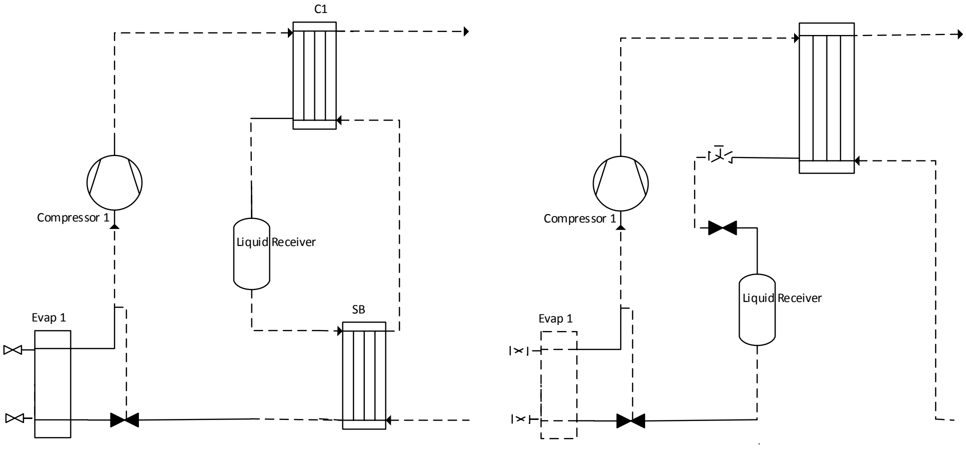


Figure 15: System design 1, configuration A on the left and B on the right

With that design it is possible to do an easy comparison between the two configurations A and B because the third heat exchanger should have the same area of the other two's sum; but the presence of three heat exchangers involve an extra cost, a higher amount of space needed for the installation and a complication in the construction disposition due to the bypass pipes.

The second possible system design (Figure 16) is using only two heat exchangers for a less expensive and a less bulky installation; the two heat exchangers of the configuration A could be used in series in the configuration B working like a single heat exchanger. In this way the exchange area is once again the same in both the configurations.

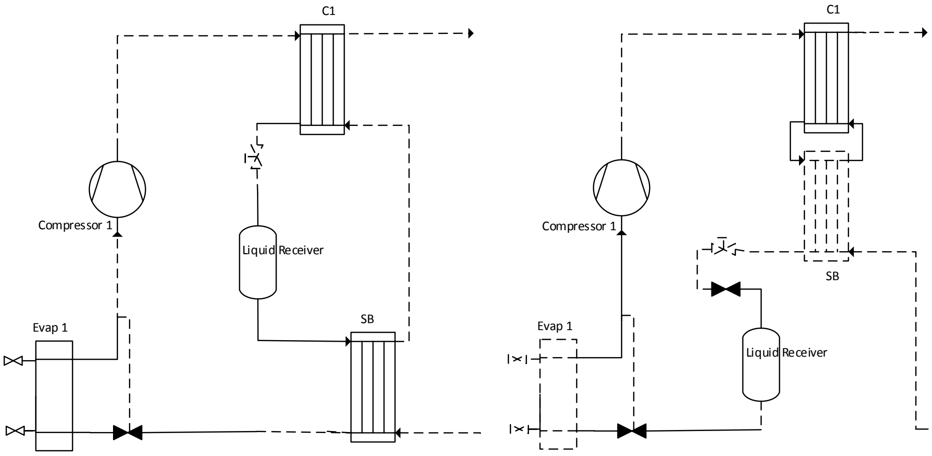


Figure 16: System design 2, configuration A and B

The last possible system design taken into account uses once again only two different heat exchangers but this time in the configuration B only one big heat exchanger will condense and subcool the fluid while in configuration A the big heat exchanger will cool down the propane till the saturated liquid point and a smaller subcooler will lead the fluid to the desired final temperature.

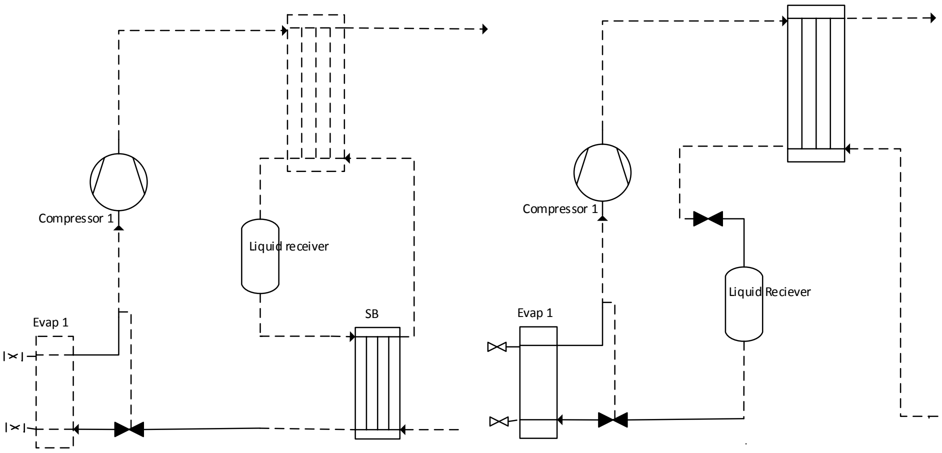


Figure 17: System design 3, configuration A and B

The last system design does not assure an equal heat exchange area in A and B configurations so the comparison of the results will be more complicated.

As one of the aims of the NxtHPG project is to develop an innovative heat pump keeping the final product commercial attractive, the first design has been refused for the necessity of using three heat exchangers.

The same issue for the second design with the two heat exchangers designed for configuration A and used in series for configuration B; in effect the Alfa Laval proposal for the two HX optimized for the A configuration was AC30-30 and CB20-40H for Condenser1 and Condenser2 respectively but a calculation on the pressure drop in the water side showed an exponential increase with the water inlet temperature.

As can be seen in the following graph these heat exchangers for the configuration B are not suitable for a water inlet temperature bigger than 35ºC.

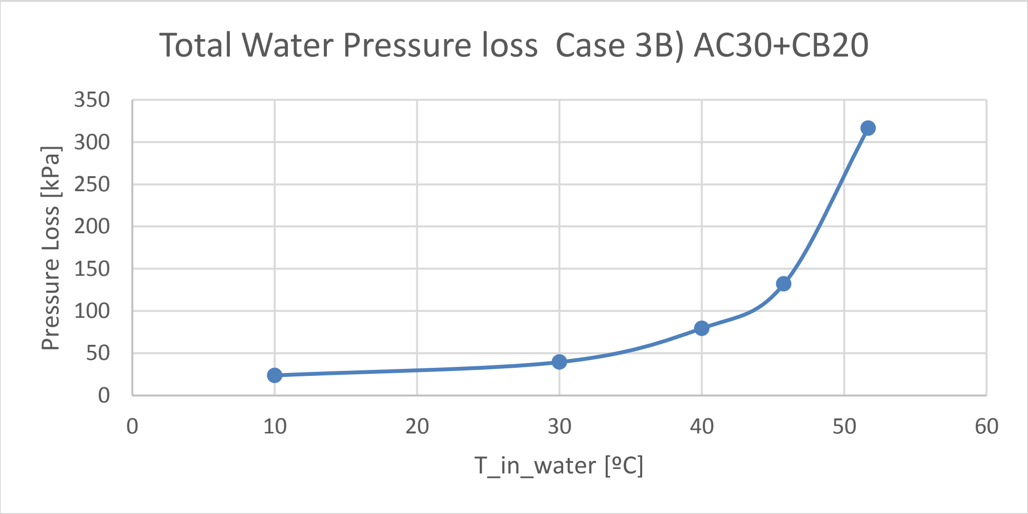


Table 4: Pressure losses in case 3B

These calculations were made with IMST-ART with the following input data: water inlet temperature from 10 to 51.7 ºC ; mf=403.2 kg/h of propane for both heat exchanger in series ; inlet temperature for AC30-30 (propane) T1=81 ºC ; dew point at the inlet of CB20-40H (propane); water outlet temperature of AC30-30 T2 = 60 ºC

In conclusion the chosen system design is the third one with Alfa Laval CB62-62 as condenser CB and Alfa Laval CB20-41 as subcooler.

3.2 Propane liquid receiver

Once the evaporator and the condenser have been selected, the liquid receiver can be sized. It has to be able to accommodate a quantity of mass equal to the difference between the maximum and minimum mass needed in the system.

The mass contained in the subcooler is not taken into account in this study because it is supposed to be filled up with liquid in both A and B configurations so the change in mass inventory will not change significantly from point to point. In the same way the pipeline that are connected and disconnected to switch between the two configurations are not taken into account. By the other hand, the pipes that are filled up with gas have a change in the mass inventory from point to point, and it will be fixed with an extra volume for security.

The whole cycle has been calculated with IMST-ART for different operating points, the components inserted for the simulation of the cycle are:

Refrigerant: R290(propane)
Compressor: PSH038
Evaporator: AC11260H
Condenser: CB62-62

The calculations for the different inlet and outlet water temperature in evaporating side are made for two boundary values of subcooling, respectively 1.5 and 40 K, in the first case we will have the maximum quantity of refrigerant in the liquid receiver and in the second the minimum.

The case with the maximum mass inventory in the circuit (and so the minimum mass quantity in the liquid receiver) is when the subcooling is 40 K and the water in evaporator side is W35-25 (that is the operative point with inlet water at 35ºC and outlet at 25ºC), and the minimum is when the subcooling is 1,5 K and W20-15, respectively 1.7478 kg and 0.58944 kg.

Hence the volume needed to host this mass in liquid state at the exit of the condenser is:

$$V_{LR} = \frac{(m_{MAX} - m_{MIN})}{\rho_{MIN}} + V_{ex} = \frac{(1.7478 - 0.58944)}{428.31} + 0.001 = 0.0037\, m^3 = 3.7\, l$$

Where ρ_{MIN} is the liquid density when the inventory mass in the cycle is minimum that is when the biggest quantity of refrigerant has to stay into the liquid receiver; V_{ex} is an extra volume for security.

3.3 Condenser

This condenser is placed at the end of the chain in the refrigerant cycle and it rejects the heat to the external ambient exchanging with the air. As said that condenser was the first selection of the auxiliary circuits for the yet explained reasons at the beginning of the chapter.

As it was not possible to calculate for first the maximum heat load of the condenser without having fixed the other components of the upstream circuits, this parameter had been supposed.

The reasoning was the following: the heat pump has to furnish 50 kW of heat power to the consumer (Circuit 2), we suppose that Evaporator1 will exchange 35 kW of heat power while the compressor in the main cycle will contribute with the remaining power of 15 kW (supposed COP of 3,3). Imagining that all the 35 kW exchanged with Circuit 1 come from the heat exchange through the recovery heat exchanger HX_1; in Circuit 2 will be drained 15 kW plus other 5 kW due to the water pumps. In total in HX_3 will be exchanged 20 kW that summed with the power of the compressor 2 (10 kW were wondered from the previous experiences in the laboratory) give the supposed maximum heat load that condenser 2 has to reject: 30 kW.

So the condenser has been selected with Alfa Laval's software: "Alfa Select Air", for the following conditions:

Required Capacity: 30 kW
Refrigerant: R407C
Inlet air Temperature: 24 °C
Condensation Temperature: 40 °C
DTsh: 25 K

And the consequent selected condenser is AGS403B.

As it could be seen the dimensioning was made for the R407C refrigerant but finally R410A is used for the cycle. That substitution is caused by the fact that at the beginning a tandem compressor working with R407C and available in the Thermal Laboratory was supposed to be used. Later one of the two compressors revealed an incorrect working during a test so the tandem compressor has been replaced by a R410A scroll compressor.

The choice to use the refrigerant R410A instead of R407C is mostly due to the bigger cooling capacity as can be seen in the following Table 5.

	Refrigerant	**R407C**	**R410A**
Chemical properties	Chlorine content	no	no
	Zeotropic	zeotropic mixture	near azeotropic mixture
	Composition	R32/R125/R134a (23/25/52 wt%)	R32/R125 (50/50 wt%)
Environmental impact	ODP	0	0
	GWP	1526	1725
	Vapour pressure (bar) at 25ºC	11.9	16.5
Thermodynamic	Cooling capacity of liquid (kJ/kgK)at 25ºC	1.54	1.84
properties	Cooling capacity of vapour (kJ/kgK)at 1 atm, 25ºC	0.829	0.833
	Temperature glide (ºC)	7.4	<0.2

Table 5: Main refrigerants properties.

This change in refrigerant involves different conditions in the operation of the condenser, its working has been studied simulating the entire circuit 4 with IMST-ART. The results are higher condensation temperature and higher pressure compared with the same points calculated with R407C as refrigerant, furthermore the heat exchange capacity calculated with the new refrigerant is bigger than the previously calculated one.

In Appendix can be seen the summary results of the calculation made with IMST-ART for the two different refrigerants in six different working conditions.

3.4 Recovery heat exchanger water-water

The aim of this component is to warm the water in circuit 1 exchanging heat with the water at 60ºC in circuit 2.

Because of in some cases HX_1 can recovery more heat than the one absorbed in the evaporator of the main circuit, a bypass governed by a three way valve is necessary to limit the temperature of the water entering in the evaporator.

The symbols used in this paragraph are the following:

$\dot{m}_r$ [kg/s] : water flow through the heat exchanger HX_1 evaporator side;

$\dot{m}_b$ [kg/s] : water flow through the bypass;

$\dot{m}_{Wevap}$[kg/s]: water flow going to the evaporator;

$\dot{m}_{Wcond}$[kg/s]: water flow through the heat exchanger HX_1 condenser side;

T_{ci} [ºC]: water inlet temperature in HX_1 evaporator side;

T_{co}[ºC]: water outlet temperature in HX_1 evaporator side;

T_{evapin}[ºC]: water temperature after the mixing between $\dot{m}_r$ and $\dot{m}_b$;

$Target\ T_{evapin}$ [ºC]: desired water inlet temperature in the evaporator;

T_{hi}[ºC]: water inlet temperature in HX_1 condenser side;

T_{ho}[ºC]: water outlet temperature in HX_1 condenser side;

$Target\ (T_{ho})$ [ºC]: desired water return temperature to the condenser;

$\dot{Q}_{maxrecovery}$ [W]: maximum recoverable heat flux;

$\dot{Q}_{rec}$ [W]: effectively recovered heat flux;

$\dot{Q}_{GW}$ [W]: heat power that will be dissipated through the water-glycol circuit;

c_p [kJ/kgK]: specific heat;

$\dot{Q}_{cond}$ [W]: heat power exchanged in the condenser(s) of the main cycle;

$\dot{Q}_{evap}$ [W]: heat power exchanged in the evaporators of the main cycle;

$x_b = \dot{m}_b/(\dot{m}_b + \dot{m}_r)$

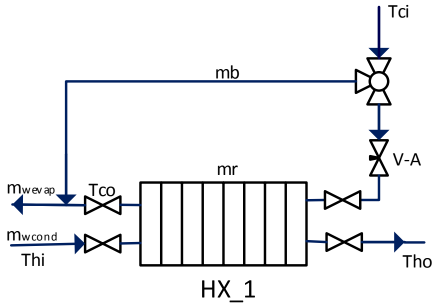


Figure 18: Recovery heat exchanger connections

The dimensioning of HX_1 has been done taking into account the control on the mass flow through the heat exchanger as follows:

First T_{co} is calculated considering $\dot{m}_r = \dot{m}_{Wevap}$ that is when the bypass is closed, this happens only when T_{co}<$Target\, T_{evapin}$ and in this case the electric heat tank will warm the water until its temperature joins the desired $Target\, T_{evapin}$.

The equations used for this situation are:

$$\dot{Q}_{maxrecovery} = \left(\dot{m}c_p\right)_{min}(T_{hi} - T_{ci})$$

$$e_{recovery} = \frac{\dot{Q}_{recovery}}{\dot{Q}_{maxrecovery}}$$

$$T_{co} = T_{ci} + \frac{\dot{Q}_{recovery}}{\dot{m}_{Wevap} * c_{p_{water}}}$$

$$T_{ho} = T_{hi} - \frac{\dot{Q}_{recovery}}{\dot{m}_{Wcond} * c_{p_{water}}}$$

When the relation T_{co}< $Target\, T_{evapin}$ is not verified other relations are taken into account considering also the mass flow $\dot{m}b$ through the bypass, in order to assure that the inlet temperature in Evaporator1 will be the target one.

In this case the following equations are used:

$$\dot{Q}_{maxrecovery} = \left(\dot{m}c_p\right)_{min}(T_{hi} - T_{ci})$$

$$e_{recovery} = \frac{\dot{Q}_{recovery}}{\dot{Q}_{maxrecovery}}$$

$$\dot{m}_{Wevap} = \dot{m}_r + \dot{m}_b$$

$$\dot{m}_r * c_{p_{water}} * (T_{co} - T_{evapin}) = \dot{m}_b * c_{p_{water}} * (T_{evapin} - T_{ci})$$

$$\dot{Q}_{recovery} = \dot{m}_r * c_{p_{water}} * (T_{co} - T_{ci})$$

Once known the inlet and outlet temperatures of the recovery heat exchanger it is possible to calculate the UA for each situation with the long mean temperature difference (LMTD) thanks to the following equations:

$$\begin{aligned} dti &= T_{hi} - T_{co} \\ dto &= T_{ho} - T_{ci} \end{aligned}$$

$$LMTD = \frac{(dto - dti)}{LN\left(\frac{dto}{dti}\right)}$$

$$Q_{recovery} = UA_{recovery} * LMTD$$

The system equation to know the temperatures, the mass flows, the heat exchanged and the UA is solved with the software EES: Engineer Equation Solver. The simulation is made for the possible situations considering the different temperature of the inlet and outlet water in the evaporator (respectively 10÷30ºC - 20÷35ºC) and of the inlet water temperature in the condenser (10÷55ºC) in steps of 5ºC.

A first simulation is made supposing that the efficiency of the heat exchanger is constant and equal to 0.7; the results for the different operating points can be seen in the following Table 6 where the worst operating point (the one that needs the greater UA) is underlined: W35-30W35-60, that is water inlet and outlet temperature in the evaporator: 35 and 30°C, water inlet and outlet temperature in the condensing side: 35 and 60°C.

Some input data for EES: $\dot{Q}_{cond}$, $\dot{Q}_{evap}$, $\dot{m}_{Wcond}$, $\dot{m}_{Wevap}$ are calculated with the software IMST-ART that simulates the working of the main cycle with the selected components.

Q cond [kW]	Q evap [kW]	Target (Tho) [ºC]	Thi [ºC]	Target T_evapin [ºC]	Tci [ºC]	Tco [ºC]	T_evapin [ºC]	Tho [ºC]	m_b [kg/s]	m_r [kg/s]	X_b [-]	m Wcond [kg/s]	m Wevap [kg/s]	Q_max recovery [kW]	Q recovery [kW]	LMTD [ºC]	UA recovery [kW/K]	Q_GW [kW]
43.85	35.09	10	60	20	10	18.75	18.75	25	0	0.8395	0	0.2098	0.8395	43.85	30.7	25.95	1.183	13.15
49.02	39.82	10	60	20	15	18.88	18.88	28.5	0	1.905	0	0.2345	1.905	44.12	30.88	24.8	1.245	18.14
50.16	40.87	10	60	25	15	22.73	22.73	28.5	0	0.9777	0	0.24	0.9777	45.15	31.6	23.41	1.35	18.56
55.81	46.01	10	60	25	20	23.4	23.4	32	0	2.202	0	0.267	2.202	44.65	31.25	22.06	1.417	24.56
57.04	47.13	10	60	30	20	26.78	26.78	32	0	1.127	0	0.2729	1.127	45.63	31.94	20.84	1.533	25.1
63.63	53.1	10	60	30	25	27.94	27.94	35.5	0	2.541	0	0.3044	2.541	44.54	31.18	19.32	1.614	32.45
65.27	54.59	10	60	35	25	30.86	30.86	35.5	0	1.306	0	0.3123	1.306	45.69	31.98	18.26	1.751	33.29
72.73	61.31	10	60	35	30	32.49	32.49	39	0	2.933	0	0.348	2.933	43.64	30.54	16.57	1.844	42.19
39.86	31.02	25	60	20	10	45	20	32.76	0.5301	0.2121	0.7143	0.2725	0.7422	44.32	31.02	18.61	1.667	8.84
44.63	35.35	25	60	20	15	46.5	20	32.28	1.423	0.2685	0.8413	0.3051	1.691	50.5	35.35	15.31	2.308	9.28
45.72	36.33	25	60	25	15	46.5	25	32.18	0.5933	0.2759	0.6825	0.3125	0.8692	51.91	36.33	15.27	2.38	9.39
51.15	41.23	25	60	25	20	24.96	24.96	32	0	1.973	0	0.3496	1.973	58.45	40.92	21.5	1.903	10.23
52.44	42.4	25	60	30	20	29.9	29.9	32	0	1.014	0	0.3584	1.014	59.93	41.95	19.68	2.131	10.49
58.49	47.84	25	60	30	25	29.28	29.28	35.5	0	2.289	0	0.3998	2.289	58.49	40.95	18.84	2.174	17.54
60.02	49.21	25	60	35	25	33.54	33.54	35.5	0	1.177	0	0.4103	1.177	60.02	42.02	17.27	2.433	18
66.82	55.27	25	60	35	30	33.63	33.63	39	0	2.644	0	0.4567	2.644	57.27	40.09	16.16	2.481	26.73
38.43	29.55	30	60	20	10	45	20	36.93	0.5049	0.202	0.7143	0.3064	0.7069	42.21	29.55	20.39	1.449	8.88
43.02	33.7	30	60	20	15	46.5	20	36.5	1.356	0.2559	0.8413	0.3431	1.612	48.14	33.7	17.19	1.96	9.32
44.05	34.64	30	60	25	15	46.5	25	36.41	0.5655	0.263	0.6825	0.3513	0.8286	49.48	34.63	17.15	2.019	9.42
49.28	39.33	30	60	25	20	48	25	36.06	1.546	0.3361	0.8214	0.393	1.882	56.19	39.33	13.93	2.824	9.95
50.51	40.44	30	60	30	20	48	30	35.98	0.6219	0.3455	0.6429	0.4028	0.9674	57.77	40.44	13.9	2.91	10.07
56.36	45.68	30	60	30	25	49.5	30	35.68	1.74	0.446	0.7959	0.4494	2.186	65.26	45.68	10.59	4.313	10.68
57.83	46.98	30	60	35	25	49.5	35	35.63	0.6652	0.4588	0.5918	0.4612	1.124	67.12	46.98	10.56	4.448	10.85
64.46	52.88	30	60	35	30	34.27	34.27	39	0	2.53	0	0.514	2.53	64.46	45.12	15.93	2.833	19.34

Q cond [kW]	Q evap [kW]	Target (Tho) [ºC]	Thi [ºC]	Target T_evapout [ºC]	Tci [ºC]	Tco [ºC]	T_evapout [ºC]	Tho [ºC]	m_b [kg/s]	m_r [kg/s]	X_b [-]	m Wcond [kg/s]	m Wevap [kg/s]	Q_max recovery [kW]	Q recovery [kW]	LMTD [ºC]	UA recovery [kW/K]	Q_GW [kW]
36.94	28.01	35	60	20	10	45	20	41.04	0.4786	0.1914	0.7143	0.3535	0.67	40.01	28.01	22.06	1.27	8.93
41.34	31.98	35	60	20	15	46.5	20	40.66	1.287	0.2428	0.8413	0.3956	1.53	45.68	31.98	18.94	1.689	9.36
42.32	32.86	35	60	25	15	46.5	25	40.59	0.5366	0.2496	0.6825	0.405	0.7861	46.94	32.86	18.9	1.738	9.46
47.35	37.36	35	60	25	20	48	25	40.27	1.468	0.3192	0.8214	0.4531	1.787	53.37	37.36	15.78	2.368	9.99
48.51	38.4	35	60	30	20	48	30	40.21	0.5905	0.3281	0.6429	0.4642	0.9186	54.85	38.4	15.75	2.438	10.11
54.13	43.42	35	60	30	25	49.5	30	39.95	1.653	0.4239	0.7959	0.518	2.077	62.02	43.42	12.59	3.447	10.71
55.53	44.65	35	60	35	25	49.5	35	39.9	0.6321	0.436	0.5918	0.5313	1.068	63.78	44.65	12.57	3.551	10.88
61.9	50.29	35	60	35	30	51	35	39.69	1.833	0.5729	0.7619	0.5924	2.406	71.84	50.28	9.341	5.383	11.62
35.38	26.39	40	60	20	10	45	20	45.08	0.4509	0.1804	0.7143	0.4232	0.6313	37.7	26.39	23.64	1.116	8.99
39.59	30.17	40	60	20	15	46.5	20	44.76	1.214	0.2291	0.8413	0.4736	1.443	43.1	30.17	20.57	1.467	9.42
40.51	31	40	60	25	15	46.5	25	44.7	0.5062	0.2354	0.6825	0.4846	0.7416	44.29	31	20.54	1.509	9.51
45.32	35.28	40	60	25	20	48	25	44.43	1.387	0.3014	0.8214	0.5421	1.688	50.4	35.28	17.48	2.018	10.04
46.42	36.26	40	60	30	20	48	30	44.38	0.5576	0.3098	0.6429	0.5552	0.8673	51.79	36.25	17.46	2.076	10.17
51.8	41.04	40	60	30	25	49.5	30	44.16	1.563	0.4007	0.7959	0.6197	1.964	58.63	41.04	14.4	2.851	10.76
53.12	42.19	40	60	35	25	49.5	35	44.11	0.5974	0.412	0.5918	0.6354	1.009	60.27	42.19	14.38	2.934	10.93
59.22	47.56	40	60	35	30	51	35	43.94	1.734	0.5418	0.7619	0.7084	2.276	67.94	47.56	11.29	4.213	11.66
32.11	22.78	55	60	20	10	45	20	56.45	0.3893	0.1557	0.7143	1.536	0.545	32.54	22.78	27.82	0.8187	9.33
35.88	26.14	55	60	20	15	46.5	20	56.36	1.052	0.1985	0.8413	1.717	1.251	37.34	26.14	24.88	1.05	9.74
36.68	26.85	55	60	25	15	46.5	25	56.34	0.4384	0.2039	0.6825	1.755	0.6423	38.35	26.85	24.88	1.079	9.83
40.98	30.64	55	60	25	20	48	25	56.26	1.204	0.2618	0.8214	1.961	1.466	43.78	30.64	21.94	1.397	10.34
41.92	31.47	55	60	30	20	48	30	56.25	0.4839	0.2689	0.6429	2.006	0.7528	44.95	31.47	21.93	1.435	10.45
46.76	35.71	55	60	30	25	49.5	30	56.18	1.36	0.3487	0.7959	2.237	1.709	51.02	35.71	19	1.88	11.05
47.88	36.69	55	60	35	25	49.5	35	56.17	0.5195	0.3582	0.5918	2.291	0.8777	52.41	36.69	19	1.931	11.19
53.34	41.43	55	60	35	30	51	35	56.12	1.51	0.472	0.7619	2.552	1.982	59.19	41.43	16.07	2.579	11.91

Table 6: Simulation at constant efficiency

Another simulation is made supposing that the efficiency can vary more than the UA in the different operative conditions so the calculations are made for three fixed values of UA: 2, 4, 6 kW/K and with the efficiency free to change.

The results show that passing from UA=2 kW/K to 4 there is a notable improvement in the electric resistance energy consumption but it is not so relevant passing from UA=4 to 6 kW/K; this can be seen in the following Figure 19, Figure 20 and Figure 21 where the power Qresist that the electric resistance has to furnish to the water entering in Evap1 is plotted in function of the water inlet temperature in the condenser in the different cases.

As can be seen a heat exchanger with UA near to 4 is preferable in order to minimize the resistance power consumption and at the same time use a heat exchanger with a reasonable UA value.

To choose the real heat exchanger from a catalogue is defined the operative point that represents the worst case scenario. The adopted criteria for this choice is to see in the simulation with UA=4 kW/K when the water mass flow in the condensing side is the lowest and contemporary when the bypass has to be maintained closed in order to fulfil as much as possible the required heat exchange between circuit 1 and 2; that operative point is: W25-20 W10-60.

The real HX is chosen basing on the calculations done for this operative point; the characteristic values, e.g. mass flow and temperature, are used as input in the commercial software SSP G7 provided by SWEP [14] to find a suitable commercial heat exchanger.

The software suggested a heat exchanger B10T with 42 plates.

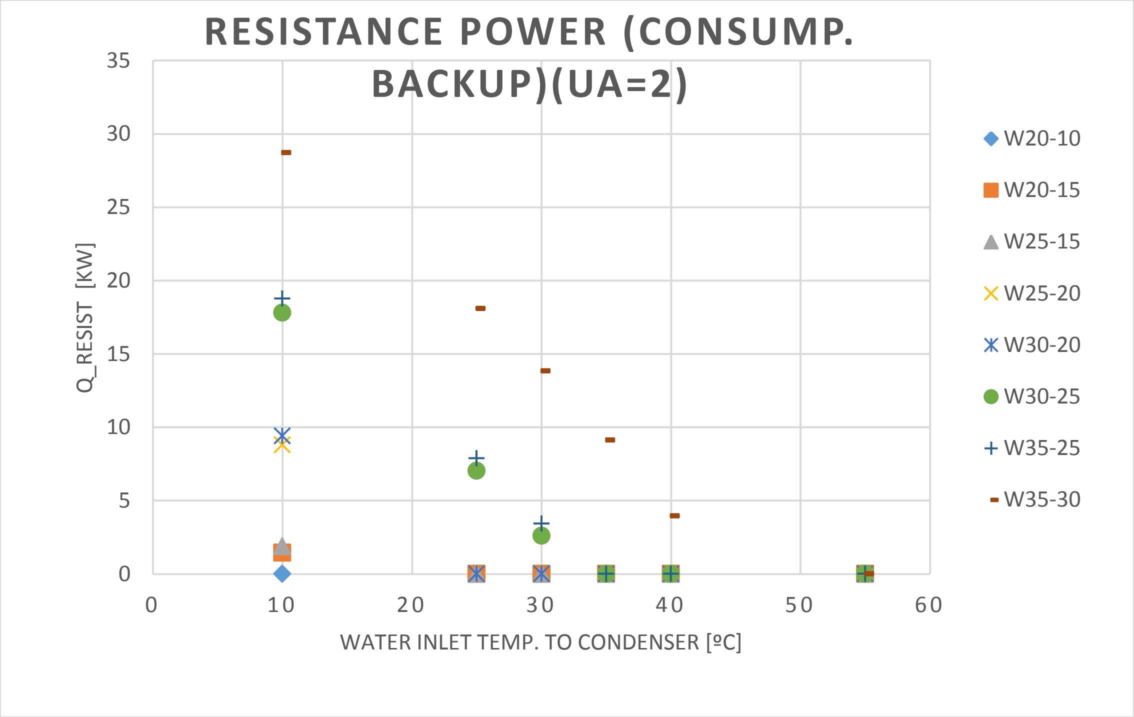


Figure 19: Resistance power, UA=2 kW/K

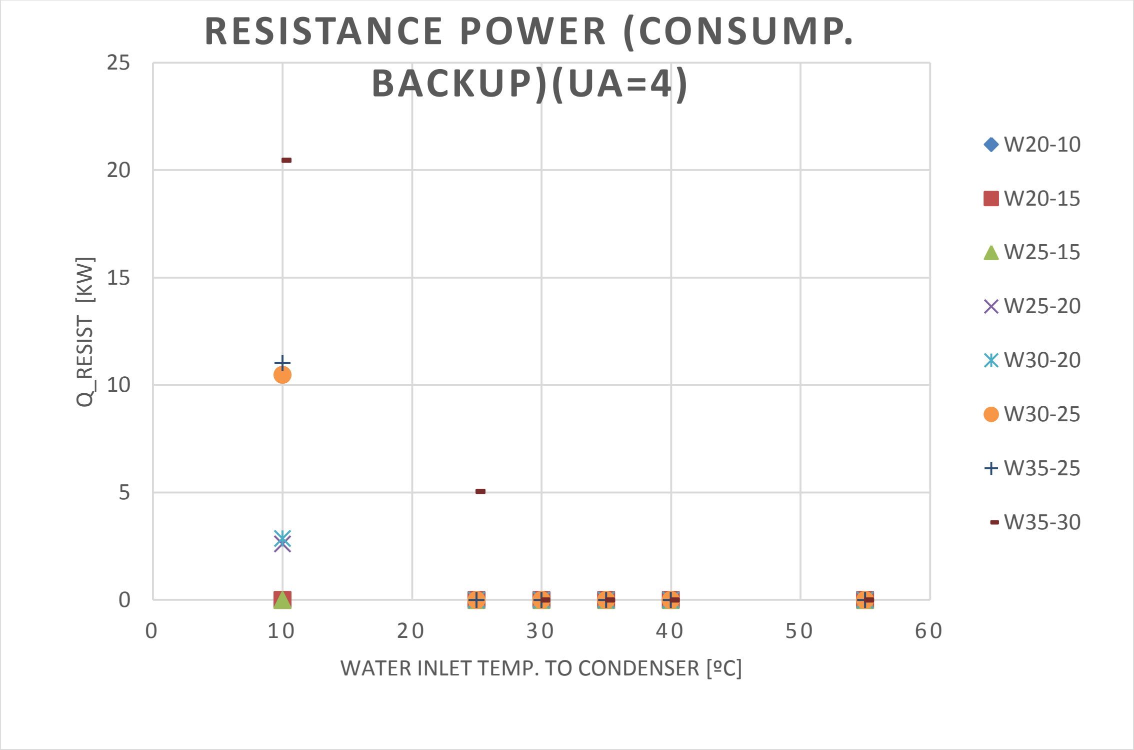


Figure 20: Resistance power, UA=4 kW/K

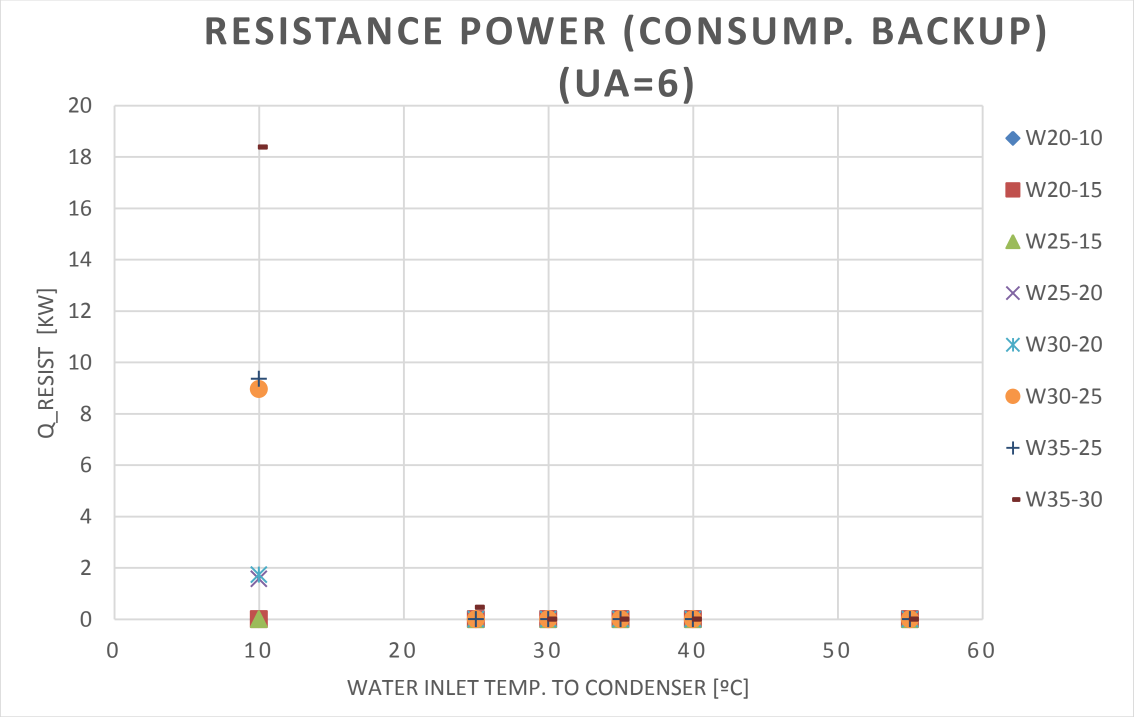


Figure 21: Resistance power, UA=6 kW/K

As actually the UA of a heat exchanger is not constant in the different working situations, in order to know exactly UA and efficiency for each operating point a parametric study has been done with the selected B10T: the UA and efficiency values are interpolated from the results of the software SSP G7.

Finally the calculations are made taking into account also the fact that the target temperature on the HX_1 outlet has to be lower of the target evaporator inlet temperature because as yet said we want to control it exactly with the electric heat tank.

The results obtained once again with EES confirm B10T as an appropriate heat exchanger: the following graph shows the power Qresist that the electric resistance has to furnish to the water entering in the Evap1 plotted in function of the water inlet temperature in the condenser in the different cases:

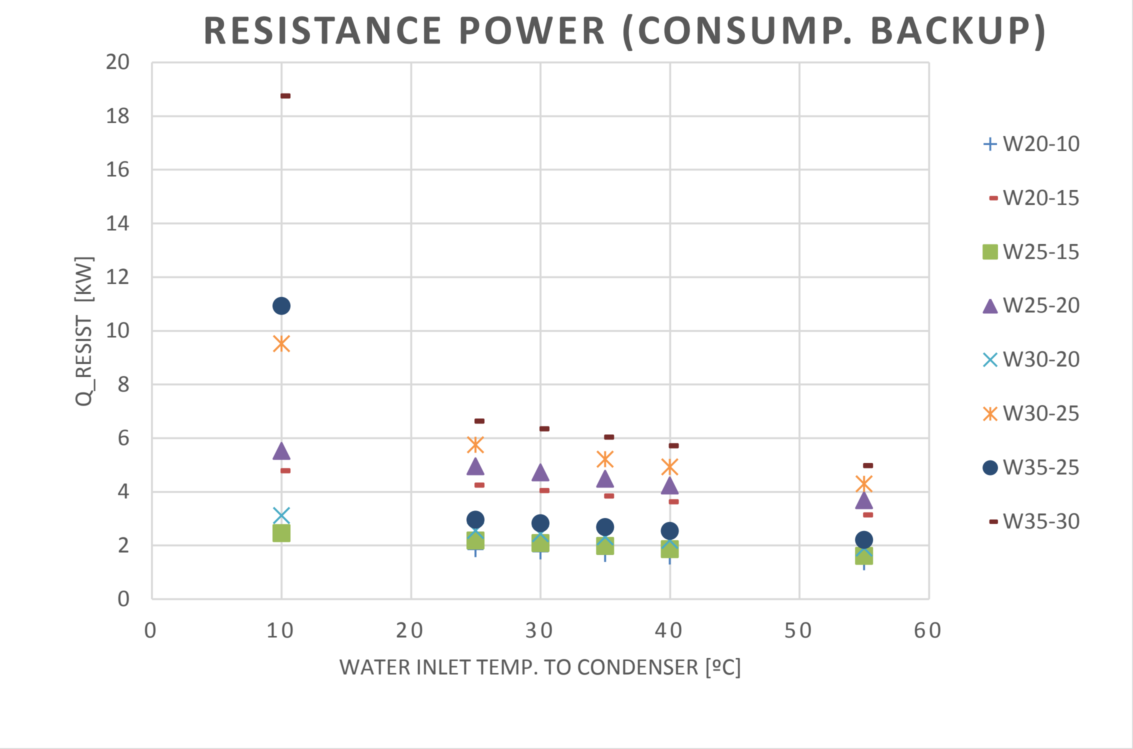


Figure 22: Resistance power for the B10T heat exchanger

As a heat exchanger with a bigger capacity than the needed one is available and not used in the UPV Thermal Laboratory, the heat exchanger Swep B25T-60 from Ciatesa Aquapark MI has been installed instead of B10T.

3.5 Heat exchanger water-water glycol

Once chose the recovery heat exchanger HX_1 and its bypass control, the heat flux that HX_1 exchanges is known so it is possible to estimate the heat power that will be dissipated downstream and finally rejected to the ambient.

The heat flux that has to be exchanged among circuit 2 and circuit 3 is not only the difference between the heat flux exchanged among the propane and the water in C1 and SB ($Qcond$) and the heat flux among HX_1 ($Qrecovery$).

Calling $Qcond$ heat flux exchanged among the propane and the water in C1 and SB; and calling $Qrecovery$ the one through HX_1, $Q_{\_WG}$ is defined as:

$Q_{\_WG} = Qcond - Qrecovery$.

So the heat flux that has to be exchanged among circuit 2 and circuit 3 is not only $Q_{\_WG}$ in fact we consider that Pump2 adds to $Q_{\_WG}$ a contribute of 10% of its value so the estimated heat exchange between circuit 2 and 3 is $Q_{WG} * 1.1$.

The design conditions for HX_2 are 5 K between outlet of water and inlet of glycol and a heating of 10 K for the glycol between 0ºC to 10ºC, under these restrictions the worst case scenario (the one that needs the higher UA) is W35-30W10-60 and has been calculated with EES knowing the point previously calculated with IMST-ART.

The software SSP G7 suggested for the characteristic values of the worst case scenario is the heat exchanger B10Tx26 so the immediately available commercial one of bigger nearer size is B10Tx30 which is installed in the circuit.

3.6 R410A compressor

The compressor had been chosen from the catalogue of the constructor.

The maximum cooling load to export from Circuit 3 is calculated multiplying the maximum value of $Q_{\_WG}$ for the coefficient 1.1, the biggest $Q_{\_WG}$ (33.17 kW) has been calculated with the previously used system on EES.

From the constructor's catalogue the smallest compressor that can satisfy that cooling capacity in the operative conditions (evaporator temperature -10ºC and condensing temperature 45ºC) is the selected Danfoss SH 240-4.

3.7 Heat exchanger water glycol-R410A

Once again to take advantage of the facilities that are in disuse in the UPV laboratory, a Swep B25T-60 from Ciatesa Aquapark MI has been installed as heat exchanger between Circuit 3 and 4.

To control if the chosen heat exchanger can satisfy the maximum cooling load to export from Circuit 3 of 36,49 kW (working point W35-30W10-60), a simulation is made with IMST-ART inserting in the software the designed evaporator, condenser and compressor of the refrigerant cycle, the values of subcooling, superheating and the massflow and inlet temperature of secondary fluids (water glycol and air).

For every working point we controlled that the heat exchange needed is lower than the feasible one with the designed components of Circuit 4, this condition is always true so the selected components are the proper ones for our installation. For example in the working point W35-30W10-60 the needed heat exchange calculated with EES is 36,49 kW while the feasible one elaborated by Circuit 4 is 41,79 kW, it means that in this working point Compressor2 will be ON for the 36,49/41,79 of the time.

3.8 R410A expansion valve

Once designed the compressor the expansion valve has been chosen from the catalogue of the constructor inserting the operative conditions:

Evaporating temperature: -10°C
Condensing temperature: 40 °C
Subcooling: 5 K
System cooling capacity: 50 kW
Pressure drop in the liquid line: 0,20 bar

The value of subcooling , evaporating and condensing temperature were fixed after the simulation with IMST-ART made with the available evaporator Swep B25T-60 (see Appendix). The pressure drop in the liquid line has been estimated by the knowledge acquired with the previous projects. The value of 50 kW for system cooling capacity is because in the simulation with IMST-ART the biggest cooling capacity that the refrigerating cycle can join is of 41.79 kW (working point W35-30W10-60) and the value of 50 kW errs in the side of safety. From the catalogue the expansion valve that can work in our capacity range is EX5 while the immediately smaller EX4 cannot exceed the capacity of 22 kW.

Finally we selected the expansion valve EX5 that has a capacity range of 6,7÷67 kW, with its controller EC3-X33.

3.9 Diameter of the pipes

The secondary circuits pipes diameters have been designed taking into account the velocity that the fluid can join into the piping, for every fluid exists a maximum velocity that does not have to be overtaken to preserve the integrity of the pipes from mechanical stress and to avoid loud noise, similarly do not descend under a minimum velocity averts the formation of deposits in the circuits.

In the following paragraph will be used the American designation for pipe sizes: the refrigerant pipes are sized by their outside diameter OD in inches; differently the water plumbing are designated by their nominal diameter NB (Normal Bore) that is 1/8" smaller than the outside diameter, the real inside diameter depends on the wall thickness indicated by the type K, L or M (in our installation L).

3.9.1 Refrigerant circuit

The following dimensioning has been done complying the prescription of Trane Company Clinic [15]: "Refrigerant Piping (TRC006EN.PPT) 2002 American Standard Inc." that is a review of refrigeration system piping considerations, design guidelines, and sizing recommendations.

3.9.1.1 Suction line

This pipe conducts low-pressure refrigerant vapour from the evaporator to the compressor. This line has to ensure an adequate velocity to return oil to compressor, it has to avoid excessive noise and to minimize the efficiency losses in terms of pressure drop. To fulfil these requirements the diameter of the suction line must be small enough that the resulting refrigerant velocity is sufficiently high to carry oil droplets in all the working conditions but contemporary, if the velocity in the pipe is too high, an undesired noise may result. Furthermore the pipe diameter should be as large as possible to minimize pressure drop and so maximize system efficiency.

The steps followed in the selection of the proper diameter of the suction line are:

- Calculation of the refrigerant velocity at both maximum and minimum system capacities for different diameters.
- Selection of the largest pipe diameter that will result in acceptable refrigerant velocity at both maximum and minimum capacities.

The diameter of the suction line must provide adequate velocity at both maximum and minimum capacities of the refrigerant cycle. Consequently it is important to verify that the refrigerant velocity at the different working points is below the recommended upper limit of 20 m/s and high enough to properly return oil to the compressor.

The minimum velocity required to carry oil droplets up to a vertical pipe (vertical riser) is higher for a larger diameter pipe than for a smaller diameter pipe; the reason is the different velocity profile of the refrigerant flowing inside the pipe.

Furthermore to assure the oil return to the compressor the diameter of a vertical riser does not necessarily need to be the same as the diameter of the horizontal or vertical drop sections of pipe, for this reason the horizontal or vertical drop sections can often be selected one diameter larger than the vertical riser in order to reduce the overall pressure drop in the entire suction line. In our case as the horizontal pipe in the refrigerant cycle is short we used the same diameter for the whole suction line.

The refrigerant velocity inside a pipe depends on the mass flow rate and density of the refrigerant, and on the inside diameter of the pipe, the relation between them is:

$$v = \frac{\dot{m}}{\rho * \pi * {}^{d^2}/_{2}}$$

With obvious meaning of the symbols.

To fulfil the predicted requirements the suggested velocity into the pipes are:

- Maximum velocity = 20 m/s;
- Minimum velocity for vertical risers depending on the diameter of the pipe;
- Minimum velocity for horizontal pipes and vertical drops equals to 75% of the minimum velocity for vertical risers of the same diameter.

In our case the minimum velocity for vertical drops is 5 m/s.

The diameter chosen for the suction line is 1. 3/8” inch and the calculations could be seen in

Table 7.

The maximum and minimum values of mass flow and the relative fluid density were calculated with the simulation of the entire refrigerant cycle made with IMST-ART for all the different operative points.

						Trane recommendation	
Mass Flow [kg/s]	Density [kg/m^3]	External Diameter [inch]	Internal Diameter [m]	Area [m^2]	Velocity [m/s]	Min. vel. [m/s]	Max. vel. [m/s]
0,17	13,933	1. 3/8	0,032131	0,00081085	15,05	5	20
0,231	22,2	1. 3/8	0,032131	0,00081085	12,83		

Table 7: Velocity calculation for suction line

As can be seen for that diameter the velocity is between the minimum and maximum value so the requirements are fulfilled, the same recommendations could be achieved with a diameter of 1. 5/8” but as this line is short we preferred to promote a better oil transportation than a bit lower pressure drop.

3.9.1.2 Discharge line

This pipe conducts hot and high-pressure refrigerant vapour from the compressor to the condenser. The design of the discharge line is less critical than that of the suction line because the refrigerant vapour is at higher temperature after leaving the compressor so the oil can be carried more easily than in the cooler suction line.

As in the suction line, the diameter of the discharge line must be small enough so the resulting refrigerant velocity is sufficiently high to carry oil droplets in all the working conditions but contemporary if the velocity in the pipe is too high an undesired noise may result.

Furthermore the pipe diameter should be as large as possible to minimize pressure drop and maximize compressor efficiency.

The steps followed in the selection of the proper diameter of the discharge line are the same followed for the suction line:

- Calculation of the refrigerant velocity at both maximum and minimum system capacities for different diameters.
- Selection of the largest pipe diameter that will result in acceptable refrigerant velocity at both maximum and minimum capacities.

The diameter of the discharge line must provide adequate velocity at both maximum and minimum capacities of the refrigerant cycle. Consequently it is important to verify that the refrigerant velocity at these two working points is below the recommended upper limit of 17,5 m/s and high enough to properly carry oil.

To fulfil the predicted requirements the suggested velocity into the pipes are:

- Maximum velocity = 17,5 m/s;
- Minimum velocity for vertical risers depending on the diameter of the pipe;
- Minimum velocity for horizontal pipes and vertical drops equals to 75% of the minimum velocity for vertical risers of the same diameter.

In our case the minimum velocity for vertical risers is 5 m/s.

The diameter chosen for the discharge line is 1. 1/8” inch and the calculations could be seen in Table 8.

The maximum and minimum values of mass flow and the relative fluid density were calculated with the simulation of the entire refrigerant cycle made with IMST-ART for all the different operative points.

						Trane recommendation	
Mass Flow [kg/s]	Density [kg/m^3]	External Diameter [inch]	Internal Diameter [m]	Area [m^2]	Velocity [m/s]	Min. vel. [m/s]	Max. vel. [m/s]
0,17	45,12	1,125	0,026035	0,00053236	7,08	5	17,5
0,231	76,85	1,125	0,026035	0,00053236	5,65		

Table 8: Velocity calculation for discharge line.

As can be seen for the chosen diameter the velocity is between the minimum and maximum value so the requirements are fulfilled.

3.9.1.3 Liquid line

This section of pipe carries the warm, high-pressure liquid refrigerant from the condenser to the expansion device and evaporator.

This line must be designed to assure that only liquid refrigerant, and not vapour, enters in the expansion device. The presence of refrigerant vapour upstream the expansion device can affect the valve operation and reduce the system capacity, lowering the mass flow through it.

In order to fulfil this requirement the condenser must ensure an adequate subcooling and the pressure drop through the liquid line and accessories must not be high enough to cause flashing upstream of the expansion device. Subcooling is necessary because it allows the liquid refrigerant to experience a certain pressure drop when it flows through the liquid line, without the risk of flashing.

As regards the oil transportation there are no problems like in discharge and suction line because oil and liquid refrigerant mix easily.

However, the design of the liquid line should be done minimizing the system refrigerant charge. The reason is that, between the three lines, the liquid one has the biggest impact on the quantity of refrigerant into the cycle’s circuit. So the diameter of the liquid line must be as small as possible to minimize the refrigerant charge minimizing also the installation cost.

Contrary, if the pipe is too small, the consequent pressure drop may cause flashing upstream of the expansion device decreasing the capacity of the expansion valve and the refrigerant velocity could lead to erosion of the inner surfaces of the piping and to undesired noises.

In conclusion we can say that the amount of subcooling available at the expansion device is as dependent on the ability of the condenser to provide subcooling as it is on the pressure drop in the liquid-line.

These considerations reveal a compromise in the diameter of the liquid line, it must be as small as possible to minimize the total refrigerant charge but contemporary it does not have to be as small to produce an excessive pressure drop and a following flashing of the liquid refrigerant before the expansion device.

To select the minimum diameter has been used the maximum velocity criteria choosing a value next to the limit one, while for the subcooling necessity the circuit will be filled with a calculated quantity of refrigerant mass as explained in the following lines.

The suggested minimum value of the subcooling downstream the expansion valve is 2.8ºC in order to assure a safety margin; we think it will be respected assuring a subcooling of 5ºC at the condenser exit, it will be done putting a refrigerant charge in the circuit equal to the biggest charge needed to assure the subcooling of 5ºC in each working condition, the calculation of the mass charge is made with IMST-ART fixing the subcooling at 5ºC.

The steps followed in the selection of the proper diameter of the suction line are:

- Calculation of the refrigerant velocity at both maximum and minimum system capacities for different diameters.
- Selection of the pipe diameter that will involve a refrigerant velocity near to the maximum acceptable at both maximum and minimum capacities.

To fulfil the predicted requirements the suggested velocity into the pipes are:

- Maximum velocity = 3 m/s;
- No minimum velocity is suggested for this line because of the easy mix between liquid refrigerant and oil.

The diameter chosen for the discharge line is 5/8” inch and the calculations can be seen in Table 9.

The maximum and minimum values of mass flow and liquid density were calculated with the simulation of the entire refrigerant cycle made with IMST-ART for all the different operative points.

						Trane recommendation	
Mass Flow [kg/s]	Density [kg/m^3]	External Diameter [inch]	Internal Diameter [m]	Area [m^2]	Velocity [m/s]	Min. vel. [m/s]	Max. vel. [m/s]
0,17	1104,4	0,625	0,013843	0,0001505	1,02	-	3
0,231	993,8	0,625	0,013843	0,0001505	1,54		

Table 9: velocity calculation for liquid line

Once again the diameter chose fulfils the requirement.

3.9.2 Water and water-glycol circuits

For water and water-glycol circuits, as for the refrigerant liquid line, the mass flow through the tubes does not have to be at high velocity in order to avoid excessive noise, to elude erosion of internal surfaces and to minimize the pressure drop.

The most common recommended velocity values can be seen in the following Table 9.

Fluid	Type of flux	Velocity [m/s]
Low viscosity fluids	Gravity flux	0,15÷0,30
	Pump input	0,3÷0,9
	Pump output	1,2÷3
	Connection line	1,2÷2,4
Viscous fluids	Pump entering	0,06÷0,15
	Pump discharge	0,15÷0,6
Water vapour		9÷15
Air or gas		9÷30

Table 10: Recommended velocities for fluids in pipes, values from [16]

Once fixed the maximum velocity and knowing the mass flow it is possible to calculate the minimum diameter of the pipes, the effective one will be the immediately bigger available in the market; in that way it is possible to fulfil the requirements with the lower possible cost.

From IMST-ART and EES results we know in every working point the mass flow needed from circuits 1, 2 and 3; with these values we can calculate the velocity of the fluid for three different standard diameters: 1”, 1.1/4” and 1.1/2” and compare them to the recommended velocity on

Fluid	Type of flux	Velocity [m/s]
Low viscosity fluids	Gravity flux	0,15÷0,30
	Pump input	0,3÷0,9
	Pump output	1,2÷3
	Connection line	1,2÷2,4
Viscous fluids	Pump entering	0,06÷0,15
	Pump discharge	0,15÷0,6
Water vapour		9÷15
Air or gas		9÷30

Table 10.

In our case the recommended velocities are:

- Pump input 0,3÷0,9 [m/s]
- Pump output 1,2÷3 [m/s]
- Connection line 1,2÷2,4 [m/s]

For the calculation of pressure drops we used Fanning equation that gives the value of the pressure drop in meters of water column:

$$H = \frac{(4 * f * L)}{d} * \frac{v^2}{2g}$$

Where:

H is the pressure drops in meter of water column;

f is a dimensionless coefficient of friction;

L is the length of the pipes in meters;

d is the diameter of the pipe in meters;

v is the velocity of the fluid in m/s;

g is the gravity acceleration in m/s^2.

f is in function of the flux and is calculated with the following:

1. If laminar flux ($Re \leq 2000$):

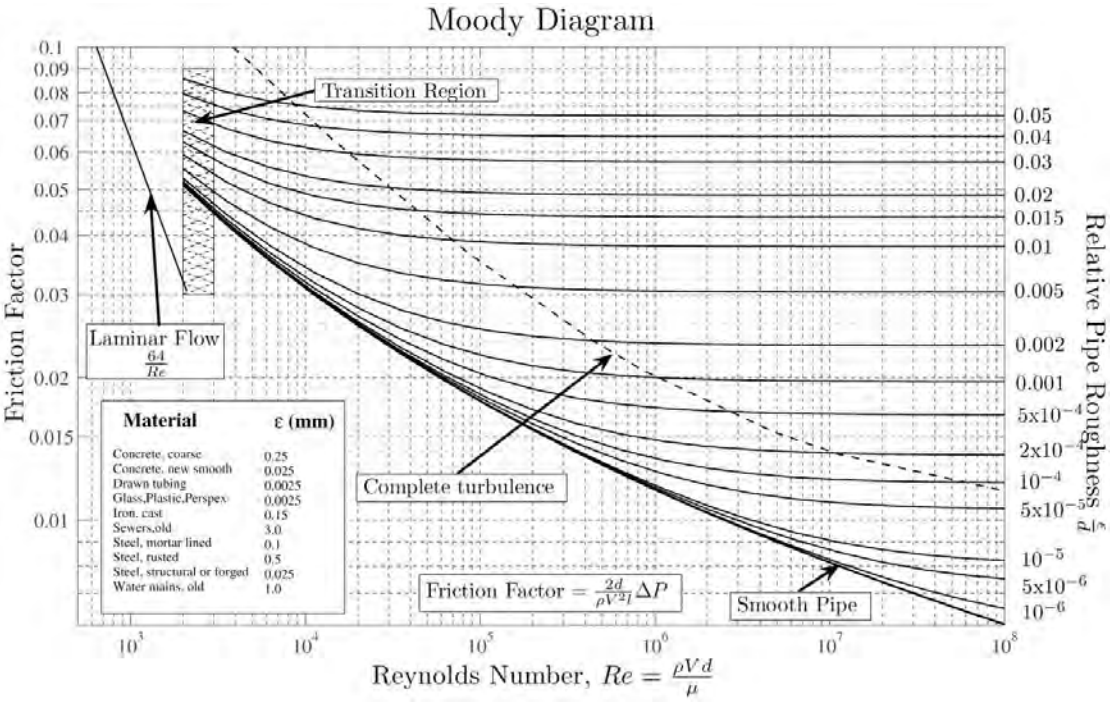
   $$f = \frac{16}{Re}$$

   Where Re is Reynolds number:

   $$Re = \frac{d * v * \rho}{\mu}$$

   Where:
   μ is the dynamic viscosity of the fluid in Pa*s,
   ρ is the density of the fluid in kg/m^3.

2. If turbulent flux ($Re>4000$) or in the transition zone ($2000<Re<4000$) it is necessary to recur to Moody diagram (Figure 23) that correlates f with Re and relative roughness ε/d where ε is a coefficient dependent of the material and d the internal diameter of the pipe (in that diagram is plotted in the y axes the Darcy-Weisbach friction factor f_D from wich it could be obtained the fanning friction factor f from the relation: $f=f_D/4$).



Material	ε (mm)
Concrete, coarse	0.25
Concrete, new smooth	0.025
Drawn tubing	0.0025
Glass,Plastic,Perspex	0.0025
Iron, cast	0.15
Sewers,old	3.0
Steel, mortar lined	0.1
Steel, rusted	0.5
Steel, structural or forged	0.025
Water mains, old	1.0

Figure 23: Moody diagram

For smooth pipes it is possible to use the Morrison correlation factor [17] for all the regimes:

$$f = \frac{0.076 * \left(\frac{3170}{Re}\right)^{0.165}}{1 + \left(\frac{3170}{Re}\right)^{7}} + \frac{16}{Re}$$

Finally to calculate the pressure drop in bar we used:

$$h = H * \rho * \mu * 10^{-5}$$

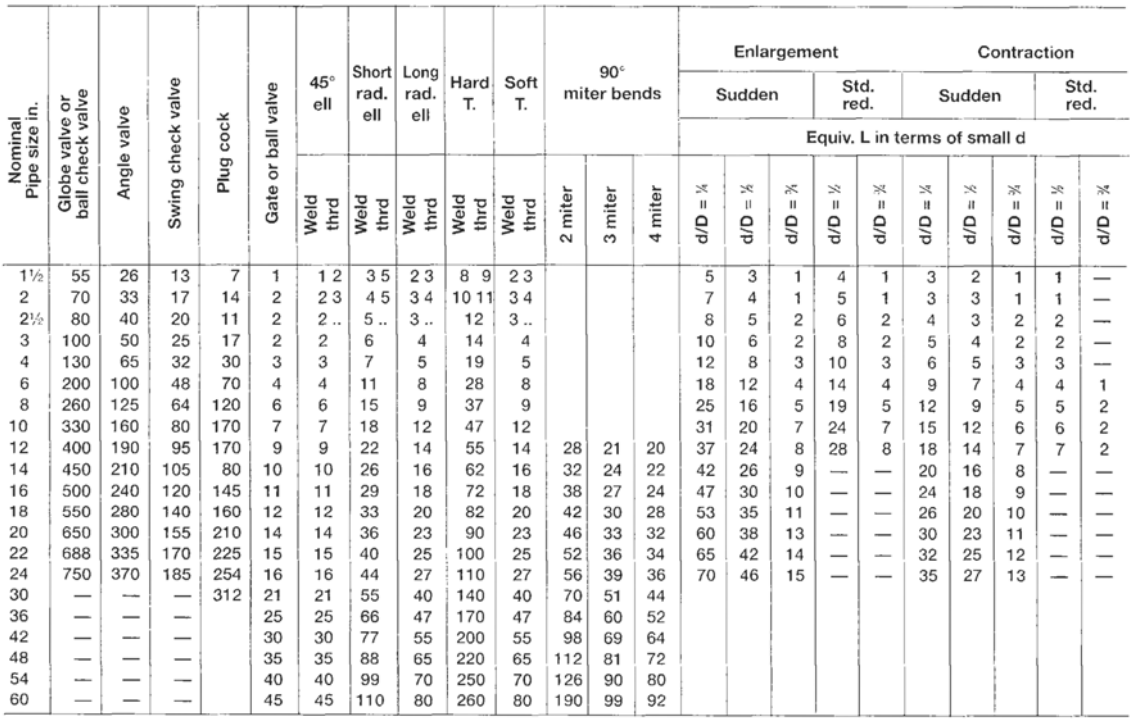
The results for the different circuits and for the different operative points are shown in Table 12, Table 13 and Table 14.

Consequently the diameters chosen for the pipes are: for the bypass lines DN 25 and for the other pipes DN 32.

As it could be seen only in few cases (mostly in the water loop in the evaporator side) the velocities overrun the recommended values but the working in these non-optimal conditions will be accepted also considering that the pressure drop in the pipes is acceptable.

To the pressure drop in the pipes has to be added the contribution of the different components present in the circuit (valves, curves, mass flow meters, heat exchangers etc.), to estimate that contribute it is possible to use a tabulated value called equivalent length; it represents the length of a straight pipe that produces the same pressure drop of the component.

These values can be found in charts in function of the type of component and of the diameter of the pipe, like in the following Table 11.



Nominal Pipe size in.	Globe valve or ball check valve	Angle valve	Swing check valve	Plug cock	Gate or ball valve	45° ell	Short rad. ell	Long rad. ell	Hard T.	Soft T.	90° miter bends			Enlargement					Contraction				
														Sudden			Std. red.		Sudden			Std. red.	
														Equiv. L in terms of small d									
						Weld thrd	Weld thrd	Weld thrd	Weld thrd	Weld thrd	2 miter	3 miter	4 miter	d/D = ¼	d/D = ½	d/D = ¾	d/D = ½	d/D = ¾	d/D = ¼	d/D = ½	d/D = ¾	d/D = ½	d/D = ¾
1½	55	26	13	7	1	1 2	3 5	2 3	8 9	2 3				5	3	1	4	1	3	2	1	1	—
2	70	33	17	14	2	2 3	4 5	3 4	10 11	3 4				7	4	1	5	1	3	3	1	1	—
2½	80	40	20	11	2	2 ..	5 ..	3 ..	12	3 ..				8	5	2	6	2	4	3	2	2	—
3	100	50	25	17	2	2	6	4	14	4				10	6	2	8	2	5	4	2	2	—
4	130	65	32	30	3	3	7	5	19	5				12	8	3	10	3	6	5	3	3	—
6	200	100	48	70	4	4	11	8	28	8				18	12	4	14	4	9	7	4	4	1
8	260	125	64	120	6	6	15	9	37	9				25	16	5	19	5	12	9	5	5	2
10	330	160	80	170	7	7	18	12	47	12				31	20	7	24	7	15	12	6	6	2
12	400	190	95	170	9	9	22	14	55	14	28	21	20	37	24	8	28	8	18	14	7	7	2
14	450	210	105	80	10	10	26	16	62	16	32	24	22	42	26	9	—	—	20	16	8	—	—
16	500	240	120	145	11	11	29	18	72	18	38	27	24	47	30	10	—	—	24	18	9	—	—
18	550	280	140	160	12	12	33	20	82	20	42	30	28	53	35	11	—	—	26	20	10	—	—
20	650	300	155	210	14	14	36	23	90	23	46	33	32	60	38	13	—	—	30	23	11	—	—
22	688	335	170	225	15	15	40	25	100	25	52	36	34	65	42	14	—	—	32	25	12	—	—
24	750	370	185	254	16	16	44	27	110	27	56	39	36	70	46	15	—	—	35	27	13	—	—
30	—	—	—	312	21	21	55	40	140	40	70	51	44										
36	—	—	—		25	25	66	47	170	47	84	60	52										
42	—	—	—		30	30	77	55	200	55	98	69	64										
48	—	—	—		35	35	88	65	220	65	112	81	72										
54	—	—	—		40	40	99	70	250	70	126	90	80										
60	—	—	—		45	45	110	80	260	80	190	99	92										

Table 11: Equivalent length of straight pipes in foot for different components

From the experience gained on previous tests on the UPV Thermal Laboratory the pressure drop on the accessories is estimated as the 21% of the pressure drop in the pipes.

The pressure drop on the coriolis is estimated as 50 kPa in every situation in the side of safety and for the three way valve 40 kPa.

The pressure drops on the heat exchangers are calculated in different ways:

for CB20-40, CB62-62, AC112-60 the following correlations are used where $\dot{m}$ is the mass flow in kg/s through the heat exchanger and Δp is in kPa.

$$\Delta p = 82{,}983 * \dot{m}^2 + \dot{m} * 20{,}762 + 0{,}9946 \quad \text{for CB20-40}$$

$$\Delta p = 10{,}686 * \dot{m}^2 + \dot{m} * 2{,}482 + 4{,}2159 \quad \text{for CB62-62}$$

$$\Delta p = 4{,}398 * \dot{m}^2 + \dot{m} * 2{,}7078 - 5{,}0544 \quad \text{for AC112-60}$$

These correlations are calculated interpolating values obtained from simulations in IMST-ART.

Differently, for the heat exchangers SWEP B10TH-30 and SWEP B25T-60, the pressure drop has been obtained for every working condition using the software of the producer: SWEP SSP G7.

In that way we calculated the total pressure drop in the different circuits for the operative points as sum of all the contributes: pressure drop on the pipes, accessories, heat exchangers.

In the following: Table 12, Table 13 and Table 14 are presented the results.

In the tables v_Wb is the velocity in m/s of the water passing through the bypasses in circuit 1 and 2, while v_Wr is the velocity of the water passing through the heat exchangers HX_1 and HX_2.

The v_Wb and v_Wr values are calculated only to have an idea of the velocities the fluid can join in these short conducts while the Reynolds number and the following pressure drops are referred to the water velocity in the rest of the circuit (v_Wevap and v_Wcond).

	Pipe sizing: Evaporator Water Loop							**Pressure Drop: Evaporator Water loop**							
	v_Wevap	v_Wb	v_Wr	Re	f	H	h	P Drop AC112x60	P Drop B25Tx60	P Drop Pipes	P Drop Accessories	3-Way valve	Coriolis	Total P_Drop	Total P_Drop
	[m/s]	[m/s]	[m/s]	[-]		m.c.l	[bar]	kPA	kPA	kPA	kPA	kPA	kPA	kPA	m.c.l.
	2,3687	3,1417	0,4464	116075,9	0,0043	4,6472	0,4559	16,0644	0,8737	45,5886	9,5736	40	50	106,9	10,9
	1,2157	1,2574	0,5839	59573,4	0,0050	1,3989	0,1372	1,7971	1,4094	13,7231	2,8819	40	50	93,2	9,5
	2,7380	3,1680	0,9783	134172,7	0,0042	6,0405	0,5926	22,2332	3,7352	59,2573	12,4440	40	50	116,0	11,8
	1,4013	0,0000	2,1227	68670,6	0,0048	1,8047	0,1770	3,5833	16,9172	17,7036	3,7178	40	50	110,5	11,3
*	3,1595	0,0000	4,7859	154828,8	0,0041	7,8307	0,7682	30,2226	82,0585	76,8190	16,1320	40	50	202,3	20,6
	1,6239	0,0000	2,4598	79577,5	0,0047	2,3519	0,2307	5,9834	22,4499	23,0724	4,8452	40	50	118,4	12,1
*	3,6469	0,0000	5,5243	178714,1	0,00400	10,1600	0,9967	40,7213	108,0230	2,21477	0,4651	40	50	238,7	24,3
	0,9228	1,1239	0,2740	45223,9	0,0053	0,8555	0,0839	-0,6220	0,3741	8,3923	1,7624	40	50	89,8	9,1
	2,1026	2,8516	0,3332	103036,4	0,0044	3,7468	0,3676	12,1005	0,5184	36,7566	7,7189	40	50	102,6	10,5
	1,0808	1,2618	0,3756	52962,3	0,0051	1,1336	0,1112	0,6219	0,6384	11,1207	2,3354	40	50	91,3	9,3
	2,4532	3,2415	0,4733	120219,3	0,0043	4,9515	0,4857	17,4083	0,9428	48,5745	10,2006	40	50	108,4	11,0
	1,2608	1,3614	0,5488	61785,3	0,0049	1,4932	0,1465	2,2133	1,2216	14,6487	3,0762	40	50	93,4	9,5
	2,8461	3,5278	0,7847	139473,8	0,0042	6,4798	0,6357	24,1872	2,4191	63,5668	13,3490	40	50	116,6	11,9
	1,4635	1,2433	0,9738	71717,2	0,0048	1,9509	0,1914	4,2254	3,6841	19,1385	4,0191	40	50	97,9	10,0
	3,2875	3,2245	1,7554	161104,8	0,0041	8,4161	0,8256	32,8503	11,5828	82,5616	17,3379	40	50	134,4	13,7
	0,8790	1,0749	0,2563	43073,0	0,0053	0,7845	0,0770	-0,9425	0,3324	7,6959	1,6161	40	50	89,4	9,1
	2,0044	2,7273	0,3096	98222,7	0,0045	3,4368	0,3372	10,7390	0,4556	33,7152	7,0802	40	50	101,2	10,3
	1,0303	1,2149	0,3458	50488,4	0,0051	1,0409	0,1021	0,2089	0,5518	10,2109	2,1443	40	50	90,8	9,3
	2,3401	3,1172	0,4276	114674,4	0,0043	4,5462	0,4460	15,6191	0,7878	44,5981	9,3656	40	50	106,4	10,8
	1,2029	1,3388	0,4833	58945,8	0,0050	1,3726	0,1347	1,6811	0,9750	13,4655	2,8277	40	50	92,7	9,4
	1,3976	1,3618	0,7551	68487,8	0,0048	1,7960	0,1762	3,5455	2,2428	17,6191	3,7000	40	50	95,8	9,8
	3,1458	3,6314	1,1333	154158,5	0,0041	7,7693	0,7622	29,9475	4,9368	76,2168	16,0055	40	50	124,9	12,7

	0,8331	1,0210	0,2409	40824,6	0,0054	0,7133	0,0700	-1,2659	0,2977	6,9972	1,4694	40	50	89,0	9,1
	1,9024	2,5936	0,2878	93226,3	0,0045	3,1279	0,3068	9,3838	0,4005	30,6848	6,4438	40	50	99,8	10,2
	0,9774	1,1608	0,3198	47898,8	0,0052	0,9476	0,0930	-0,2080	0,4808	9,2962	1,9522	40	50	90,3	9,2
	2,2220	2,9759	0,3908	108885,9	0,0044	4,1399	0,4061	13,8289	0,6728	40,6120	8,5285	40	50	104,5	10,7
	1,1422	1,2932	0,4370	55972,3	0,0050	1,2513	0,1227	1,1441	0,8165	12,2749	2,5777	40	50	92,0	9,4
	2,5825	3,3620	0,5515	126556,2	0,0043	5,4339	0,5331	19,5424	1,2144	53,3067	11,1944	40	50	110,8	11,3
	1,3279	1,3823	0,6295	65075,6	0,0049	1,6387	0,1608	2,8540	1,5715	16,0759	3,3759	40	50	94,4	9,6
	2,9916	3,6709	0,8609	146602,9	0,0041	7,0926	0,6958	26,9199	2,8886	69,5781	14,6114	40	50	119,8	12,2
	0,7850	0,9642	0,2249	38466,5	0,0055	0,6419	0,0630	-1,5922	0,2636	6,2968	1,3223	40	50	88,7	9,0
	1,7942	2,4504	0,2690	87925,2	0,0046	2,8146	0,2761	8,0107	0,3554	27,6117	5,7985	40	50	98,4	10,0
	0,9221	1,1001	0,2966	45187,3	0,0053	0,8543	0,0838	-0,6275	0,4211	8,3803	1,7599	40	50	89,8	9,2
	2,0989	2,8196	0,3599	102853,6	0,0044	3,7349	0,3664	12,0478	0,5822	36,6389	7,6942	40	50	102,6	10,5
	1,0784	1,2343	0,3993	52846,5	0,0051	1,1292	0,1108	0,6023	0,6969	11,0774	2,3263	40	50	91,3	9,3
	2,4420	3,2019	0,4959	119670,9	0,0043	4,9107	0,4817	17,2281	1,0037	48,1743	10,1166	40	50	108,2	11,0
	1,2546	1,3441	0,5569	61480,6	0,0049	1,4801	0,1452	2,1553	1,2363	14,5196	3,0491	40	50	93,4	9,5
	2,8300	3,5636	0,7231	138681,7	0,0042	6,4133	0,6291	23,8910	2,0542	62,9141	13,2120	40	50	115,9	11,8
	0,6777	0,8333	0,1931	33208,1	0,0056	0,4950	0,0486	-2,2723	0,2000	4,8560	1,0198	40	50	87,9	9,0
	1,5555	2,1246	0,2305	76226,2	0,0047	2,1768	0,2135	5,2159	0,2695	21,3545	4,4844	40	50	95,5	9,7
	0,7986	0,9564	0,2531	39136,8	0,0054	0,6618	0,0649	-1,5008	0,3169	6,4925	1,3634	40	50	88,8	9,1
	1,8228	2,4561	0,3047	89326,6	0,0046	2,8960	0,2841	8,3672	0,4331	28,4098	5,9661	40	50	98,8	10,1
	0,9360	1,0832	0,3347	45869,8	0,0052	0,8773	0,0861	-0,5236	0,5094	8,6068	1,8074	40	50	90,0	9,2
	2,1250	2,8102	0,4074	104133,2	0,0044	3,8192	0,3747	12,4184	0,7094	37,4662	7,8679	40	50	103,1	10,5
	1,0913	1,2054	0,4477	53480,2	0,0051	1,1535	0,1132	0,7103	0,8354	11,3157	2,3763	40	50	91,5	9,3
*	2,4644	3,1774	0,5558	120767,6	0,0043	4,9925	0,4898	17,5892	1,2246	48,9761	10,2850	40	50	108,8	11,1
Port velocity on secondary side is high > 5.5 m/s															

Table 12: Pipe sizing and pressure drops for Circuit 1

	Pipe sizing: Condenser Water Loop							**Pressure Drop: Condenser Water loop**									
	v_Wcond	v_Wb	v_Wr	Re	f	H	h	P Drop CB 20x40	P Drop CB 62x62	P Drop B 25Tx60	P Drop B 10Tx30	P Drop Pipes	P Drop Accessories	3-Way valve	Coriolis	Total P_Drop	Total P_Drop
	[m/s]	[m/s]	[m/s]	[-]		m.c.l	[bar]	kPA	kPA	kPA	kPA	kPA	kPA	kPA	kPA	kPA	m.c.l.
	0,2916	0,1613	0,2803	14288,6	0,0070	0,1145	0,0112	10,4265	5,3856	0,8803	0,6704	1,1235	0,2359	40	50	108,7	11,1
	0,2984	0,1870	0,2650	14623,7	0,0070	0,1191	0,0117	10,7573	5,4271	0,9245	0,6075	1,1687	0,2454	40	50	109,1	11,1
	0,3320	0,1847	0,3181	16268,9	0,0068	0,1429	0,0140	12,4538	5,6404	1,1049	0,8463	1,4023	0,2945	40	50	111,7	11,4
	0,3393	0,2126	0,3012	16628,4	0,0067	0,1484	0,0146	12,8407	5,6891	1,1553	0,7684	1,4559	0,3057	40	50	112,2	11,4
*	0,3785	0,1844	0,3889	18547,8	0,0065	0,1791	0,0176	15,0037	5,9616	1,4084	1,2190	1,7569	0,3690	40	50	115,7	11,8
*	0,3883	0,1803	0,4078	19029,1	0,0065	0,1872	0,0184	15,5720	6,0332	1,4754	1,3275	1,8364	0,3856	40	50	116,6	11,9
	0,4327	0,1381	0,5172	21204,4	0,0063	0,2258	0,0221	18,2693	6,3738	1,8174	2,0364	2,2148	0,4651	40	50	121,2	12,4
	0,3388	0,3618	0,1516	16604,0	0,0067	0,1480	0,0145	12,8143	5,6857	1,1236	0,2151	1,4522	0,3050	40	50	111,6	11,4
	0,3794	0,3893	0,1852	18590,4	0,0065	0,1798	0,0176	15,0536	5,9679	1,3974	0,3097	1,7639	0,3704	40	50	114,9	11,7
	0,3886	0,4215	0,1669	19041,3	0,0065	0,1874	0,0184	15,5865	6,0351	1,4646	0,2566	1,8384	0,3861	40	50	115,6	11,8
	0,4347	0,4515	0,2068	21301,9	0,0063	0,2276	0,0223	18,3952	6,3897	1,8191	0,3790	2,2325	0,4688	40	50	119,7	12,2
	0,4456	0,4906	0,1848	21838,1	0,0063	0,2376	0,0233	19,0949	6,4781	1,9103	0,3095	2,3310	0,4895	40	50	120,6	12,3
	0,4971	0,5215	0,2313	24360,7	0,0061	0,2874	0,0282	22,5593	6,9163	2,3612	0,4652	2,8196	0,5921	40	50	125,7	12,8
	0,5102	0,5679	0,2053	25000,5	0,0060	0,3007	0,0295	23,4831	7,0332	2,4858	0,3751	2,9502	0,6195	40	50	126,9	12,9
	0,5679	0,5999	0,2599	27827,7	0,0059	0,3628	0,0356	27,7848	7,5783	3,0614	0,5760	3,5586	0,7473	40	50	133,3	13,6
	0,3810	0,4413	0,1358	18669,6	0,0065	0,1811	0,0178	15,1466	5,9796	1,4089	0,1754	1,7769	0,3731	40	50	114,9	11,7
	0,4266	0,4801	0,1661	20905,8	0,0063	0,2203	0,0216	17,8866	6,3254	1,7530	0,2529	2,1610	0,4538	40	50	118,8	12,1
	0,4368	0,5127	0,1488	21405,5	0,0063	0,2295	0,0225	18,5294	6,4066	1,8359	0,2071	2,2513	0,4728	40	50	119,7	12,2
	0,4887	0,5554	0,1847	23946,4	0,0061	0,2790	0,0274	21,9707	6,8418	2,2802	0,3068	2,7365	0,5747	40	50	124,7	12,7
	0,5008	0,5944	0,1639	24543,5	0,0061	0,2912	0,0286	22,8213	6,9494	2,3929	0,2475	2,8567	0,5999	40	50	125,9	12,8
	0,5735	0,6869	0,1813	28101,9	0,0059	0,3690	0,0362	28,2210	7,6336	3,1124	0,2975	3,6203	0,7603	40	50	133,6	13,6
	0,6391	0,7372	0,2307	31319,2	0,0057	0,4465	0,0438	28,6409	8,3148	3,8404	0,4610	4,3799	0,9198	40	50	136,6	13,9

	0,4395	0,5424	0,1232	21539,5	0,0063	0,2320	0,0228	18,7037	6,4286	1,8571	0,1462	2,2759	0,4779	40	50	119,9	12,2
	0,4919	0,5946	0,1505	24104,8	0,0061	0,2822	0,0277	22,1948	6,8701	2,3082	0,2105	2,7682	0,5813	40	50	124,9	12,7
	0,5036	0,6285	0,1344	24677,5	0,0061	0,2940	0,0288	23,0145	6,9739	2,4166	0,1715	2,8840	0,6056	40	50	126,1	12,9
	0,5634	0,6863	0,1667	27608,4	0,0059	0,3578	0,0351	27,4382	7,5343	3,0022	0,2535	3,5096	0,7370	40	50	132,5	13,5
	0,5772	0,7268	0,1475	28284,7	0,0059	0,3733	0,0366	28,5136	7,6707	3,1476	0,2032	3,6617	0,7690	40	50	134,0	13,7
	0,6441	0,7907	0,1852	31562,9	0,0057	0,4526	0,0444	28,6409	8,3689	3,8916	0,3071	4,4401	0,9324	40	50	136,6	13,9
	0,6606	0,8374	0,1625	32373,3	0,0057	0,4733	0,0464	28,6409	8,5510	4,0896	0,2423	4,6429	0,9750	40	50	137,1	14,0
	0,7366	0,9092	0,2070	36096,2	0,0055	0,5735	0,0563	28,6409	9,4364	5,0497	0,3764	5,6265	1,1816	40	50	140,3	14,3
	0,5262	0,6846	0,1130	25786,5	0,0060	0,3174	0,0311	24,6432	7,1801	2,6298	0,1244	3,1142	0,6540	40	50	128,3	13,1
	0,5889	0,7541	0,1376	28857,5	0,0058	0,3866	0,0379	28,6409	7,7882	3,2693	0,1781	3,7928	0,7965	40	50	134,5	13,7
	0,6026	0,7896	0,1226	29527,8	0,0058	0,4025	0,0395	28,6409	7,9281	3,4190	0,1445	3,9489	0,8293	40	50	134,9	13,8
	0,6740	0,8690	0,1518	33031,4	0,0056	0,4904	0,0481	28,6409	8,7017	4,2475	0,2128	4,8105	1,0102	40	50	137,6	14,0
	0,6903	0,9110	0,1341	33829,6	0,0056	0,5115	0,0502	28,6409	8,8878	4,4501	0,1702	5,0175	1,0537	40	50	138,2	14,1
	0,7705	0,9998	0,1680	37759,7	0,0055	0,6211	0,0609	28,6409	9,8577	5,5051	0,2562	6,0934	1,2796	40	50	141,6	14,4
	0,7901	1,0499	0,1473	38716,3	0,0054	0,6493	0,0637	28,6409	10,1073	5,7808	0,2020	6,3694	1,3376	40	50	142,4	14,5
	0,8808	1,1478	0,1872	43164,4	0,0053	0,7875	0,0772	28,6409	11,3367	7,1364	0,3121	7,7250	1,6222	40	50	146,8	15,0
	1,9099	2,8026	0,0934	93591,9	0,0045	3,1501	0,3090	28,6409	33,2397	31,9825	0,0873	30,9021	6,4895	40	50	221,3	22,6
	2,1349	3,1285	0,1124	104620,6	0,0044	3,8515	0,3778	28,6409	39,9808	39,7021	0,1221	37,7835	7,9345	40	50	244,2	24,9
	2,1822	3,1963	0,1002	106936,0	0,0044	4,0069	0,3931	28,6409	41,4850	41,4292	0,0991	39,3075	8,2546	40	50	249,2	25,4
	2,4383	3,5730	0,1226	119488,1	0,0043	4,8972	0,4804	28,6409	50,1763	51,3888	0,1429	48,0413	10,0887	40	50	278,5	28,4
	2,4943	3,6653	0,1083	122230,0	0,0043	5,1024	0,5005	28,6409	52,1956	53,7074	0,1142	50,0544	10,5114	40	50	285,2	29,1
	2,7815	4,0872	0,1343	136305,4	0,0042	6,2155	0,6097	28,6409	63,2427	66,3627	0,1686	60,9745	12,8046	40	50	322,2	32,8
	2,8486	4,1926	0,1178	139595,7	0,0042	6,4901	0,6367	28,6409	65,9896	69,5139	0,1329	63,6675	13,3702	40	50	331,3	33,8
*	3,1732	4,6579	0,1482	155499,0	0,0041	7,8923	0,7742	28,6409	80,1447	85,7124	0,2015	77,4233	16,2589	40	50	378,4	38,6
*=Port velocity on primary side is higher (> 5.5 m/s:).																	

Table 13: Pipe sizing and pressure drops for Circuit 2

Pipe sizing: Water Glycol Loop					**Pressure Drop: Water Glycol Loop**					
v	Re	f	H	h	P Drop BT10T	P Drop BT25T	P Drop Pipes	P Drop Accessories	Total P_Drop	Total P_Drop
[m/s]	[-]		m.c.l	[bar]	kPA	kPA	kPA	kPA	kPA	m.c.l.
0,6988	6334,3	0,0093	2,8782	0,2824	8,0573	5,5330	28,2354	5	46,8	4,8
0,5874	5324,6	0,0098	2,1544	0,2113	5,8508	4,4140	21,1348	5	36,4	3,7
0,7661	6944,7	0,0090	3,3479	0,3284	9,5513	6,3974	32,8434	5	53,8	5,5
0,6510	5901,3	0,0095	2,5604	0,2512	7,0758	5,0070	25,1175	5	42,2	4,3
1,0025	9088,0	0,0081	5,2149	0,5116	15,6967	9,7681	51,1581	5	81,6	8,3
1,0798	9788,7	0,0079	5,8998	0,5788	18,0044	11,0493	57,8773	5	91,9	9,4
1,5074	13665,2	0,0071	10,3405	1,0144	33,3278	19,0490	101,4407	5	158,8	16,2
0,5352	4851,4	0,0100	1,8320	0,1797	4,9057	3,9705	17,9715	5	31,8	3,2
0,6765	6133,0	0,0094	2,7290	0,2677	7,5673	5,2987	26,7713	5	44,6	4,6
0,5783	5241,9	0,0098	2,0975	0,2058	5,6631	4,3222	20,5768	5	35,6	3,6
0,7433	6737,9	0,0091	3,1859	0,3125	9,0058	6,0869	31,2537	5	51,3	5,2
0,6296	5707,6	0,0096	2,4220	0,2376	6,6299	4,8012	23,7602	5	40,2	4,1
0,8197	7431,0	0,0087	3,7415	0,3670	10,7941	7,0859	36,7040	5	59,6	6,1
0,6888	6244,0	0,0093	2,8109	0,2758	7,8281	5,4150	27,5753	5	45,8	4,7
0,9093	8242,6	0,0084	4,4374	0,4353	13,0757	8,3539	43,5314	5	70,0	7,1
0,5329	4830,7	0,0100	1,8180	0,1783	4,8617	3,9420	17,8345	5	31,6	3,2
0,6683	6057,9	0,0094	2,6740	0,2623	7,3901	5,1877	26,2319	5	43,8	4,5
0,5747	5209,3	0,0099	2,0752	0,2036	5,5919	4,2915	20,3573	5	35,2	3,6
0,7333	6647,6	0,0091	3,1161	0,3057	8,7759	5,9560	30,5687	5	50,3	5,1
0,6252	5667,4	0,0096	2,3935	0,2348	6,5369	4,7671	23,4807	5	39,8	4,1
0,6833	6193,9	0,0093	2,7739	0,2721	7,7045	5,3752	27,2117	5	45,3	4,6
0,8965	8127,3	0,0085	4,3355	0,4253	12,7281	8,2031	42,5317	5	68,5	7,0
0,5306	4810,0	0,0101	1,8040	0,1770	4,8183	3,9420	17,6977	5	31,5	3,2

0,6601	5984,0	0,0094	2,6202	0,2570	7,2178	5,1148	25,7038	5	43,0	4,4
0,5719	5184,3	0,0099	2,0580	0,2019	5,5366	4,2915	20,1893	5	35,0	3,6
0,7238	6561,7	0,0091	3,0502	0,2992	8,5598	5,8695	29,9220	5	49,4	5,0
0,6210	5629,3	0,0096	2,3667	0,2322	6,4495	4,7346	23,2171	5	39,4	4,0
0,7966	7221,0	0,0088	3,5694	0,3502	10,2188	6,8068	35,0161	5	57,0	5,8
0,6779	6145,0	0,0094	2,7378	0,2686	7,5849	5,2987	26,8576	5	44,7	4,6
0,8829	8003,3	0,0085	4,2271	0,4147	8,5849	8,0033	41,46757365	6	64,1	6,5
0,5287	4792,6	0,0101	1,7923	0,1758	4,7814	3,9135	17,5825	5	31,3	3,2
0,6524	5914,3	0,0095	2,5698	0,2521	7,0574	5,0432	25,2098	5	42,3	4,3
0,5688	5156,0	0,0099	2,0387	0,2000	5,4756	4,2610	19,9996	5	34,7	3,5
0,7138	6470,3	0,0092	2,9806	0,2924	8,3337	5,7408	29,2399	5	48,3	4,9
0,6169	5592,3	0,0097	2,3407	0,2296	6,3654	4,7011	22,9618	5	39,0	4,0
0,7847	7113,3	0,0089	3,4824	0,3416	9,9305	6,6234	34,1627	5	55,7	5,7
0,6728	6099,3	0,0094	2,7042	0,2653	7,4739	5,2621	26,5285	5	44,3	4,5
0,8688	7876,0	0,0086	4,1169	0,4039	11,9899	7,8060	40,3872	5	65,2	6,6
0,5347	4847,0	0,0100	1,8290	0,1794	4,8703	3,9705	17,9426	5	31,8	3,2
0,6443	5840,3	0,0095	2,5166	0,2469	6,8801	4,9728	24,6881	5	41,5	4,2
0,5719	5184,3	0,0099	2,0580	0,2019	5,5176	4,2915	20,1893	5	35,0	3,6
0,7010	6355,0	0,0092	2,8937	0,2839	8,0435	5,5725	28,3873	5	47,0	4,8
0,6169	5592,3	0,0097	2,3407	0,2296	6,3498	4,7011	22,9618	5	39,0	4,0
0,7666	6949,0	0,0090	3,3514	0,3288	9,4897	6,3974	32,8772	5	53,8	5,5
0,6697	6071,0	0,0094	2,6835	0,2633	7,3918	5,2237	26,3254	5	43,9	4,5
0,8439	7649,7	0,0086	3,9242	0,3850	11,3357	7,4181	38,4959	5	62,2	6,3

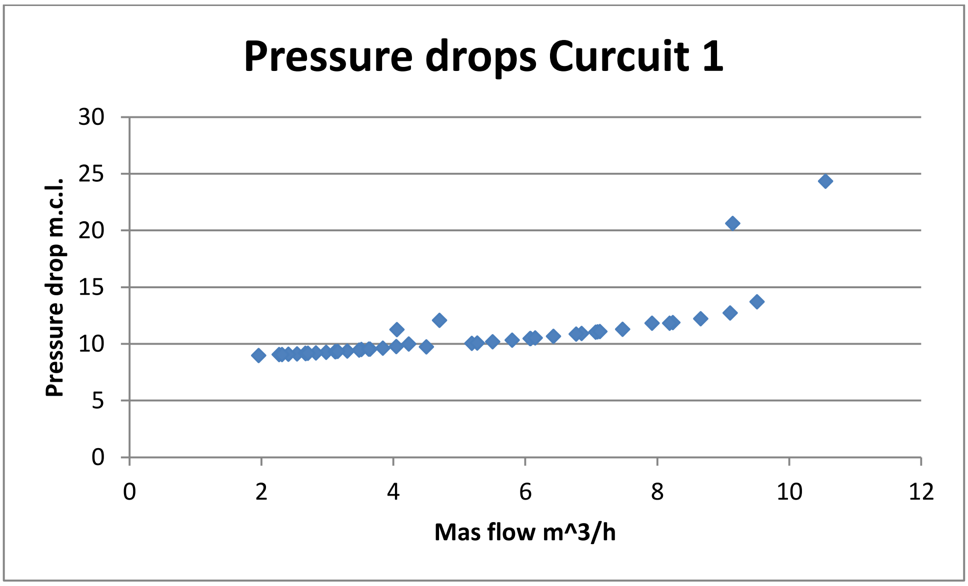
Table 14: Pipe sizing and pressure drops for Circuit 3

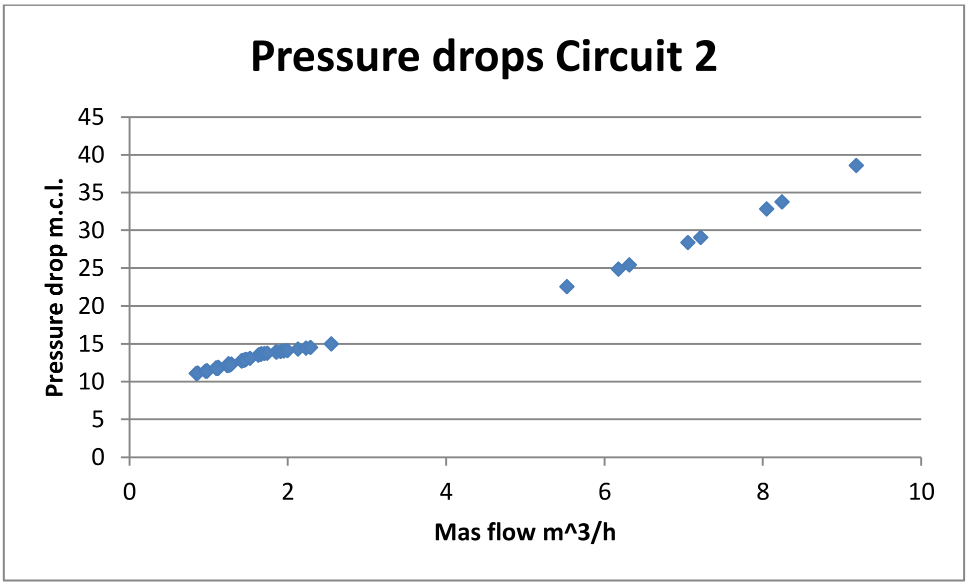
3.10 Pump dimensioning

For the dimensioning of the pumps the method followed is:

1. Calculate for every pump the friction losses in the different operative conditions;
2. Calculate the Net Positive Suction Head available $NPSH_A$;
3. Decide the adequate model of the pump with the characteristic curves of the pump;
4. Verify that $NPSH_A > NPSH_R$

The friction losses (or total pressure drop) are calculated for every circuit and working point in the previous chapter 3.9.2, once known these information we designed the pumps from the constructor catalogue; in Table 15 are shown the values of the pressure drops calculated for the working points of the different circuits.





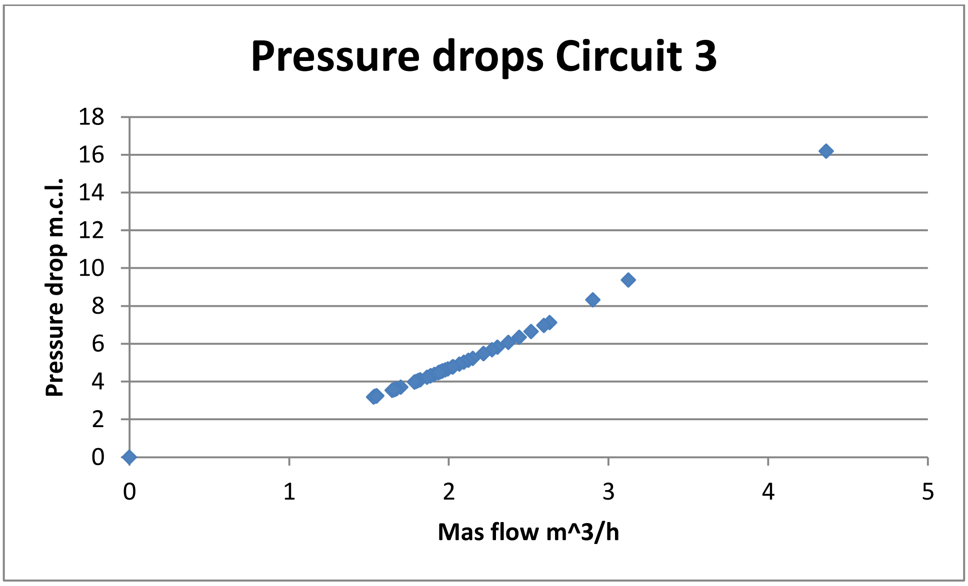


Table 15: Values of pressure drop in function of mass flow for circuits 1, 2 and 3.

The visible points represent the characteristics of the circuits in terms of mass flow and pressure drop that the pumps have to be able to satisfy. As could be seen some values of pressure drop are much more bigger than the others, that is because we used some correlations to calculate the pressure drops in the heat exchangers valid in a range of velocity that in some cases is overcome, these are in specific the cases marked with a * in Table 12 and Table 13: W30-25W10-60, W35-30W10-60 and W35-30W55-60.

During the put into operation will be clear the real pressure drop of these working points.

With these results we calculated the Net Positive Suction Head available: $NPSH_A$ with the following formula:

$$NPSH_A = \frac{p_0 - p_v}{\rho * g} - Y$$

Where:

p_0 is the pressure of the fluid at the outlet of the pump (3 bar for every circuit);

p_v is the vapour pressure of the fluid, for water it has been calculated with the equation 3.10.1 while for water glycol it is p_v = 2169 Pa from the characteristic of the fluid [18].

$$p_v = 6.11 * 10^{\frac{7.5*t}{237.7+t}} \tag{3.10.1}$$

where t is the temperature in ºC and p_v is in mbar,

ρ is the density of the fluid (1000kg/m^3 for water and 1029kg/m^3 for water-glycol at 20ºc);

g the gravity acceleration (9,81 m/s^2);

Y the pressure drop in the circuit between the outlet and the inlet of the pump.

The results of the calculation are in the following Table 16, the values of $NPSH_A$ are calculated for the worst case that is when the temperature and the pressure drop are the highest .

	tmax [ºC]	pv [mbar]	pv [Pa]	pv [m.c.l.]	p_0 [bar]	$NPSH_A$ [m]
circuit 1	34,4	54,228	5422,781	0,553	3	5,698
circuit 2	56,58	169,058	16905,781	1,723	3	3,458
circuit 3	20	21,69	2169	0,215	3	13,504

Table 16: $NPSH_A$ calculation

From the constructor catalogue are chosen the following pumps that satisfy the requirements.

3.10.1 Pump1

The selected pump is Grundfoss CRE 10-2 that can satisfy the big range of mass flow and the pressure drop for the different working point varying the velocity with an inverter.

In order to regulate in a precise way the low mass flow situations has been installed a bypass with a needle valve taking into account the regulation difficulties observed in the previous installations in the Thermal Laboratory with pumps at variable velocity and low mass flow.

As can be seen from Figure 24, also in the case when the mass flow is maximum (10,56 kg/h, case W35-30W10-60), the minimum calculated value of $NPSH_A$=5,698 m is bigger than the $NPSH_R$ one. The $NPSH_R$ value in that working point is close to 4 meters so the safety requirement: $NPSH_A > NPSH_R + 0{,}5$ m is fulfilled and the cavitation risk avoided.

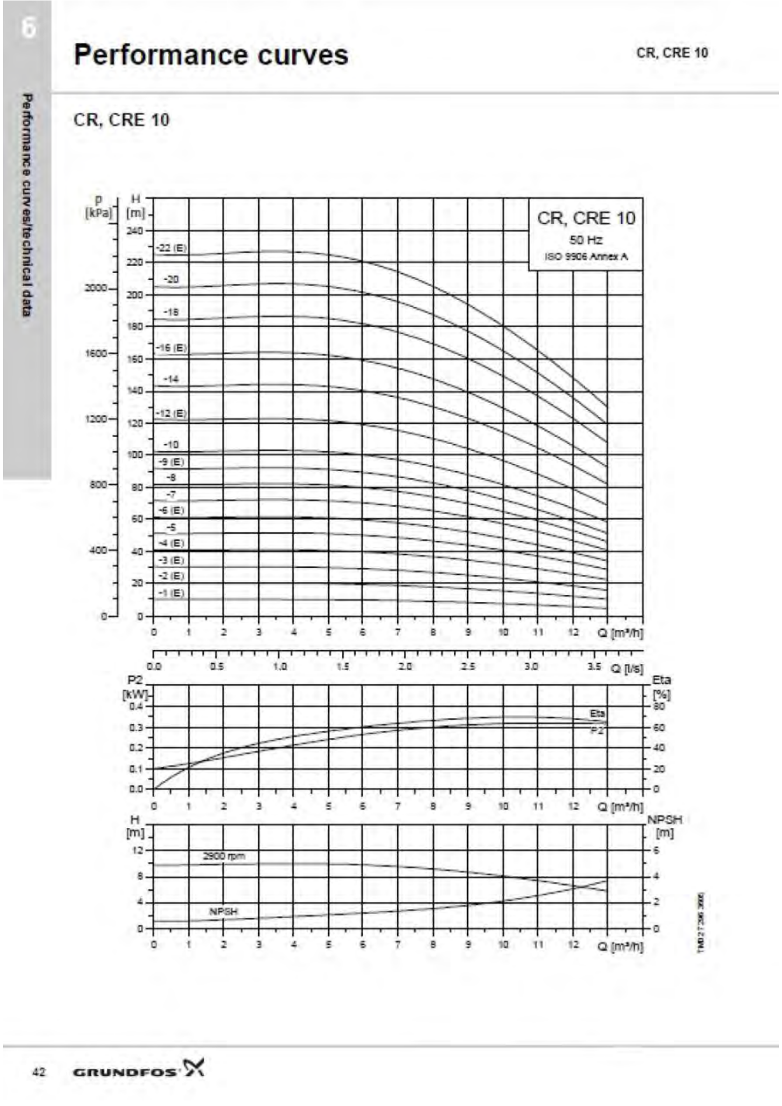


Figure 24: Pump performance curves from Grundfos Catalogue

3.10.2 Pump2

The selected pump is Grundfos CME 10-2 that can satisfy the big range of mass flow and pressure drop for the different working point varying the velocity with an inverter.

In order to regulate in a precise way the low mass flow situations has been installed a bypass with a needle for the same reasons explained for Pump 1.

As can be seen from Figure 25 the minimum calculated value of $NPSH_A$=3,458 m is near to the $NPSH_R$ one (but every time bigger) only for the 3 last cases when the mass flow is maximum, that is for example in the working point W35-30W55-60. As these points represent limit situations (high temperature for the water available in the evaporator side: 30or 35ºC) we think it is not necessary to buy a pump with a smaller $NPSH_R$ only for these particular working points.

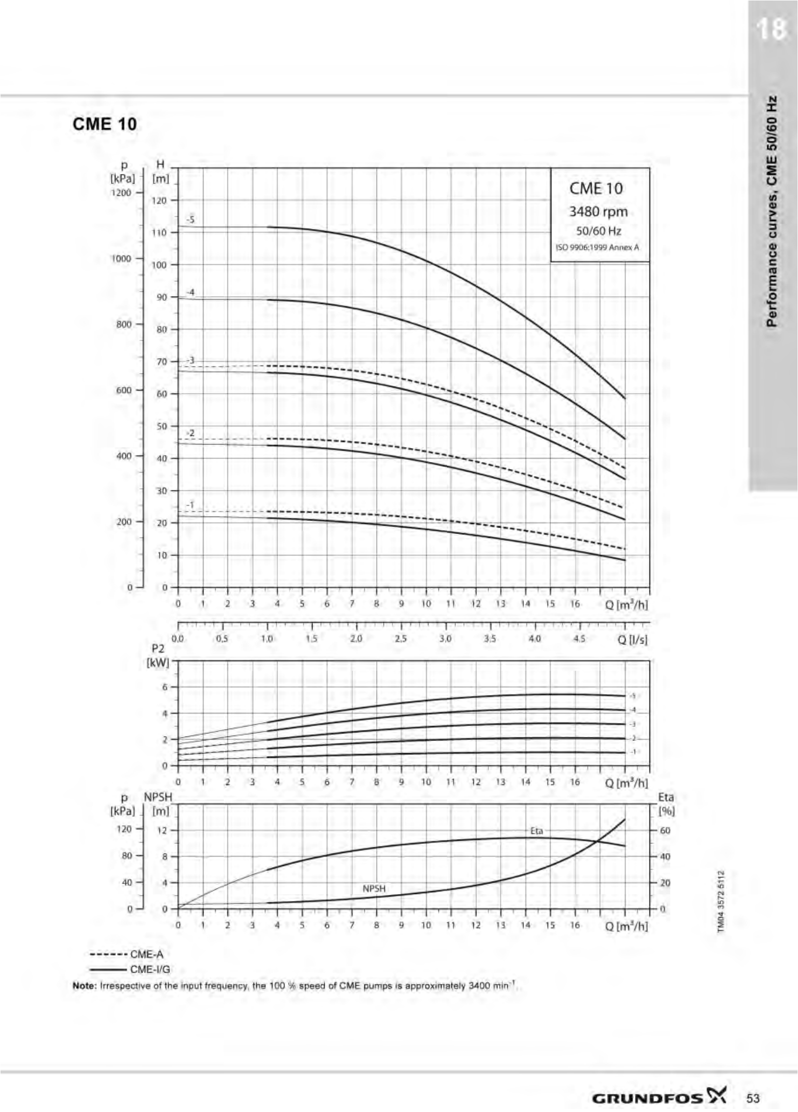


Figure 25: Pump performance curves from Grundfos Catalogue

3.10.3 Water glycol pump: pump3

The selected pump is Grundfoss CM 5-2 that can satisfy the requirements of mass flow and pressure drop for the circuit, as it works at fixed velocity the different values of mass flow will be reached modifying the bypass flow with the needle valve.

As can be seen from Figure 26 the value of $NPSH_A$ is bigger than the $NPSH_R$ in the worst case scenario: $NPSH_A$=13,5 m while $NPSH_R$ is smaller than 5 meters so the risk of cavitation is avoided.

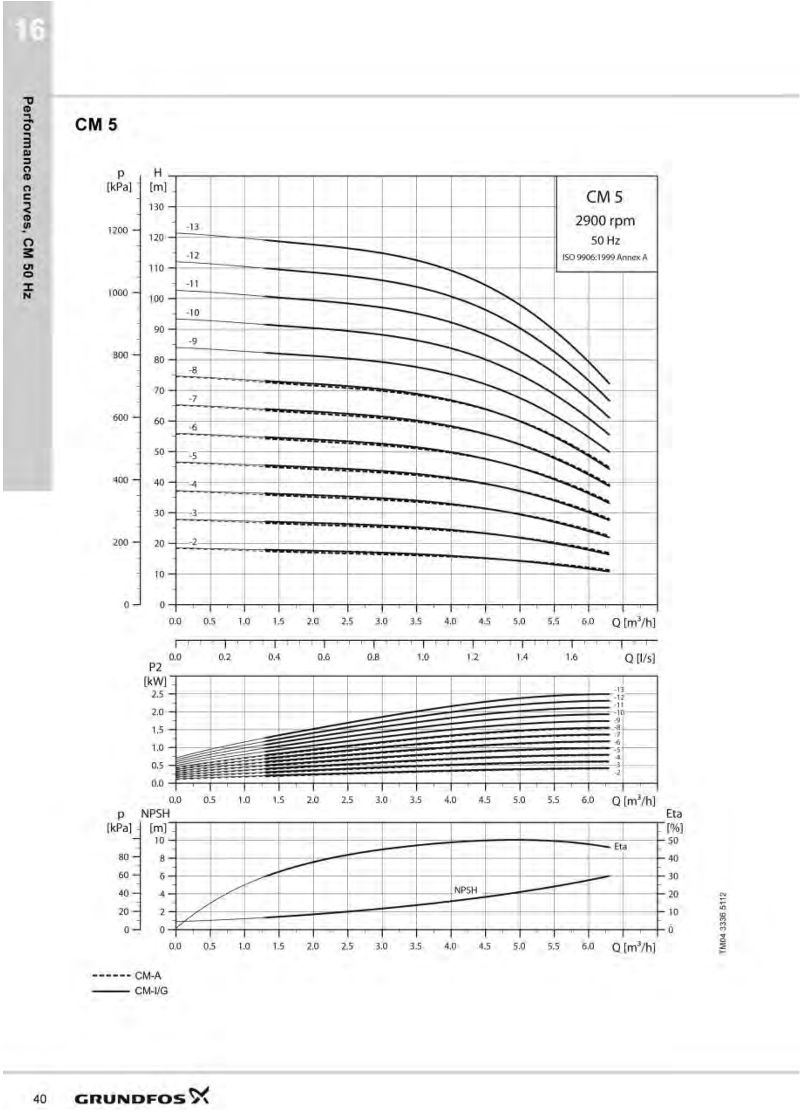


Figure 26: Pump performance curves from Grundfos Catalogue

3.11 Electric heat tank

The electric heat tank is necessary to warm up the water entering in Evaporator1 when its temperature is lower than the desired one, furthermore it is used to adjust the temperature with precision also when the recovery heat exchanger HX_1 can fulfil all the necessary heat exchange.

The size of the electric heat tank is related to the heat power that the electric resistances have to provide. As can be seed in Figure 22 the maximum electric power needed is near to 19 kW, however from the simulations we know that this value occurs only in one case that is W35-30W10-60 and in all the other working points the value is lower than 11 kW.

For that reason the decision to use only two electric resistances in the tank, one of 9 kW and the other of 4,5 kW; in that way it will be possible to test all the working points except for W35-30W10-60 and eventually in the future a new resistance of 4,5 kW can be added to investigate also in this point.

The bought resistances are the following models from the producer Salvador Escodia: ED210 4.5 kW and ED011 9 kW.

3.12 Water-glycol buffer tank

To avoid freezing risks in the heat exchanger HX_2 and to assure a reasonable heat exchange the outlet temperature of the tank has to be kept between tmin=-1°C and tmax=1°C.

Called Q_{WG} the thermal power exchanged through HX_2 and Q_c the thermal power exchanged through HX_3, it is clear that when compressor2 will be ON Qc will be bigger than Q_{WG} so the water-glycol temperature will decrease; in reverse when the compressor2 will be OFF Qc will be near to zero and the water-glycol temperature will raise up again.

As the compressor works at fixed velocity and its ON OFF control is made on the water-glycol temperature in the tank outlet, this swing between tmin and tmax will repeat over and over during the working of the installation.

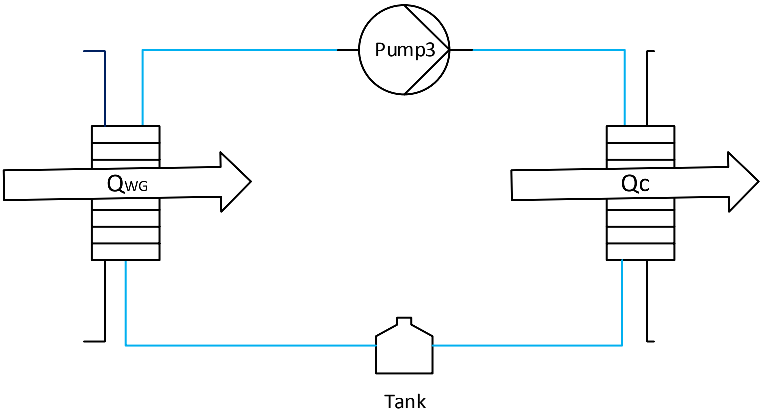


Figure 27: Water-glycol loop

To prevent a continuous turn on and switch off of the compressor, the value of tmin, tmax and the tank volume have been studied with the following differential equation in the time dt:

$$M_T * c_p * \frac{dT}{dt} = Q_{WG} - Q_C$$

Where M_T is the mass of water-glycol in the tank , c_p is the specific heat capacity of the water-glycol solution, T the temperature inside of the tank and Q_{WG} and Q_C the heat power transferred.

In this calculation the mass of the fluid into the pipes and into the other components has been neglected because of its low quantity in front of the mass in the tank.

The differential equation can be integrated in the time when the compressor is ON and when it is OFF.

- Compressor OFF:
  In that case the heat power exchanged through HX_3 (Q$_C$) is equal to zero so the differential equation became:

$$M_T * c_p * \frac{dT}{dt} = Q_{WG} - Q_C \xrightarrow{Q_C=0} \Delta T = \frac{Q_{WG}}{M_T * c_p} \Delta t$$

  Consequently the time when the compressor is OFF is:

$$t_{OFF} = \frac{M_T * c_p * (T_{Cmax} - T_{Cmin})}{Q_{WG}}$$

- Compressor ON
  In that case the heat power exchanged through HX_3 (Q$_C$) is not zero so the differential equation became:

$$M_T * c_p * \frac{dT}{dt} = Q_{WG} - Q_C \;\rightarrow\; \Delta T = \frac{Q_{WG} - Q_C}{M_T * c_p} \Delta t$$

  Consequently the time when the compressor is ON is:

$$t_{ON} = \frac{M_T * c_p * (T_{Cmin} - T_{Cmax})}{Q_{WG} - Q_C}$$

Then we can calculate the number of cycles per hour:

$$CPH = \frac{3600}{t_{OFF} + t_{ON}} = \frac{3600}{\frac{1}{Q_{WG}} + \frac{1}{Q_{WG} - Q_C}} * \frac{1}{M_T * c_p * (T_{Cmax} - T_{Cmin})}$$

To maintain the integrity of the compressor the number of cycles per hour must be limited so we impose a maximum value for CPH=10.

Introducing the previously calculated values of Q$_{WG}$ and Q$_C$ for the different operative points, the medium c_p of water glycol between Tcmin and Tcmax and the values of the latters (T_{Cmin}=-1ºC and T_{Cmax}=+1ºC) we can calculate the value of CPH in every test condition with the tank of 370 litres yet available in the laboratory.

In every test configuration it has been verified that the CPH value is lower than 10 so that 370 liters tank (Lapesa G3701-E4) fulfilling the requirements has been adopted for the loop.

After the installation that tank revealed to be broken as explained in paragraph 4.1, for this reason it has been replaced with a new tank of 500 litres, the choose of a bigger capacity is due only to stock availability.

3.13 Expansion tanks

The expansion tanks are situated in water and water-glycol loops to balance the fluid dilatations at the different temperatures.

The input data for the calculation are the volume of the liquid in each circuit, its density at the different temperatures and the minimum and maximum temperature that the fluid can reach.

The volume has been calculated knowing the diameter of the pipes in each circuit and its length, the latter has been estimated thanks to a 3D model of the disposition of the whole components and pipes in the lab; see Figure 29. Furthermore the volumes of the heat exchangers and of the tanks have been added to the volume computation.

For the temperatures the minimum and maximum value has to take into account the operative and laboratory conditions, for that reason Tmax for circuit 3 is 30 º C instead of the maximum operative temperature when the circuit is working that is near to 15º C.

The density values at different conditions of water and ethylene glycol-water (25/75 %) are inferred from [19].

For the calculation we took into account an ambient temperature during the circuit filling of 20ºC and a security coefficient equal to 65%.

The calculation procedure is reported in the following Table 17 and the size really bought is the commercial one immediately higher.

	Circuit1 H_2O	Circuit2 H_2O	Circuit3 H_2O+glycol25%
Pipes legth [m]	23	32	6,5
Pipes volume [m^3]	0,0221	0,0308	0,0063
Heat exchangers volume [m^3]	0,006	0,0105	0,0045
Tank volume [m^3]	0,01	0	0,37
Total volume [m^3]	0,0381	0,0413	0,3808
Density at 20ºC [kg/m^3]	998,2	998,2	1029,1
Mass (filled at 20 ºC) [kg]	38,0488	41,1977	391,8304
Mass [kg]	38,0488	41,1977	391,8304
Tmax	40	60	30
Tmin	5	10	-10
Density at tmax [kg/m^3]	992,2	980,6	1025
Density at tmin [kg/m^3]	1000	999,7	1036,5
Expansion volume [m^3]	0,0003	0,0008	0,0042
Security coefficient	1,65	1,65	1,65
Expansion tank volume [m^3]	0,0005	0,0013	0,0070
Expansion tank volume [liters]	0,5	1,3	7,0

Table 17: Calculations for the expansion tanks

3.14 Isolating material

The pipes, heat exchangers, pumps and the other components are isolated from the Laboratory ambient in order to reduce the heat losses (or the heat gains in case of low fluid temperature) and to prevent the possible condensation of the ambient humidity.

In particular in the refrigerant cycle (circuit 4) the isolating thickness is the most dependent on the part of the cycle, the adopted dimensioning procedure has been followed taking into account the expedients described in the Trane Company Clinic [15] .

The suction line (from the evaporator to the pump) is entirely isolated to minimize the capacity losses of the cycle and also to avoid condensation of air humidity since the suction line temperature is lower than the dew point temperature.

Contrary the discharge line (from the pump to the condenser) is isolated only to prevent injury to someone who may come in contact with the piping; in effect the pipe temperature is very hot at the discharge of the compressor and a heat loss through the ambient reduces the heat rejection load of the condenser improving its efficiency. For that reason only the discharge line at a few feet off the ground is isolated with a thin isolating.

Finally the liquid line (from the condenser to the evaporator) normally is at higher temperature in respect to the ambient one but anyway it has been isolated because in some working conditions this statement can be false and a consequent subcooling loss can occur.

The selected material to realize the insulation is elastomeric foam of different diameters in function of the pipe diameter and fluid temperature; the minimum isolating thickness has been determined for all the secondary circuits with the following the prescription in the manual [20] and reported in Table 18 and Table 19.

External diameter (mm)	Maximum fluid temperature (ºC)		
	40÷60	60÷100	100÷180
D≤35	25	25	30
35<D≤60	30	30	40
60<D≤90	30	30	40
90<D≤140	30	40	50
140<D	35	40	50

Table 18: Minimum isolating thickness (mm) for pipes with hot fluid

External diameter (mm)	Maximum fluid temperature (ºC)		
	-10÷0	0÷10	>10
D≤35	30	20	20
35<D≤60	40	30	20
60<D≤90	40	30	03
90<D≤140	50	40	30
140<D	50	40	30

Table 19: Minimum isolating thickness (mm) for pipes with cold fluid

4 Installation

In this chapter will be explained the procedures and the most relevant expedients followed during the circuit installation.

As in the UPV Thermal Laboratory the available space is constricted by other equipments concerning several projects, the NxtHPG installation has been set up dismantling an older one and reutilizing a part of the electrical cabinet.

One priority is to obtain an installation as compact as possible in order to limit the length of the pipes and consequently the pressure drop, the charge of fluids and the cost of the equipment. The position of the fan coil was bounded by the necessity to extract and throw out the required air flux, as it was not possible to install it in an outside restricted area, it had to be set up into the lab considering the necessity to build also an air conduct connecting the external ambient to the installation.

The only available place was above the garage door taking into account the limit due to the opening and closing movement, also the air conduct has been installed above the door reducing the encumbrance exploiting an unused zone of the laboratory.

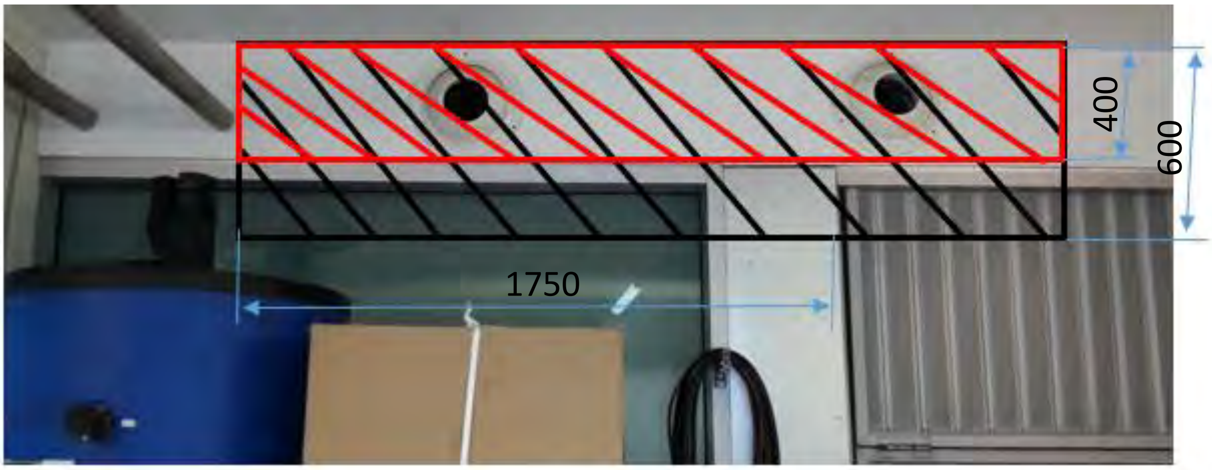


Figure 28: Available space in the Thermal Laboratory for condenser2 installation, in red can be seen the area of the hole and in black the condenser encumbrance

Under the fan coil the circuit 3 and 4 are installed in a space limited by the garage door and a workstation; from here two pipes passing near to the ceiling connect the HX_2 to the rest of circuit 2 and circuit 1, they are built in place of another circuit that had been dismantled for the occasion.

Near to circuit 2 a pre-existing electric panel will be modified to manage the electric part of the installation.

Finally the main circuit that arrived prebuilt by the manufacturer has been placed next to circuit 1.

The optimal positioning of the circuit components and the preferable way to connect them was studied using a 3D model of the Thermal Laboratory. I've carried out this task using the 3D modelling software SketchUp and taking into account the limited space available in the Thermal Laboratory and the constraints due to other installations and workspaces. The pipes connection designed aims to be as short as possible in order to minimize the pressure drops and the cost of the material but also permits a comfortable inspection of the installed devices.

The 3D model has been used not only as a reference guide during the components installation but also to estimate the length of the pipes and to make the first order.

In the following Figure 29 an image of the 3D model permits to visualize the entire installation.

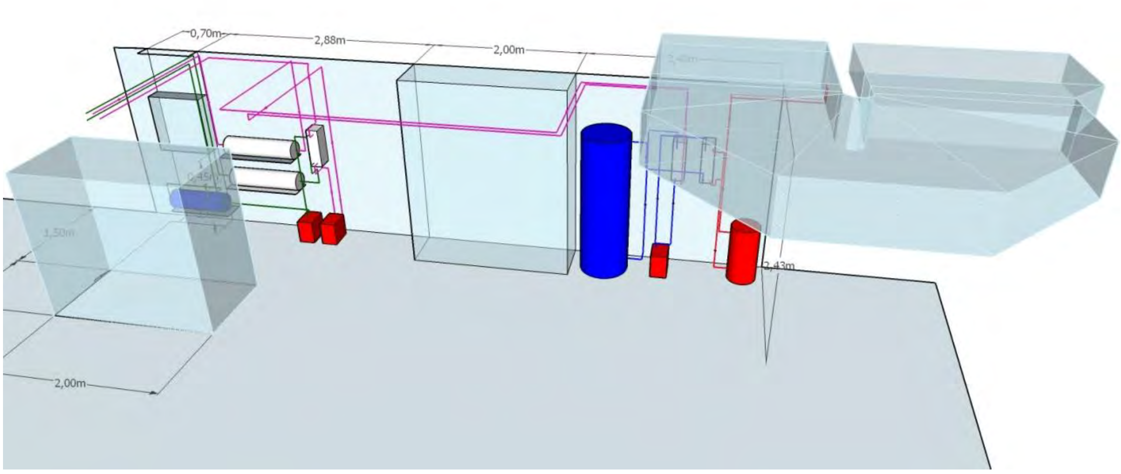


Figure 29: The entire installation in the Thermal Laboratory. In red the compressor and the pumps, in blue the tank in Circuit4 and the electric heat tank, in white the coriolis and the heat exchangers. In the upper part on the right the air conduct and the light blue box on the left represents the propane heat pump. Every circuit is of a different colour: Circuit1is green, Circuit2 is violet, Circuit3 is blue and Circuit4 is red.

Hereinafter will be explained in detail the procedure followed in the construction of the different circuits and systems that set up the installation, the order of the description is the one really followed in the laboratory.

4.1 Circuit 4

After the positioning of the fan coil the air conduct has been placed in order to extract the air from the external ambient and bring it to the fan coil; the conduct has been built with 25 mm panels of glass mineral wool covered in both the faces by aluminium foils (URSA AIR - P5858 Panel Al-Al), these insulating panels assembled in the desired configuration are resistant enough to be fixed above the ceiling with metal bars limiting the heat exchange with the internal laboratory ambient. The line followed in the construction was to maintain the passing area constant limiting the pressure drops compatibly with the available space.

Figure 30: Air conduct above the ceiling and circuit 4 positioning on the left.

The refrigerant circuit is made with different diameter pipes as the working fluid exchanges its state. In the following all the refrigerant copper pipes will be named with their external diameter size expressed in inch as in chapter 3.9; e.g. $1"\,{}^{3}\!/_{8}$ means an external diameter of one inch and three eighth.

In the pump suction side the external diameter is $1"\,{}^{3}\!/_{8}$, in the discharge side it is $1"\,{}^{1}\!/_{8}$ and in the liquid side is ${}^{5}\!/_{8}$; the different diameters fits the inlet and outlet of the compressor and in the liquid side it is the smallest with the purpose to limit the refrigerant charge.

In order to avoid some inconvenient some precautions has been taken:

- Two flexible sections are installed at the entrance and exit of the compressor muffling the vibrations that could damage the pipes, in the exit of the compressor the pipe is sloped in the direction of an elbow tube that can receive a certain quantity of oil preventing the contingent return of it to the compressor from the discharge side when the system is off; we want to avert this unlucky event because it could damage permanently the compressor. (see Figure 31)

- The horizontal section of the discharge line is pitched so that the eventual condensed refrigerant can drain to the condenser, this prevent from condensed refrigerant flowing back to the compressor when the system is off. For the same reason an elbow at the entering of the condenser permit to join an altitude higher than the top of it (see Figure 31).
  To avoid a possible stagnation of refrigerant vapour and its eventual presence at the entrance of the expansion valve, also the horizontal liquid line is sloped forcing the contingent vapour present in the pipe to return in the condenser.

- A syphon after the evaporator does not allow a free drain of refrigerant and oil from the evaporator to the compressor when the system is off, for the same reason the horizontal pipe before the compressor is sloped in the opposite direction to the latter. (see Figure 31)

Two sight glasses are installed; they are components that allow to determine the state of the refrigerant watching it straight through a glass or looking the colour of an indicator. The first is upstream the expansion device as close as possible to it permitting a direct control of the absence of liquid, as yet said; the second is before the suction of the compressor allowing to see the eventual presence of liquid drops.

Figure 31: From the left to the right, details of the flexible sections and elbow downstream the compressor; elbow at the entering of the condenser; syphon after the evaporator.

4.2 Circuit 3

In that circuit a water-glycol loop transfers the heat that has to be rejected to the ambient from circuit 2 to circuit 4; a pump of constant velocity assure the circulation of the fluid, the mass flow can be adjusted with a needle valve and two pressure indicators show the pressure difference between the suction and discharge line.

The installed expansion tank is of 18 litres, this value is much more bigger than the value calculated during the dimensioning (7 litres) because an unused device was available in the Laboratory.

As it can be seen in Figure 32 before the expansion tank a valve is installed, it goes against the prescript of RITE (Reglamento de Instalaciones Térmicas en los Edificios) that forbids to put a valve between the proper circuit and the above mentioned device.

That choice is made because in case it is necessary to disassemble the tank for normal maintenance, i.e. refill or change the charge of the expansion gas, with this expedient it will not be necessary to empty the whole circuit that contains more than 500 litres of water-glycol but only the 18 litres tank. As preventative measure to avoid the mistaken closure of valve during the normal working and the consequent non compensation of fluid dilatations, the lever valve is removed.

In the higher parts of the circuit, air purges are positioned in order to evacuate the air in the circuit during the filling; its presence could lead to problems in the pump operation, lower heat exchange coefficient in the heat exchangers, loudness during the working and corrosion of the iron components like the tank due to the oxygen presence (see Figure 32).

The previous 370 litres installed tank presented some problems due to the corrosion so in the following pages will be explained the chemical mechanism that leads to that phenomenon in order to better analyse what happened in the tank and to avoid it keeps happening.

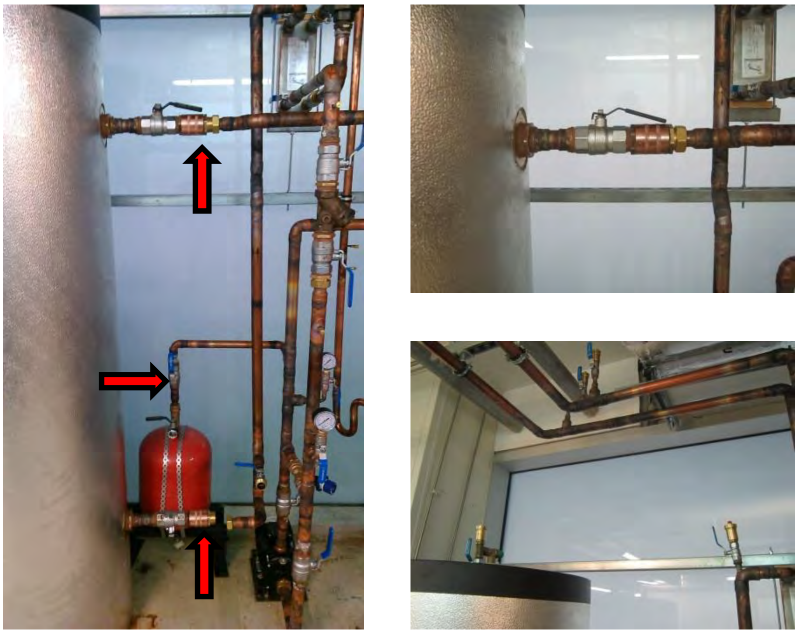


Figure 32: In counter-clockwise from the left, interception valve before the expansion tank and the two anti-electrolysis sleeves, detail of the upper anti-electrolysis sleeve and air purges.

4.3 Types of Corrosion

Corrosion is a chemical phenomenon that if not averted can damage a hydraulic circuit easily bringing it to a premature break.

The following part will briefly explain the principal corrosion formation ways and their causes focusing on the ones that mainly affect the hydraulic circuits and on the most common methods to avoid or delay the corrosion progress.

Particularly the explanation will focus on the possible corrosion that could happen in an iron tank connected to a copper circuit like the one present in the NxtHPG installation. The information and the images are obtained from [21].

4.3.1 General corrosion

Corrosion is the deterioration of a material due to interaction with its environment; a kind of corrosion is electrolysis that is the decomposition by electric current, it can occur when a metal is in contact with an electrolyte that is an electricity conducting fluid in which are present positive and negative ions that can move giving rise to an electric current, e.g. water is an electrolyte having a certain number of dissociated H^+ and OH^- ions.

When iron is in contact with water the superficial atoms can go into solution as Fe^{++} ions; the result is that the metal became negatively charged and simultaneously the electrolyte (water) became positively charged.

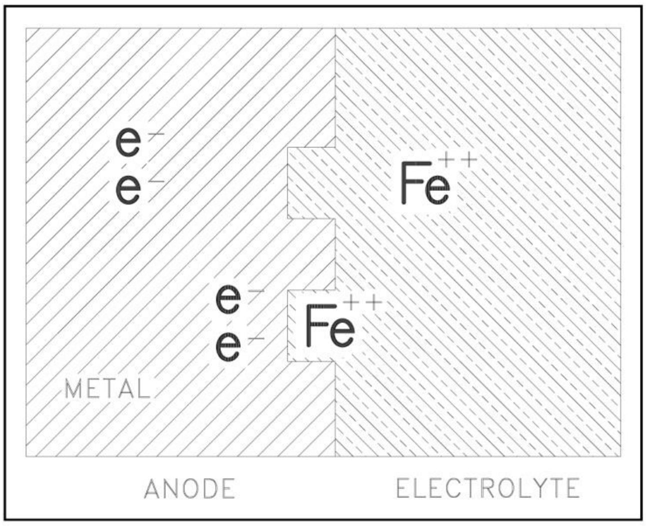


Figure 33: Formation of ferrous ions during the iron corrosion.

$$Fe \longrightarrow Fe^{2+}+2e^-$$

This process in which electrons are given up and positive metal ions are formed is called oxidation, when it takes place Fe^{++} ions goes into solution and a potential difference begins between iron and electrolyte so a electrolytic cell arises.

The move of ions from the metal to the electrolyte causes the metal deterioration that is dangerous for the integrity of a hydraulic circuit.

The iron can go into solution as Fe^{++} ions until the potential difference between the positively-charged solution and the negatively-charged metal stops the iron ions from leaving the surface.

An oxidation process cannot continue without a simultaneous reduction process (gain of electrons), for example a common reduction for water in contact with metal is the reduction of hydronium ions:

$$H_3O^+ + e^- \rightarrow \frac{1}{2} H_2 + H_2O$$

Normally the oxidation process stops when on the surface of the metal a concrete layer of metal oxide is produced forming a barrier that separates the metal surface to the electrolyte, the barrier is formed when the corrosion products (Fe^{2+}) are not soluble anymore in the electrolyte; as for the continuation of the reaction the reactant must diffuse through the oxide and this process if present is very slow, we can say that after a certain period this corrosion almost stops.

In conclusion the reactions that occur between iron and water are the following:

$$Fe \rightarrow Fe^{2+} + 2e^- \quad \text{(oxidation)}$$

$$H_3O^+ + e^- \rightarrow \frac{1}{2} H_2 + H_2O \quad \text{(reduction)} \qquad (4.3.1.1)$$

So the overall reaction is:

$$Fe + H_3O^+ \rightarrow Fe^{2+} + \frac{1}{2} H_2 + H_2O$$

The Fe^{2+} combines with OH^- ions at the metal surface first forming $Fe(OH)_2$ which later decomposes to FeO forming a layer on the metal surface :

$$Fe^{2+} + 2\,OH^- \rightarrow Fe(OH)_2 \rightarrow FeO + H_2O$$

The hydrogen atoms formed by the reaction of Equation (4.3.1.1) absorb on the metal surface and remain there until removed by one of the two following processes: combination of two hydrogen atoms to form molecular hydrogen, which is then released as a gas, or reaction with dissolved oxygen to form water; in the absence of oxygen the first process happens:

$$\frac{1}{2} H_2 + \frac{1}{2} H_2 \rightarrow H_2 \qquad (4.3.1.2)$$

The presence of hydrogen atoms in the metal surface blocks the sites in which the reaction (4.3.1.1) could happen and as the velocity at ambient temperature of reaction (4.3.1.2) is very slow compared to velocity of reaction in equation (4.3.1.1), the whole corrosion reaction is controlled by the recombination of hydrogen atoms.

The presence of oxygen in the water in contact with iron increases the corrosion rate because of the rapid reaction between the oxygen and the hydrogen absorbed on the oxide layer:

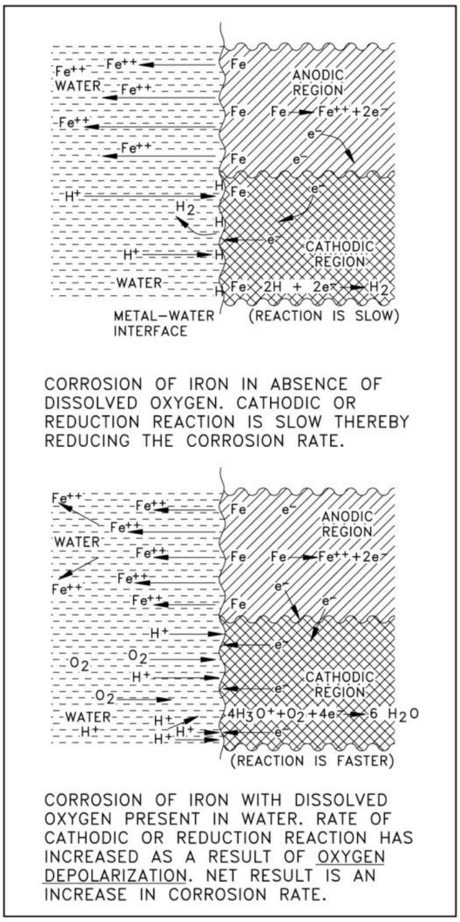
$$O_2+4H \rightarrow 2H_2O$$

Combining that equation with Equation (4.3.1.2) we obtain:

$$O_2+4H_3O^{+}+4e^{-}\rightarrow 6H_2O$$

Furthermore also the direct reaction between oxygen and Fe can accelerate the corrosion:

$$2Fe+O_2 \rightarrow 2FeO$$



CORROSION OF IRON IN ABSENCE OF DISSOLVED OXYGEN. CATHODIC OR REDUCTION REACTION IS SLOW THEREBY REDUCING THE CORROSION RATE.

CORROSION OF IRON WITH DISSOLVED OXYGEN PRESENT IN WATER. RATE OF CATHODIC OR REDUCTION REACTION HAS INCREASED AS A RESULT OF OXYGEN DEPOLARIZATION. NET RESULT IS AN INCREASE IN CORROSION RATE.

Figure 34: Representation of cathodic depolarization by oxygen

The condition and composition of the metal surface affect the corrosion rate; deposits, scale, or irregular surfaces create areas on the metal where corrosion can locally initiate and proceed at a faster rate than normal.

4.3.2 Galvanic Corrosion

Galvanic corrosion can happen when two different metals with different electrode potential are electrically connected and are both in contact with an electrolyte solution. The most noble metal between the two (in our case copper) will be the cathodic site of the reaction while the most active (iron) will act as the anode; an electron flow from the anode to the cathode caused by the different potential will corrode the less active metal so the anode metal dissolves into the electrolyte, and the deposit collects on the cathodic metal damaging the installation as shown in Figure 35.

The electrolyte provides a way for metallic ions migration from the anode to the cathode while the electrons move between the metals through a conducting path.

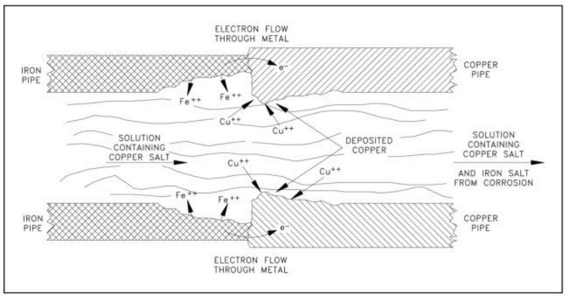


Figure 35: Galvanic corrosion for iron-copper interface

The corrosion seen in Figure 35 is explained by the following reaction:

$$Cu^{2+} + Fe \rightarrow Cu + Fe^{2+}$$

The velocity of the galvanic corrosion is correlated to the parameter i_{el} that takes into account the current through the junction and the ratio between the areas of the cathode Sc and the anode .

$$i_{el} = \frac{Sc}{Sa} * \frac{\Delta U}{R_{el} + R_{pa} + R_{pc}}$$

Where ΔU is the potential difference between the two metals, R_{el} the resistance of the electrolyte and R_{pc} and R_{pa} the bias resistance of the cathode and the anode.

In conclusion there are different variables that can affect the velocity of the galvanic corrosion, for example if the area of the anode is much bigger in comparison to that of the cathode, the cathode will not be able to provide enough current to sustain the corrosion of the anode.

For the formation of the galvanic corrosion some conditions are necessary:

- Different potential corrosion between two metals in the same system;
- Electric connection between the two metals;
- Connection of the two metals through an electrolyte.

It is sufficient to avoid one of these conditions to prevent this negative phenomenon; when two different metals has to be used in the same hydraulic circuit it is possible to interrupt the electric connection between the two: if they are not in electrical contact, no galvanic couple will occur, this could be realized for example connecting the two materials with a isolating plastic pipe spool.

Other methods could be used like the cathodic protection that consists in forcing a different tension in the two materials with an external voltage cancelling the potential difference between them, or isolating the metals from the electrolyte for example using a covering, but in the last case also a little hole in it could make vane the procedure.

Also the use of sacrificial anodes, e.g. zinc, could be another method to preserve the integrity of an iron component but usually that is not applied in plumbing because of the release of particles that could cause potential mechanical damage to circuit components like circulating pumps or heat exchangers.

As said the previous installed tank presented a problem after the installation, it had been used for some years in another installation and when we dismantled it we thought it could have been used in the NxtHPG one. After the installation we tested Circuit3 charging it with water at 3.5 bars; after some hours the pressure was slowly decreasing and we found a little leakage in the top of the tank.

Probably its presence was due to two main reasons: an air bubble in the top of the tank in the previous installation could have initiate the corrosion of the top of the iron tank; after the arrest of this corrosion pursuant to the formation of an oxide layer or to the consumption of the oxygen present in the circuit (also the previous circuit was a closed loop) the corrosion continued as galvanic corrosion (no anti-electrolysis sleeves were installed).

In the installation of the new tank we took some precautions to avoid the repeating of that phenomenon: we took care about empty the tank from any residual bubble of air, and to prevent galvanic corrosion two anti-electrolysis sleeves electrically isolate the iron tank from the copper circuit. (See Figure 32).

4.4 Circuits 1 and 2

These two loops are the source and the sink of heat for the heat pump prototype; in each circuit a variable velocity pump assures the circulation of the fluid at different flow rate, the mass flow can be adjusted with precision with a needle valve and two pressure indicators show the pressure difference between the suction and discharge line.

In both circuits the installed expansion tanks are of 5 litres; this value is much more bigger than the values calculated during the dimensioning, that are respectively 0,5 and 1,3 litres, because that is the minimum available size.

A security valve in every circuit is put in order to empty them in case of an extra pressure, they are calibrated at 6 bars and installed after the pump so if a block occurs and the pressure in the discharge line starts increasing, this component avoids the breakage of the pump.

As required from the constructor, the coriolis mass flow meters are installed under head, that is under the connected pipe height, to assure they are always filled with water and never crossed by bubbles of air in order to obtain a precise measurement.

Furthermore they have been placed in detached supports and connected to the rest of the circuit with flexible sections so as to prevent the vibration interference between the two coriolis and between the coriolis and the pipes connected to the pumps. Once again that preventative measure is done to obtain a precise measurement.

Finally, as in the water-glycol loop, air purges are installed in the upper parts of the circuit to evacuate the possible bubbles of air present within.

Figure 36: From the right, coriolis installation in detached supports and under head, in the figure could be seen also the flexible sections; detail of one flexible section.

5 Measuring instruments and data logging system

5.1 Temperature measurement

The measure of the temperatures in the different points of the installation is fundamental to characterize the propane heat pump and also to control the working conditions of the secondary circuits during the functioning.

We want a meticulous data survey to monitor the working conditions of the propane heat pump while the measurements of the secondary circuits can be less precise as they have only to assure a correct working of the previous machine.

As these two main necessities require different levels of accuracy, distinct measurement devices have been set up in the installation: thermocouples connected to an isothermal block measure the temperature in the propane heat pump points and at inlet and outlet of the heat exchangers directly connected to this cycle; thermocouples with a lower precision connection are put in the other heat exchangers inlet and outlet of the secondary circuits.

In the following will be explained the working of the temperatures transducers.

5.1.1 Thermocouple

When two wires composed by dissimilar metals are joined at both ends and the two junctions are at different temperatures, continuous current flows in the circuit; if the circuit is broken, the open circuit voltage is proportional to the temperature difference and function of the two metals.

$$Vs = \propto (T1 - T2)$$

Where Vs is the Seebeck voltage, $\propto$ the Seebeck coefficient and $T1$and $T2$ the temperatiure of the two junctions, it has to be taken into account that the Seebeck coefficient could be considered constant only for small temperature intervals.

However connecting a voltmeter to the open circuit creates other two junctions; for example in our case we used type T thermocouples that are made of copper and constantan. When a voltmeter is connected its copper clamps are in contact with the copper and constantan wires and so two more junctions are created (J3 and J2 as can be seen in Figure 37). As the first one is copper to copper it does not create a thermal voltage but it is not possible to say the same for J2.

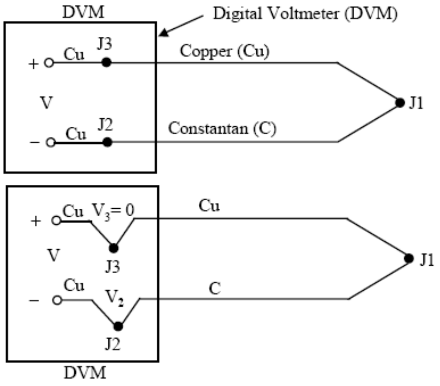


Figure 37: Connection of the type T thermocouple with a digital voltmeter

In conclusion the measured tension V is the difference between the voltage V1 in the junction J1 and the voltage V2 in the junction J2 which in turn are related to the relative temperatures T1 and T2.

It is possible to know the temperature in junction 1, knowing the measured tension V and the temperature in junction 2; from now the latter will be called also reference junction because its temperature is directly estimated with an external device.

To achieve this it is created an isothermal bath that is a block of high thermal conductive material (copper) surrounded by high isolating material, in that way the isothermal block temperature is all at the same value eliminating stratification effects, furthermore it slowly changes with the ambient temperature thanks to the high thermal inertia.

Putting the reference junction of the installed thermocouples into the perforated copper block, and filling the holes with conductive paste, it is possible to maintain all the junctions of the different thermocouples at the same temperature (see Figure 38).

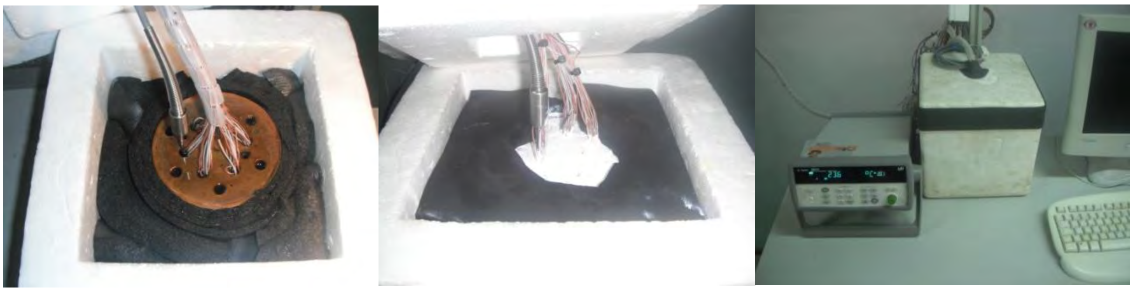


Figure 38: From the left, isothermal copper block into an isolating container and with the connected junctions, filling with conductive paste, final positioning in a close and isolated box

A resistance temperature detector (RTD) permits to calculate the temperature value of the isothermal block determining the resistance of a platinum wire, which is function of the temperature. The RTD used in the installation is a Pt100, class 1/10; from the constructor catalogue it has a precision of ±0.03ºC at 0ºC.

The utilisation of a RTD, that is a more expensive device compared to a thermocouple, is justified because we want to use its measurement for all the thermocouples connected to the isothermal block and so a low precision on its temperature measurement will wreak a lower precision on all the correlate measurements.

Contemporary we do not want to use RTDs for all the temperature measurements because of their cost but also because of their slower response to temperature changes. The latter property is compatible to the big thermal inertia of the isothermal block because the RTD is a good device for measuring a slowly changing temperature.

The other junction of the thermocouple, positioned in the point in which we want to know the temperature, is put into a thin copper cylinder (thermowell) which is directly in contact with the internal fluid in the pipes in order to obtain a better measurement.

Once known the reference temperature Tref of the isothermal block with the RTD, it is possible to obtain the reference voltage associated to that junction: Vref; for this we used an equation suggested by [22] where the units are mV and ºC.

$$Vref = c0 + c1 * Tref + c2 * Tref^2 + \cdots + c8 * Tref^8$$

With the relative cn coefficient from [22] in the range 0÷400ºC:

cn	value
c0	0
c1	3.874 810 636 4 * 10^-2
c2	3.329 222 788 0 * 10^-5
c3	2.061 824 340 4 * 10^-7
c4	-2.188 225 684 6 * 10^-9
c5	1.099 688 092 8 * 10^-11
c6	-3.081 575 877 2 * 10^-14
c7	4.547 913 529 0 * 10^-17
c8	-2.751 290 167 3 * 10^-20

Finally the correlation used to obtain the temperature value of the measuring point is:

$$T1 = a0 + a1 * (V + Vref) + a2 * (V + Vref)^2 + \cdots + a7 * (V + Vref)^7$$

With the relative an coefficient from [22]:

In the range -200÷0ºC:

an	value
a0	0
a1	2.594 919 2 * 10^1
a2	-2.131 696 7 * 10^-1
a3	7.901 869 2 * 10^-1
a4	4.252 777 7 * 10^-1
a5	1.330 447 3 * 10^-1
a6	2.024 144 6 * 10^-2

a7	1.266 817 1 * 10-3

And in the range 0÷400ºC :

an	value
a0	0
a1	2.592 800 * 10^1
a2	-7.602 961 * 10^-1
a3	4.637 791 * 10^ -2
a4	-2.165 394 * 10^-3
a5	6.048 144 * 10^-5
a6	-7.293 422 * 10^-7
a7	0

The voltmeter is in the data logger so the two copper wires connect the data logger port to the two junctions J1 and J2.

In this way, solving with a computer the described correlations, it is possible to know $Vref$ and finally to obtain $T1$ with accuracy near to ±0.03 K [22].

For the points of the secondary circuits where we do not need a so high accuracy, the data logger port is directly connected with the copper wire and the constantan wire. In that way the reference junction J2 is located in one of the data logger ports and not in the isothermal block. As it is not possible to know exactly the temperatures of all the different ports, $Tref$ is estimated by a thermistor put in the case of the data logger.

That device measures the cage temperature that could be slightly different from the temperature in the junction J2, i.e. in the port, and furthermore the internal correlation of the data logger for the calculation of $Vref$ and $T1$ are simplified in respect of the ones earlier described.

For these reasons the temperature measure with the thermocouple has a lower precision with this configuration and the accuracy we expect is ±0.5 K.

5.2 Pressure measurement

The absolute pressure is measured in the propane cycle before the condenser and after the evaporator in order to know respectively the condensation and the evaporation pressure.

As we want to know exactly these values, two high accuracy pressure devices will be used: for the high pressure side a Rosemount model 2088 (range 0/50 bar) and for the low pressure side a Rosemount model 3051 (range 0/20 bar). Both the devices have a precision of 0,065%.

All of the other pressure meters are Emerson model PT5-18M for the low pressure line (range 0/18 bar) and Emerson PT5-50M for the high pressure line (range 0/50 bar) with a lower precision.

Three differential pressure transducers are installed in the water side of every heat exchanger of the propane cycle in order to estimate the pressure drop throw them. The devices adopted are Rosemount 1151SMART with range0-0.373 bar.

5.3 Mass flow measurement

The measure of the mass flow in circuit one and two is obtained with two coriolis mass flow meters; knowing these values is necessary to evaluate the entity of the heat exchanged through the evaporator, the condenser and the subcooler of the main circuit and so to test the working of the propane heat pump.

As yet said, in circuit two is put also a magnetic mass flow meter for the reasons explained in section 2.3.

Both the coriolis are of Siemens model MASS 2100, with a signal converter of Siemens model MASS 6000; the magnetic mass flow meter is of Siemens model MAG 5100W, with a signal converter of Siemens model MAG 6000.

The selection is made consulting the producer catalogue taking into account the maximum and minimum mass flow and the relative maximum error (and pressure drop for the coriolis).

The minimum and maximum mass flows are calculated with IMST-ART and can be seen in Table 6, they are respectively:

- For circuit one:

  case W20-10 W55-60, 0.545 kg/s that is 1962 kg/h or 1.96 m^3/h

  case W35-30 W55-60, 2.933 kg/s that is 10559 kg/h or 10.56 m^3/h

- For circuit two:

  case W20-10 W10-60, 0.2098 kg/s that is 755 kg/h or 0.76 m^3/h

  case W35-30 W55-60, 2.552 kg/s that is 9187 kg/h or 9.19 m^3/h

The test values can be seen in Appendix, paragraph 7.2.

5.4 Liquid level sensor

A level sensor Danfoss AKS 4100 is used to know the liquid level in the liquid receiver of the propane cycle. The utilization of this device will be fundamental to study the working of the heat pump and to perform the subcooling control in configuration B (see paragraph 2.1).

The output of this instrument is a current signal in the range 4-20 mA.

5.5 Data logger

The acquisition of the data is possible thanks to data loggers model Aglilent-HP 34972A each one with 3 slots HP 34901A, every slot has 20 input channels for the voltage measures and 2 for the current measures.

Figure 39: In the right the front and the back of a data logger used in the installation, in the right one of the three slots present in every data logger

In our case the channels for the voltage measures will be used for the temperature measures with the thermocouples while the current channels for the mass flows, the pressure sensors and the liquid receiver. In total we need two data loggers because the current measures are eleven: 3 for the mass flow meters and 8 for the pressure sensors.

To communicate with the data loggers it is used the software HP BenchLink Data Logger that permits to analyse in real time the scanned values.

The inputs from all the measurement devices are in the range 0÷10V or 4÷20mA.

5.6 PID controllers

These controllers are not connected to the data logging system but directly to the device they are commanding.

The two three way valve that vary the mass flow through the bypass in circuit one and two, are governed by motorized valves that are connected to PID regulators. These devices, knowing the temperature measured downstream with a RTD, set the mass flow through the valves in order to reach the set point value.

Also the three way valve before the subcooler will have a PID control but for the investigation work we suppose to use it only in totally opened/closed position with a manual control; in case of further future necessities it will be possible also to control its opening relating it to a pressure value (for example to limit the pressure drop in the SB).

The expansion valves in circuit 4 and in the propane one are regulated by a PID that evaluates the temperature and pressure after the evaporator supervising the exit point of the fluid in the evaporator.

The expansion valve before the liquid receiver (that will be used in configuration B, see paragraph 2.1) will be governed by a PID but the preferable value to be used as input will be determined during the investigation period; the three candidates are the propane temperature before the expansion valve, the water temperature before the heat exchanger C1 or the liquid level indicator in the liquid receiver.

Finally another PID regulates the heat power transferred from the electric resistances to the water in circuit 1 electric heat tank, in that case the control is done measuring the water temperature after the tank.

6 Conclusions

This document presents a study of the optimal design of a test rig for the propane heat pump which is the focus of the investigation that will be carried out in Universitat Politècnica de València for the NxtHPG project.

As we contemporary need to supply and to absorb heat power from the machine, a recovery heat exchanger furnishes to the heat source part of the energy needed extracting it from the heat sink.

In that way two important objectives are reached. First of all it is avoided the necessity to directly furnish, e.g. with electric resistances or a boiler, all the heat power needed by the machine in the heat source side. Second we reduce the heat power that has to be rejected with a refrigerant cycle to the external air.

These expedients lead to a double side economic saving: from a point of view they reduce the electric (or natural gas) consumption that should have been used to feed the electric resistances (or boiler) and the higher capacity compressor and auxiliaries; from the other point they permit to buy lower capacity components, above all a smaller compressor in refrigerant cycle.

The choice of install three different loops to simulate the hot water consumption by a consumer in the condenser side, is due to the selection of a fixed velocity compressor in the refrigerant cycle. In fact, to assure the desired return water temperature to the propane heat pump and decouple it from the starts and stops of the fixed velocity compressor, a water glycol loop with a 500 litres tank has been installed.

Its task is useful to soften the sudden variation of heat exchange pursuant to a start or stop of the compressor, assuring that the fluid exchanging with the water in the condenser side stays in the desired range ±1ºC. This guarantees a certain stability to the return water temperature in the condenser side avoiding the risk of water freezing. The same ability could not have been reached simply using a variable velocity compressor in the refrigerant cycle.

During the thesis work in the UPV Thermal Laboratory I participated in the last part of the dimensioning phase and in the selection of the components and contemporary in the construction and assembly of the four secondary circuits. That allowed me to improve both the theoretical knowledge and the practical one directly in the field.

The work on the UPV Thermal Lab will hereafter continue with the put into operation of the whole installation and the test of the propane heat pump in the different working conditions and configurations. The results will be discussed and compared with the other project partners in order to study the feasibility of that innovative propane heat pump and to suggest improvement for the construction of a second prototype.

7 Appendix

7.1 IMST-ART summary results

N.B.: The two simulations are made using the same components also if the compressor of SH 240 can be used only with R410A, this is only to justify the bigger heating capacity in the condenser using R410A instead of R407C; in the results are present only the last six of the sixty-six working point analysed.

7.1.1 Refrigerant cycle with R407C

Input Data

Refrigerant: R407C
Compressor: SH 240
Evaporator: EvapB25T60
Condenser: Exchanger
Expansion Device: TXV 10 °C
Tubing: Simplified
Condenser Outlet: Subcooling 5 °C
Parametrics Studies: YES

Parametric Studies

	Units	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Evaporator Sec. Fluid Inlet Flow Rate (*)	m³/h	1.66	2.03	1.79	2.22	1.94	2.44

Summary

	Units	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Condensation Temp.	°C	36.244	36.739	36.44	36.929	36.636	37.107
Evaporation Temp.	°C	-6.6187	-5.2884	-6.074	-4.7906	-5.5585	-4.3234
Condensation Press.	kPa	1594.2	1614.2	1601.9	1622	1609.9	1629.3
Evaporation Press.	kPa	363.34	381.6	370.49	388.69	377.75	395.36
Condenser SubCooling	K	5	5	5	5	5	5
Total SubCooling	K	5	5	5	5	5	5
Evaporator Superheat	K	10	10	10	10	10	10
Total SuperHeat	K	10	10	10	10	10	10
COP	--	2.89	3.0026	2.9366	3.0449	2.9799	3.085
COP Auxiliary Included	--	2.5522	2.6553	2.5946	2.6941	2.6344	2.7307
EER	--	9.8699	10.254	10.029	10.399	10.177	10.536
EER Auxiliary Included	--	8.7162	9.0683	8.8611	9.2007	8.9969	9.3257
Total Carnot Eff.	%	46.477	47.111	46.747	47.336	46.987	47.545
Total Carnot Eff. Auxiliary Included	%	41.044	41.662	41.302	41.883	41.539	42.084
COP(HPA)	--	3.84	3.9526	3.8866	3.9949	3.9299	4.035
COP(HPA) Auxiliary Included	--	3.3911	3.4954	3.434	3.5346	3.4743	3.5716
EER(HPA)	--	13.114	13.499	13.274	13.643	13.421	13.78
EER(HPA) Auxiliary Included	--	11.581	11.937	11.728	12.071	11.865	12.197
TCE(HPA)	%	53.199	53.605	53.373	53.749	53.526	53.882
TCE(HPA) Auxiliary Included	%	46.98	47.405	47.157	47.557	47.321	47.693
Cooling Capacity	kW	26.017	27.405	26.563	27.942	27.113	28.446
Heating Capacity	kW	34.569	36.075	35.156	36.66	35.757	37.206
Mass Flowrate	kg/s	0.15403	0.16228	0.15726	0.16549	0.16054	0.1685
Comp. Disch. Temp.	°C	81.375	80.411	80.953	80.087	80.593	79.793
Comp. Power Input	kW	9.0024	9.127	9.0453	9.1768	9.0988	9.2208
Global Power Input	kW	10.194	10.321	10.238	10.372	10.292	10.417
Isentropic Eff.	%	69.683	70.642	70.068	70.999	70.445	71.327
Compressor Eff.	%	66.199	67.11	66.564	67.449	66.923	67.761
Volumetric Eff.	%	94.348	94.823	94.541	94.994	94.727	95.15
Refrigerant	--	R407C	R407C	R407C	R407C	R407C	R407C

7.1.2 Refrigerant cycle with R410A

Input Data

Refrigerant:	R410A
Compressor:	SH 240
Evaporator:	EvapB25T60
Condenser:	Exchanger
Expansion Device:	TXV 10 °C
Tubing:	Simplified
Condenser Outlet:	Subcooling 5 °C
Parametrics Studies:	YES

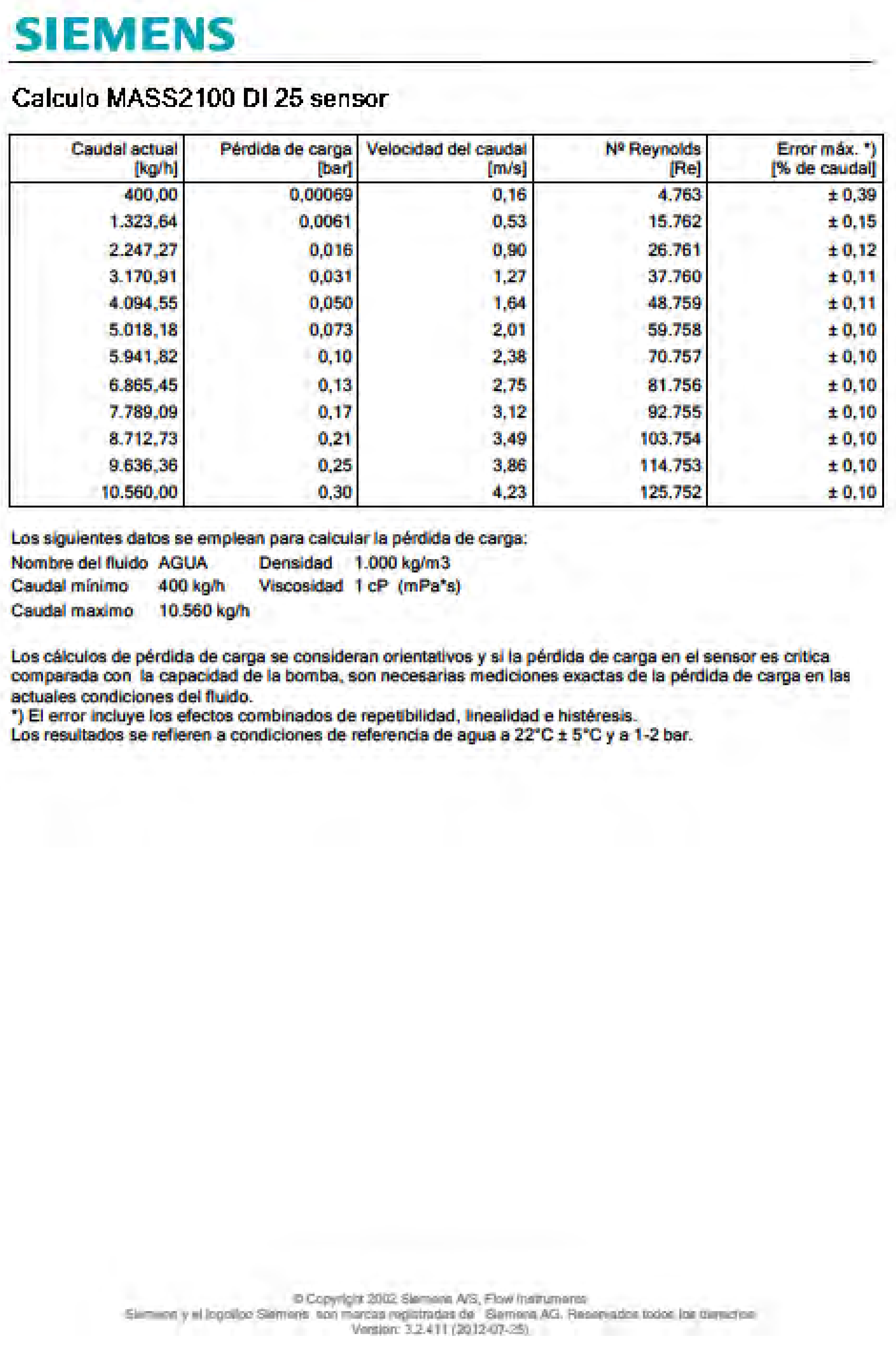
Parametric Studies

	Units	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Evaporator Sec. Fluid Inlet Flow Rate (*)	**m³/h**	1.66	2.03	1.79	2.22	1.94	2.44

Summary

	Units	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6
Condensation Temp.	**°C**	40.411	41.247	40.728	41.607	41.061	41.962
Evaporation Temp.	**°C**	-12.923	-11	-12.19	-10.225	-11.414	-9.444
Condensation Press.	**kPa**	2451.1	2500.4	2469.8	2521.7	2489.4	2543.6
Evaporation Press.	**kPa**	517.53	553.32	531.01	568.72	545.29	584.27
Condenser SubCooling	**K**	5	5	5	5	5	5
Total SubCooling	**K**	5	5	5	5	5	5
Evaporator Superheat	**K**	10	10	10	10	9.9999	10
Total SuperHeat	**K**	10	10	10	10	9.9999	10
COP	**--**	2.3166	2.4267	2.3579	2.4714	2.4026	2.5167
COP Auxiliary Included	**--**	2.1342	2.2391	2.1736	2.2818	2.2161	2.3249
EER	**--**	7.9115	8.2875	8.0526	8.4402	8.2053	8.595
EER Auxiliary Included	**--**	7.2888	7.6468	7.4233	7.7926	7.5684	7.9399
Total Carnot Eff.	**%**	47.479	48.364	47.814	48.721	48.169	49.06
Total Carnot Eff. Auxiliary Included	**%**	43.741	44.625	44.078	44.982	44.43	45.321
COP(HPA)	**--**	3.2666	3.3767	3.3079	3.4214	3.3526	3.4667
COP(HPA) Auxiliary Included	**--**	3.0095	3.1156	3.0494	3.1589	3.0924	3.2025
EER(HPA)	**--**	11.156	11.532	11.297	11.685	11.45	11.839
EER(HPA) Auxiliary Included	**--**	10.278	10.64	10.414	10.788	10.561	10.937
TCE(HPA)	**%**	55.562	56.114	55.769	56.342	55.99	56.554
TCE(HPA) Auxiliary Included	**%**	51.188	51.776	51.411	52.019	51.644	52.244
Cooling Capacity	**kW**	32.475	34.771	33.342	35.748	34.258	36.731
Heating Capacity	**kW**	45.792	48.383	46.775	49.49	47.805	50.596
Mass Flowrate	**kg/s**	0.19215	0.20651	0.19756	0.21271	0.20329	0.21895
Comp. Disch. Temp.	**°C**	94.841	93.163	94.173	92.555	93.509	91.989
Comp. Power Input	**kW**	14.018	14.329	14.14	14.465	14.259	14.595
Global Power Input	**kW**	15.216	15.529	15.339	15.667	15.459	15.799
Isentropic Eff.	**%**	67.754	68.969	68.22	69.467	68.703	69.949
Compressor Eff.	**%**	64.366	65.521	64.809	65.994	65.268	66.452
Volumetric Eff.	**%**	93.324	93.981	93.581	94.238	93.841	94.482
Refrigerant	**--**	R410A	R410A	R410A	R410A	R410A	R410A

7.2 Mass flow meters data



SIEMENS

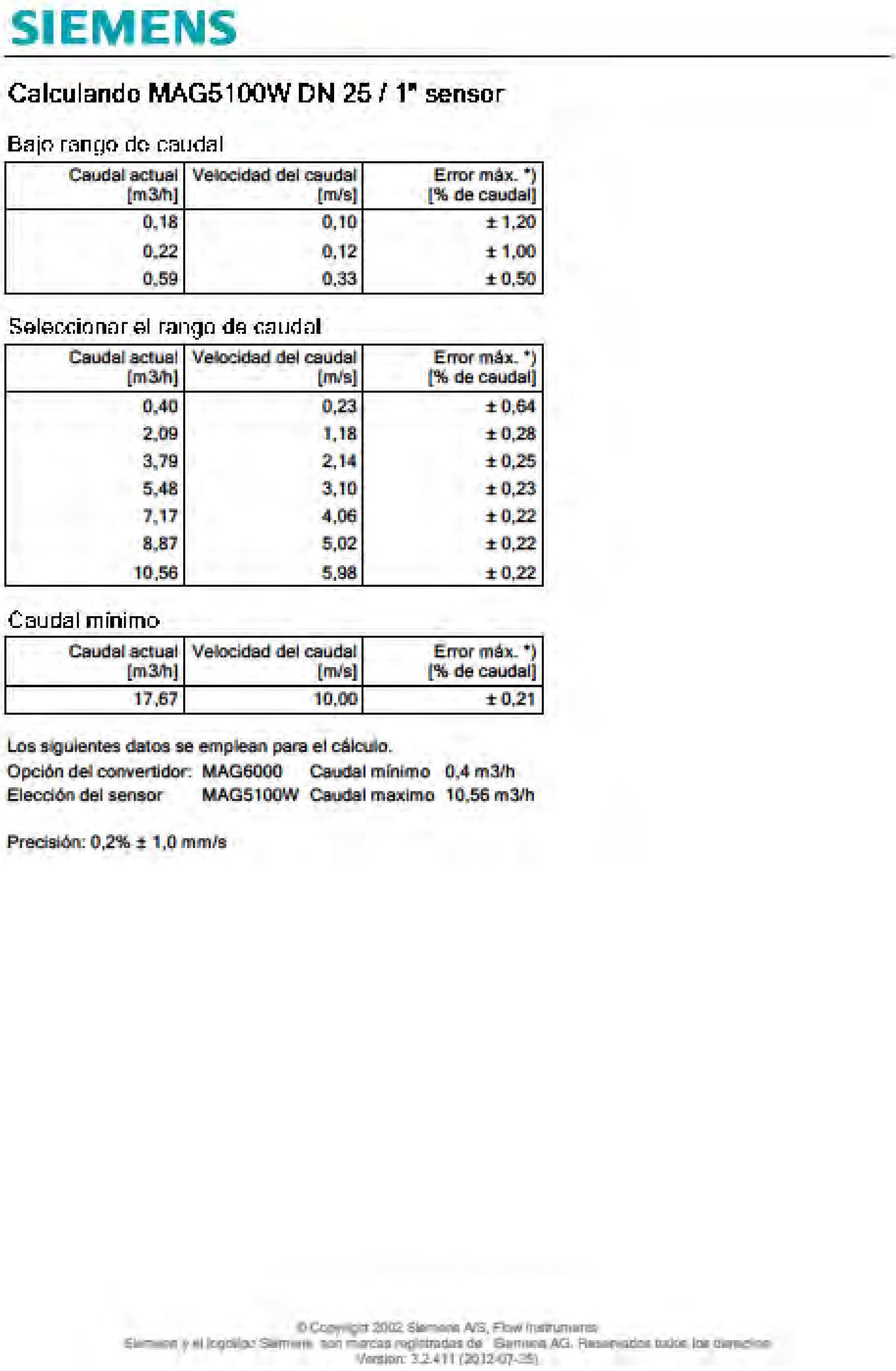
Calculo MASS2100 DI 25 sensor

Caudal actual [kg/h]	Pérdida de carga [bar]	Velocidad del caudal [m/s]	Nº Reynolds [Re]	Error máx. *) [% de caudal]
400,00	0,00069	0,16	4.763	± 0,39
1.323,64	0,0061	0,53	15.762	± 0,15
2.247,27	0,016	0,90	26.761	± 0,12
3.170,91	0,031	1,27	37.760	± 0,11
4.094,55	0,050	1,64	48.759	± 0,11
5.018,18	0,073	2,01	59.758	± 0,10
5.941,82	0,10	2,38	70.757	± 0,10
6.865,45	0,13	2,75	81.756	± 0,10
7.789,09	0,17	3,12	92.755	± 0,10
8.712,73	0,21	3,49	103.754	± 0,10
9.636,36	0,25	3,86	114.753	± 0,10
10.560,00	0,30	4,23	125.752	± 0,10

Los siguientes datos se emplean para calcular la pérdida de carga:

Nombre del fluido	AGUA	Densidad	1.000 kg/m3
Caudal mínimo	400 kg/h	Viscosidad	1 cP (mPa*s)
Caudal maximo	10.560 kg/h		

Los cálculos de pérdida de carga se consideran orientativos y si la pérdida de carga en el sensor es critica comparada con la capacidad de la bomba, son necesarias mediciones exactas de la pérdida de carga en las actuales condiciones del fluido.
*) El error incluye los efectos combinados de repetibilidad, linealidad e histéresis.
Los resultados se refieren a condiciones de referencia de agua a 22°C ± 5°C y a 1-2 bar.



SIEMENS

Calculando MAG5100W DN 25 / 1" sensor

Bajo rango de caudal

Caudal actual [m3/h]	Velocidad del caudal [m/s]	Error máx. *) [% de caudal]
0,18	0,10	± 1,20
0,22	0,12	± 1,00
0,59	0,33	± 0,50

Seleccionar el rango de caudal

Caudal actual [m3/h]	Velocidad del caudal [m/s]	Error máx. *) [% de caudal]
0,40	0,23	± 0,64
2,09	1,18	± 0,28
3,79	2,14	± 0,25
5,48	3,10	± 0,23
7,17	4,06	± 0,22
8,87	5,02	± 0,22
10,56	5,98	± 0,22

Caudal mínimo

Caudal actual [m3/h]	Velocidad del caudal [m/s]	Error máx. *) [% de caudal]
17,67	10,00	± 0,21

Los siguientes datos se emplean para el cálculo.

Opción del convertidor:	MAG6000	Caudal mínimo	0,4 m3/h
Elección del sensor	MAG5100W	Caudal maximo	10,56 m3/h

Precisión: 0,2% ± 1,0 mm/s

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