UNIVERSITY
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DEPARTMENT OF BUILDING ENERGY AND ENVIRONMENT ENGINEERING

EXPLOTATION SYSTEM OF THE EXTERNAL COLD FOR REFRIGERATED AREAS APPLIED IN WILLYS SUPERMARKET

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MASTER PROGRAM IN ENERGY SYSTEM
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PREFACE

In no way would have been possible to carry out this project, from the most obvious start to finish, without the help of many collaborators and friends, at who from the sincerity we would like to express our deepest gratitude.

Thanks to our supervisor, Roland, for having always very useful comments to our questions and problems, and of course, its fearsome but valuables corrections. He always had the voice of experience.

Thanks too for our examiner, Taghi, who besides being a judge was also part of this work, providing views, solutions and advices.

To Willy’s supermarket staff, for outstanding heroism withstanding our presence and requests with the utmost of patience.

To the master's degree teachers, who indirectly were involved from their subjects, they gave us the scientific basis for carrying out this project.

And finally, how to forget our friends, they never denied an answer or good advice in the moments that we needed them.

And so many others forgotten, each and every one of them thanks.

1. INTRODUCTION

Nowadays it can be found multitude of processes with a cold necessity. From the home fridge to assembly line on the automobilist industry, food storage areas, conditioned air, so on.

All these process have no more option that getting their cold necessities from the one way available on the market to generate cold, the refrigerating cycle.

On the other hand much people on the world lives in countries where the most of the year the weather has very low temperatures (A good example is Sweden, specifically Gävle). Hence it can be thought to develop a system to exploit the external cold totally free to try to decrease the electric energy used on this kind of processes.

2. OBJECTIVE AND LIMITATIONS

On the present project it is going to be designed an innovator and new system to takes external free cold as a complementary purpose for the standard system to supply cold.

The main objective is to design a real system that could be installed with guaranty of work. This engine much be the most efficient possible to save the most of electric energy cost.

How the utilization of this system is totally linked to the external weather it will be studied also the different work conditions as well as the application ranges and the total cost saved when the system is working on parallel with other standard system.

An important aspect that is being studied is the particular and specifies requirements of emplacement and meteorological conditions where the system will be placed, that is due to the kind of the internal cycle used for the system to take the cold is totally different of the others cooling machines.

When the prototype designed is finished, it will make an example of application on Willys supermarket in order to quantify the total electricity cost is possible to save.

3. DESCRIPTION

Before presenting the final structure of the proposed system to develop it is important to know all the technology available in the market to produce cold, and the processes to be followed to achieve this. (Cooling cycle)

3.1 Refrigeration cycle

The refrigeration cycle is a cycle rather complex if it is compared with the system that is going to be studied in this project.

In order to perform the complete cycle requires a system in which components must perform a particular function; we are talking about the evaporator, compressor, condenser and expansion valve.

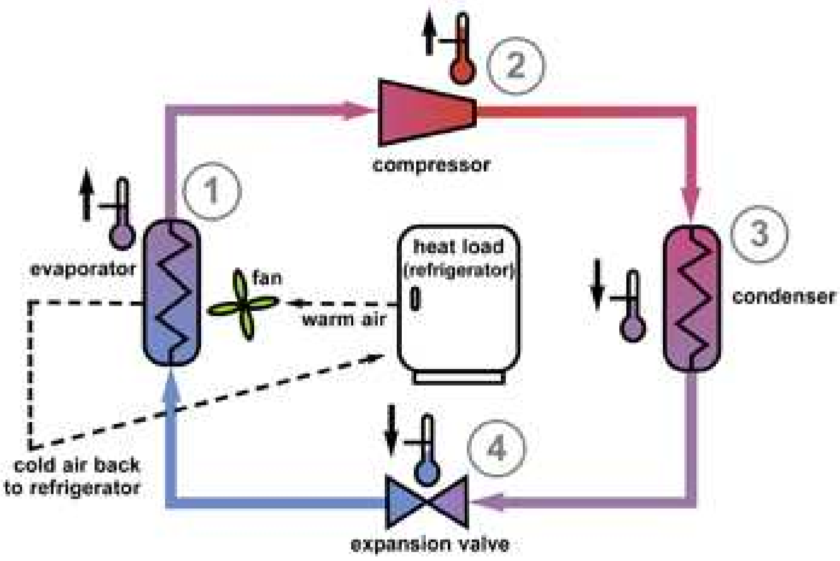


Figure 77; Components schematic of refrigeration cycle

Cooling is achieved by evaporating a liquid refrigerant through an expansion device within the EVAPORATOR. In the evaporation process refrigerant liquid absorbs heat around the evaporator. This amount of heat extracted is called thermal load (important for calculations concept on our system later). From the evaporator, the refrigerant is conducted to the COMPRESSOR which is responsible of increasing the gas pressure for later condense it into another heat exchanger, the CONDENSER, makes the gas liquid again. In this heat exchanger cooling system, the heat is extracted in the evaporator. Because of that the refrigerant change into liquid state again thus leads to the EXPANSION VALVE to return to evaporate and the refrigeration cycle starts again.

From all the previous cycle it is important to keep the concept that in the evaporator there is an extraction of heat from the desired location and in the condenser is released to the outside, being led by a compressor refrigerant at high pressure.

3.2 Purposed system.

Of all the components that make up the refrigeration cycle are going to be studied the evaporator and condenser, which are what performs the function being sought.

Moreover, the compressor is the one that provides the mechanical energy to the fluid to circulate throughout the hydraulic circuit, which also is going to be studied.

3.2.1 Exchangers

In the market can be found three mains exchangers models.

· ***Radiators***
· ***Condensers***
· ***Dry coolers***

Now it is going to be explained each one of them;

-Radiators: are heat exchangers used to transfer from a medium to another the thermal energy, with this action it can be possible heating or cooling. In the case of the system purposed the idea is to cool a fluid or coolant.

The radiators studied for making a choice were outside radiators. It consists of two different parts; the fans and the casing. The fans are usually direct driven fan units and the casing is usually stainless or efficiently corrosion protected.

Finally after looking for a company with good quality products it was found Thermokey which one didn´t offer radiators, but searching in other companies it was checked that the radiators were not the product that was needed to implement the new system because the radiators have not forced convection (no fans).

Figure 31; Radiators

-Condensers: the mission of a condenser is to condense a substance from it gaseous state to its liquid state, this process is made by cooling the substance. The power of the condenser can vary a lot for example it is possible to find condensers of 10KW until 202KW just only in Thermokey company but the power range is higher than this one. It is possible to find condensers of lower and higher power too. There are different types of condensers, ones that are flat that have 1-3 fans and the V-type that have 2-4 fans.

Speak of condensers mean speak of compressors too. The condenser is part of the refrigerant cycle, and it is composed by an expansion valve, an evaporator, a compressor and the condenser. For the new system it is not needed any compressor so condensers are not chosen because of this reason, which mean that the new system is not going to have higher energy consumption as if condensers were used.

Figure 32; condenser

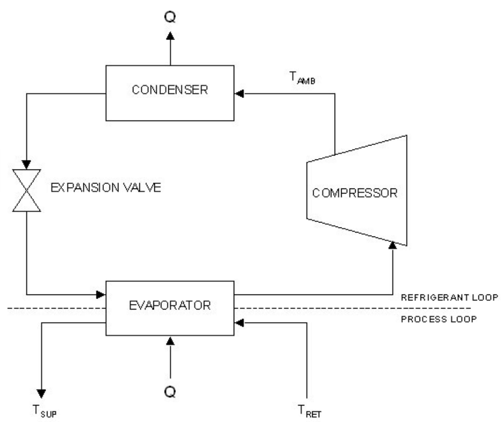


Figure 33; Refrigerated cycle

-Dry coolers: This one is the type that has been selected for the Project. The main reasons are the good relation between the heat absorbed and the size, also another good relation is the heat absorbed and the energy absorbed. Another important point to mention is that these dry coolers are used for cooling the ambient so is the best option for the areas that are shown in the project.

The most important point to remark is that the COP is quite high, as it is shown in the calculations done below.

The selection of these dry coolers is because the areas 1, 2, 3, 4 and 5 are quite similar because are closed areas. The main problem could be with the area 2 which is smaller comparing it with others areas. This difference of volume is traduced in an extra power dry cooler, the dry cooler selected for this area is the one with the lowest power capacity but it is not necessary so much power to cool down the area.

For the rest of the areas the chosen dry coolers meet the requirements for cooling each area.

Figure 34; Dry cooler

3.2.2 Pump

To perform the extraction of heat from the inside room and carry it outside, dry coolers are used. But now must be chosen how it is going to circulate the refrigerant liquid throughout the piping system, for it can only be chosen between a pump and a compressor.

Because their functions are the same but their fields of application different it is necessary to define where you can implemented each of them;

Compressor; a compressor is used for compressible fluids, such as gas or steam, it can work in a high pressure circuit and can drive a caudal flow medium or low.

Pump; unlike compressor, the pump is used for incompressible fluids such as water or liquid coolants, their range of use is very diverse and ranges from very low flows to very high, usually pumps are installed on low pressure circuits.

As our system is anticipated that will operate at low pressure and with a liquid refrigerant it will use a pump.

3.3 Sketch of the first idea

So you can already make a sketch of an initial idea of the system design, later the calculations are going to be done;

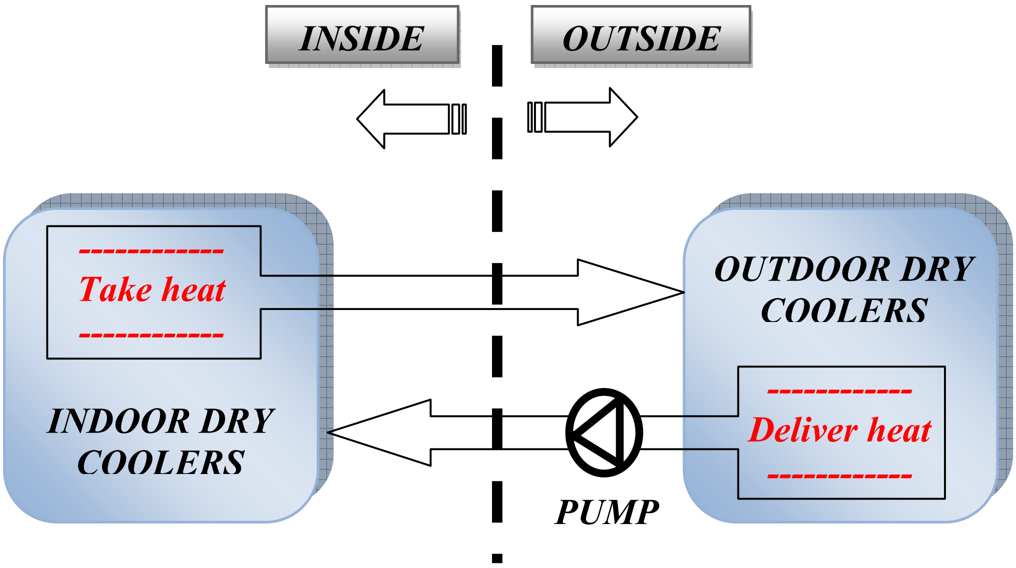


Figure 85; previous sketch of the dry cooler system

3.4 Implantation site

To achieve a running considerable time per year it should be chosen a city where outside temperatures is within a specific range.

This fact is very important because, although the system has excellent performance, it cannot save energy if it is not running. Intuitively believe that Scandinavian cities can be a good proposal for every application of the device. Specifically the study will be made on the Willys supermarket, in Gävle.

Gävle is a Swedish city with coordinates +60º 40' Latitude and +17º10' Longitude, and with a height above sea level of 16m.

On particular, Willy's is the company which has the suitable conditions to implant the cooling system. This company is dedicated to the selling of every type of aliment. To realize its activities Willys needs an enclosure where can be put the showcases, freezers and all the elements to conserve cold products. In these areas is where the system will be placed.

Figure 86; Location

As it is possible to see in the table xxx Gävle is an excellent place for the installation of the prototype, with an average monthly temperature in winter months of;

Month 2009	Tmean	Tmin	Tmax
January	-3	-17	7
February	-6	-21	2
March	-1	-22	9
April	6	-5	21
October	4	-5	13
November	4	-2	11
December	-5	-23	6

Table 82; Temperature resume

4. METHODOLOGY

The real handicap to develop this project has been the absence of information about similar subjects. This fundamental handicap has been the main motive to take a methodology of cycle *"study -> test -> error"*.

Through this method all different systems and components have been checked to get the best solution. Therefore it has been needed to make constantly recalculations to find the more efficient solution.

On the other hand, the method "study-test-error" has permitted to find the exact system with the exact components to improve until the maximum the total COP of the designed system. That has been done in this way because each repeated iteration has optimized the system until find the best solution.

Below it is showed a short schema with the method resumed;

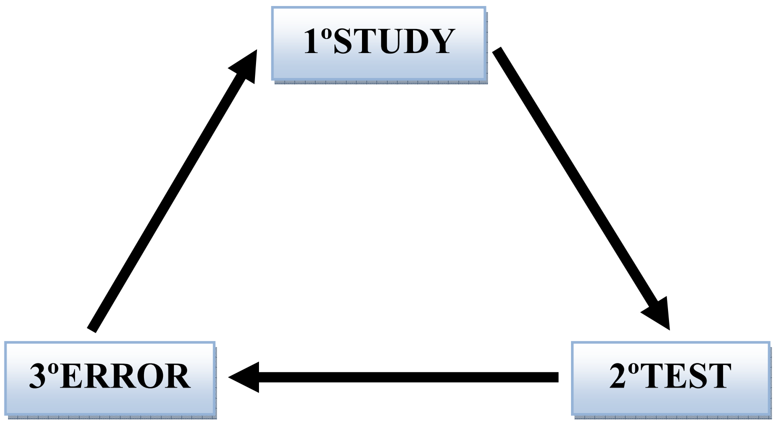


Figure 87; Cycle of methodology "study-test-error"

This method may be applied from a general approach to specifics calculations. To understand the exact process followed on the project it has to be seen the picture below (**figure 88**).

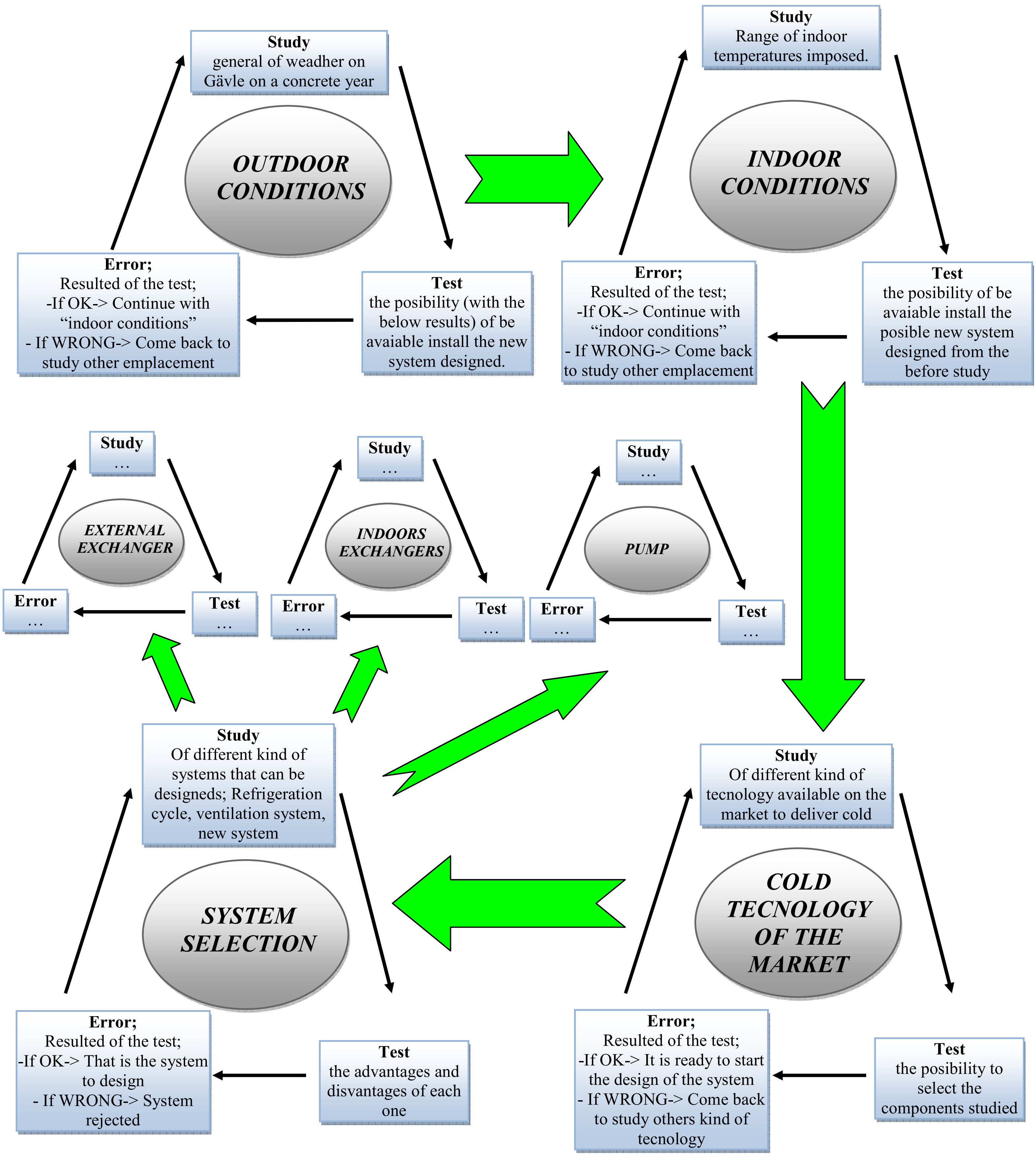


Figure 88; Application of the method

That is only a small example of the method followed across the project. A study from the general concept to the particular components has been carried out subjected at the "study-test-error" method.

Therefore the project has been studied under a constant feedback, this assure a revision continued that minimize the possibility apparition of failures. Moreover it gets to improve the quality of the final solution and contributes to finals result with high reliability and viability.

5. THEORICAL FRAMEWORK

Theoretical procedures used to carry out the different kinds of calculations are based on equations and selection procedures demonstrated for others authors.

Throughout the whole project there are four distinct types of calculation; calculation of internal heat exchangers, calculation of external heat exchangers, pump calculations and calculation of the efficiency of the system depending of the externals temperatures.

Other types of calculations have been realized in order to estimate the total time that the system is working. It has been done other kinds of calculations to estimate how much time over one year is running the system. It should be borne in mind that these calculations are always an estimation subjects to slight variations depending on the weather of each year.

Due to the complexity of explaining all the calculations together are separated then to specify in detail each of them.

The first step is checking the ranges of external and internal temperatures to take an idea of the project viability. When it will be demonstrated this fact then it will be available to continue with the system components calculations. The internals exchangers must be the first on calculate, this is because we expect the completion of the calculations follow a pyramid structure. The the result of the calculations are made as the needs of the following calculations to be performed;

- The indoor dry coolers should be calculated based on the needs of each chilled room.

- The outdoor dry coolers are going to be calculated based on the needs of indoor dry coolers.

- Pump should be calculated based on the need for coolant flow from the previous two.

So if for some reason it is needed to change any cold room, you should recalculate the other components of the installation in the order listed.

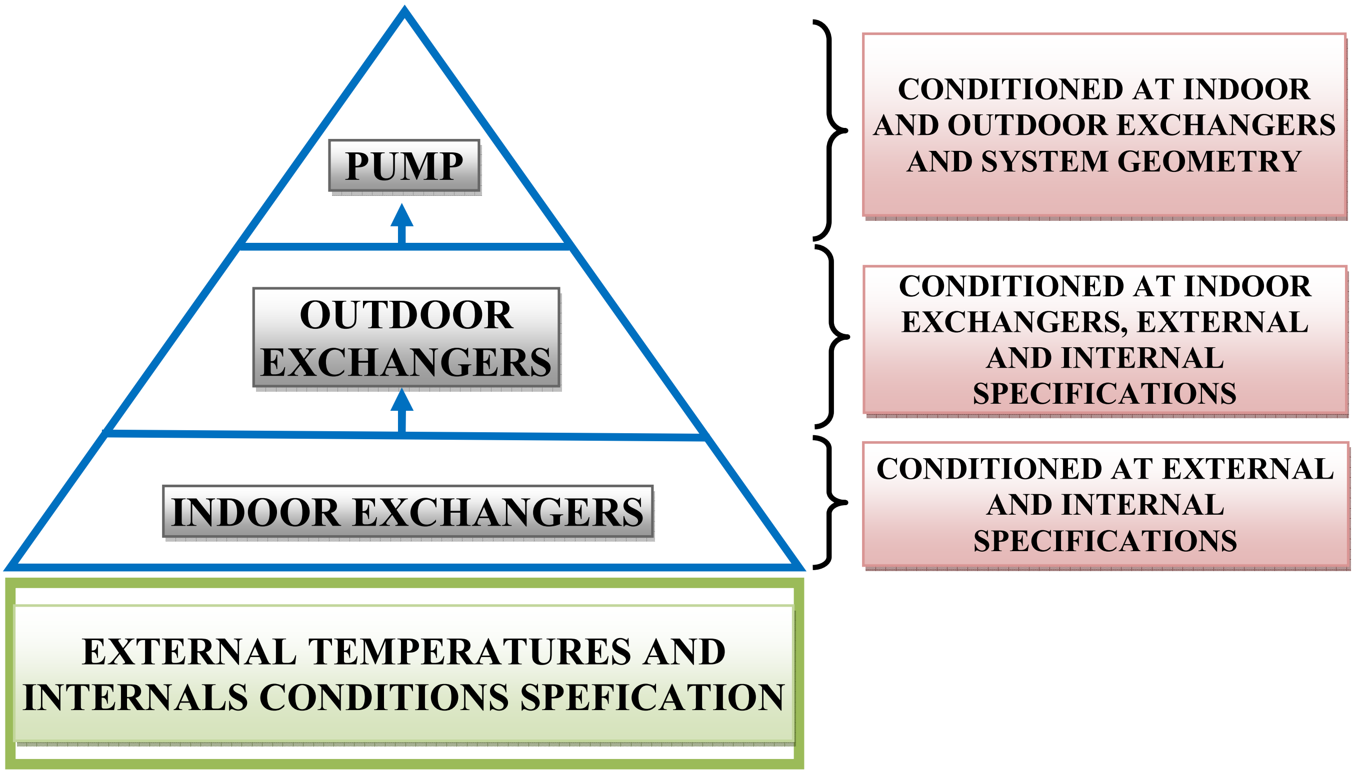


Figure 89; calculations with pyramidal structure

5.1 Procedure to estimate the outdoor temperatures probability

One of the great challenges and obstacles of this project is making an accurate estimation of the operation hours of the system.

It has been used a historical data provided by the "Sveriges meteorologiska och hydrologiska institute" (SMHI) to get temperatures of the reference year, 2009.

Once taken the necessary data has been used MINITAB program to probabilistic studies, to estimate with great accuracy the likelihood that appears throughout the year the temperatures searched.

After processing the data with MINITAB, it has been obtained graphics for each month from which can be deduced the probability of appears a certain temperature.

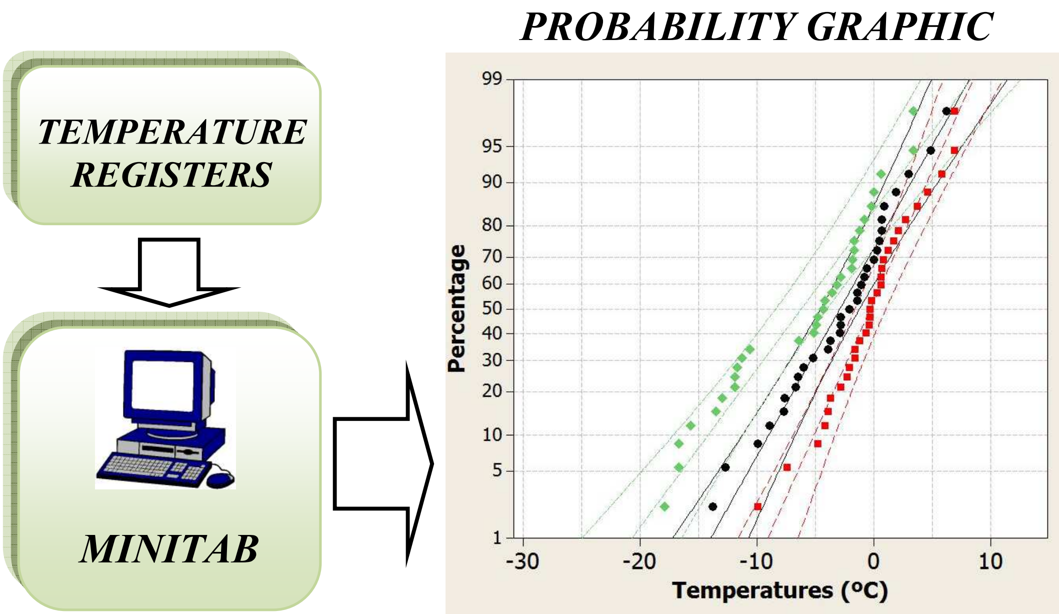


Figure 90; obtaining of the probability graphics

5.2 Procedure to measure indoor temperatures

Unlike the previous case in which it had accurate records, inside Willys supermarket there are no previous measurements of the temperatures.

On the other hand it has to be distinguished two zones within the Willys where the thermal characteristics are totally different, open spaces and closed rooms.

To determine the temperatures in enclosed spaces, it has been checked from the Swedish legislation of refrigerated areas, where it is specified the range of temperatures, that the temperature in each enclosed area should be based on the type of food that it contains.

The procedure chosen for the taking of measures in open areas has been using an infrared temperature gun. With the emissivity calibration parameter can be obtained a very precise measurement of any object pointed with the gun.

There is one problem and the cause of this problem is that the space is very large and cannot be considered an average temperature for the entire campus. So only it can be done an accurate estimation by considering the temperature profile from the ground each 1m, 2m and 3m.

Finally, because of the large number of measures that were had to be taken, it has devised an instrument to measure temperatures in the shortest possible time.

Using a very thin polyethylene film painted on black, it was obtained a surface with very low emissivity and it was able to match the temperature of its surroundings in a very short time

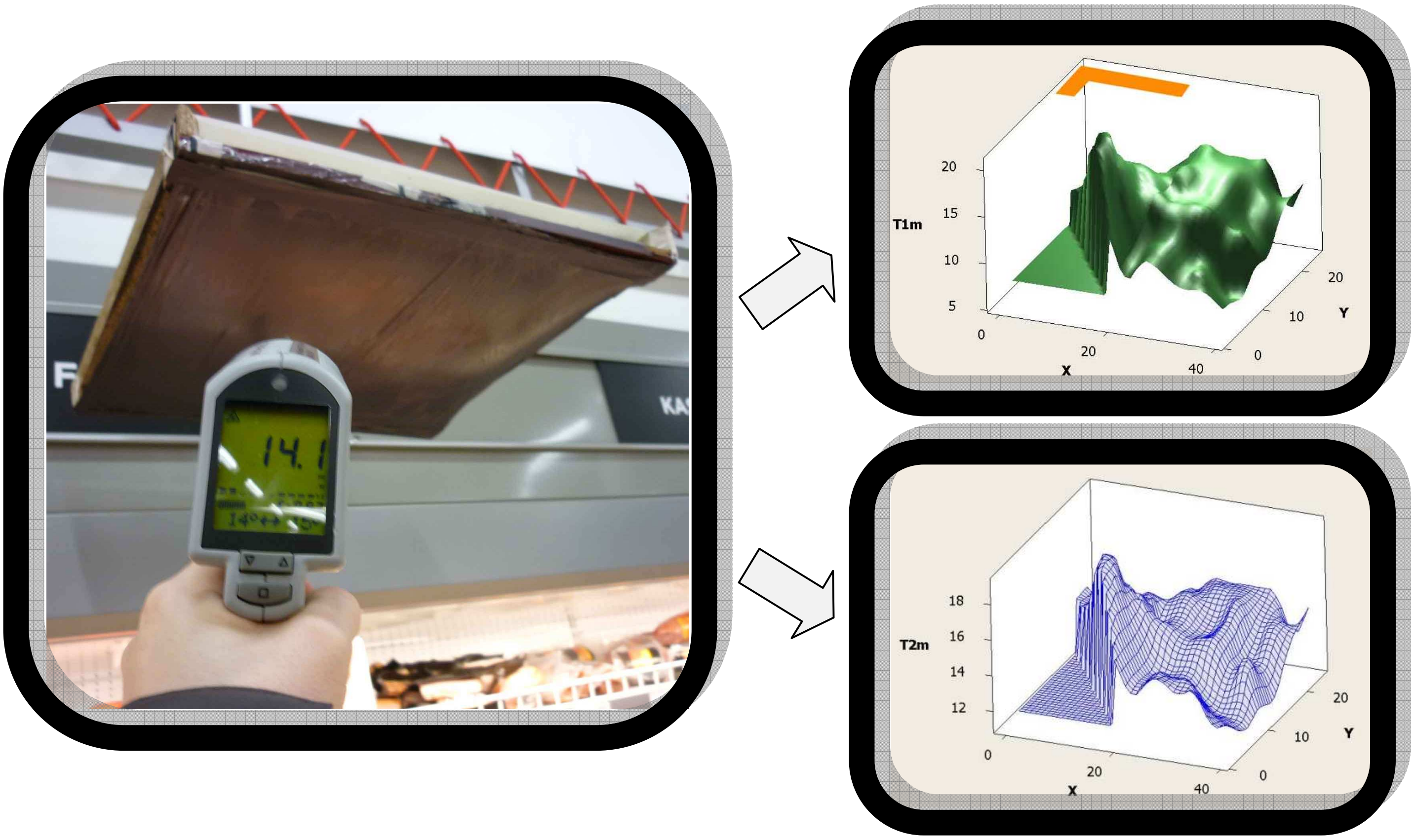


Figure 91; Example of temperature measure

5.3 Procedure to calculate the internal exchangers

Bring to dig the study of a heat exchanger is a process that is in itself a full project, however there are businesses that are specifically dedicated to it.

In this project the exchangers that are going to be used have been designed by Thermokey, manufacturer and distributor of cooling systems worldwide.

The procedure for the selection of the exchanger is basically to determine the exchanger required through a series of variables parameters.

On the other hand it must be have in mind that each site has different thermal cooling needs, which is also necessary to determine what those powers before starting calculations.

This problem has been solved thanks to the old cooling system, in a previous project was already necessary to determine the thermal requirements of each area. In our case only had to have in mind that our exchanges had to have the same power or more than those that are already installed.

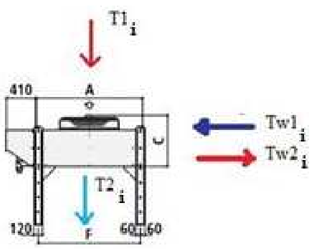
The criteria for determining the exchanger, once calculated the minimum necessary, has not been to take an exchanger similar to the old one installed. It has been taking in care the fact of having a good COP, although the selected cooling power has been somewhat greater than the one required.

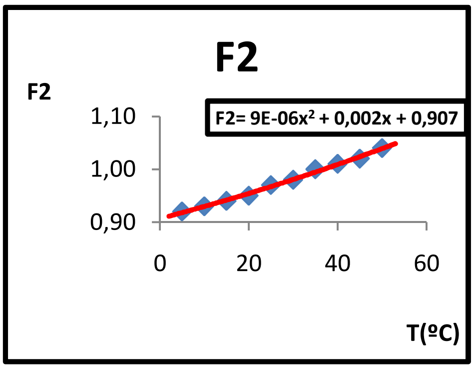
THERMAL NEEDS

DIFERENTS ROOMS

CALCULATIONS

Minimum capacity > Qreal*F1*F2*F3*F4*F5*F6





MINIMUM THERMAL POWER OF THE DRY COOLER

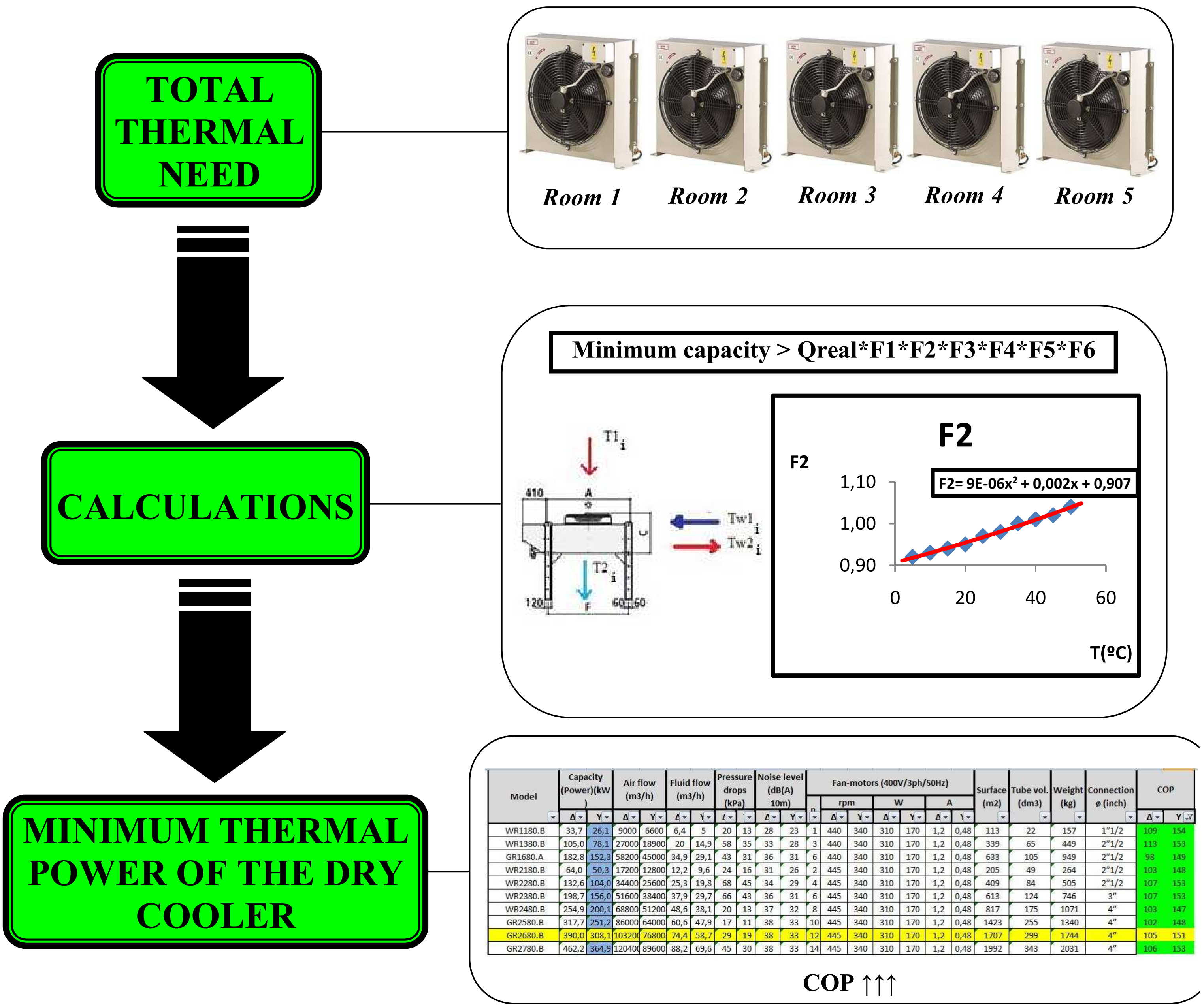
Model	Capacity (Power)(kW)		Air flow (m3/h)		Fluid flow (m3/h)		Pressure drops (kPa)		Noise level (dB(A) 10m)		Fan-motors (400V/3ph/50Hz)							Surface (m2)	Tube vol. (dm3)	Weight (kg)	Connection ø (inch)	COP	
											n	rpm		W		A							
	Δ	Y	Δ	Y	Δ	Y	Δ	Y	Δ	Y		Δ	Y	Δ	Y	Δ	Y					Δ	Y
WR1180.B	33,7	26,1	9000	6600	6,4	5	20	13	28	23	1	440	340	310	170	1,2	0,48	113	22	157	1”1/2	109	154
WR1380.B	105,0	78,1	27000	18900	20	14,9	58	35	33	28	3	440	340	310	170	1,2	0,48	339	65	449	2”1/2	113	153
GR1680.A	182,8	152,3	58200	45000	34,9	29,1	43	31	36	31	6	440	340	310	170	1,2	0,48	633	105	949	2”1/2	98	149
WR2180.B	64,0	50,3	17200	12800	12,2	9,6	24	16	31	26	2	445	340	310	170	1,2	0,48	205	49	264	2”1/2	103	148
WR2280.B	132,6	104,0	34400	25600	25,3	19,8	68	45	34	29	4	445	340	310	170	1,2	0,48	409	84	505	2”1/2	107	153
WR2380.B	198,7	156,0	51600	38400	37,9	29,7	66	43	36	31	6	445	340	310	170	1,2	0,48	613	124	746	3”	107	153
WR2480.B	254,9	200,1	68800	51200	48,6	38,1	20	13	37	32	8	445	340	310	170	1,2	0,48	817	175	1071	4”	103	147
GR2580.B	317,7	251,2	86000	64000	60,6	47,9	17	11	38	33	10	445	340	310	170	1,2	0,48	1423	255	1340	4”	102	148
GR2680.B	390,0	308,1	10320(	76800	74,4	58,7	29	19	38	33	12	445	340	310	170	1,2	0,48	1707	299	1744	4”	105	151
GR2780.B	462,2	364,9	12040(	89600	88,2	69,6	45	30	38	33	14	445	340	310	170	1,2	0,48	1992	343	2031	4”	106	153

COP ↑↑↑

Figure 92; Resume of the dry cooler selection

5.4 Procedure to calculate the external exchanger

The procedure to calculate the external heat exchanger it is similar to the internal one, but in this case the thermal needs are the sum of the internals thermal needs required.



Model	Capacity (Power)(kW)		Air flow (m3/h)		Fluid flow (m3/h)		Pressure drops (kPa)		Noise level (dB(A) 10m)		Fan-motors (400V/3ph/50Hz)							Surface (m2)	Tube vol. (dm3)	Weight (kg)	Connection ø (inch)	COP	
											n	rpm		W		A							
	Δ	Y	Δ	Y	Δ	Y	Δ	Y	Δ	Y		Δ	Y	Δ	Y	Δ	Y					Δ	Y
WR1180.B	33,7	26,1	9000	6600	6,4	5	20	13	28	23	1	440	340	310	170	1,2	0,48	113	22	157	1"1/2	109	154
WR1380.B	105,0	78,1	27000	18900	20	14,9	58	35	33	28	3	440	340	310	170	1,2	0,48	339	65	449	2"1/2	113	153
GR1680.A	182,8	152,3	58200	45000	34,9	29,1	43	31	36	31	6	440	340	310	170	1,2	0,48	633	105	949	2"1/2	98	149
WR2180.B	64,0	50,3	17200	12800	12,2	9,6	24	16	31	26	2	445	340	310	170	1,2	0,48	205	49	264	2"1/2	103	148
WR2280.B	132,6	104,0	34400	25600	25,3	19,8	68	45	34	29	4	445	340	310	170	1,2	0,48	409	84	505	2"1/2	107	153
WR2380.B	198,7	156,0	51600	38400	37,9	29,7	66	43	36	31	6	445	340	310	170	1,2	0,48	613	124	746	3"	107	153
WR2480.B	254,9	200,1	68800	51200	48,6	38,1	20	13	37	32	8	445	340	310	170	1,2	0,48	817	175	1071	4"	103	147
GR2580.B	317,7	251,2	86000	64000	60,6	47,9	17	11	38	33	10	445	340	310	170	1,2	0,48	1423	255	1340	4"	102	148
GR2680.B	390,0	308,1	10320(	76800	74,4	58,7	29	19	38	33	12	445	340	310	170	1,2	0,48	1707	299	1744	4"	105	151
GR2780.B	462,2	364,9	12040(	89600	88,2	69,6	45	30	38	33	14	445	340	310	170	1,2	0,48	1992	343	2031	4"	106	153

Figure 93; Resume of the outdoor exchanger selection

5.5 Procedure to calculate the pump

The pump is a key element of the system. With the choice of a suitable pump is going to be possible to get a system which is going to perform it function with the least possible energy expenditure, with a COP as high as possible.

The choice of the pump has been accomplished through the determination of its power from the equation of the pump. (Setting the flows to every part of the hydraulic installation, loss of freight, so on)

The specific steps undertaken to determine the best pump for the system are;

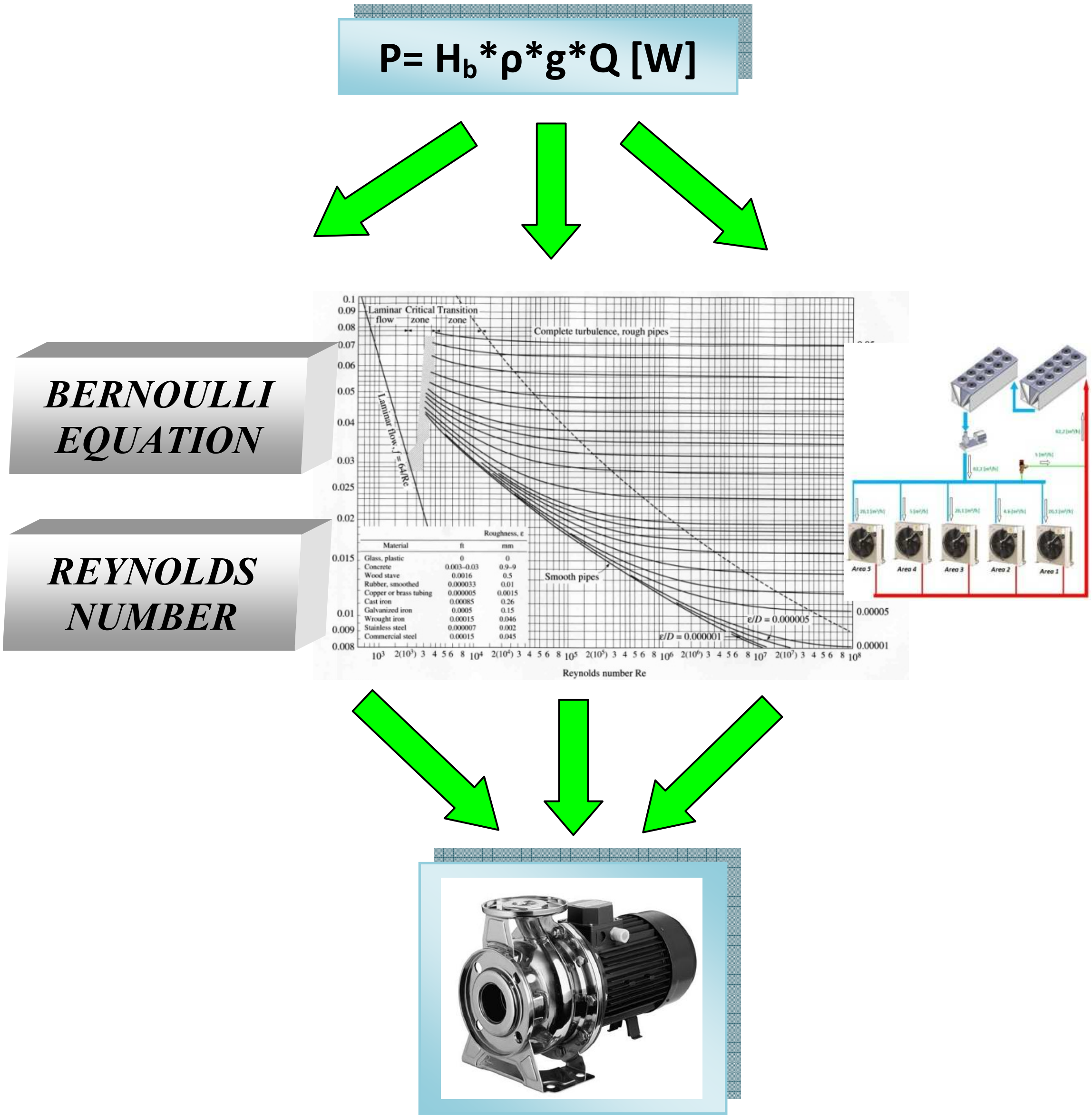


Figure 94; Resume of the pump selection

5.6 Procedure to estimate the working time and the energy saved

One of the biggest challenges of the Project has been the fact of making a reliable estimation of the working range of the system.

It is necessary to take into account that for each certain outdoor temperature the indoor temperature of the rooms can vary between 8ºC and 0ºC, consequently it can be found around 160 situations.

The idea consist in define exactly which are the temperatures that can be reached inside the refrigerated rooms in function of the outdoor temperatures. When the outdoor temperature starts to increase then a value will be reach where the save of energy will start to decrease (this is because the wanted minimum temperature is impossible to be reached). In this moment is when the system has to start working in parallel with the old system, this is the reason of because the energy saved decreases considerably for temperature values near to 0ºC.

When this problem is solved, then the next step is apply the results to a determined interval of time (in this case are the months of December 2009, January 2010, February 2010 and March 2010). The idea is for guessing the time in hours during in which the system is working and in what working conditions it is working.

Finally it must be done a comparison between the consumed energy by the system that is now installed in Willys and the consumption of this system working in parallel with the new one. Then can be obtained by the difference of both the fridge energy saved and consequently the costs in electrical energy.

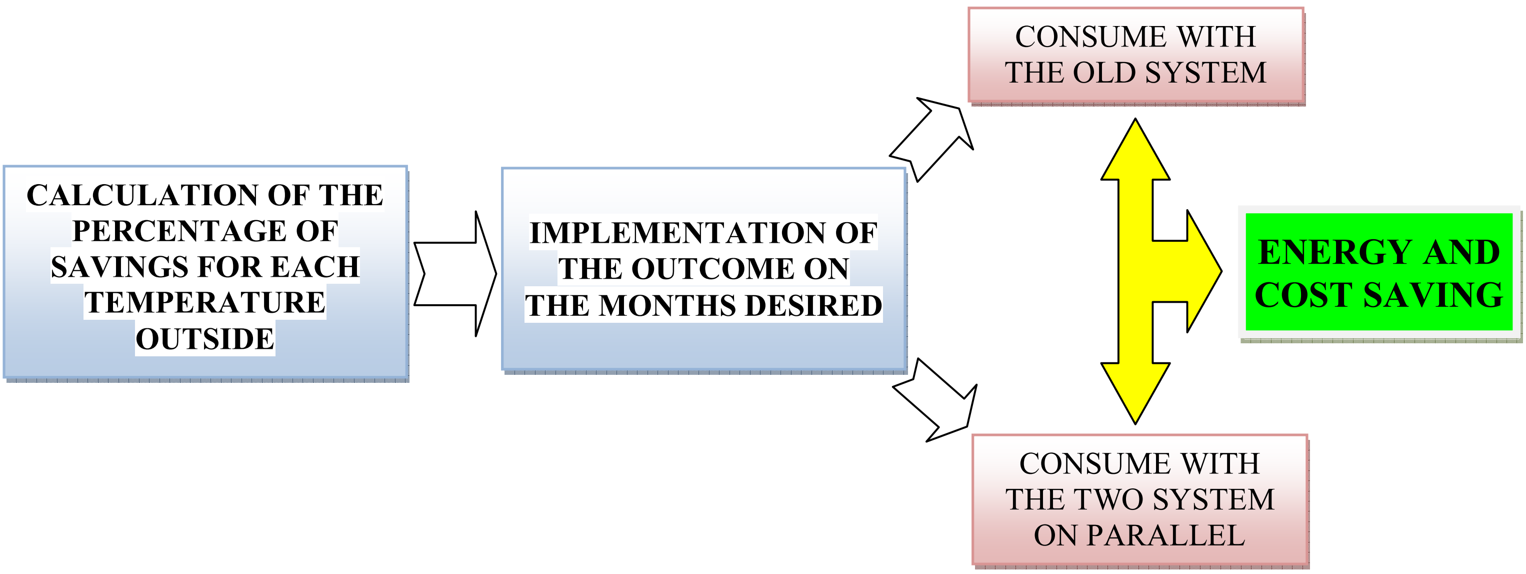


Figure 95; Resume of how the systems can work

6. RESULTS

All results obtained are relatives of the final structure of the project are subject to the geometry of where the system is implemented, the internal conditions of each room (needs refrigeration and temperature), the outside temperature, etc.

By this it is meant that the types of dry cooler chosen, the pump and all other system add-ons are specifically suited to these needs. If you ever want to use the same system for other needs (for example, cooling a car paint shop) have to be redone all calculations with the new conditions specified.

6.1 Indoor dry coolers, outdoor dry coolers and pump selected.

The final system chosen was five dry internal coolers (one for each room) and one external. It has considered the possibility of installing two external dry coolers, then if one fails the system can continue operating at least with a 50% power.

It was not chosen this option because after studying all the possible combinations, with two external dry coolers the total COP is 4 points less than using only one external cooler, with the consequent losses of savings.

Specific characteristics for each element are;

Line	Exchanger Model	Cooling Power [kW]	Refrigerant flux [m^3/h]|[m^3/s]	Minimum diameter [mm^2]	Longitude [m]
1	***GR2680.B***	308	58,7 | 0,016	76	55
2	***WR1480.B***	105,7	20,1 | 0.0055	76	3
3	***WR1263.A***	24,3	4,6 | 0.0012	38	7
4	***WR1480.B***	105,7	20,1 | 0.0055	76	3
5	***WR1263.B***	26,1	5 | 0.0014	51	7
6	***WR1363.B***	39,5	7,5 | 0.002	51	3
1'	go out	326	57,3 | 0.0173	76	55
2'	go out	105,7	20,1 | 0.0055	76	3
3'	go out	24,3	4,6 | 0.0012	38	7
4'	go out	105,7	20,1 | 0.0055	76	3
5'	go out	26,1	5 | 0.0014	51	7
6'	go out	39,5	7,5 | 0.002	51	3
A	*Auxiliary line*	*6,5*	*1,5* | 0.0014	*30*	*≈0*
		Total indoor flux	57,3 |	**L total**	156
		Total outdoor flux	58,7 |		

Table 62; system characteristics

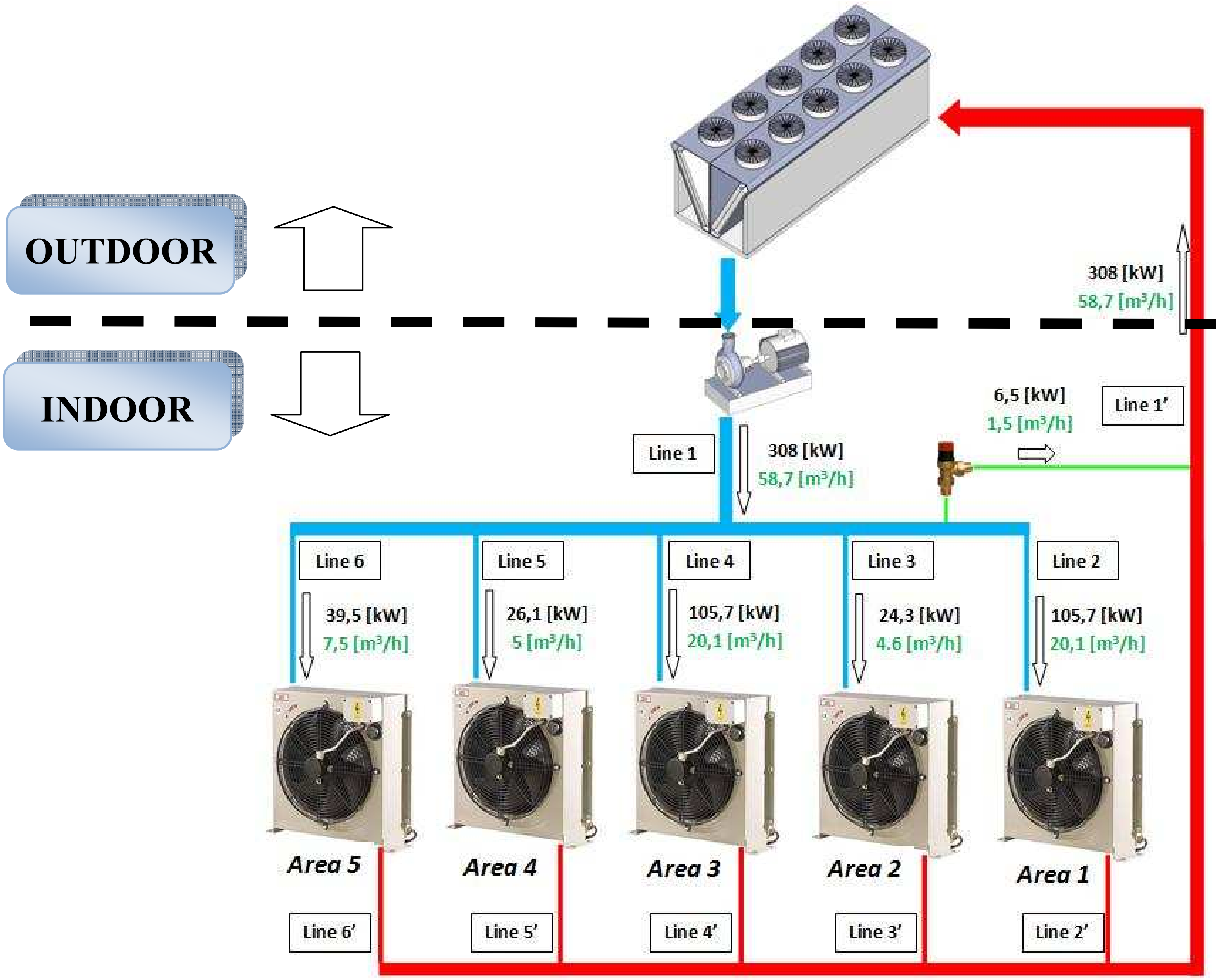


Figure 52; System schematic. Cooling and refrigerant fluxes.

6.2 System schematic. Total COP

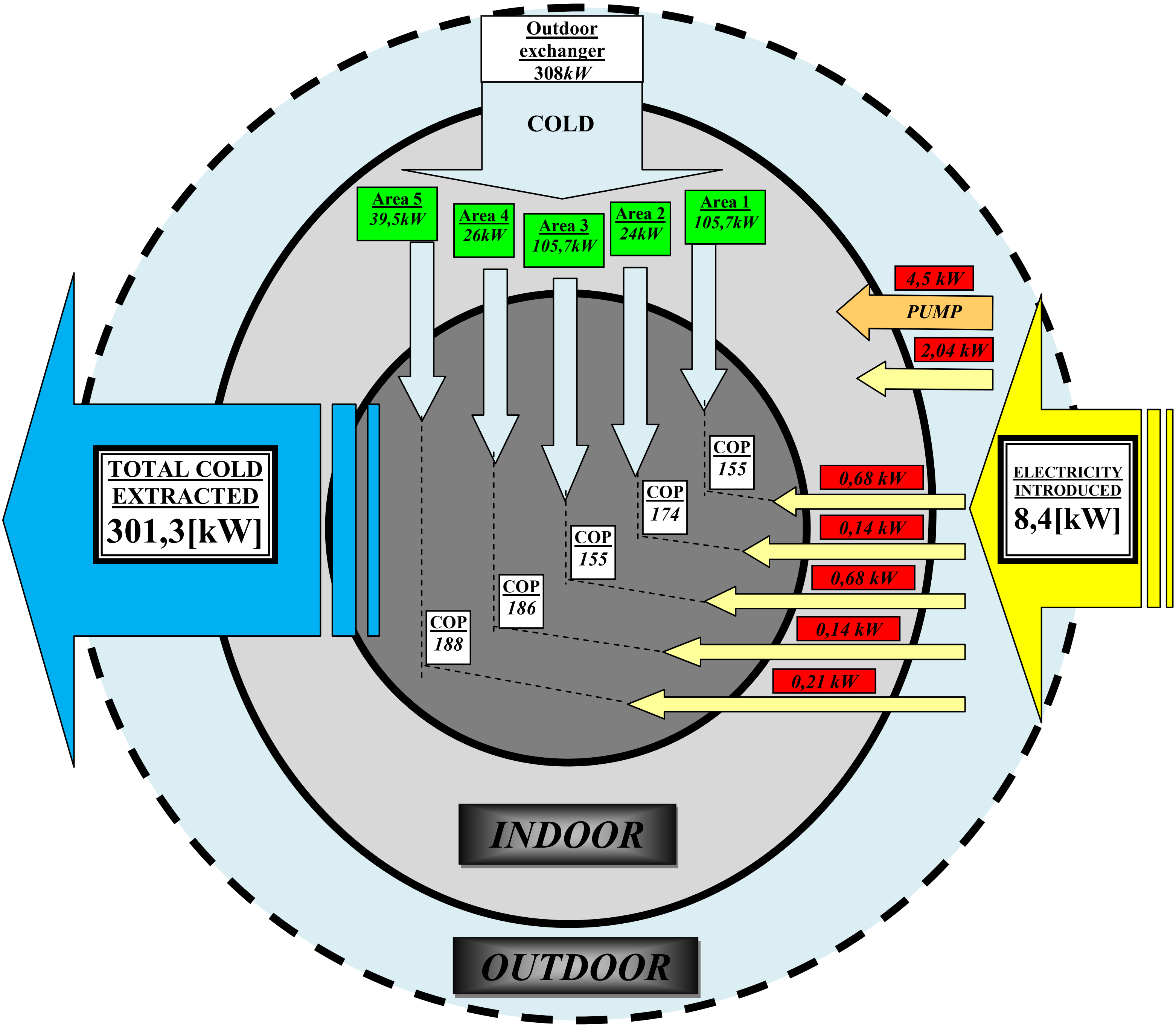


Figure 75; Cooling flux and electricity fluxes. Local and total COP.

Taking into account the consumption of all the components **(8,4 kW)**, the cooling energy delivered within the cold rooms **(301,3kW)** and losses **(≈ 7kW)**, the total COP of the system is;

$$COP = 36$$

6.3 Energy and cost saving

In the **table 67** it is shown the percentage of energy saved in function of the different outdoor temperature. For understand the table it is important to look the first and the last column (painted in green). The first one is for the outdoor temperature that is going to be necessary to obtain the saving percentage shown in the last column.

As information it is shown in the blue column the temperature reached inside the indoor rooms. The system attends to three working types;

- **From -20ºC to -9ºC→** working alone→ *Case "T"*
- **From -8ºC to -4ºC→** working on parallel→ *Cases "1", "2", "3", "4", "5"*
- **From -4ºC in forward→** *Not working*

T outside [ºC]	*T inside got [ºC]*	*CASE*	*CASE DESCRIPTION*	*ENERGY SAVED (%)*
-20->-11	0	***T***	Dry cooler system working alone	***91,7***
-10	1	***T***	Dry cooler system working alone	***91,7***
-9	2	***T***	Dry cooler system working alone	***91,7***
-8	3	***1***	From 8ºC->3ºC / 3ºC->0ºC	***57,3***
-7	4	***2***	From 8ºC->4ºC / 4ºC->0ºC	***45,3***
-6	5	***3***	From 8ºC->5ºC / 5ºC->0ºC	***34,3***
-5	6	***4***	From 8ºC->6ºC / 6ºC->0ºC	***23***
-4	7	***5***	From 8ºC->7ºC / 7ºC->0ºC	***12***
-3	NOT WORKING	NOT WORKING	NOT WORKING	**NOT WORKING**
-2	NOT WORKING	NOT WORKING	NOT WORKING	**NOT WORKING**
-1	NOT WORKING	NOT WORKING	NOT WORKING	**NOT WORKING**
0	NOT WORKING	NOT WORKING	NOT WORKING	**NOT WORKING**

Table 67; Relationship between the outside temperature/inside temperatures reached/Energy saved

DECEMBER 2009

The estimated energy savings and costs for the month of December is ***5750[kWh]*** and ***4140[SEK]***.

If a coefficient is considered with a working time of 0.85 (a downtime of the cooling system 15%) the total savings would be ***4888[kWh]*** and ***3519[SEK]***.

CASE	SAVED (%)	HOURS	[kWh]BEFORE	[kWh]AFTER
T	91,7	258	5392	448
1	53,3	0	0	0
2	45,3	48	1003	549
3	34,3	24	502	330
4	23	6	125	97
5	12	60	1254	1104
	TOTAL [kWh]		**8276**	**2526**
	[kWh/month] SAVED		***5750***	
	[SEK/ month] SAVED		***4140***	

Table 70; calculation of energy and money saved in December

JANUARY 2009

The estimated energy savings and costs for the month of January is ***10295[kWh]*** and ***7412[SEK]***.

If a coefficient is considered with a working time of 0.85 (a downtime of the cooling system 15%) the total savings would be ***8751[kWh]*** and ***6300[SEK]***.

CASE	SAVED (%)	HOURS	[kWh]BEFORE	[kWh]AFTER
T	91,7	441	9217	765
1	53,3	54,5	1139	532
2	45,3	35	732	400
3	34,3	78,5	1641	1078
4	23	48	1003	772
5	12	26	543	478
	TOTAL [kWh]		**14275**	**4026**
	[kWh/month] SAVED		***10295***	
	[SEK/ month] SAVED		***7412***	

Table 73; calculation of energy and money saved in January

FEBRUARY 2009

The estimated energy savings and costs for the month of February is ***7974[kWh]*** and ***5741[SEK]***.

If a coefficient is considered with a working time of 0.85 (a downtime of the cooling system 15%) the total savings would be ***6778[kWh]*** and ***4880SEK]***.

CASE	SAVED (%)	HOURS	[kWh]BEFORE	[kWh]AFTER
T	91,7	339	7085	588
1	53,3	0	0	0
2	45,3	90	1881	1029
3	34,3	18	376	247
4	23	75	1568	1207
5	12	54	1129	993
	TOTAL [kWh]		**12038**	**4064**
	[kWh/month] SAVED		***7974***	
	[SEK/ month] SAVED		***5741***	

Table 76; calculation of energy and money saved in February

MARCH 2009

The estimated energy savings and costs for the month of December is ***2396[kWh]*** and ***1725[SEK]***.

If a coefficient is considered with a working time of 0.85 (a downtime of the cooling system 15%) the total savings would be ***2037[kWh]*** and ***1467[SEK]***.

CASE	SAVED (%)	HOURS	[kWh]BEFORE	[kWh]AFTER
T	91,7	37,5	784	65
1	53,3	37,5	784	366
2	45,3	62,5	1306	715
3	34,3	25	523	343
4	23	82,5	1724	1328
5	12	24	502	441
	TOTAL [kWh]		**5622**	**3258**
	[kWh/month] SAVED		***2396***	
	[SEK/ month] SAVED		***1725***	

Table 79; calculation of energy and money saved in March

SUMARY

The total estimated savings for the winter of 2009/2010 is (assuming a coefficient of running time of 100%);

MONTH	ENERGY SAVED [kWh/year]	COST SAVED [SEK/year]	% Saved over total
December 09	5750	4140	22
January 10	10295	7412	39
February 10	7974	5741	30
March 10	2396	1725	9
TOTAL	***26415 [kWh/year]***	***19190 [SEK/year]***	**-**

Table 80; saving summary

If it is considered as downtime of the cooling system by 15% (time since the system stops until it returns to work to cool the rooms), the saving would be ***22453[kWh]*** and ***16312[SEK]***.

6.4 Resume of feasibility and budget

The objective followed through all the project has been to get a system as efficient as it is possible. It has kept all time in mind to don't use more elements than the necessary ones.

Despite the energy saved has a very good value, it can occur that the total cost of the project is as high as it is not feasible to build.

It must be separated two kinds of costs, the investment cost and the life cycle cost. The first one is only the cost of the components, elements, materials and man work, without the maintenance cost accumulated on the system life. The second one has into account the total system life (estimated on 15 years) and the capital cost of the bank loan (2%).

Below it is showed a sketch of the budget, into the circumference appears the particular percentage over total cost;

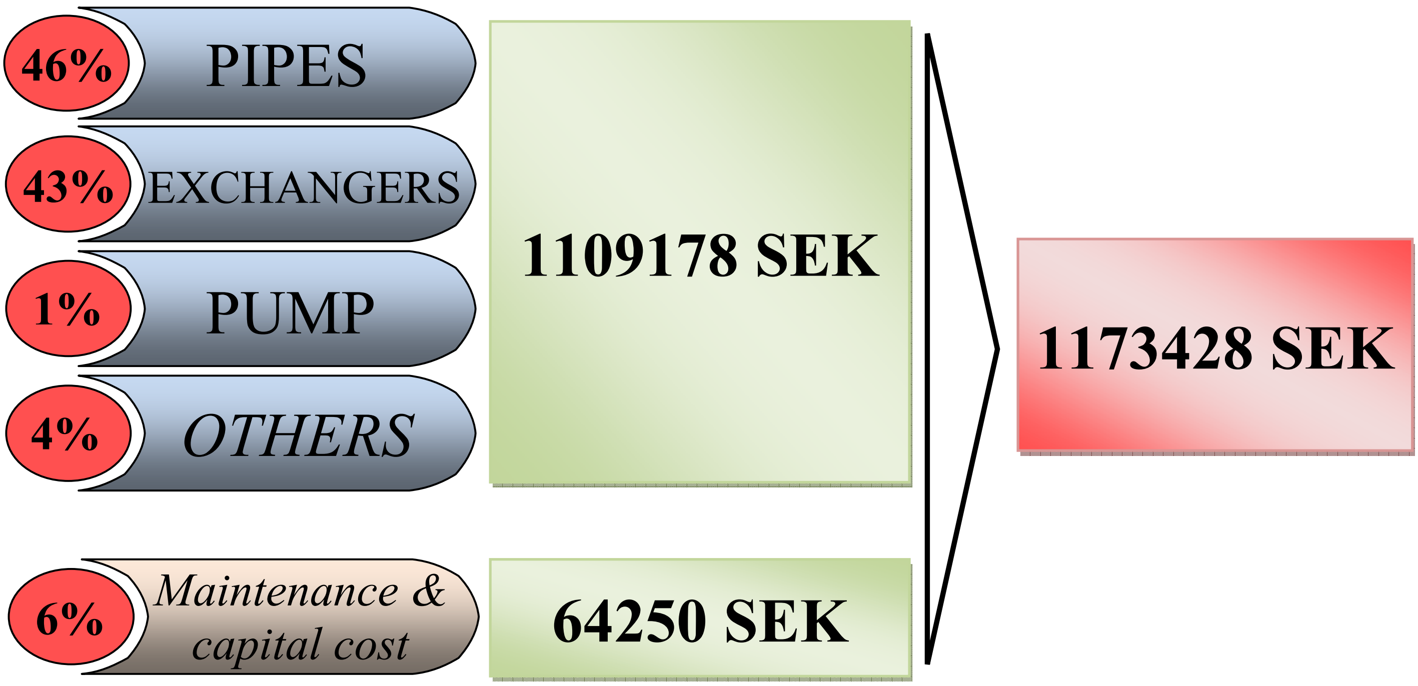


Figure 96; Budget sketch

From de **figure 96** it is deduced that the highest amount of the final cost is due to pipes and exchangers. The pump, maintenance, capital cost and others (As valves cost, refrigerant liquid, etc.) are not very expensive, around 11% of the life cycle cost.

Therefore, to improve the expenses it should be studied the different providers from the market, although the exchanger and pipes prices don't varies too much from a provider to other. So it is very difficult to decrease significantly the total cost.

On the other hand it is needed to know if the money saved is higher than the sum of the cost on the system life. If this occurs, the system will be feasible.

So, to calculate the total amount of money saved it must be multiplied the saved cost on one year by the total years of life, giving a total amount of **287850 SEK**. Hence;

287850 SEK << 1173428 SEK

To be economically efficient the system should not cost more than 287850 SEK. So, despite the system is very good saving energy, but from the economic point of view is not viable.

7. ANALYSIS/DISCUSSION

As it has been shown in the results the total amount of energy saved and the consequently the total amount of money saved is quite significant, as it was expected. It can be appreciated that the system for some certain temperatures can save more than a 90% of energy. It is important to say that the system is not going to work if the temperature is higher than -4ºC. As it has been shown in the results the month with the highest amount of energy saved is January which has the lowest mean temperature, the amount of money saved is 7412 SEK that is an important amount of money for one month.

It has to take into account that in the beginning it was hoped a working range between -20ºC and 0ºC. After doing some calculations it has been demonstrated that it is not possible to improve the utilization range of -20ºC/-4ºC. It is like this due to dry coolers characteristics. It could be possible to think about select other kind of exchanger but, on the market, it is not possible to find exchangers with better advantages than dry coolers. So it can be assured that it is not possible to build a system of these characteristics with better utilization range until being available technology type on the market.

Initially it was hoped that the new system installation work without the conventional installation (i.e. not simultaneously generate cold). After the study of temperatures and ranges of use it has been deduced that if the system works alone, the operation hours per year would decrease drastically. It is therefore very important that both systems work in parallel. This fact does not increase the efficiency of the system but increases the operation hours per year, and therefore the total savings.

To carry out this type of work it is essential to drive the system through an electronic control. Managing a simultaneous operation of both systems does not require a very powerful control. A simple PLC control would be sufficient to do it, this does not increase costs over 5000 totals SEK.

The final COP of the system is 36, when the system was thought the COP that could be expected was higher than 36 but otherwise 36 is a very good result. It is not possible to find a refrigerated system with this COP. The usual COP founded in a refrigerated installation is around 3 so it is 12 times bigger. It is a significant result that can make possible the fact of using this system in the future. It can be added that the COP of the dry coolers is very high and finally decreases until 36 mainly because of the consumption of the pump.

Among the highlights selected components that dry coolers are of a size approximately 4 times greater than required if they are calculated under the design conditions. This phenomenon occurs because it is tried to draw as much energy as possible from both indoor air as the outdoor air. To absorb the minimum amount required of heat and also for an starting operation of the system at -4 º C are required an oversize heat exchangers.

From that it can be deduced that it has been followed a criteria across the project in which the aim has been to increase the annual working hours and also the total COP of the system. It has been moved to a secondary plane the size of the facility or space requirements in the areas of implantation. (On our case there are no problems with that)

It is important to say that the final cost of the whole system is really high and this fact becomes in a handicap for making it real applied in Willys.

Finally only commenting that contrary to a ventilation system, with a cooler dry air system the air is not injected directly from the outside, but through the system it is extracted the cold outside and it is got into the required areas. This does not change the relative humidity of the areas that should be refrigerated, that makes dry cooling system to be perfectly valid for rooms where moisture control is required.

8. CONCLUSIONS

Along the project it has been explained how the system works, the results obtained and the different parts of the system. So after study carefully the system and the different results obtained it is possible to assure different points.

The first point to comment is that the system studied has to be used in “cold countries”. For start the system working it has to appear a certain cold temperature (-4ºC) so for becoming a profitable system it has to work the highest time as possible between -20ºC and -4ºC. This reason makes the system being appropriated for countries with long and hard winters. The system must be placed on countries with special conditions.

Another relevant aspect of the project is that after making a study of how the system is going to work in Gävle, it has been deduced that the total amount of money that it is going to be saved per year is around 19200 SEK (About 14% of the total). This amount of money is quite significant because the system has a very long life, around 15 years. It is important to take on mind that the economy results changes on function of the year and the emplacement.

In addition to the point explained before, it can be said that this system is a friendly environmental system. The total amount of energy that is going to be saved using the system makes the company stops emitting the amount of pollutants that were emitted before to the atmosphere.

The system is not going to work at high pressure so the installation is not going to be a risk installation. In addition to that the refrigerant used is a mix of glycol with water. This refrigerant has the advantage that it is innocuous so it is not a harmful to the atmosphere. It has the advantage too of being cheap and it is easy to find in the refrigerant market. There is no change of state in the refrigerant (like in the refrigerant cycle) so it is no necessary too many security components. Also the components that are part of the whole system (Exchangers and pump) are components that have been used in the industry since years so the system is going to be reliable, that make the system reliable and secure.

For all those characteristics it can be concluded that although the system produces a very significant savings, the system is very simple if it is compared to the normal refrigeration cycle. Also the maintenance is practically zero.

It is important to mention that the final COP of the system is going to be unless 36, this means that the system has a COP twelve times higher than a good installation done in this industry field. It is concluded that although the system does not work the full year, it is interesting because it has a COP greater than any other conventional system, producing a saving in our case more or less 19200 [SEK/year].

On the other hand it has to be said that the total cost of the system is around 1.175.000 SEK so it is not going to be profitable to build it. The main problem that increases the final cost is the price of the dry coolers and the pipelines. It could decrease the total cost in a new installation, due to exchangers would be installed to do lower the pipe lengths.

Other important point directly linked with the feasibility is the low price of electricity. The electricity price in Sweden is very low, so As much as a lot of energy is saved, the cost savings is not representative.

9. SUGGESTIONS

This project explains how works the new system created but there are many things that can be studied and considered to apply them in this the new system for the future;

The first thing to say is that it could be important, applied to Sweden, to make a careful study of the outdoor temperatures in a northern place than Gävle, for example in Kiruna or Lulea. In these northern cities the system could be even more profitable.

Other suggestion that can be considered by Willys is that indoor areas where the food is storage can be in a range of temperature between 0-8ºC as the Swedish law says. Normally, in Willys, the temperature of these rooms is fixed so it can be considered the option of working in this range of temperature.

In case of the system stops working by a failure in it, it has been thought two ideas. The first one consist in having two pumps working in parallel so the total stop of the system is minimized in a 99.9 %. The second is the possibility of install outdoor two dry coolers and in this case if one fails then the other can continue working unless with the 50% of the power, however the COP is going to decrease a little.

Another aspect that can be study is how to apply the system to the open areas. In this project has been studied the closed areas and in them the amount of money that can be saved is quite significant so it could be interesting the fact of make a deep study for the open areas. This study would be another project in itself, there are available temperature data for different open areas heights because they were measured at the beginning of this project in order to estimate whether it was possible to implement the system in open areas.

Other field that can be studied and it has relation with the project is make a study of where can be placed this system. The system uses the outdoor air for cooling a place. It can be added to some places that work at high temperatures, so it could be not necessary to have very low temperatures outside. So with this idea the field of searching is bigger; for example slaughterhouses or specific machining lines automobile industry (sheet metal, and paint)

It is possible to add that it is going to be easier and cheaper to install the system when the place (for example a supermarket) is being built. Then the cost of the installation is going to be lower because the emplacement of outside dry coolers, pipe lines and pump are not conditioned at a building built.

Finally it can be studied carefully the market to find solutions for decreasing the final cost of the system which is the biggest handicap of the project.

It can also set up the possibility of installing the system in another country, where electricity prices were higher, which would produce greater savings.

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11. APPENDIX

ANNEX I; CALCULATIONS

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ANNEX I

CALCULATIONS

CALCULATION INDEX

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1 SITE AREAS

The present project is focused at cooling different refrigeration areas to save energy, so because of that it has been looked for the areas appropriated to implant the system.

As it is shown in the picture below, throughout all Willys building the next areas have been distinguished because of their working parameters;

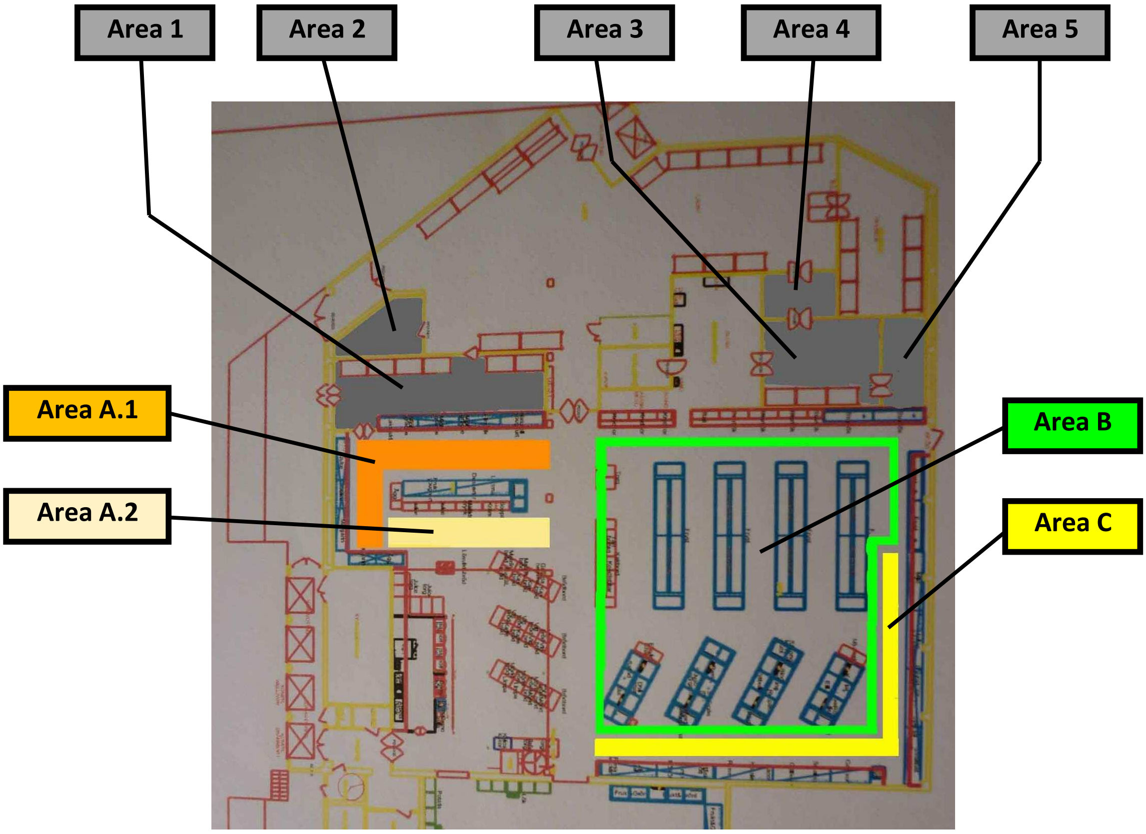


Figure 1; Willy´s map

1.1 Definition, thermal conditions of each area

1.1.1 Area 1

The Area 1 is a closed room and it is defined by three walls. The other wall has holes to communicate with the shelving refrigerated, which is located in the Area A.1, used to keep the packed milk products in a proper temperature. So the Area 1 and the Area A.1 are communicated by the shelving refrigerated. The location of this area is shown in the **figure 1.**

It exist a implicit problem with this room, the cause of the problem is the opened wall, one opened wall means that to get the objective temperature it will be necessary to deliver more cold than if the wall is closed. This problem will be solved on the next calculations.
The relativity humidity has been measured with an instrument which can make the measure in the place in a few seconds. This instrument is shown in **figure 2.**

Figure 2, Measure of the relativity humidity

The room characteristics are shown below;

Product stored	Packed milk products
Dimensions (m)	5x15x3.5
Volume (m^3)	262.5
Temperature range (ºC)	0-7
Relative humidity (%rH)	58.1

Table 1; Mean characteristics of the Area 1

1.1.2 Area 2

The Area 2 is a closed room perfectly defined by four walls. This area is particularly small and it can have some space problem to place the dry cooler. The location of the area it is shown in **picture 1**.

The room characteristics are shown below;

Product stored	Packed milk products
Dimensions (m)	Polygonal base
Volume (m^3)	70
Temperature range (ºC)	0-7
Relative humidity (%rH)	58.4

Table 2; Mean characteristics of the Area 2

1.1.3 Area 3

This area is defined by four walls and the room is communicated (as it is shown in **picture 1**) with Area 4 and Area 5 which has the same refrigerating system. The Area 4 is separated by a door but not the Area 5 which has no door of separation. In this Area the stored products are meat and cheese. There are three dry coolers and it is because this area is quite big in comparison to the other areas.

Product stored	meat
Dimensions (m)	6x7x 3.5
Volume (m^3)	147
Temperature range (ºC)	0-4
Relative humidity (%rH)	60.9

Table 3; Mean characteristics of the Area 3

1.1.4 ***Area 4***

This area is defined by four walls. This Area is small so there is one dry cooler. In this room is stored meat and the temperature is quite low, around 2ºC.

Product stored	Meat and cheese
Dimensions (m)	3x5x3.5
Volume (m^3)	52.5
Temperature range (ºC)	0-4
Relative humidity (%rH)	72.4

Table 4; Mean characteristics of the Area 4

1.1.5 Area 5

This Area is defined by four walls and it is located near to the Area 3 but without door. There are two dry coolers in this Area. The main thing to say about this room is that the half of the room normally is without material to store.

Product stored	meat
Dimensions (m)	3x6x3.5
Volume (m^3)	63
Temperature range (ºC)	0-4
Relative humidity (%rH)	70.1

Table 5; Mean characteristics of the Area 5

1.1.6 Open areas

Unlike others rooms, open areas have no walls which limited the geometry and, therefore, the air volume is too big to know with accuracy the air comportment. The consequence of having an opener area is that the study of the temperature is more difficult compared to the studies done in the closed areas. So a very important fact to know is the different temperature at different height levels.

Another thing to take into account is the customer comfort, it is very important to take care about the indoor temperature in the area because the customer who is buying there has to be in comfort.

The temperatures have been measured with a laser gun and a homemade instrument. The instrument consists in a metal bar of 4 meters of height and a wood square with thin plastic in the middle painted of black as it is shown in **figure 3**. It was decided to put this thin plastic because after testing different materials this one reaches the ambient temperature in a few seconds, so pointing there with the gun the measures have been taken. The laser gun was well calibrated and it was taken into account the emissivity of the plastic. To check if the laser gun was well calibrated it was taken as reference the temperature that the refrigerated showcases have in their thermometers. This fact is shown in **figure 4**.

Figure **3; Instrument used to take the measures**

Figure 4; Laser gun measuring the temperature inside a refrigerated showcase

1.1.6.1 Area A.1

The area A.1 is a zone close to Area 1 but without walls. In **figure 1** is shown as the area painted in orange. It is completely surrounded of refrigerated showcases with an internal temperature of 2ºC.

This factor of being surrounded by refrigerated showcases produce that the temperatures in this Area are quite low as it is shown in the **Figure 5** shown below.

Below is explained an extensive study to know the means temperatures on different levels; 1m, 2m, 3m.

As it is shown in the **Figure 5** the temperature increases as high we are but it is important to say that the most significant change is between the 1m and the 2m height. This change is because the refrigerated showcases have a great repercussion at the height of 1m.

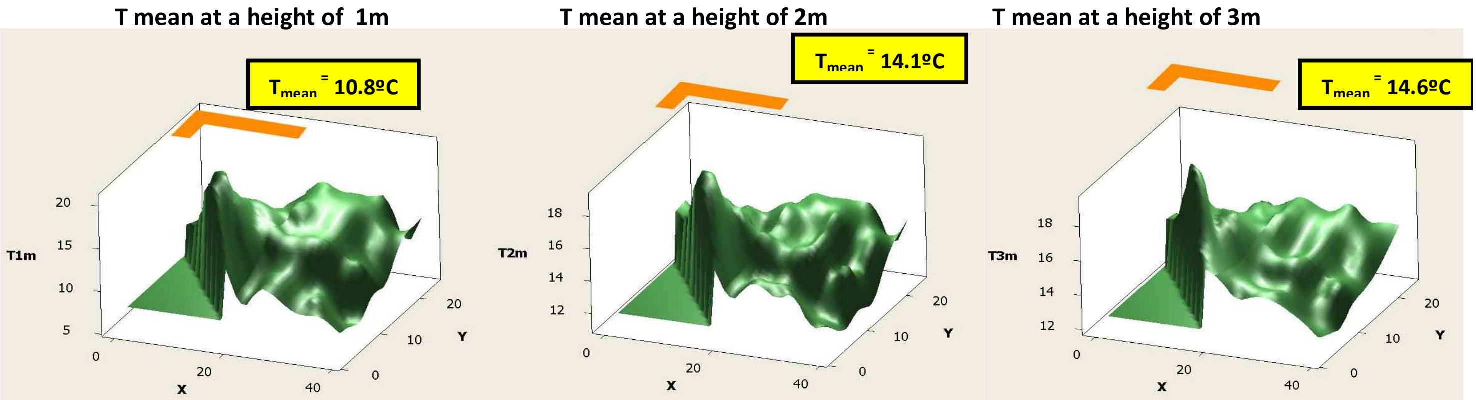


Figure 5, Temperature distribution in opened area A.1

Figure 6; Area A.1

1.1.6.2 Area A.2

This area is separated to the Area A.1 because as it is shown in **figure 1** there is only one refrigerated showcase closed to this Area. This Area is shown in the **figure 1** as the area painted in cream color. This means that the temperatures in this area are going to be higher as it is shown in the **figure 7**.

There is an important increase in the mean temperature from the Area A.1 to this Area A.2, the most significant change is that the temperature at 1m has increased almost 3ºC and this fact is because of the influence of the refrigerated showcases.

The difference of temperature between the 2m and 3m is not so significant but it is shown how the temperature changes as high as the measure is taken.

T mean at a height of 1m **T mean at a height of 2m** **T mean at a height of 3m**

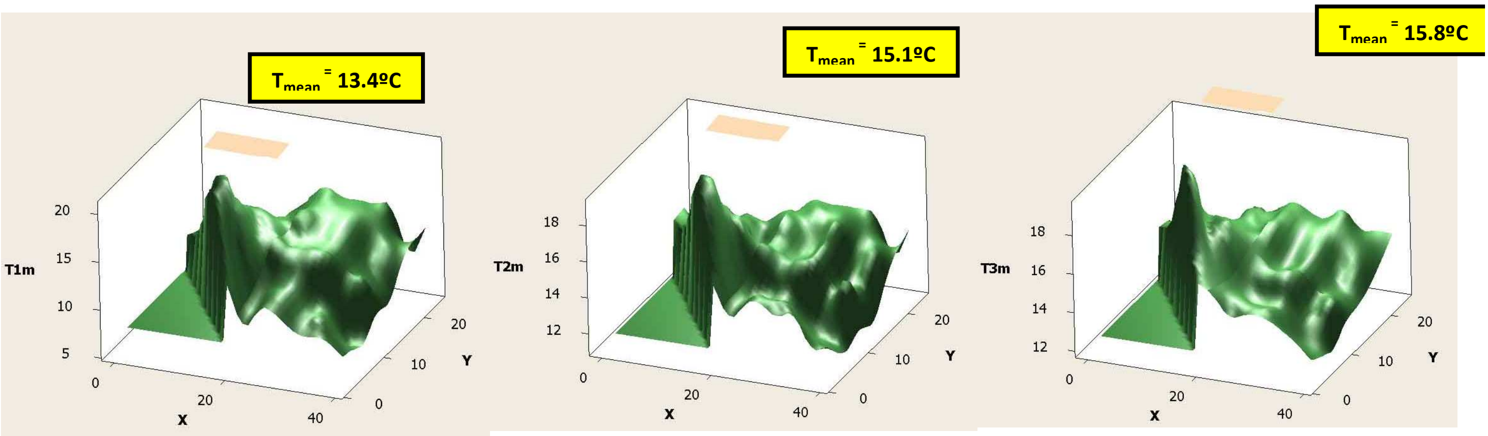


Figure 7; Temperature distribution in opened area A.2

Figure 8; Area A.2

1.1.6.3 Area B

This Area is quite huge in comparison to the others that have been shown before. As it is shown in the **figure 1** this area is surrounded by green color. This Area has been selected like this because the temperature is quite similar in all the points measured, and it is because there are no open refrigerated showcases. This Area has closed fridges so the cold of these fridges goes to the ambient when somebody opens them but it is not a long period of time that they are open.

Another important fact is that in the part which is more closed to the Area C the temperature is lower than the part that is closer to the Area 3 and Area 5 and this is because in the Area C there are open refrigerated showcases.

In this Area the changes in the temperature between the 1m and the 2m are quite similar to the ones that are shown in the Area A.2 with an increase of 2ºC (**see figure 9**).

Between the 2m and 3m the change of temperature is not so significant.

T mean at a height of 1m **T mean at a height of 2m** **T mean at a height of 3m**

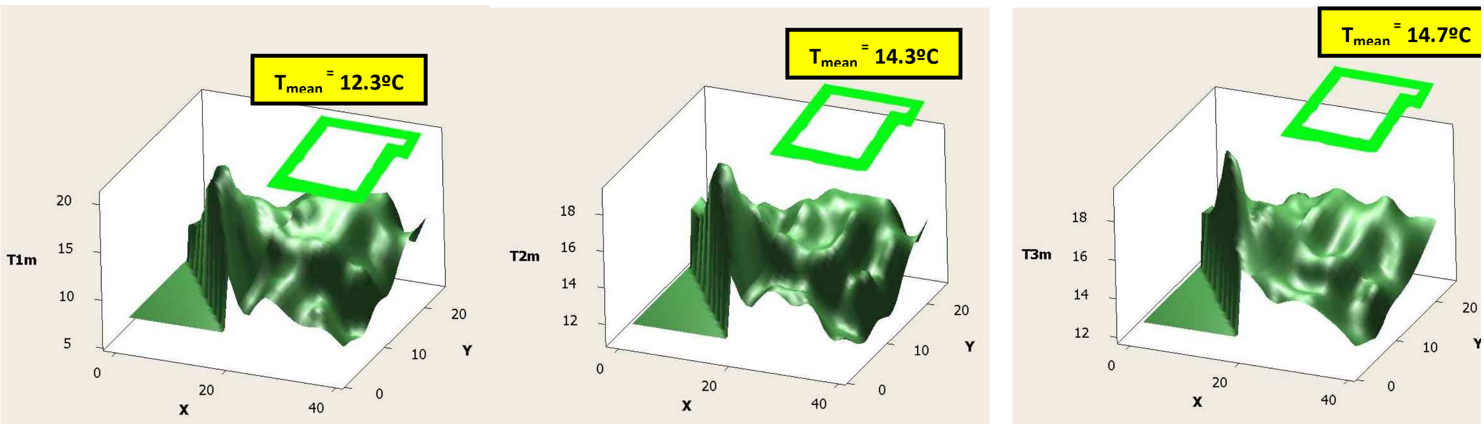


Figure 9; Temperature distribution in opened area B

Figure 10; Area B

1.1.6.4 Area C

This Area is closed to two modules of refrigerated showcases so it is influenced by the cold that is transmitted to the space which in this case is the Area C. This Area is shown in the **figure 1** as the area painted in yellow.

The mean temperature at 1m, as it is shown in **figure 11**, is the lowest found in the whole opened-Area. The change between the 1m and 2m is the highest found in the opened-Area so the influence of the refrigerated showcases is very important.

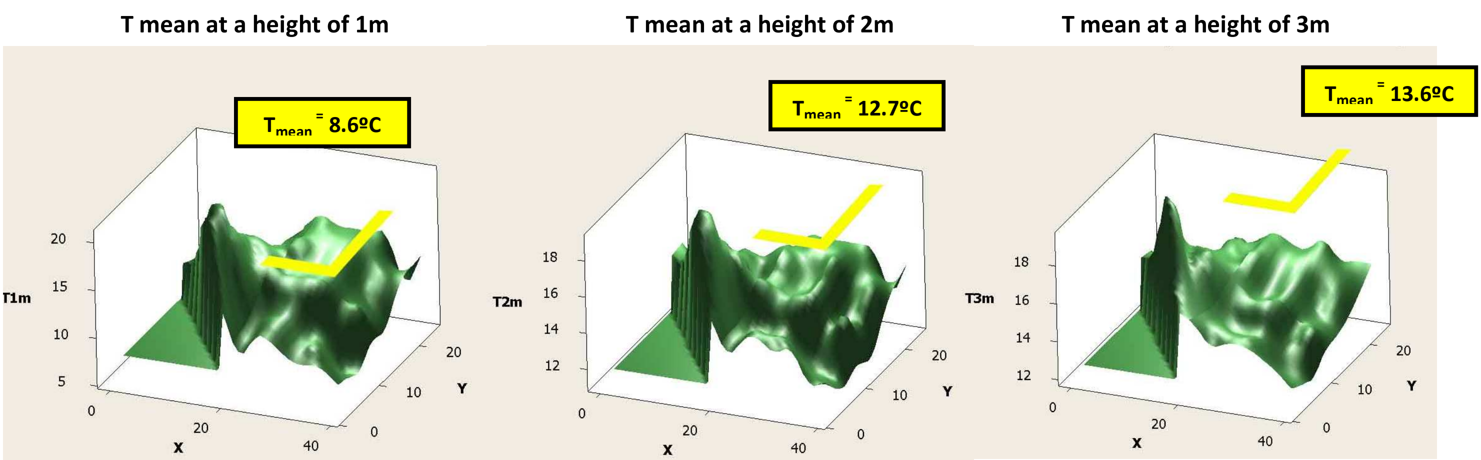


Figure 11; Temperature distribution in opened area C


Figure 12; Area C

1.2 Resume table

In the **tables 6 and 7** are shown the mean characteristic of the different areas studied above. The resume tables are divided in open areas and closed areas.
In **table 6** are shown the different temperatures taken at different heights. In **table 7** are shown the temperature, products storage there, dimensions, areas, volumes and relativity humidity.

- **OPENED-AREAS**

	T mean (1m) ºC	T mean (2m) ºC	T mean (3m) ºC
AREA A.1	10.8	14.1	14.6
AREA A.2	13.4	15.1	15.8
AREA B	12.3	14.3	14.7
AREA C	8.6	12.7	13.6

Table 6; Different temperatures comparison in opened areas

- **CLOSED AREAS**

	RANGE OF TEMPERATURES (ºC)	PRODUCTS	DIMENSIONS	VOLUME (m3)	RELATIVITY HUMIDITY (rh%)
AREA 1	0-7	Packed milk products	5x15x3.5	262,5	58.1
AREA 2	0-7	Cheese	Polygonal base	70	58.4
AREA 3	0-4	Meat	6x7x3.5	147	60.9
AREA 4	0-4	Meat and cheese	3x5x3.5	52,5	72.4
AREA 5	0-4	Meat	3x6x3.5	63	70.1

Table 7; Closed areas characteristics

For making the different measures it has been considered some factors that have influence in the measures taken. The measures have been taken with the influence of the lights inside the different areas, the open and close of the doors (in the closed areas) and the influence of the people which is more important in the open area.

2 TEMPERATURE CONDITIONS

Initially to start with the calculations it must be calculated the cool requirements on the places where the new dispositive will be installed.

On Willys are found some different refrigerated rooms and others open spaces which, at first appearance, seem nice places where the new cool system can be implement. It is important not to forget that the new study system has very different features that the commons systems that are normally used on cooling industries and, therefore, to continue with the idea of the new installation will be necessary to know the different data of every room or space to decide which are the best options where the system can be installed.

Another point to mention is that the energy balance of each room is not going to be calculated because it is not the objective of this project, so to satisfy the cooling demand on every refrigerated room it has been taken the air coolers that were installed previously.

Those air coolers have been placed according with the cool necessity, hence are going to be selected the dry coolers with the same cooling power or a little more to assure that the system can affront the cool demand.

To understand which is the best place where the system can be placed, previously it is necessary to know how the dry cooler system works.

2.1 Working needs

It is always necessary to take into account that the new device is only going to start working when there are some particular circumstances, besides, it should work on sometimes replacing the old cooling system.

On the other hand, the external weather influence strongly over the system functioning and due to that it has to be established the working ranges on which the efficiency of system is going to be optimized.

The system works in a range of temperatures, the outside temperature has to be unless 0ºC to start working and the lowest temperature that can afford the system is -20ºC. For terms of protection of the different elements of the installation the lowest temperature is -20ºC.

The system is going to work in an internal temperature range too. The internal temperature is going to be different in the different rooms. The system is going to work when the inside temperature reach a limit temperature and is going to be working until reach a minimum temperature which is going to vary depending of the room too.

Another important aspect to say is that the hot limit temperature in the inside of the room for our system (activate the system) has to be lower than the one that has the system placed nowadays in Willys. With this method the new system is going to work before the other system starts working.

2.2 Outside temperature variation

The outside temperature determines when the device is going to start working or stop working, so the first step to do is a study of the outside temperatures. After doing this first study it is going to be necessary to know the economy viability, utilization time, working temperature range and a lot of other important variables as switch on and switch off time.

It is not possible to foresee how the weather will be and this is an assumption that must be assumed in the project, however, it is possible to have an idea of how can the weather be doing a probability study based on others past years. As is shown below it has been done a temperature description though every month of the last year (2009);

JANUARY

JANUARY 2009			
Day	T (ºC)	Tmax (ºC)	Tmin (ºC)
1	-6	0	-12
2	-7	-4	-12
3	-8	-4	-11
4	-14	-10	-18
5	-10	-5	-17
6	-3	0	-6
7	-8	-2	-12
8	-5	3	-13
9	2	6	-4
10	1	2	-5
11	3	5	1
12	6	7	3
13	5	7	3
14	1	4	-2
15	-7	0	-11
16	-13	-7	-17
17	-9	-4	-16
18	-3	-2	-14
19	-1	0	-2
20	1	1	-1
21	1	2	0
22	-1	0	-4
23	-1	1	-3
24	0	1	-1
25	0	1	0
26	-1	1	-2
27	-1	-1	-2
28	-2	-1	-4
29	-4	-2	-5
30	-3	-2	-3
31	-4	-3	-5
Average	*-3*	*0*	*-6*

Table 8, January temperatures

Tmax	Tmin
6.9	-17.9

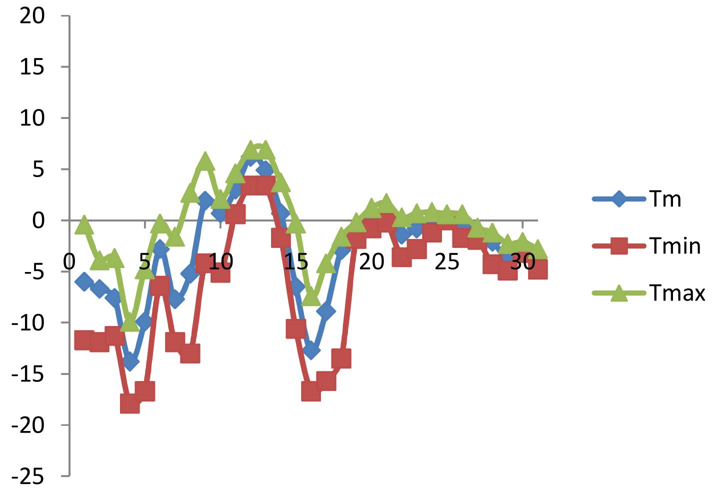


Figure 13; Temperature distribution graphic

It is possible to appreciate in the **figure 13** that the temperatures are mostly below of 0ºC across all month, but this data is not enough to make a decision that will be reliable.

It must be taking into account that the profitability of the system is hardly linked to the correct outside temperature selected on which the device switches on and switches off. To solve this problem has been done a graph where it can be estimated at the same time the total percentage in a month in which can appear a certain temperature vs outsides temperatures. With this graphic it is possible to quantify the days that are going to be able to use the new system.

For example as it is shown in the **figure 14** the possibility of having a mean temperature (black color) of -10ºC is approximately 10%.

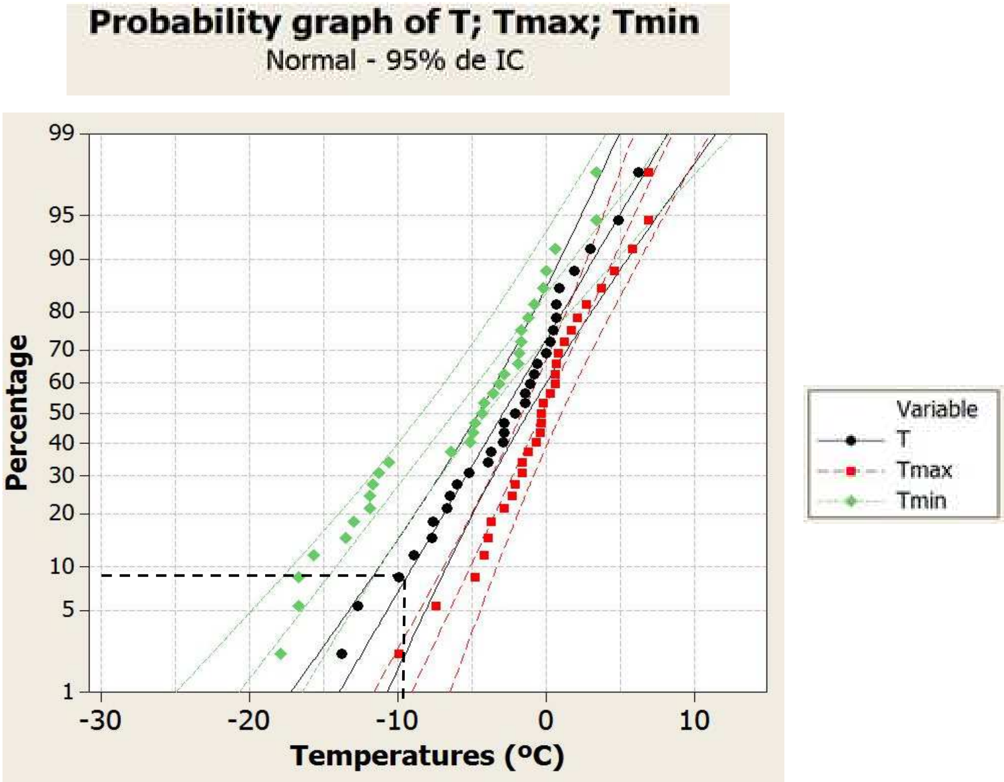


Figure 14; Probability graphic of appear a temperature

In **graphic 6**, the black points represent the mean temperatures throughout all January, the green points represent the mean of minimum temperatures and finally the red points are the ones which represent the mean of maximum temperatures.

The new cooling system will switch on between -5ºC and 0 ºC but it is still very soon to fix this value because before that it is necessary knowing which are the cooling losses along the pipes (Calculation are done in next points).

So to continue with the calculations and to assure that the project is going to be viable there have been taken some interesting values.

Temperature (ºC)	Probability Tmean	Probability Tmin.mean
0	70%	85%
-2	50%	75%
-5	30%	55%
-7	20%	40%
-20	0%	2%

Table 9, Temperature probability table

It has been deduced that if the design temperature to connect the system is 0ºC then the dry coolers will work the 70% of time in January as it is shown in **table 9**. (And it is important not to forget that this is a probability assumption and there can be some variations around the number)

On futures calculations these values are going to be used to find the best efficiency.

Another point to remark is that in all the piping system there are going to be found losses, these losses are traduced in an increment of the temperature of the refrigerant from the initial point to the final point. This fact has to be calculated for having an exactly idea of the temperature that has to be taken as the higher one to start working the system. All the calculations are done in next points.

It is important to say that the system is going to stop when the outside temperature is -20ºC or lower, this is because of terms of protection of the installation. The motor of the external installation is designed to work until temperatures of -26ºC so for terms of prevention the temperature adopted to stop the installation is -20ºC.

This fact represents approximately the 2% peak (is shown is **figure 14**) lower temperatures so it is not as important as the mean temperature but it has to be taken in care.

FEBRUARY

FEBRUARY 2009			
Day	T (ºC)	Tmax (ºC)	Tmin (ºC)
1	-6	-4	-9
2	-7	-3	-11
3	-5	-3	-9
4	-3	-1	-4
5	-2	-1	-3
6	-2	-1	-5
7	-5	-4	-5
8	-6	-4	-7
9	-7	-4	-10
10	-10	-6	-13
11	-9	-4	-14
12	-8	-2	-16
13	-12	-2	-19
14	-9	0	-17
15	-6	-3	-11
16	-6	-5	-7
17	-7	-5	-7
18	-7	-5	-10
19	-10	-5	-13
20	-9	-5	-14
21	-10	-4	-21
22	-1	0	-4
23	-2	0	-5
24	-3	2	-12
25	-3	2	-9
26	0	2	-6
27	-2	1	-7
28	-8	-2	-17
Average	*-6*	*-2*	*-10*

Table 10; February temperatures

Tmax	Tmin
2.2	-21.2

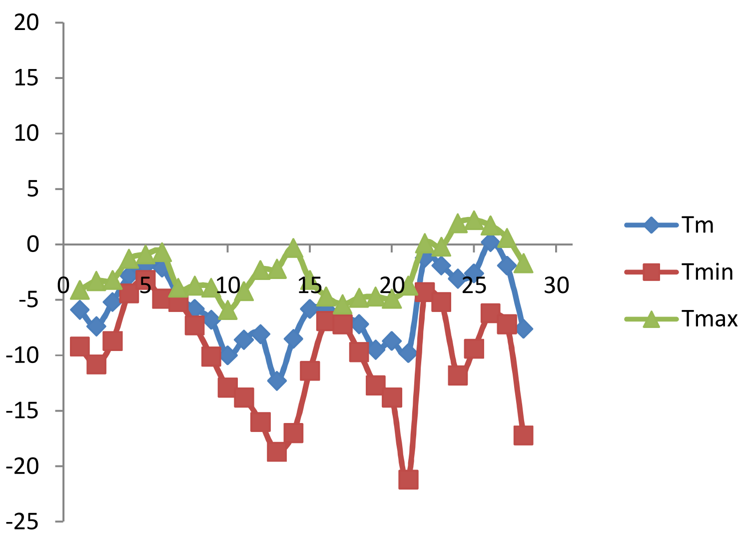


Figure 15; Temperature distribution graphic

As it is shown in the **table 10** and **figure 15** February has even lower temperatures than January, so it could be possible that in this month the new system can work the whole month. However there have to be taking into account the minimum temperatures because maybe someday are too low to let the system work for terms of security of the installation.

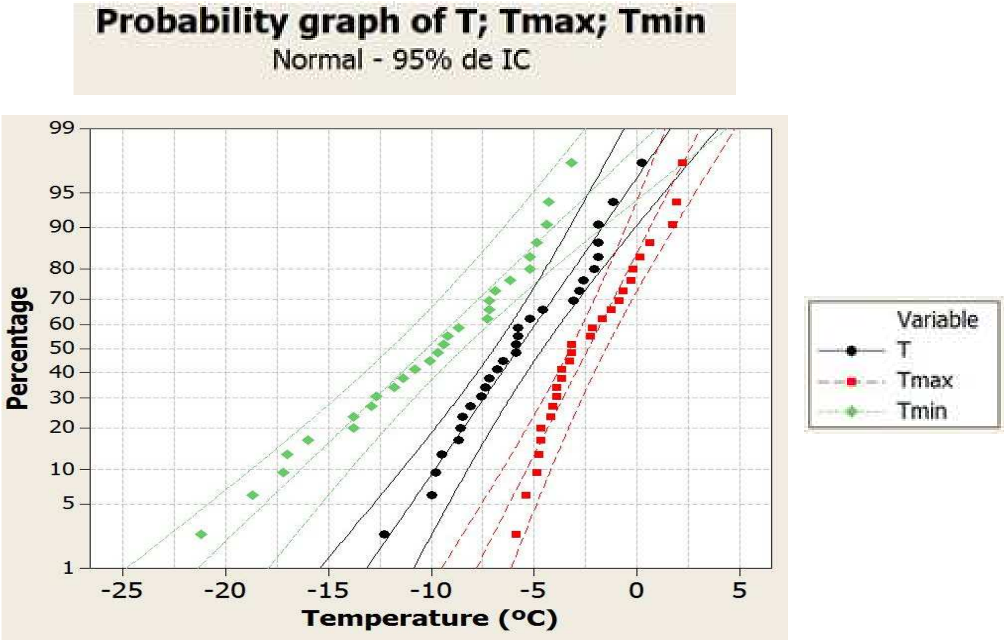


Figure 16; Probability graphic of appear a temperature

Temperature (ºC)	Probability Tmean	Probability Tmin.mean
0	97%	98%
-2	89%	95%
-5	60%	80%
-7	30%	60%
-20	0%	3%

Table 11; Temperature probability table

As it is shown in the **table 11** and **figure 16** above the percentage of time with a temperature of 0ºC is 97% of the time but it is important to take care of the peaks of -20ºC or less, during these temperatures the system is going to stop.

MARCH

MARCH 2009			
Day	T (ºC)	Tmax (ºC)	Tmin (ºC)
1	-9	-3	-22
2	0	0	-2
3	0	2	-6
4	1	3	-2
5	0	2	-2
6	0	1	-2
7	0	1	-2
8	0	2	-3
9	1	4	0
10	-2	0	-3
11	-3	-2	-4
12	-2	-1	-4
13	0	2	-2
14	1	3	-5
15	-2	3	-8
16	1	5	-3
17	2	7	-5
18	0	5	-5
19	-2	1	-7
20	3	8	-8
21	2	7	-4
22	-1	3	-5
23	-2	-1	-4
24	-7	-2	-17
25	-7	2	-19
26	-3	2	-11
27	-2	2	-15
28	1	3	-1
29	2	6	0
30	3	6	-2
31	4	9	-3
Average	*-1*	*3*	*-6*

Table 12; March temperatures

Tmax	Tmin
9.2	-21.7

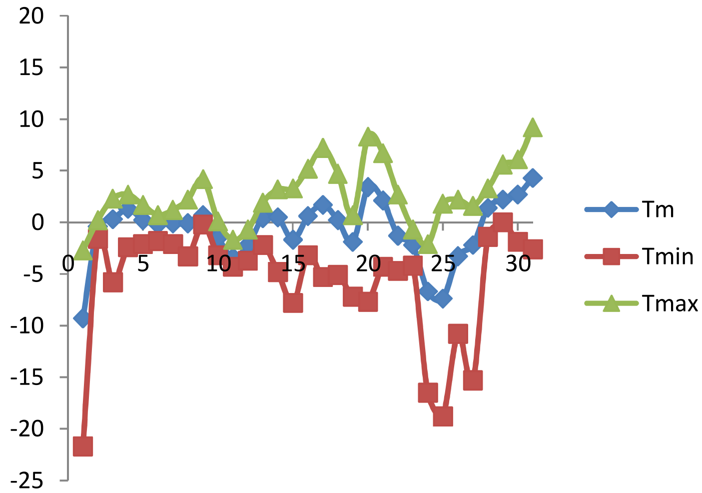


Figure 17; Temperature distribution graphic

As it is shown in the graphics March is another month were the new system can work, in this month the percentage is around the 60% if the selected temperature were 0ºC, but this will be calculated later.

March is a month where the temperature starts to increase to in the next months the percentage will go down.

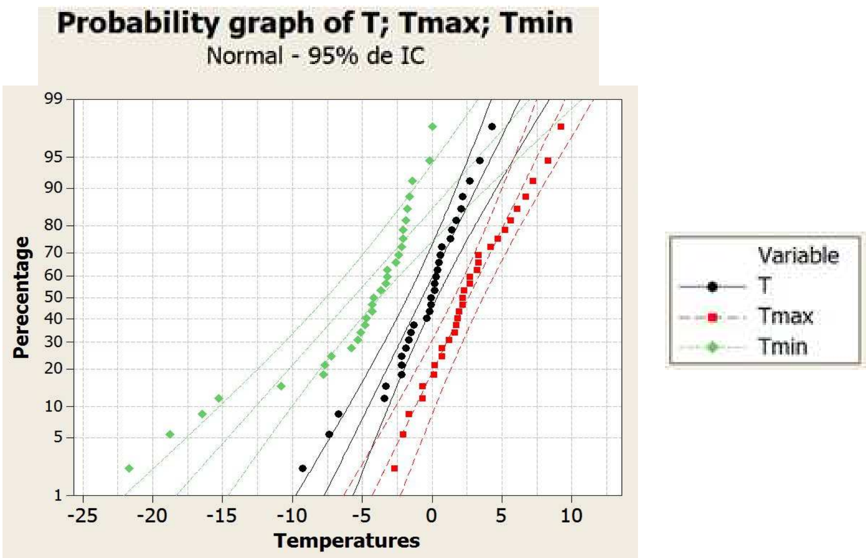


Figure 18; Probability graphic of appear a temperature

Temperature (ºC)	Probability Tmean	Probability Tmin.mean
0	60%	75%
-2	30%	70%
-5	8%	40%
-7	1%	20%
-20	0%	3%

Table 12; Temperature probability table

As it is said before the installation is going to stop when the temperature reach to -20ºC.

APRIL

APRIL 2009			
Dia	T	Tmax	Tm
1	3	8	-3
2	7	12	-2
3	6	14	0
4	6	14	-1
5	2	8	-2
6	6	12	-2
7	0	5	-1
8	2	5	0
9	4	8	1
10	7	15	1
11	5	12	-1
12	9	17	-2
13	5	14	0
14	4	9	-1
15	3	6	1
16	5	9	1
17	6	12	1
18	1	8	-3
19	6	14	-3
20	2	11	-2
21	3	9	-4
22	5	11	-1
23	7	14	-1
24	11	19	-1
25	14	20	7
26	13	19	6
27	15	22	6
28	11	19	4
29	7	12	0
30	6	12	-1
Average	*6*	*12*	*0*

Table 13; April temperatures

Tmax	Tmin
9.2	-21.7

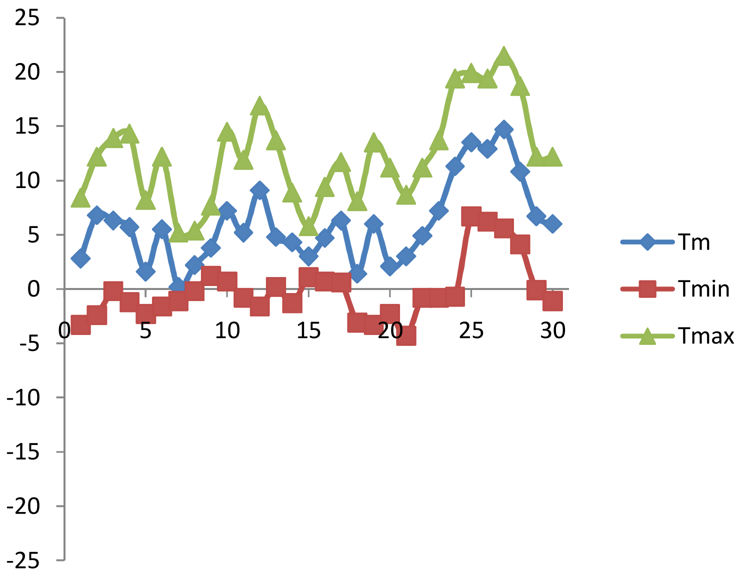


Figure 19; Temperature distribution graphic

Here in April the main percentage shown in **table 15** is lower and it is around 5% but as it is shown in the **figure 19,** there are minimum peaks that can be very useful for the system.

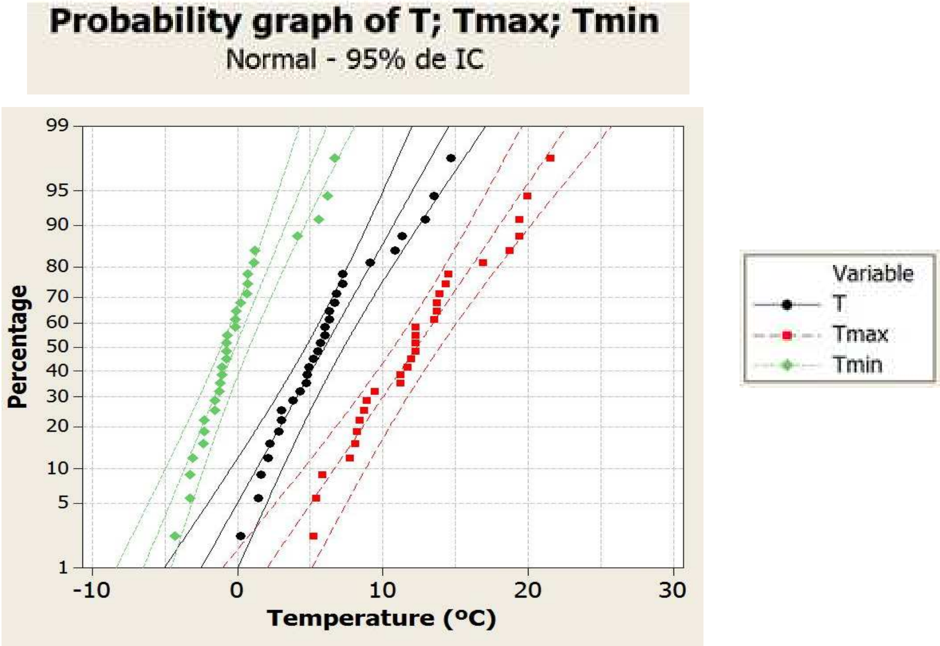


Figure 20; Probability graphic of appear a temperature

Temperature (ºC)	Probability Tmean	Probability Tmin.mean
0	5%	40%
-2	1%	20%
-5	0%	3%
-7	0%	0%
-20	0%	0%

Table 15; Temperature probability table

As it is shown on the last table the probability of appearing a Tmín < -2ºC is about 20%, so the system will work sometimes but it is not possible quantify so much. On the other hand there are not already temperatures below -20ºC so the system is not going to lose time due to protection stops.

OCTOBER

OCTOBER 2009			
	T	TM	Tm
1	3	9	-4
2	4	10	-1
3	3	8	-4
4	6	8	4
5	5	10	0
6	4	8	-2
7	10	13	5
8	5	10	0
9	3	9	-1
10	1	9	-3
11	1	6	-5
12	3	6	1
13	2	4	-1
14	1	4	-2
15	2	9	-4
16	5	6	-1
17	4	6	3
18	2	5	-1
19	2	5	-1
20	2	9	-2
21	2	5	-1
22	6	7	4
23	4	6	3
24	4	5	3
25	6	6	4
26	6	7	6
27	6	7	4
28	3	6	-1
29	2	5	-1
30	2	6	-1
31	3	6	-1
Average	*4*	*7*	*0*

Table 16; October temperatures

Tmax	Tmin
12.8	-4.6

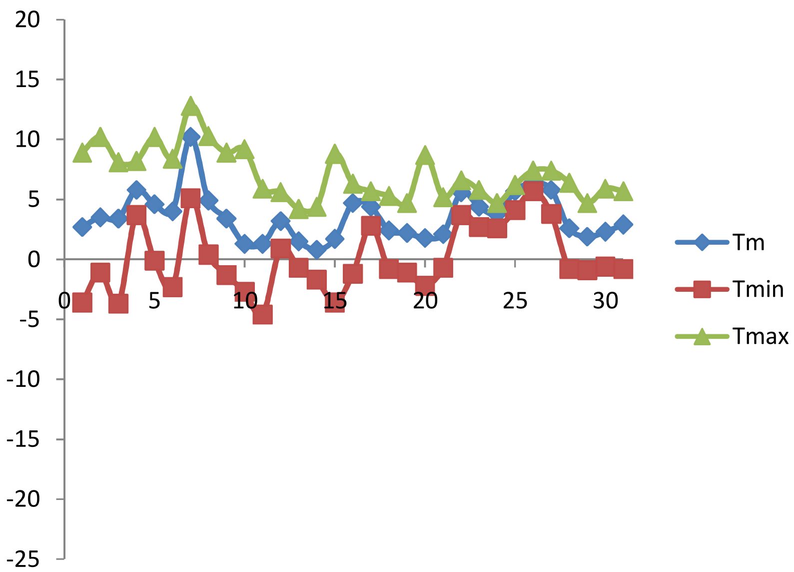


Figure 21; Temperature distribution graphic

October is a month which is quite similar to April, the main percentage is not high but there are some useful minimum peaks for the system as is shown in **figure 21**.

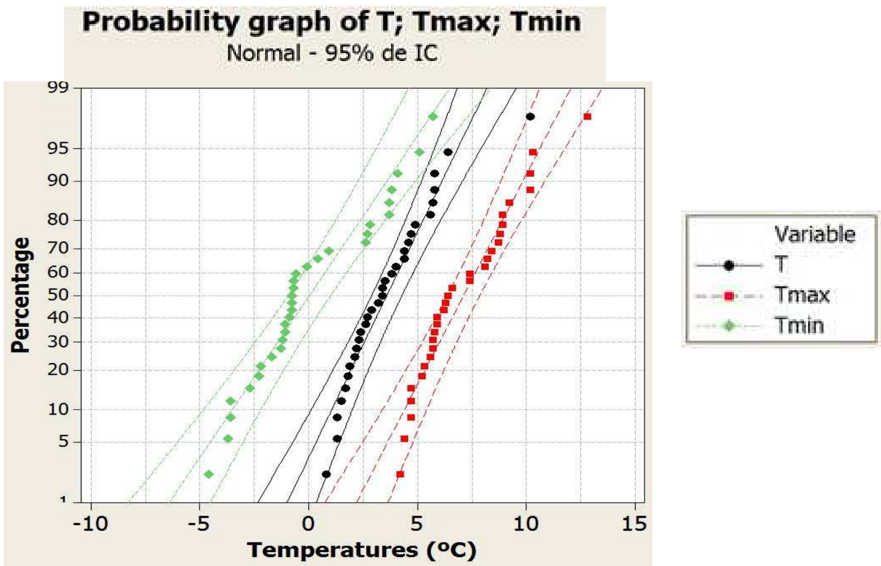


Figure 22; Probability graphic of appear a temperature

Temperature (ºC)	Probability Tmean	Probability Tmin.mean
0	4%	50%
-2	0%	18%
-5	0%	3%
-7	0%	0%
-20	0%	0%

Table 17; Temperature probability table

NOVEMBER

OCTOBER 2009			
	T	TM	Tm
1	3	9	-4
2	4	10	-1
3	3	8	-4
4	6	8	4
5	5	10	0
6	4	8	-2
7	10	13	5
8	5	10	0
9	3	9	-1
10	1	9	-3
11	1	6	-5
12	3	6	1
13	2	4	-1
14	1	4	-2
15	2	9	-4
16	5	6	-1
17	4	6	3
18	2	5	-1
19	2	5	-1
20	2	9	-2
21	2	5	-1
22	6	7	4
23	4	6	3
24	4	5	3
25	6	6	4
26	6	7	6
27	6	7	4
28	3	6	-1
29	2	5	-1
30	2	6	-1
31	3	6	-1
Average	*4*	*7*	*0*

Table 18; November temperatures

Tmax	Tmin
11.4	-2.2

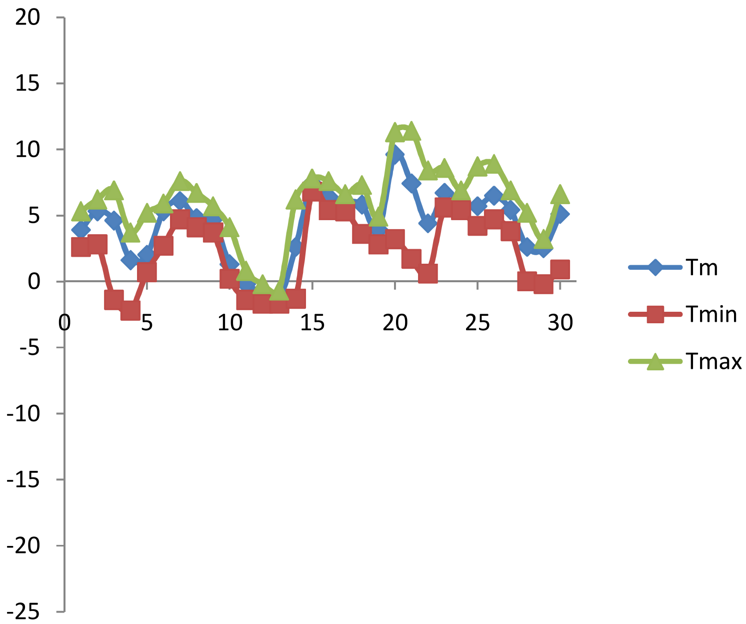


Figure 23; Temperature distribution graphic

It happens something special with November and it is that the minimum peaks, shown in **figure 23**, are less than the ones of October, and there is a significant change between November and December, so a thing to comment is that maybe is not a representative November and the normal one has more minimum peaks.

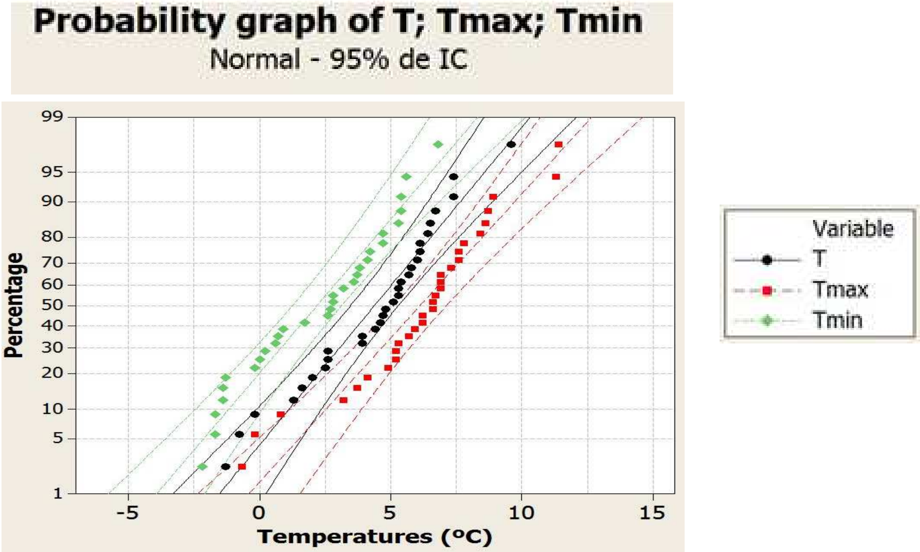


Figure 24; Probability graphic of appear a temperature

Temperature (ºC)	Probability Tmean	Probability Tmin.mean
0	5%	20%
-2	1%	5%
-5	0%	0%
-7	0%	0%
-20	0%	0%

Table 19; Temperature probability table

DECEMBER

DECEMBER 2009			
	T	TM	Tm
1	0	6	-4
2	-5	-2	-10
3	-3	0	-10
4	0	2	-3
5	3	5	0
6	3	5	2
7	3	4	2
8	2	3	1
9	2	3	1
10	2	2	1
11	0	2	-2
12	0	2	-2
13	-1	0	-4
14	-3	0	-5
15	-3	-3	-5
16	-3	-2	-4
17	-5	-4	-8
18	-14	-4	-18
19	-13	-5	-19
20	-4	-3	-13
21	-13	-5	-19
22	-13	-9	-16
23	-7	-5	-16
24	-6	-3	-13
25	-1	1	-4
26	-2	0	-4
27	-11	-3	-17
28	-8	-6	-10
29	-16	-8	-22
30	-20	-11	-23
31	-11	-8	-22
Average	*-5*	*-1*	*-9*

Tmax	Tmin
6.1	-23.3

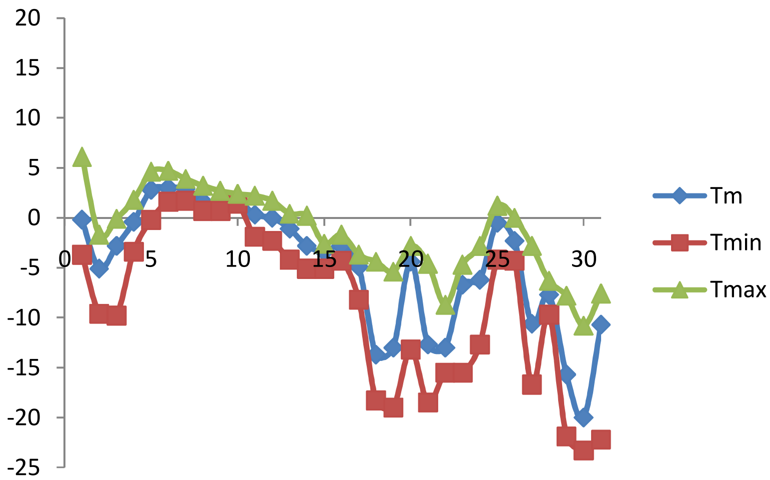


Table 20; December temperatures **Figure 25; Temperature distribution graphic**

As it is shown in the **figure 26** and **table 21**, here in December the percentage of operative days for having a temperature of unless 0ºC is high again with a main percentage of 80%. So it is possible to assume that the system could be working a lot of hours of the month.

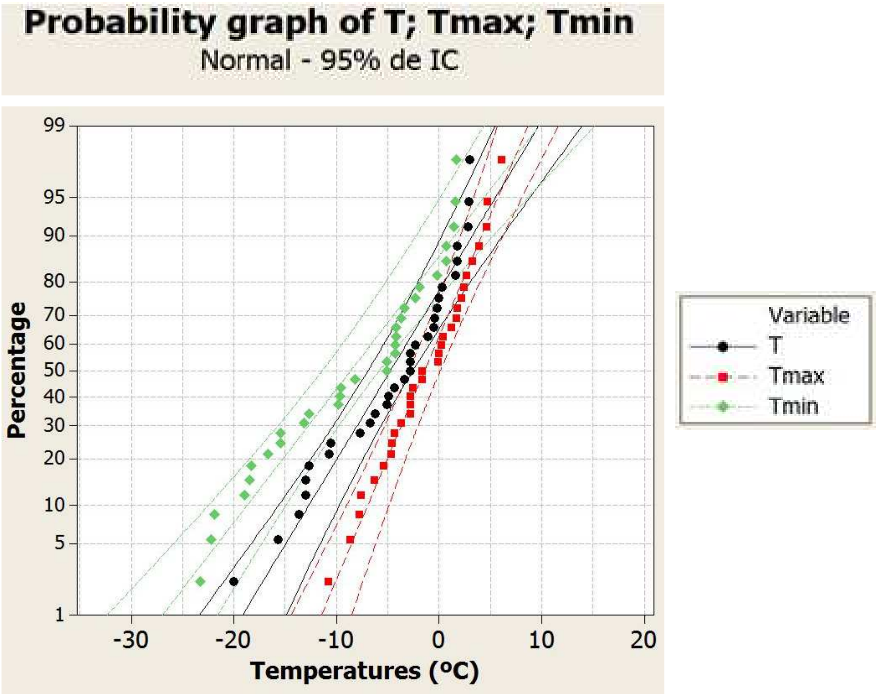


Figure 26; Probability graphic of appear a temperature

Temperature (ºC)	Probability Tmean	Probability Tmin.mean
0	80%	90%
-2	60%	80%
-5	50%	70%
-7	30%	60%
-20	0%	7%

Table 21; Temperature probability table

As it is shown in **table 21** this is the month with the highest percentage of lowest peak temperatures so the system is going to stop during the period of these peaks.

2.2.1 Resume table. Outsides characteristics.

	JANUARY			FEBRUARY			MARCH			APRIL		
T (ºC)	% Tm	%Tmin	Time	%Tm	%Tmin	Time	%Tm	%Tmin	Time	%Tm	%Tmin	Time
0	70%	85%	521h	97%	98%	698h	60%	75%	447h	5%	40%	36h
-2	50%	75%	372h	89%	95%	641h	30%	70%	223h	1%	20%	8h
-5	30%	55%	223h	60%	80%	432h	8%	40%	60h	0%	3%	h
-7	20%	40%	149h	30%	60%	216h	1%	20%	8h	0%	0%	h
-20	0%	2%	0	0%	3%	0h	0%	3%	0h	0%	0%	h

	OCTOBER			NOVEMBER			DECEMBER		
T (ºC)	% Tm	%Tmin	Time	%Tm	%Tmin	Time	%Tm	%Tmin	Time
0	4%	50%	30h	5%	20%	36h	80%	90%	595h
-2	0%	18%	0h	1%	5%	8h	60%	80%	447h
-5	0%	3%	0h	0%	0%	1h	50%	70%	372h
-7	0%	0%	0h	0%	0%	1h	30%	60%	224h
-20	0%	0%	0h	0%	0%	1h	0%	7%	0h

Table 22; Resume temperature table

On the **table 22** it can be seen a data resume of outside temperatures and the utilization time measured in hours. It has been highlighted on red color the months where it can appear temperatures peaks below of -20ºC and on green color the data on which the utilization time is more interesting due to long time of use.

TOTAL UTILIZATION HOURS			
	0ºC	-2ºC	-5ºC
January	521h	372h	223h
February	698h	641h	432h
March	447h	223h	60h
April	36h	8h	0h
October	30h	0h	0h
November	36h	8h	1h
December	595h	447h	372h
TOTAL	***2363h***	***1699h***	***1088h***

Table 23, Total utilization hours

In view of the results it is sure that the temperature that is going to give the longer utilization time is 0ºC (2363h, about 27% total time of year) but it is necessary to take into account that the system can't give out inside cool rooms a temperature lesser than the external one plus 5ºC as it is shown in the calculations done in the next parts. That is a good reason to consider the utilization of both old and new system coordinate to take the new system advantages.

2.3 Working range. Indoor temperature

On the building selected to install the new cooling system there are two areas very well differentiated, open areas and closed areas.

A) Open areas

The open areas are; Area A.1, Area A.2, Area B and Area C as are shown in **picture 1**. The problem with this kind of areas is that it is very difficult to assure a good economic result because there are quite uncertain variables which each one of them is object of a different project, the most important are;

· *Huge total volume*; all areas are closed and very diffusedly differenced. So it appears effects as a strong temperature gradient on only three meters or in cold zones and heat zones which are susceptible of change their temperatures due to, for instance, convection of air and heat focus (customers, lights, so on). If a dry cooler is placed on this area there is no guaranty of saving energy. This is because the exchanger out jet itself generating a hard convection and a complicated mix between outside air and internal air that difficult too much the calculations.

· *Customer area*; Effect over customer is maybe the best option to rule out the purpose on this area kind. To get a significant saving of energy it is necessary to guarantee a temperature around freezer between 0ºC and 2ºC for long time. There are not normalized PPD and PDM graphs for this case specifically but actually the temperature of open areas at 1m is more on less between 7ºC and 10ºC (see **figure 27**) .So it may be sure that if the temperature is down from 10ºC to 2ºC it might be a suggestive motive that affect negatively at customer.

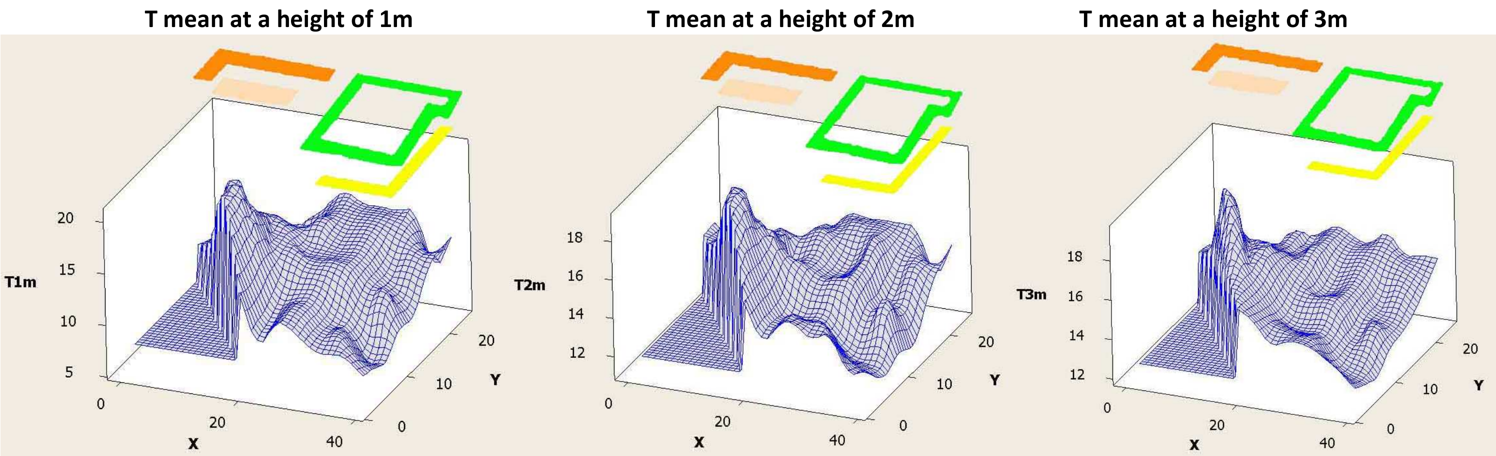


Figure 27, Temperature distribution in open areas

B) Closed areas

The closed areas are; Area 1, Area 2, Area 3, Area 4 and Area 5. In these areas the problem of no quantifying the save of energy disappears. This assumption is because of some factors:

-Range of temperature: The range of temperature used in these Areas is the one that permits to the system works and cool the Area saving energy stopping during some periods (depending of the external temperatures) the other system installed in Willys. This areas can work at this temperatures because are not open to the clients.

-Small volume: the other factor which permits calculate the amount of energy saved is that these Areas are small closed Areas so there are not too many ambient interferences in the volume of the different areas.

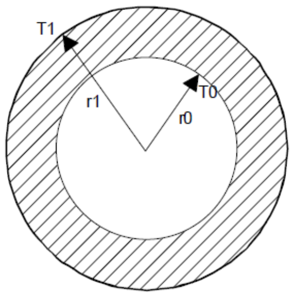
3 PIPE LOSSES

3.1 Heat losses

It is logical to think that it is not possible to place the external exchanger and the indoor one with a huge distance. If not it is going to appear big heat losses on the long travel of refrigerant from one exchanger to the other one. So it will never get the necessary temperature to get to the objective that is purposed in the project.

With this little explanation it is introduced the concept of maximum placement distance. To know the maximum distance where the external exchanger could be installed it must be known the Celsius degrees that drop across a lineal meter of pipe.

The appropriate equation is;



$$Qr = \frac{(T0 - T1)}{\frac{\ln\left(\frac{r1}{r0}\right)}{\lambda \cdot 2 \cdot \pi \cdot L}} \quad (1)$$

Picture 28; Pipe diameters

Equation 1; Heat flow

Q_r = Heat flow (W)
λ= thermal conductivity (W/m.K)
r_0 = inner radius (m)
r_1 = outer radius (m)
T_0 = inner temperature (K)
T_1 = outer temperature (K)
L = length of the pipe (m)

As it is possible to see in the **equation 1** the pipe radius affect indirectly to the pipes heat losses, besides it is logic to deduce that if the pipe radius rises, the heat lost is going to decrease.

With this equation it is obtained the amount of heat lost per meter but it is more interesting than that to know how many degrees drop per meter, so;

Q_{lost} = m·c_p·(Tb-Ta) → $\Delta T = \frac{Qlost \cdot 3{,}6}{m \cdot cp}$ [W/m] (2)

Equation 2; Temperature difference W/m

Q_{lost} = Heat flow lost (W/m)
m = mass flow of refrigerant (kg/h)
c_p = specific heat capacity (kJ/kg·K)
T_a = temperature on A point (K)
T_b = temperature on B point (K)

1→Pipe
2→Polyurethane insulation
3→Frame

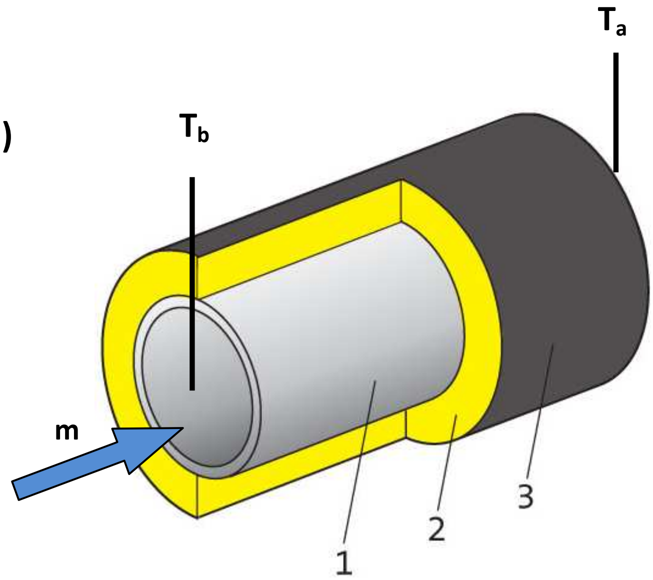


Figure 29; Pipe components

If it replaces the **equation 2** on the **equation 1**, it is obtained a single equation, **equation 3**, with all the variables;

$$\Delta T = \frac{(T0 - T1) \cdot 3{,}6}{m \cdot cp \cdot \frac{\ln\left(\frac{r1}{r0}\right)}{\lambda \cdot 2 \cdot \pi \cdot L}} \quad \left[\frac{^{\circ}C}{m}\right]$$

Equation 3; Temperature difference

As it is shown in the equation 3 the only parameters that can vary are the ones highlighted in red, the insulation thickness (r1) and the thermal conductivity of insulation (λ).

Checking the **table 24** which is shown below it is possible to see that the polyurethane is the best option, therefore there is now only one variable, the thickness.

To select the insulation material it is necessary to know all the material existent on the market and theirs characteristics;

THERMICS INSULATION MATERIALS	Apparent density kg/m^3	Thermal conductivity λ *kcal/hm °C*	Thermal conductivity λ *W/m °C*
Glass fiber:			
Type I	10 – 18	0,038	0,044
Type II	19 – 30	0,032	0,037
Type III	31 – 45	0,029	0,034
Type IV	46 – 65	0,028	0,033
Type V	66 – 90	0,028	0,033
Type VI	91	0,031	0,036
Mineral wool:			
Type I	30 – 50	0,036	0,042
Type II	51 – 70	0,034	0,040
Type III	71 – 90	0,033	0,038
Type IV	91 – 120	0,033	0,038
Type V	121 – 150	0,033	0,038
- Expanded perlite	130	0,040	0,047
Expanded polystyrene UNE 53.310			
Type I	10	0,049	0,057
Type II	12	0,038	0,044
Type III	15	0,032	0,037
Type IV	20	0,029	0,034
Type V	25	0,028	0,033
-Extruded polystyrene	33	0,028	0,033
-Crosslinked polyethylene	30	0,033	0,038
-Polisocianurato foam	35	0,022	0,026
Polyurethane forming foam			
Type I	32	0,020	0,023
Type II	35	0,020	0,023
Type III	40	0,020	0,023
Type IV	80	0,034	0,040
In situ applied polyurethane foam			
Type I	35	0,020	0,023
Type II	40	0,02	0,023
- Urea formaldehyde foam	10 – 12	0,029	0,034
-Urea formaldehyde foam	12 – 14	0,030	0,035
-Vermiculite expanded	120	0,030	0,035
-Cell-glass	160	0,038	0,044

Table 24; Types of insulation and its characteristics

The refrigerant flux (m), specific heat (cp) and pipe radius are fixed, but it is necessary to distinguish the worst cases if it is taken into account the external and the internal pipe temperatures;

A) *When the temperature difference between T0 and T1 is maximum*. This case appears if the outside temperature is the minimum namely before the security stop of the system, approximately -20ºC.
So replacing the data in the **equation 3**;

T0=15 [ºC]
T1= -20 [ºC]
m= 132.7 (m3/h)= $132{,}7 \cdot 10^3$ [kg/h]
cp= 4,2 [kJ/kg·K]
λ= 0.023 [W/m°C]
L= 1 [m]
r0= 100mm = 0,1 [m]

$$\Delta T = \frac{(15-(-20)) \cdot 3{,}6}{m \cdot cp \cdot \frac{\ln\left(\frac{r1}{0{,}1}\right)}{\lambda \cdot 2 \cdot \pi \cdot L}} = \frac{126}{88{,}7 \cdot 10^3 \cdot \frac{\ln\left(\frac{r1}{0{,}1}\right)}{\lambda}} \left[\frac{ºC}{m}\right]$$

As it is demonstrated above due to the very high refrigerant flux it is obtained a very low ΔT, so it has been checked the results to some different thickness;

Thickness [mm]	ΔT [ºC/m]
0	$3 \cdot 10^{-3}$
50	$8{,}1 \cdot 10^{-5}$
100	$4{,}7 \cdot 10^{-5}$
150	$3{,}6 \cdot 10^{-5}$

Table 25; Temperature variation per meter depending of the insulation thickness for a Tout of -20ºC

As it is possible to see in **table 25** it is totally negligible the dropping of temperature in the pipe.

B) *When the temperature difference between T0 and T1 is minimum*. This case appears when the outside temperature is maximum so just before the system stops working, approximately -2ºC.

So replacing the parameters in the **equation 3**;

T0=15 [ºC]
T1= -2 [ºC]
m= 132.7 (m3/h)= $132,7\cdot10^{3}$ [kg/h]
cp= 4,2 [kJ/kg·K]
λ= 0.023 [W/m°C]
L= 1 [m]
r0= 100mm = 0,1 [m]

$$\Delta T=\frac{(15-(-2))\cdot 3,6}{m\cdot cp\cdot\frac{\ln\left(\frac{r1}{0,1}\right)}{\lambda\cdot 2\cdot\pi\cdot L}}=\frac{61.2}{88,7\cdot 10^{3}\cdot\frac{\ln\left(\frac{r1}{0,1}\right)}{\lambda}}\left[\frac{ºC}{m}\right]$$

Like on the calculation done to the case presented before, due to a very high refrigerant flux it is obtained a ΔT that is going to be very low, so it has been checked the results to some different thickness;

Thickness [mm]	ΔT [ºC/m]
0	$1,5\cdot10^{-3}$
50	$3,9\cdot10^{-5}$
100	$2,3\cdot10^{-5}$
150	$1,7\cdot10^{-5}$

Table 26; Temperature variation per meter depending of the insulation thickness for a Tout of -2ºC

The risk of this case was that the dropping temperature could be high. The temperature that appears in the inlet of the exchangers of the different areas could be not useful for cold down the area. If that occurs all calculation would be wrong and it would be not possible to save energy as much as it was thought.

To demonstrate that there is no risk to have a high difference of temperature between the inlet temperature on the external exchanger and the inlet temperature on the indoor exchanger it is shown below, in **figure 30**, the possible maximums longitudes of pipes;

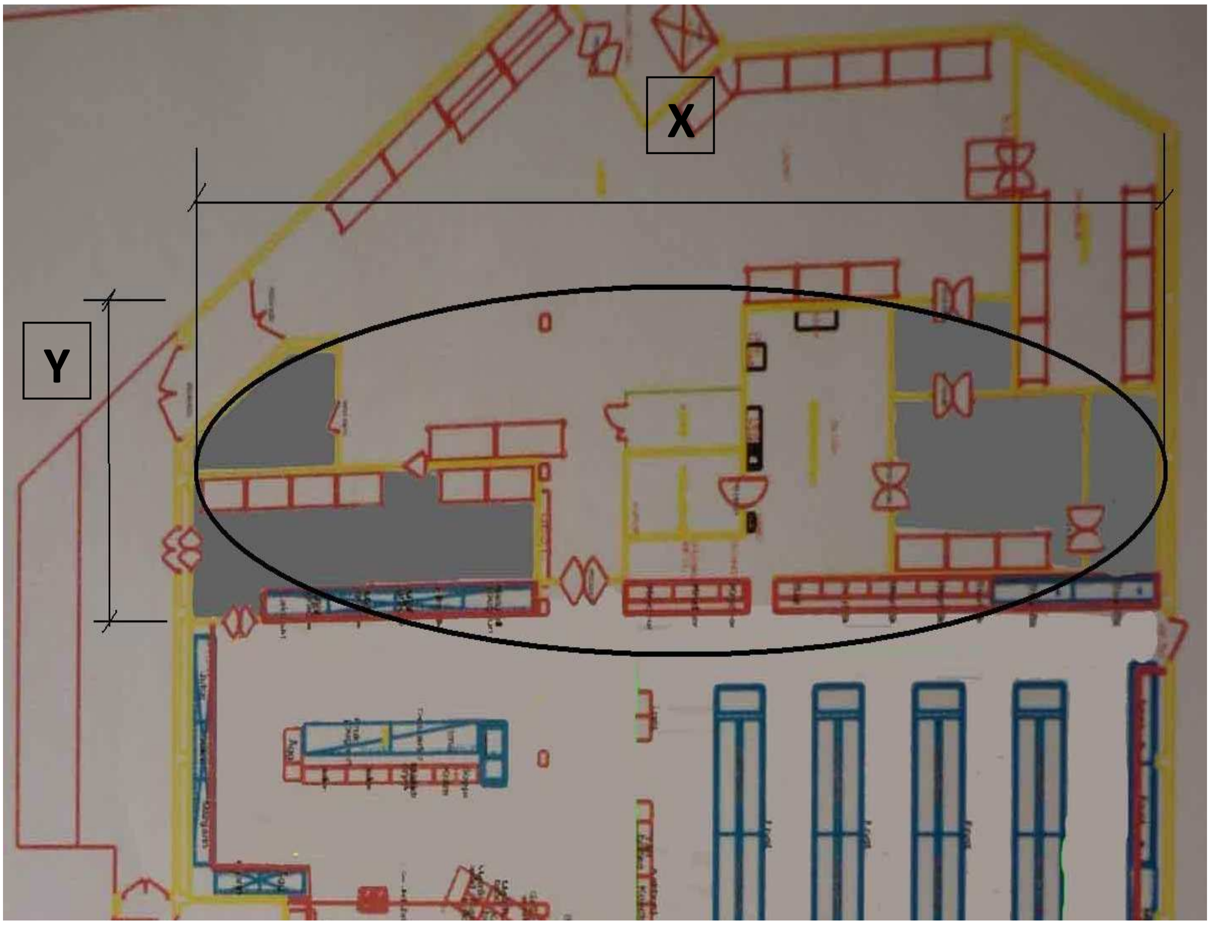


Figure 30; Area for placing the system

X= 40 m / Y= 15 m

As it is shown in **figure 30** the length of the pipes can be a maximum of 40m in the axis X and a maximum of 15m in the axis Y. So it doesn´t matter where the pumps room and where the heat exchangers are going to be installed because the maximum length is going to be 55m. As it has been calculated before the losses in 55m are not going to be high, so it is not a problem the longitude of the pipes.

4 INTERNAL EXCHANGER

4.1 Introduction

To solve the problem of “extracting” the outside cold and to deliver it inside of refrigeration rooms it is important to select a kind of exchanger. It has to have the best relation between the cool air flux deliver and the electricity consume of all devices that the exchanger utilizes (Like fans and pumps). To take the best selection it must be known all the exchangers models that the market offer, and what are their uses too.

4.2 Exchanger models. Model selected

-Radiators: are heat exchangers used to transfer from a medium to another the thermal energy, with this action it can be possible heating or cooling. In the case of the system purposed the idea is to cool a fluid or coolant.

The radiators studied for making a choice were outside radiators. It consists of two different parts; the fans and the casing. The fans are usually direct driven fan units and the casing is usually stainless or efficiently corrosion protected. See **figure 31**.

Finally after looking for a company with good quality products it was found Thermokey which one didn´t offer radiators, but searching in other companies it was checked that the radiators were not the product that was needed to implement the new system because the radiators have not forced convection (no fans).

Figure 31; Radiators

-Condensers: the mission of a condenser is to condense a substance from it gaseous state to its liquid state, this process is made by cooling the substance. The power of the condenser can vary a lot for example it is possible to find condensers of 10KW until 202KW just only in Thermokey company but the power range is higher than this one, it is possible to find condensers of lower and higher power too. There are different types of condensers, ones that are flat that have 1-3 fans and the V-type that have 2-4 fans. In **figure 32** there is an example of a condenser V-type of 4 fans.

Speak of condensers mean speak of compressors too. The condenser is part of the refrigerant cycle (see figure 33), and it is composed by an expansion valve, an evaporator, a compressor and the condenser. For the new system it is not needed any compressor so condensers are not chosen because of this reason, which mean that the new system is not going to have higher energy consumption as if condensers were used.

Figure 32; condenser

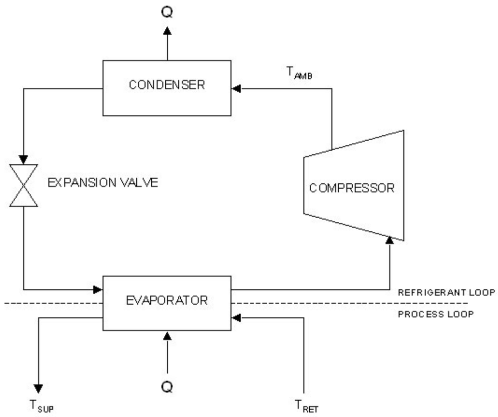


Figure 33; Refrigerated cycle

-Dry coolers: This one is the type that has been selected for the Project. The main reasons are the good relation between the heat absorbed and the size, also another good relation is the heat absorbed and the energy absorbed. Another important point to mention is that these dry coolers are used for cooling the ambient so is the best option for the areas that are shown in the project.

The most important point to remark is that the COP is quite high, as it is shown in the calculations done below.

The selection of these dry coolers is because the areas 1, 2, 3, 4 and 5 are quite similar because are closed areas. The main problem could be with the area 2 which is smaller comparing it with others areas. This difference of volume is traduced in an extra power dry cooler, the dry cooler selected for this area is the one with the lowest power capacity but it is not necessary so much power to cool down the area.

For the rest of the areas the chosen dry coolers meet the requirements for cooling each area.

Figure 34; Dry cooler

4.3 Calculation procedure

The kinds of exchanger selected are dry coolers. On the market it is possible to find a lot of companies that sell these types of device. Between all companies checked, “THERMOKEY” is the one that offers the best relation between cooling power and energy consumed, however theirs commodities are a little more expensive than other companies. All THERMOKEY products use a mix of glycol and water but with a different percentage of glycol in the mix. On this way the refrigerant is going to have 35 % of glycol and the particular characteristics of the mix are shown on another point which is specifically of refrigerants.

To continue with the calculation it is important not to forget that it is necessary to cover the two principal objectives;

· Satisfy the minimal cool demand of each room
· Choosing an exchanger as efficient as it is possible

Te exchangers purposed for THERMOKEY are designed to particulars conditions, according with the ENV1048 standards;

- Air inlet temperature T1 : 25°C
- Glycolwater inlet temperature Tw1 : 40°C
- Glycolwater outlet temperature Tw2 : 35°C
- Glycol : 35%
- ΔT1 → Tw1i – T1i : 15 K
- ΔTw → Tw1i – Tw2i : 5 K

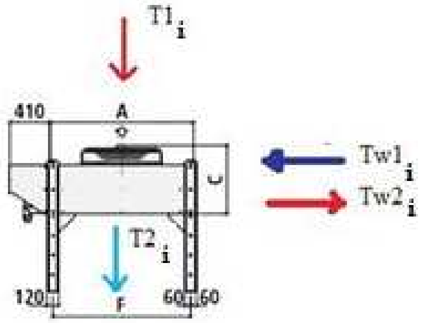


Figure 35; Dry cooler working diagram

It can happen that the air temperature differs from 25°C, the altitude from the sea level may differ and the glycol concentration is possible to change and there are more factors. So it's possible to choose the dry-cooler in the catalogue by multiplying the requested capacity with some factors that are explained in the following pages. To choose the dry cooler has been used the following formula:

Table capacity > Qreal*F1*F2*F3*F4*F5*F6

F1= Glycol mixture factor
F2= Entering temperature air factor
F3= Water rise factor (Tin water - Tout water)
F4= Water-Air factor (Tout water - Tin air)
F5= Fins pitch factor
F6= Altitude Factor

A) Glycol mixture factor (F1)

Mainly, Glycol is used on the refrigerant mixture to improve the freeze point of the mix, so the selection of the percentage of the amount of glycol in the mix is important to avoid the refrigerant freezing while the system is working.

Our working range of external temperatures is between -20ºC and -2ºC, so the mixture selected has to guaranty the correct working at least at -20ºC;

FREEZING POINTS (Mixture water + glycol) % on volume of distilled and deionized water	
% Propylene glycol	**UNTIL ºC**
0%	0
10%	-3
20%	-8
30%	-14
40%	-22
50%	**-34**
60%	-48
100%	-59

Table 27; Temperature limit per propylene glycol percentage

As it is shown in the **table 27** there are different temperatures for different percentages of glycol so as safety margin it has been consider 50% of glycol in the mix as the best option. This selection leads to raise the mixture density, but it is not as significantly as to consider changing to other mixture which could generate risk of freezing (40%).

Glycol %	*0*	*5*	*10*	*15*	*20*	*25*	*30*	*35*	*40*	*45*	*50*
F1	1,00	1,01	1,02	1,03	1,05	1,07	1,09	1,11	1,14	1,17	**1,20**

Table 28; Glycol mixture factor

As it is possible to appreciate in the **table 28** It is not too much the difference between F1 (40%) and F1 (50%), so it is a good option to pay with a little more of cooling power to down the freezing risk.

So **F1= 1,2**

B) Entering temperature air factor (F2)

Entering temperature air factor is not totally defined on the data specifications provided by THERMOKEY. It is possible to see on the **table 29** that it is not presented the valor used on this project (use range of entering temperature air between -20ºC and 0ºC)

After revising all the design parameters from all dry cooler components it is possible to assure that the fans and pump are the less resistant in terms of cooling, with a limit of using temperature of -26ºC.

So to solve the absence of data it has been thought to study the mathematical comportment of experimental data shown on the correction factor table;

Tin AIR (ºC)	5	10	15	20	25	30	35	40	45	50
F2	0,92	0,93	0,94	0,95	0,97	0,98	1,00	1,01	1,02	1,04

Table 29; Entering temperature air factor

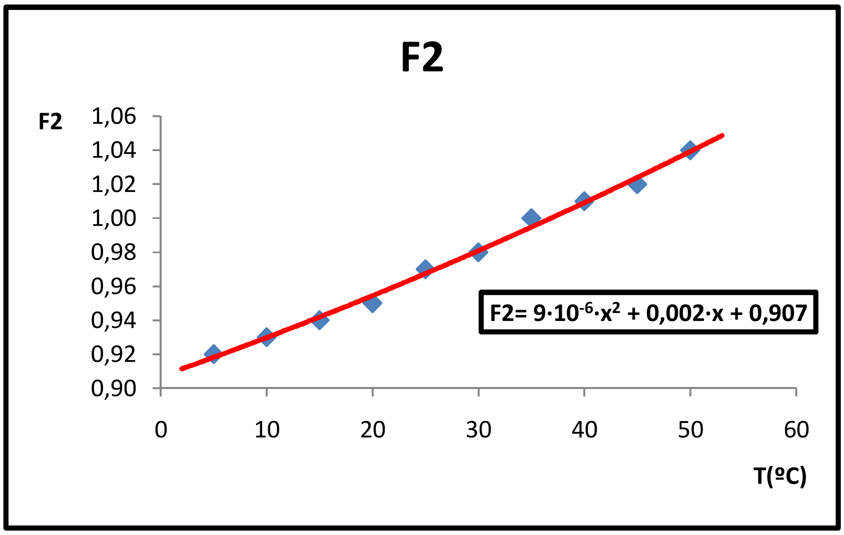


Figure 36, Factor 2 line

Figure 37; Inlet dry cooler

As it is shown in **figure 36**, realizing a mathematical regression with the data and approximating with a polynomial line it has been obtained the equation of comportment of F2 factor on function of the temperature.

Now only it is necessary have to distinguish between the two extreme cases that can appear during the working period;

- **-20ºC** → F2 = $9 \cdot 10^{-6}x^2 + 0{,}002x + 0{,}907$ → F2= 0.87
- **0ºC** → F2 = $9 \cdot 10^{-6}x^2 + 0{,}002x + 0{,}907$ → F2= 0.91

The worse selection is when the external temperature air is 0ºC, so **F2=0.91**

C) Water rise factor (Tw1 - Tw2) (F3)

This design parameter is very important to define the maximum temperature that is going to decrease on the outing air. If it is wanted to get heat transference between two flux currents it is necessary to keep a minimal difference of temperature between both two fluxes. So it is going to be advisable arriving to this point to fix objectives temperatures of outlet air for the two different extremes ways;

CASE (Outside Tº)	Tw1i	Tw2i	T2i (Objective)
-20ºC	-20ºC	-17ºC	-16ºC
-2ºC	-2ºC	1ºC	2ºC

Table 30; Different temperatures for the two cases studied

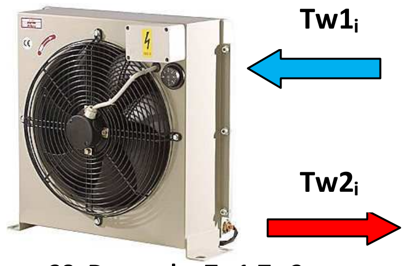


Figure 38; Dry cooler Tw1-Tw2

It should be understood that the difference of one degree between Tw2i and T21 is necessary to keep a good heat transference.

Also, this parameter quantify the maximum exchanger that it should be used on the room, that means that if it is selected an exchanger with more power than the correct, it might not decrease more than the T2i temperature or extract more heat from the dry cooler.

Water rise (ºC)	*|Tw1-Tw2|*										
	3	*4*	*5*	*6*	*7*	*8*	*9*	*10*	*12*	*15*	*20*	*25*
F3	**1,14**	1,12	1,10	1,06	1,05	1,03	1,02	1,00	0,97	0,92	0,86	0,81

Table 31; Water rise factor

D) Water-Air factor (Tw2i-T1i)(F4)

This parameter defines the amount of surface finned into the exchanger to get the complete heat energy transfer that is hoped.

To fix this factor it could be foreseen if it is wanted to transfer heat with a few degrees of difference, between T1i and Tw2i it would be needed a very big finned surface to get it.

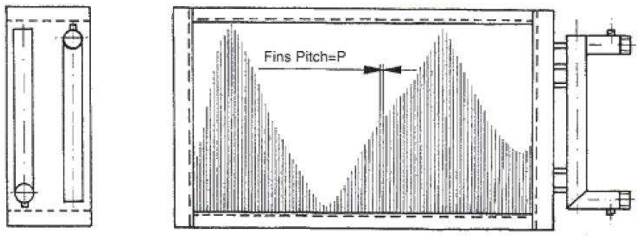


Figure 39; Fins

As has been done before to be sure that the system satisfies this coefficient it is necessary to study it on the worse cases.

In this case there are more than two cases because the two flux currents can vary into theirs ranges;

Water-Air (ºC)	*| T1i - Tw2i|*										
	20	*25*	*30*	*35*	*40*	*45*	*50*	*55*	*60*	*65*	*70*
F4	2,39	1,95	1,64	1,42	1,24	1,11	1,00	0,91	0,84	0,76	0,71

Table 32; Water-Air factor

CASE	T1i	Tw2i	|T1i-Tw2i|
1	-20ºC	8ºC	28ºC
2	-20ºC	0ºC	20 ºC
3	-2ºC	8ºC	**10 ºC**
4	-2ºC	2ºC	**4 ºC**

Table 33; Temperature difference between T1i and Tw2i

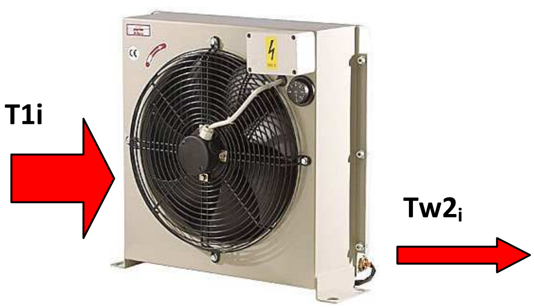


Figure 40; Dry cooler (T1i-Tw2i)

Newly it appears some values outrange, so to solve the absence of data it must be studied the experimental data, shown on the **table 32,** to know the comportment of the factor on function of the difference of temperatures |T1i-Tw2i|;

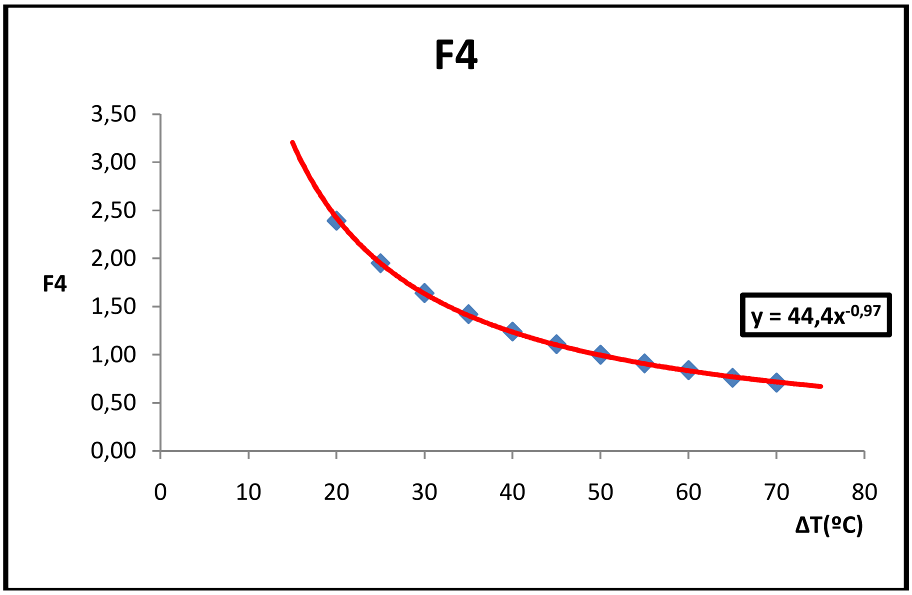


Figure 41; Factor 4 curve

Realizing a mathematical regression with the available data and approximating it with a potential line it has been obtained the comportment equation of F4 factor on function of difference of temperature. This curve can be seen in the **figure 41**.

Now it has to be solved the equation to the four cases to find the worst one;

CASE	T1i	Tw2i	|T1i-Tw2i|	Equation	F4
1	-20ºC	8ºC	28ºC	$44,4 \cdot x^{-0,97}$	1,75
2	-20ºC	0ºC	20 ºC	$44,4 \cdot x^{-0,97}$	2,39
3	-2ºC	8ºC	10 ºC	$44,4 \cdot x^{-0,97}$	**4,5**
4	-2ºC	2ºC	4 ºC	$44,4 \cdot x^{-0,97}$	**11**

Table 34; Factor 4 for the different cases studied

The worst result, as it is shown in the **table 34**, is the case 4, that result a **F4 = 11**

E) Fins pitch factor (F5)

Fins pitch factor is referenced to the distance between two consecutives fins, to do that concept more visual it is shown in the **figure 42**;

So if the distance is small it will have more fins on the same space, therefore it can be extracted more amount of heat than if on the other hand the distance was bigger. For that it is hoped that if the fins distance is going to increase the F5 factor will increase too.

Pitch (mm)	*2,1*	*2,5*	*3,2*	*4,2*
F5	**0,93**	1,00	1,09	1,25

Table 35; Fins pitch factor table

The pinch selected is 2,1mm, so **F5 = 0,93**

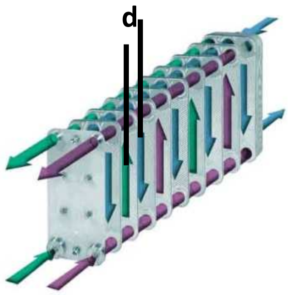


Figure 42; Fins distance

F) Altitude Factor (F6)

ALTITUDE (m)	Density values for ambient air (kg/m^3)		
	Minimum	Mean	Maximum
0	1,1405	1,2254	1,3167
300	1,1101	1,1886	1,2735
600	1,0812	1,1533	1,2302
900	1,0524	1,1197	1,2222
1000	1,0444	1,1101	1,1902
1200	1,0252	1,0861	1,1501
1500	0,9996	1,0556	1,1133
1800	0,9739	1,0236	1,0764
2000	0,9595	1,0076	1,0572
2200	0,9483	0,9931	1,0412
2400	0,9243	0,9643	1,0060
2700	0,8986	0,9355	0,9723
3000	0,8794	0,9115	0,9467

DECREASES

DECREASES

Table 36; Density values depending of the altitude

The absorbing capacity of an air flux is directly proportional at its density, so if it is taken as reference the sea level, it is going to be needed more surface of exchanger to deliver the same heat;

Altitude (m)	0	200	400	600	800	1000	1200	1400	1600	1800	2200	2600	3000
F6	**1,00**	1,01	1,03	1,04	1,06	1,07	1,09	1,11	1,12	1,14	1,18	1,22	1,26

Table 37; Altitude factor

Gävle has a mean altitude of 16m, so **F6 = 1** as it is shown in **table 37.**

4.4 Cool demands

To select the appropriate dry cooler exchanger it is necessary to know the cooling demand of each refrigeration room. It is logical thinking that the cooling demand is going to be different in each room because the rooms have inside different food and different geometry. To make these calculations for all the rooms it could be a project in itself, but there is other way shorter.

In the different rooms there are installed different old exchangers which satisfy without problem the cooling demand of each room, so the way followed to solve the problem of calculating the balance energy is resolved.

All what is necessary to do is identifier the thermal power of the different old exchangers and after that select the dry coolers with the same power or if not a little more.

4.5 Area 1

4.5.1 Old cooling method. Area 1

Unfortunately the old exchanger type (see **figure 43**) installed in the Area 1 is discontinued but fortunately the same supply company "Fincoil" disposes of other equivalent model;

Figure 43; Equipment installed in Area 1 and its characteristics

The characteristics of this exchanger are shown in **table 38**;

Model	PCG-8-5-7R1 1/2
Fin spacing (mm)	7
Nº Fans	1
Cool power-1400rpm(kW)	**7.4**
Cool power-1150rpm(kW)	6.53
Internal volume (l)	10
Weight (kg)	60

Table 38; Actual equipment characteristics

4.5.2 Dry cooler calculation. Area 1

Now it is disposed of the particular and general condition to define the dry cooler for the Area 1, so it is shown below the **table 39** joining the data;

FACTOR	Valor		Description
	Option 1	Option 2	
F1	**1,2**	-	Glycol mixture factor
F2	0,87	**0,91**	Entering Tº factor
F3	**1,14**	-	Water-rise |Tw2i-w1i| factor
F4	**4,5**	**11**	Water-Air |T1i-Tw2i| factor
F5	**0,93**	-	Fins pinch factor
F6	**1**	-	Altitude factor
Qreal	**7,4 (kW)**		Heat real power required

Table 39; new dry cooler characteristics

In the **table 39** there are two column for "valor", this fact is because sometimes exists the possibility of finding more than one case that can be the worst one.

It is possible to see on the **table 39** that the F4 factor has an especial influence on the dry cooler selection, F4 can increase the size from 4 to 11 times bigger. Therefore, it is necessary to make the two different calculations;

Table capacity > Qreal*F1*F2*F3*F4*F5*F6

- ***F4= 4,5*** → Table capacity > 7,4·1.2·0,91·1,14·**4,5**·0,93·1 → Table capacity > **38 kW**
- ***F4= 11*** → Table capacity > 7,4·1.2·0,91·1,14·**11**·0,93·1 → Table capacity > **94 kW**

So as it can be appreciated in the results above the highest power is for the one that has the factor 4 of 11. So checking the **table 40** it has been selected a dry cooler with a power of 105.7 KW with a COP of 155.

Model	Capacity (Power)(kW)		Air flow (m3/h)		Fluid flow (m3/h)		Pressure drops (kPa)		Noise level (dB(A) 10m)		n	Fan-motors (400V/3ph/50Hz) rpm		W		A		Surface (m2)	Tube vol. (dm3)	Weight (kg)	Connection ø (inch)	COP	
	Δ	Y	Δ	Y	Δ	Y	Δ	Y	Δ	Y		Δ	Y	Δ	Y	Δ	Y					Δ	Y
WQ1150.A	10,1	8,5	3250	2600	1,9	1,6	30	22	33	28	1	660	510	110	70	0,27	0,13	27	5	48	1"	92	121
WQ1150.B 10,13	11,7	9,3	3050	2300	2,2	1,8	32	21	33	28	1	660	510	110	70	0,27	0,13	41	8	52	1"	106	133
WQ1250.A	20,1	17,0	6500	5200	3,8	3,3	26	19	36	31	2	660	510	110	70	0,27	0,13	54	10	90	1"	91	121
WQ1250.B	18,5	14,9	6100	4600	3,5	2,9	12	8	36	31	2	660	510	110	70	0,27	0,13	54	11	98	1"1/2	84	106
WQ1163.A	22,5	18,2	7300	5500	4,3	3,5	35	24	39	31	1	630	470	330	190	0,73	0,37	58	11	79	1"	68	96
WQ1163.B	25,6	20,2	6800	5100	4,9	3,9	22	14	39	31	1	630	470	330	190	0,73	0,37	87	17	88	1"1/2	78	106
WQ1263.A	43,9	35,5	14600	11000	8,4	6,8	19	13	41	33	2	630	470	330	190	0,73	0,37	115	24	151	2"	67	93
WQ1263.B	52,4	41,3	13600	10200	10	7,9	39	25	41	33	2	630	470	330	190	0,73	0,37	173	34	168	2"	79	109
WQ1363.A	69,1	56,0	21900	16500	13,2	10,7	61	42	43	35	3	630	470	330	190	0,73	0,37	173	34	222	2"	70	98
WQ1363.B	78,4	61,8	20400	15300	15	11,8	35	23	43	35	3	630	470	330	190	0,73	0,37	259	49	248	2"	79	108
WQ1463.B	106,8	84,2	27200	20400	20,4	16,1	80	52	44	36	4	630	470	330	190	0,73	0,37	345	66	328	2"1/2	81	111
WR1163.A	15,8	11,9	4550	3200	3	2,3	19	11	30	22	1	400	290	140	70	0,41	0,16	58	11	79	1"	113	170
WR1163.B	17,8	13,0	4200	2950	3,4	2,5	35	20	30	22	1	400	290	140	70	0,41	0,16	87	16	88	1"	127	186
WR1263.A	32,2	24,3	9100	6400	6,1	4,6	31	19	32	24	2	400	290	140	70	0,41	0,16	115	22	151	1"1/2	115	174
WR1263.B	35,6	26,1	8400	5900	6,8	5	38	22	32	24	2	400	290	140	70	0,41	0,16	173	34	168	2"	127	186
WR1363.A	48,4	36,5	13650	9600	9,2	7	32	20	34	26	3	400	290	140	70	0,41	0,16	173	34	222	2"	115	174
WR1363.B	53,9	39,5	12600	8850	10,3	7,5	59	34	34	26	3	400	290	140	70	0,41	0,16	259	49	248	2"	128	188
WR1463.A	66,0	49,7	18200	12800	12,6	9,5	73	45	35	27	4	400	290	140	70	0,41	0,16	230	44	294	2"	118	178
WR1463.B	71,3	52,3	16800	11800	13,6	10	39	23	35	27	4	400	290	140	70	0,41	0,16	345	66	328	2"1/2	127	187
WR1180.A	29,6	24,5	9700	7500	5,7	4,7	30	22	28	23	1	440	340	310	170	1,2	0,48	76	16	146	1"1/2	95	144
WR1180.B	33,7	26,1	9000	6600	6,4	5	20	13	28	23	1	440	340	310	170	1,2	0,48	113	22	157	1"1/2	109	154
WR1280.A	59,4	49,1	19400	15000	11,3	9,4	31	22	31	26	2	440	340	310	170	1,2	0,48	151	30	280	2"	96	144
WR1280.B	70,1	54,9	18000	13400	13,4	10,5	63	41	31	26	2	440	340	310	170	1,2	0,48	226	44	303	2"	113	161
WR1380.A	85,1	70,3	29100	22500	16,2	13,4	12	9	33	28	3	440	340	310	170	1,2	0,48	226	43	414	2"	92	138
WR1380.B	105,0	78,1	27000	18900	20	14,9	58	35	33	28	3	440	340	310	170	1,2	0,48	339	65	449	2"1/2	113	153
WR1480.A	118,7	98,1	38800	30000	22,7	18,7	30	21	34	29	4	440	340	310	170	1,2	0,48	301	59	595	2"1/2	96	144
WR1480.B	134,8	105,7	36000	26800	25,7	20,1	18	12	34	29	4	440	340	310	170	1,2	0,48	452	88	642	3"	109	155
GR1580.A	148,6	123,9	48500	37500	28,4	23,7	25	18	35	30	5	440	340	310	170	1,2	0,48	527	89	733	2"1/2	96	146
GR1580.B	168,4	132,9	45000	33500	32,1	25,3	15	10	35	30	5	440	340	310	170	1,2	0,48	791	133	813	3"	109	156

Table 40; Dry cooler selection

4.6 Area 2

4.6.1 Old cooling method. Area 2

Fortunately the refrigeration model of Area 2 it is commercialized today, so their characteristics are shown in **table 41**;

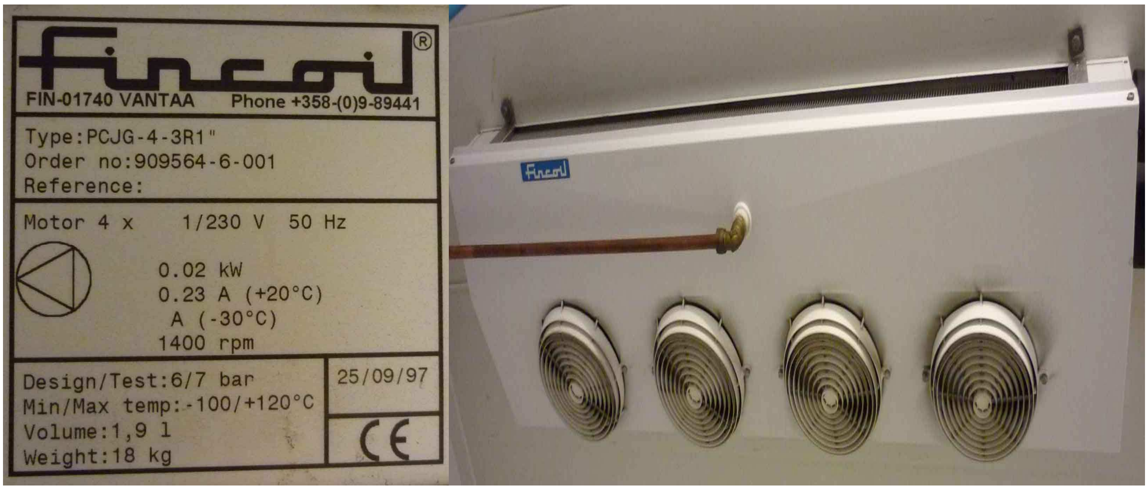


Figure 44; Actual equipment in Area 2

Model	PCJG -4- 3R1”
Fin spacing (mm)	5
Nº Fans	4
Cool power-1400 rpm (kW)	**1.9**
Internal volume (l)	1.9
Weight (kg)	18

Table 41; Actual equipment characteristics

4.6.2 Dry cooler calculation. Area 2

Now it is disposes of the particular and general condition to define the dry cooler for the Area 2, so it is shown below a table joining the data;

FACTOR	Valor		Description
	Option 1	Option 2	
F1	**1,2**	-	Glycol mixture factor
F2	0,87	**0,91**	Entering Tº factor
F3	**1,14**	-	Water-rise |Tw2i-w1i| factor
F4	**4,5**	**11**	Water-Air |T1i-Tw2i| factor
F5	**0,93**	-	Fins pinch factor
F6	**1**	-	Altitude factor
Qreal	**1,9 (kW)**		Heat real power required

Table 42; new equipment characteristics

In the **table 42** there are two columns for "valor", that is because sometimes exists the possibility to find more than one case that can be the worst.

It is possible see on the **table 42** that the F4 factor has an especial influence on the dry cooler selector, F4 can increase the size from 4 to 11 times bigger. Therefore, it is necessary to make the two different calculations;

Table capacity > Qreal*F1*F2*F3*F4*F5*F6

· ***F4= 4,5*** → Table capacity > 1,9·1.2·0,91·1,14·4,5·0,93·1 → Table capacity > **10 kW**
· ***F4= 11*** → Table capacity > 1,9·1.2·0,91·1,14·11·0,93·1 → Table capacity > **24 kW**

This room is small, but it is viable to select the bigger exchanger because the size difference between both options is not relevant;

Model	Capacity (Power)(kW)		Air flow (m3/h)		Fluid flow (m3/h)		Pressure drops (kPa)		Noise level (dB(A) 10m)		Fan-motors (400V/3ph/50Hz)							Surface (m2)	Tube vol. (dm3)	Weight (kg)	Connection ø (inch)	COP	
											n	rpm		W		A							
	Δ	Y	Δ	Y	Δ	Y	Δ	Y	Δ	Y		Δ	Y	Δ	Y	Δ	Y					Δ	Y
WQ1150.A	10,1	8,5	3250	2600	1,9	1,6	30	22	33	28	1	660	510	110	70	0,27	0,13	27	5	48	1"	92	121
WQ1150.B 10,13	11,7	9,3	3050	2300	2,2	1,8	32	21	33	28	1	660	510	110	70	0,27	0,13	41	8	52	1"	106	133
WQ1250.A	20,1	17,0	6500	5200	3,8	3,3	26	19	36	31	2	660	510	110	70	0,27	0,13	54	10	90	1"	91	121
WQ1250.B	18,5	14,9	6100	4600	3,5	2,9	12	8	36	31	2	660	510	110	70	0,27	0,13	54	11	98	1"1/2	84	106
WQ1163.A	22,5	18,2	7300	5500	4,3	3,5	35	24	39	31	1	630	470	330	190	0,73	0,37	58	11	79	1"	68	96
WQ1163.B	25,6	20,2	6800	5100	4,9	3,9	22	14	39	31	1	630	470	330	190	0,73	0,37	87	17	88	1"1/2	78	106
WQ1263.A	43,9	35,5	14600	11000	8,4	6,8	19	13	41	33	2	630	470	330	190	0,73	0,37	115	24	151	2"	67	93
WQ1263.B	52,4	41,3	13600	10200	10	7,9	39	25	41	33	2	630	470	330	190	0,73	0,37	173	34	168	2"	79	109
WQ1363.A	69,1	56,0	21900	16500	13,2	10,7	61	42	43	35	3	630	470	330	190	0,73	0,37	173	34	222	2"	70	98
WQ1363.B	78,4	61,8	20400	15300	15	11,8	35	23	43	35	3	630	470	330	190	0,73	0,37	259	49	248	2"	79	108
WQ1463.B	106,8	84,2	27200	20400	20,4	16,1	80	52	44	36	4	630	470	330	190	0,73	0,37	345	66	328	2"1/2	81	111
WR1163.A	15,8	11,9	4550	3200	3	2,3	19	11	30	22	1	400	290	140	70	0,41	0,16	58	11	79	1"	113	170
WR1163.B	17,8	13,0	4200	2950	3,4	2,5	35	20	30	22	1	400	290	140	70	0,41	0,16	87	16	88	1"	127	186
WR1263.A	32,2	24,3	9100	6400	6,1	4,6	31	19	32	24	2	400	290	140	70	0,41	0,16	115	22	151	1"1/2	115	174
WR1263.B	35,6	26,1	8400	5900	6,8	5	38	22	32	24	2	400	290	140	70	0,41	0,16	173	34	168	2"	127	186
WR1363.A	48,4	36,5	13650	9600	9,2	7	32	20	34	26	3	400	290	140	70	0,41	0,16	173	34	222	2"	115	174
WR1363.B	53,9	39,5	12600	8850	10,3	7,5	59	34	34	26	3	400	290	140	70	0,41	0,16	259	49	248	2"	128	188
WR1463.A	66,0	49,7	18200	12800	12,6	9,5	73	45	35	27	4	400	290	140	70	0,41	0,16	230	44	294	2"	118	178
WR1463.B	71,3	52,3	16800	11800	13,6	10	39	23	35	27	4	400	290	140	70	0,41	0,16	345	66	328	2"1/2	127	187
WR1180.A	29,6	24,5	9700	7500	5,7	4,7	30	22	28	23	1	440	340	310	170	1,2	0,48	76	16	146	1"1/2	95	144
WR1180.B	33,7	26,1	9000	6600	6,4	5	20	13	28	23	1	440	340	310	170	1,2	0,48	113	22	157	1"1/2	109	154
WR1280.A	59,4	49,1	19400	15000	11,3	9,4	31	22	31	26	2	440	340	310	170	1,2	0,48	151	30	280	2"	96	144
WR1280.B	70,1	54,9	18000	13400	13,4	10,5	63	41	31	26	2	440	340	310	170	1,2	0,48	226	44	303	2"	113	161
WR1380.A	85,1	70,3	29100	22500	16,2	13,4	12	9	33	28	3	440	340	310	170	1,2	0,48	226	43	414	2"	92	138
WR1380.B	105,0	78,1	27000	18900	20	14,9	58	35	33	28	3	440	340	310	170	1,2	0,48	339	65	449	2"1/2	113	153
WR1480.A	118,7	98,1	38800	30000	22,7	18,7	30	21	34	29	4	440	340	310	170	1,2	0,48	301	59	595	2"1/2	96	144

Table 43; Dry cooler selection

4.7 Area 3

4.7.1 Old cooling method. Area 3

Radiators as it has been said before don´t use fans, so to get the heat transference necessary the contact surface between fins and cold air must be bigger.

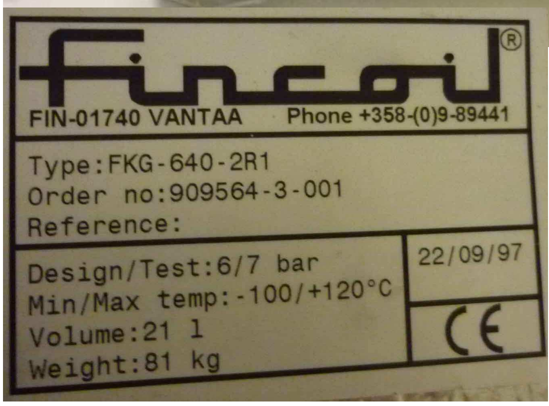


Figure 44; Actual equipment installed

Model	FKG-640-2R1
Nº Fans	0
Cool power (kW)	**2,1**
Length (mm)	3910
Wide (mm)	770
Fins spacing (mm)	7.5
Internal volume (l)	21
Weight (kg)	81

Table 44; Actual equipment characteristics

In this room there are three big radiators. Each one with 2,5kW of cooling power, it must be observed that the total cooling power in the room is around 6,3kW and that power could have been obtained with only one condenser with forced convection of the “PCG” type.

It exist an explication for that, it is possible that on the firsts periods of utilization of the room it was important to have a low noise level and maybe it was also important to have the absence of jets currents of air.

Today in the room it is stored packaged meat, therefore to implant the new system there will be no problem.

4.7.2 Dry cooler calculation. Area 3

Now it is disposed of the particular and general condition to define the dry cooler for the Area 3, so it is shown below the **table 45** joining the data;

FACTOR	***Valor***		***Description***
	Option 1	***Option 2***	
F1	**1,2**	-	Glycol mixture factor
F2	0,87	**0,91**	Entering Tº factor
F3	**1,14**	-	Water-rise |Tw2i-w1i| factor
F4	**4,5**	**11**	Water-Air |T1i-Tw2i| factor
F5	**0,93**	-	Fins pinch factor
F6	**1**	-	Altitude factor
Qreal	**6,3 (kW)**		Heat real power required

Table 45; New equipment characteristics

On the last table there are two columns for "valor", that is because sometimes exists the possibility of finding more than one case that may be the worst.

It is possible to see on the **table 45** that the F4 factor has an especial influence on the dry cooler selection, F4 can increase the size from 4 to 11 times bigger. Therefore, it is necessary to make the two different calculations;

Table capacity > Qreal*F1*F2*F3*F4*F5*F6

· ***F4= 4,5*** → Table capacity > 6,3·1.2·0,91·1,14·4,5·0,93·1 → Table capacity > **33 kW**
· ***F4= 11*** → Table capacity > 6,3·1.2·0,91·1,14·11·0,93·1 → Table capacity > **80 kW**

The valor is similar as the one of the area 1, it is like this because the room volume of both, area 1 and area 3, is similar. So as it is shown in **table 46** the dry cooler selected is the same as it was selected for the area 1.

Model	Capacity (Power)(kW)		Air flow (m3/h)		Fluid flow (m3/h)		Pressure drops (kPa)		Noise level (dB(A) 10m)		Fan-motors (400V/3ph/50Hz)							Surface (m2)	Tube vol. (dm3)	Weight (kg)	Connection ø (inch)	COP	
	Δ	Y	Δ	Y	Δ	Y	Δ	Y	Δ	Y	n	rpm Δ	rpm Y	W Δ	W Y	A Δ	A Y					Δ	Y
WQ1150.A	10,1	8,5	3250	2600	1,9	1,6	30	22	33	28	1	660	510	110	70	0,27	0,13	27	5	48	1"	92	121
WQ1150.B 10,13	11,7	9,3	3050	2300	2,2	1,8	32	21	33	28	1	660	510	110	70	0,27	0,13	41	8	52	1"	106	133
WQ1250.A	20,1	17,0	6500	5200	3,8	3,3	26	19	36	31	2	660	510	110	70	0,27	0,13	54	10	90	1"	91	121
WQ1250.B	18,5	14,9	6100	4600	3,5	2,9	12	8	36	31	2	660	510	110	70	0,27	0,13	54	11	98	1"1/2	84	106
WQ1163.A	22,5	18,2	7300	5500	4,3	3,5	35	24	39	31	1	630	470	330	190	0,73	0,37	58	11	79	1"	68	96
WQ1163.B	25,6	20,2	6800	5100	4,9	3,9	22	14	39	31	1	630	470	330	190	0,73	0,37	87	17	88	1"1/2	78	106
WQ1263.A	43,9	35,5	14600	11000	8,4	6,8	19	13	41	33	2	630	470	330	190	0,73	0,37	115	24	151	2"	67	93
WQ1263.B	52,4	41,3	13600	10200	10	7,9	39	25	41	33	2	630	470	330	190	0,73	0,37	173	34	168	2"	79	109
WQ1363.A	69,1	56,0	21900	16500	13,2	10,7	61	42	43	35	3	630	470	330	190	0,73	0,37	173	34	222	2"	70	98
WQ1363.B	78,4	61,8	20400	15300	15	11,8	35	23	43	35	3	630	470	330	190	0,73	0,37	259	49	248	2"	79	108
WQ1463.B	106,8	84,2	27200	20400	20,4	16,1	80	52	44	36	4	630	470	330	190	0,73	0,37	345	66	328	2"1/2	81	111
WR1163.A	15,8	11,9	4550	3200	3	2,3	19	11	30	22	1	400	290	140	70	0,41	0,16	58	11	79	1"	113	170
WR1163.B	17,8	13,0	4200	2950	3,4	2,5	35	20	30	22	1	400	290	140	70	0,41	0,16	87	16	88	1"	127	186
WR1263.A	32,2	24,3	9100	6400	6,1	4,6	31	19	32	24	2	400	290	140	70	0,41	0,16	115	22	151	1"1/2	115	174
WR1263.B	35,6	26,1	8400	5900	6,8	5	38	22	32	24	2	400	290	140	70	0,41	0,16	173	34	168	2"	127	186
WR1363.A	48,4	36,5	13650	9600	9,2	7	32	20	34	26	3	400	290	140	70	0,41	0,16	173	34	222	2"	115	174
WR1363.B	53,9	39,5	12600	8850	10,3	7,5	59	34	34	26	3	400	290	140	70	0,41	0,16	259	49	248	2"	128	188
WR1463.A	66,0	49,7	18200	12800	12,6	9,5	73	45	35	27	4	400	290	140	70	0,41	0,16	230	44	294	2"	118	178
WR1463.B	71,3	52,3	16800	11800	13,6	10	39	23	35	27	4	400	290	140	70	0,41	0,16	345	66	328	2"1/2	127	187
WR1180.A	29,6	24,5	9700	7500	5,7	4,7	30	22	28	23	1	440	340	310	170	1,2	0,48	76	16	146	1"1/2	95	144
WR1180.B	33,7	26,1	9000	6600	6,4	5	20	13	28	23	1	440	340	310	170	1,2	0,48	113	22	157	1"1/2	109	154
WR1280.A	59,4	49,1	19400	15000	11,3	9,4	31	22	31	26	2	440	340	310	170	1,2	0,48	151	30	280	2"	96	144
WR1280.B	70,1	54,9	18000	13400	13,4	10,5	63	41	31	26	2	440	340	310	170	1,2	0,48	226	44	303	2"	113	161
WR1380.A	85,1	70,3	29100	22500	16,2	13,4	12	9	33	28	3	440	340	310	170	1,2	0,48	226	43	414	2"	92	138
WR1380.B	105,0	78,1	27000	18900	20	14,9	58	35	33	28	3	440	340	310	170	1,2	0,48	339	65	449	2"1/2	113	153
WR1480.A	118,7	98,1	38800	30000	22,7	18,7	30	21	34	29	4	440	340	310	170	1,2	0,48	301	59	595	2"1/2	96	144
WR1480.B	134,8	105,7	36000	26800	25,7	20,1	18	12	34	29	4	440	340	310	170	1,2	0,48	452	88	642	3"	109	155
GR1580.A	148,6	123,9	48500	37500	28,4	23,7	25	18	35	30	5	440	340	310	170	1,2	0,48	527	89	733	2"1/2	96	146
GR1580.B	168,4	132,9	45000	33500	32,1	25,3	15	10	35	30	5	440	340	310	170	1,2	0,48	791	133	813	3"	109	156

Table 46; Dry cooler selection

4.8 Area 4

4.8.1 Old cooling method. Area 3

As it happened in the case presented before, in this room is used a radiator too.

Figure 45; Actual equipment installed

Model	FKG-640-2R1
Nº Fans	0
Cool power (kW)	**2,1**
Length (mm)	3910
Wide (mm)	770
Fins spacing (mm)	7.5
Internal volume (l)	21
Weight (kg)	81

Table 47; Actual equipment characteristics

The explanation is exactly the same as before, but with a little difference. This difference is the size of the room, in this case the room is smaller so it has been only installed one radiator, the total cooling power of the enclosure area is 2,1kW.

4.8.2 Dry cooler calculation. Area 4

Now it is disposed of the particular and general condition to define the dry cooler for the Area 4, so it is shown below the **table 48** joining the data;

FACTOR	***Valor***		***Description***
	Option 1	***Option 2***	
F1	**1,2**	-	Glycol mixture factor
F2	0,87	**0,91**	Entering Tº factor
F3	**1,14**	-	Water-rise |Tw2i-w1i| factor
F4	**4,5**	**11**	Water-Air |T1i-Tw2i| factor
F5	**0,93**	-	Fins pinch factor
F6	**1**	-	Altitude factor
Qreal	**2,1 (kW)**		Heat real power required

Table 48; new equipment characteristics

As it is shown in **table 48** there are two columns for “valor”, this fact is because sometimes exists the possibility of finding more than one case that can be the worst one.

It is possible to see on the **table 48** that the F4 factor has an especial influence on the dry cooler selection, F4 can increase the size from 4 to 11 times bigger. Therefore, it is necessary to make the two different calculations;

Table capacity > Qreal*F1*F2*F3*F4*F5*F6

· ***F4= 4,5*** → Table capacity > 2,1·1.2·0,91·1,14·4,5·0,93·1 → Table capacity > **11 kW**
· ***F4= 11*** → Table capacity > 2,1·1.2·0,91·1,14·11·0,93·1 → Table capacity > **27 kW**

This room is small so it happens the same as it happened before in the area 2, it is viable to select the bigger exchanger because the size difference between both options is not very relevant.

It has been chosen the dry cooler of 26.1KW because the COP of this dry cooler is the most interesting for our system.

Model	Capacity (Power)(kW)		Air flow (m3/h)		Fluid flow (m3/h)		Pressure drops (kPa)		Noise level (dB(A) 10m)		Fan-motors (400V/3ph/50Hz)							Surface (m2)	Tube vol. (dm3)	Weight (kg)	Connection ø (inch)	COP	
											n	rpm		W		A							
	Δ	Y	Δ	Y	Δ	Y	Δ	Y	Δ	Y		Δ	Y	Δ	Y	Δ	Y					Δ	Y
WQ1150.A	10,1	8,5	3250	2600	1,9	1,6	30	22	33	28	1	660	510	110	70	0,27	0,13	27	5	48	1"	92	121
WQ1150.B 10,13	11,7	9,3	3050	2300	2,2	1,8	32	21	33	28	1	660	510	110	70	0,27	0,13	41	8	52	1"	106	133
WQ1250.A	20,1	17,0	6500	5200	3,8	3,3	26	19	36	31	2	660	510	110	70	0,27	0,13	54	10	90	1"	91	121
WQ1250.B	18,5	14,9	6100	4600	3,5	2,9	12	8	36	31	2	660	510	110	70	0,27	0,13	54	11	98	1"1/2	84	106
WQ1163.A	22,5	18,2	7300	5500	4,3	3,5	35	24	39	31	1	630	470	330	190	0,73	0,37	58	11	79	1"	68	96
WQ1163.B	25,6	20,2	6800	5100	4,9	3,9	22	14	39	31	1	630	470	330	190	0,73	0,37	87	17	88	1"1/2	78	106
WQ1263.A	43,9	35,5	14600	11000	8,4	6,8	19	13	41	33	2	630	470	330	190	0,73	0,37	115	24	151	2"	67	93
WQ1263.B	52,4	41,3	13600	10200	10	7,9	39	25	41	33	2	630	470	330	190	0,73	0,37	173	34	168	2"	79	109
WQ1363.A	69,1	56,0	21900	16500	13,2	10,7	61	42	43	35	3	630	470	330	190	0,73	0,37	173	34	222	2"	70	98
WQ1363.B	78,4	61,8	20400	15300	15	11,8	35	23	43	35	3	630	470	330	190	0,73	0,37	259	49	248	2"	79	108
WQ1463.B	106,8	84,2	27200	20400	20,4	16,1	80	52	44	36	4	630	470	330	190	0,73	0,37	345	66	328	2"1/2	81	111
WR1163.A	15,8	11,9	4550	3200	3	2,3	19	11	30	22	1	400	290	140	70	0,41	0,16	58	11	79	1"	113	170
WR1163.B	17,8	13,0	4200	2950	3,4	2,5	35	20	30	22	1	400	290	140	70	0,41	0,16	87	16	88	1"	127	186
WR1263.A	32,2	24,3	9100	6400	6,1	4,6	31	19	32	24	2	400	290	140	70	0,41	0,16	115	22	151	1"1/2	115	174
WR1263.B	35,6	26,1	8400	5900	6,8	5	38	22	32	24	2	400	290	140	70	0,41	0,16	173	34	168	2"	127	186
WR1363.A	48,4	36,5	13650	9600	9,2	7	32	20	34	26	3	400	290	140	70	0,41	0,16	173	34	222	2"	115	174

Table 49; dry cooler selection

4.9 Area 5

In this area like in area 3 and area4 the equipment used for cooling the room are radiators. In **table 50** are shown the characteristics of this radiators.

Model	FKG-621-1R ¾"
Nº Fans	0
Cool power (kW)	1,4
Length (mm)	2130
Wide (mm)	770
Fins spacing (mm)	7.5
Internal volume (l)	11.1
Weight (kg)	43

Model	FKG-640-2R1
Nº Fans	0
Cool power (kW)	1,8
Length (mm)	2800
Wide (mm)	770
Fins spacing (mm)	7.5
Internal volume (l)	16
Weight (kg)	59

Table 50; Actual equipment characteristics

The difference with the other rooms is that now there are two radiators with different power, the total cooling power of the room is 3,2kW.

4.9.1 Dry cooler calculation. Area 5

Now it is disposed of the particular and general condition to define the dry cooler for the Area 4, so it is shown below the **table 51** joining the data;

FACTOR	*Valor*		*Description*
	Option 1	*Option 2*	
F1	**1,2**	-	Glycol mixture factor
F2	0,87	**0,91**	Entering Tº factor
F3	**1,14**	-	Water-rise |Tw2i-w1i| factor
F4	**4,5**	**11**	Water-Air |T1i-Tw2i| factor
F5	**0,93**	-	Fins pinch factor
F6	**1**	-	Altitude factor
Qreal	**3,2 (kW)**		Heat real power required

Table 51; new system characteristics

On the last table there are two columns for "valor", that is because sometimes exists the possibility of finding more than one case that can be considered the worst.

It is possible to see on the **table 51** that the F4 factor has an especial influence on the dry cooler selection, F4 can increase the size from 4 to 11 times bigger. Therefore, it is necessary to make the two different calculations for these cases;

Table capacity > Qreal*F1*F2*F3*F4*F5*F6

- ***F4= 4,5*** → Table capacity > 3,2·1.2·0,91·1,14·4,5·0,93·1 → Table capacity > **17 kW**
- ***F4= 11*** → Table capacity > 3,2·1.2·0,91·1,14·11·0,93·1 → Table capacity > **41 kW**

This room is relatively small, it is viable to select the bigger exchanger because the size difference between both options is not very relevant so there won´t be any space problem;

Model	Capacity (Power)(kW)		Air flow (m3/h)		Fluid flow (m3/h)		Pressure drops (kPa)		Noise level (dB(A) 10m)		Fan-motors (400V/3ph/50Hz)							Surface (m2)	Tube vol. (dm3)	Weight (kg)	Connection ø (inch)	COP	
	Δ	Y	Δ	Y	Δ	Y	Δ	Y	Δ	Y	n	rpm Δ	rpm Y	W Δ	W Y	A Δ	A Y					Δ	Y
WQ1150.A	10,1	8,5	3250	2600	1,9	1,6	30	22	33	28	1	660	510	110	70	0,27	0,13	27	5	48	1"	92	121
WQ1150.B 10,13	11,7	9,3	3050	2300	2,2	1,8	32	21	33	28	1	660	510	110	70	0,27	0,13	41	8	52	1"	106	133
WQ1250.A	20,1	17,0	6500	5200	3,8	3,3	26	19	36	31	2	660	510	110	70	0,27	0,13	54	10	90	1"	91	121
WQ1250.B	18,5	14,9	6100	4600	3,5	2,9	12	8	36	31	2	660	510	110	70	0,27	0,13	54	11	98	1"1/2	84	106
WQ1163.A	22,5	18,2	7300	5500	4,3	3,5	35	24	39	31	1	630	470	330	190	0,73	0,37	58	11	79	1"	68	96
WQ1163.B	25,6	20,2	6800	5100	4,9	3,9	22	14	39	31	1	630	470	330	190	0,73	0,37	87	17	88	1"1/2	78	106
WQ1263.A	43,9	35,5	14600	11000	8,4	6,8	19	13	41	33	2	630	470	330	190	0,73	0,37	115	24	151	2"	67	93
WQ1263.B	52,4	41,3	13600	10200	10	7,9	39	25	41	33	2	630	470	330	190	0,73	0,37	173	34	168	2"	79	109
WQ1363.A	69,1	56,0	21900	16500	13,2	10,7	61	42	43	35	3	630	470	330	190	0,73	0,37	173	34	222	2"	70	98
WQ1363.B	78,4	61,8	20400	15300	15	11,8	35	23	43	35	3	630	470	330	190	0,73	0,37	259	49	248	2"	79	108
WQ1463.B	106,8	84,2	27200	20400	20,4	16,1	80	52	44	36	4	630	470	330	190	0,73	0,37	345	66	328	2"1/2	81	111
WR1163.A	15,8	11,9	4550	3200	3	2,3	19	11	30	22	1	400	290	140	70	0,41	0,16	58	11	79	1"	113	170
WR1163.B	17,8	13,0	4200	2950	3,4	2,5	35	20	30	22	1	400	290	140	70	0,41	0,16	87	16	88	1"	127	186
WR1263.A	32,2	24,3	9100	6400	6,1	4,6	31	19	32	24	2	400	290	140	70	0,41	0,16	115	22	151	1"1/2	115	174
WR1263.B	35,6	26,1	8400	5900	6,8	5	38	22	32	24	2	400	290	140	70	0,41	0,16	173	34	168	2"	127	186
WR1363.A	48,4	36,5	13650	9600	9,2	7	32	20	34	26	3	400	290	140	70	0,41	0,16	173	34	222	2"	115	174
WR1363.B	53,9	39,5	12600	8850	10,3	7,5	59	34	34	26	3	400	290	140	70	0,41	0,16	259	49	248	2"	128	188
WR1463.A	66,0	49,7	18200	12800	12,6	9,5	73	45	35	27	4	400	290	140	70	0,41	0,16	230	44	294	2"	118	178
WR1463.B	71,3	52,3	16800	11800	13,6	10	39	23	35	27	4	400	290	140	70	0,41	0,16	345	66	328	2"1/2	127	187
WR1180.A	29,6	24,5	9700	7500	5,7	4,7	30	22	28	23	1	440	340	310	170	1,2	0,48	76	16	146	1"1/2	95	144
WR1180.B	33,7	26,1	9000	6600	6,4	5	20	13	28	23	1	440	340	310	170	1,2	0,48	113	22	157	1"1/2	109	154
WR1280.A	59,4	49,1	19400	15000	11,3	9,4	31	22	31	26	2	440	340	310	170	1,2	0,48	151	30	280	2"	96	144
WR1280.B	70,1	54,9	18000	13400	13,4	10,5	63	41	31	26	2	440	340	310	170	1,2	0,48	226	44	303	2"	113	161
WR1380.A	85,1	70,3	29100	22500	16,2	13,4	12	9	33	28	3	440	340	310	170	1,2	0,48	226	43	414	2"	92	138
WR1380.B	105,0	78,1	27000	18900	20	14,9	58	35	33	28	3	440	340	310	170	1,2	0,48	339	65	449	2"1/2	113	153
WR1480.A	118,7	98,1	38800	30000	22,7	18,7	30	21	34	29	4	440	340	310	170	1,2	0,48	301	59	595	2"1/2	96	144
WR1480.B	134,8	105,7	36000	26800	25,7	20,1	18	12	34	29	4	440	340	310	170	1,2	0,48	452	88	642	3"	109	155
GR1580.A	148,6	123,9	48500	37500	28,4	23,7	25	18	35	30	5	440	340	310	170	1,2	0,48	527	89	733	2"1/2	96	146
GR1580.B	168,4	132,9	45000	33500	32,1	25,3	15	10	35	30	5	440	340	310	170	1,2	0,48	791	133	813	3"	109	156
GR1680.A	182,8	152,3	58200	45000	34,9	29,1	43	31	36	31	6	440	340	310	170	1,2	0,48	633	105	949	2"1/2	98	149
GR1680.B	206,8	163,0	54000	40200	39,4	31,1	26	17	36	31	6	440	340	310	170	1,2	0,48	949	158	1039	3"	111	160
GR1780.B	245,0	193,0	63000	46900	46,7	36,8	41	27	36	31	7	440	340	310	170	1,2	0,48	1107	182	1210	3"	113	162
WR2180.A	54,3	45,1	18800	14600	10,4	8,6	16	12	31	26	2	445	340	310	170	1,2	0,48	137	32	243	2"	88	133

Table 52; Dry cooler selection

4.10 Resume table. Cooling needed.

AREA	VOLUME [m³]	DESCRIPTION	TOTAL COOL POWER REQUIRED [kW]	
			Δ|T1-Tw2|=10ºC	Δ|T1-Tw2|=8ºC
1	247	Packeted milk products	38	94
2	70	Packeted milk products	10	24
3	151	Meat	33	80
4	45	Meat and cheese	11	27
5	63	Meat	17	41
TOTAL COOL POWER REQUIRED			**109**	**266**

Table 53; Cooling demanded

EXHANGER	Model	Capacity (Power)(kW)		Air flow (m3/h)		Fluid flow (m3/h)		Press. drops (kPa)		Fan-motors (400V/3ph/50Hz)							COP	
										n	rpm		W		A			
		Δ	Y	Δ	Y	Δ	Y	Δ	Y		Δ	Y	Δ	Y	Δ	Y	Δ	Y
Area 1	WR1480.B	134,8	105,7	36000	26800	25,7	20,1	18	12	4	440	340	310	170	1,2	0,48	109	155
Area 2	WR1263.A	32,2	24,3	9100	6400	6,1	4,6	31	19	2	400	290	140	70	0,41	0,16	115	174
Area 3	WR1480.B	134,8	105,7	36000	26800	25,7	20,1	18	12	4	440	340	310	170	1,2	0,48	109	155
Area 4	WR1263.B	35,6	26,1	8400	5900	6,8	5	38	22	2	400	290	140	70	0,41	0,16	127	186
Area 5	WR1363.B	53,9	39,5	12600	8850	10,3	7,5	59	34	3	400	290	140	70	0,41	0,16	128	188
TOTAL INDOOR	-	-	**301,3**	-	-	-	**57.3**	-	-	-	-	-	-	-	-	-	-	-
Outdoor	GR1680.B	207	163	54000	40200	39,4	31,1	26	17	36	31	6	440	340	310	170	111	160
TOTAL OUTDOOR	2x GR1680.B	-	**326**	-	-	-	**62.2**	-	-	-	-	-	-	-	-	-	-	-

Table 54; Resume table of the different dry coolers selected

5 EXTERNAL EXCHANGER

5.1 Introduction

To complete the cycle of the external extraction of "cold" and internal deliberation of "cold" the internal exchangers have been selected. Another essential component to carry out the process is the external exchanger.

The function of this device is to absorb the outside cold through a fins battery and forced convection and transfer it to the refrigerant. The main problem of this selection is to choose the appropriate exchanger because if it is selected an exchanger with more power, the cold absorbed from it is not going to be taken, consequently the COP of all the system is going to decrease. On the other hand if the exchanger is with less power, the internal temperature wished is not going to be reached.

So in the next calculations it is going to be tried the fact of optimizer this relation.

5.2 Assumptions

The exchanger is placed outside, so it can be considered the external space like a cooling focus; it can take as cold as it wants and the external temperature is not going to vary.

It exist a lot of external conditions which could influence with more or less extent in the exchanger, some important conditions are;

- *The sun*; the installation of the external exchanger should be placed over the north side of the building. With this measure it is minimized the amount of solar radiation absorbed for the metal casing of the exchanger. If this occurs the metal casing would heat the air around the exchanger, hence the cold absorbs is less.

- *The snow*; Snow is a meteorological condition that can ruin all the project, many problems can then appear. One of the most important is if the snow goes inside the exchanger, so it is necessary a safety structure for placing outside the exchanger. This necessity comes in itself a problem, the exchanger enclosed means that the air will have problems to flux freely, so it may appear over press points or under press points, as well as a micro climate but with a little higher temperature. The structure must be designed large enough. It is important also to assure that the structure is not going to be completely closed for the snow, the snow acts as an insulator so the exchanger enclosed would not take long time to break.

- *The rain*; the exchanger has inside electric circuits to drive the fan velocity, to supply electric power so on. It is not possible let that rain comes into external exchanger.

- *Good ventilation*; this necessity is totally linked at the problem with the snow.

5.3 Calculation procedure

All the general considerations for calculate the outdoor exchanger are exactly the same as the ones used to calculate the indoor ones. It is going to be utilized the same equation used before that was provided by the company, however the factors value can change. Hence it is necessary check again all factors on the new conditions.

G) Glycol mixture factor (F1)

The refrigerant and their composition no change, so the factor F1 no change too;
F1= 1,2

H) Entering temperature air factor (F2)

The temperature range where the external temperature moves is the same, so;

- $\boxed{-20ºC} \rightarrow F2 = 9 \cdot 10^{-6} x^2 + 0{,}002x + 0{,}907 \rightarrow F2 = 0.87$
- $\boxed{0ºC} \rightarrow F2 = 9 \cdot 10^{-6} x^2 + 0{,}002x + 0{,}907 \rightarrow F2 = 0.91$

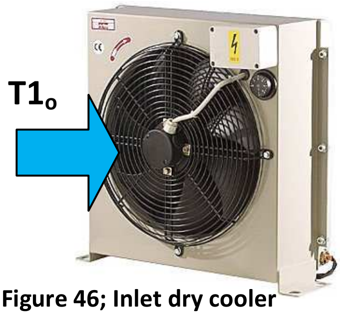


Figure 46; Inlet dry cooler

The worst selection is when the external temperature air is 0ºC, so **F2=0.91**

I) Water rise factor (Tw1 - Tw2) (F3)

This design parameter is very important to define the maximum temperature that has decreased on the outing air. If it is wanted to get heat transference between two flux currents it is necessary to keep a minimal difference of temperature between both two fluxes. So it would be advisable arrived to this point to fix objectives temperatures of the outlet air for the two different extremes ways;

As the other case |Tw1 - Tw2|=3, so **F3=1,14**

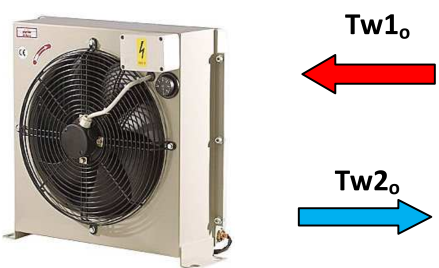


figure 47; Dry cooler Tw1o-Tw2o

J) Water-Air factor (Tw2i-T1i)(F4)

The importance of this parameter remains vital to the final selection of the external exchanger.

It is important again to be sure that the system satisfies this coefficient so it is necessary to apply it on the worst cases.

For the study of this factor there are more than two cases because the two flux currents can vary into theirs ranges;

Water-Air (ºC)	*| Tw2o-T1o|*										
	20	*25*	*30*	*35*	*40*	*45*	*50*	*55*	*60*	*65*	*70*
F4	2,39	1,95	1,64	1,42	1,24	1,11	1,00	0,91	0,84	0,76	0,71

Table 55; Factor 4 table

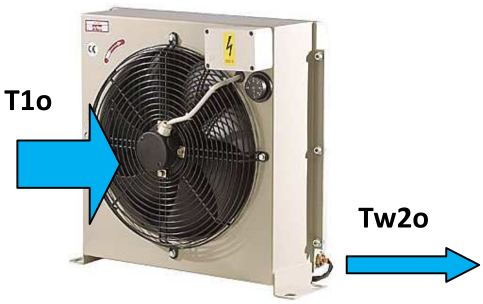


Figure 48; Dry cooler Tw2i-T1i

Again it appears some values outrange, so to solve the absence of data it must be studied the experimental data that is shown in **table 55.** Then it is known how is the comportment of the factor on function of the difference of temperatures |T1i-Tw2i|, so it has been done the **figure 49**;

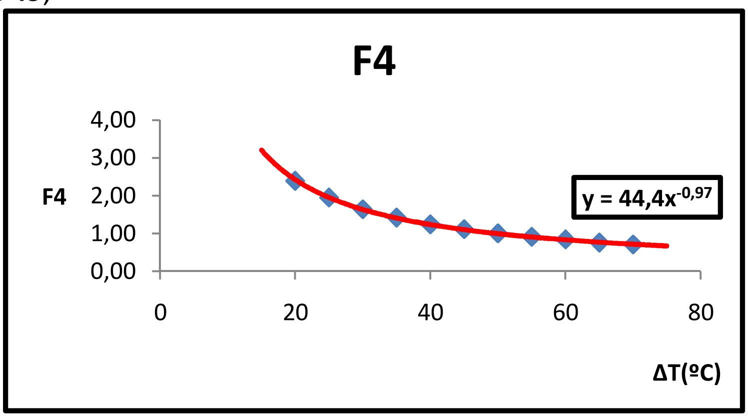


Figure 49; Factor 4 curve

So realizing a mathematical regression with the available data and approximating them with a potential line it has been obtained the comportment equation of F4 factor on function of difference of temperature.

Now it has to be solved the equation to all the different cases to find the worst one;

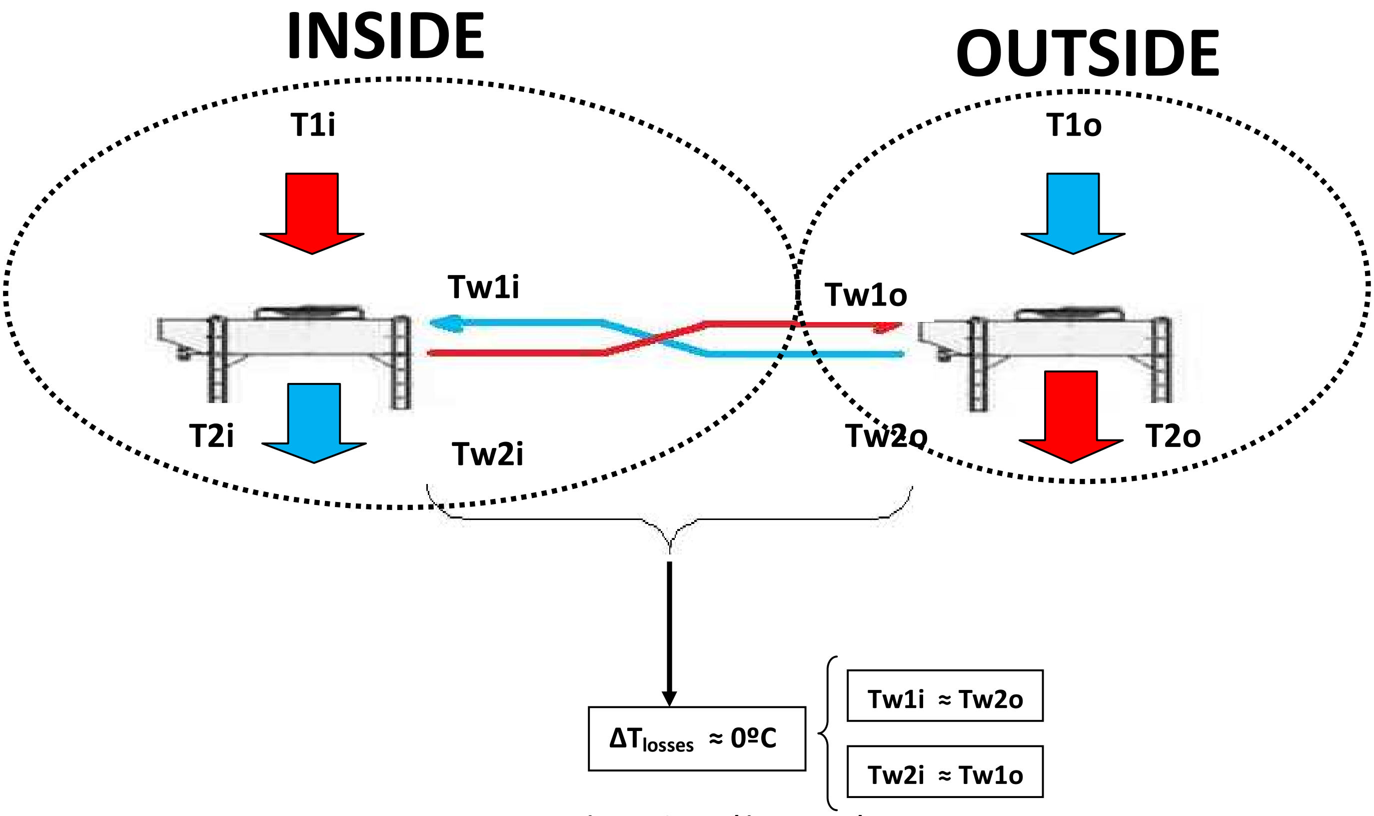


Figure 50; Working example

THERMAL FOCUS									
OUTSIDE [ºC]			INSIDE [ºC]			|T1-Tw2|[ºC]			
T1o	|Tw1-Tw2|=3[ºC]		T1i	|Tw1-Tw2|=3[ºC]		OUTSIDE	F4	INSIDE	F4
	Tw1o	Tw2o		Tw1i	Tw2i				
-20	-13	-16	8	-16	-13	4	12	21	2
-20	-13	-16	2	-16	-13	4	12	15	3
-20	-13	-16	0	-16	-13	4	12	13	4
-19	-12	-15	8	-15	-12	4	12	20	2
-19	-12	-15	2	-15	-12	4	12	14	3
-19	-12	-15	0	-15	-12	4	12	12	4
-18	-11	-14	8	-14	-11	4	12	19	3
-18	-11	-14	2	-14	-11	4	12	13	4
-18	-11	-14	0	-14	-11	4	12	11	4
-17	-10	-13	8	-13	-10	4	12	18	3
-17	-10	-13	2	-13	-10	4	12	12	4
-17	-10	-13	0	-13	-10	4	12	10	5
-16	-9	-12	8	-12	-9	4	12	17	3
-16	-9	-12	2	-12	-9	4	12	11	4
-16	-9	-12	0	-12	-9	4	12	9	5
-15	-8	-11	8	-11	-8	4	12	16	3
-15	-8	-11	2	-11	-8	4	12	10	5
-15	-8	-11	0	-11	-8	4	12	8	6
-14	-7	-10	8	-10	-7	4	12	15	3
-14	-7	-10	2	-10	-7	4	12	9	5
-14	-7	-10	0	-10	-7	4	12	7	7
-13	-6	-9	8	-9	-6	4	12	14	3
-13	-6	-9	2	-9	-6	4	12	8	6
-13	-6	-9	0	-9	-6	4	12	6	8
-12	-5	-8	8	-8	-5	4	12	13	4
-12	-5	-8	2	-8	-5	4	12	7	7
-12	-5	-8	0	-8	-5	4	12	5	9
-11	-4	-7	8	-7	-4	4	12	12	4
-11	-4	-7	2	-7	-4	4	12	6	8
-11	-4	-7	0	-7	-4	4	12	4	12
-10	-3	-6	8	-6	-3	4	12	11	4

-10	-3	-6	2	-6	-3	4	12	5	9
-10	-3	-6	0	-6	-3	4	12	3	15
-9	-2	-5	8	-5	-2	4	12	10	5
-9	-2	-5	2	-5	-2	4	12	4	12
-9	-2	-5	0	-5	-2	4	12	2	23
-8	-1	-4	8	-4	-1	4	12	9	5
-8	-1	-4	2	-4	-1	4	12	3	15
-8	-1	-4	0	-4	-1	4	12	1	44
-7	0	-3	8	-3	0	4	12	8	6
-7	0	-3	2	-3	0	4	12	2	23
-7	0	-3	0	-3	0	4	12	0	-
-6	1	-2	8	-2	1	4	12	7	7
-6	1	-2	2	-2	1	4	12	1	44
-6	1	-2	0	-2	1	4	12	-1	-
-5	2	-1	8	-1	2	4	12	6	8
-5	2	-1	2	-1	2	4	12	0	-
-5	2	-1	0	-1	2	4	12	-2	-
-4	3	0	8	0	3	4	12	5	9
-4	3	0	2	0	3	4	12	-1	-
-4	3	0	0	0	3	4	12	-3	-
-3	4	1	8	1	4	4	12	4	12
-3	4	1	2	1	4	4	12	-2	-
-3	4	1	0	1	4	4	12	-4	-
-2	5	2	8	2	5	4	12	3	15
-2	5	2	2	2	5	4	12	-3	-
-2	5	2	0	2	5	4	12	-5	-
-1	6	3	8	3	6	4	12	2	23
-1	6	3	2	3	6	4	12	-4	-
-1	6	3	0	3	6	4	12	-6	-!
0	7	4	8	4	7	4	12	1	44
0	7	4	2	4	7	4	12	-5	-
0	7	4	0	4	7	4	12	-7	-

Table 56; working range

K) Fins pitch factor (F5)

Pitch (mm)	2,1	2,5	3,2	4,2
F5	0,93	1,00	1,09	1,25

Table 57; Factor F5 table

The pinch selected is 2,1mm, so **F5 = 0,93**

L) Altitude Factor (F6)

Altitude (m)	0	200	400	600	800	1000	1200	1400	1600	1800	2200	2600	3000
F6	1,00	1,01	1,03	1,04	1,06	1,07	1,09	1,11	1,12	1,14	1,18	1,22	1,26

Table 58; Factor F6 table

Gävle has a mean altitude of 16m, so **F6 = 1**

5.4 Dry cooler calculation. Outside.

Now it disposes of the particular and general condition to define the dry cooler to Area 5, so it shows below a table joining the data;

FACTOR	Valor		Description
	Option 1	Option 2	
F1	**1,2**	-	Glycol mixture factor
F2	0,87	**0,91**	Entering Tº factor
F3	**1,14**	-	Water-rise |Tw2i-w1i| factor
F4	**4,5**	**11**	Water-Air |T1i-Tw2i| factor
F5	**0,93**	-	Fins pinch factor
F6	**1**	-	Altitude factor
$Q_{real\ area1}$	**7,4(kW)**		Heat real power required. Area 1
$Q_{real\ area2}$	**1,9(kW)**		Heat real power required. Area 2
$Q_{real\ area3}$	**6,3(kW)**		Heat real power required. Area 3
$Q_{real\ area4}$	**2,1(kW)**		Heat real power required. Area 4
$Q_{real\ area5}$	**3,2 (kW)**		Heat real power required. Area 5
Q_{TOTAL}	**20,9 (kW)**		Total heat demand to extract

Table 59; new equipment characteristics

In the **table 59** there are two column for "valor", that is because sometimes it exists the possibility of finding more than one case that can be the worst.

It is possible to see on the **table 59** that the F4 factor has an especial influence on the dry cooler selection, F4 can increase the size from 4 to 11 times bigger. Therefore, it is necessary to make the two different calculations;

Table capacity > Qreal*F1*F2*F3*F4*F5*F6

- ***F4= 4,5*** → Table capacity > 20,9·1.2·0,91·1,14·4,5·0,93·1 → Table capacity > **109 kW**
- ***F4= 11*** → Table capacity > 20,9·1.2·0,91·1,14·11·0,93·1 → Table capacity > **266 kW**

Model	Capacity (Power)(kW)		Air flow (m3/h)		Fluid flow (m3/h)		Pressure drops (kPa)		Noise level (dB(A) 10m)		Fan-motors (400V/3ph/50Hz)							Surface (m2)	Tube vol. (dm3)	Weight (kg)	Connection ø (inch)	COP	
											n	rpm		W		A							
	Δ	Y	Δ	Y	Δ	Y	Δ	Y	Δ	Y		Δ	Y	Δ	Y	Δ	Y					Δ	Y
WR1180.B	33,7	26,1	9000	6600	6,4	5	20	13	28	23	1	440	340	310	170	1,2	0,48	113	22	157	1"1/2	109	154
WR1380.B	105,0	78,1	27000	18900	20	14,9	58	35	33	28	3	440	340	310	170	1,2	0,48	339	65	449	2"1/2	113	153
GR1680.A	182,8	152,3	58200	45000	34,9	29,1	43	31	36	31	6	440	340	310	170	1,2	0,48	633	105	949	2"1/2	98	149
WR2180.B	64,0	50,3	17200	12800	12,2	9,6	24	16	31	26	2	445	340	310	170	1,2	0,48	205	49	264	2"1/2	103	148
WR2280.B	132,6	104,0	34400	25600	25,3	19,8	68	45	34	29	4	445	340	310	170	1,2	0,48	409	84	505	2"1/2	107	153
WR2380.B	198,7	156,0	51600	38400	37,9	29,7	66	43	36	31	6	445	340	310	170	1,2	0,48	613	124	746	3"	107	153
WR2480.B	254,9	200,1	68800	51200	48,6	38,1	20	13	37	32	8	445	340	310	170	1,2	0,48	817	175	1071	4"	103	147
GR2580.B	317,7	251,2	86000	64000	60,6	47,9	17	11	38	33	10	445	340	310	170	1,2	0,48	1423	255	1340	4"	102	148
GR2680.B	390,0	308,1	10320(	76800	74,4	58,7	29	19	38	33	12	445	340	310	170	1,2	0,48	1707	299	1744	4"	105	151
GR2780.B	462,2	364,9	12040(	89600	88,2	69,6	45	30	38	33	14	445	340	310	170	1,2	0,48	1992	343	2031	4"	106	153

Table 60; exchager selection

5.5 Resume table. Selected exchangers

EXHANGER	Model	Capacity (Power)(kW)	Fluid flow (m3/h)	Fan-motors (400V/3ph/50Hz)			COP
				nº fans	Consum [kW/fan]	Total consum [W]	
Area 1	WR1480.B	105,7	20,1	4	0,17	0,68	155
Area 2	WR1263.A	24,3	4,6	2	0,07	0,14	174
Area 3	WR1480.B	105,7	20,1	4	0,17	0,68	155
Area 4	WR1263.B	26,1	5	2	0,07	0,14	186
Area 5	WR1363.B	39,5	7,5	3	0,07	0,21	188
TOTAL INDOOR	-	**301,3**	57,3	-	-	1,85	**163**
Outdoor	GR2680.B	308	58,7	12	170	2040	151
TOTAL OUTDOOR	GR2680.B	**308**	58,7	12	170	2040	**151**

Table 61; Resume of the different exchangers selected

The electrical motor connection selected is Y.

To calculate the total COP, it has been realized a balance of the total energy consumed by all the internal exchangers. After that, the total COP has been calculated multiplying the contribution percentage of each exchanger on the total COP to after that sum all the partial COPs.

It is logical to think that the COP of all the system is the lesser COP between both indoor and outdoor, so COP = 160.

6 PUMP

6.1 Introduction

An important aspect to keep low the total COP is the pump selected. All the time it is necessary to take into account that the internal and the external dry coolers, fans, pump… are going to be considered in the same total system for the calculation of the global COP. So if it is selected a pump with more power than the necessary, the COP is going to decrease.

6.2 System schematic

To carry out the pump calculation it is necessary know with quite detail the complete system and which are the pipe grid, heat flux distributions, refrigerant flux and how large are the distribution pipes.

Previously all that, the pump emplacement must be chosen;

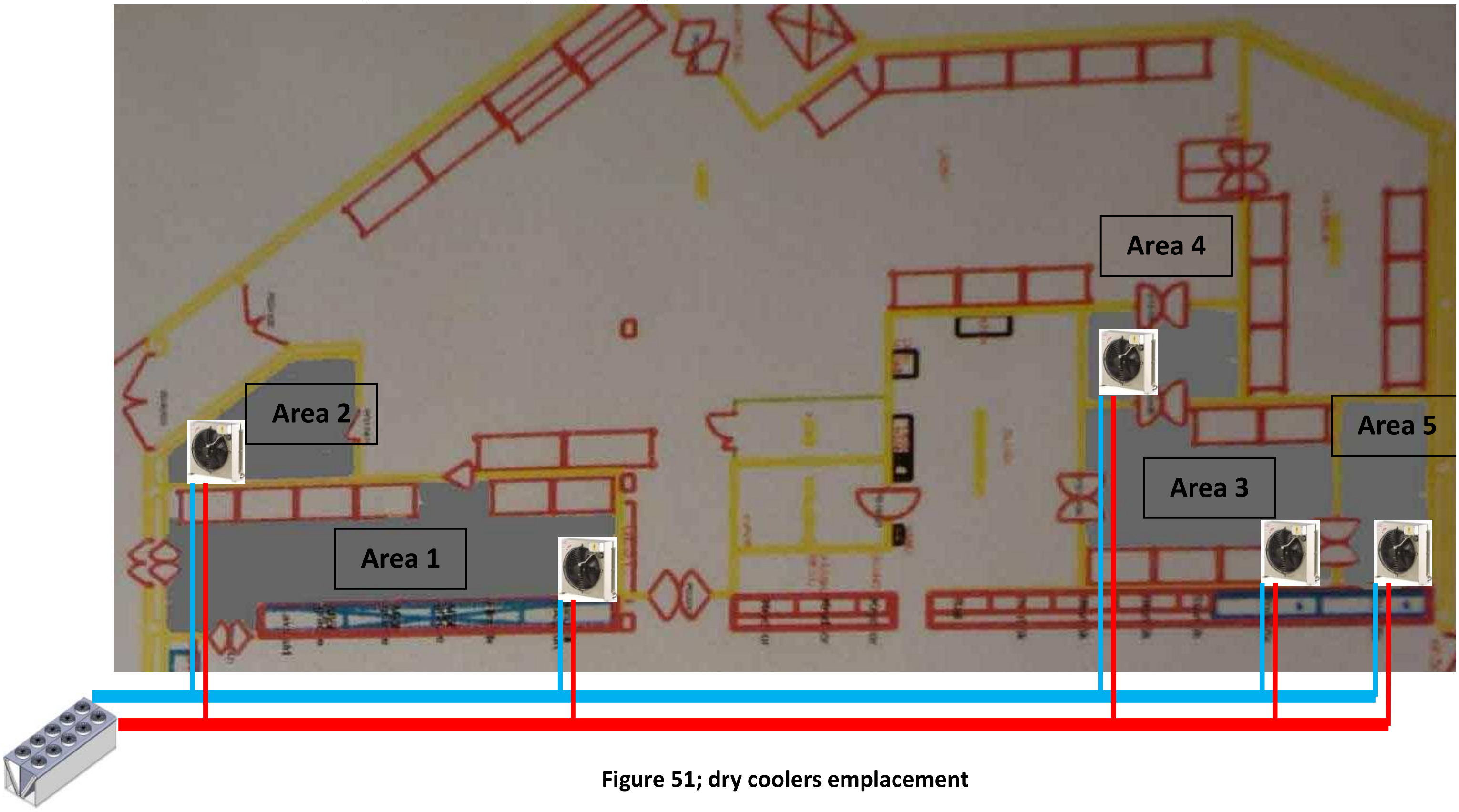


Figure 51; dry coolers emplacement

So, It can be seen on the **figure 51** that the external exchanger is placed on the west side of Willys, on the roof. The calculations realized below are going to carry out with the technical conditions that appear due to the last emplacement specifications. To build the system it is not important to change the emplacements, because the system conditions are going to vary just a little.

Below it is shown a schematic figure of the system (see **figure 52**) with its correspondent table (see **table 62**) where it can be found the geometrical characteristics of the pipe net, heat and refrigerant fluxes and an illuminating schema of installation system;

Line	Exchanger Model	Cooling Power [kW]	Refrigerant flux $[m^3/h]\|[m^3/s]$	Minimum diameter $[mm^2]$	Longitude [m]
1	GR2680.B	308	58,7 | 0,016	76	55
2	WR1480.B	105,7	20,1 | 0.0055	76	3
3	WR1263.A	24,3	4,6 | 0.0012	38	7
4	WR1480.B	105,7	20,1 | 0.0055	76	3
5	WR1263.B	26,1	5 | 0.0014	51	7
6	WR1363.B	39,5	7,5 | 0.002	51	3
1'	go out	326	57,3 | 0.016	76	55
2'	go out	105,7	20,1 | 0.0055	76	3
3'	go out	24,3	4,6 | 0.0012	38	7
4'	go out	105,7	20,1 | 0.0055	76	3
5'	go out	26,1	5 | 0.0014	51	7
6'	go out	39,5	7,5 | 0.002	51	3
A	*Auxiliary line*	*6,5*	*1,5* | 0.0014	*30*	-
		Total indoor flux	57,3 |	**L total**	156
		Total outdoor flux	58,7 |		

Table 62; system characteristics

If it is compared the total indoor flux and the total outdoor flux It can be appreciated that the indoor flux is bigger than the outdoor flux. Then the total cooling demand of the indoor exchangers may be satisfied with the outdoor one.

But there is a problem, it exists an overload of flux, about (58,7-57,3)= ≈ **1,5 $[m^3/h]$**, so to solve this problem it is necessary to install a bypass valve.

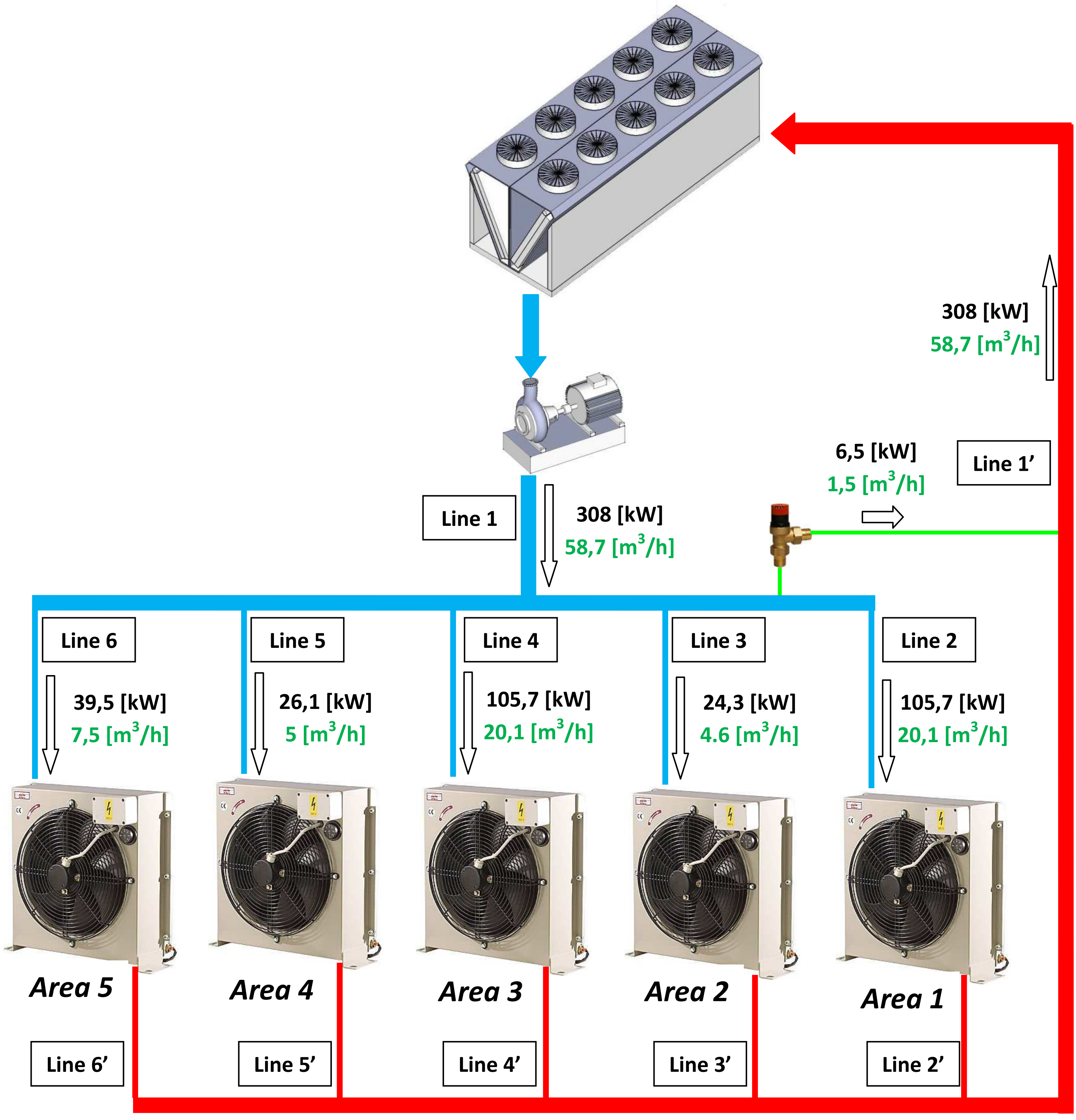


Figure 52; System schematic. Cooling and refrigerant fluxes.

6.3 Calculation pump

To calculate the pump it is going to be used the standard procedure normalized to do it.

So the equation purposed is the below one;

$$\boxed{P = H_b * \rho * g * Q \ [W]}$$

Equation 4; power of the pump

Where;

- H_b; pump height [m]
- ρ; refrigerant density [kg/m^3]
- g; gravity [m/s^2]
- Q; caudal [m^3/s]

From the equation we know;

- **g= 9,8 [m/s^2]**

  The refrigerant density can be found on the characteristics graphic of the refrigerant;

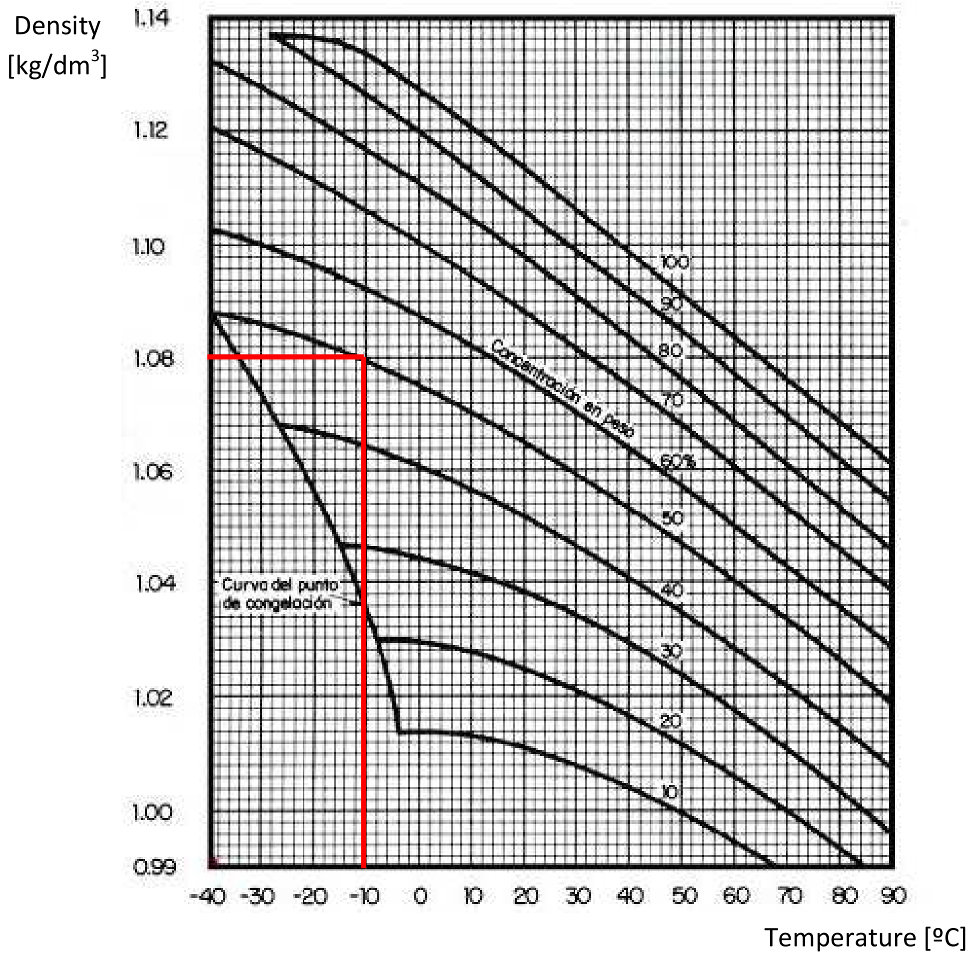


Figure 53; Density of a glycol + water mixture

As it can be appreciated in the **figure 53** for a T= -10ºC and a refrigerant with a 50% mix of glycol;

- **ρ= 1080 [kg/m^3]**

The caudal that has been taken is the one for the maximum demand, and it can be taken from the **table 62**;

- **Q= 58,7 [m^3/h] = 0,0163[m^3/s]**

Now it is only needed to obtain the pump height to can continue with the final calculation of the pump.

6.3.1 Pump height (Hb)

The pump height is the factor that has the higher influence on the selection of the pump, so it is convenient to do an exact calculation.

The equation used to this is the Bernoulli equation, applied on a closed pipes system with pressure;

$$\underbrace{\frac{P_1}{\rho \cdot g} + \frac{V_1^2}{2 \cdot g} + Z_1 + H_b}_{\boxed{\text{INLET}}} = \underbrace{\frac{P_2}{\rho \cdot g} + \frac{V_2^2}{2 \cdot g} + Z_2 + h_f}_{\boxed{\text{OUTLET}}}$$

Equation 5; Bernoulli equation

Where;

- P_1 ; Fluid pressure at inlet system [Pa]
- $V^2{}_1$; Fluid velocity at inlet system [m/s]
- Z_1 ; Extraction height of fluid respect at the referent point [m]
- P_2 ; Fluid pressure at outlet system [Pa]
- $V^2{}_2$; Fluid velocity at outlet system [m/s]
- Z_2 ; Expulsion height of fluid respect at the referent point [m]
- H_b; Height pump [m]
- h_f; losses across the pipes [m]

To can solve the system modeled, it is necessary to make a referent point and see what happen on the inlet and the outlet of the fluid.

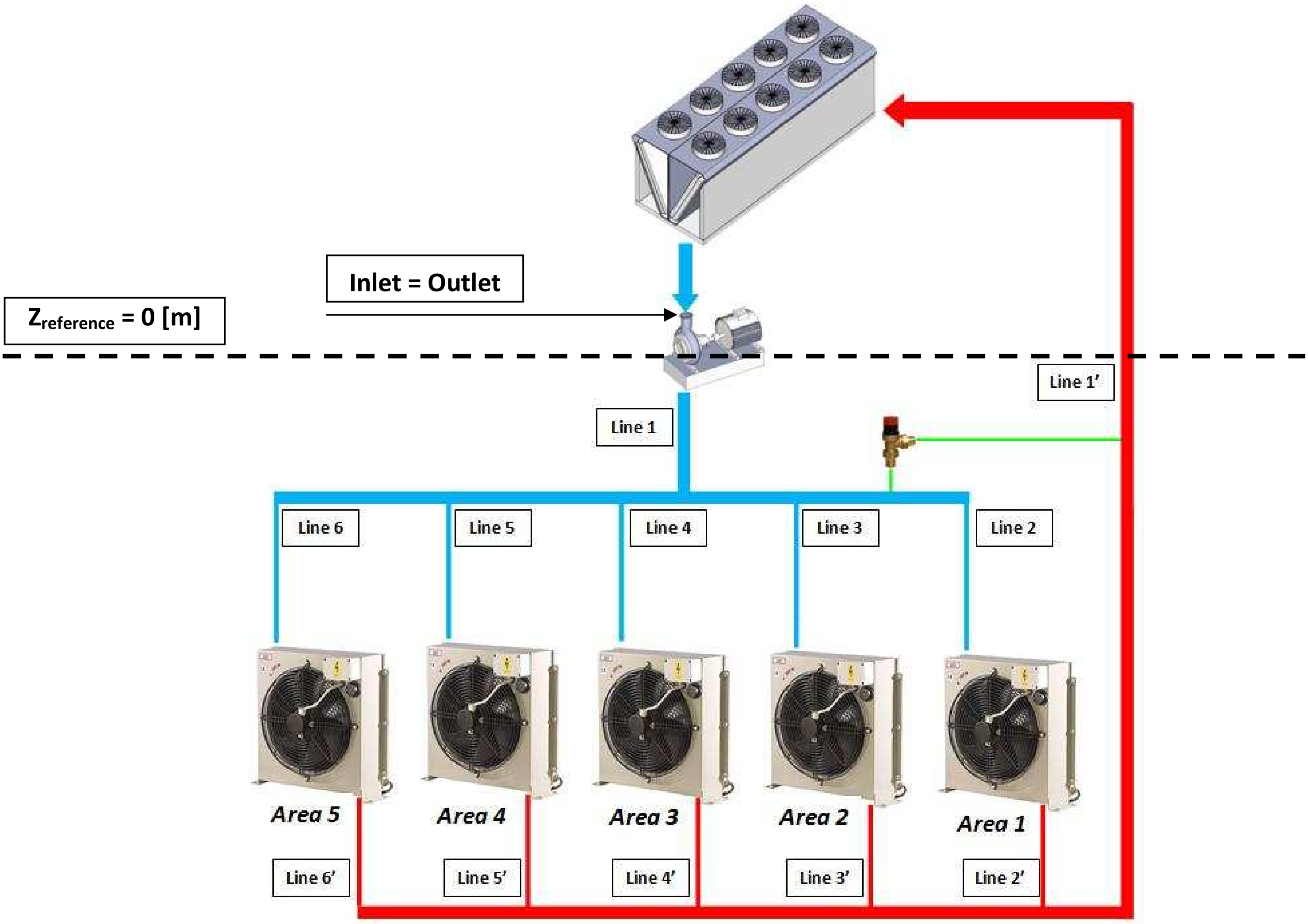


Figure 54; System reference point (Z). Inlet and outlet fluid.

Bernoulli formula is a general equation applicable for open and closed systems. On particular our system is closed, therefore there is not an inlet and an outlet differenced, so Inlet = Outlet. This system is going to simplify a lot the operations made on the **equation 5**, it is like that because;

- $P_1 = P_2$
- $V_1 = V_2$
- $Z_1 = Z_2$

It is operate the Bernoulli equation to get Hb on function of the others variables;

$$H_b = \frac{P_2}{\rho \cdot g} + \frac{V_2^2}{2 \cdot g} - (\frac{P_1}{\rho \cdot g} + \frac{V_1^2}{2 \cdot g} + Z_1) + Z_2 + h_f$$

Now, if it is replaced on the last equation the assumptions applicable to a system closed are obtained;

$$H_b = \frac{P_2}{\rho \cdot g} + \frac{V_2^2}{2 \cdot g} - \frac{P_2}{\rho \cdot g} - \frac{V_2^2}{2 \cdot g} - Z_2 + Z_2 + h_f$$

$$\boxed{H_b = h_f}$$

It has been found a result very simplified as was hoped. The Bernoulli equation has been simplified still deducing that the height of the pump is exactly the same value as the pipe losses.

6.3.2 Losses across the pipes (h_f)

There are two different types of losses inside the pipes;

- *Load losses through pipes*; this kind of losses are due to friction between the fluid and the internal face of the pipes.

- *Load losses on punctual elements*; this other kind of losses are produced because of the elements installed on the system. They present much opposition at the circulation of the fluid.

Losses are measures in “m.c.a”, so instead of to have units on kPa it has to be done the conversion from m.c.a. to kPa, with the next relation; 1m.c.a = 9,81 kPa.

To calculate the losses it is going to be used the next formula;

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g}$$

Equation 6; friction losses

- f: friction coefficient
- L_{eq}; equivalent longitude [m]
- V; flux velocity [m/s]
- D; internal diameter of pipe [m]
- g; gravity [m/s^2]

As it is shown in the **equation 6** h_f is in function of different variables. These variables change with the different geometry of each pipe. So it must be separated the calculation of each pipe branch.

A) Line 1- Line 1';

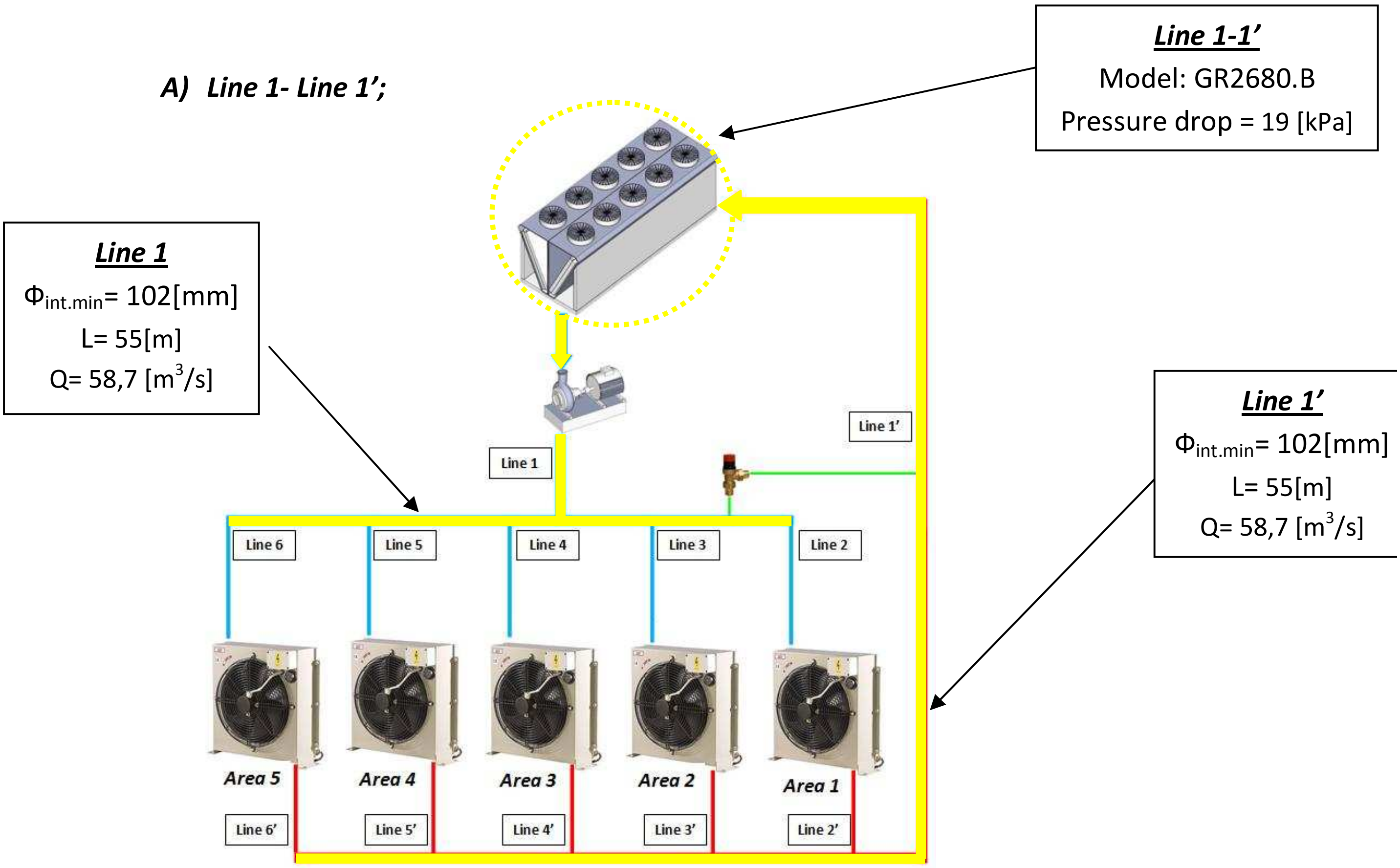


Figure 55; specifications line 1/line 1'

With these specifications it can start to solve the **equation 5,** the one of the losses across the pipes;

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g}$$

It is known;

- **g= 9,81 [m/s^2]**

- **D= 102 [mm] = 0'102 [m]**

For the flux velocity, it must be applied the caudal formula;

$$Q = A \cdot V \;\; [\mathrm{m^3/s}]$$

$$V = \frac{Q}{A} \; [\mathrm{m/s}]$$

Equation 7; caudal formula

Where;
Q = 58,7 [m^3/h] = 0,0163 [m^3/s]

It has been selected an internal diameter to the main pipes (line 1-1') of 102mm, so;

A = π·(D^2/4) = π·(102^2/4)= 8171 mm^2 = 8,2·10^{-3} m^2

$$V = \frac{0{,}0163}{8{,}2 \cdot 10^{-3}} \approx 2[\mathrm{m/s}]$$

- **V= 2 [m/s]**

It is very important the fact of getting a velocity above zero. This reason is because the velocity influx is quadratically on the pump calculation, so if it is selected an adequate diameter pipe then the power needed of the pump is going to decrease.

Finally the diameter pipe selected to line 1 and 1' is **102mm.**

The next variable that it is going to be calculated is the equivalent longitude. The equivalent pipe longitude is composed by the lineal longitude of the pipes and the accessories installed on it. The valor of the equivalent longitude of the accessories installed can be taken from the accessory datasheet when it will be available (This can be done with the dry cooler). If the equivalent longitude is not available, it can be estimated with a simple formula took from the European normative for pressure circuits. So;

L_{eq}= L_{lineal} + $L_{accesoires}$

Equation 7; equivalent longitude

It is estimated that there are approximately five 90º elbows, the load losses of one elbow is estimate like L_{1elbow}=(L_{lineal}·ϕ_{int})= [(55+55)*0.102]= 11 [m/accessorie] . Also it must be taken into account the external dry cooler.

- Model: GR2680.B → Pressure drop = 19 [kPa] = 19/9,81 [m.c.a]= 1,94 m

Finally;

L_{eq}= L_{lineal} + $L_{accesoires}$= (55+55) + 5*11 + 1'94≈ 167 [m]

So;

- **L_{eq}= 167 [m]**

The last step to solve the calculation of the pump is to know the friction coefficient "f". This coefficient gives an idea about how the fluid behaves across pipes. There are two different possibilities on which the flux can appear across one pipe, these possibilities are laminar flux and turbulent flux.

To determinate what kind of flux is on our pipe, it is necessary calculate the Reynolds number;

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu}$$

Equation 8; Reynolds number

- D; internal diameter of pipe [m]
- V; flux velocity [m/s]
- ρ; density [kg/m^3]
- μ; dynamic viscosity [cP]

A flux is considerate laminar if the N_{Re}< 2000, moreover if the Reynolds number is 2000 < N_{Re} < 4000 it is on the unstable zone and if the N_{Re} > 4000 the flux is considered turbulent.

For applying the Reynolds equation it is known that D=0,102[m], V=2[m/s] and ρ=1080[kg/m^3], the viscosity has been calculated using the graphic of the figure 56;

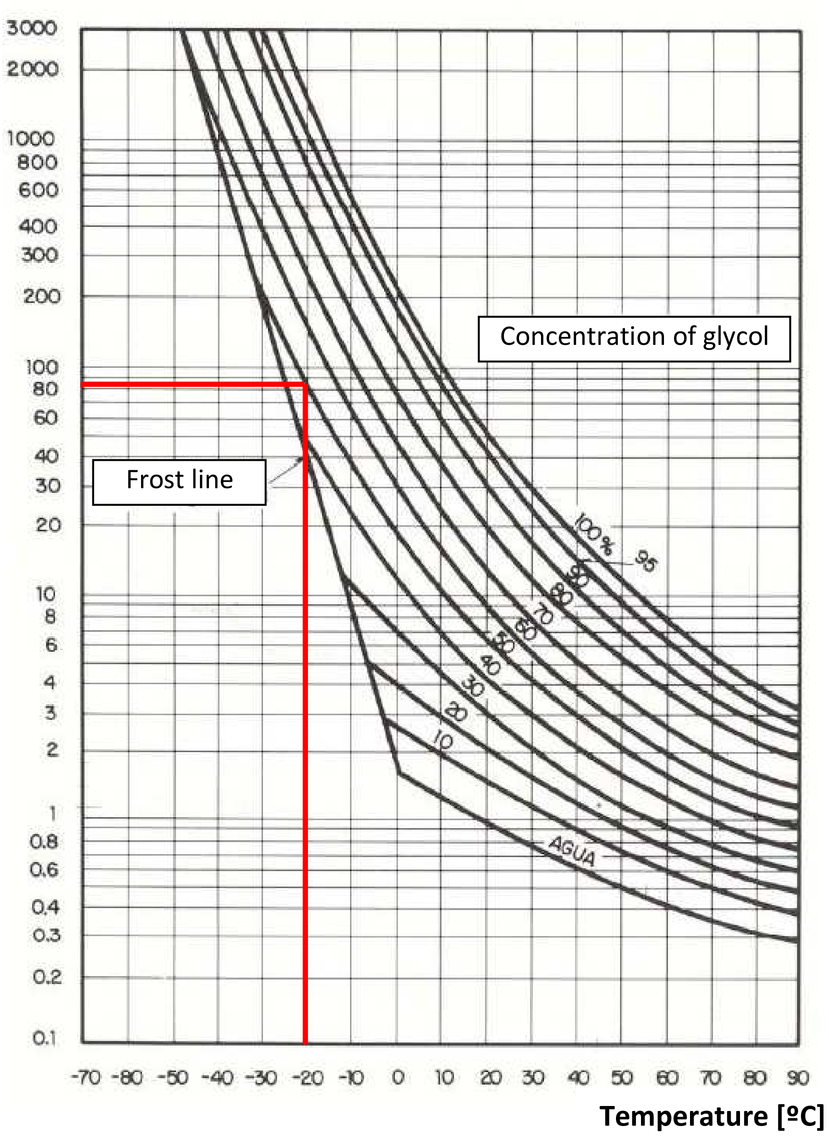


Figure 56; Dynamic viscosity to different glycol-water mixtures

From the **figure 56** it has been selected μ= 85 [cP], but to substitute on the Reynolds equation (equation 8) the viscosity must be on Pa·s;

100 [cP]=0.1 [Pa·s] → (85·0,1)/100 = 0,085 [cP]

- **μ= 0,085 [cp]**

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu} = \frac{0,102 \cdot 2 \cdot 1080}{0,085} \approx 2600$$

On this case 2000 < N_{Re} < 4000, so the system is going to work on the unstable regime. It can be used the Moody diagram (see **figure 57**) to get the friction coefficient;

4

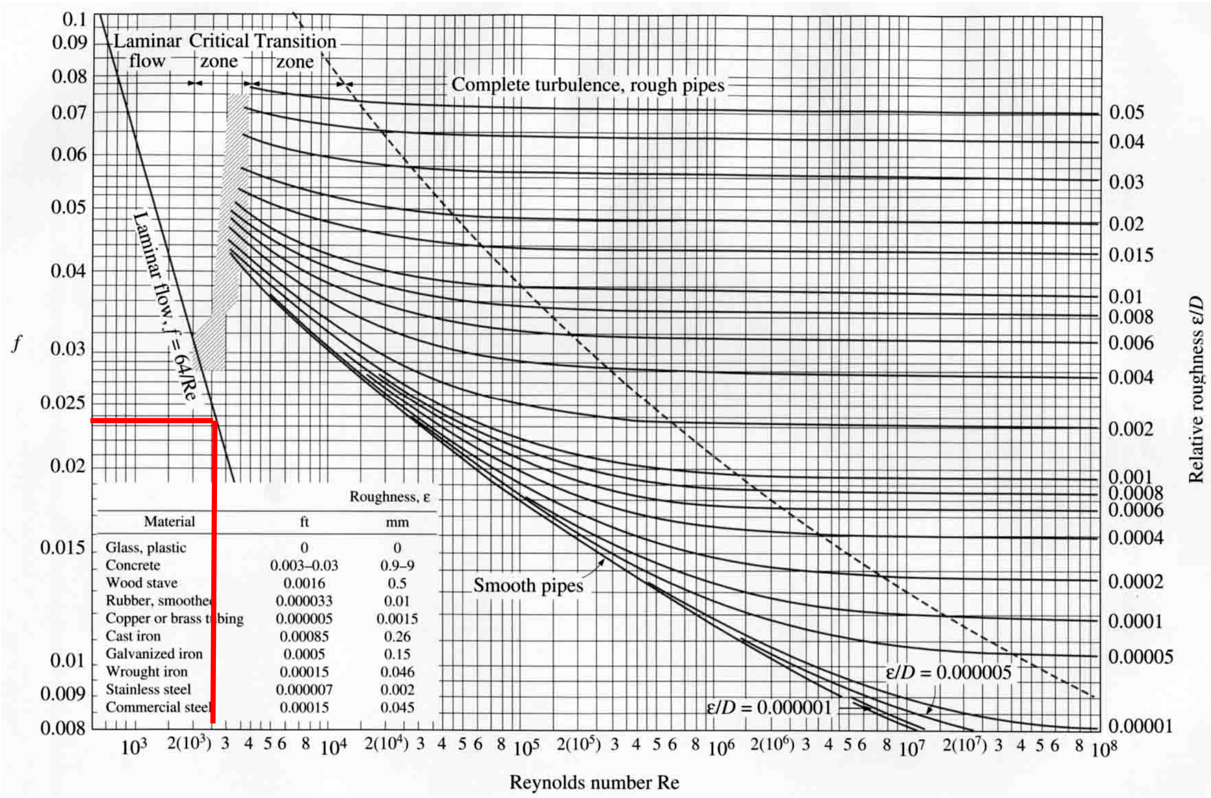


Figure 57; Moody diagram

Finally, if it is used the graphic above (**figure 57**) for a $N_{Re} \approx 2600$, it is obtained a friction coefficient of approximately f≈0,023, so;

- **f= 0,023**

Now, it is available to substitute all the values calculated on the losses pipes equation (**equation 6**);

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g} = \frac{0,023 \cdot 167 \cdot 2^2}{2 \cdot 0,102 \cdot 9,8} = 7,7\ [m]$$

B) Line 2- Line 2';

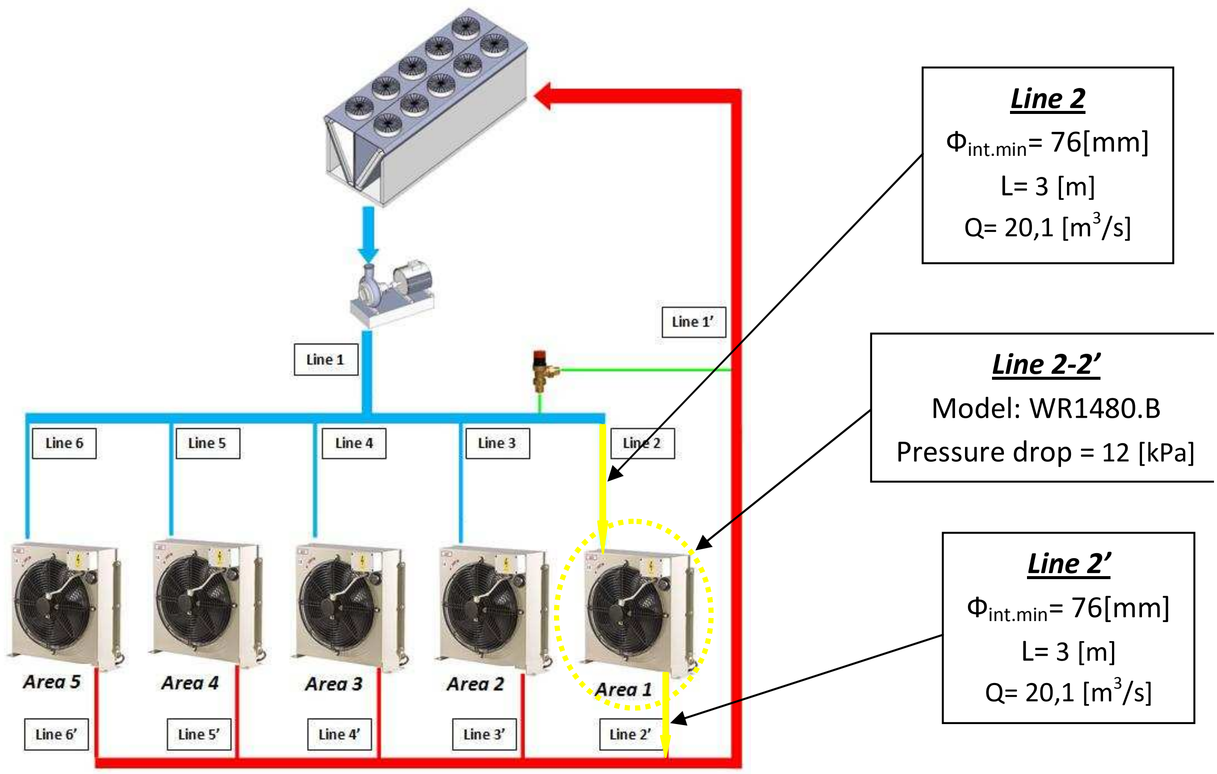


Figure 58; specifications line 2/line 2'

With these specifications it can start to solve the equation to losses across the pipes;

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g}$$

It is known that;

- **g= 9,81 [m/s^2]**
- **D= 76 [mm] = 0,076 [m]**

For the flux velocity, it must be applied the caudal formula;

$$Q = A \cdot V \ \ [\mathrm{m^3/s}]$$

$$V = \frac{Q}{A} \ [\mathrm{m/s}]$$

Where;
Q = 20,1 [m^3/s] = 0,0055 [m^3/h]

It has been selected an internal diameter to the main pipes (line 2-2') of 76mm, so;

A = π·(D^2/4) = π·(76^2/4)= 4536 mm^2 = 4,5·10^{-3} m^2

$$V = \frac{0{,}0055}{4{,}5 \cdot 10^{-3}} = 1{,}2[\mathrm{m/s}]$$

- **V= 1,2 [m/s]**

It is very important to have a velocity above zero. This reason is because the velocity influx is quadratically on the pump calculation, so if it is selected an adequate diameter pipe the power needed of the pump is going to decrease.

Finally the diameter pipe selected to the line 2 and 2' is the minimal one; **76mm.**

The next variable that it is going to be calculated is the equivalent longitude. The equivalent pipe longitude is composed by the lineal longitude of the pipes and the accessories installed on it. The value of the equivalent longitude of the accessories can be taken from the accessory datasheet when it is available (This can be done with the dry cooler). If the equivalent longitude is not available, it can be estimated with a simple formula took from the European normative for pressure circuits. So;

$$L_{eq}= L_{lineal} + L_{accesoires}$$

It is estimated that there are approximately two 90º elbows, the load losses of one elbow is estimate like L_{1elbow}=(L_{lineal}·ϕ_{int})= [(3+3)*0.076]= 0,46 [m/accessorie] . Also it must be taken into account the external dry cooler.

- Model: WR1480.B → Pressure drop = 12 [kPa] = 12/9,81 [m.c.a]= 1,23 m

Finally;

$$L_{eq}= L_{lineal} + L_{accesoires}= (3+3) + 2*0{,}46 + 1'23 \approx 9 \text{ [m]}$$

So;
- **L_{eq}= 9 [m]**

The last step to solve the calculation of the pump is to know the friction coefficient "f". This coefficient gives an idea about how the fluid behaves across the pipes. There are two different possibilities on which the flux can appear across a pipe, laminar flux and turbulent flux.

To determinate what kind of flux is on our pipe, it is necessary calculate the Reynolds number;

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu}$$

- D; internal diameter of pipe [m]
- V; flux velocity [m/s]
- ρ; density [kg/m^3]
- μ; dynamic viscosity [cP]

A flux is considered laminar if the N_{Re}< 2000, moreover if the Reynolds number is 2000 < N_{Re} < 4000 it is on the unstable zone and if N_{Re} > 4000 the flux is considered turbulent.

From the Reynolds equation it know D= 0,076 [m], V= 1,2 [m/s] and ρ= 1080 [kg/m^3], to calculate the viscosity it must find it from the next figure;

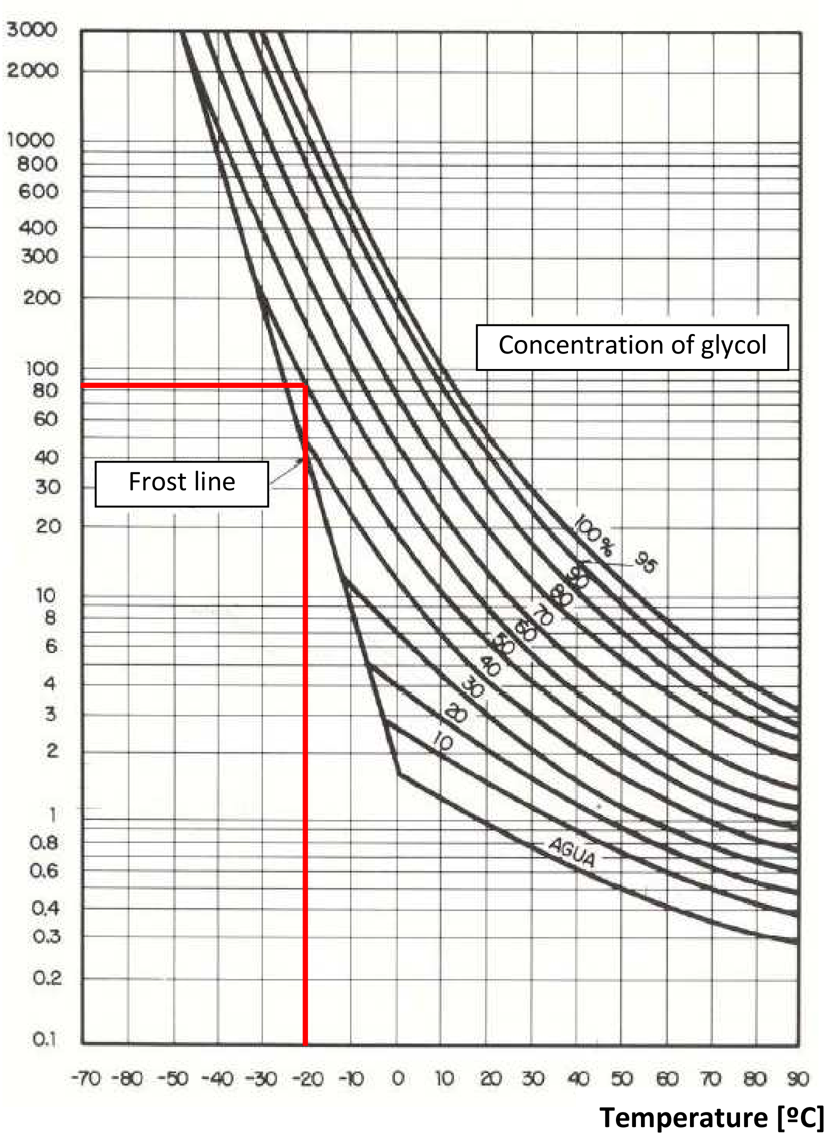


Figure 59; Dynamic viscosity to different glycol-water mixtures

From the figure 59 it has been selected μ= 85 [cP], but to substitute it on the Reynolds equation the viscosity must be on Pa·s;

100 [cP]=0.1 [Pa·s] → (85·0,1)/100 = 0,085 [cP]

- **μ= 0,085 [cp]**

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu} = \frac{0,076 \cdot 1,2 \cdot 1080}{0,085} \approx 1160$$

On this case N_{Re} < 2000, so our system works on the laminar regime. It can be used the Moody diagram to get the friction coefficient;

4

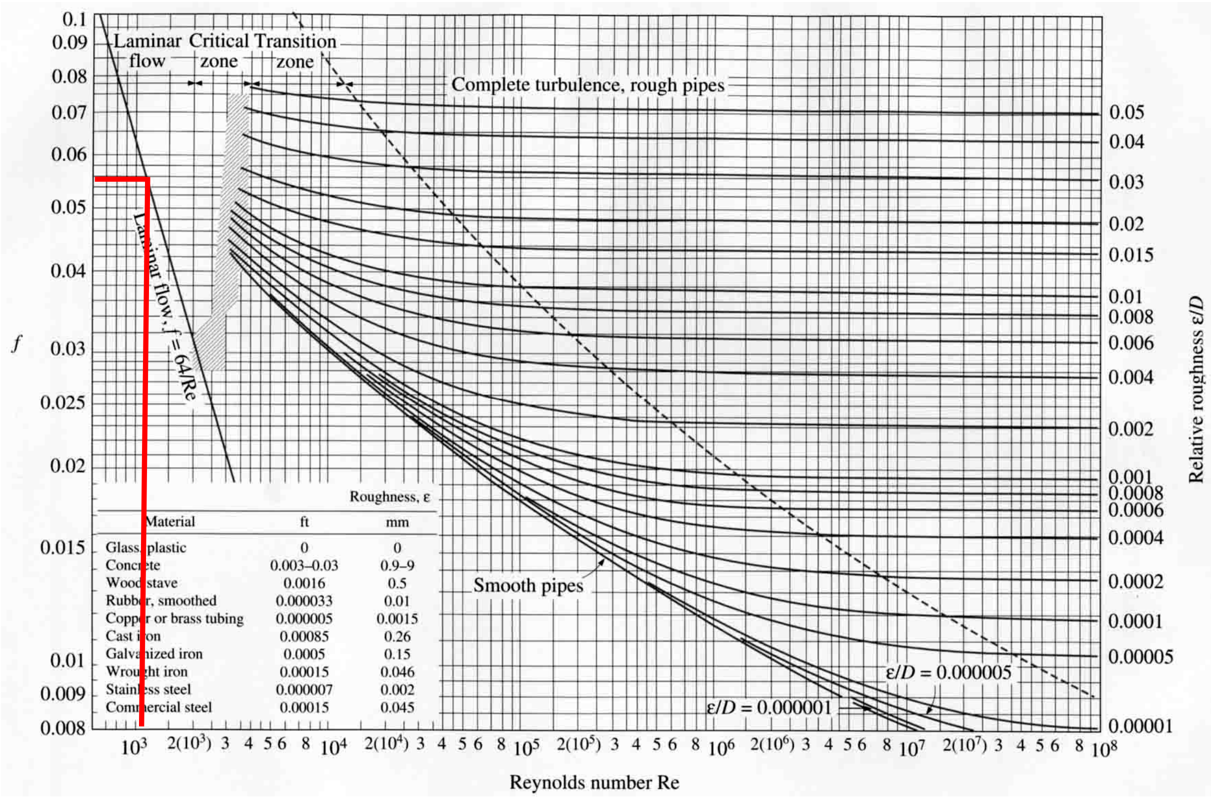


Figure 60; Moody diagram

Finally, if it used the graphic for a N_{Re} ≈ 1160, it is obtained a friction coefficient of around f≈0,055, so;

- **f= 0,055**

Now, it is available to substitute all the values calculated on the losses pipes equation (**equation 6**);

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g} = \frac{0,055 \cdot 9 \cdot 1,2^2}{2 \cdot 0,076 \cdot 9,8} \approx 0,48\ [m]$$

C) Line 3- Line 3';

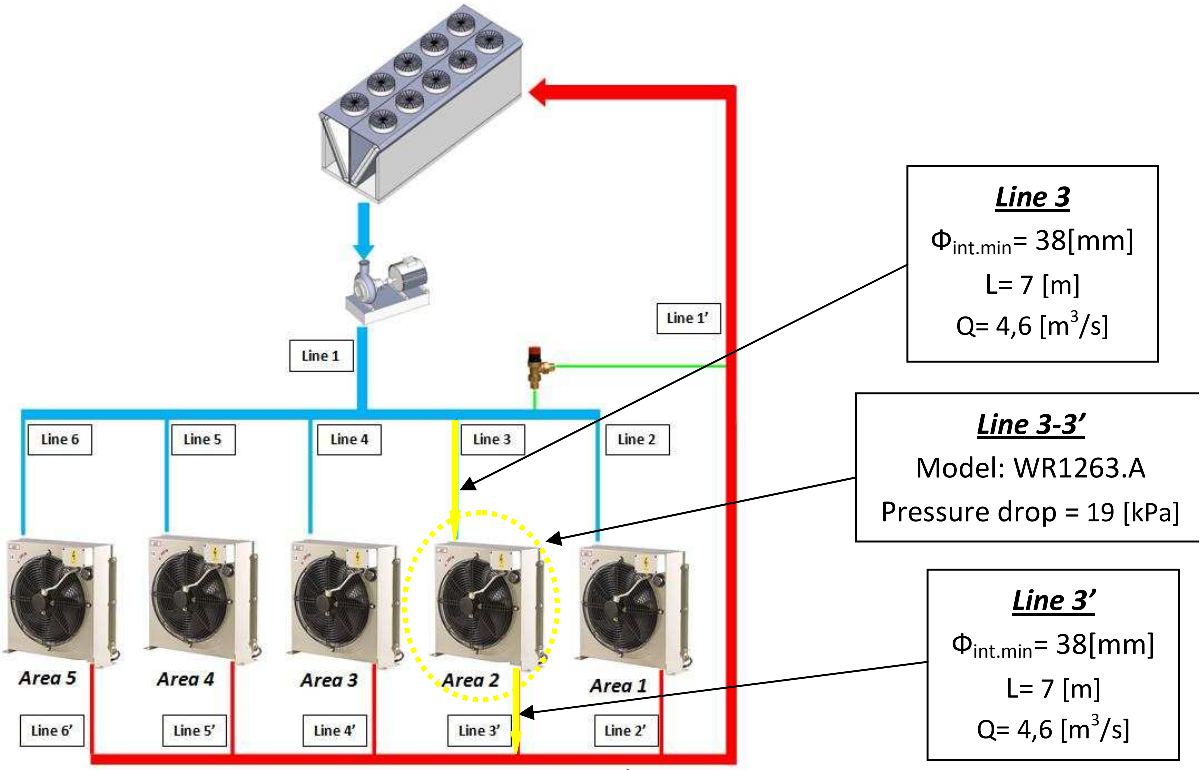


Figure 61; specifications line 3/line 3'

With these specifications it can be solve the equation of the losses across the pipes;

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g}$$

It is known that;

- **g= 9,81 [m/s^2]**
- **D= 38 [mm] = 0,038 [m]**

For the flux velocity, it must be applied the caudal formula;

$$Q = A \cdot V \ \ [\mathrm{m}^3/\mathrm{s}]$$

$$V = \frac{Q}{A} \ [\mathrm{m/s}]$$

Where;

Q = 4,6 [m^3/s] = 0,0013 [m^3/h]

It has been selected an internal diameter to the main pipes (line 3-3') of 38mm, so;

A = π·(D^2/4) = π·(38^2/4)= 1134 mm^2 = 1,2·10^{-3} m^2

$$V = \frac{0{,}0013}{1{,}2 \cdot 10^{-3}} = 1{,}1[\mathrm{m/s}]$$

- **V= 1,1 [m/s]**

It is very important to have a velocity above zero. This reason is because the velocity influx is quadratically on the pump calculation, so if it is selected an adequate diameter pipe the power needed of the pump is going to decrease.

Finally the diameter pipe selected to the line 3 and 3' is the minimal; **38mm.**

The next variable that it is going to be calculated is the equivalent longitude. The equivalent pipe longitude is composed by the lineal longitude of the pipes and the accessories installed on it. The value of the equivalent longitude of the accessories can be taken from the accessory datasheet when it is available (This can be done with the dry cooler). If the equivalent longitude is not available, it can be estimated with a simple formula took from the European normative for pressure circuits. So;

$$L_{eq}= L_{lineal} + L_{accesoires}$$

It is estimated that there are approximately two 90º elbows, the load losses of one elbow is estimate like L_{1elbow}=(L_{lineal}·ϕ_{int})= [(7+7)*0.038]= 0,53 [m/accessorie] . Also it must be taken into account the external dry cooler.

- Model: WR1263.A → Pressure drop = 19 [kPa] = 19/9,81 [m.c.a]= 1,94 m

Finally;

$$L_{eq}= L_{lineal} + L_{accesoires}= (7+7) + 2*0{,}53 + 1'94 \approx 17 \text{ [m]}$$

So;

- **L_{eq}= 17 [m]**

The last step to solve the calculation of the pump is to know the friction coefficient "f". This coefficient gives an idea about how the fluid behaves across the pipes. There are two different possibilities on which the flux can appear across a pipe, laminar flux and turbulent flux.

To determinate what kind of flux is on our pipe, it is necessary to calculate the Reynolds number;

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu}$$

- D; internal diameter of pipe [m]
- V; flux velocity [m/s]
- ρ; density [kg/m^3]
- μ; dynamic viscosity [cP]

A flux is considered laminar if the N_{Re}< 2000, moreover if the Reynolds number is 2000 < N_{Re} < 4000 it is on the unstable zone and if N_{Re} > 4000 the flux is considered turbulent.

From the Reynolds equation it has been known that D= 0,038 [m], V= 1,1 [m/s] and ρ= 1080 [kg/m^3], to calculate the viscosity it must be found it from the **figure 62**;

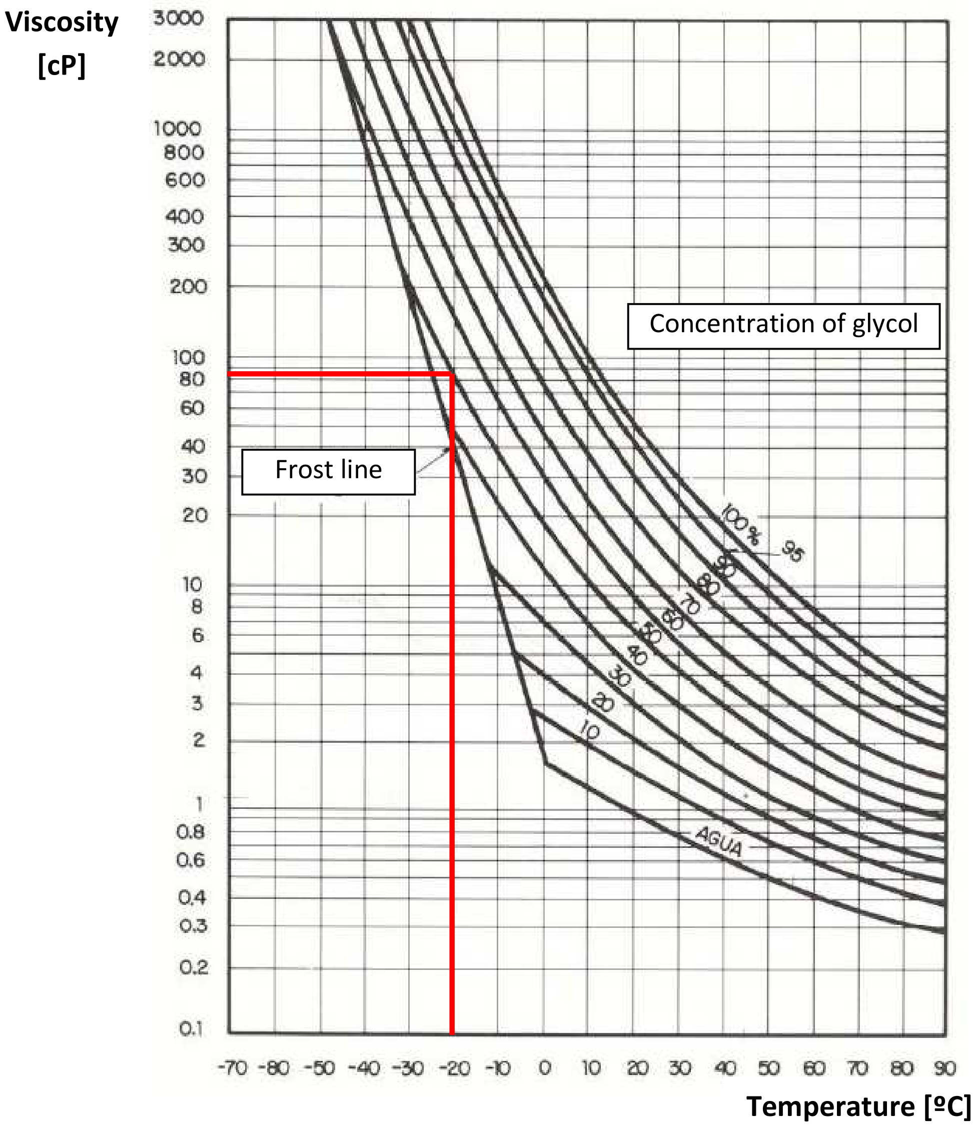


Figure 62; Dynamic viscosity to different glycol-water mixtures

From the figure 62 it has been selected μ= 85 [cP], but to substitute it on the Reynolds equation the viscosity must be on Pa·s;

100 [cP]=0.1 [Pa·s] → (85·0,1)/100 = 0,085 [cP]

- **μ= 0,085 [cp]**

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu} = \frac{0,038 \cdot 1,1 \cdot 1080}{0,085} \approx 550$$

On this case N_{Re} < 2000, so our system works on a laminar regime. It can be used the Moody diagram to get the friction coefficient;

4

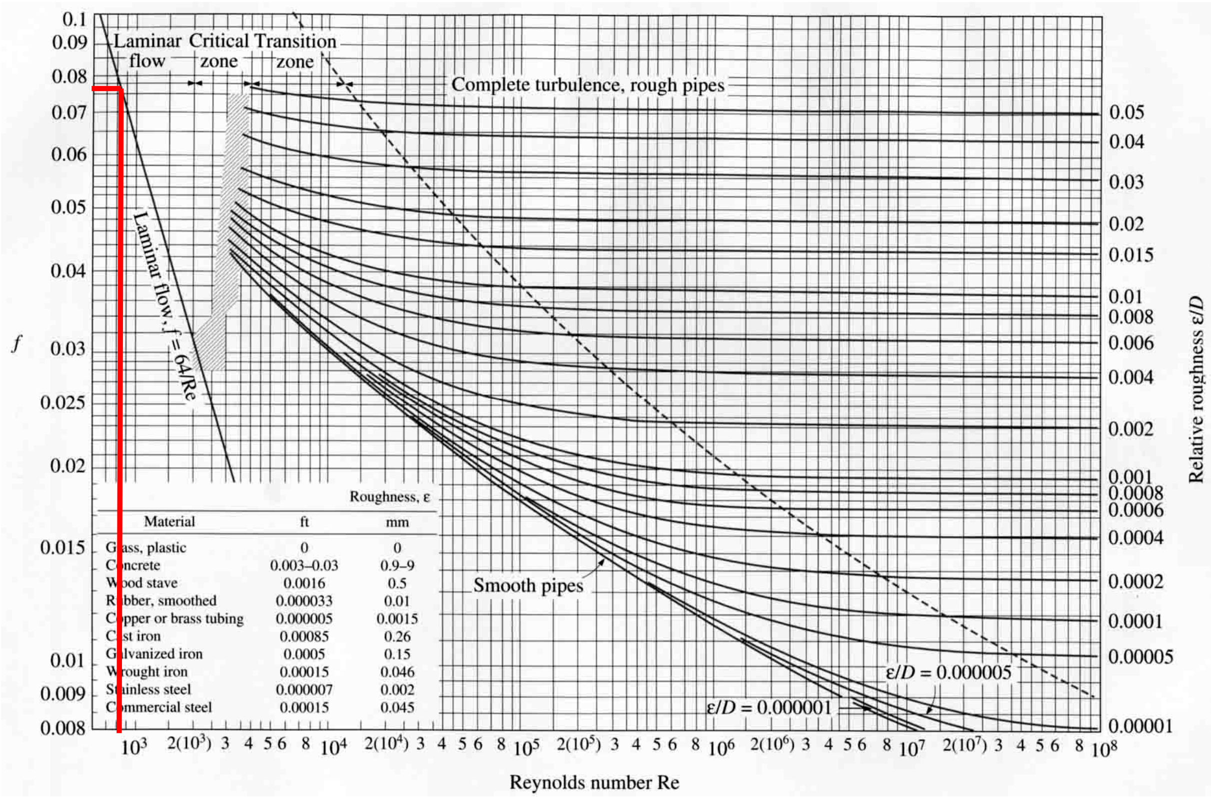


Figure 63; Moody diagram.

Finally, if it used the graphic for a NRe ≈ 550, it is obtained a friction coefficient around f≈0,077, so;

- **f= 0,077**

Now, it is available to substitute all the values calculated on the losses pipes equation (**equation 6)**;

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g} = \frac{0,077 \cdot 17 \cdot 1,1^2}{2 \cdot 0,038 \cdot 9,8} = 2,1\ [m]$$

D) Line 4- Line 4';

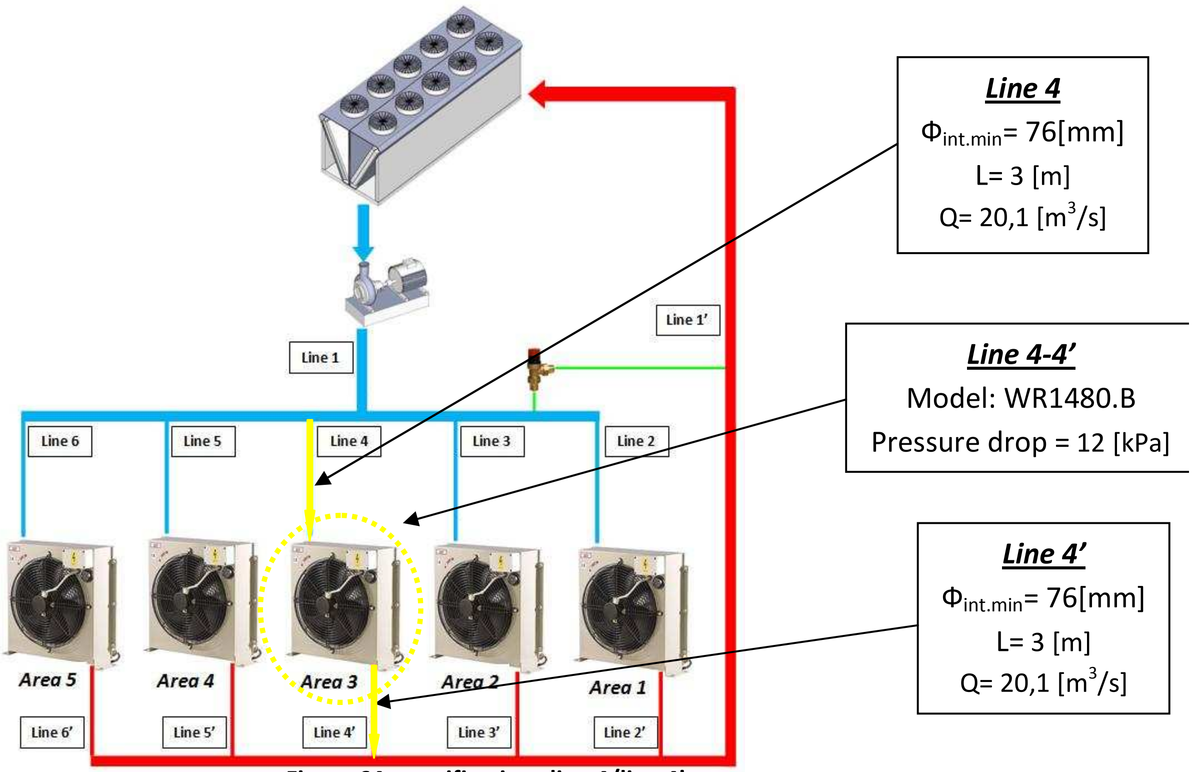


Figure 64; specifications line 4/line 4'

With these specifications it can be started to solve the equation for the losses across the pipes;

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g}$$

Now it is known;

- **g= 9,81 [m/s²]**
- **D= 76 [mm] = 0,076 [m]**

For the flux velocity, it must be applied the caudal formula;

$$Q = A \cdot V \ \ [\mathrm{m}^3/\mathrm{s}]$$

$$V = \frac{Q}{A}\ [\mathrm{m/s}]$$

Where;
Q = 20,1 [m^3/s] = 0,0055 [m^3/h]

It has been selected an internal diameter to main pipes (line 4-4') of 76mm, so;

A = π·(D^2/4) = π·(76^2/4)= 4536 mm^2 = $4{,}5 \cdot 10^{-3}$ m^2

$$V = \frac{0{,}0055}{4{,}5 \cdot 10^{-3}} = 1{,}2[\mathrm{m/s}]$$

- **V= 1,2 [m/s]**

It is very important to have a velocity above zero. This reason is because the velocity influx is quadratically on the pump calculation, so if it is selected an adequate diameter pipe the power needed of the pump is going to decrease.

.

Finally the diameter pipe selected to line 4 and 4' is the minimal; **76mm.**

The next variable that it is going to be calculated is the equivalent longitude. The equivalent pipe longitude is composed by the lineal longitude of the pipes and the accessories installed on it. The value of the equivalent longitude of the accessories can be taken from the accessory datasheet when it is available (This can be done with the dry cooler). If the equivalent longitude is not available, it can be estimated with a simple formula took from the European normative for pressure circuits. So;

$$L_{eq} = L_{lineal} + L_{accesoires}$$

It has been estimated that there are approximately two 90º elbows, the load losses of one elbow is estimate like L_{1elbow}=(L_{lineal}·ϕ_{int})= [(3+3)*0.076]= 0,46 [m/accessorie] . Also it must be taken into account the external dry cooler.

- Model: WR1480.B → Pressure drop = 12 [kPa] = 12/9,81 [m.c.a]= 1,23 m

Finally;

$$L_{eq} = L_{lineal} + L_{accesoires} = (3+3) + 2*0{,}46 + 1'23 \approx 9\ [\mathrm{m}]$$

So;

- **L_{eq}= 9 [m]**

The last step to solve the calculation of the pump is to know the friction coefficient "f". This coefficient gives an idea about how the fluid behaves across the pipes. There are two different possibilities on which the flux can appears across a pipe, laminar flux and turbulent flux.

To determinate what kind of flux is on our pipe, it is necessary to calculate the Reynolds number;

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu}$$

- D; internal diameter of pipe [m]
- V; flux velocity [m/s]
- ρ; density [kg/m^3]
- μ; dynamic viscosity [cP]

A flux is considered laminar if the $N_{Re} < 2000$, moreover if the Reynolds number is $2000 < N_{Re} < 4000$ it is on unstable zone and if $N_{Re} > 4000$ the flux is considered turbulent.

From the Reynolds equation it is known that D= 0,076 [m], V= 1,2 [m/s] and ρ= 1080 [kg/m^3], to calculate the viscosity it must be found in the next figure;

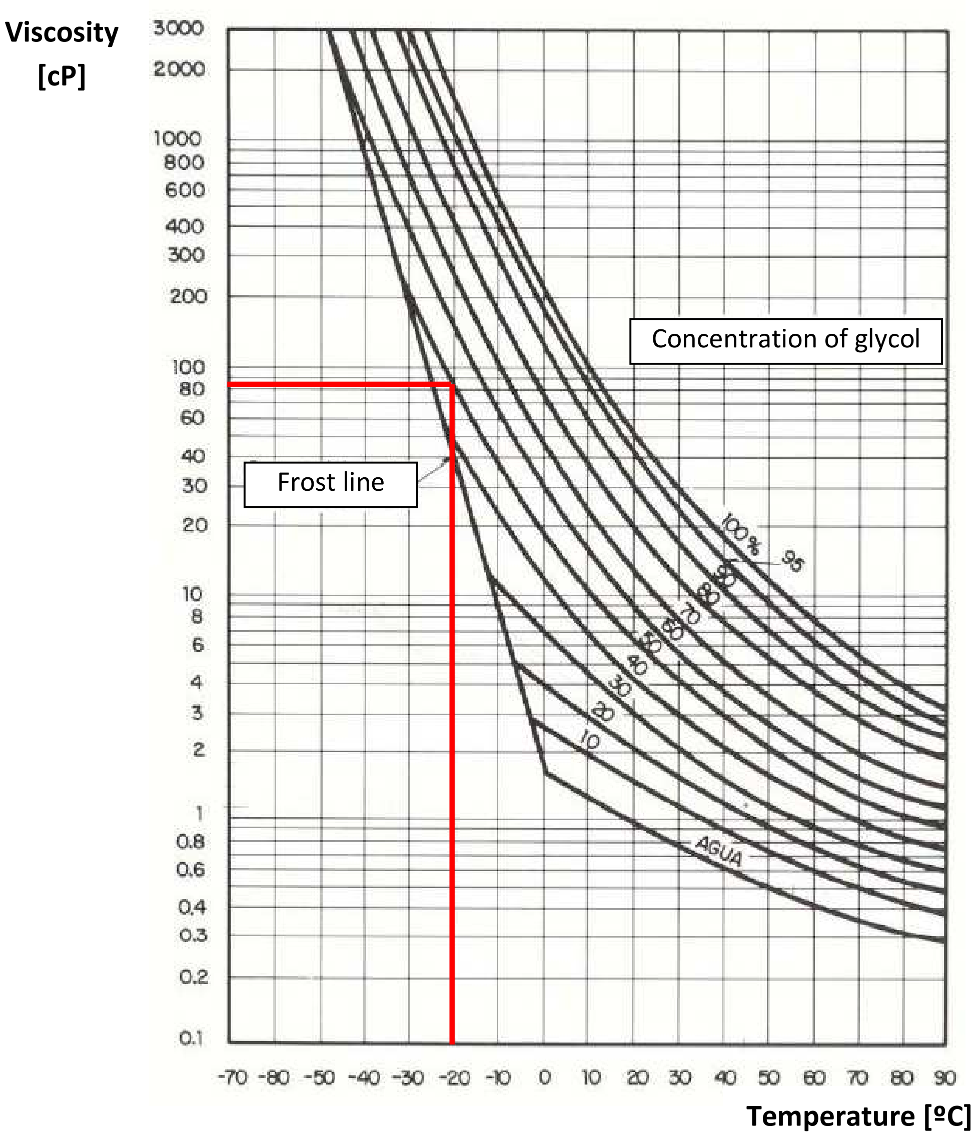


Figure 65; Dynamic viscosity to different glycol-water mixtures

From the **figure 65** has been selected µ= 85 [cP], but to substitute it on the Reynolds equation the viscosity must be on Pa·s;

100 [cP]=0.1 [Pa·s] → (85·0,1)/100 = 0,085 [cP]

- **µ= 0,085 [cp]**

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu} = \frac{0,076 \cdot 1,2 \cdot 1080}{0,085} \approx 1160$$

On this case N_{Re} < 2000, so our system works on the laminar regime. It can be used the Moody diagram to get the friction coefficient;

4

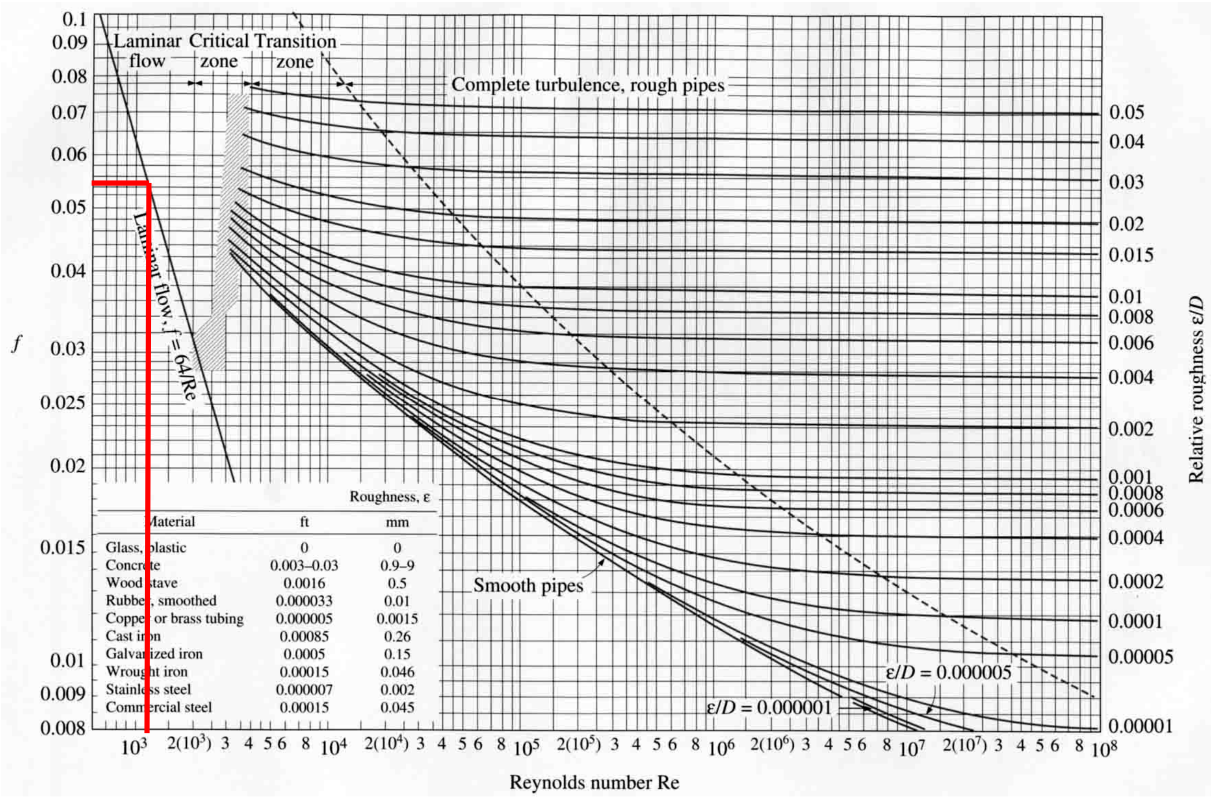


Figure 66; Moody diagram

Finally, if it is used the **figure 66** for a NRe ≈ 1160, it is obtained a friction coefficient of around f≈0,055, so;

- **f= 0,055**

Now, it is available to substitute all the values calculated on the losses pipes equation (**equation 6**);

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g} = \frac{0,055 \cdot 9 \cdot 1,2^2}{2 \cdot 0,076 \cdot 9,8} = 0,48\ [m]$$

E) Line 5- Line 5';

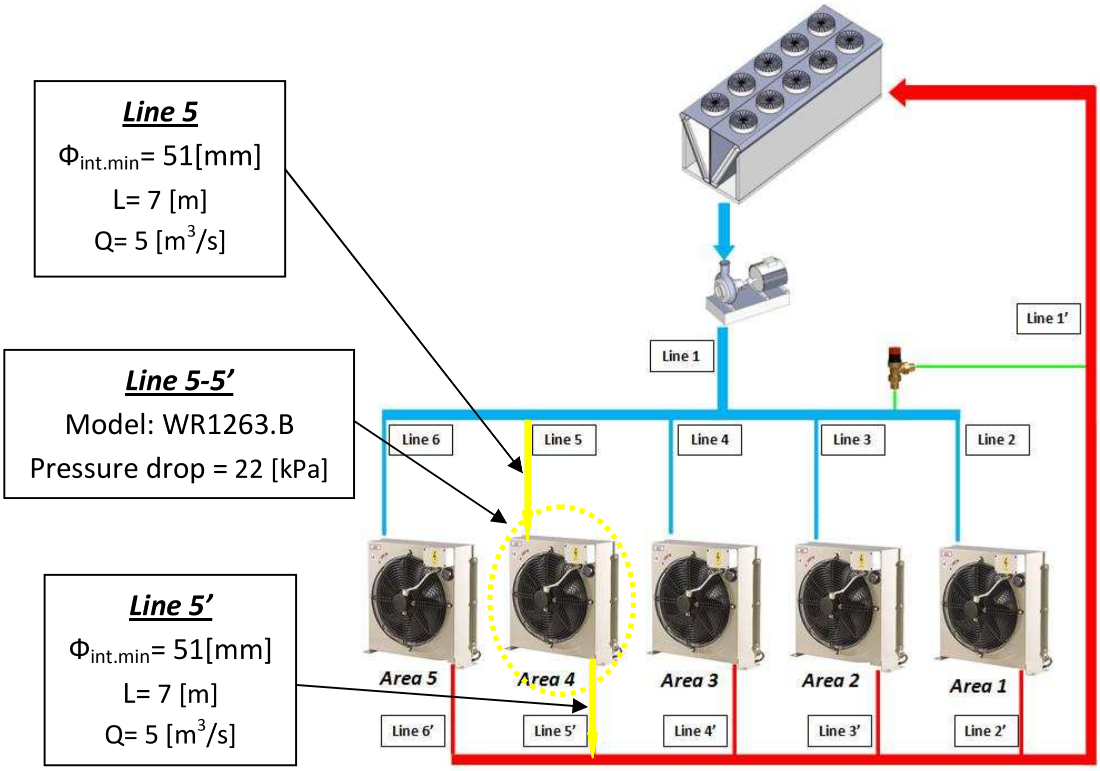


Figure 67; specifications line 5/line 5'

With these specifications it can be solved the equation for losses across the pipes;

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g}$$

It is known that;

- **g= 9,81 [m/s²]**
- **D= 51 [mm] = 0,051 [m]**

For the flux velocity, it must be applied the caudal formula;

$$Q = A \cdot V \ \ [\mathrm{m^3/s}]$$

$$V = \frac{Q}{A} \ [\mathrm{m/s}]$$

Where;

Q = 5 [m^3/s] = 0,0014 [m^3/h]

It has been selected an internal diameter to the main pipes (line 5-5') of 51mm, so;

A = π·(D^2/4) = π·(51^2/4)= 2043 mm^2 = 2,0·10^{-3} m^2

$$V = \frac{0{,}0014}{2{,}0 \cdot 10^{-3}} = 0{,}7[\mathrm{m/s}]$$

- **V= 0,7 [m/s]**

It is very important to have a velocity above zero. This reason is because the velocity influx is quadratically on the pump calculation, so if it is selected an adequate diameter pipe the power needed of the pump is going to decrease.

Finally the diameter pipe selected to line 5 and 5' is the minimal; **51mm.**

The next variable that it is going to be calculated is the equivalent longitude. The equivalent pipe longitude is composed by the lineal longitude of the pipes and the accessories installed on it. The value of the equivalent longitude of the accessories can be taken from the accessory datasheet when it is available (This can be done with the dry cooler). If the equivalent longitude is not available, it can be estimated with a simple formula took from the European normative for pressure circuits. So;

$$L_{eq}= L_{lineal} + L_{accesoires}$$

It has been estimated that there are approximately two 90º elbows, the load losses of one elbow is estimated like L_{1elbow}=(L_{lineal}·ϕ_{int})= [(7+7)*0.051]= 0,72 [m/accessorie] . Also it must be taken into account the external dry cooler.

- Model: WR1263.A → Pressure drop = 22 [kPa] = 22/9,81 [m.c.a]= 2,3 m

Finally;

$$L_{eq}= L_{lineal} + L_{accesoires}= (7+7) + 2*0{,}72 + 2'3 \approx 18 \text{ [m]}$$

So;

- **L_{eq}= 18 [m]**

The last step to solve the calculation of the pump is to know the friction coefficient “f”. This coefficient gives an idea about how the fluid behaves across the pipes. There are two different possibilities on which the flux can appear across a pipe, laminar flux and turbulent flux.

To determine what kind of flux is on our pipe, it is necessary to calculate the Reynolds number;

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu}$$

- D; internal diameter of pipe [m]
- V; flux velocity [m/s]
- ρ; density [kg/m^3]
- µ; dynamic viscosity [cP]

A flux is considered laminar if the N_{Re}< 2000, moreover if the Reynolds number is 2000 < N_{Re} < 4000 it is on unstable zone and if N_{Re} > 4000 the flux is considered turbulent.

From the Reynolds equation it is know that D= 0,051 [m], V= 0,7 [m/s] and ρ= 1080 [kg/m^3], to calculate the viscosity it must be found it in the next figure;

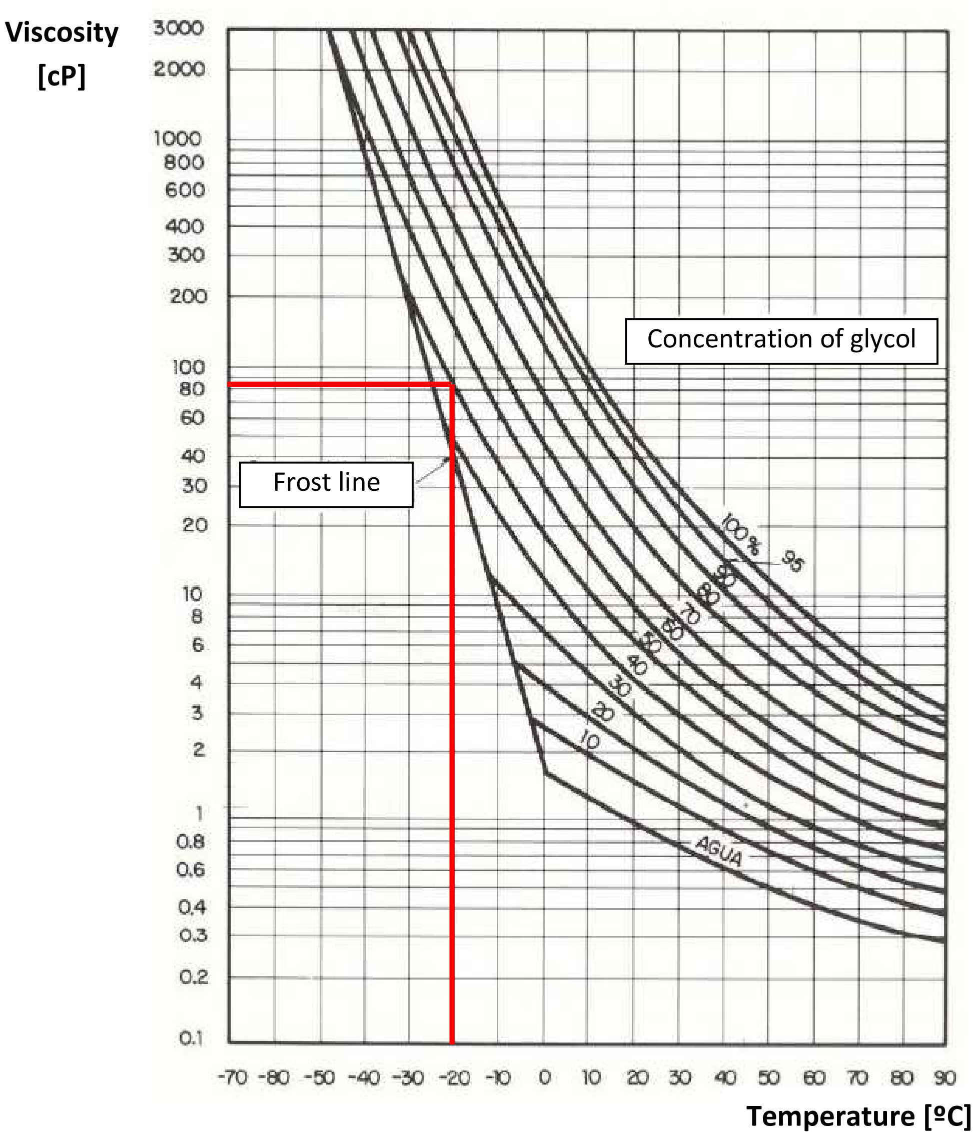


Figure 68; Dynamic viscosity to different glycol-water mixtures

From the figure 68 it has been selected μ= 85 [cP], but to substitute it on the Reynolds equation the viscosity must be on Pa·s;

100 [cP]=0.1 [Pa·s] → (85·0,1)/100 = 0,085 [cP]

- **μ= 0,085 [cp]**

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu} = \frac{0,051 \cdot 0,7 \cdot 1080}{0,085} \approx 455$$

On this case the $N_{Re} < 2000$, so our system works on a laminar regime. It can be used the Moody diagram to get the friction coefficient;

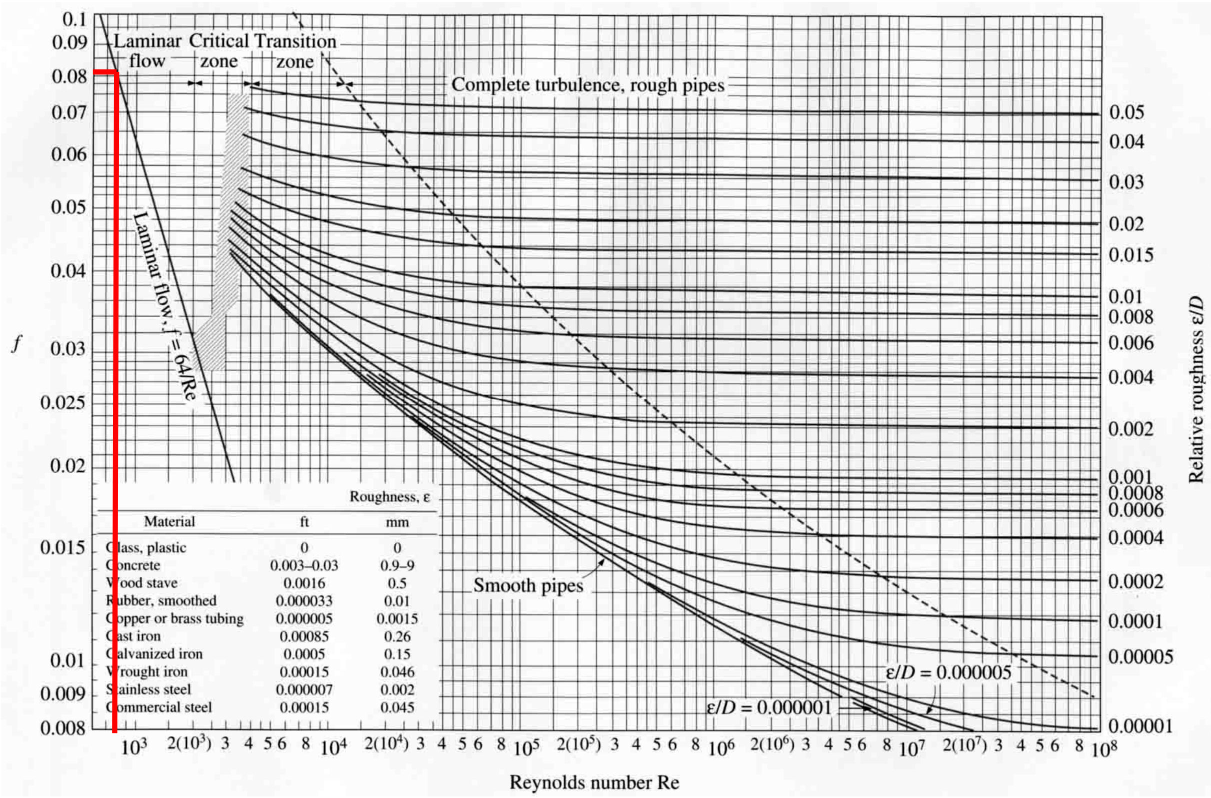


Figure 69; Moody diagram

Finally, if it has been selected in the **figure 69** a NRe ≈ 455, it is obtained a friction coefficient around f≈0,08 , so;

- **f= 0,08**

Now, it is available to substitute all the values calculated on the losses pipes equation (**equation 6**);

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g} = \frac{0,08 \cdot 18 \cdot 0,7^2}{2 \cdot 0,051 \cdot 9,8} = 0,7\ [m]$$

F) Line 6- Line 6';

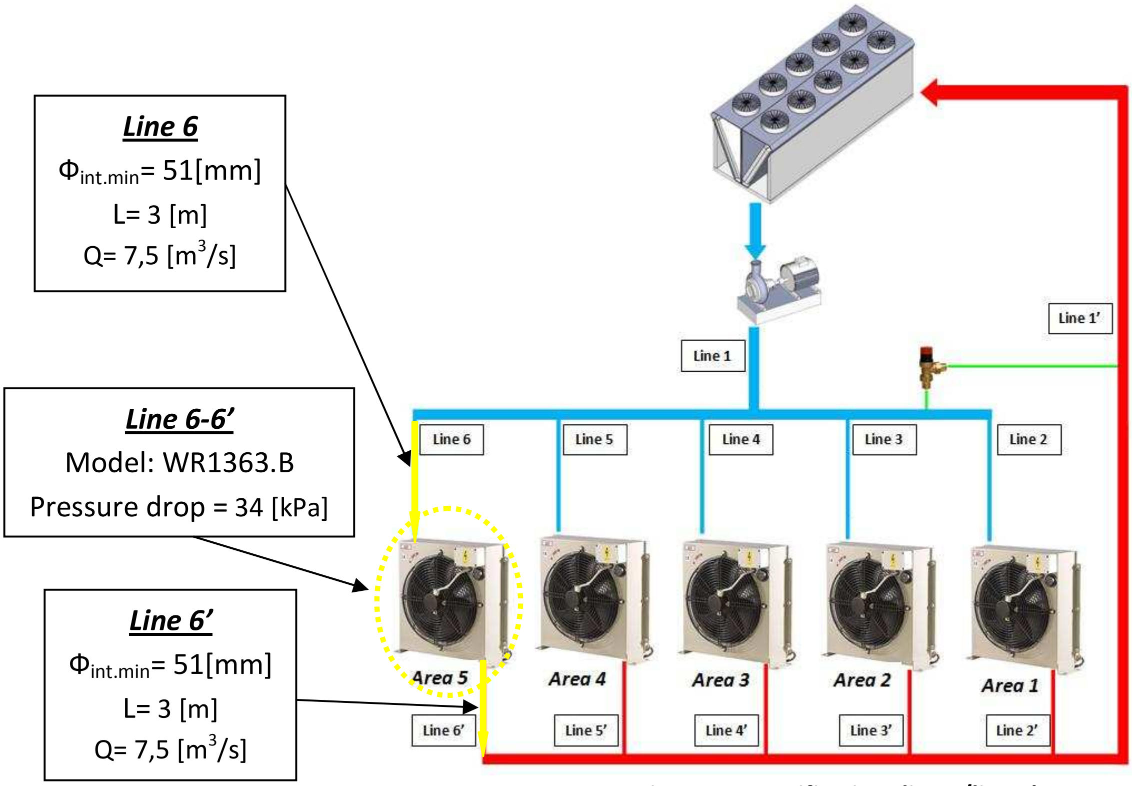


Figure 70; specifications line 6/line 6'

With these specifications it can be solved the equation of the losses across the pipes;

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g}$$

Now it knows;

- **g= 9,81 [m/s²]**
- **D= 51 [mm] = 0,051 [m]**

For the flux velocity, it must be applied the caudal formula;

$$Q = A \cdot V \;\; [\mathrm{m^3/s}]$$

$$V = \frac{Q}{A} \; [\mathrm{m/s}]$$

Where;

Q = 7,5 [m^3/s] = 0,002 [m^3/h]

It has been selected an internal diameter for the main pipes (line 6-6') of 51mm, so;

A = $\pi\cdot(D^2/4) = \pi\cdot(51^2/4)$= 2043 mm^2 = $2{,}0\cdot10^{-3}$ m^2

$$V = \frac{0{,}002}{2{,}0 \cdot 10^{-3}} = 1\ [\mathrm{m/s}]$$

- **V= 1 [m/s]**

It is very important to have a velocity above zero. This reason is because the velocity influx is quadratically on the pump calculation, so if it is selected an adequate diameter pipe the power needed of the pump is going to decrease.

Finally the diameter pipe selected to line 6 and 6' is the minimal; **51mm.**

The next variable that it is going to be calculated is the equivalent longitude. The equivalent pipe longitude is composed by the lineal longitude of the pipes and the accessories installed on it. The value of the equivalent longitude of the accessories can be taken from the accessory datasheet when it is available (This can be done with the dry cooler). If the equivalent longitude is not available, it can be estimated with a simple formula took from the European normative for pressure circuits. So;

$$L_{eq} = L_{lineal} + L_{accesoires}$$

It has been estimated that there are approximately two 90º elbows, the load losses of one elbow are estimated like $L_{1elbow}=(L_{lineal}\cdot\phi_{int})$= [(3+3)*0.051]= 0,31 [m/accessorie] . Also it must be taken into account the external dry cooler.

- Model: WR1263.A → Pressure drop = 34 [kPa] = 34/9,81 [m.c.a]= 3,4 m

Finally;

$$L_{eq} = L_{lineal} + L_{accesoires} = (3+3) + 2*0{,}31 + 3'4 \approx 10\ [m]$$

So;

- **L_{eq}= 10 [m]**

The last step to solve the calculation of the pump is to know the friction coefficient “f”. This coefficient gives an idea about how the fluid behaves across the pipes. There are two different possibilities on which the flux can appear across a pipe, laminar flux and turbulent flux.

To determinate what kind of flux is on our pipe, it is necessary to calculate the Reynolds number;

$$\boldsymbol{N_{Re} = \frac{D \cdot V \cdot \rho}{\mu}}$$

- D; internal diameter of pipe [m]
- V; flux velocity [m/s]
- ρ; density [kg/m^3]
- µ; dynamic viscosity [cP]

A flux is considered laminar if the N_{Re}< 2000, moreover if the Reynolds number is 2000 < N_{Re} < 4000 it is on unstable zone and if N_{Re} > 4000 the flux is considered turbulent.

From the Reynolds equation it is known that D= 0,051 [m], V= 1 [m/s] and ρ= 1080 [kg/m^3], to calculate the viscosity it must be found it in the next figure;

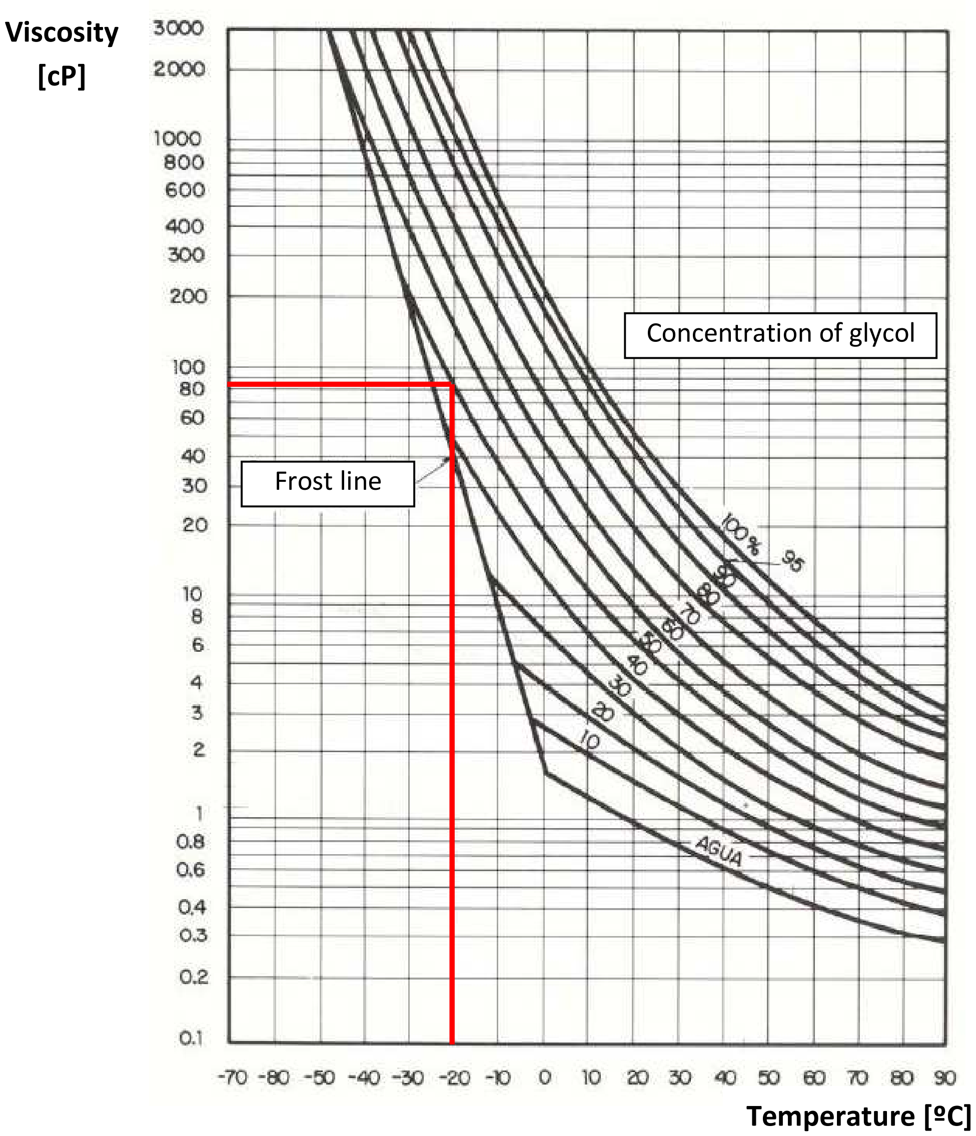


Figure 71; Dynamic viscosity to different glycol-water mixtures

From the **figure 71** it has been selected μ= 85 [cP], but to substitute it on the Reynolds equation the viscosity must be on Pa·s;

100 [cP]=0.1 [Pa·s] → (85·0,1)/100 = 0,085 [cP]

- **μ= 0,085 [cp]**

$$N_{Re} = \frac{D \cdot V \cdot \rho}{\mu} = \frac{0,051 \cdot 1 \cdot 1080}{0,085} \approx 650$$

On this case the $N_{Re} < 2000$, so our system works on a laminar regime. It can be used the Moody diagram to get the friction coefficient;

4

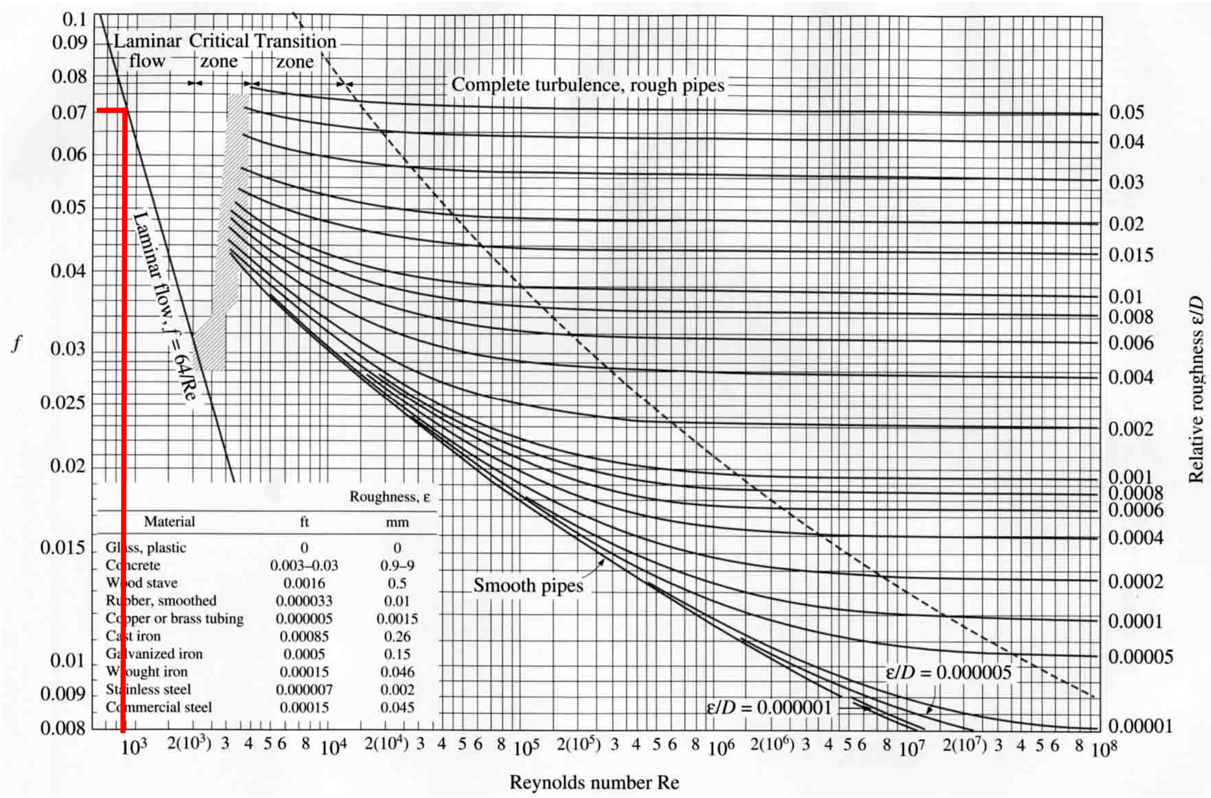


Figure 72; Moody diagram

Finally, if it is used the graphic for a NRe ≈ 650, it is obtained a friction coefficient around f≈0,07, so;

- **f= 0,07**

Now, it is available to substitute all the values calculated on the losses pipes equation (**equation 6**);

$$h_f = \frac{f \cdot L_{eq} \cdot V^2}{2 \cdot D \cdot g} = \frac{0,07 \cdot 10 \cdot 1^2}{2 \cdot 0,051 \cdot 9,8} \approx 0,7\ [m]$$

6.3.3 Resume table of the losses across the pipes

Now all necessary data are ready to calculate the pump, however, before that it has been taken into account other decision. As can be seen on the last figures (see **figure 70**) the pipe net is composed but a main branch and five derivations that go into each enclosure area. So there is a quite different emplacement to the pump and it has to be decided to take a pump for derivation or a single pump into the main branch.

The project objective is to decrease as much as possible the final COP, so because of that try to reduce the COP is the criteria to make the selection;

LINE	Hb=h_f [m]
1-1’	**7,7**
2-2’	0,48
3-3’	2,1
4-4’	0,48
5-5’	0,7
6-6’	0,7
TOTAL	***12,16***

Table 63; Pipe load losses resume

On the **table 63** it can be appreciated that the more influent line is the 1-1’. If it is installed a pump on the main branch them practically the pump can drive all the system flux. So the final option is to take a pump installed into line 1-1’ with the total load losses.

6.3.4 Pump selection

The pump much satisfies three principal conditions; the pump power, the total caudal driven and the minimal Hb, after calculations it has been got;

- ***Q= 58,7 [m³/h]***
- ***Hb= 12 [m]***

As it can be deduced from the last data this pump is not a normal one, this is because the caudal needed is very high on relation with the low Hb. Finally the best pump supplier found is EBARA;

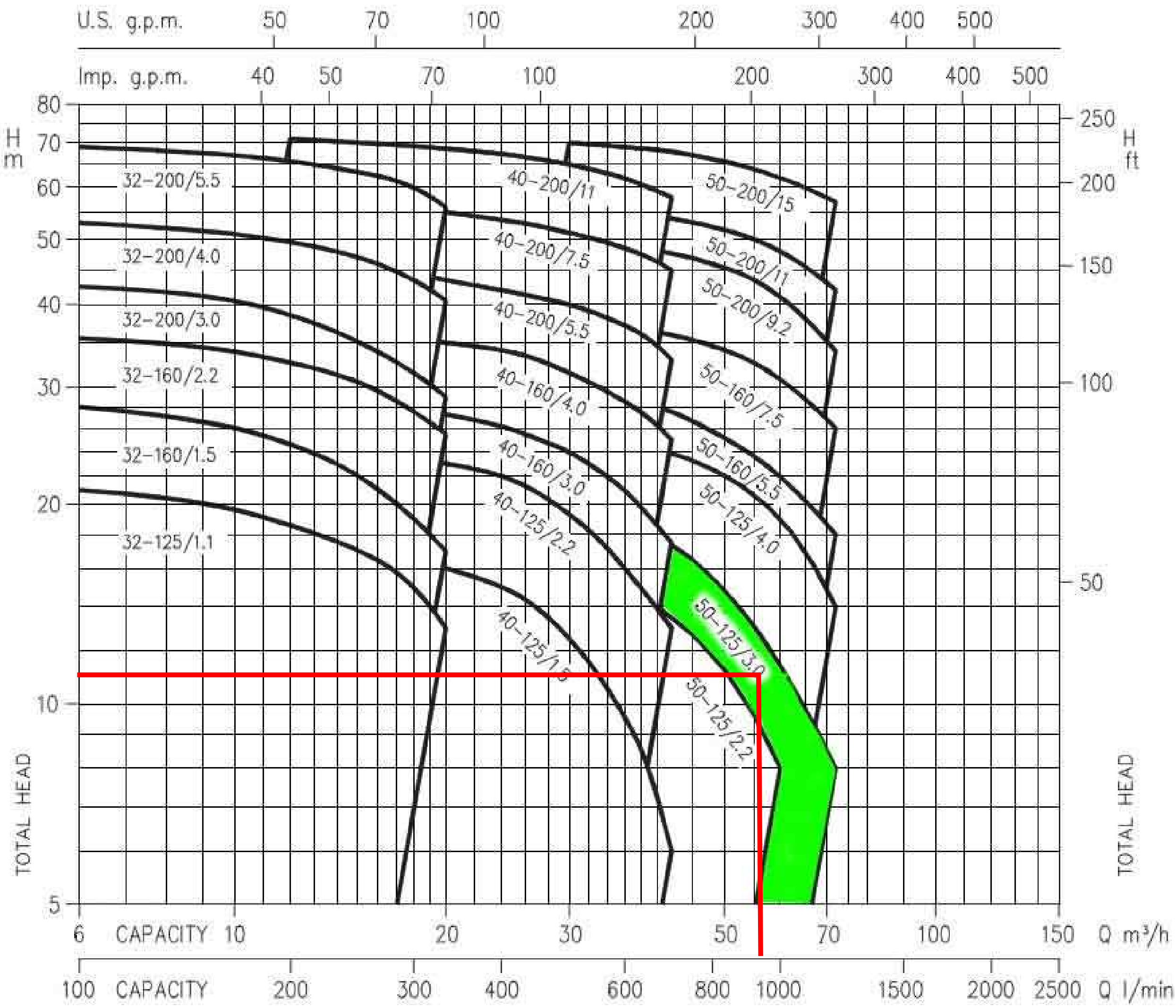


Figure 73; Working ranges to 3M-3S-3P-(L) model

To know which type of pump has to be taken, it is necessary to go into the **figure 73,** like is showed, with the next data;

- ***Q= 58,7 [m^3/h]= 58,7 ·(1000/60) [l/min]= 979 [l/min]***
- ***Hb= 12 [m]***

Lines intersect into 50-125/3.0 zone, so it has to be used that kind of pump but it is necessary specify with accuracy the working conditions.

Bomba	kW	Intensidad absorbida (A) Trifásica 230V	400V	690V	l/min	100	150	200	250	300	333	400	450	500	550	600	650	700	800	100
					m³/h	6	9	12	15	18	20	24	27	30	33	36	39	42	48	60
						Q=Caudal / H=Altura manométrica total en m														
32-125/1.1	1,1	5,0	2,9	-		21	20	18,5	17	15	13	-	-	-	-	-	-	-	-	-
32-160/1.5	1,5	5,9	3,4	-		28	26,5	24,5	22	19	17	-	-	-	-	-	-	-	-	-
32-160/2.2	2,2	8,3	4,8	-		35,5	34,5	32,5	30,5	27,5	25,5	-	-	-	-	-	-	-	-	-
32-200/3.0	3,0	11,8	6,8	-		42,5	41	38,5	35	31,5	29	-	-	-	-	-	-	-	-	-
32-200/4.0	4,0	15,6	9,0	-		53	51,5	49,5	47	43,5	40,5	-	-	-	-	-	-	-	-	-
32-200/5.5	5,5	-	11,8	6,8		69	67,6	65,5	63	60	56	-	-	-	-	-	-	-	-	-
40-125/1.5	1,5	5,9	3,4	-		-	-	18	17,5	17	16	15	14	12,5	11	9,5	8	6	-	-
40-125/2.2	2,2	8,3	4,8	-		-	-	26	25	24,2	23	22	21	19	17,5	16	14,3	13	-	-
40-160/3.0	3,0	11,8	6,8	-		-	-	30	29	28,5	27,3	26,2	25,4	24	22,5	21	19,2	17,5	-	-
40-160/4.0	4,0	15,9	9,2	-		-	-	38	37	36	35	34	33	31,3	30	28,5	27	25	-	-
40-200/5.5	5,5	-	11,1	6,4		-	-	46	45	44	43,5	42	41	40	38,5	37	35,1	33	-	-
40-200/7.5	7,5	-	15,1	8,7		-	-	56,5	56	55,3	55	53,5	52,5	51,2	49,8	48,5	47	45	-	-
40-200/11	11	-	20,0	11,6		-	-	71	70	69,3	68,8	67,5	66,2	65	63,5	62	60	58	-	-
50-125/2.2	2,2	8,3	4,8	-		-	-	-	-	-	-	17	16,6	16,1	15,5	14,9	14,2	13,4	11,8	8
50-125/3.0	3,0	11,8	6,8	-		-	-	-	-	-	-	20,5	20	19,5	19	18,5	18	17,3	15,5	12,
50-125/4.0	4,0	15,9	9,2	-		-	-	-	-	-	-	26	25,9	25,7	25,3	24,7	24,2	23,3	22,2	19
50-160/5.5	5,5	-	11,5	6,6		-	-	-	-	-	-	31	30,5	30	29,5	29	28	27,6	26	22,
50-160/7.5	7,5	-	15,5	9,0		-	-	-	-	-	-	39	38,5	38	37,5	37	36,5	36	34,5	31
50-200/9.2	9,2	-	17,4	10,0		-	-	-	-	-	-	-	-	50	49,5	49	48,4	47,5	46	41
50-200/11	11	-	22,0	12,7		-	-	-	-	-	-	-	-	56	55,5	55	54,5	53,8	52	48
50-200/15	15	-	31,3	18,0		-	-	-	-	-	-	-	-	70	69,5	69	68,5	68	66	62

Table 64; Characteristics to 3M-3S-3P-(L) model

From the **table 64** has to be extracted all the data of the pump on nominal conditions of work;

- Model; 3M-3S-3P-(L)
- Kind; 50-125/3.0
- **Max pump power; 3.0 kW**
- Q= 60 [m^3/h] ***> 58,7 [m^3/h]. OK***
- Hb= 12,3 [m] ***> 12 [m]. OK***

Now it is known the pump power but it is needed to know the motor power that drives the pump, therefore it is needed the pump efficiency, that is going to be selected in the **figure 74**;

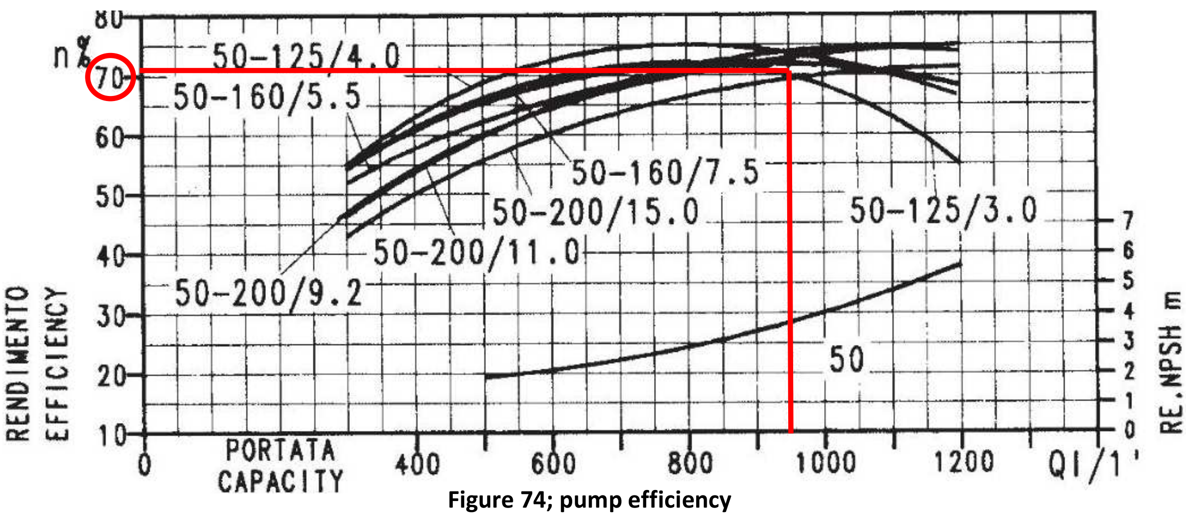


Figure 74; pump efficiency

The pump efficiency for a regimen of Q= 955 [l/min] is around η= 0,7, hence the driving motor of the pump must have a minimal power of;

$$Pmotor = \frac{Ppump}{\eta_{pump}} = \frac{3}{0{,}7} = 4{,}3\ [kW]$$

The motor that has been chosen to drive the pump is the immediately superior available on the market;

P_{motor}= 4,5kW

7 TOTAL SYSTEM COP

The total COP gives an idea of how much cost the cold needed to refrigerate the indoor enclosure rooms, so the COP is the main objective to improve the energy saves.

Through the project it has appeared divers COP referred to the different system parts but not the total COP with which the dry cooler system is going to work. The COP calculated had around 150 and 160 points. It has been calculated like this because locally for each dry cooler with their fans it has not taken into account the consumption of the pump.

On the other hand believe that the COP from the internal dry coolers is the total COP is to commit an error, as it has been demonstrated. That is due to the total electricity consumed; the outdoor and indoor dry coolers fans, as well as the pump. It is important not to forget that only is recovered the cold from the outdoor dry cooler. It is necessary the indoor dry cooler to carry it into the rooms and the pump has to drive the refrigerant flux, but these two last systems are necessary for not increase the recovered cold. These systems only consume electricity and this fact increases the electricity consume, consequently decreasing the total COP.

Below it is showed a explaining picture with all the system COPs and with the total one calculated for the system;

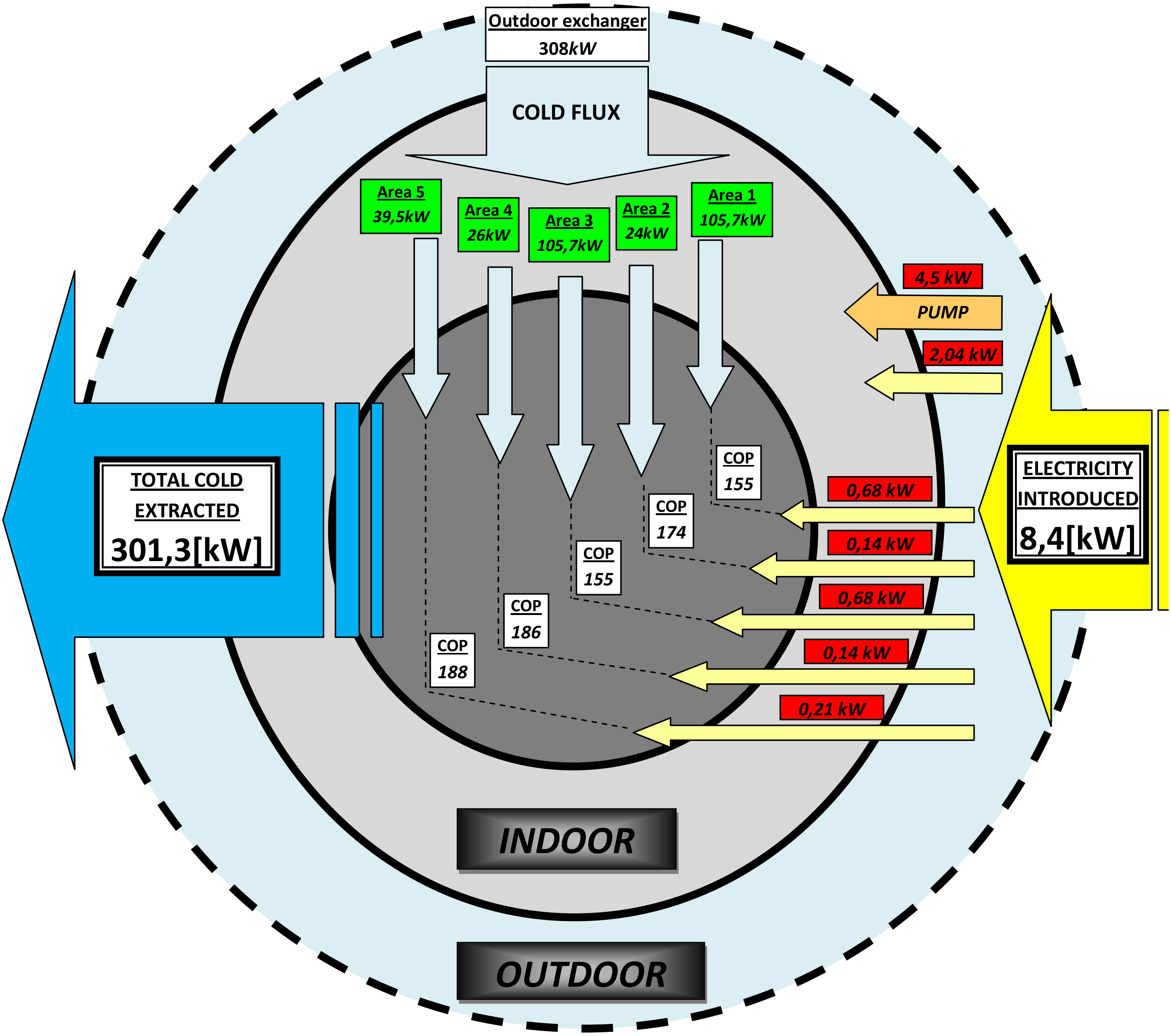


Figure 75; Cooling flux and electricity fluxes. Local and total COP.

$$COP_{total}\frac{\textit{Total cold delivered into rooms}}{\textit{Total electricity consumed}}=\frac{301,3}{8,4}=36$$

Initially it was thought to have a COP of around 150 or 160 but it has been demonstrated that is not possible with this kind of system. However a COP=36 is an exceptional number if it is compared with the conventional system for getting cold, a COP of a standard system cannot exceed 4 points.

So, it has been designed a system to save energy **12 times more efficient** than any standard system available on the market.

8 IMPLEMENTATION OF THE NEW SYSTEM TO SUPPORT A STANDARD SYSTEM

Dry cooler system depends for start working of the external temperatures and these temperatures change with the year's seasons, therefore it cannot work alone. So to exploit the very high efficiency offered for the new system it must be installed combined with an autonomy system to supply cold.

To explain how the new system works it is necessary understand how a standard system works. On the next points first it is going to be shown a normal system to supply cold and after that it will be designed the working method for the new system.

8.1 Working mode of a standard system to supply cold

A standard system to supply cold works accord the refrigeration cycle (see **figure 77**);

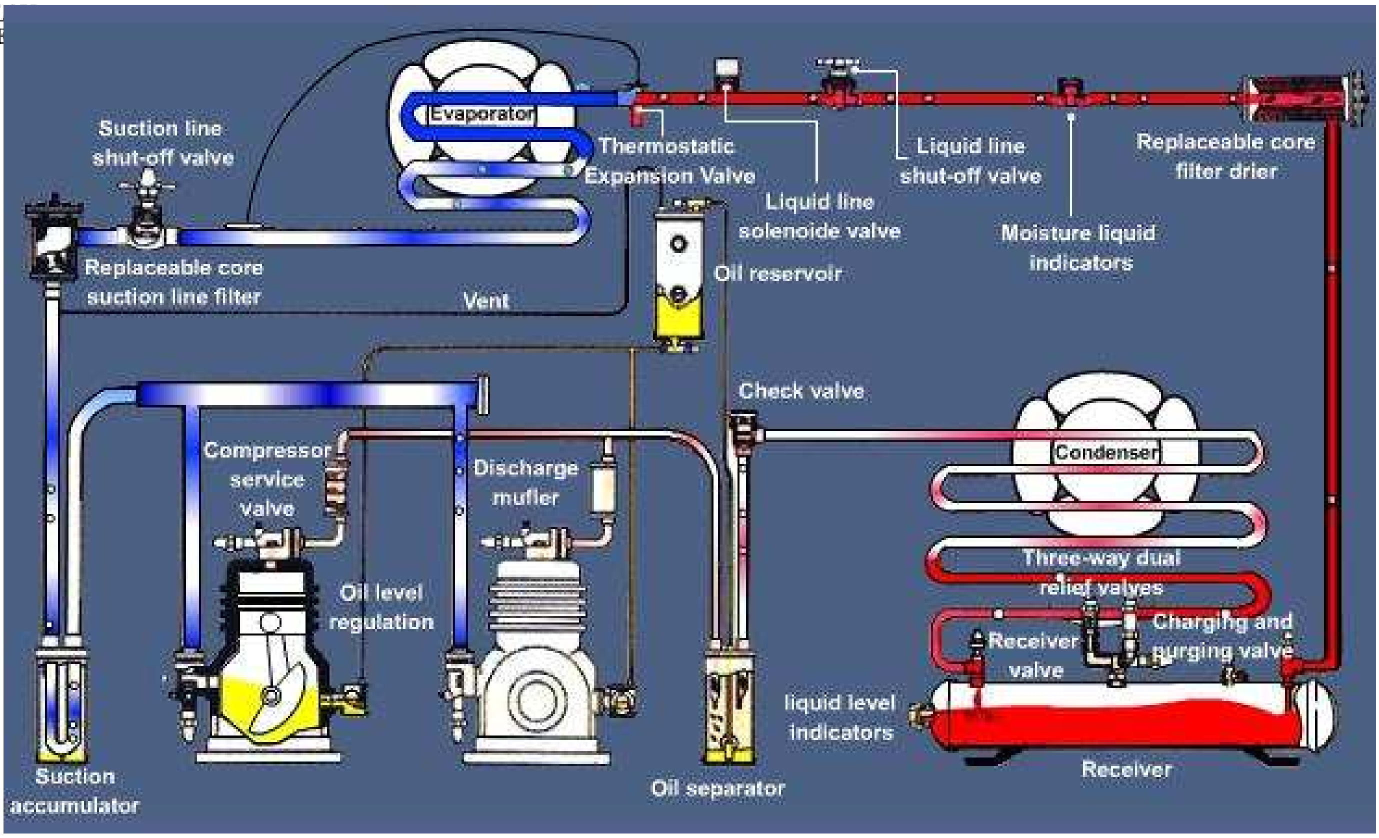


Figure 76; Refrigeration cycle on Willys. Schematic model

This is the system used on all common refrigeration cases, it has the advantage to be very safety to supply cold but it is very expensive if it is compared with the dry cooler system.

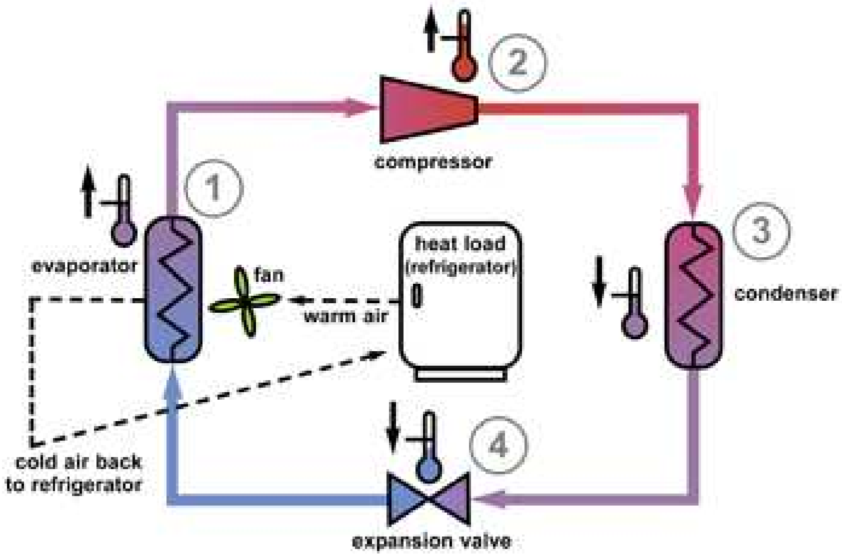


Figure 77; Components schematic of refrigeration cycle

Between all the components of the **figure 76** the evaporator/expansion valve is the one that goes into the room. It is built of different sharps but the common particularity is that it has to be placed where the cold is required, like in our system the indoor dry coolers.

The model of work of this device is digital, i.e., it switches on and switches off at different intervals between a temperature range. To understand what this mean it showed below the **figure 78**;

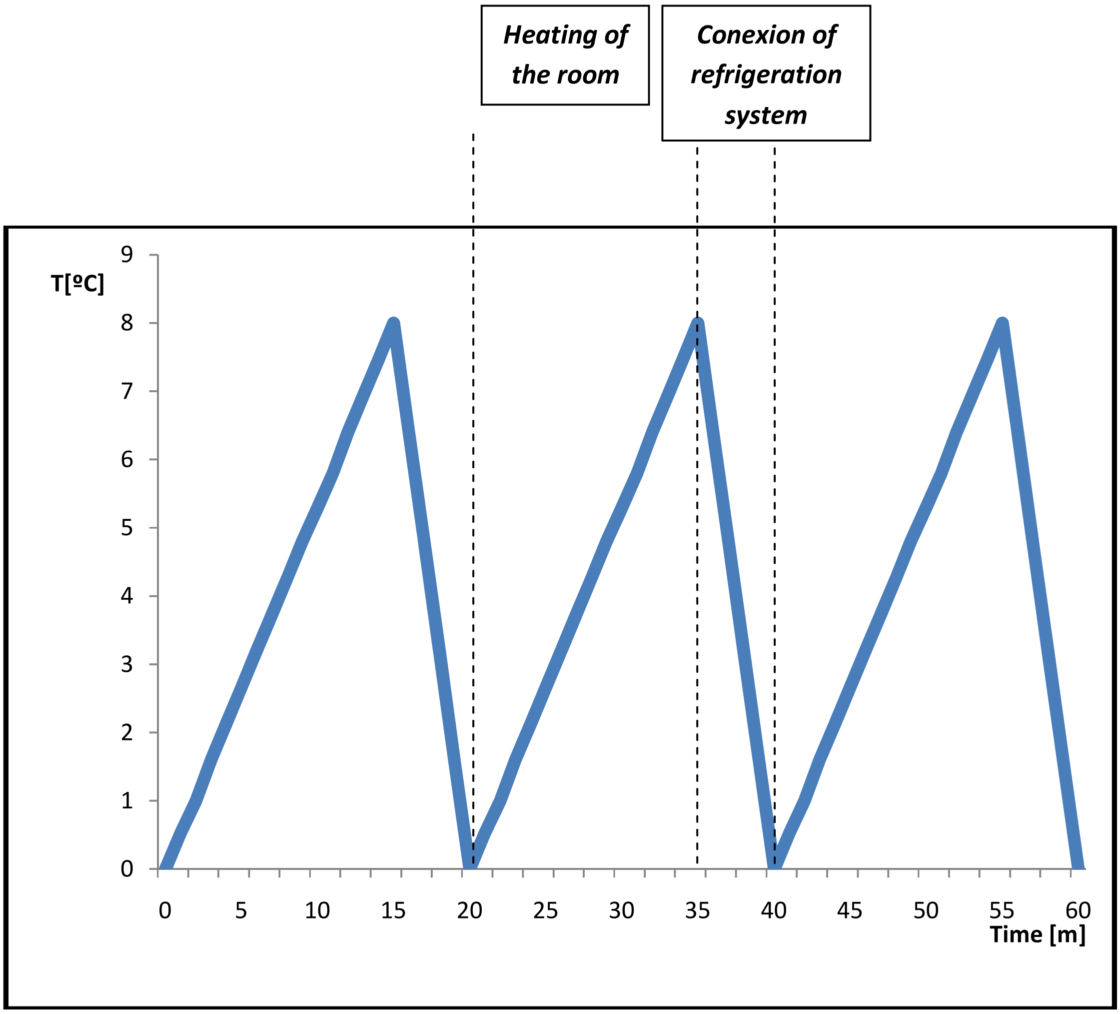


Figure 78; Model of work of a common system to deliver cold

In the **figure 78** the line with positive slope represent the heating room through the time, it is intuitive to think that if the insulation materials of the room walls have a very low U-value the slope of ascendant line is going to decrease. Consequently the system would have to switch on less time to keep the room temperature on range.

8.2 Working on parallel with the new system

It has to be differentiated two working ways of the supporting system. When the system works alone and when the system is working on parallel with the other cooling system.

8.2.1 System working alone

When the system is working alone all the energy delivered to keep the indoor temperature on range is carried from the new system. This mean that the cold comes from a COP=36, therefore the electricity cost is going to decrease around 10 times.

So it must be designed the system to switch on before the standard system starts working. The control could be done with a simple electronic control.

The standard system starts to work when the indoor temperature is 8ºC and it is going to switch off when the indoor temperature becomes 0ºC. To assure that it is going only to work our system the working conditions for the dry cooler system are going to be designed between 7'5 ºC and 2ºC. It is going to be sure through this way that the new system is going to work before the old system starts when the external temperatures are the ones required for the new system. This is shown in the **figure 79**.

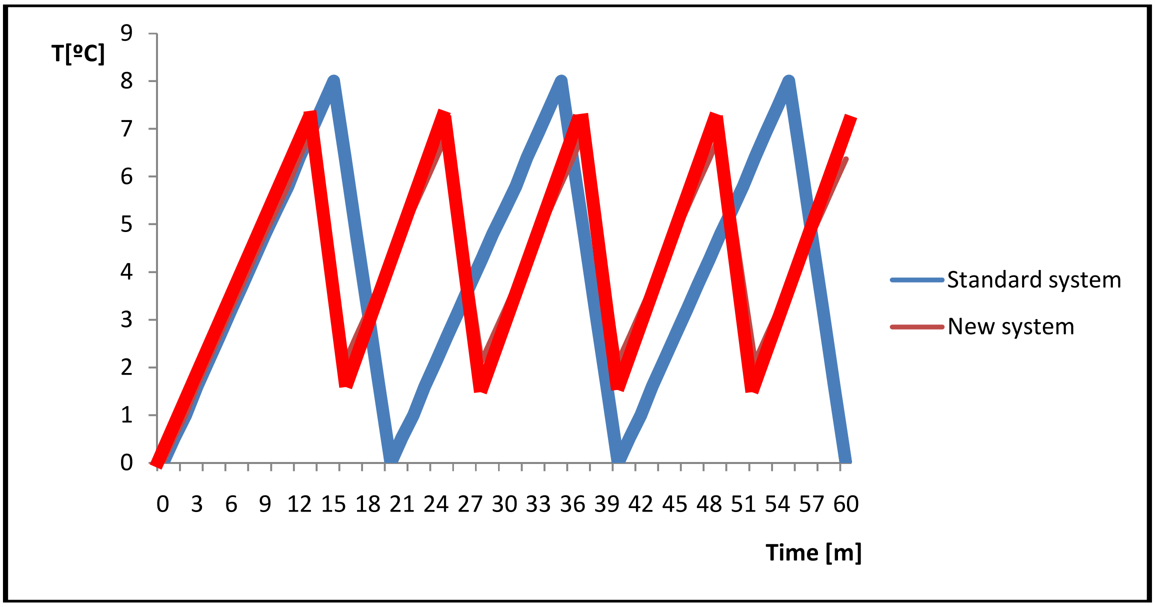


Figure 79; the new system working alone

Now it is necessary to quantify the amount of energy saved on this way;

Willys system gets cold to refrigerate theirs rooms with a relation of;

$$\mathbf{1[kW_{elect}] \rightarrow 3[kW_{cold}]}$$

With the new system working the relation to get cold is;

$$\mathbf{1[kW_{elect}] \rightarrow 36[kW_{cold}]}$$

So the electricity used to deliver the same cold is;

$$\left(\frac{3}{36}\cdot 100\right) = 8{,}3\ \%\ \text{of the before total, hence;}$$

$$\boxed{\textit{91,7 \% of electricity is saved}}$$

8.2.2 System working on parallel

When both new and old systems are working on parallel it has to be differentiated five different cases;

A) New system working from 8ºC->3ºC. Old system working from 3ºC->0ºC

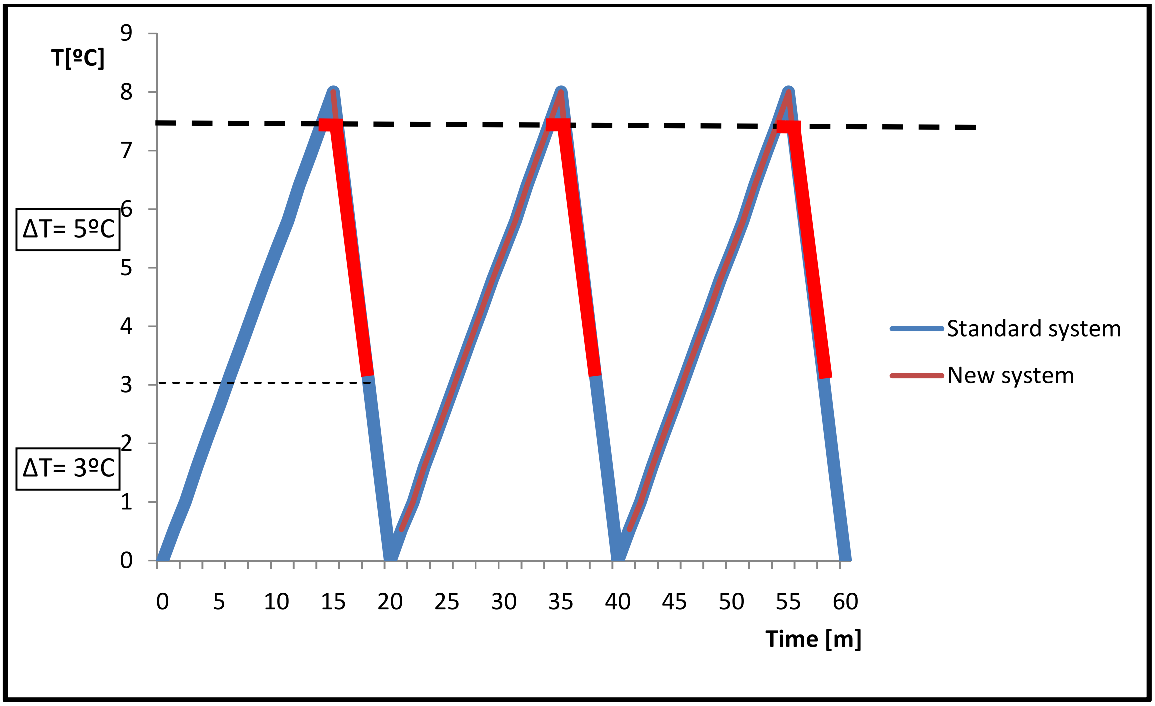


Figure 80; New and old system working on parallel (8ºC-3ºC)

The total amount of energy needed is the energy needed to decrease from 8ºC to º0C, this amount of energy is named X [kW_{cold}]

Immediately it has been deduced that the electric energy to get that cold power with the old system is;

$$X[kWcold]\cdot\left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right]=\left(\frac{1}{3}\right)\cdot X[kWe]$$

So to find the total amount of energy consumed with both systems working on parallel it has to be done an equation pondered with each COP influence;

$$X \cdot \left(\frac{5}{8}\right)[kWcold] \cdot \left(\frac{1}{36}\right)\left[\frac{kWe}{kWcold}\right] + X \cdot \left(\frac{3}{8}\right)[kWcold] \cdot \left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right] = (0{,}0174 + 0{,}125)X[kWe]$$

So the total energy consumed respect the normal one is;

$$(100 - \frac{(0{,}0174 + 0{,}125) \cdot X[kWe]}{\left(\frac{1}{3}\right) \cdot X[kWe]} \cdot 100) = \mathbf{57{,}3\%[kWe]}$$

From the formula it has been deduced that the cold demand has no influence on the total energy saved;

57,3 % of electricity is saved

B) New system working from 8ºC->4ºC. Old system working from 4ºC->0ºC

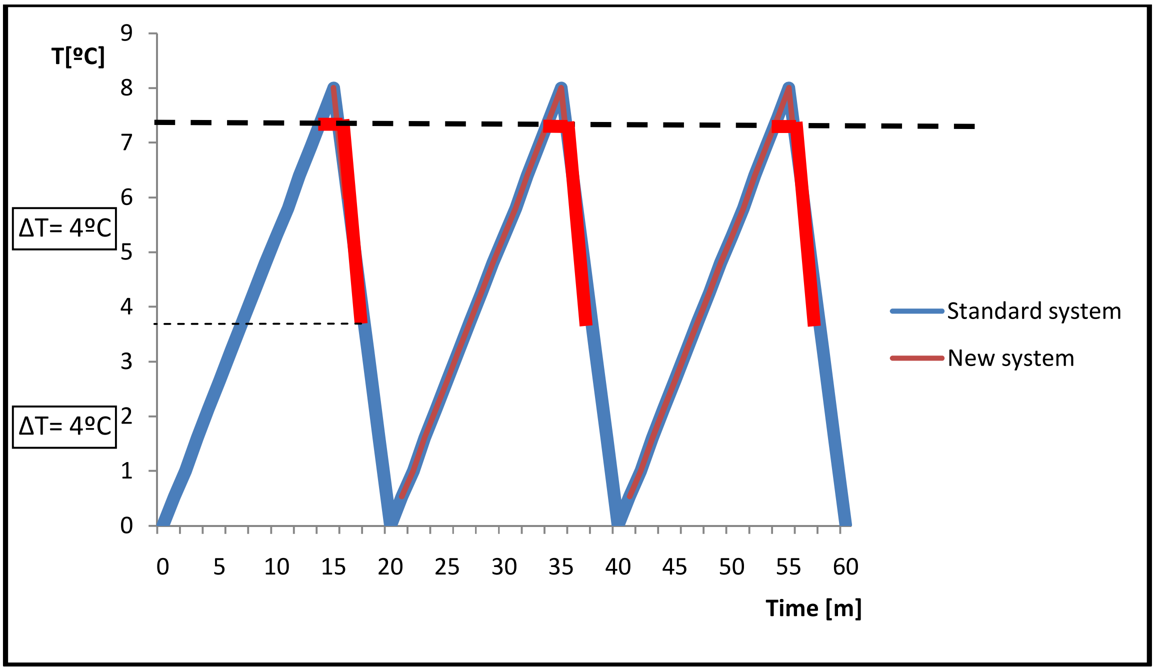


Figure 81; new and old system working on parallel (8ºC-4ºC)

The total amount of energy needed is the energy needed to decrease from 8ºC to º0C, this amount of energy is named X [kW_{cold}]

Immediately it has been deduced that the electric energy to get that cold power with the old system is;

$$X[kWcold]\cdot\left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right]=\left(\frac{1}{3}\right)\cdot X[kWe]$$

So to find the total amount of energy consumed with both systems working on parallel it has to be done an equation pondered with each COP influence;

$$X\cdot\left(\frac{4}{8}\right)[kWcold]\cdot\left(\frac{1}{36}\right)\left[\frac{kWe}{kWcold}\right]+X\cdot\left(\frac{4}{8}\right)[kWcold]\cdot\left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right]$$
$$=(0{,}0139+0{,}166)X[kWe]$$

So the total energy consumed respect the normal one is;

$$(100-\frac{(0{,}0139+0{,}166)\cdot X[kWe]}{\left(\frac{1}{3}\right)\cdot X[kWe]}\cdot 100)=\ \mathbf{45,3\%[\mathit{kWe}]}$$

From the formula it has been deduced that the cold demand has no influence on the total energy saved;

45,3 % of electricity is saved

C) New system working from 8ºC->5ºC. Old system working from 5ºC->0ºC

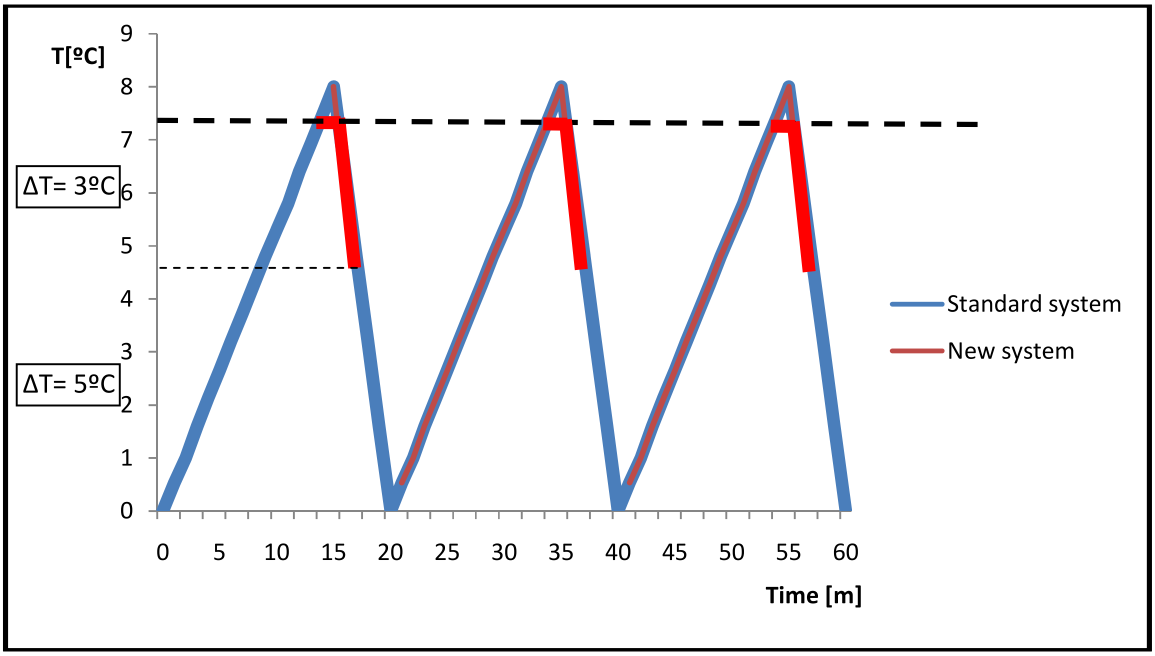


Figure 82; new and old system working in parallel (8ºC-5ºC)

The total amount of energy needed is the energy needed to decrease from 8ºC to º0C, this amount of energy is named X [kW_{cold}]

Immediately it has been deduced that the electric energy to get that cold power with the old system is;

$$X[kWcold] \cdot \left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right] = \left(\frac{1}{3}\right) \cdot X[kWe]$$

So to find the total amount of energy consumed with both systems working on parallel it has to be done an equation pondered with each COP influence;

$$X \cdot \left(\frac{3}{8}\right)[kWcold] \cdot \left(\frac{1}{36}\right)\left[\frac{kWe}{kWcold}\right] + X \cdot \left(\frac{5}{8}\right)[kWcold] \cdot \left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right]$$
$$= (0{,}0104 + 0{,}208)X[kWe]$$

So the total energy consumed respect the normal one is;

$$\left(100 - \frac{(0{,}0104 + 0{,}208) \cdot X[kWe]}{\left(\frac{1}{3}\right) \cdot X[kWe]} \cdot 100\right) = \mathbf{34,3\%[kWe]}$$

From the formula it has been deduced that the cold demand has no influence on the total energy saved;

34,3 % of electricity is saved

D) New system working from 8ºC->6ºC. Old system working from 6ºC->0ºC

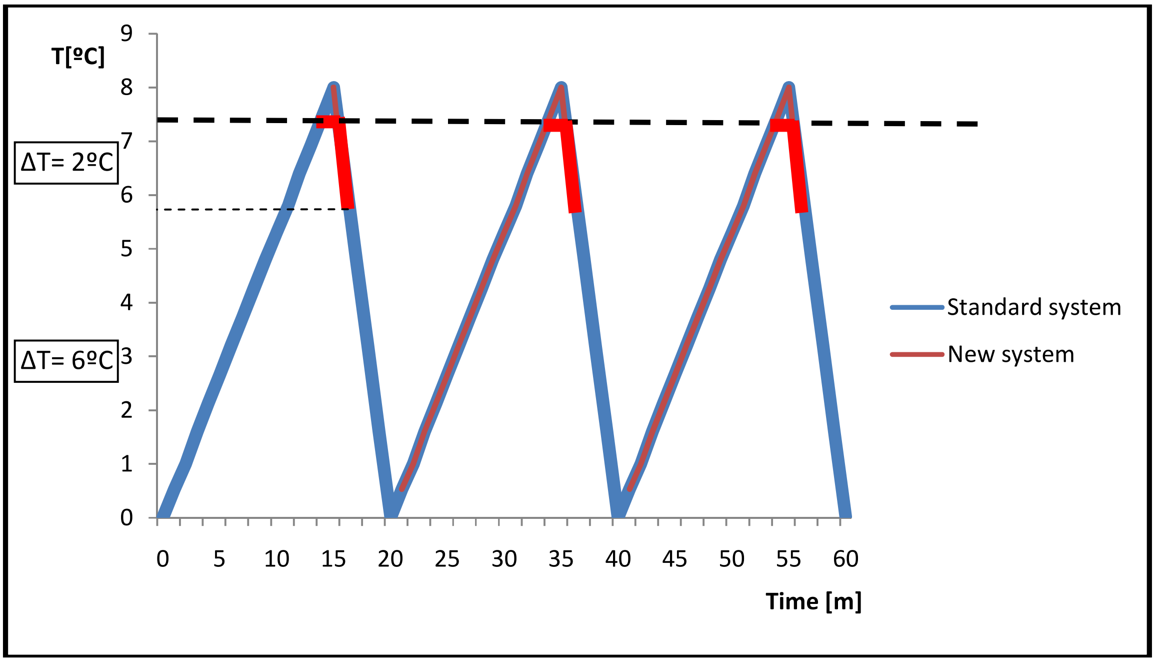


Figure 83; new and old system working on parallel (8ºC-6ºC)

The total amount of energy needed is the energy needed to decrease from 8ºC to º0C, this amount of energy is named X [kW_{cold}]

Immediately it has been deduced that the electric energy to get that cold power with the old system is;

$$X[kWcold] \cdot \left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right] = \left(\frac{1}{3}\right) \cdot X[kWe]$$

So to find the total amount of energy consumed with both systems working on parallel it has to be done an equation pondered with each COP influence;

$$X \cdot \left(\frac{2}{8}\right)[kWcold] \cdot \left(\frac{1}{36}\right)\left[\frac{kWe}{kWcold}\right] + X \cdot \left(\frac{6}{8}\right)[kWcold] \cdot \left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right] = (0{,}0069 + 0{,}25)X[kWe]$$

So the total energy consumed respect the normal one is;

$$\left(100 - \frac{(0{,}0069 + 0{,}25) \cdot X[kWe]}{\left(\frac{1}{3}\right) \cdot X[kWe]} \cdot 100\right) = \mathbf{23}\%[\boldsymbol{kWe}]$$

From the formula it has been deduced that the cold demand has no influence on the total energy saved;

23 % of electricity is saved

E) New system working from 8ºC->7ºC. Old system working from 7ºC->0ºC

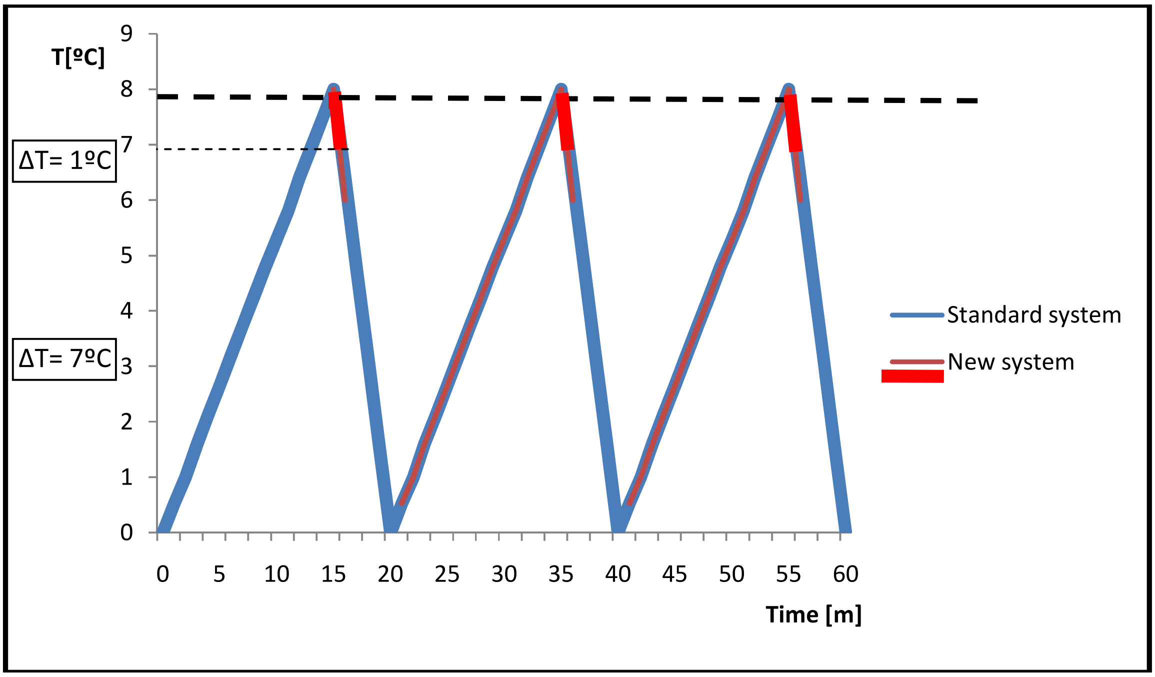


Figure 84; new and old system working on parallel (8ºC-7ºC)

The total amount of energy needed is the energy needed to decrease from 8ºC to º0C, this amount of energy is named X [kW_{cold}]

Immediately it has been deduced that the electric energy to get that cold power with the old system is;

$$X[kWcold]\cdot\left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right]=\left(\frac{1}{3}\right)\cdot X[kWe]$$

So to find the total amount of energy consumed with both systems working on parallel it has to be done an equation pondered with each COP influence;

$$X\cdot\left(\frac{1}{8}\right)[kWcold]\cdot\left(\frac{1}{36}\right)\left[\frac{kWe}{kWcold}\right]+X\cdot\left(\frac{7}{8}\right)[kWcold]\cdot\left(\frac{1}{3}\right)\left[\frac{kWe}{kWcold}\right]$$
$$=(0{,}0034+0{,}29)X[kWe]$$

So the total energy consumed respect the normal one is;

$$\left(100 - \frac{(0{,}0034 + 0{,}29)\cdot X[kWe]}{\left(\frac{1}{3}\right)\cdot X[kWe]}\cdot 100\right) = \mathbf{12}\%[\boldsymbol{kWe}]$$

From the formula it has been deduced that the cold demand has no influence on the total energy saved;

12 % of electricity is saved

8.3 Resume table

SITUATION		ELECTRIC ENERGY SAVED
Dry cooler system working along		**91,7%**
Working on parallel Standard system/New system	From 8ºC->3ºC / 3ºC->0ºC	**57,3%**
	From 8ºC->4ºC / 4ºC->0ºC	**45,3%**
	From 8ºC->5ºC / 5ºC->0ºC	**34,3%**
	From 8ºC->6ºC / 6ºC->0ºC	**23%**
	From 8ºC->7ºC / 7ºC->0ºC	**12%**

Table 65; resume table of the different saves

As it is shown in the **table 65** the different saving percentages are quite high so the system can be quite profitable for a company which is placed in Nordic countries as Sweden. It can be added that if the company is located the most to the north then the total energy saved is going to be higher because the temperatures are going to be lower than in the south.

9 DETERMINATION OF SAVINGS DEPENDING OF THE OUTSIDE TEMPERATURES RANGE

At this point it is known the percentage of electricity savings in terms of operating ranges of temperatures of the rooms. Throughout the entire project has been made studies in terms of absolute values, now it is going to determine the specific amount of energy savings depending on the outside temperatures.

To demonstrate what has been said it has been prepared one table with all the possible cases that can occur when the outside temperature is between -20 º C and 0 º C, the specific range during which the dry coolers system will be running.

To understand the development of the table it must be followed three basic rules, these design parameters have been fixed and it cannot be change, in our case they are the system limitations, these are;

Another key factor is that the outside is considered as a heat focus, so it can be taken all the necessary cold without changing its temperature (This had been said on the external exchanger calculations)

To finish only remains to indicate that the blue columns are referring to the outside and yellow columns to the inside cooled enclosures. Green boxes mean that inside temperature indicated can be achieved with the outside temperature of the same row, if it is red means that the temperature cannot be achieved.

When all the boxes in the last column are red for an outside temperature or coincides only with 8 º C in the interior, the system will be stopped because they cannot decrease at least one degree the inside temperature.

THERMAL FOCUS											
OUTSIDE [ºC]			INSIDE [ºC]			|T1-Tw2|[ºC]=4					
T1o	|Tw$_o$1-Tw$_o$2|=3[ºC]		T1i	|Twi1-Twi2|=3[ºC]		OUTSIDE	F4	INSIDE	F4	CASE	PERCENTAGE OF ENERGY SAVED
	Tw1o	Tw2o		Tw1i	Tw2i						
-20	-13	-16	**8**	-16	-13	4	12	21	2		
-20	-13	-16	**7**	-16	-13	4	12	20	2		
-20	-13	-16	**6**	-16	-13	4	12	19	3		
-20	-13	-16	**5**	-16	-13	4	12	18	3		
-20	-13	-16	**4**	-16	-13	4	12	17	3	***Dry cooler system working alone***	***91,7%***
-20	-13	-16	**3**	-16	-13	4	12	16	3		
-20	-13	-16	**2**	-16	-13	4	12	15	3		
-20	-13	-16	**1**	-16	-13	4	12	14	3		
-20	-13	-16	**0**	-16	-13	4	12	13	4		
-19	-12	-15	**8**	-15	-12	4	12	20	2		
-19	-12	-15	**7**	-15	-12	4	12	19	3		
-19	-12	-15	**6**	-15	-12	4	12	18	3		
-19	-12	-15	**5**	-15	-12	4	12	17	3		
-19	-12	-15	**4**	-15	-12	4	12	16	3	***Dry cooler system working alone***	***91,7%***
-19	-12	-15	**3**	-15	-12	4	12	15	3		
-19	-12	-15	**2**	-15	-12	4	12	14	3		
-19	-12	-15	**1**	-15	-12	4	12	13	4		
-19	-12	-15	**0**	-15	-12	4	12	12	4		
-18	-11	-14	**8**	-14	-11	4	12	19	3		
-18	-11	-14	**7**	-14	-11	4	12	18	3		
-18	-11	-14	**6**	-14	-11	4	12	17	3		
-18	-11	-14	**5**	-14	-11	4	12	16	3		
-18	-11	-14	**4**	-14	-11	4	12	15	3	***Dry cooler system working alone***	***91,7%***
-18	-11	-14	**3**	-14	-11	4	12	14	3		
-18	-11	-14	**2**	-14	-11	4	12	13	4		
-18	-11	-14	**1**	-14	-11	4	12	12	4		
-18	-11	-14	**0**	-14	-11	4	12	11	4		
-17	-10	-13	**8**	-13	-10	4	12	18	3		
-17	-10	-13	**7**	-13	-10	4	12	17	3	***Dry cooler system working***	***91,7%***
-17	-10	-13	**6**	-13	-10	4	12	16	3		

-17	-10	-13	**5**	-13	-10	4	12	15	3	***alone***	
-17	-10	-13	**4**	-13	-10	4	12	14	3		
-17	-10	-13	**3**	-13	-10	4	12	13	4		
-17	-10	-13	**2**	-13	-10	4	12	12	4		
-17	-10	-13	**1**	-13	-10	4	12	11	4		
-17	-10	-13	**0**	-13	-10	4	12	10	5		
-16	-9	-12	**8**	-12	-9	4	12	17	3	***Dry cooler system working alone***	***91,7%***
-16	-9	-12	**7**	-12	-9	4	12	16	3		
-16	-9	-12	**6**	-12	-9	4	12	15	3		
-16	-9	-12	**5**	-12	-9	4	12	14	3		
-16	-9	-12	**4**	-12	-9	4	12	13	4		
-16	-9	-12	**3**	-12	-9	4	12	12	4		
-16	-9	-12	**2**	-12	-9	4	12	11	4		
-16	-9	-12	**1**	-12	-9	4	12	10	5		
-16	-9	-12	**0**	-12	-9	4	12	9	5		
-15	-8	-11	**8**	-11	-8	4	12	16	3	***Dry cooler system working alone***	***91,7%***
-15	-8	-11	**7**	-11	-8	4	12	15	3		
-15	-8	-11	**6**	-11	-8	4	12	14	3		
-15	-8	-11	**5**	-11	-8	4	12	13	4		
-15	-8	-11	**4**	-11	-8	4	12	12	4		
-15	-8	-11	**3**	-11	-8	4	12	11	4		
-15	-8	-11	**2**	-11	-8	4	12	10	5		
-15	-8	-11	**1**	-11	-8	4	12	9	5		
-15	-8	-11	**0**	-11	-8	4	12	8	6		
-14	-7	-10	**8**	-10	-7	4	12	15	3	***Dry cooler system working alone***	***91,7%***
-14	-7	-10	**7**	-10	-7	4	12	14	3		
-14	-7	-10	**6**	-10	-7	4	12	13	4		
-14	-7	-10	**5**	-10	-7	4	12	12	4		
-14	-7	-10	**4**	-10	-7	4	12	11	4		
-14	-7	-10	**3**	-10	-7	4	12	10	5		
-14	-7	-10	**2**	-10	-7	4	12	9	5		
-14	-7	-10	**1**	-10	-7	4	12	8	6		
-14	-7	-10	**0**	-10	-7	4	12	7	7		
-13	-6	-9	**8**	-9	-6	4	12	14	3	***Dry cooler system***	***91,7%***
-13	-6	-9	**7**	-9	-6	4	12	13	4		

-13	-6	-9	**6**	-9	-6	4	12	12	4	***working alone***	
-13	-6	-9	**5**	-9	-6	4	12	11	4		
-13	-6	-9	**4**	-9	-6	4	12	10	5		
-13	-6	-9	**3**	-9	-6	4	12	9	5		
-13	-6	-9	**2**	-9	-6	4	12	8	6		
-13	-6	-9	**1**	-9	-6	4	12	7	7		
-13	-6	-9	**0**	-9	-6	4	12	6	8		
-12	-5	-8	**8**	-8	-5	4	12	13	4	***Dry cooler system working alone***	***91,7%***
-12	-5	-8	**7**	-8	-5	4	12	12	4		
-12	-5	-8	**6**	-8	-5	4	12	11	4		
-12	-5	-8	**5**	-8	-5	4	12	10	5		
-12	-5	-8	**4**	-8	-5	4	12	9	5		
-12	-5	-8	**3**	-8	-5	4	12	8	6		
-12	-5	-8	**2**	-8	-5	4	12	7	7		
-12	-5	-8	**1**	-8	-5	4	12	6	8		
-12	-5	-8	**0**	-8	-5	4	12	5	9		
-11	-4	-7	**8**	-7	-4	4	12	12	4	***Dry cooler system working alone***	***91,7%***
-11	-4	-7	**7**	-7	-4	4	12	11	4		
-11	-4	-7	**6**	-7	-4	4	12	10	5		
-11	-4	-7	**5**	-7	-4	4	12	9	5		
-11	-4	-7	**4**	-7	-4	4	12	8	6		
-11	-4	-7	**3**	-7	-4	4	12	7	7		
-11	-4	-7	**2**	-7	-4	4	12	6	8		
-11	-4	-7	**1**	-7	-4	4	12	5	9		
-11	-4	-7	**0**	-7	-4	4	12	4	12		
-10	-3	-6	**8**	-6	-3	4	12	11	4	***Dry cooler system working alone***	***91,7%***
-10	-3	-6	**7**	-6	-3	4	12	10	5		
-10	-3	-6	**6**	-6	-3	4	12	9	5		
-10	-3	-6	**5**	-6	-3	4	12	8	6		
-10	-3	-6	**4**	-6	-3	4	12	7	7		
-10	-3	-6	**3**	-6	-3	4	12	6	8		
-10	-3	-6	**2**	-6	-3	4	12	5	9		
-10	-3	-6	**1**	-6	-3	4	12	4	12		
-10	-3	-6	**0**	-6	-3	4	12	3	15		
-9	-2	-5	**8**	-5	-2	4	12	10	5	***Dry cooler***	***91,7%***

-9	-2	-5	**7**	-5	-2	4	12	9	5	*system working alone*	
-9	-2	-5	**6**	-5	-2	4	12	8	6		
-9	-2	-5	**5**	-5	-2	4	12	7	7		
-9	-2	-5	**4**	-5	-2	4	12	6	8		
-9	-2	-5	**3**	-5	-2	4	12	5	9		
-9	-2	-5	**2**	-5	-2	4	12	4	12		
-9	-2	-5	**1**	-5	-2	4	12	3	15		
-9	-2	-5	**0**	-5	-2	4	12	2	23		
-8	-1	-4	**8**	-4	-1	4	12	9	5	**From 8ºC->3ºC / 3ºC->0ºC**	***57,3%***
-8	-1	-4	**7**	-4	-1	4	12	8	6		
-8	-1	-4	**6**	-4	-1	4	12	7	7		
-8	-1	-4	**5**	-4	-1	4	12	6	8		
-8	-1	-4	**4**	-4	-1	4	12	5	9		
-8	-1	-4	**3**	-4	-1	4	12	4	12		
-8	-1	-4	**2**	-4	-1	4	12	3	15		
-8	-1	-4	**1**	-4	-1	4	12	2	23		
-8	-1	-4	**0**	-4	-1	4	12	1	44		
-7	0	-3	**8**	-3	0	4	12	8	6	***From 8ºC->4ºC / 4ºC->0ºC***	***45,3%***
-7	0	-3	**7**	-3	0	4	12	7	7		
-7	0	-3	**6**	-3	0	4	12	6	8		
-7	0	-3	**5**	-3	0	4	12	5	9		
-7	0	-3	**4**	-3	0	4	12	4	12		
-7	0	-3	**3**	-3	0	4	12	3	15		
-7	0	-3	**2**	-3	0	4	12	2	23		
-7	0	-3	**1**	-3	0	4	12	1	44		
-7	0	-3	**0**	-3	0	4	12	0	-		
-6	1	-2	**8**	-2	1	4	12	7	7	***From 8ºC->5ºC / 5ºC->0ºC***	***34,3%***
-6	1	-2	**7**	-2	1	4	12	6	8		
-6	1	-2	**6**	-2	1	4	12	5	9		
-6	1	-2	**5**	-2	1	4	12	4	12		
-6	1	-2	**4**	-2	1	4	12	3	15		
-6	1	-2	**3**	-2	1	4	12	2	23		
-6	1	-2	**2**	-2	1	4	12	1	44		
-6	1	-2	**1**	-2	1	4	12	0	-		
-6	1	-2	**0**	-2	1	4	12	-1	-		

-5	2	-1	**8**	-1	2	4	12	6	8	*From 8ºC->6ºC / 6ºC->0ºC*	***23,0%***
-5	2	-1	**7**	-1	2	4	12	5	9		
-5	2	-1	**6**	-1	2	4	12	4	12		
-5	2	-1	**5**	-1	2	4	12	3	15		
-5	2	-1	**4**	-1	2	4	12	2	23		
-5	2	-1	**3**	-1	2	4	12	1	44		
-5	2	-1	**2**	-1	2	4	12	0	-		
-5	2	-1	**1**	-1	2	4	12	-1	-		
-5	2	-1	**0**	-1	2	4	12	-2	-		
-4	3	0	**8**	0	3	4	12	5	9	*From 8ºC->7ºC / 7ºC->0ºC*	***12,0%***
-4	3	0	**7**	0	3	4	12	4	12		
-4	3	0	**6**	0	3	4	12	3	15		
-4	3	0	**5**	0	3	4	12	2	23		
-4	3	0	**4**	0	3	4	12	1	44		
-4	3	0	**3**	0	3	4	12	0	-		
-4	3	0	**2**	0	3	4	12	-1	-		
-4	3	0	**1**	0	3	4	12	-2	-		
-4	3	0	**0**	0	3	4	12	-3	-		
-3	4	1	**8**	1	4	4	12	4	12	*NOT WORKING*	*NOT WORKING*
-3	4	1	**7**	1	4	4	12	3	15		
-3	4	1	**6**	1	4	4	12	2	23		
-3	4	1	**5**	1	4	4	12	1	44		
-3	4	1	**4**	1	4	4	12	0	-		
-3	4	1	**3**	1	4	4	12	-1	-		
-3	4	1	**2**	1	4	4	12	-2	-		
-3	4	1	**1**	1	4	4	12	-3	-		
-3	4	1	**0**	1	4	4	12	-4	-		
-2	5	2	**8**	2	5	4	12	3	15	*NOT WORKING*	*NOT WORKING*
-2	5	2	**7**	2	5	4	12	2	23		
-2	5	2	**6**	2	5	4	12	1	44		
-2	5	2	**5**	2	5	4	12	0	-		
-2	5	2	**4**	2	5	4	12	-1	-		
-2	5	2	**3**	2	5	4	12	-2	-		
-2	5	2	**2**	2	5	4	12	-3	-		
-2	5	2	**1**	2	5	4	12	-4	-		

-2	5	2	**0**	2	5	4	12	-5	-		
-1	6	3	**8**	3	6	4	12	2	23	***NOT WORKING***	***NOT WORKING***
-1	6	3	**7**	3	6	4	12	1	44		
-1	6	3	**6**	3	6	4	12	0	-		
-1	6	3	**5**	3	6	4	12	-1	-		
-1	6	3	**4**	3	6	4	12	-2	-		
-1	6	3	**3**	3	6	4	12	-3	-		
-1	6	3	**2**	3	6	4	12	-4	-		
-1	6	3	**1**	3	6	4	12	-5	-		
-1	6	3	**0**	3	6	4	12	-6	-		
0	7	4	**8**	4	7	4	12	1	44	***NOT WORKING***	***NOT WORKING***
0	7	4	**7**	4	7	4	12	0	-		
0	7	4	**6**	4	7	4	12	-1	-		
0	7	4	**5**	4	7	4	12	-2	-		
0	7	4	**4**	4	7	4	12	-3	-		
0	7	4	**3**	4	7	4	12	-4	-		
0	7	4	**2**	4	7	4	12	-5	-		
0	7	4	**1**	4	7	4	12	-6	-		
0	7	4	**0**	4	7	4	12	-7	-		

Table 66; Resume of all possible cases

T outside [ºC]	T inside got [ºC]	CASE	CASE DESCRIPTION	ENERGY SAVED (%)
-20->-11	0	T	Dry cooler system working alone	91,7
-10	1	T	Dry cooler system working alone	91,7
-9	2	T	Dry cooler system working alone	91,7
-8	3	1	From 8ºC->3ºC / 3ºC->0ºC	57,3
-7	4	2	From 8ºC->4ºC / 4ºC->0ºC	45,3
-6	5	3	From 8ºC->5ºC / 5ºC->0ºC	34,3
-5	6	4	From 8ºC->6ºC / 6ºC->0ºC	23
-4	7	5	From 8ºC->7ºC / 7ºC->0ºC	12
-3	NOT WORKING	NOT WORKING	NOT WORKING	NOT WORKING
-2	NOT WORKING	NOT WORKING	NOT WORKING	NOT WORKING
-1	NOT WORKING	NOT WORKING	NOT WORKING	NOT WORKING
0	NOT WORKING	NOT WORKING	NOT WORKING	NOT WORKING

Table 67; Relationship between the outside temperature/inside temperatures reached/Energy saved

From the table it can deduced that the operating system range depending on the outside temperature is very wide, from -20 º C to -4 º C, but the amount of energy saved changes substantially.

This fact requires a very precise study of the externals temperatures for estimating the cost savings accurately.

10 SAVING ESTIMATED WITH THE NEW SYSTEM APPLICATED INTO WILLYS SUPERMARKET

It is impossible to make an accurate prediction of the future climate, but it can be got a good estimation of the range of cost savings that the system can get. Do not forget that the parameters have been chosen with the worst cases when designing the system, so in any case the results would be equal or better than those which have been calculated. It is important to say that the external temperature profile provided don´t vary.

The season in which has been examined the system has been the winter of 09/10. The first and most important step is to achieve a given temperature record in the shortest time interval possible. This has not been possible, but from SMHI it has been able to achieve average daily temperature records, with maximum and minimum daily temperature.

As we have seen in the initial estimation of temperature it is not necessary to have to take into account the daily average, because they might be out of range of use and this fact has got erroneous results. It must be said that on the same day the temperature can vary greatly.

So to get a more objective result, it has been made the following assumptions;

- The ambient temperatures change strongly with the hours of the day and night (It have been split the hours of daylight and night for each winter month in Gävle).

- Since it is not true that the system does not work on a day when the average temperature is below -4 º C; it has been assumed that the temperature of daylight is the average of the daily average temperature and the maximum one. For the overnight hours the opposite happens, the average of the daily average temperature and the minimum.

Thus below a table shows the temperatures for each month; the average temperature, maximum, minimum, and temperatures estimated during the day and night.

The columns named as "CASE" makes reference to the case that is occurring in that interval. (In the daylight hours or night hours) cases that can occur have been calculated before and are shown on yellow into the **table 66**.

Inside the columns "case" values appear in green and white, when the value is green means that the temperature is enough to enable the system working.

If the box is white means that there are no conditions for the system to produce cold, so it is not possible to save a representative amount of money.

DECEMBER 2009

DECEMBER 2009							
Day	T [ºC]	Tmax [ºC]	Tmin [ºC]	Tday [ºC]	*CASE*	Tnigth	*CASE*
1	0	6	-4	3	-	-2	-
2	-5	-2	-10	-3	-	-7	*2*
3	-3	0	-10	-1	-	-6	*3*
4	0	2	-3	1	-	-2	-
5	3	5	0	4	-	1	-
6	3	5	2	4	-	2	-
7	3	4	2	3	-	2	-
8	2	3	1	2	-	1	-
9	2	3	1	2	-	1	-
10	2	2	1	2	-	2	-
11	0	2	-2	1	-	-1	-
12	0	2	-2	1	-	-1	-
13	-1	0	-4	0	-	-3	-
14	-3	0	-5	-1	-	-4	*5*
15	-3	-3	-5	-3	-	-4	*5*
16	-3	-2	-4	-2	-	-4	*5*
17	-5	-4	-8	-4	*5*	-7	*2*
18	-14	-4	-18	-9	*T*	-16	*T*
19	-13	-5	-19	-9	*T*	-16	*T*
20	-4	-3	-13	-4	-	-9	*T*
21	-13	-5	-19	-9	*T*	-16	*T*
22	-13	-9	-16	-11	*T*	-14	*T*
23	-7	-5	-16	-6	*3*	-11	*T*
24	-6	-3	-13	-5	*4*	-9	*T*
25	-1	1	-4	0	-	-2	-
26	-2	0	-4	-1	-	-3	-
27	-11	-3	-17	-7	*2*	-14	*T*
28	-8	-6	-10	-7	*2*	-9	*T*
29	-16	-8	-22	-12	*T*	-19	*T*
30	-20	-11	-23	-15	*T*	-22	*T*
31	-11	-8	-22	-9	*T*	-16	*T*

Table 68; resume of the different cases in December

Now only it has to be remained to summarize the total hours in which the system is operating in each different case. Keep in mind that the hours of daylight and night are different in the different months, for December, the average hours of day and night is; ***Light hours = 6h, night hours = 18h***, so;

RESUME OF WORKING HOURS	*HOURS WORKING*											
	CASE											
	T		*1*		*2*		*3*		*4*		*5*	
	D	*N*	*D*	*N*	*D*	*N*	*D*	*N*	*D*	*N*	*D*	*N*
Nº of occurrences/month	7	12	0	0	2	2	1	1	1	0	1	3
Hours (Day&Night)/month	42	216	0	0	12	36	6	18	6	0	6	54
Total hours/month	***258***		***0***		***48***		***24***		***6***		***60***	

Table 69; resume of working hours to the different cases in December

To know the energy consumption in the hours calculated before, it is needed to know the total system power, this data is obtained from the **table 59**, ***Ptotal = 20,9 [kW]***

CASE	SAVED (%)	HOURS	[kWh]BEFORE	[kWh]AFTER
T	91,7	258	5392	448
1	57,3	0	0	0
2	45,3	48	1003	549
3	34,3	24	502	330
4	23	6	125	97
5	12	60	1254	1104
	TOTAL [kWh]		**8276**	**2526**
	[kWh/month] SAVED		***5750***	
	[SEK/ month] SAVED		***4140***	

Table 70; calculation of energy and money saved in December

The total cost saved in December of 2009 could have been of **4140 SEK** using the dry cooling system working on parallel with the old one.

JANUARY 2009

JANUARY 2010							
Day	T [ºC]	Tmax [ºC]	Tmin [ºC]	Tday [ºC]	*CASE*	Tnigth	*CASE*
1	-6,1	-4,7	-8,6	**-5**	***4***	**-7**	***2***
2	-9,3	-5,7	-16,7	**-8**	***1***	**-13**	***T***
3	-8,7	-3,7	-17,5	**-6**	***3***	**-13**	***T***
4	-9,3	-3,2	-16	**-6**	***3***	**-13**	***T***
5	-19,3	-10,7	-25,6	**-15**	***T***	**-22**	***T***
6	-23,3	-14,3	-30,3	**-19**	***T***	**-27**	***T***
7	-12,5	-11,1	-22,5	**-12**	***T***	**-18**	***T***
8	-18,3	-11,9	-23,8	**-15**	***T***	**-21**	***T***
9	-24,4	-16,7	-28,2	**-21**	***T***	**-26**	***T***
10	-21,2	-14,3	-25,9	**-18**	***T***	**-24**	***T***
11	-16,5	-10,2	-21,8	**-13**	***T***	**-19**	***T***
12	-15,5	-8,9	-18	**-12**	***T***	**-17**	***T***
13	-17,1	-10,6	-20,8	**-14**	***T***	**-19**	***T***
14	-14,9	-10,2	-20,9	**-13**	***T***	**-18**	***T***
15	-9,8	-5,3	-16,9	**-8**	***1***	**-13**	***T***
16	-3,8	-3,3	-6,6	**-4**	***5***	**-5**	***4***
18	-2,7	-2,2	-3,7	**-2**	-	**-3**	-
19	-3,8	-2,6	-5,4	**-3**	-	**-5**	***4***
20	-3,6	-2,2	-7,9	**-3**	-	**-6**	***3***
21	-5,3	-2,3	-11,2	**-4**	***5***	**-8**	***1***
22	-4,9	-3,4	-7,1	**-4**	***5***	**-6**	***3***
23	-4,7	-3,6	-6,7	**-4**	***5***	**-6**	***3***
24	-6,3	-4,7	-7,7	**-6**	***3***	**-7**	***2***
25	-9,6	-6,1	-17,8	**-8**	***1***	**-14**	***T***
26	-11,1	-8,4	-19,8	**-10**	***T***	**-15**	***T***
27	-7,6	-4,8	-10,7	**-6**	***3***	**-9**	***T***
28	-6,5	-3,9	-9,2	**-5**	***4***	**-8**	***1***
29	-10,4	-8,3	-12,5	**-9**	***T***	**-11**	***T***
30	-12,9	-11,8	-13,5	**-12**	***T***	**-13**	***T***
31	-16,3	-11,4	-26,2	**-14**	***T***	**-21**	***T***

Table 71; Resume of the different cases in January

Now only it has to be remained to summarize the total hours in which the system is operating in each different case. Keep in mind that the hours of daylight and night are different in the different months, for December, the average hours of day and night is; ***Light hours = 6,5h, night hours = 17,5h***, so;

RESUME OF WORKING HOURS	HOURS WORKING											
	CASE											
	T		1		2		3		4		5	
	D	N	D	N	D	N	D	N	D	N	D	N
Nº of occurrences/month	14	20	3	2	0	2	4	3	2	2	4	0
Hours (Day/Night)/month	91	350	20	35	0	35	26	52,5	13	35	26	0
Total hours/month	*441*		*54,5*		*35*		*78,5*		*48*		*26*	

Table 72; resume of working hours to the different cases in January

To know the energy consumed in the hours calculated before, it needs to know the total system power, this data is obtained from the **table 59**, P_{total} ***= 20,9 [kW]***

CASE	SAVED (%)	HOURS	[kWh]BEFORE	[kWh]AFTER
T	91,7	441	9217	765
1	57,3	54,5	1139	486
2	45,3	35	732	400
3	34,3	78,5	1641	1078
4	23	48	1003	772
5	12	26	543	478
	TOTAL [kWh]		**14275**	**3980**
	[kWh/month] SAVED		*10295*	
	[SEK/ month] SAVED		*7412*	

Table 73; calculation of energy and money saved in January

The total cost saved in January of 2010 could have been of **7412 SEK** using the dry cooling system working on parallel with the old one.

FEBRUARY 2010

FEBRUARY 2010							
Day	T [ºC]	Tmax [ºC]	Tmin [ºC]	Tday [ºC]	*CASE*	Tnigth	*CASE*
1	-10,2	-3,3	-26,4	-7	2	-18	T
2	-9,9	-4,2	-17,9	-7	2	-14	T
3	-2,8	-0,7	-18,8	-2	-	-11	T
4	-1,4	-0,6	-2,6	-1	-	-2	-
5	-3,3	-1,9	-7,2	-3	-	-5	4
6	-4,9	-1,2	-15,4	-3	-	-10	T
7	-9,1	-2,6	-17,5	-6	3	-13	T
8	-3,9	1,3	-10,2	-1	-	-7	2
9	-3,8	-1,3	-5,1	-3	-	-4	5
10	-5,8	-4,1	-7,9	-5	4	-7	2
11	-8,6	-5,1	-15,8	-7	2	-12	T
12	-7,3	-2,2	-17,3	-5	4	-12	T
13	-10,2	-4,7	-18,8	-7	2	-15	T
14	-7,6	0,7	-16,7	-3	-	-12	T
15	-11,5	-1,1	-19,2	-6	3	-15	T
16	-6,3	-3,1	-16,2	-5	4	-11	T
17	-5,5	-3,9	-7,8	-5	4	-7	2
18	-9,6	-5,2	-10,9	-7	2	-10	T
19	-9,1	-8,1	-10,9	-9	T	-10	T
20	-12,1	-8,2	-12,8	-10	T	-12	T
21	-15,5	-11,4	-22,9	-13	T	-19	T
22	-19	-13,5	-29,3	-16	T	-24	T
23	-14,4	-12,2	-16,2	-13	T	-15	T
24	-13,8	-6,9	-24,1	-10	T	-19	T
25	-7,3	-2,2	-14,5	-5	4	-11	T
26	-1,4	0,4	-7,2	-1	-	-4	5
27	-2,5	0,2	-4,6	-1	-	-4	5
28	-4,8	-3,2	-5,3	-4	5	-5	4

Table 74; resume of the different cases in February

Now only it has to be remained to summarize the total hours in which the system is operating in each different case. Keep in mind that the hours of daylight and night are different in the different months, for December, the average hours of day and night is; ***Light hours = 9h, night hours = 15h***, so;

RESUME OF WORKING HOURS	*HOURS WORKING*											
	CASE											
	T		*1*		*2*		*3*		*4*		*5*	
	D	*N*	*D*	*N*	*D*	*N*	*D*	*N*	*D*	*N*	*D*	*N*
Nº of occurrences/month	6	19	0	0	5	3	2	0	5	2	1	3
Hours (Day/Night)/month	54	285	0	0	45	45	18	0	45	30	9	45
Total hours/month	***339***		***0***		***90***		***18***		***75***		***54***	

Table 75; resume of working hours to the different cases in February

To know the energy consumed in the hours calculated before, it needs to know the total system power, this data is obtained from the **table 59**, ***P_{total} = 20,9 [kW]***

CASE	SAVED (%)	HOURS	[kWh]BEFORE	[kWh]AFTER
T	91,7	339	7085	588
1	57,3	0	0	0
2	45,3	90	1881	1029
3	34,3	18	376	247
4	23	75	1568	1207
5	12	54	1129	993
	TOTAL [kWh]		**12038**	**4064**
	[kWh/month] SAVED		*7974*	
	[SEK/ month] SAVED		***5741***	

Table 76; calculation of energy and money saved in February

The total cost saved in February of 2010 could have been of **5741 SEK** using the dry cooling system working on parallel with the old one.

MARCH 2010

MARCH 2010							
Day	T	TM	Tm	Tdia	*CASE*	Tnoche	*CASE*
1	-5,2	-4,4	-6,3	**-5**	***4***	**-6**	***3***
2	-5,8	-3,2	-9,4	**-5**	***4***	**-8**	***1***
3	-5,8	-4,1	-8,7	**-5**	***4***	**-7**	***2***
4	-5,8	-3,2	-8,8	**-5**	***4***	**-7**	***2***
5	-9,5	0	-18,4	**-5**	***4***	**-14**	***T***
6	-11,1	3,1	-21,5	**-4**	***5***	**-16**	***T***
7	-6,9	1,7	-19,2	**-3**	**-**	**-13**	***T***
8	0,4	8,2	-9,2	**4**	**-**	**-4**	***5***
9	0,4	4,2	-3,2	**2**	**-**	**-1**	**-**
10	4,4	9,2	-0,2	**7**	**-**	**2**	**-**
11	3,2	7,2	-0,4	**5**	**-**	**1**	**-**
12	-0,7	4,6	-5,9	**2**	**-**	**-3**	**-**
13	-2	2,8	-8,7	**0**	**-**	**-5**	***4***
14	-2,2	1,8	-5,9	**0**	**-**	**-4**	**-**
15	-3,4	2,2	-9,9	**-1**	**-**	**-7**	***2***
16	-5,5	-1,4	-11,2	**-3**	**-**	**-8**	***1***
17	-2	4,1	-11,7	**1**	**-**	**-7**	***2***
18	1,9	4,4	-5,2	**3**	**-**	**-2**	**-**
19	3,4	5,4	-1,9	**4**	**-**	**1**	**-**
20	5,6	11,3	0,2	**8**	**-**	**3**	**-**
21	-2,4	3,3	-8,3	**0**	**-**	**-5**	***4***
22	-1,9	4,9	-11,7	**2**	**-**	**-7**	***2***
23	-3,4	0,2	-8,8	**-2**	**-**	**-6**	***3***
24	-2,5	3,7	-13,7	**1**	**-**	**-8**	***1***
25	1,1	2,7	-1,1	**2**	**-**	**0**	**-**
26	1,5	5,4	0,1	**3**	**-**	**1**	**-**
27	-0,2	2,2	-1,2	**1**	**-**	**-1**	**-**
28	0,2	1,2	-0,7	**1**	**-**	**0**	**-**
29	-0,7	0	-1,7	**0**	**-**	**-1**	**-**
30	0	2,2	-2,3	**1**	**-**	**-1**	**-**
31	-0,6	0,7	-3,4	**0**	**-**	**-2**	**-**

Table 77; Resume of the different cases in March

Now only it has to be remained to summarize the total hours in which the system is operating in each different case. Keep in mind that the hours of daylight and night are different in the different months, for December, the average hours of day and night is; ***Light hours = 11,5h, night hours = 12,5h***, so;

RESUME OF WORKING HOURS	*HOURS WORKING*											
	CASE											
	T		*1*		*2*		*3*		*4*		*5*	
	D	*N*	*D*	*N*	*D*	*N*	*D*	*N*	*D*	*N*	*D*	*N*
Nº of occurrences/month	0	3	0	3	0	5	0	2	5	2	1	1
Hours (Day/Night)/month	0	38	0	38	0	62,5	0	25	58	25	12	13
Total hours/month	***37,5***		***37,5***		***62,5***		***25***		***82,5***		***24***	

Table 78; resume of working hours to the different cases in March

To know the energy consumed in the hours calculated before, it needs to know the total system power, this data is obtained from the **table 59**, ***P_{total} = 20,9 [kW]***

CASE	SAVED (%)	HOURS	[kWh]BEFORE	[kWh]AFTER
T	91,7	37,5	784	65
1	57,3	37,5	784	335
2	45,3	62,5	1306	715
3	34,3	25	523	343
4	23	82,5	1724	1328
5	12	24	502	441
	TOTAL [kWh]		**5622**	**3227**
	[kWh/month] SAVED		***2396***	
	[SEK/ month] SAVED		***1725***	

Table 79; calculation of energy and money saved in March

The total cost saved in March of 2010 could have been of **1725 SEK** using the dry cooling system working on parallel with the old one.

10.1 RESUME TABLE

MONTH	ENERGY SAVED [kWh/year]	COST SAVED [SEK/year]	% Saved over total
December 09	5750	4140	22
January 10	10295	7412	39
February 10	7974	5741	30
March 10	2396	1725	9
TOTAL	**26415 [kWh/year]**	**19190 [SEK/year]**	**-**

Table 80; saving summary

These results have been obtained with a running time of the cooling system of 100%. To figure out which would be savings with a working time of 85%, or what is the same, a stopping time of 15% (Appearing the warm up of the rooms). The saving should be calculated by multiplying those values by 0.85.

11 FEASIBILITY AND BUDGET

In the previous section can be seen that the annual savings is quite good. However, to know if the implementation of the system is successful talking economically, it is first necessary to make an estimate of the budget as accurate as possible.

For this, we have checked several suppliers of components needed for installation, to finally produce the following table to deduce the total cost;

Type of element	Description	Characteristic	Longitude [m]	Cost per unity [kr/m]	Total cost [kr]
Pipe	Main pipe, from external exchanger to main branch.	ϕ=100mm	55	3990	219450
Pipe	Secondary pipe, from main branch to Area 1	ϕ=76mm	3	2910	8730
Pipe	Secondary pipe, from main branch to Area 2	ϕ=38mm	7	1836	12852
Pipe	Secondary pipe, from main branch to Area 3	ϕ=76mm	3	2910	8730
Pipe	Secondary pipe, from main branch to Area 4	ϕ=51mm	7	2416	16912
Pipe	Secondary pipe, from main branch to Area 5	ϕ=51mm	3	2416	7248
Pipe'	Main pipe, from indoor to outdoor exchanger	ϕ=100mm	55	3990	219450
Pipe'	Secondary pipe, from area 1 to main branch	ϕ=76mm	3	2910	8730
Pipe'	Secondary pipe, from area 2 to main branch	ϕ=38mm	7	1836	12852
Pipe'	Secondary pipe, from area 3 to main branch	ϕ=76mm	3	2910	8730
Pipe'	Secondary pipe, from area 4 to main branch	ϕ=51mm	7	2416	16912
Pipe'	Secondary pipe, from area 5 to main branch	ϕ=51mm	3	2416	7248
Internal exchanger	Inside Area 1; WR1480.B	105,7 [kW]	-	92289	92289
Internal exchanger	Inside Area 2; WR1263.A	24,3 [kW]	-	32529	32529

Internal exchanger	Inside Area 3; WR1480.B	105,7 [kW]	-	92289	92289
Internal exchanger	Inside Area 4; WR1263.B	26,1 [kW]	-	36576	36576
Internal exchanger	Inside Area 5; WR1363.B	39,5 [kW]	-	47861	47861
External exchanger	GR2680.B	308 [kW]	-	201472	201472
Pump	Pump for driving of the hydraulic circuit	P= 3 [kW] Q= 60 [m3/h] Hb= 12,3 [m]	-	5500	5500
				SUBTOTAL	***1056360***
Others	Not estimated costs; switchgear safety, hydraulics elements, etc.	-	-	5% of the subtotal	52818
				TOTAL	***1109178***

Table 81; Brief budget

It is important to have into account that all the last prices include both manpower and material cost.

Therefore the total investment cost to realize the installation is **1109178 kr**.

Here it has a reference to can estimate if the installation will be feasible or not. To calculate this it is going to use the **L**ife **C**ycle **C**ost equation;

$$\mathbf{L = K + D}$$
$$\mathbf{L = K + d \cdot f}$$

Where;

- L = Life cycle cost
- K = investment cost
- D = maintenance cost through life cycle
- d = maintenance cost per year
- f = factor that relations the capital cost and the life cycle

From the LCC formula it knows the investment cost and it is necessary to calculate the maintenance cost to one year and the 'f' factor;

The maintenance cost has been estimated in **5000kr/year**, because for this system is not necessary practically realizing maintenance. This cost is quite smaller than other for a standard installation with the same power.

Finally to calculate the 'f' factor it has been done by the next way;

$$f(r,n) = \frac{1}{r} \cdot \left[1 - \left(\frac{1}{1+r}\right)^{n}\right]$$

Where;

- r = capital cost, for a energy installation investment is about 2%
- n = installation life, for an installation of this kind is around 15 years.

So;

$$f(0.02,15) = \frac{1}{0.02} \cdot \left[1 - \left(\frac{1}{1+0.02}\right)^{15}\right] = 12{,}85$$

To end, the total cost in 15 year, with a capital cost of 2% is;

L= 1109178 + 5000*12,85 = 1173428 SEK

The energy saved in that time is;

$$\mathbf{E_{saved} = 15\ [years] \cdot 19190\ [SEK/year] = 287850\ SEK}$$

For the system is feasible it must occur that $\mathbf{E_{saved} > L}$, thus;

$$\mathbf{E_{saved} > L}$$

287850 SEK << 1173428 SEK

Hence, in spite of the good energy saving, it is not economical feasible due to the components and their installation is very expensive.

HÖGSKOLAN I GÄVLE

ANNEX II

LIST OF FIGURES AND TABLES

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