SUPERMARKET REFRIGERATION WITH DECENTRALISED PUMPS

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Supermarkets are intensive users of energy in most countries. Electricity consumption in large supermarkets in US and France is estimated to 4% of the national electric energy use. The electrical energy used in the Swedish supermarkets account for 3% (1.8TWh) of the annual national electricity consumption. Further, the refrigeration system in the supermarket accounts for about 40-50% of the electricity used. Thus, this area is an interesting field when looking at energy saving in supermarkets.

This article focuses on the indirect supermarket refrigeration system and specifically the energy saving potential of the distribution energy. A concept for decentralised pumps in indirect refrigeration systems has been evaluated and applied to a miniature supermarket which was installed in a laboratory with controlled conditions. The system was well instrumented to measure the required system parameters. Three steps were taken in the evaluation where the first step was fixed pump speed, followed by capacity controlled pump conditions and at last the decentralised pump concept was evaluated. This study had a special focus on the pump energy saving and the system level energy saving.

Supermarket refrigeration systems
There are mainly three types of systems for the supermarket refrigeration; the stand alone equipment, condensing units and centralised units. In the latter category the indirect systems are found which implies a central refrigeration unit and a secondary fluid is used for the distribution to cabinets, cold rooms, etc.

Figure 1. A common type of display cabinet with an indirect refrigeration system.

Normally, there are two temperature levels in the supermarkets: medium and low temperature level. Medium temperature ranges from 1°C to 14°C with typical evaporation temperature from -15°C to 5°C, and low temperature ranges from -18°C to -12°C with the typical evaporation temperature from -40°C to -30°C. The choice of different temperatures depends on the food property, display cabinets and the refrigeration system.

Indirect refrigeration systems typically use 15-40% of the input electrical energy for the energy distribution on the cold and warm side. This includes brine and coolant pumps, and fans on cabinets and dry coolers. Normally the brine pumps account for about half of the mentioned parasite energies, which is a considerable amount of energy. The latter should be put in the context that you pay more than once since in a cooling system you need to compensate for the added energy from the brine pump. In reality you pay 1.3 kWh per kWh invested in the pump (if COP=3).

Still today many common installations use is a fixed speed pump in the main loop regardless the flow requirement. When a part of the system is controlled by on-off solenoid valve, the pressure difference in the main loop is changing severely. Better control methods based on the flow adaptation are coming up.

However, decentralised pump systems would offer further reduced flow losses and potentially higher pump efficiencies if controlled properly.

Refrigeration and distribution system

The refrigeration system has two separate identical chillers with the refrigerant of R404A. Two main parts of the distribution system are the pump unit and the display cabinets group. The central pump is installed on the return line to ensure working at highest possible brine temperature. It can run at two different capacity controlled modes which are constant pressure control and variable pressure control.

Different distribution system requires different pump capacity, in other words, different speed. A too high speed produces more flow than necessary. In this case, more heat will be generated and it needs to be covered by more cooling capacity from the chillers.

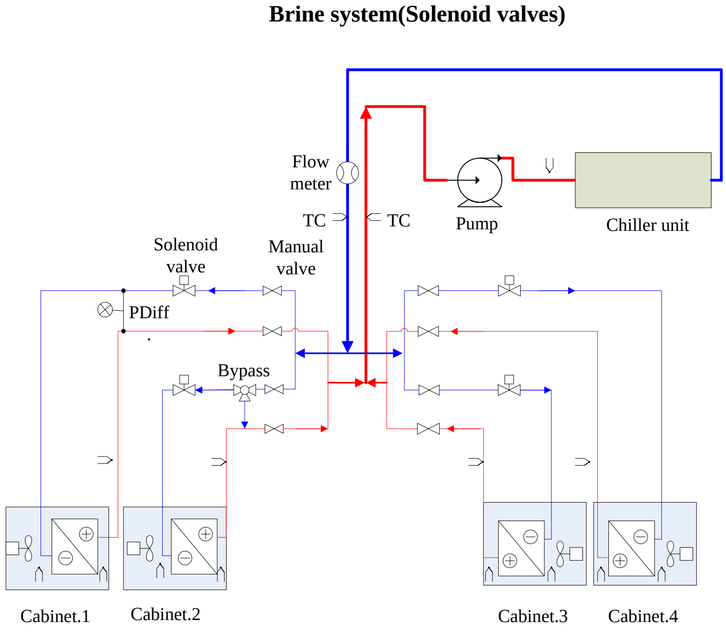


Figure 2. The cold side distribution system (secondary loop) of the test system.

Figure 3 shows the layout of a "standard" indirect refrigeration system with a central pump and the traditional solenoid valve control. The draw back of this system solution is that in order to get proper flow in all parts of the system, restrictions that "force" the flow to every extremity need to be introduced. In fact the largest pressure drops in the system are the ones deliberately introduced for flow adjustment. Another disadvantage is that if the central pump is capacity controlled the distribution can still be affected.

To evaluate the different systems solutions measurement equipment is applied that can monitor temperature levels, cooling capacity, brine flow, energy, etc. To compare the performance of different distribution system, there are different conditions that must be kept constant. One is the food temperature in the cabinets, the other one is the room condition. The food temperature is measured from three food simulators in each cabinet. There are three test steps in this section; first is the fixed pump speed; second test is the capacity controlled conditions and the last is the decentralised pump condition.

Results

The fixed speed started with the maximum speed of the pump and worked its way down to the point where not enough flow was provided to keep stable conditions in the cabinets. During the test, the control equipments in the cabinet were solenoid valves where each has a power consumption of 10W. When the cabinets call for cooling, the solenoid valves to open and let the brine through.

An external signal can be used to run the pump at different speeds. The pump input power ranges from 0.3 to 5.45kW depending on the flow and pump speed.

Since the reference fixed speed was minimised and the load changes in such a laboratory based system is minimal the influence of the capacity control is negligible, so further analysis with regards to this specific system is not necessary.

Decentralised pumps
Decentralised pump, also called multi pump system, which means instead of only having one pump in the main loop, there are several pumps installed in the branches. The potential advantages of decentralised pump system are; lower energy consumption, better flow distribution for each cabinet, because each cabinet will be taken care of by its own pump resulting in more stable temperatures; no need for control valves; potentially lower installation cost (less labour work, no need to adjust the flow of each cabinet). On the negative side is a slightly higher component cost.

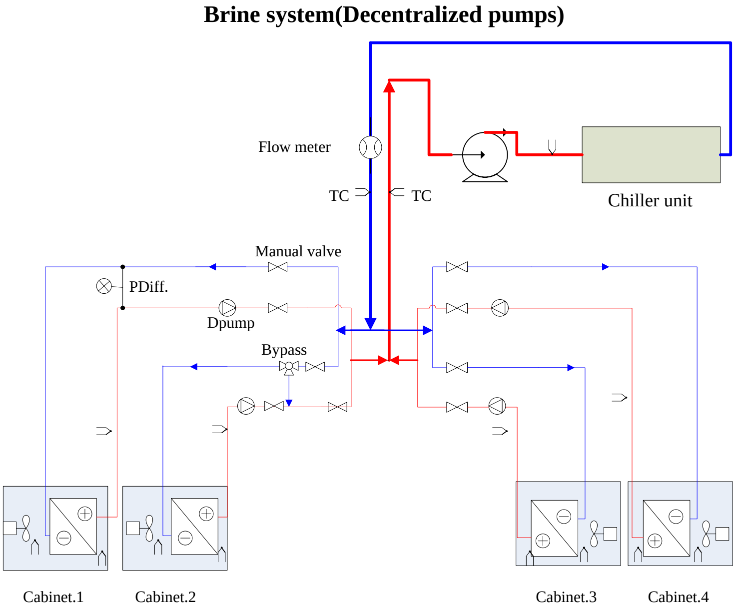


Figure 3. System sketch of the decentralised pump system.

In this test system (3), the pump model is Wilo-stratos-PARA (4) with the max power of 70 W. In the past such small pumps have had poor efficiency but today they have approached the larger pumps and have comparable efficiency.

Wilo-stratos-PARA 15/1-7	
Power supply P_1	0.07kW
Shaft power P_2	0.05kW
Rotation speed *n*	1100 to 4450 rpm
External signal	3 to 10 V
Pressure difference	1 to 7 m

Figure 4. Picture of the decentralised pumps and the technical data

The average power over 24 hours for the solenoid valves and the decentralised pumps are 10W and 23W respectively. The average power of the central pump over 24 hours was also reduced from 778W (central pump with lowest speed) to 550W.
The energy consumption over 24 hrs of the different distribution system concepts are compared in figure 5. *Cpump* stands for the central pump, *Svalve* is all four solenoid valves, *Dpump* is all four decentralised pumps, *Cab*. is all fans and cabinet controllers, *comp1* is the compressor of the main chiller and *comp2* is the compressor of the second chiller. From 10V to 6.5V are the fixed pump speed tests, followed by capacity controlled modes, then the decentralised pump test.

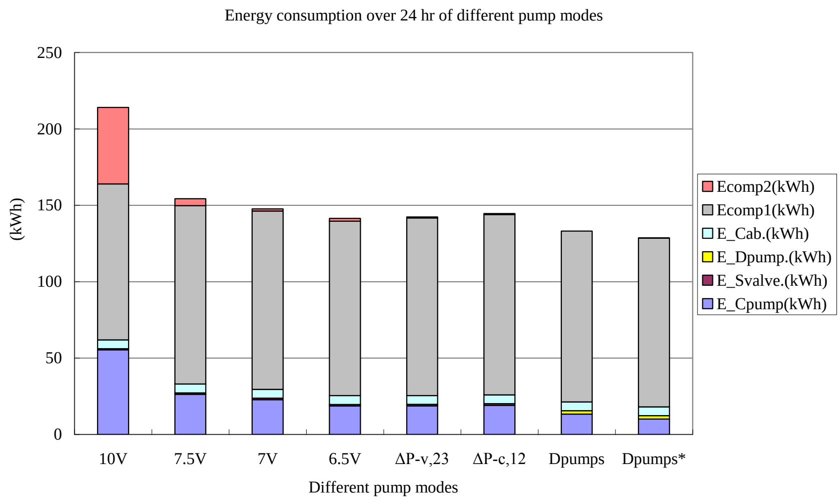


Figure 5. Energy consumption of each equipment over 24 hr for different pump modes.

The central pump used in the test had too high capacity which resulted in poor efficiency when controlled to a low capacity. The presented result *Dpump** is based on the assumption that a proper sized central pump is used and the pump power is calculated based on the real conditions from *Dpump*.

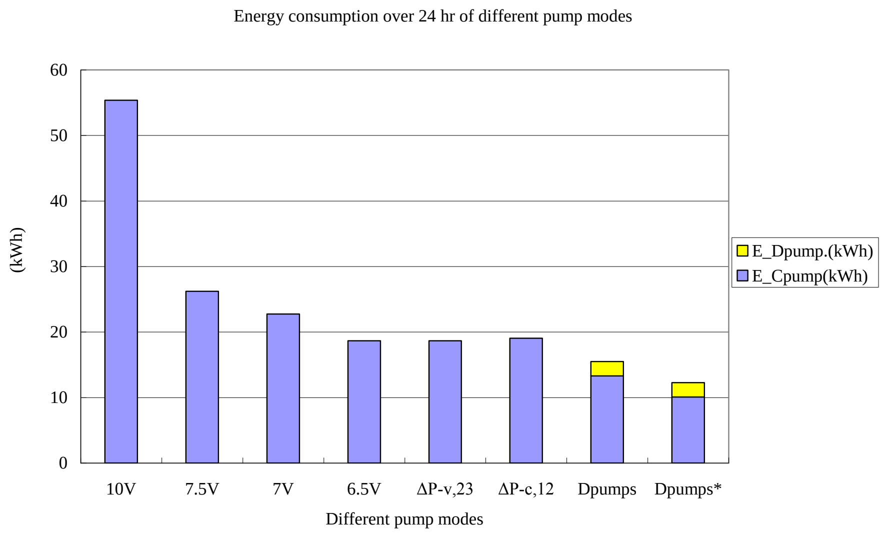


Figure 6. Energy consumption of the pumps only (central and decentralised) over 24 hrs.

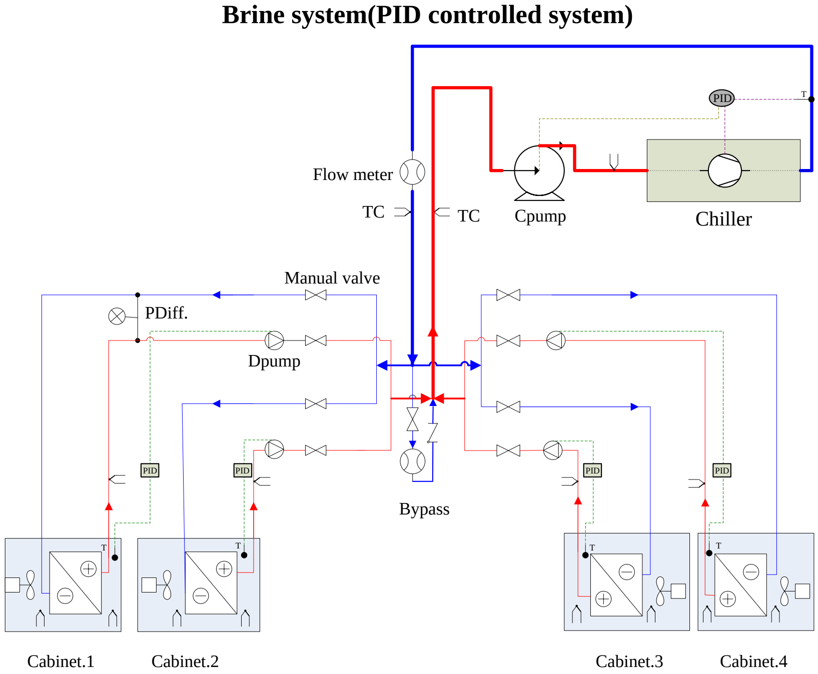
When only the pump energy consumption is considered, figure 6, the saving from best possible centralised condition (6.5V) compared to the measured decentralised, *Dpumps*, is 17%. If then the central pump is changed to a proper size, *Dpumps**, the saving is 34%. However, in reality most systems do not operate at the "optimised" central pump setting so the actual saving potential is most likely higher.

As far as the overall energy saving is concerned, with the optimized centralised pump condition as reference, 6.0% total energy and 17% pump energy can be saved. If the central pump would have been properly sized the results are 9.0% total energy and 34% pump energy respectively.
Other advantages with the decentralised concept can be mentioned such as; shorter start up time for the installation of the decentralised pumps system by avoiding the flow adjustment. The decentralised pump concept would be a "plug and play" solution. This also applies for a potential extension of a refrigeration system, which normally would require a new adjustment of the system.

A further advantage is the improved controllability of the cabinet air temperature. It is shown that the supply air temperature can be more accurately controlled. At the end of the day all oscillations implies losses.
Finally the cost aspect indicates an increased investment of about 100 Euro per cabinet, which will have to be put into perspective considering the potential savings mentioned above.

Decentralised Pumps in Supermarket Refrigeration

- Phase 2 -



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ABSTRACT

This project reports results from the second phase of a study on decentralised pumps used in refrigeration system application. The project has mainly been financed by KYS (Kylbranschens Samarbetsstiftelse).

The project has shown interesting results with respect to the decentralised pumps in indirect refrigeration systems. In phase 1 a traditional system solution with a central pump was compared with the first generation decentralised pump system. The conclusion was an energy saving of 6% on a system level and 17% of the pump distribution energy.

The improvement in phase 1 consisted of two major reasons; the control strategy and the decentralised pumps. The latter will take a part of the work from the central pump but the adaption to the required flow is the main driver for the reduction in energy usage.

In phase 2 an adaptation of the secondary refrigerant piping reduced the flow resistance, however, the main driver was the coordinated compressor and pump control. Due to the pump work which is adapted to the cooling capacity the pump energy is significantly reduced.

Overall in the project a 17% energy reduction was achieved on a system level. The cold side distribution energy was reduced from 21% to 18.6% in phase 1. In phase 2 this was further reduced to 12% and in relation to the refrigeration energy the corresponding figures are 9.7%, 8.4% och 5.6%.

The interesting part is the relative improvement rather the than the absolute level of the distribution energy, since the system design highly influences the pump work. The present system is of an ultimate design and therefore the pump work is relatively high.

The increased cost of the present concept is expected to be approximately 100 Euro per display case. This will have to be trade off against the energy saving, the reduced assembly and adjustment time and the improved temperature control.

PREFACE

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1 INTRODUCTION

With the rising awareness of the global energy situation, the efficiency of all energy systems gradually comes in focus. This applies to all the energy related fields which range from huge country electricity transmission system to tiny computer chips. Supermarket refrigeration system is one of the energy systems of interest.

In terms of electricity consumption, supermarkets in Sweden account for 3% of total amount of 1.8TWh (Sjöberg, 1997) in the national level. The refrigeration energy consumption accounts for around a half of this. There is a huge potential to save the energy in the supermarket refrigeration field.

In the previous project phase 1 (Jan 2009 to June 2009) a saving in pump energy of 17% by introducing decentralized pumps was realised. On a system level the saving was between 7 and 9% depending on the reference.
To further improve the concept a second project phase was performed during the time from October 2009 to March 2010.

1.1 Background

Pump is used to provide the flow in the distribution loop. The normal condition is with a fixed speed uncontrolled pump in the main loop no matter how the flow changes. Since each branch is controlled by on-off solenoid valve, the pressure difference in the main loop is changing considerably. Thus, some better control methods based on flow variation are coming up. The decentralized pump system is one of those leading methods.

The success of decentralized pump usage in heating system has been proven. The pump manufacturer Wilo did field tests years ago and came up with the conclusion that the decentralized pump system could decrease the energy consumption by up to 20%.
Despite the advantage of energy saving, there are three other advantages:

1. It is more convenient for the customer because the quick heating process and the better room temperature stability can be achieved;
2. A pleasant automatic hydraulic balancing is achieved, which leads to the easier job for the craftsmen in terms of the system installation and flow adjustment.
3. It provides the better feasibility for the building central control in terms of the data diagnosis and energy management.

To investigate whether if the decentralized pump system can have the same advantages on the cooling usage, this project was carried out.

1.1 Aim of the project

The aim of the project is to:

- Show the decentralized pumps influence on the indirect refrigeration system overall performance
- Optimise a test system as reference
- Show practical gains with a decentralized pump arrangement on the cold brine side
- Deliver practical results from the test system
- Show a cost estimate of the system solution
- Verify that the cold side distribution energy can be reduced to maximum 4% of the total refrigeration system energy input
- Demonstrate that a typical supermarket system can save 10 MWh per year

2 PHASE 2 SYSTEM IMPROVEMENT

This chapter describes the main changes in the system compared to phase 1. Inherent weaknesses in the system discovered in the first phase were focused on, such as adoptions allowing evaluating new ideas.

2.1 Reconstruction and modification of the brine system

In response to previous test results the brine system has been upgraded in order to optimize it even further. The new layout is obtained from Figure 2-1. In comparison with the old system, the new brine system consists of larger brine tubes at the chiller side. The purpose of the larger tubes is to decrease the pressure loss when the brine is operating at low temperatures and consequently at high viscosity. It has been proven from the previous tests that one chiller is enough to cover the cooling load. Thus, the back up chiller has been disconnected to avoid the electricity consumption during its standby time.

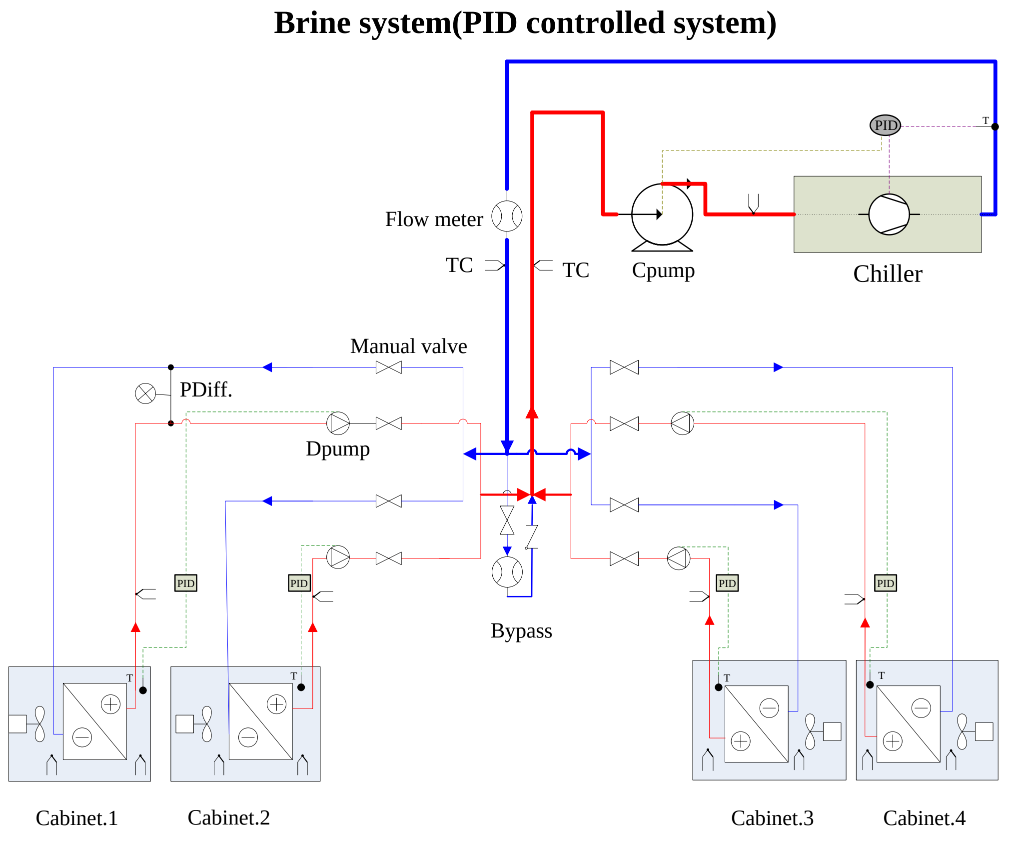

Figure 2-1. The new and improved brine system.

The construction of the brine system forced the old central pump to operate at low region of the pump property flow chart. The operating point had a negative impact on the efficiency of the pump. By replacing the larger pump by a smaller sized pump (as can be obtained from Figure 2-2) this could be solved. The main flow could now be maintained at a central region of the pump property curve and thereby improving the pump efficiency.

Figure 2-2 The new and smaller sized central pump.

The position of the bypass function is the last modification of the old system. Previously, the bypass function has been located at one branch of the distribution system, which has proven to be a poor design. In the new setup, the bypass is located at the main branch and associated with a flow meter in order to supervise the bypass brine flow.

2.2 System control

2.2.1 PID controller of compressor

The frequency controlled compressor is regulated by the brine out temperature at the chiller side. A T-type thermocouple is installed at the brine out spot and the signal is transferred to the PID frequency controller (Figure 2-3) through a converter (Figure 2-4). The operational range of the converter is between -15°C to 30°C and with the current output ranging from 0 to 20 mA. A linear function between the temperature and ampere can be obtained to correlate these two parameters.

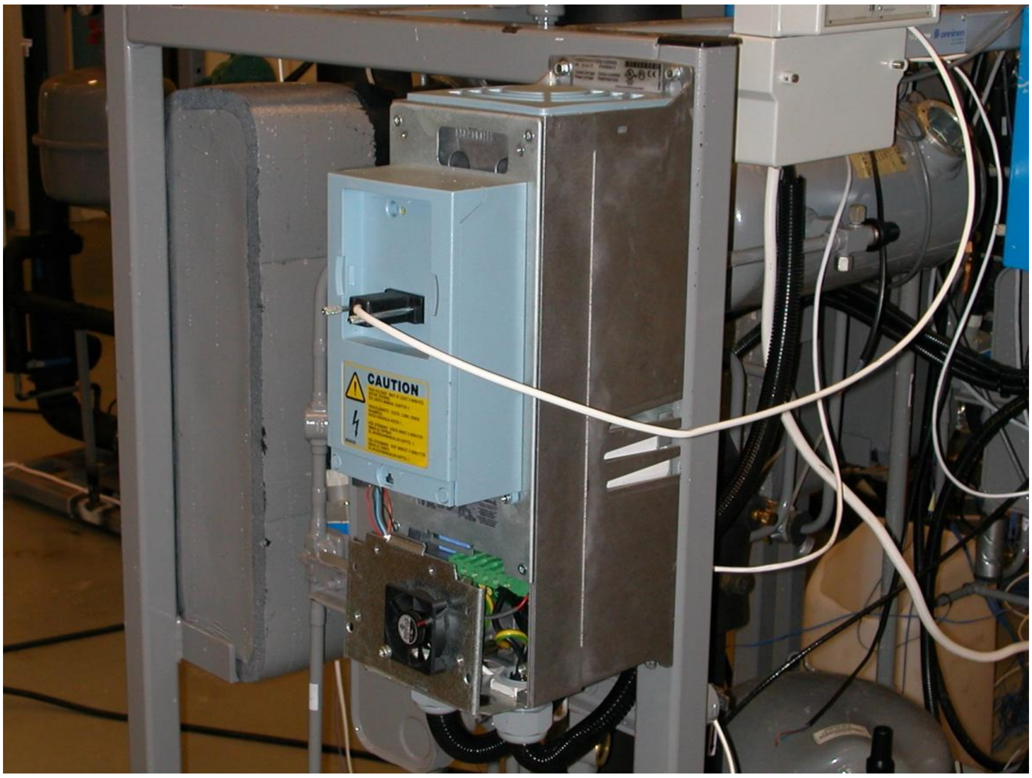

Figure 2-3 Frequency controller for the compressor control

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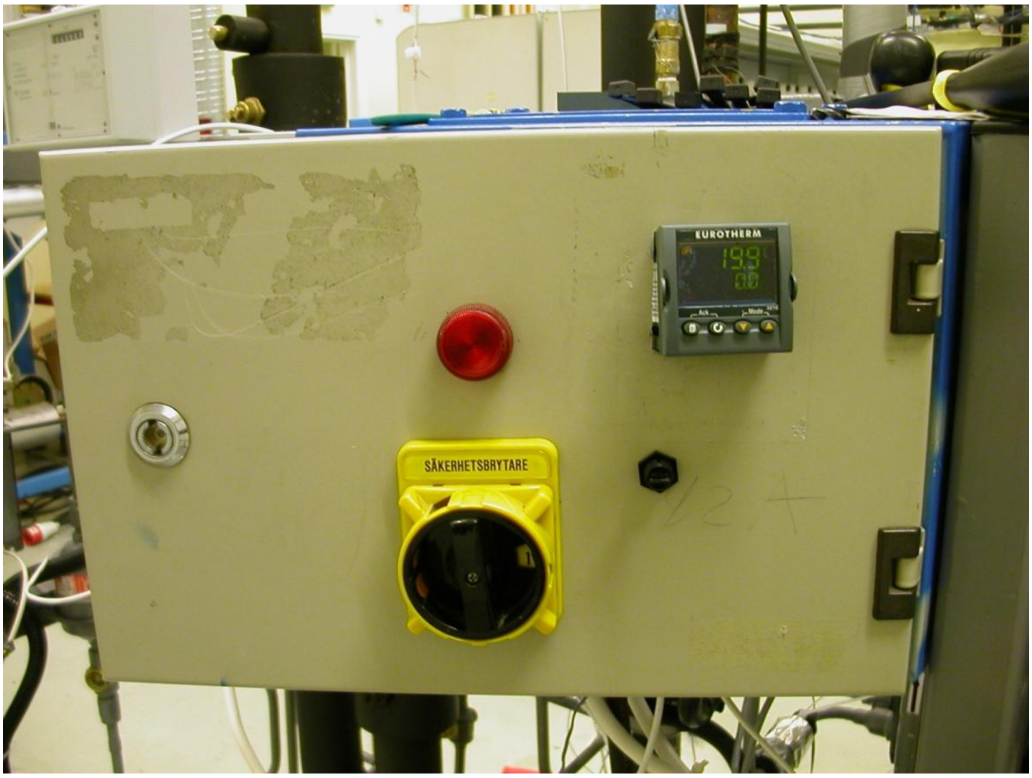

Figure 2-4 Temperature converter for the compressor control.

2.2.2 PID control of decentralized pumps

Each decentralized pump is regulated by a PID controller (Figure 2-5). The input signals are obtained from the air-out temperature of the cabinet and the set point of the controllers is adjusted to 2°C.

Figure 2-5 PID controllers of decentralized pumps.

3 TEST EQUIPMENT - SYSTEM TEST AND TUNING

In the following section the verification of the test system including tuning is presented. The aim is to give a picture of the test environment and repeatability of the tests.

3.1 Test system – ambient conditions

In order to compare the results between the various tests, the surrounding conditions must remain stable and fixed during the entire period.

The basic requirements are:

1. The food temperatures are controlled between 3 and 6^{o}C (Figure 3-1).
2. Room temperature is controlled around 20 ^{o}C (Figure 3-2).
3. Room humidity is controlled at around 36% (Figure 3-1).
   (It shows temporarily a 10% lower value in the figure below)

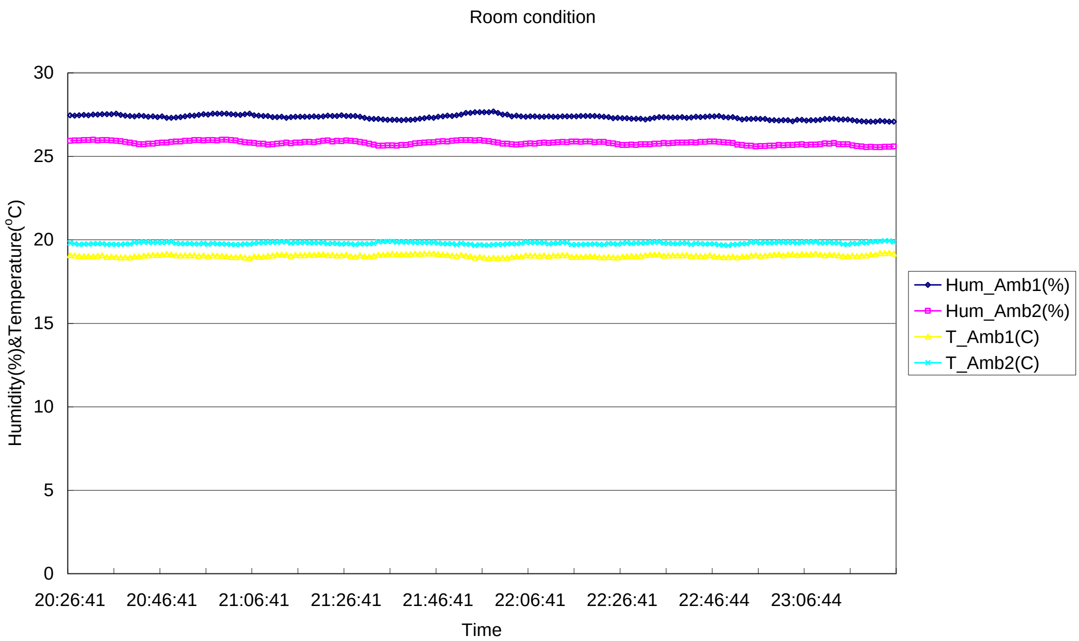

Figure 3-1 Controlled room conditions with temperature and humidity.

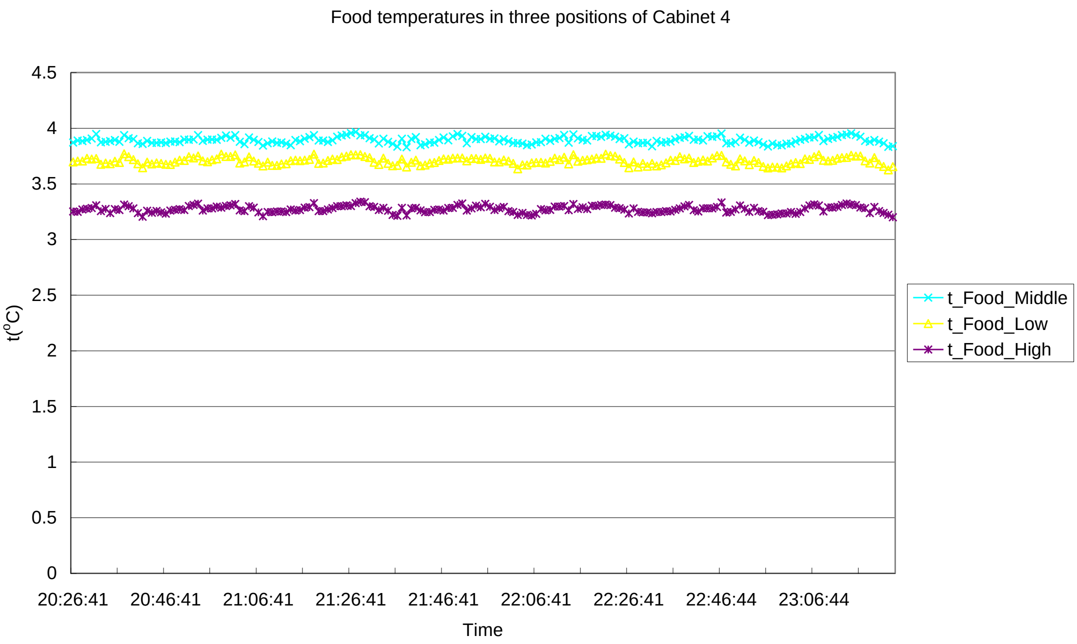

Figure 3-2 Controlled food simulator temperatures in one cabinet.

3.2 Test 1: Bypass open without check valve.

At test number one, the decentralized pumps were running at the PID-mode (set-point = 2°C) and the central pump was set to ΔP-c=10m. The compressor control parameter, t_setting, was selected to -4°C. At the established operating condition, the required outlet air temperature of 2°C at the cabinets could not be fulfilled even thus all of the decentralized pumps were operating at the full load.

The phenomenon could be explained by backflow in the brine circuit. The backflow short cut the brine loop so that the cabinets cannot get the low temperature brine flow from the chiller. No matter how hard the decentralized pump worked, they are still pumping the old cabinets flow over and over, only with a little mixture of the fresh brine flow from the chiller.

Thus, a check valve is necessary in the bypass to avoid the backflow.

3.3 Test 2: Bypass closed and changed t_setting of compressor controller.

During this test, t_setting was reduced gradually from a start value of -2°C down to -6°C, with a temperature interval of 1°C. The bypass backflow was totally avoided in this test by using a check valve. Although the backflow had been solved a new problem occurred.

From the logger, it was clear that with decreasing t_setting, the food temperature in each cabinet was also decreasing. At some moment, the decentralized pump stopped operating occurring to the PID regulation. Even thou the decentralized pumps were in off-mode the cabinet outlet air temperature were still getting below 2^{o}C. Both above two phenomenon proved that the central pump was overriding the decentralized pumps which made the cabinet temperatures loss control.

Thus, the obtained test data were not valid for analyzing the system.

3.4 Test 3: Decreased ΔP-pump of central pump.

In order to not override the decentralized pumps the condition of the Central pump was set to ΔP-c=6m. The t_setting was changed from -6 ^{o}C up to -2 ^{o}C with the interval of 2^{o}C, in other words, three steps were conducted in this test. During each step, the “air out temperature” of each cabinet could be kept around the set value (2^{o}C). Since the boundary conditions could be kept rather constant during each step of t_setting, the data was valid to be analyzed.

3.4.1 Central pump parameters curve in different t_setting.

T_setting is the reference value to control the compressor speed. The input signal comes from the brine out temperature at the evaporator side. During this test, the central pump was always working at ΔP-c=6m mode while the t_setting of compressor was changing at 3 steps, which were -6(Figure 1), -4 (Figure 2) and -2°C (Figure 3) respectively.

As can be seen from the figures, with increasing t_setting, the Flow_BriM and Eff_pump will increase accordingly. Pdiff_Cpump is kept constant as it is set to “constant pressure control”-mode.

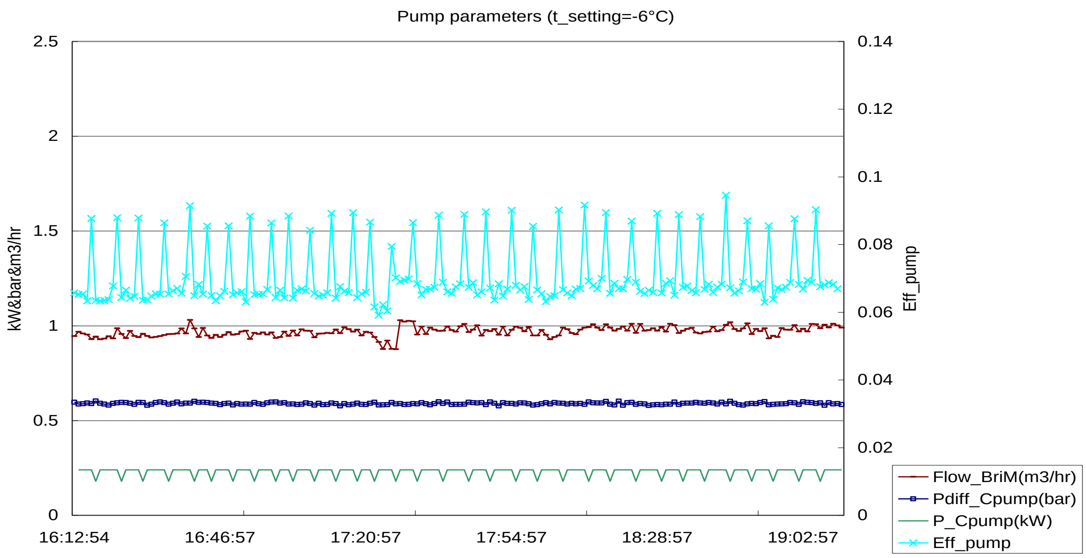

Figure 1 Pump parameters at t_setting=-6^{o}C

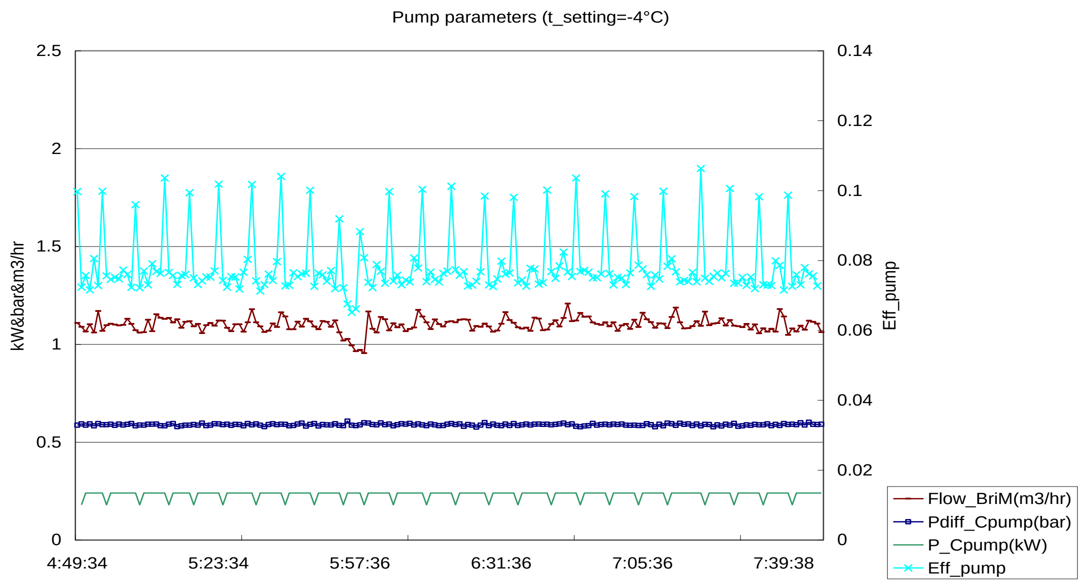

Figure 2 Pump parameters at t_setting=-4^{o}C

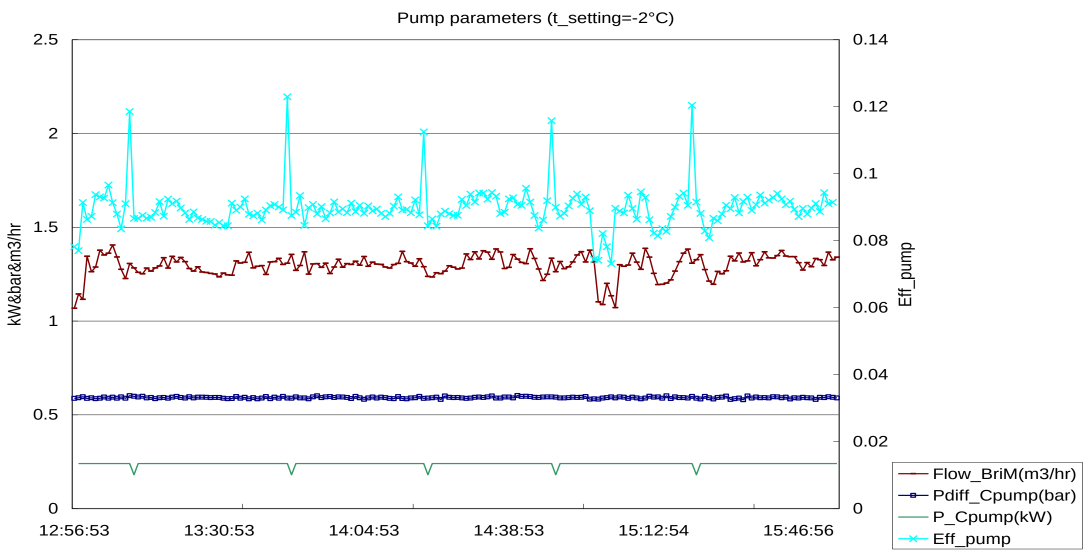

Figure 3 Pump parameters at t_setting=-2^{o}C

3.4.2 Q_Controller VS P_Comp over 3 hours in each condition

The aim of this section is to evaluate the frequency controller losses. It is not only of general interest but also specifically important to make the correct assumptions for the compressor energy balance.

Q_Controller stands for the energy loss from the frequency controller, which is calculated with the equation:

$$Q\_Controller = AirFlow * \rho_{air} * C_{p\_air} * \Delta t \qquad \text{Equation 1}$$

Where AirFlow (m^3/s) stands for the air volume flow pass through the frequency controller, ρ_{air}(m^3/kg) is the air density, Cp_air (KJ/(kg*K)) is the specific heat capacity of air, Δt is the temperature difference of air in and air out of the frequency controller.
The air in temperature is the room temperature and the air out temperature is measured by a thermocouple (Figure 4) at the air outlet duct of frequency controller.

Figure 4 The frequency controller mounted with the thermocouple at the air outlet.

P_Comp actually consists of the power of both compressor and frequency controller (Q_Controller), since the power supply is connected to the same electricity socket, only one energy meter is used.

Figure 12, Figure 13 and Figure 14 show the Q_Controller value varies from 0.12 to 0.18 kW and the proportion over P_Comp varies from 2 to 7%. The value of Q_Controller was not significantly influenced by the changing of P_Comp, so it didn't appear a trend when P_Comp was increasing. In this case, the proportion over P_Comp will surely decrease when the P_Cump was increasing.

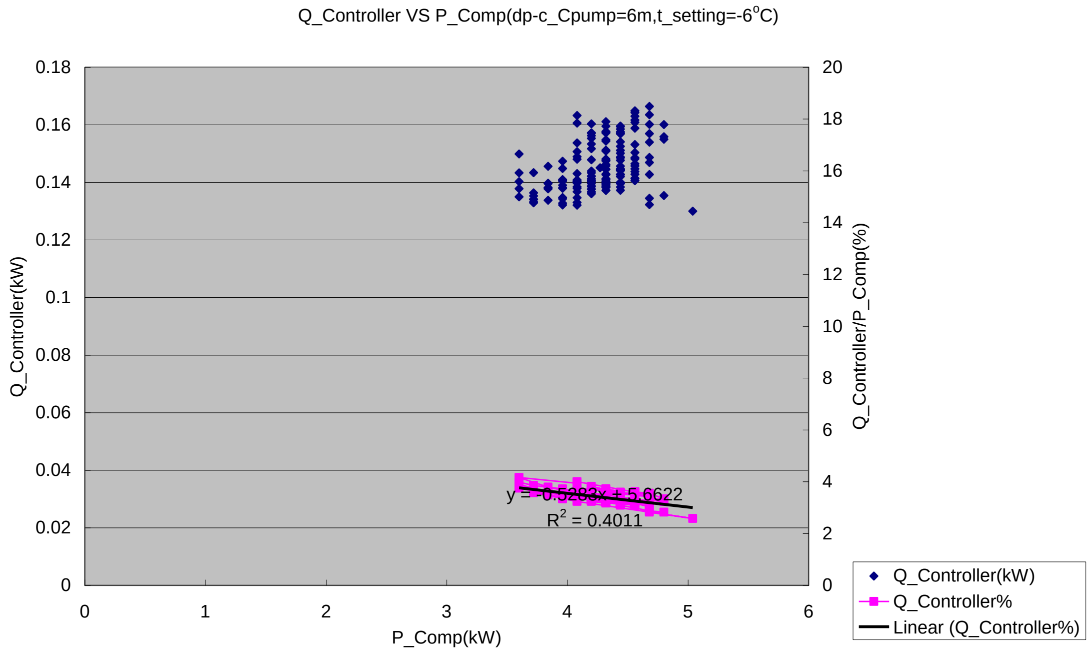

Figure 5 Q_Controller's real value and proportion VS P_Comp when t_setting=-6^{o}C.

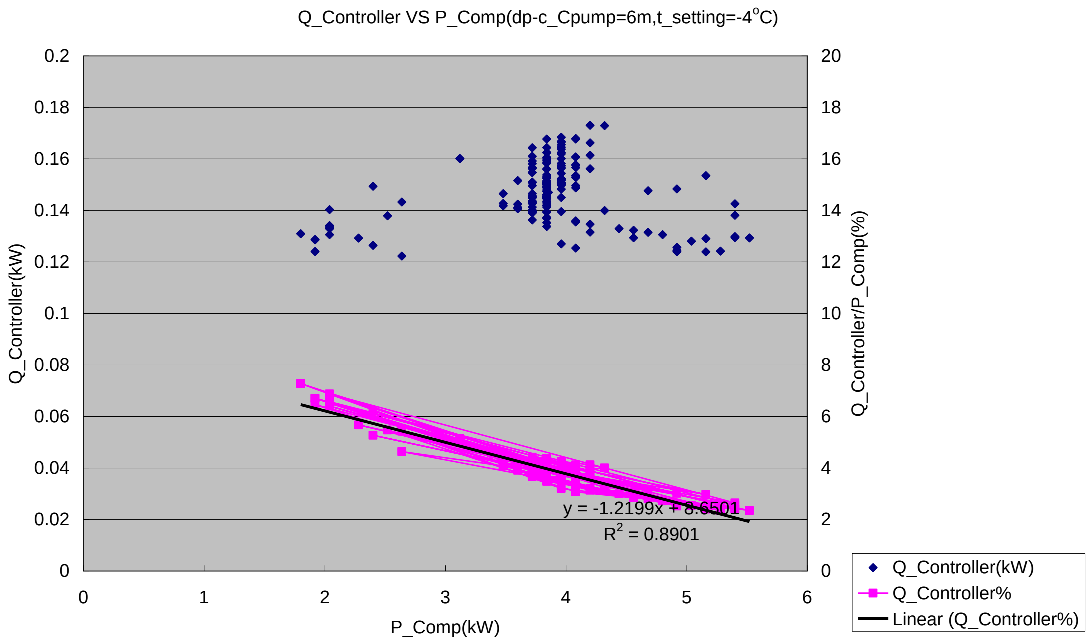

Figure 6 Q_Controller's real value and proportion VS P_Comp when t_setting=-4^{o}C.

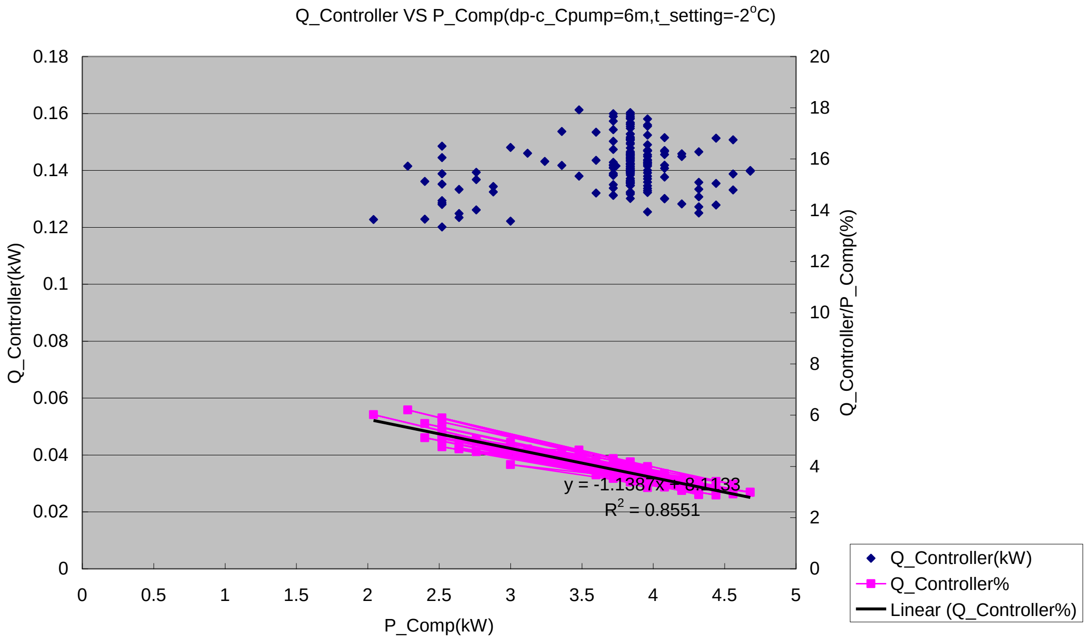

Figure 7 Q_Controller's real value and proportion VS P_Comp when t_setting=-2^{o}C.

4 TEST RESULTS

This chapter first presents the test procedure followed by the results obtained for the respective test section.

4.1 Test procedure

The test steps were followed as below:

1. **PID controlled compressor, ΔP-c mode Cpump and Dpumps:**
   - Adjusted compressor PID controller to ensure the stable brine forward temperature at chiller side.
   - All pumps are set at ΔP-c mode, in this case, the brine flow is rather stable.
   - Analyze the energy consumption of the system over 3 hours.

2. **Enable one D-pump to have the PID controlled function and adjust the cabinet temperature setting, in this case, the optimal setting point can be obtained.**
   - Cpump was set at ΔP-c mode to ensure the rather stable main brine flow.

3. **PID controlled compressor, PID controlled Dpumps and ΔP-c mode Cpump.**
   - Enable all Dpumps to have PID controlled function, all the temperature settings are the same as the optimal setting point which has been obtained from the previous test.
   - Analyze the energy consumption over 3 hours.

4. **PID controlled Dpumps and integrated PID controlled Cpump and compressor.**
   - Analyze the energy consumption over 3 hours.

In practice, the test started from step 3 directly, because the PID controllers can maintain the temperature quite stable and did not need further PID parameters adjustment.

4.1.1 Energy consumption of each equipment over 3 hours in each condition

In this refrigeration system, the main energy consumers are:

- the compressor (with frequency controller)
- the central pump
- the decentralized pumps
- fans and cabinets controllers

As can be seen from Figure 8, with the increasing of t_setting, the E_Dpumps was increasing a little bit, E_Cpump was almost constant, E_Comp was decreasing obviously and E_Fans&Controllers was constant. The total energy consumption was decreasing.

As shown in Figure 10, the increasing of the t_setting (brine out temperature at the evaporator side) results in the increasing of evaporation temperature (t_Eva). The increasing evaporation temperature provides a larger COP_c. This means that with the same amount of cooling load, the compressor consumes less energy. The higher brine out temperature makes lower brine density, which leads to higher brine flow (Flow_Bri) and thus the higher pump energy consumption. Higher brine flow makes the temperature difference of the brine in and out (dt_RF_Bri) smaller. But, one conclusion can be made; by increasing the t_setting, the energy saving from the increasing chiller COP is more than the increasing of the pump energy consumption. The total energy consumption will decrease (Figure 8).

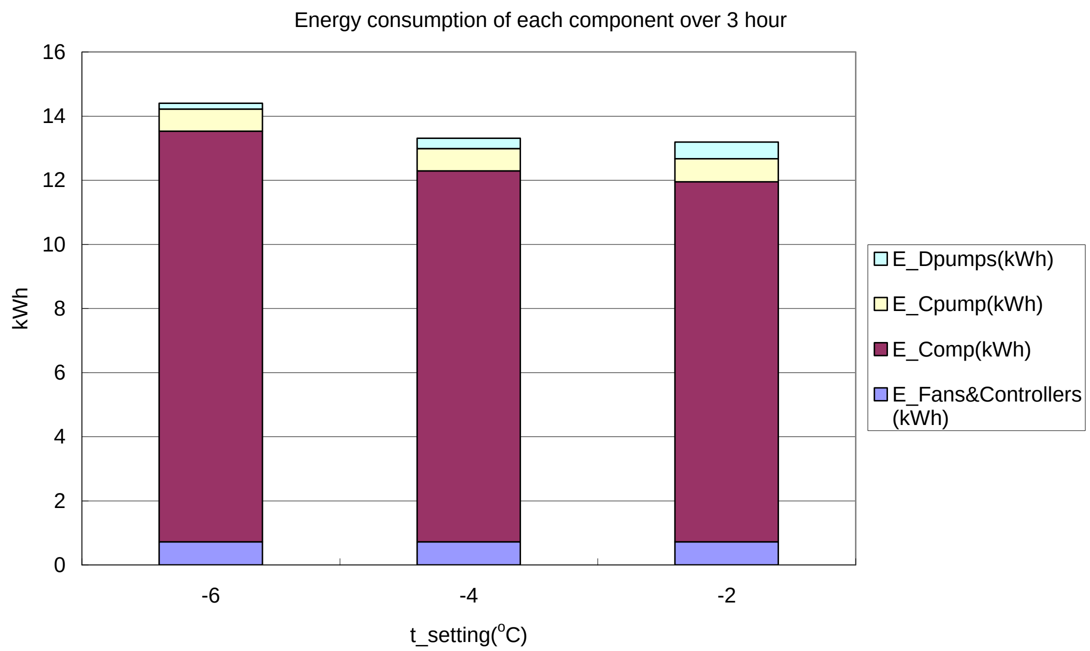

Figure 8 Energy consumption of each component over 3 hours.

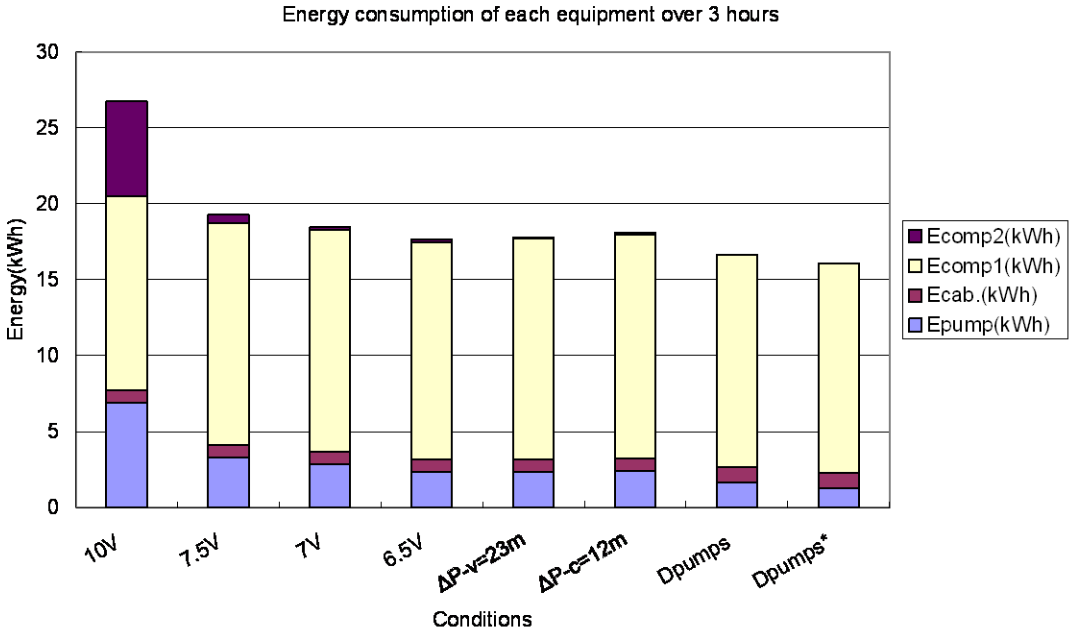

Figure 9 Energy consumption of each component over 3 hours (Phase 1).

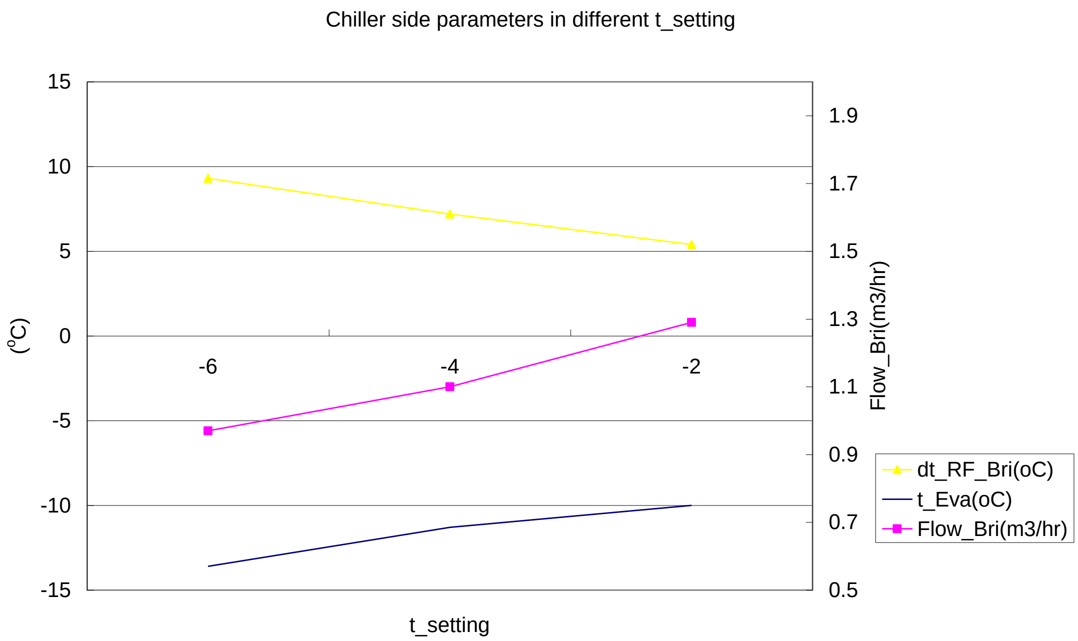

Figure 10 Chiller side parameters in different t_setting

4.1.2 Ratios between different parameters at different t_setting

Figure 11 shows that the ratio (E_Dpumps/E_Cpump) and (E_Cpump+E_Dpumps)/E_Comp are increasing as t_setting is increasing.

As the flow was increasing, owing to increasing t_setting, decentralized pumps are prone to increase the power more than the central pump, thus the increasing of E_Dpumps/E_Cpump.

Since t_setting is increasing, the brine flow and evaporation temperature will increase, which lead to an increase of (E_Cpump+E_Dpumps) and decrease of E_Comp. In this case, the (E_Cpump+E_Dpumps)/E_Comp will increase for sure.

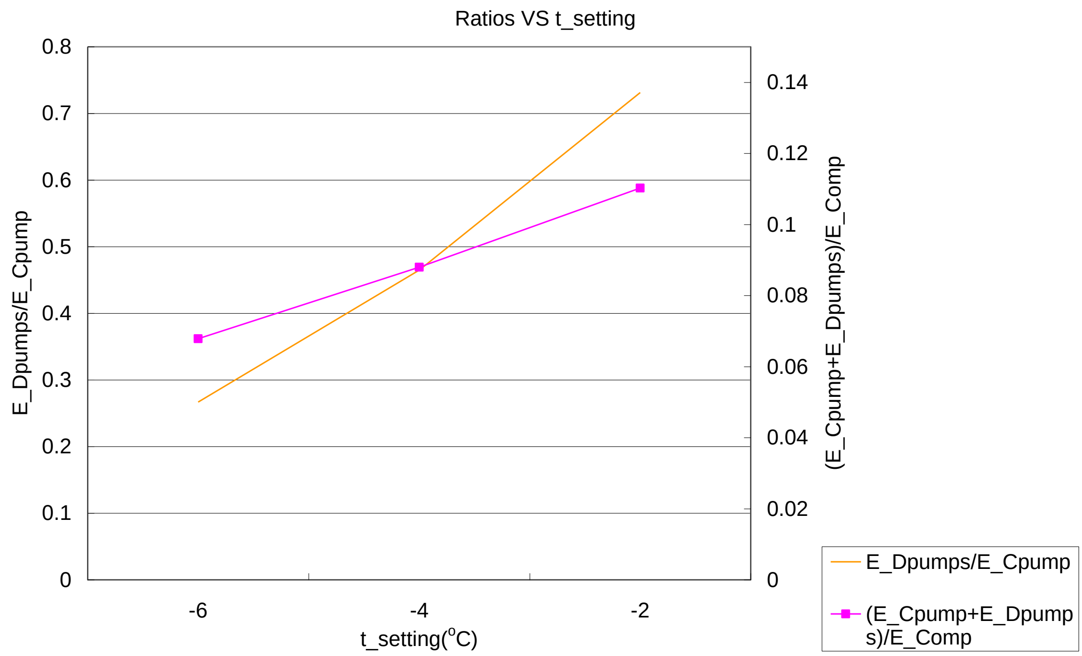

Figure 11 Equipment power ratios VS t_setting

4.2 Test 4: Pressure controlled central pump

The check valve is used to prevent the backward flow in the bypass line. In this case, the maximum brine flow can be ensured. The COP of the chiller may increase owing to the higher brine flow.

This test is included with 3 steps, each step is varying the central pump Δp-c setting between 10m, 8m and 6m, in each step it is included with 3 sub-steps which are varying the t_setting between -6 ^{o}C, -4 ^{o}C and -2 ^{o}C.

As can be seen from Figure 12, the first 3 solid bars represent the energy logged directly from the energy meters; the compressor energy (E_Comp), central pump energy (E_Cpump) and decentralized pump energy (E_Dpumps).
The other 4 pattern bars represent the calculated energy, which are cooling capacity calculated from the refrigerant side (Q2), cooling capacity calculated from the brine side (Q2_Bri), cabinet cooling load (Q_Cab) and controller heat loss (Q_Controller).

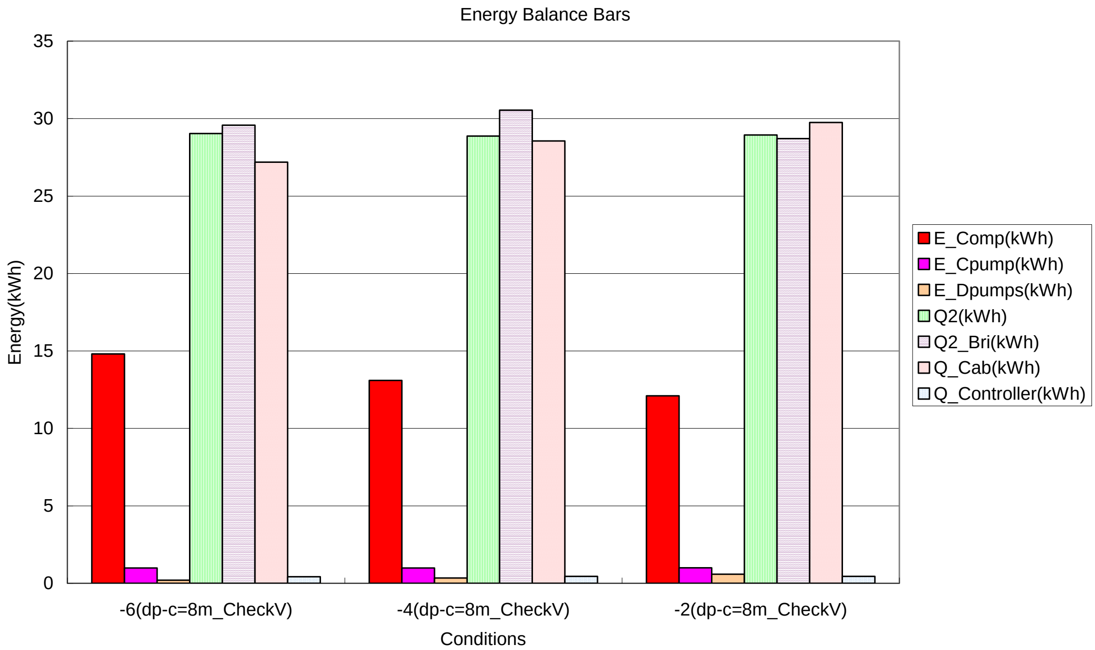

Figure 12 Energy balance bars at Δp-c=8m.

In the condition of Δp-c=10m, Q_Cab changed a lot in different t_setting, some unexpected factors might have influenced the test, so this step is not to be analyzed. In the condition of Δp-c=6m, when the t_setting was up to -2^{o}C, the food temperatures in one cabinet start to rise, which means the flow is not enough to meet the cooling load. Thus, this step is also eliminated. Only in the condition of Δp-c=8m, where all four calculated energy values kept relatively stable, the data could be used for analyzing.

When comparing Figure 12 and Figure 13, it can be seen that with the same Q_2 in each condition, the increasing of t_setting leads to the increasing of evaporation temperature (t_2). A increase of t2 will increase COP and thus the compressor energy consumption became less. On the other hand, the increasing of the t_setting resulted in the relative higher working temperature of brine, thus the flow will increase owing to the lower brine viscosity, and consequently, the increase of the decentralized pumps energy.

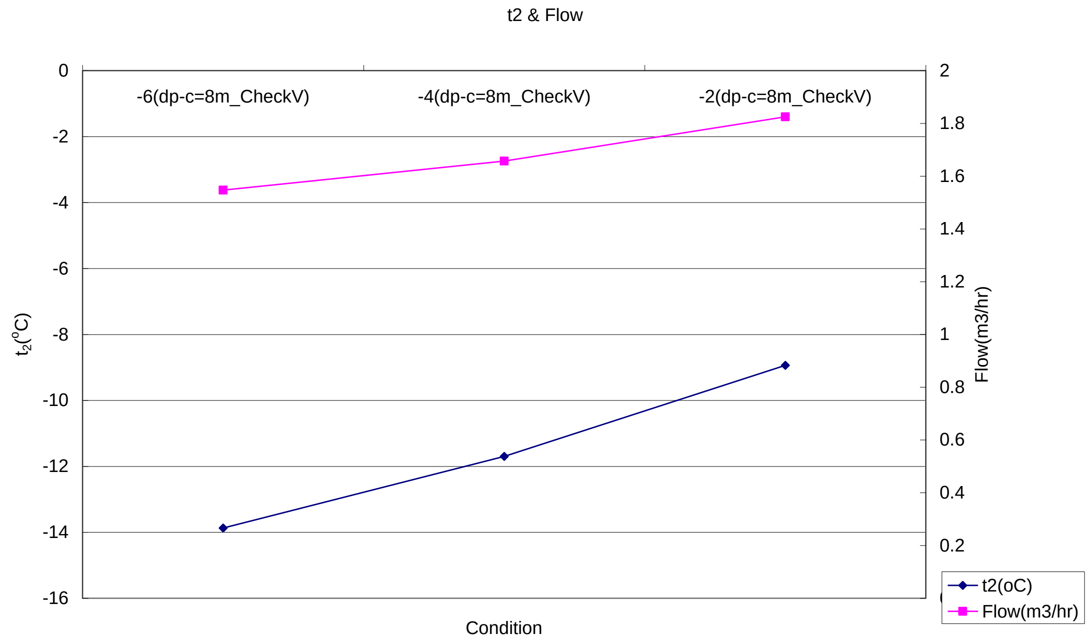

Figure 13 Main brine flow and t_2 in each condition.

The proportion of bypass flow over main flow is decreasing with the increasing of t_setting (brine forward temperature). The cabinet1 flow is increasing according to the main flow. The flow curves are shown in Figure 14 Three flows VS different t_setting. This phenomenon shows that the higher brine forward temperature makes it easier to pump the brine through each cabinet. Higher brine temperature gives lower brine viscosity, and thus the lower cabinet pressure resistance. In other words, the higher brine forward temperature enables good flow distribution in each cabinet and avoids the big proportion of bypass flow.

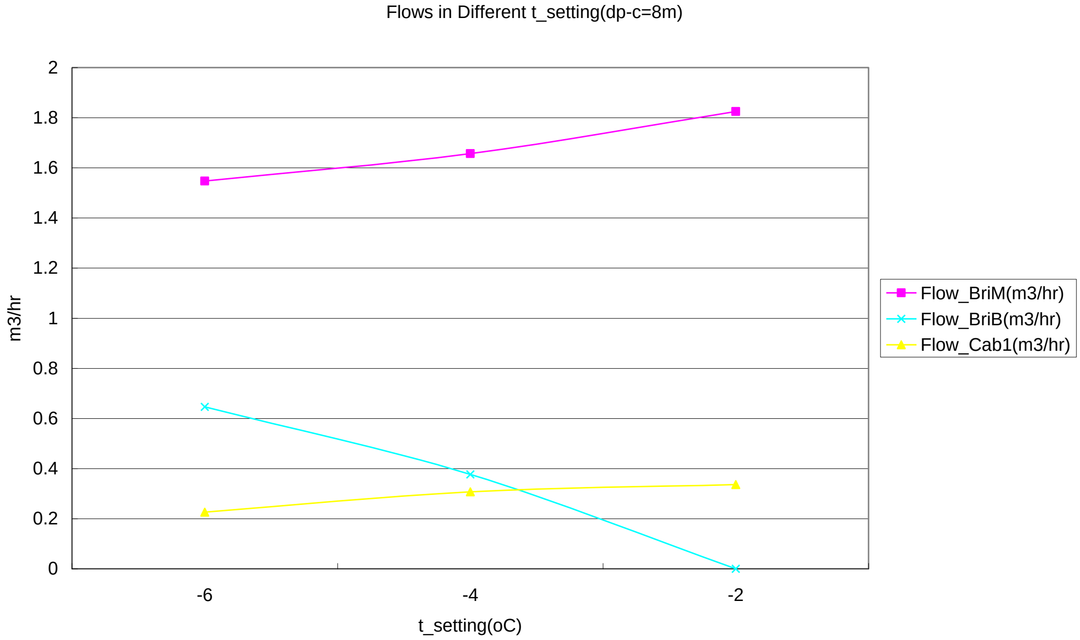

Figure 14 Three flows VS different t_setting

4.3 Test 5: Dynamical change of cabinet load

The purpose of this test was to evaluate the response of the cabinet control system. As mentioned previously the quick and accurate response of the cabinet is one advantage of the current concept.

A dynamical change of load was performed by putting a heater into one cabinet for a short period of time. The heater was then removed after a short period in order to bring the system back to normal operation again.

During the test, the heater was placed in front of cabinet 1. By changing the angel of the heater, the heat load could be changed. Figure 15 shows the heater facing outside of the cabinet in order to have a small heat input. Figure 16 shows the heater facing the inside of the cabinet, providing a large heat input. The influence of the heater for the system can be seen from the curves shown in Figure 17.

Figure 15 Dynamic heat load change. (low load)

Figure 16 Dynamic heat load change. (high load)

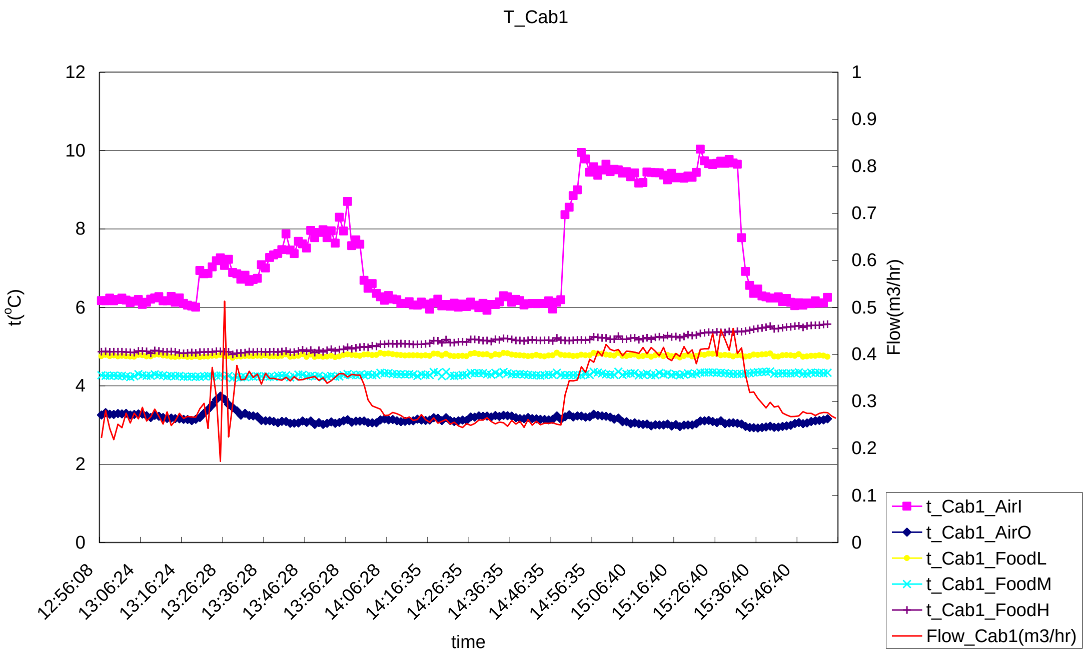

Figure 17 Temperatures and flow curves for Cabinet 1 (t_setting=80%, dp-c=8m).

In Figure 17, all the temperatures and brine flow logged for Cabinet 1:

- The air out temperature (t_Cab1_AirO) of the cabinet.
- The air in temperature (t_Cab1_AirI) of the cabinet.
- The food simulators which are located at; low (t_Cab1_FoodL), middle (t_Cab1_FoodM) and high (t_Cab1_FoodH) position.

Two "peaks" are obvious in the t_Cab1_AirI and Flow_Cab1 curve. The t_Cab1_AirI is quite sensitive to the heater because its thermocouple is mounted right after one of the cabinets fan. The brine flow, Flow_Cab1, is controlled by the decentralized pump which has a thermocouple mounted at the air-out side position of the cabinet heat exchanger (HX). The peak of the flow matches very well with the peak of the air-in temperature.

This proves that the PID controlled decentralized pump has a fast response time. Not only that, since the "air-out" is almost not affected it means the flow increase can absorb the full added heat load. It is a sign that this concept has a favourable effect on the cabinet controllability.

The temperature t_Cab1_FoodH was increasing gradually during this period while the other two food temperatures were stable. The brine temperatures (Figure 18) at the evaporator side were not influenced that much by the dynamic heat load change. Only a minor temperature rise can be noticed for the brine in temperature curve (t_RF_Bri_I), mainly at the right part.

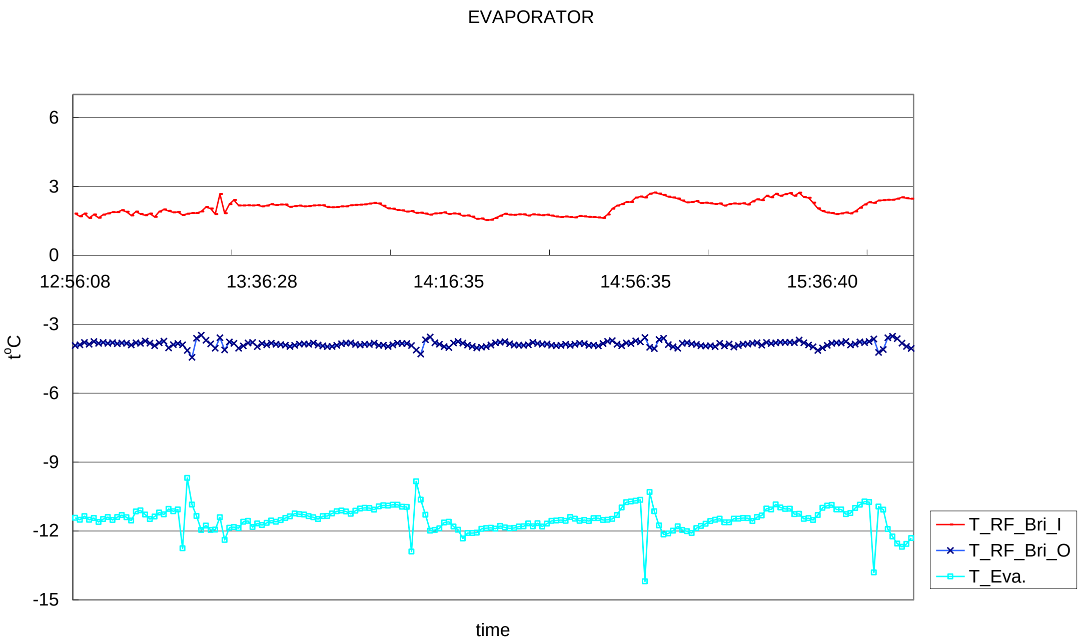

Figure 18 Temperatures at the evaporator side (t_setting=80%, dp-c=8m).

4.4 Test 6: Central pump and compressor integrated control

In this section the control of the central pump is integrated with the compressor control. In practice the compressor control signal is used in parallel to speed control the pump.

The ideal system would operate in the following way:
When the cooling load is low, both the compressor power and pump power should be decreased, due to less cooling capacity less flow is required.

In this test, the central pump was set at the external signal control mode in order to have it controlled by the compressor frequency controller. The pump speed is depending on the external signal value (working voltage ranges from 3-10V) obtained from the frequency controller.

During the operating time, the frequency controller will send the analogue signal (voltage) to the central pump simultaneously as it is controlling the compressor. In this case, the speed of central pump and compressor are changing at the same pace.
Figure 19 shows the connection of external signal wires with the pump. The batteries are used to ensure that the external signal is always above 3V. This will provide a minimum brine flow in the system even when the compressor stops.

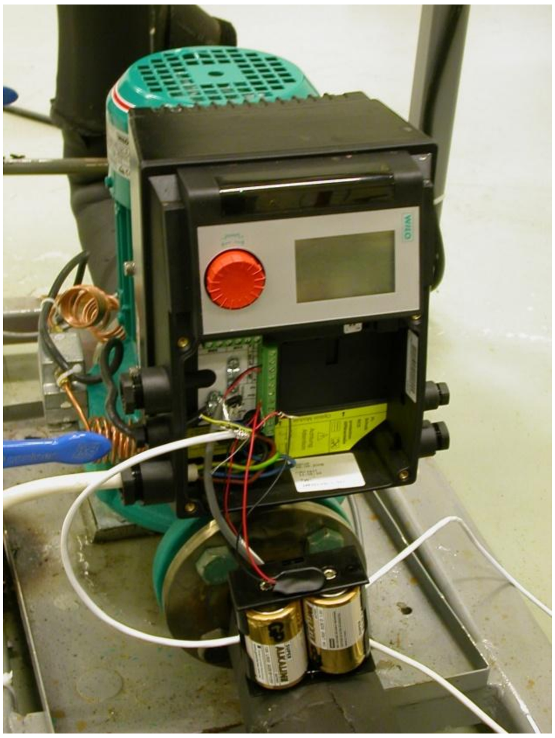

Figure 19 Pump is controlled by the external signal sent from the compressor frequency controller.

A change in speed of the central pump will lead to a different brine flow (Flow_Bri). At the same time, the brine out temperature (T_RF_Bri_O) at the evaporator side will also change. The change in brine flow and brine out temperature are normally in the same direction, either increasing or decreasing. For example, when the brine out temperature is decreasing, the frequency controller will adjust the compressor speed and the power will go down. At the same time, the central pump speed will also decrease; causing a reduced amount of flow through the evaporator. In this case, the decreasing brine out temperature will be accelerated.

As the brine flow changes, the brine out temperature will behave in the same manner. This will introduce oscillation curves regarding the brine temperatures (Figure 20). Since the compressor is controlled by the frequency controller in a good manner, the evaporation temperature can be maintained relatively stable.

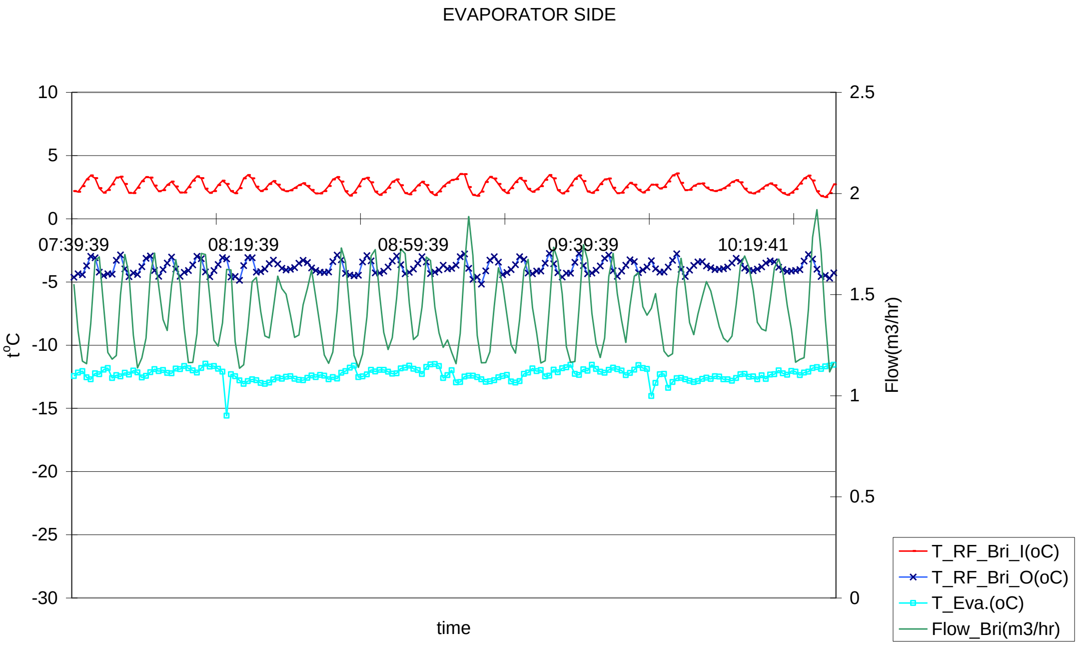

Figure 20 Main brine flow and the temperatures at the evaporator side.

Figure 21 shows all the equipment power in the system. P_Comp+PID stands for the power of the compressor and frequency controller. Q_Controller stands for the heat loss from the frequency controller. P_Cabs stands for all the power consumption from the cabinet side which include cabinet controllers, fans and decentralized pumps. P_Cpump stands for the central pump power.

One easily noticed phenomenon is that P_Comp+PID has the same changing trend as P_Cpump, owing to the integrated simultaneously control. The alternation of the P_Cabs curve is due to the resolution limitation by the energy meter. Curve Q_Controller performs quite stable, which means that the heat loss level from the frequency controller is not sensitive to the compressor power.

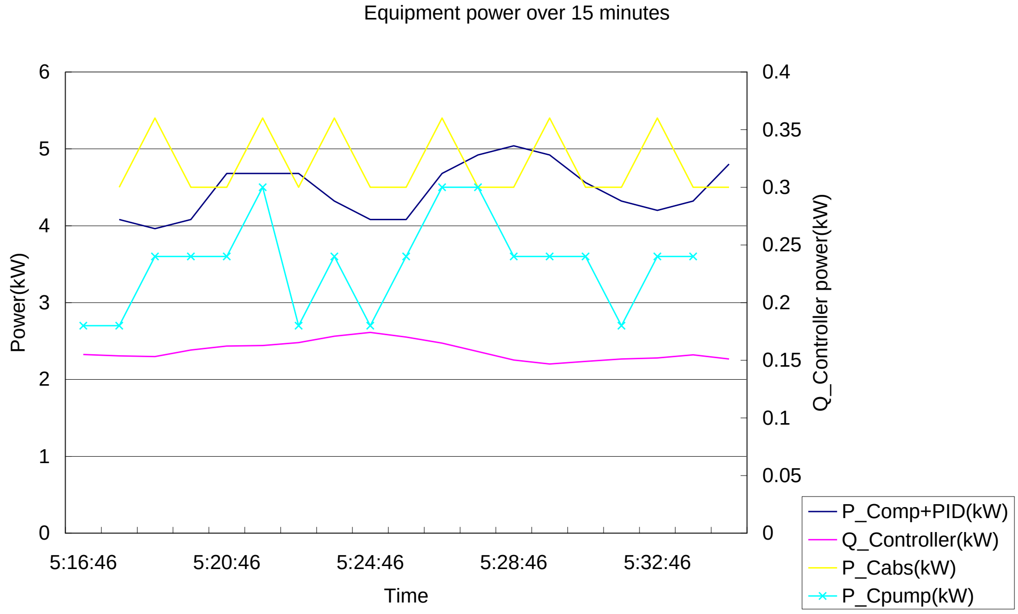

Figure 21 Equipment power consumption over 15 minutes

In order to compare energy utilization from the last test, a reference test result has been selected. As can be seen from Figure 22, the selected reference conditions are:

- Central pump set to Δp-c control mode (8m).
- Check valve is installed.
- T_setting = -4^{o}C (80%).

The reason why this particular case has been chosen has a reference, is due to the stable conditions. Q_2, Q_2_Bri and Q_Cab appear to be within the similar value, which proves that the system was operating at a relatively stable condition.

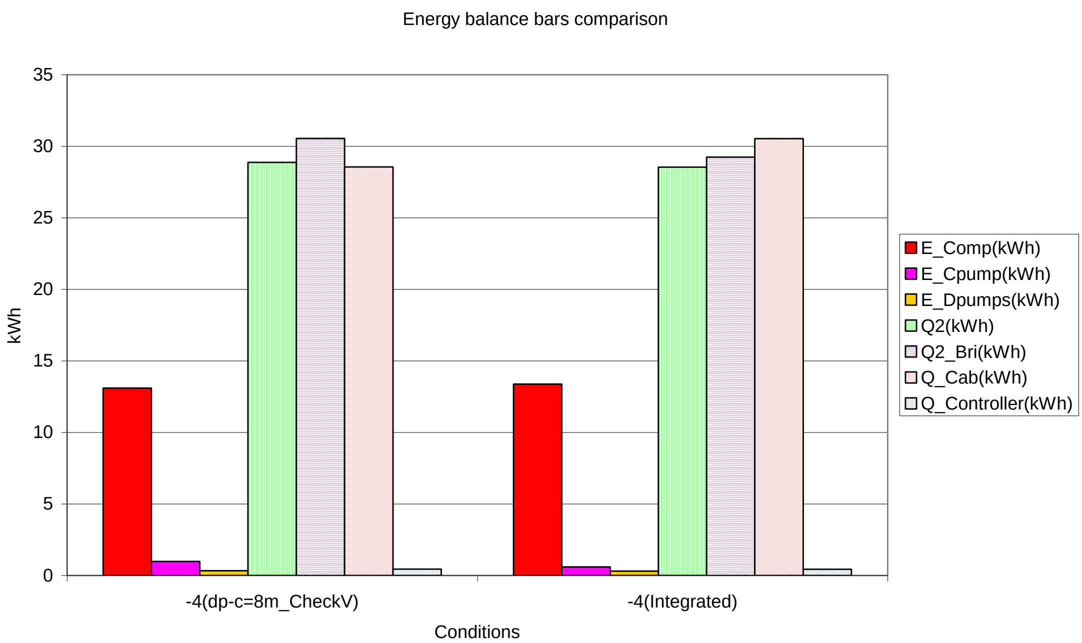

Figure 22 Energy balance comparison between condition -4^{o}C (dp-c=8m_CheckV) and -4 ^{o}C (integrated control).

At the first sight of these two groups of bars, they look quite similar. At a closer inspection a difference can be noticed. The compressor energy consumption is a little bit higher in the integrated condition, while the central pumps energy consumption is considerably lower.

The decentralized pumps energy consumption is a little bit higher in the integrated case which is difficult to notice from the Figure 22. The other four pattern bars provides information about the stability of the system and is not analyzed further.

4.5 Test 7: Exchange of brine from propylene- to ethylene glycol mixture.

In the previous tests, the brine has consisted of 50% propylene glycol (a mixture of 50 mass% propylene and 60 mass% water). The freezing point of mixture is -31°C and its density is 1208 kg/m3 at 25°C. The norm is 40%, however, it was discovered when replacing the brine that the concentration as higher than intended.

In this part of the test the propylene glycol has been exchanged and replaced by 35 % ethylene glycol. This mixture consists of 35 mass % ethylene and 65 mass % of water with a density of 1063 kg/m3 and a freezing point of - 33°C.

The system is operating in the same way as in Test 6 with an integrated control of compressor and central pump. The objective is to reduce the central pump's energy usage while the food substitutes are maintaining its temperature. The test has been performed in three different parts in which three values of t_settings has been tested (-6°C, -4°C and -2°C). As can be seen in Figure 23 the energy usage by the compressor is reduced as the t_setting is increased (higher evaporating temperature). The energy consumption by the central pump will also decrease since it is integrated with the compressor control. In order to maintain an acceptable food temperature the decentralized pumps have to operate even harder and consequently consumes more energy.

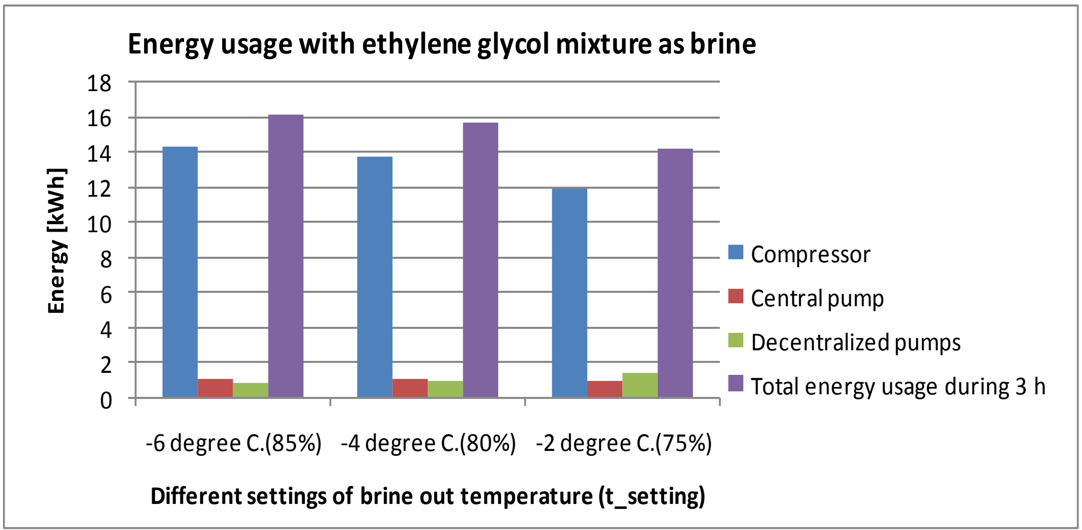

Figure 23 A comparison of energy consumption during three hours for different values of t_setting.

The Figure 24 below shows a comparison of energy consumption in the use of different brines and control systems. The Figure 24 is basically combination of Figure 22 and Figure 23 for the case of t_setting = -4°C. The compressor energy usage is a bit higher in the integrated control mode, which has already been conformed in Figure 22. Even though the compressor power is less in the pressure control mode of the central pump then the case with integrated system, the energy utilization by the central pump remains quite constant.

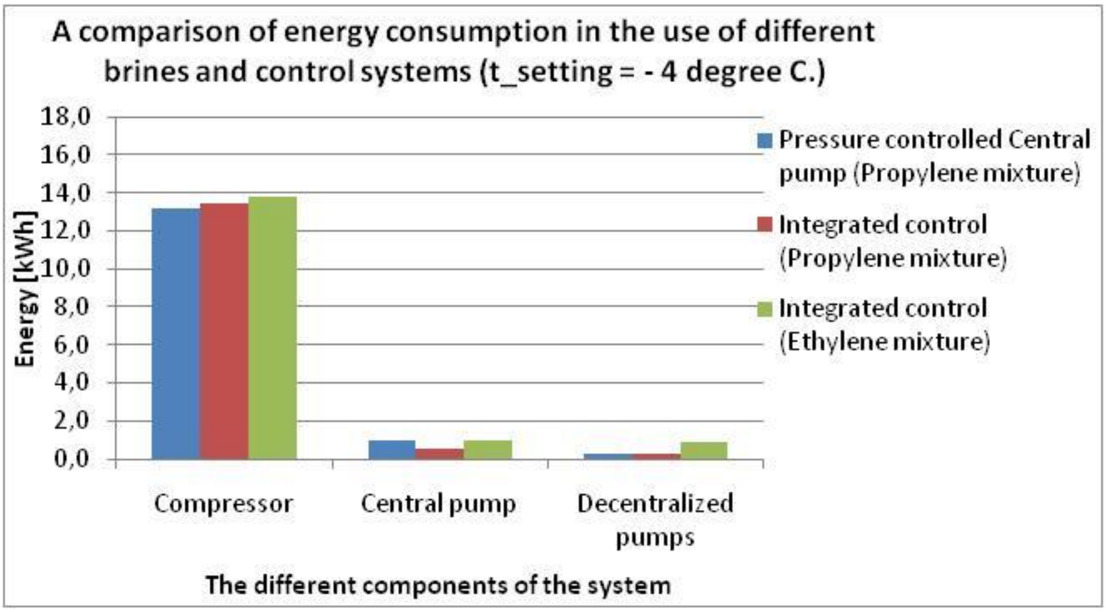

Figure 24 A comparison of energy consumption during three hours for different brines and control modes.

4.6 Over all results discussion

To conclude this report it can be said that the respective development phase has shown interesting results with respect to the decentralised pumps in indirect refrigeration systems.

In phase 1 a traditional system solution with a central pump was compared with the first generation decentralised pump system. The conclusion was an energy saving of 6% on a system level and 17% of the pump distribution energy. These measurements were based on 24 hrs testing and included a defrost cycle.

	Performance of the system during test period (3 h)		
	Reference test -- Phase 1 --	The most energy efficient system -- Phase 1 --	The most energy efficient system -- Phase 2 --
Cooling capacity, Q2 [kW]	11.0	10.7	9.51
Evaporation temp. [degree C.]	-12.7	-12.9	-12.32
Condensation temp. [degree C.]	37.7	37.7	37.30
Brine forward temp. [degree C.]	-3.7	-3.5	-3.84
Humidity [%]	30.2	31.3	24.60
Room temperature [degree C.]	20.4	20.0	19.90
Compressors	14.9	14.5	13.31
Central pump	2.3	1.7	0.59
Solenoid valves	0.9	-	-
Decentralized pumps	-	1.0	1.02
Total energy usage	18.0	17.2	14.92
Energy savings compared to reference test [%]:	-	**4.4%**	**17.0%**

Figure 25. Compilation of the results from the different development steps

I figure 25 the key results from three significant steps of the development is compared. These results are based on the second phase method with 3 hours stable operation. For this reason the results may look slightly different compared to the previously shown but the comparability over the whole project is improved.

Further, the table in figure 25 gives an overview of the basic parameters in the tests. The stated cooling capacity is based on the refrigeration machinery and the pump energy generated will therefore be included in this overall cooling capacity.
The most striking difference is the reduction in the central pump energy demand. Between the two tests in phase 1 there are two major differences; the control strategy and the decentralised pumps. The latter will take a part of the work from the central pump but mainly the adaption to the required flow is the main driver for the reduction in energy usage.

In phase 2 the adaptation of the secondary refrigerant piping improves the situation, but the main driver is the coordinated compressor and pump control. Due to the pump work which is adapted to the cooling capacity the pump energy is significantly reduced.

Overall a 17% energy reduction is achieved on a system level and the distribution part of it ($E_{Brine\ pump}/E_{comp}$) for the cold side is reduced from 21% to 18.6% in phase 1. In phase 2 this was further reduced to 12% and in relation to the refrigeration energy the corresponding figures are 9.7%, 8.4% och 5.6%.

The interesting part here is rather the relative improvement than the absolute level of the distribution energy, since the system design highly influences the pump work. The present system is of an ultimate design and therefore the pump work is relatively high.

5 CONCLUSIONS

The project test targets have been fulfilled as planned with the integrated control of compressor and the central pumps.

The achievement of frequency controlled compressor and PID controlled decentralised pumps requires a good knowledge of automation and instrumentation. The setting of controller's parameters determines how the equipment can behave in terms of system stability and energy consumption. The start-up time is dependent on the complexity of the system.

The brine out temperature setting at the evaporator side (t_setting) is important for the system energy consumption. Generally a higher t_setting is equivalent to lower system energy consumption. The reason is that a high t_setting will increase the temperature of the evaporation and less power to the compressor is needed.
Even though the brine flow increases due to the higher brine temperature (which results in higher pump power) the compressor power saving will always be dominating vs. the increase in the pump power.

The frequency controlled compressor is a good energy solution and the system is able to adapt the cooling capacity to the cooling load efficiently.

Some specific conclusions regarding to the conducted tests:

1. The integrated control of the compressor and the central pump (along with the PID regulated decentralized pumps) has improved the system even further in an energy point of view. By comparing the results from phase 1 the system is improved by 13%.

2. The compressor consumes the major part of the energy, which mainly depends on the evaporation temperature and thereby the evaporation pressure. Therefore the pump function must ensure highest possible evaporation temperature i.e. it must not render in a penalised heat transfer which lowers the evaporation temperature.

3. The most efficiency point is an important parameter for the central pump. Thus, instead of having one large central pump with speed control, two smaller central pumps can be mounted in parallel with the step control method to fulfill the task. The controlling method can be operating in the following way: when there is one or two cabinets operating, only one pump is on. If there are three or four cabinets operating, both of the pumps start to operate. In this case, each pump can work around its best efficiency point to a larger extent.

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Decentralised Pumps in Supermarket Refrigeration - Phase 2 -

Brine system(PID controlled system)

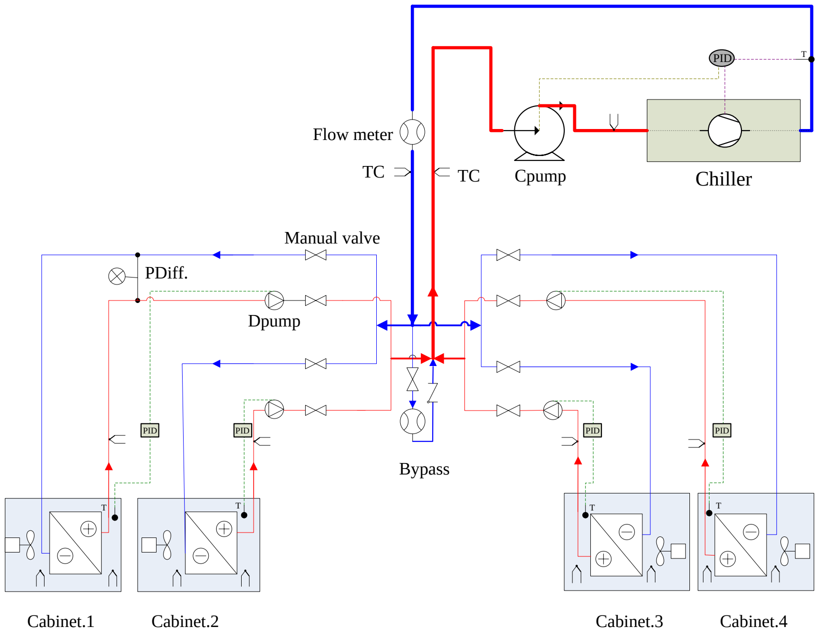

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TABLE OF CONTENTS

ABSTRACT

This project reports results from the second phase of a study on decentralised pumps used in refrigeration system application. The project has mainly been financed by KYS (Kylbranschens Samarbetsstiftelse).

The project has shown interesting results with respect to the decentralised pumps in indirect refrigeration systems. In phase 1 a traditional system solution with a central pump was compared with the first generation decentralised pump system. The conclusion was an energy saving of 6% on a system level and 17% of the pump distribution energy.

The improvement in phase 1 consisted of two major reasons; the control strategy and the decentralised pumps. The latter will take a part of the work from the central pump but the adaption to the required flow is the main driver for the reduction in energy usage.

In phase 2 an adaptation of the secondary refrigerant piping reduced the flow resistance, however, the main driver was the coordinated compressor and pump control. Due to the pump work which is adapted to the cooling capacity the pump energy is significantly reduced.

Overall in the project a 17% energy reduction was achieved on a system level. The cold side distribution energy was reduced from 21% to 18.6% in phase 1. In phase 2 this was further reduced to 12% and in relation to the refrigeration energy the corresponding figures are 9.7%, 8.4% och 5.6%.

The interesting part is the relative improvement rather the than the absolute level of the distribution energy, since the system design highly influences the pump work. The present system is of an ultimate design and therefore the pump work is relatively high.

The increased cost of the present concept is expected to be approximately 100 Euro per display case. This will have to be trade off against the energy saving, the reduced assembly and adjustment time and the improved temperature control.

PREFACE

The authors direct their gratitude to the main financers of phase 2, Kylbranschens Samarbetsstiftelse (KYS) and Effsys 2. Phase 1 was financed by Effsys 2 (the Swedish Energy Agency (Energimyndigheten)).

A special thanks to the contributing companies Wilo AB, Wica Cold AB and Cupori for their support and contributions.

1 INTRODUCTION

With the rising awareness of the global energy situation, the efficiency of all energy systems gradually comes in focus. This applies to all the energy related fields which range from huge country electricity transmission system to tiny computer chips. Supermarket refrigeration system is one of the energy systems of interest.

In terms of electricity consumption, supermarkets in Sweden account for 3% of total amount of 1.8TWh (Sjöberg, 1997) in the national level. The refrigeration energy consumption accounts for around a half of this. There is a huge potential to save the energy in the supermarket refrigeration field.

In the previous project phase 1 (Jan 2009 to June 2009) a saving in pump energy of 17% by introducing decentralized pumps was realised. On a system level the saving was between 7 and 9% depending on the reference.
To further improve the concept a second project phase was performed during the time from October 2009 to March 2010.

1.1 Background

Pump is used to provide the flow in the distribution loop. The normal condition is with a fixed speed uncontrolled pump in the main loop no matter how the flow changes. Since each branch is controlled by on-off solenoid valve, the pressure difference in the main loop is changing considerably. Thus, some better control methods based on flow variation are coming up. The decentralized pump system is one of those leading methods.

The success of decentralized pump usage in heating system has been proven. The pump manufacturer Wilo did field tests years ago and came up with the conclusion that the decentralized pump system could decrease the energy consumption by up to 20%.
Despite the advantage of energy saving, there are three other advantages:

1. It is more convenient for the customer because the quick heating process and the better room temperature stability can be achieved;
2. A pleasant automatic hydraulic balancing is achieved, which leads to the easier job for the craftsmen in terms of the system installation and flow adjustment.
3. It provides the better feasibility for the building central control in terms of the data diagnosis and energy management.

To investigate whether if the decentralized pump system can have the same advantages on the cooling usage, this project was carried out.

1.1 Aim of the project

The aim of the project is to:

- Show the decentralized pumps influence on the indirect refrigeration system overall performance
- Optimise a test system as reference
- Show practical gains with a decentralized pump arrangement on the cold brine side
- Deliver practical results from the test system
- Show a cost estimate of the system solution
- Verify that the cold side distribution energy can be reduced to maximum 4% of the total refrigeration system energy input
- Demonstrate that a typical supermarket system can save 10 MWh per year

2 PHASE 2 SYSTEM IMPROVEMENT

This chapter describes the main changes in the system compared to phase 1. Inherent weaknesses in the system discovered in the first phase were focused on, such as adoptions allowing evaluating new ideas.

2.1 Reconstruction and modification of the brine system

In response to previous test results the brine system has been upgraded in order to optimize it even further. The new layout is obtained from Figure 2-1. In comparison with the old system, the new brine system consists of larger brine tubes at the chiller side. The purpose of the larger tubes is to decrease the pressure loss when the brine is operating at low temperatures and consequently at high viscosity. It has been proven from the previous tests that one chiller is enough to cover the cooling load. Thus, the back up chiller has been disconnected to avoid the electricity consumption during its standby time.

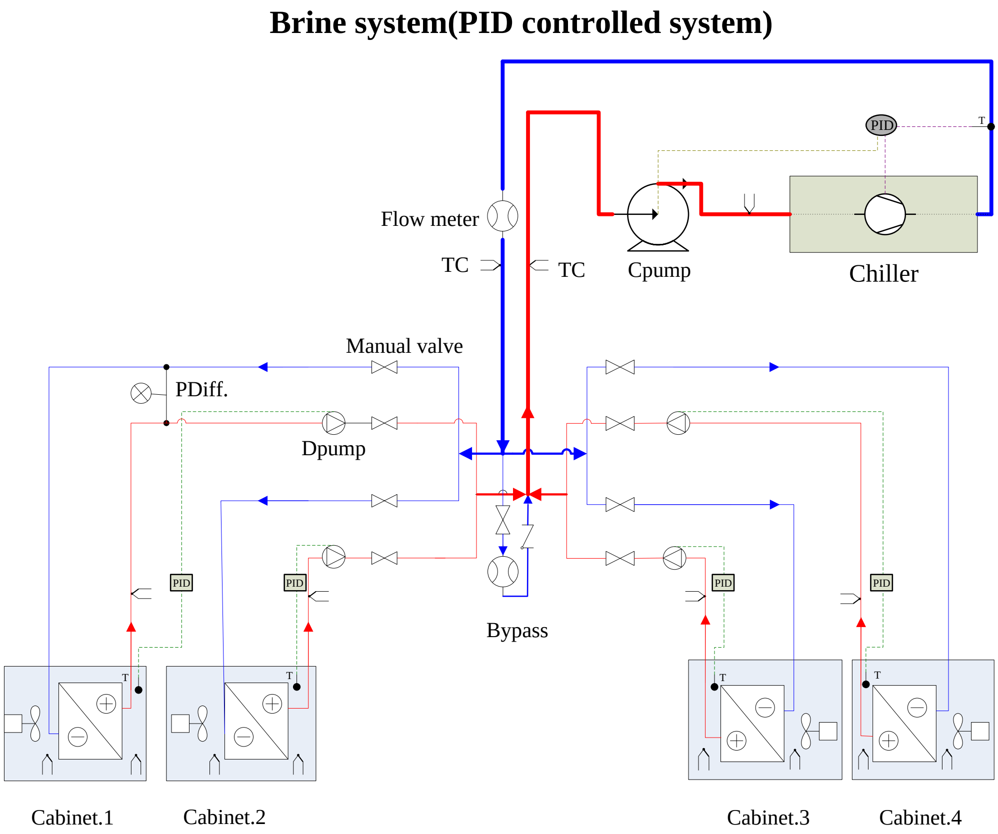

Figure 2-1. The new and improved brine system.

The construction of the brine system forced the old central pump to operate at low region of the pump property flow chart. The operating point had a negative impact on the efficiency of the pump. By replacing the larger pump by a smaller sized pump (as can be obtained from Figure 2-2) this could be solved. The main flow could now be maintained at a central region of the pump property curve and thereby improving the pump efficiency.

Figure 2-2 The new and smaller sized central pump.

The position of the bypass function is the last modification of the old system. Previously, the bypass function has been located at one branch of the distribution system, which has proven to be a poor design. In the new setup, the bypass is located at the main branch and associated with a flow meter in order to supervise the bypass brine flow.

2.2 System control

2.2.1 PID controller of compressor

The frequency controlled compressor is regulated by the brine out temperature at the chiller side. A T-type thermocouple is installed at the brine out spot and the signal is transferred to the PID frequency controller (Figure 2-3) through a converter (Figure 2-4). The operational range of the converter is between -15°C to 30°C and with the current output ranging from 0 to 20 mA. A linear function between the temperature and ampere can be obtained to correlate these two parameters.

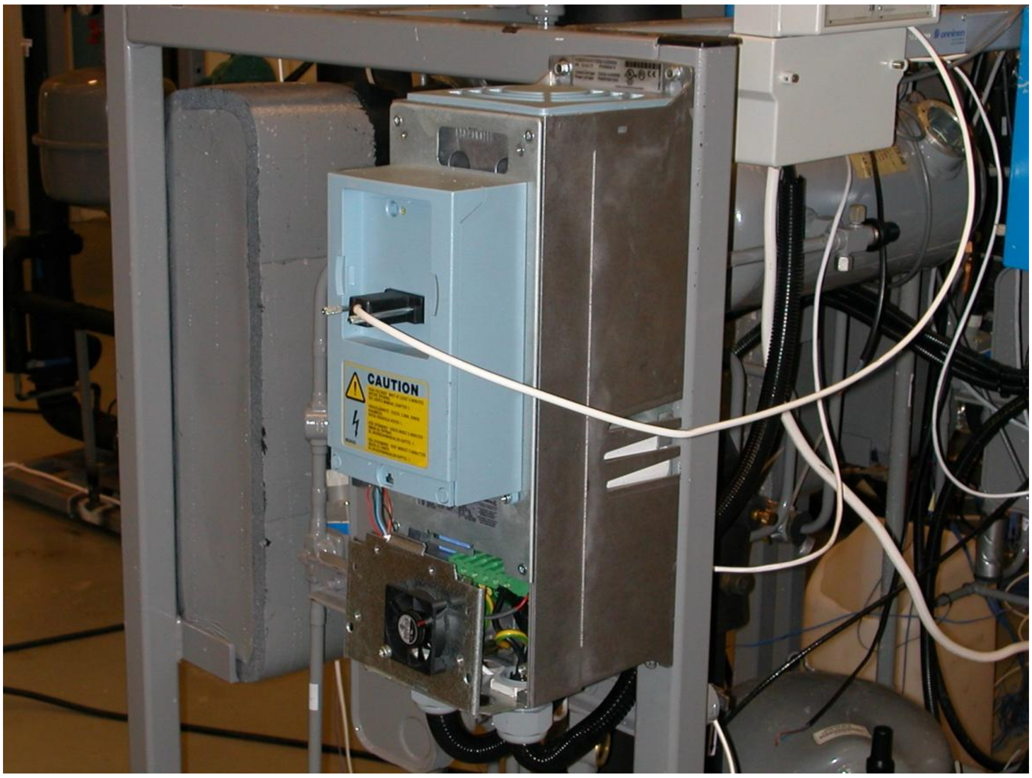

Figure 2-3 Frequency controller for the compressor control

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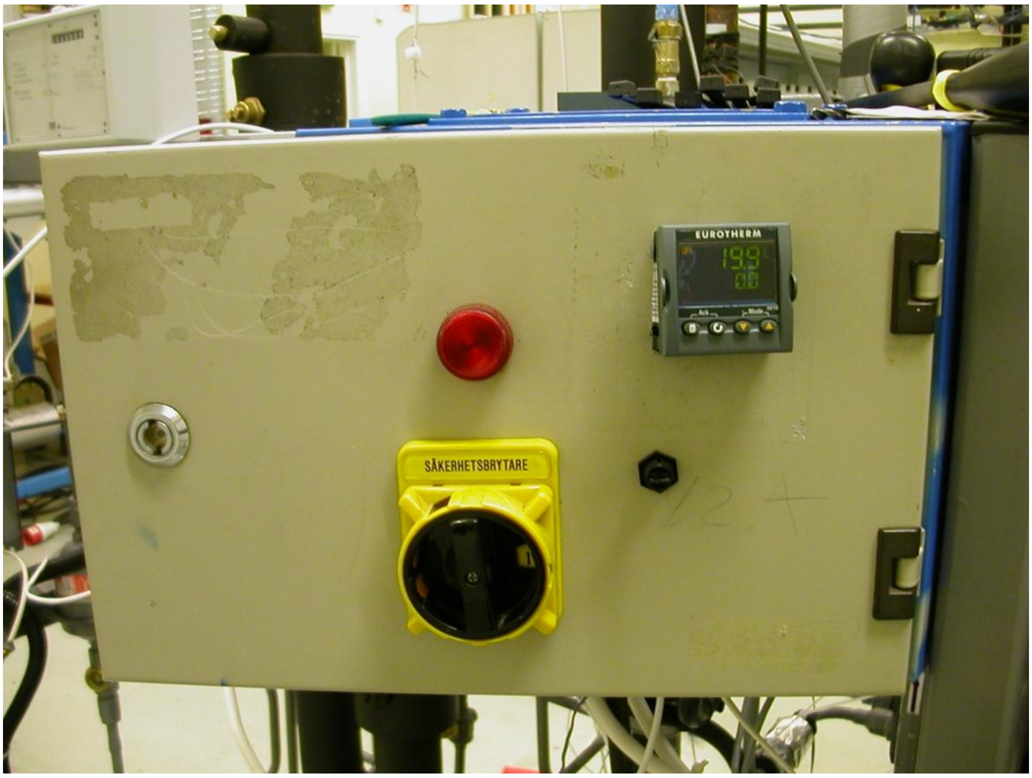

Figure 2-4 Temperature converter for the compressor control.

2.2.2 PID control of decentralized pumps

Each decentralized pump is regulated by a PID controller (Figure 2-5). The input signals are obtained from the air-out temperature of the cabinet and the set point of the controllers is adjusted to 2°C.

Figure 2-5 PID controllers of decentralized pumps.

3 TEST EQUIPMENT - SYSTEM TEST AND TUNING

In the following section the verification of the test system including tuning is presented. The aim is to give a picture of the test environment and repeatability of the tests.

3.1 Test system – ambient conditions

In order to compare the results between the various tests, the surrounding conditions must remain stable and fixed during the entire period.

The basic requirements are:

1. The food temperatures are controlled between 3 and 6^{o}C (Figure 3-1).
2. Room temperature is controlled around 20 ^{o}C (Figure 3-2).
3. Room humidity is controlled at around 36% (Figure 3-1).
   (It shows temporarily a 10% lower value in the figure below)

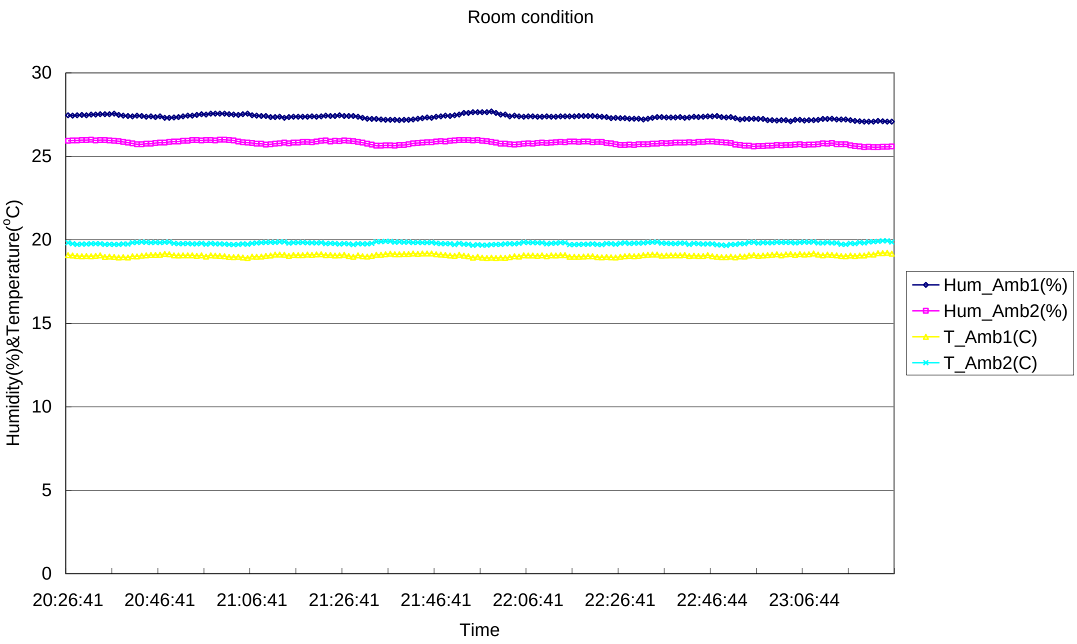

Figure 3-1 Controlled room conditions with temperature and humidity.

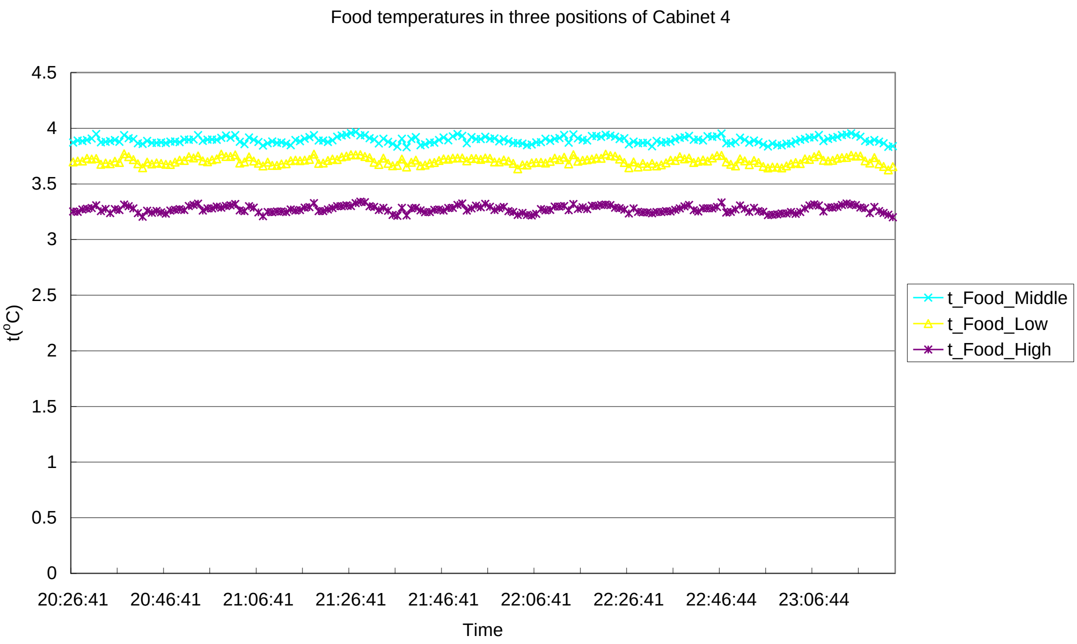

Figure 3-2 Controlled food simulator temperatures in one cabinet.

3.2 Test 1: Bypass open without check valve.

At test number one, the decentralized pumps were running at the PID-mode (set-point = 2^{o}C) and the central pump was set to ΔP-c=10m. The compressor control parameter, t_setting, was selected to -4^{o}C. At the established operating condition, the required outlet air temperature of 2^{o}C at the cabinets could not be fulfilled even thus all of the decentralized pumps were operating at the full load.

The phenomenon could be explained by backflow in the brine circuit. The backflow short cut the brine loop so that the cabinets cannot get the low temperature brine flow from the chiller. No matter how hard the decentralized pump worked, they are still pumping the old cabinets flow over and over, only with a little mixture of the fresh brine flow from the chiller.

Thus, a check valve is necessary in the bypass to avoid the backflow.

3.3 Test 2: Bypass closed and changed t_setting of compressor controller.

During this test, t_setting was reduced gradually from a start value of -2^{o}C down to -6^{o}C, with a temperature interval of 1^{o}C. The bypass backflow was totally avoided in this test by using a check valve. Although the backflow had been solved a new problem occurred.

From the logger, it was clear that with decreasing t_setting, the food temperature in each cabinet was also decreasing. At some moment, the decentralized pump stopped operating occurring to the PID regulation. Even thou the decentralized pumps were in off-mode the cabinet outlet air temperature were still getting below 2^{o}C. Both above two phenomenon proved that the central pump was overriding the decentralized pumps which made the cabinet temperatures loss control.

Thus, the obtained test data were not valid for analyzing the system.

3.4 Test 3: Decreased ΔP-pump of central pump.

In order to not override the decentralized pumps the condition of the Central pump was set to ΔP-c=6m. The t_setting was changed from -6 ^{o}C up to -2 ^{o}C with the interval of 2^{o}C, in other words, three steps were conducted in this test. During each step, the “air out temperature” of each cabinet could be kept around the set value (2^{o}C). Since the boundary conditions could be kept rather constant during each step of t_setting, the data was valid to be analyzed.

3.4.1 Central pump parameters curve in different t_setting.

T_setting is the reference value to control the compressor speed. The input signal comes from the brine out temperature at the evaporator side. During this test, the central pump was always working at ΔP-c=6m mode while the t_setting of compressor was changing at 3 steps, which were -6(Figure 1), -4 (Figure 2) and -2°C (Figure 3) respectively.

As can be seen from the figures, with increasing t_setting, the Flow_BriM and Eff_pump will increase accordingly. Pdiff_Cpump is kept constant as it is set to “constant pressure control”-mode.

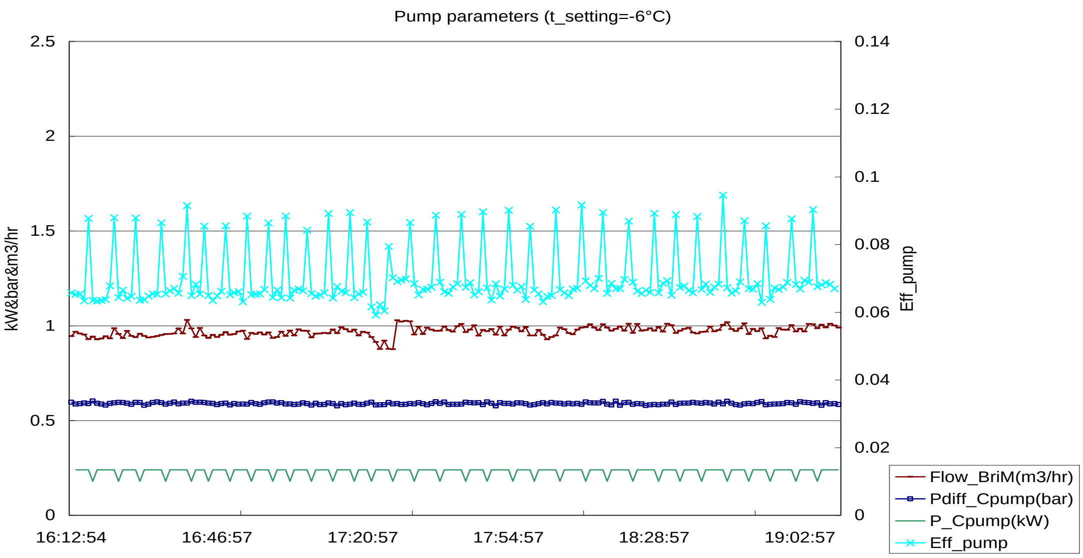

Figure 1 Pump parameters at t_setting=-6^{o}C

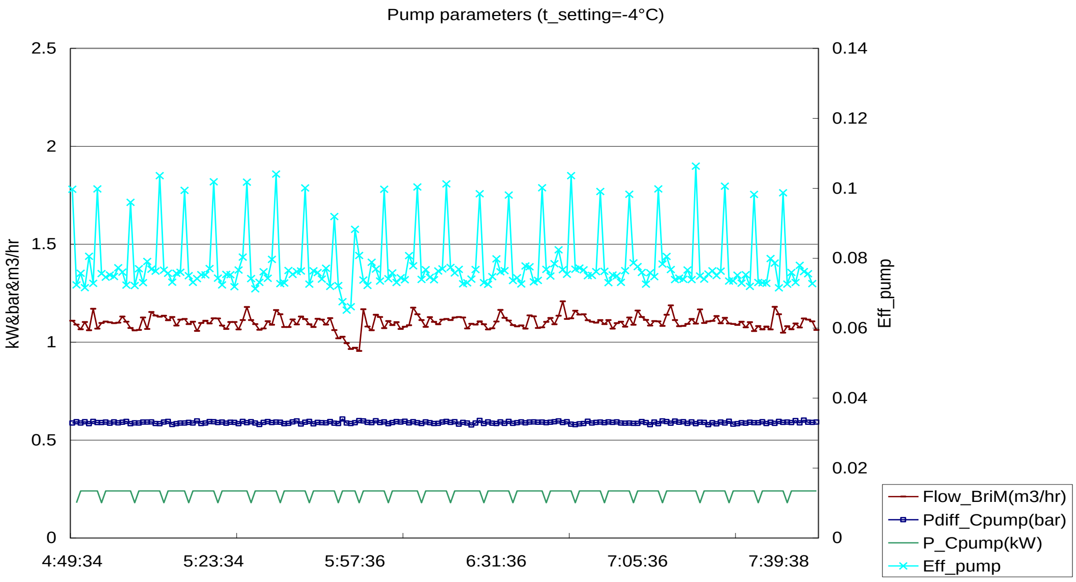

Figure 2 Pump parameters at t_setting=-4^{o}C

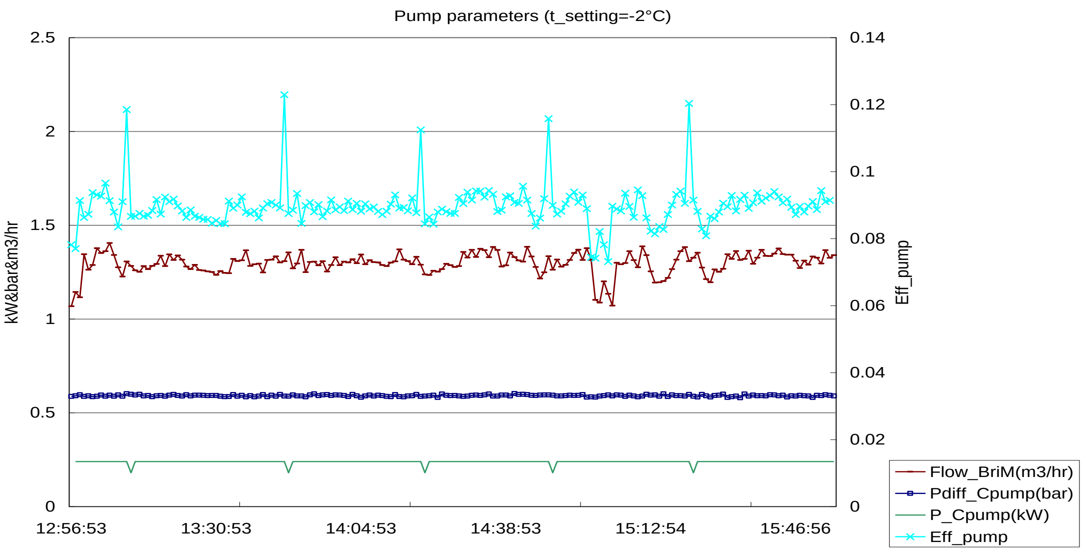

Figure 3 Pump parameters at t_setting=-2^{o}C

3.4.2 Q_Controller VS P_Comp over 3 hours in each condition

The aim of this section is to evaluate the frequency controller losses. It is not only of general interest but also specifically important to make the correct assumptions for the compressor energy balance.

Q_Controller stands for the energy loss from the frequency controller, which is calculated with the equation:

$$\text{Q\_Controller}=\text{AirFlow}*\rho_{\text{air}}*C_{\text{p\_air}}*\Delta t \qquad \text{Equation 1}$$

Where AirFlow (m^3/s) stands for the air volume flow pass through the frequency controller, ρ_{air}(m^3/kg) is the air density, Cp_air (KJ/(kg*K)) is the specific heat capacity of air, Δt is the temperature difference of air in and air out of the frequency controller.
The air in temperature is the room temperature and the air out temperature is measured by a thermocouple (Figure 4) at the air outlet duct of frequency controller.

Figure 4 The frequency controller mounted with the thermocouple at the air outlet.

P_Comp actually consists of the power of both compressor and frequency controller (Q_Controller), since the power supply is connected to the same electricity socket, only one energy meter is used.

Figure 12, Figure 13 and Figure 14 show the Q_Controller value varies from 0.12 to 0.18 kW and the proportion over P_Comp varies from 2 to 7%. The value of Q_Controller was not significantly influenced by the changing of P_Comp, so it didn’t appear a trend when P_Comp was increasing. In this case, the proportion over P_Comp will surely decrease when the P_Cump was increasing.

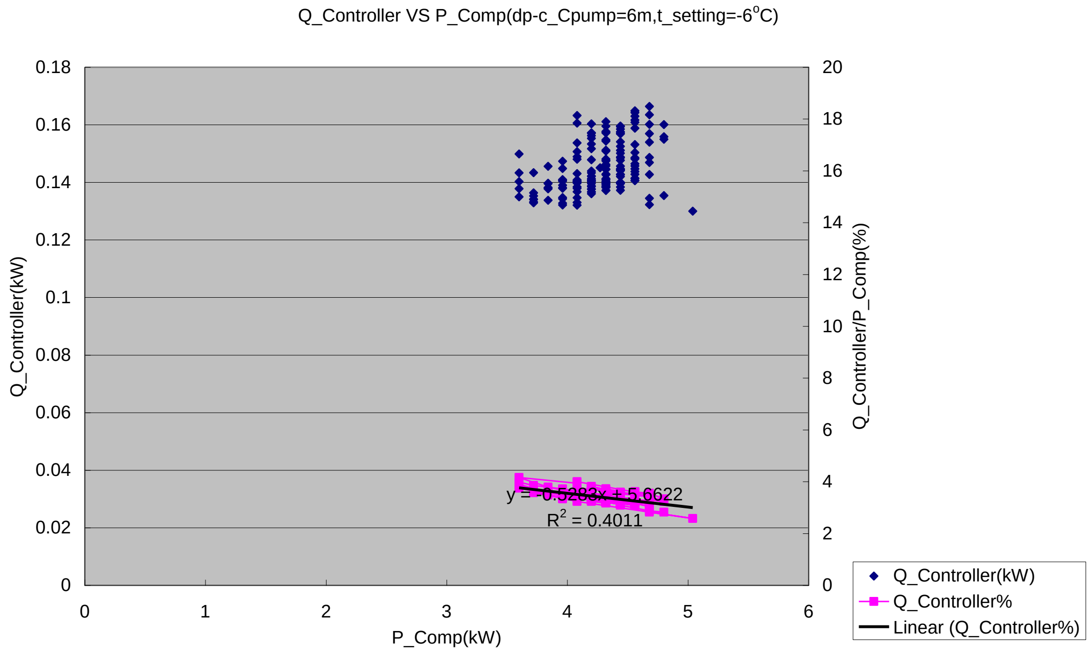

Figure 5 Q_Controller's real value and proportion VS P_Comp when t_setting=-6^{o}C.

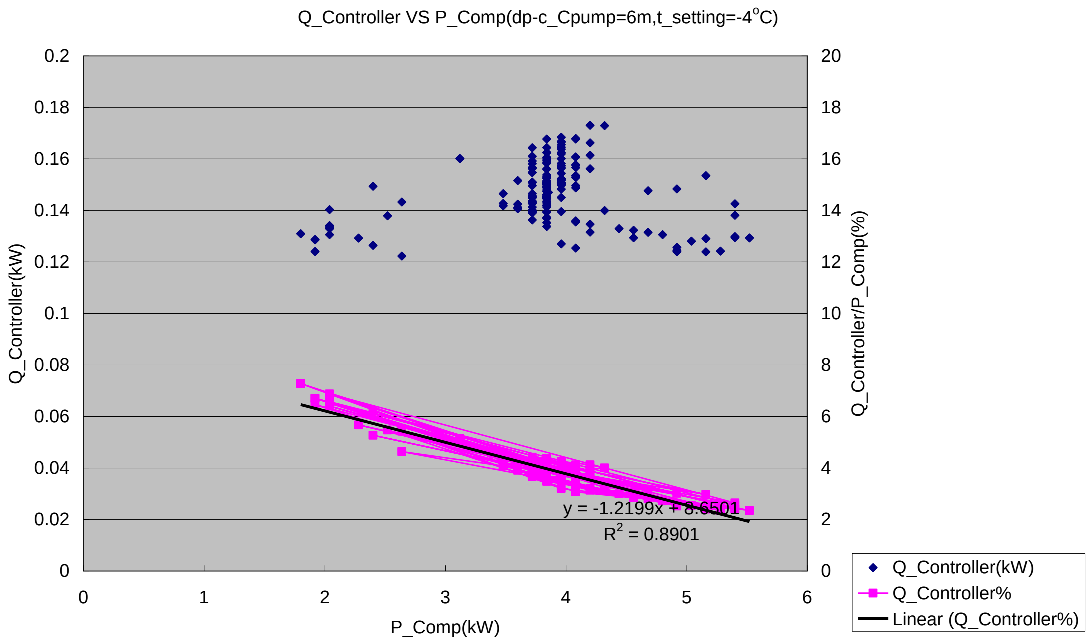

Figure 6 Q_Controller's real value and proportion VS P_Comp when t_setting=-4^{o}C.

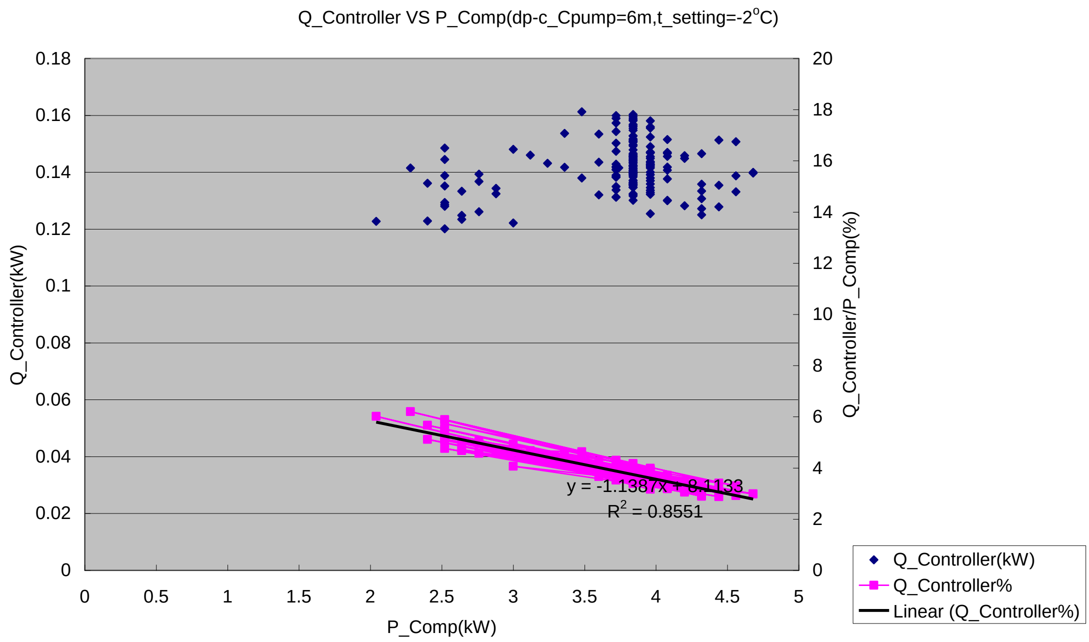

Figure 7 Q_Controller's real value and proportion VS P_Comp when t_setting=-2°C.

4 TEST RESULTS

This chapter first presents the test procedure followed by the results obtained for the respective test section.

4.1 Test procedure

The test steps were followed as below:

1. **PID controlled compressor, ΔP-c mode Cpump and Dpumps:**
   - Adjusted compressor PID controller to ensure the stable brine forward temperature at chiller side.
   - All pumps are set at ΔP-c mode, in this case, the brine flow is rather stable.
   - Analyze the energy consumption of the system over 3 hours.

2. **Enable one D-pump to have the PID controlled function and adjust the cabinet temperature setting, in this case, the optimal setting point can be obtained.**
   - Cpump was set at ΔP-c mode to ensure the rather stable main brine flow.

3. **PID controlled compressor, PID controlled Dpumps and ΔP-c mode Cpump.**
   - Enable all Dpumps to have PID controlled function, all the temperature settings are the same as the optimal setting point which has been obtained from the previous test.
   - Analyze the energy consumption over 3 hours.

4. **PID controlled Dpumps and integrated PID controlled Cpump and compressor.**
   - Analyze the energy consumption over 3 hours.

In practice, the test started from step 3 directly, because the PID controllers can maintain the temperature quite stable and did not need further PID parameters adjustment.

4.1.1 Energy consumption of each equipment over 3 hours in each condition

In this refrigeration system, the main energy consumers are:

- the compressor (with frequency controller)
- the central pump
- the decentralized pumps
- fans and cabinets controllers

As can be seen from Figure 8, with the increasing of t_setting, the E_Dpumps was increasing a little bit, E_Cpump was almost constant, E_Comp was decreasing obviously and E_Fans&Controllers was constant. The total energy consumption was decreasing.

As shown in Figure 10, the increasing of the t_setting (brine out temperature at the evaporator side) results in the increasing of evaporation temperature (t_Eva). The increasing evaporation temperature provides a larger COP_c. This means that with the same amount of cooling load, the compressor consumes less energy. The higher brine out temperature makes lower brine density, which leads to higher brine flow (Flow_Bri) and thus the higher pump energy consumption. Higher brine flow makes the temperature difference of the brine in and out (dt_RF_Bri) smaller. But, one conclusion can be made; by increasing the t_setting, the energy saving from the increasing chiller COP is more than the increasing of the pump energy consumption. The total energy consumption will decrease (Figure 8).

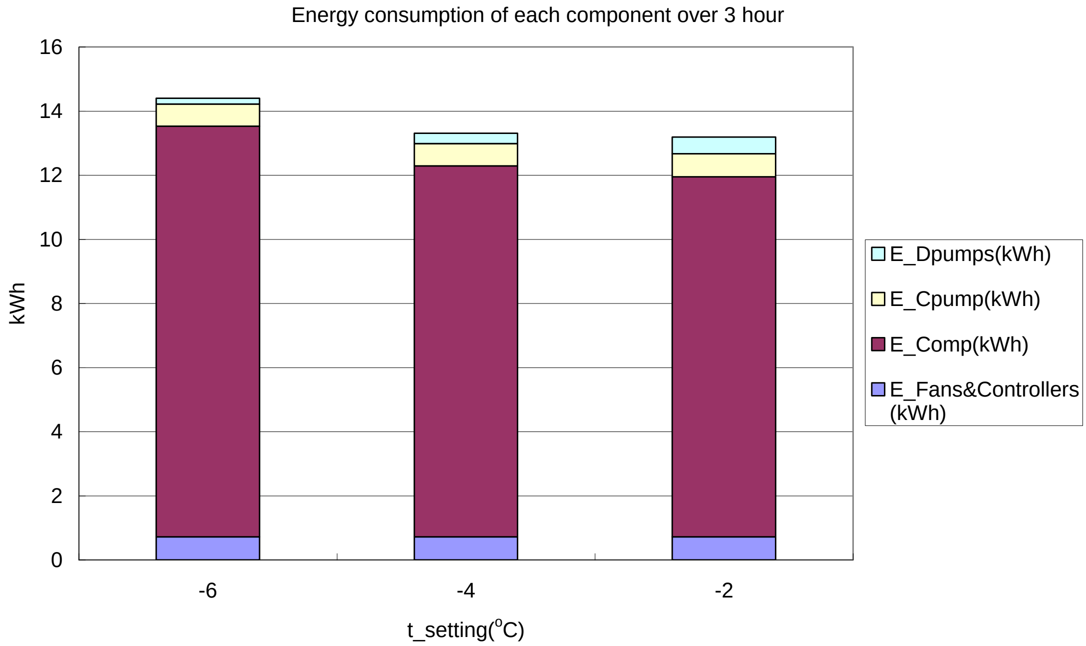

Figure 8 Energy consumption of each component over 3 hours.

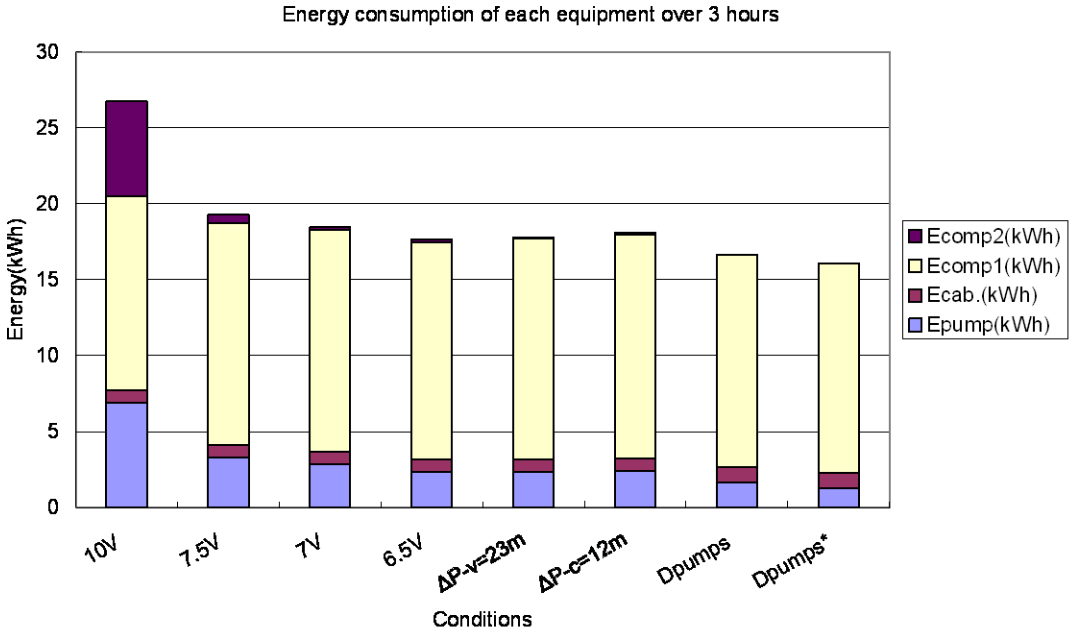

Figure 9 Energy consumption of each component over 3 hours (Phase 1).

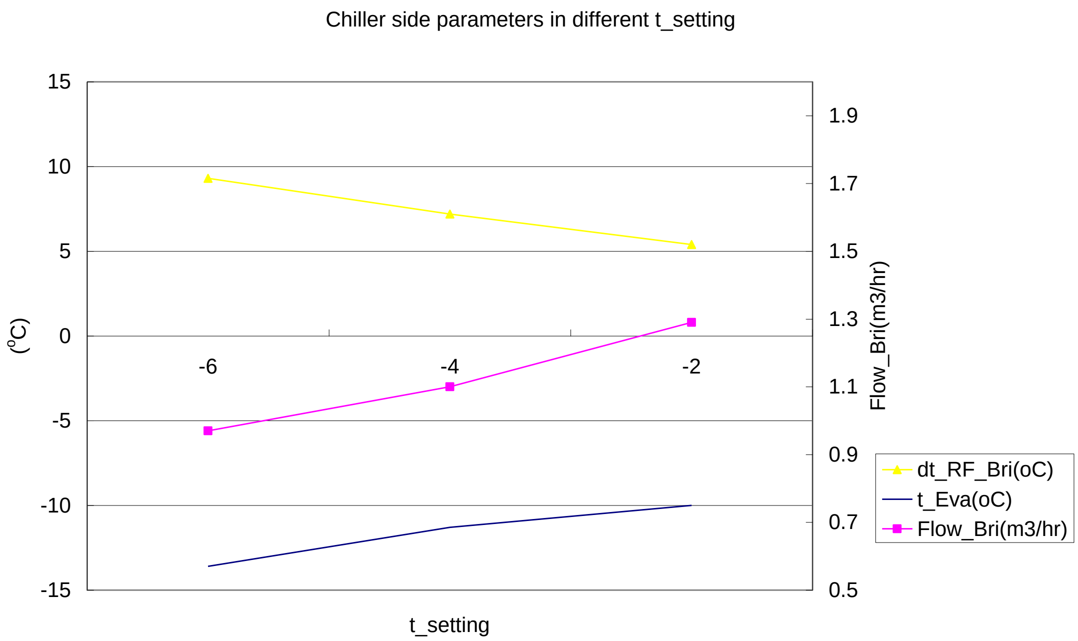

Figure 10 Chiller side parameters in different t_setting

4.1.2 Ratios between different parameters at different t_setting

Figure 11 shows that the ratio (E_Dpumps/E_Cpump) and (E_Cpump+E_Dpumps)/E_Comp are increasing as t_setting is increasing.

As the flow was increasing, owing to increasing t_setting, decentralized pumps are prone to increase the power more than the central pump, thus the increasing of E_Dpumps/E_Cpump.

Since t_setting is increasing, the brine flow and evaporation temperature will increase, which lead to an increase of (E_Cpump+E_Dpumps) and decrease of E_Comp. In this case, the (E_Cpump+E_Dpumps)/E_Comp will increase for sure.

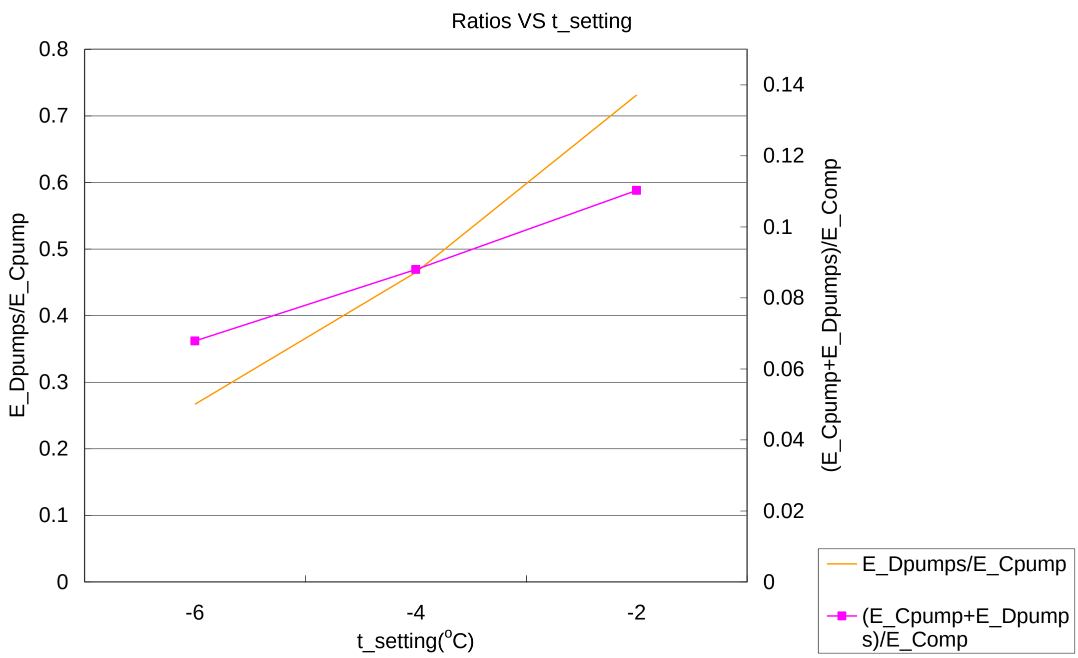

Figure 11 Equipment power ratios VS t_setting

4.2 Test 4: Pressure controlled central pump

The check valve is used to prevent the backward flow in the bypass line. In this case, the maximum brine flow can be ensured. The COP of the chiller may increase owing to the higher brine flow.

This test is included with 3 steps, each step is varying the central pump Δp-c setting between 10m, 8m and 6m, in each step it is included with 3 sub-steps which are varying the t_setting between -6 ^oC, -4 ^oC and -2 ^oC.

As can be seen from Figure 12, the first 3 solid bars represent the energy logged directly from the energy meters; the compressor energy (E_Comp), central pump energy (E_Cpump) and decentralized pump energy (E_Dpumps).
The other 4 pattern bars represent the calculated energy, which are cooling capacity calculated from the refrigerant side (Q2), cooling capacity calculated from the brine side (Q2_Bri), cabinet cooling load (Q_Cab) and controller heat loss (Q_Controller).

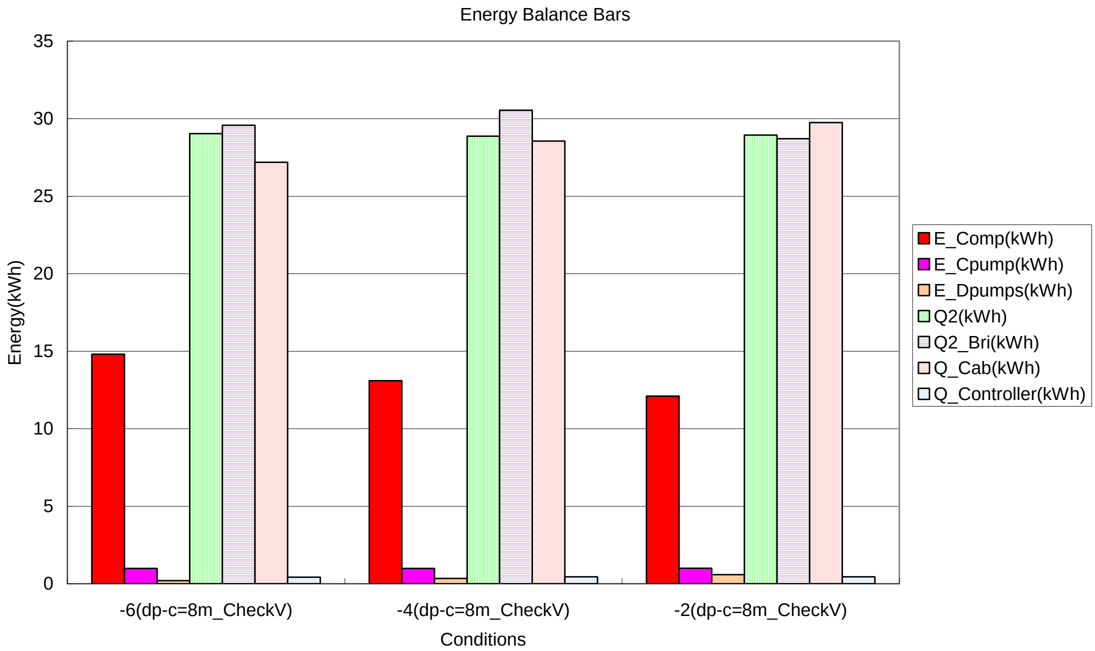

Figure 12 Energy balance bars at Δp-c=8m.

In the condition of Δp-c=10m, Q_Cab changed a lot in different t_setting, some unexpected factors might have influenced the test, so this step is not to be analyzed. In the condition of Δp-c=6m, when the t_setting was up to -2^oC, the food temperatures in one cabinet start to rise, which means the flow is not enough to meet the cooling load. Thus, this step is also eliminated. Only in the condition of Δp-c=8m, where all four calculated energy values kept relatively stable, the data could be used for analyzing.

When comparing Figure 12 and Figure 13, it can be seen that with the same Q_2 in each condition, the increasing of t_setting leads to the increasing of evaporation temperature (t_2). A increase of t2 will increase COP and thus the compressor energy consumption became less. On the other hand, the increasing of the t_setting resulted in the relative higher working temperature of brine, thus the flow will increase owing to the lower brine viscosity, and consequently, the increase of the decentralized pumps energy.

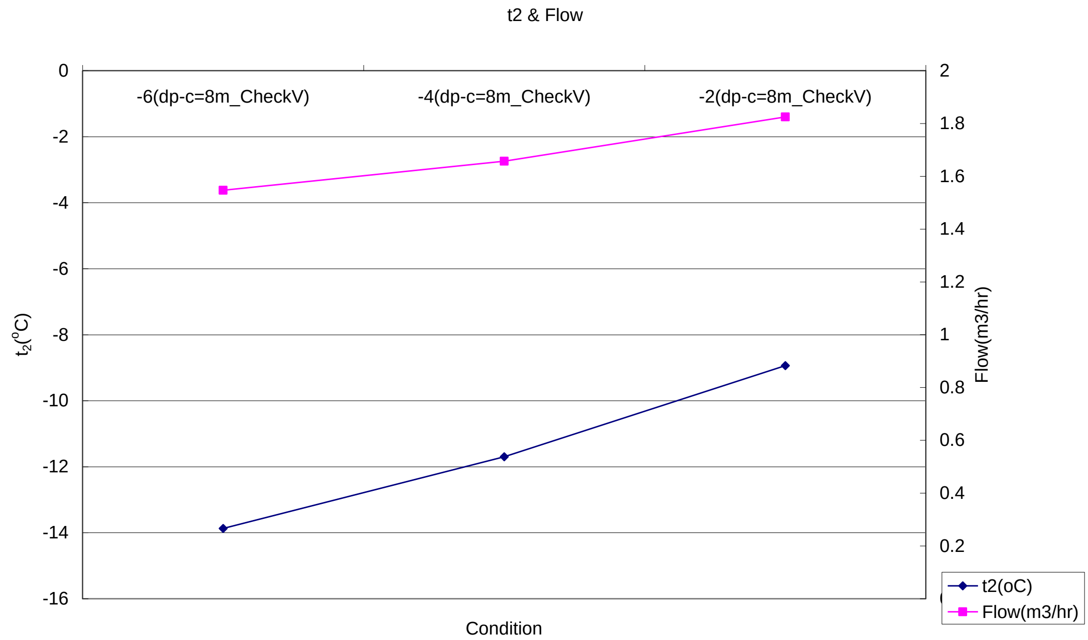

Figure 13 Main brine flow and t_2 in each condition.

The proportion of bypass flow over main flow is decreasing with the increasing of t_setting (brine forward temperature). The cabinet1 flow is increasing according to the main flow. The flow curves are shown in Figure 14 Three flows VS different t_setting. This phenomenon shows that the higher brine forward temperature makes it easier to pump the brine through each cabinet. Higher brine temperature gives lower brine viscosity, and thus the lower cabinet pressure resistance. In other words, the higher brine forward temperature enables good flow distribution in each cabinet and avoids the big proportion of bypass flow.

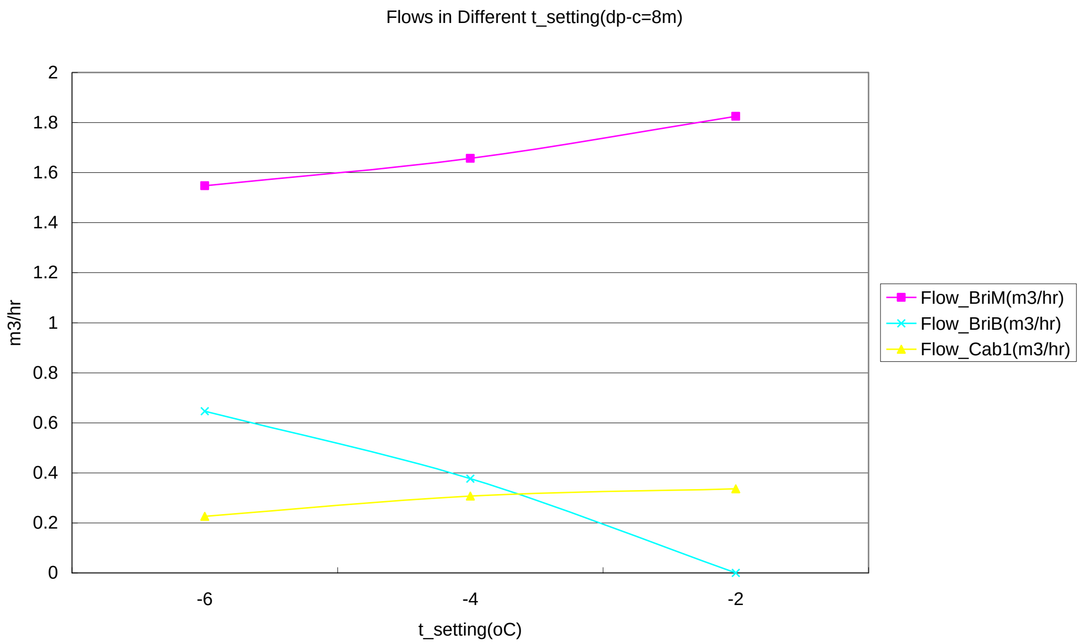

Figure 14 Three flows VS different t_setting

4.3 Test 5: Dynamical change of cabinet load

The purpose of this test was to evaluate the response of the cabinet control system. As mentioned previously the quick and accurate response of the cabinet is one advantage of the current concept.

A dynamical change of load was performed by putting a heater into one cabinet for a short period of time. The heater was then removed after a short period in order to bring the system back to normal operation again.

During the test, the heater was placed in front of cabinet 1. By changing the angel of the heater, the heat load could be changed. Figure 15 shows the heater facing outside of the cabinet in order to have a small heat input. Figure 16 shows the heater facing the inside of the cabinet, providing a large heat input. The influence of the heater for the system can be seen from the curves shown in Figure 17.

Figure 15 Dynamic heat load change. (low load)

Figure 16 Dynamic heat load change. (high load)

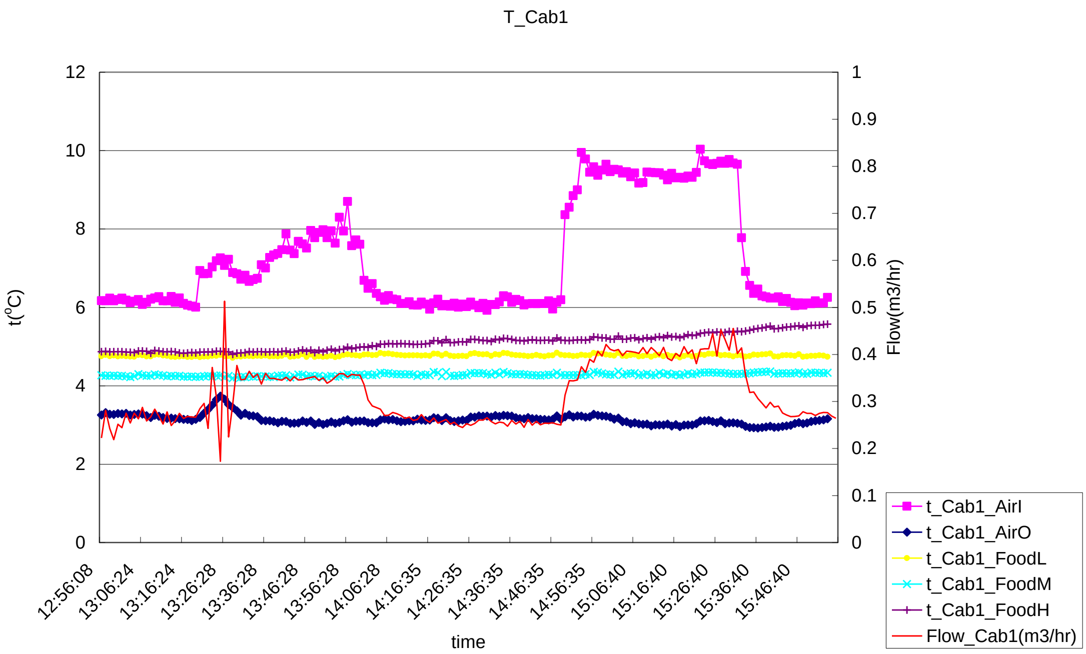

Figure 17 Temperatures and flow curves for Cabinet 1 (t_setting=80%, dp-c=8m).

In Figure 17, all the temperatures and brine flow logged for Cabinet 1:

- The air out temperature (t_Cab1_AirO) of the cabinet.
- The air in temperature (t_Cab1_AirI) of the cabinet.
- The food simulators which are located at; low (t_Cab1_FoodL), middle (t_Cab1_FoodM) and high (t_Cab1_FoodH) position.

Two "peaks" are obvious in the t_Cab1_AirI and Flow_Cab1 curve. The t_Cab1_AirI is quite sensitive to the heater because its thermocouple is mounted right after one of the cabinets fan. The brine flow, Flow_Cab1, is controlled by the decentralized pump which has a thermocouple mounted at the air-out side position of the cabinet heat exchanger (HX). The peak of the flow matches very well with the peak of the air-in temperature.

This proves that the PID controlled decentralized pump has a fast response time. Not only that, since the "air-out" is almost not affected it means the flow increase can absorb the full added heat load. It is a sign that this concept has a favourable effect on the cabinet controllability.

The temperature t_Cab1_FoodH was increasing gradually during this period while the other two food temperatures were stable. The brine temperatures (Figure 18) at the evaporator side were not influenced that much by the dynamic heat load change. Only a minor temperature rise can be noticed for the brine in temperature curve (t_RF_Bri_I), mainly at the right part.

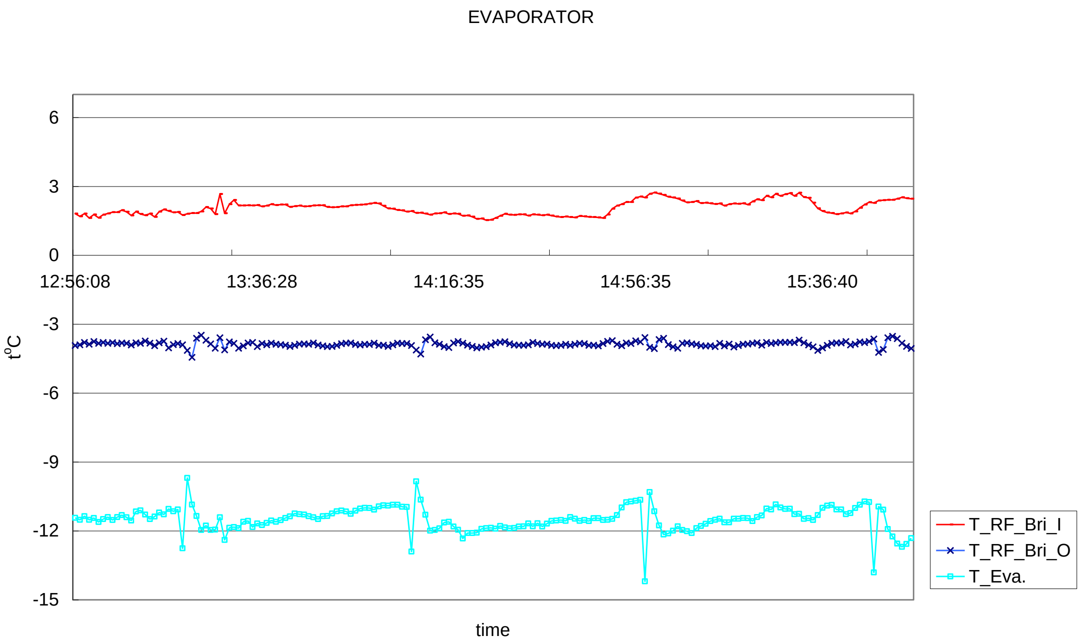

Figure 18 Temperatures at the evaporator side (t_setting=80%, dp-c=8m).

4.4 Test 6: Central pump and compressor integrated control

In this section the control of the central pump is integrated with the compressor control. In practice the compressor control signal is used in parallel to speed control the pump.

The ideal system would operate in the following way:
When the cooling load is low, both the compressor power and pump power should be decreased, due to less cooling capacity less flow is required.

In this test, the central pump was set at the external signal control mode in order to have it controlled by the compressor frequency controller. The pump speed is depending on the external signal value (working voltage ranges from 3-10V) obtained from the frequency controller.

During the operating time, the frequency controller will send the analogue signal (voltage) to the central pump simultaneously as it is controlling the compressor. In this case, the speed of central pump and compressor are changing at the same pace.
Figure 19 shows the connection of external signal wires with the pump. The batteries are used to ensure that the external signal is always above 3V. This will provide a minimum brine flow in the system even when the compressor stops.

Figure 19 Pump is controlled by the external signal sent from the compressor frequency controller.

A change in speed of the central pump will lead to a different brine flow (Flow_Bri). At the same time, the brine out temperature (T_RF_Bri_O) at the evaporator side will also change. The change in brine flow and brine out temperature are normally in the same direction, either increasing or decreasing. For example, when the brine out temperature is decreasing, the frequency controller will adjust the compressor speed and the power will go down. At the same time, the central pump speed will also decrease; causing a reduced amount of flow through the evaporator. In this case, the decreasing brine out temperature will be accelerated.

As the brine flow changes, the brine out temperature will behave in the same manner. This will introduce oscillation curves regarding the brine temperatures (Figure 20). Since the compressor is controlled by the frequency controller in a good manner, the evaporation temperature can be maintained relatively stable.

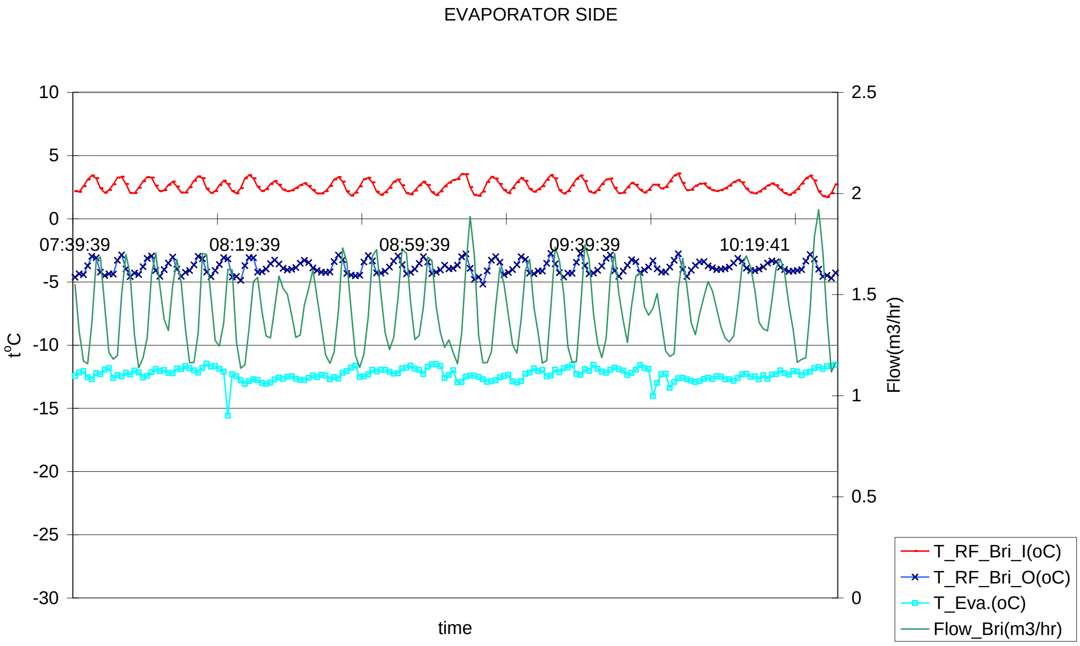

Figure 20 Main brine flow and the temperatures at the evaporator side.

Figure 21 shows all the equipment power in the system. P_Comp+PID stands for the power of the compressor and frequency controller. Q_Controller stands for the heat loss from the frequency controller. P_Cabs stands for all the power consumption from the cabinet side which include cabinet controllers, fans and decentralized pumps. P_Cpump stands for the central pump power.

One easily noticed phenomenon is that P_Comp+PID has the same changing trend as P_Cpump, owing to the integrated simultaneously control. The alternation of the P_Cabs curve is due to the resolution limitation by the energy meter. Curve Q_Controller performs quite stable, which means that the heat loss level from the frequency controller is not sensitive to the compressor power.

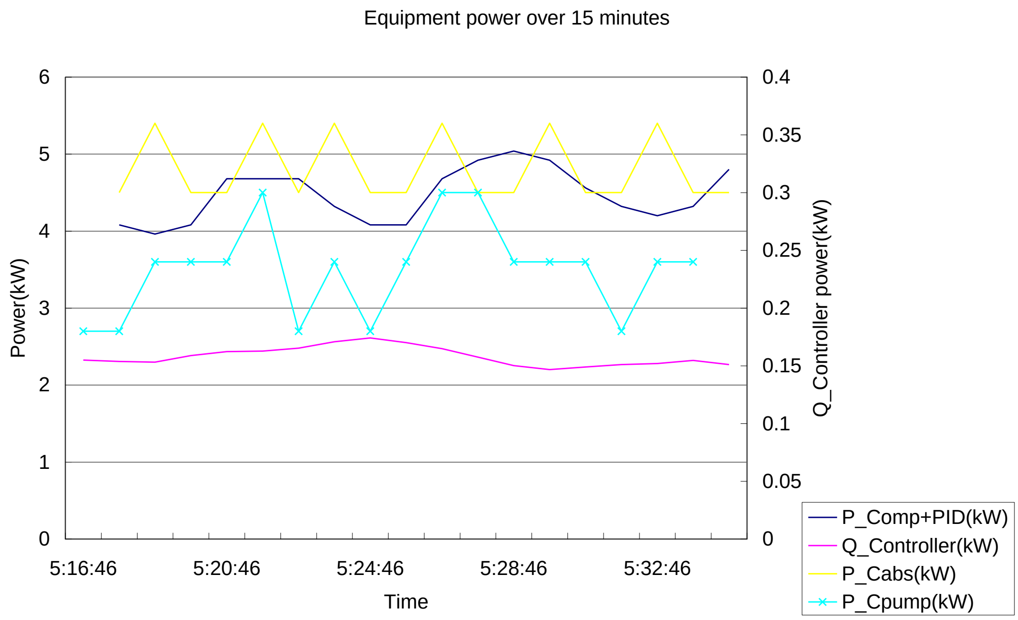

Figure 21 Equipment power consumption over 15 minutes

In order to compare energy utilization from the last test, a reference test result has been selected. As can be seen from Figure 22, the selected reference conditions are:

- Central pump set to Δp-c control mode (8m).
- Check valve is installed.
- T_setting = -4^{o}C (80%).

The reason why this particular case has been chosen has a reference, is due to the stable conditions. Q_2, Q_2_Bri and Q_Cab appear to be within the similar value, which proves that the system was operating at a relatively stable condition.

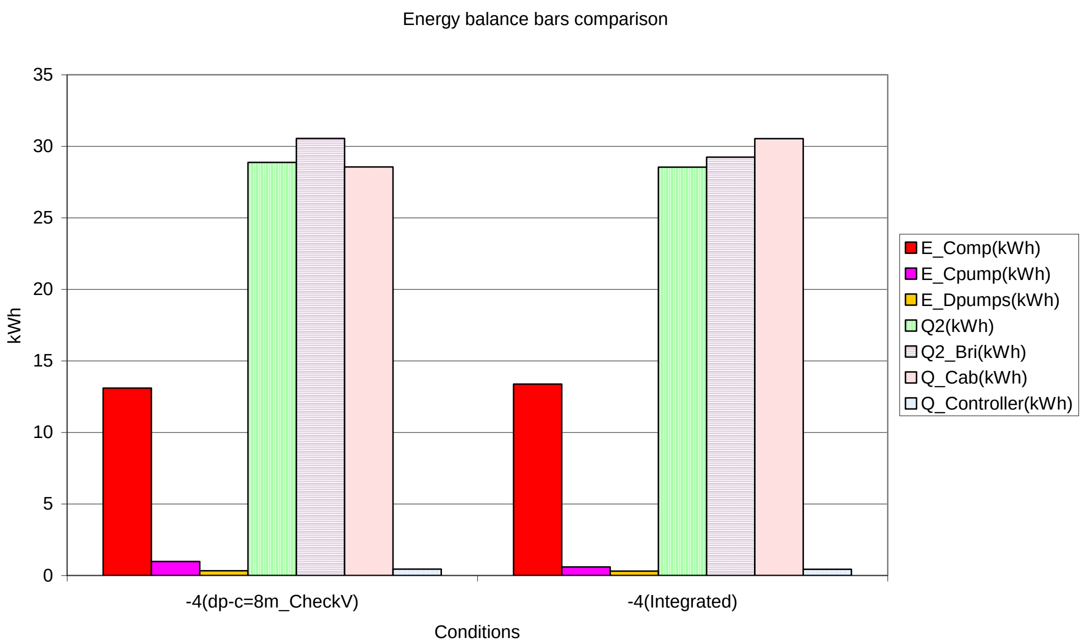

Figure 22 Energy balance comparison between condition -4^{o}C (dp-c=8m_CheckV) and -4 ^{o}C (integrated control).

At the first sight of these two groups of bars, they look quite similar. At a closer inspection a difference can be noticed. The compressor energy consumption is a little bit higher in the integrated condition, while the central pumps energy consumption is considerably lower.

The decentralized pumps energy consumption is a little bit higher in the integrated case which is difficult to notice from the Figure 22. The other four pattern bars provides information about the stability of the system and is not analyzed further.

4.5 Test 7: Exchange of brine from propylene- to ethylene glycol mixture.

In the previous tests, the brine has consisted of 50% propylene glycol (a mixture of 50 mass% propylene and 60 mass% water). The freezing point of mixture is -31°C and its density is 1208 kg/m3 at 25°C. The norm is 40%, however, it was discovered when replacing the brine that the concentration as higher than intended.

In this part of the test the propylene glycol has been exchanged and replaced by 35 % ethylene glycol. This mixture consists of 35 mass % ethylene and 65 mass % of water with a density of 1063 kg/m3 and a freezing point of - 33°C.

The system is operating in the same way as in Test 6 with an integrated control of compressor and central pump. The objective is to reduce the central pump's energy usage while the food substitutes are maintaining its temperature. The test has been performed in three different parts in which three values of t_settings has been tested (-6°C, -4°C and -2°C). As can be seen in Figure 23 the energy usage by the compressor is reduced as the t_setting is increased (higher evaporating temperature). The energy consumption by the central pump will also decrease since it is integrated with the compressor control. In order to maintain an acceptable food temperature the decentralized pumps have to operate even harder and consequently consumes more energy.

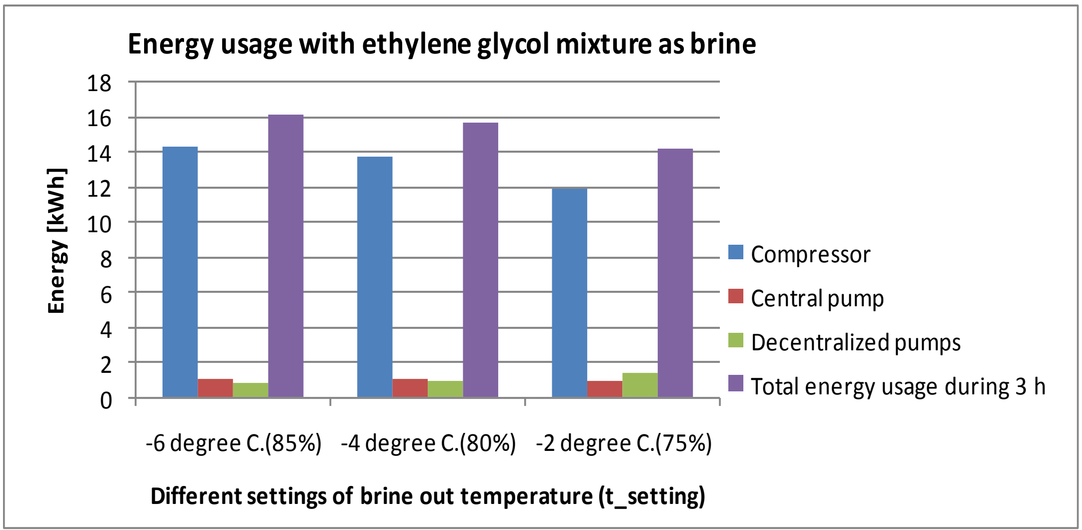

Figure 23 A comparison of energy consumption during three hours for different values of t_setting.

The Figure 24 below shows a comparison of energy consumption in the use of different brines and control systems. The Figure 24 is basically combination of Figure 22 and Figure 23 for the case of t_setting = -4°C. The compressor energy usage is a bit higher in the integrated control mode, which has already been conformed in Figure 22. Even though the compressor power is less in the pressure control mode of the central pump then the case with integrated system, the energy utilization by the central pump remains quite constant.

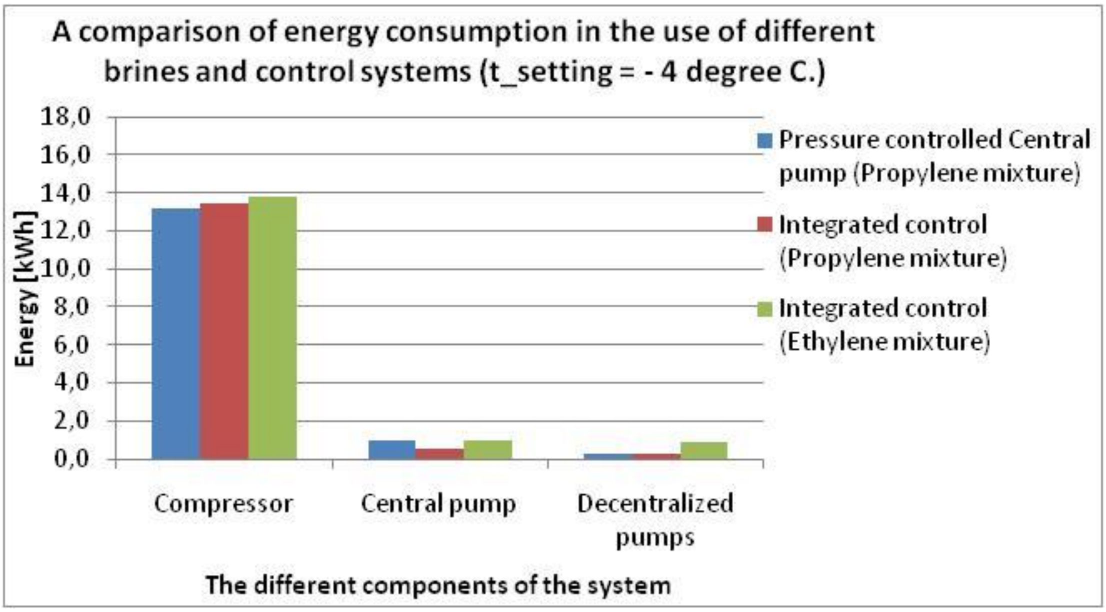

Figure 24 A comparison of energy consumption during three hours for different brines and control modes.

4.6 Over all results discussion

To conclude this report it can be said that the respective development phase has shown interesting results with respect to the decentralised pumps in indirect refrigeration systems.

In phase 1 a traditional system solution with a central pump was compared with the first generation decentralised pump system. The conclusion was an energy saving of 6% on a system level and 17% of the pump distribution energy. These measurements were based on 24 hrs testing and included a defrost cycle.

	Performance of the system during test period (3 h)		
	Reference test -- Phase 1 --	The most energy efficient system -- Phase 1 --	The most energy efficient system -- Phase 2 --
Cooling capacity, Q2 [kW]	11.0	10.7	9.51
Evaporation temp. [degree C.]	-12.7	-12.9	-12.32
Condensation temp. [degree C.]	37.7	37.7	37.30
Brine forward temp. [degree C.]	-3.7	-3.5	-3.84
Humidity [%]	30.2	31.3	24.60
Room temperature [degree C.]	20.4	20.0	19.90
Compressors	14.9	14.5	13.31
Central pump	2.3	1.7	0.59
Solenoid valves	0.9	-	-
Decentralized pumps	-	1.0	1.02
Total energy usage	18.0	17.2	14.92
Energy savings compared to reference test [%]:	-	**4.4%**	**17.0%**

Figure 25. Compilation of the results from the different development steps

I figure 25 the key results from three significant steps of the development is compared. These results are based on the second phase method with 3 hours stable operation. For this reason the results may look slightly different compared to the previously shown but the comparability over the whole project is improved.

Further, the table in figure 25 gives an overview of the basic parameters in the tests. The stated cooling capacity is based on the refrigeration machinery and the pump energy generated will therefore be included in this overall cooling capacity.
The most striking difference is the reduction in the central pump energy demand. Between the two tests in phase 1 there are two major differences; the control strategy and the decentralised pumps. The latter will take a part of the work from the central pump but mainly the adaption to the required flow is the main driver for the reduction in energy usage.

In phase 2 the adaptation of the secondary refrigerant piping improves the situation, but the main driver is the coordinated compressor and pump control. Due to the pump work which is adapted to the cooling capacity the pump energy is significantly reduced.

Overall a 17% energy reduction is achieved on a system level and the distribution part of it ($E_{Brine\ pump}/E_{comp}$) for the cold side is reduced from 21% to 18.6% in phase 1. In phase 2 this was further reduced to 12% and in relation to the refrigeration energy the corresponding figures are 9.7%, 8.4% och 5.6%.

The interesting part here is rather the relative improvement than the absolute level of the distribution energy, since the system design highly influences the pump work. The present system is of an ultimate design and therefore the pump work is relatively high.

5 CONCLUSIONS

The project test targets have been fulfilled as planned with the integrated control of compressor and the central pumps.

The achievement of frequency controlled compressor and PID controlled decentralised pumps requires a good knowledge of automation and instrumentation. The setting of controller's parameters determines how the equipment can behave in terms of system stability and energy consumption. The start-up time is dependent on the complexity of the system.

The brine out temperature setting at the evaporator side (t_setting) is important for the system energy consumption. Generally a higher t_setting is equivalent to lower system energy consumption. The reason is that a high t_setting will increase the temperature of the evaporation and less power to the compressor is needed.
Even though the brine flow increases due to the higher brine temperature (which results in higher pump power) the compressor power saving will always be dominating vs. the increase in the pump power.

The frequency controlled compressor is a good energy solution and the system is able to adapt the cooling capacity to the cooling load efficiently.

Some specific conclusions regarding to the conducted tests:

1. The integrated control of the compressor and the central pump (along with the PID regulated decentralized pumps) has improved the system even further in an energy point of view. By comparing the results from phase 1 the system is improved by 13%.

2. The compressor consumes the major part of the energy, which mainly depends on the evaporation temperature and thereby the evaporation pressure. Therefore the pump function must ensure highest possible evaporation temperature i.e. it must not render in a penalised heat transfer which lowers the evaporation temperature.

3. The most efficiency point is an important parameter for the central pump. Thus, instead of having one large central pump with speed control, two smaller central pumps can be mounted in parallel with the step control method to fulfill the task. The controlling method can be operating in the following way: when there is one or two cabinets operating, only one pump is on. If there are three or four cabinets operating, both of the pumps start to operate. In this case, each pump can work around its best efficiency point to a larger extent.

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