Final Report

German and Danish Part

"Innovation in Small Capacity Ammonia Refrigeration Plants"

Contract N^{o} ENK6-CT-2002-30020

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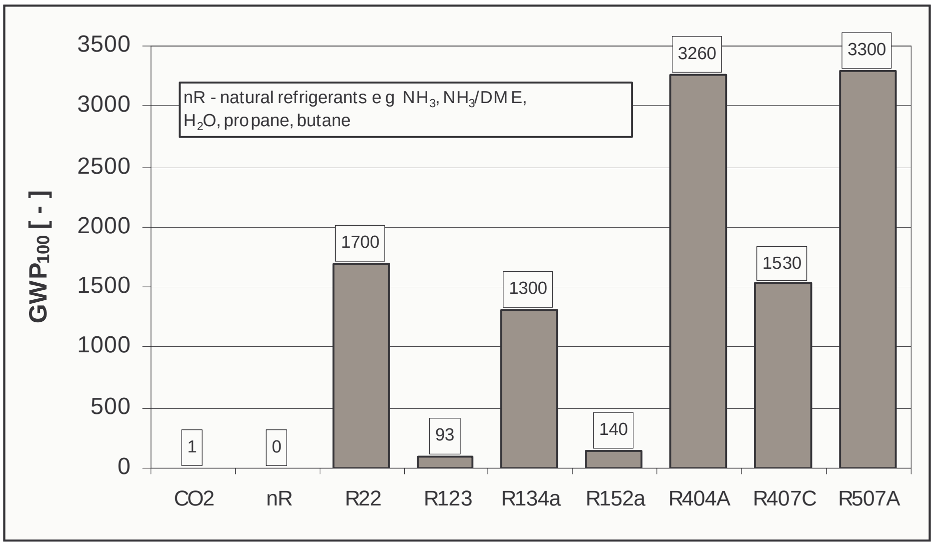
1 Introduction

In connection with compression refrigeration plants the chemical refrigerant R134a and refrigerant mixtures such as R404A, R407A, R407C, R410A or R507 are typically used. These refrigerants do not have any ozone depletion potential but they have a global warming potential. So far, the utilisation of natural refrigerants has been limited. CO_2 and H_2O refrigeration plants are being developed and butane and propane applications are limited to very small systems and ammonia (NH_3) to larger applications.

The potential of the environmental pollution of the refrigerants is characterised by the coefficient to the potential of the warming of the atmosphere of the earth (GWP = Global Warming Potential). The numerical data of this coefficient represents the respective polluting rate of the refrigerant in relation to CO_2 in which the GWP-value of CO_2 was set to 1.0. The GWP-value gets the GWP on an equivalent amount of CO_2. For R134a this factor – 1300 – should be seen over a period of 100 years. That means that an amount of 1 kg R134a corresponds to an equivalent GWP of 1300 kg CO_2. The GWP value for NH_3 is almost 0.

Figure 1.1 shows the polluting potentials of some refrigerants with regard to the greenhouse effect. Unlike the chemical refrigerants the natural refrigerants do not have a potential for the environmental pollution, because they are already part of the environment. It is to be assumed that in future natural refrigerants will mainly be used. That is why an expanded application of ammonia is to be aimed at in the small applications.

Figure 1.1: GWP for different refrigerants



The real contribution to the greenhouse effect of a refrigeration plant originates from the direct part of the refrigerant (GWP) and the indirect part from the energy consumption. This value is an evaluation number in kg CO2-equivalent and is called TEWI (Totally Equivalent Warming Impact). In order to hold the indirect part low, the refrigeration plants that are operated with natural refrigerants must also show high COP and correspond to the latest developments in technology.

A reduction of the greenhouse effect can be achieved only through innovative plants that are in operation. For that the plants with natural refrigerants must be competitive compared with the conventional refrigeration plants.

The objective of the project was the manufacturing of 6 demonstration plants, which use ammonia or the mixture of ammonia-dimethylether as natural refrigerants, which show a high COP and are economically competitive.

In the present final report the results of the development task are represented. The demo-plants was build-up, investigated and measured and regarding performance and energy consumption evaluated.

Within the project processing the EU-line for the refrigerant problematic concretised itself. The regulation proposal is represented within the final report. Because with the EU proposal the exit from the use of refrigerants with greenhouse effect was shifted, The use of R717 or R723 results less by the commitment of use of natural refrigerants than through the reduction of the energy consumption and achievement of a economic efficiency.

2 New State of Affairs of Law and Decrees within the EU

In the Midterm-Report of the project (February 2004) the state of the legislations and regulations in Germany, Denmark and EU was explained fully. In Germany and in Denmark this state did not change up to the current date. Currently, Danish national Regulations still exist on the regulation of certain industrial greenhouse gases (Regulation of certain industrial greenhouse gases BEK No. 552, that came into force on 02/07/2002). The proposed EU Regulation could still influence Danish legislation but, as it probably will take a long time before a finally adopted Regulation or Directive exists, it is still necessary to prepare the industry for compliance with current Danish Regulation.

The standard EN 378, part 1 – 4, is being revised for the time being, and it might influence the classification of R723. In the former edition of the standard both R717 and R723 was classified as B2, but now it seems that R723 will be classified as B3. This fact could be very important for future use of R723 in some applications. All though a new legislation for refrigeration systems in Denmark was released in December 2003, it will not change anything regarding the use of R723 in Denmark – in relation to this project.

The EU- modus operandi concretised itself 2004. In October 2004 a political agreement for the European F-Gas-regulation was achieved within EU. It is expected that the legislative procedure will be completed at the end of 2005 and the EU-regulation is consequently law-effective at the beginning of 2007.Currently, one EU Regulation on ozone layer depleting substances exists. The ozone Regulation i.a. comprises the following requirements and restrictions:

- From 1 January 2001 all use of CFC (R11, R12, R13, R502 etc.) will be prohibited in all EU Member States.
- All refrigeration systems with CFC that are repaired shall be converted to other types of refrigerant.
- It is prohibited to purge CFC into the atmosphere. It has to be collected and delivered for destruction.
- From 1 July 2002 it is prohibited to sell new HCFC plants in the EU.
- EU imposes each Member State to prepare an obligatory approval agreement (In Denmark a system exist called “Kølebranchen Miljøordning KMO” – in English “Environmental System within Refrigeration Business”).
- Everybody working with CFC or HCFC in the EU has to be approved by an authority.
- Minimum requirements have to be defined for personnel handling refrigerants.

- EU imposes each Member State to monitor the leakage of CFC or HCFC if the charge exceeds 3 kg.
- No minimum limit has been determined for leakage but plants have to be inspected at least once annually.

The Regulation on ozone layer depleting substances (No. 2037/2000) came into force in September 2000 and since then it has been revised twice (i.a. with Regulation No. 2038/2000/EF and Regulation No. 2039/2000/EF) – most recently in March 2003.

The law proposition for the Greenhouse Gas Arrangement has following measures, demands and restrictions to the content:

Legal basis

The environmental protection regulations of the EU contract allowed the member states to introduce nationally severe definitions. Under the domestic market definitions this is only difficultly and under specific conditions as well as after approval of the commission possible. The current arrangement chooses the domestic market definitions of the EU-contract as a basis for the product and application prohibitions and for the marking definitions. It safeguards a homogeneous domestic market with that for products and equipments in this field while to the member states a greater freedom is remaining in the organization of the emission control. There are comparable arrangements in the border range between environmental and chemicals right.

Field of application and definitions

The regulation contains an obligatory list of the most important fluorinated gases and decides, by the way, a threshold value from 150 for the GWP. This threshold value is certain of the GWP for the refrigerant R152a. In all rule now the operator is supposed to be responsible for the realization of the arrangements for emission control and the correct disposal of gases. The member states can choose also another solution however in reasonable cases.

Emission reduction

This main clause of the law contains in item 1 at first once the obligation for operators of stationary refrigerating and air-conditioning plants, heat pumps and fire fighting devices, to apply all technically possible measures for avoiding of plant`s leakage and repairing of occurred leakage as soon as possible that are not combined with disproportionately high costs.

To control the leakages these plants must be inspected regularly through qualified personnel. The frequency of this control depends on the amount of the used gases. Yearly controls are prescribed from a charge of 3 kg, semi-annual controls from 30 kg and monthly controls from 300 kg. For large-scale plants from 300 kg an automatic leakage control system is furthermore prescribed. Provided that a functioning leakage control system is fitted, the frequency of the controls can be reduced to the half. All operators of such plants have to lead furthermore a log-file, in which information on the amount and kind of the used gas, the carried out controls and refilled amounts are registered. The clauses in the commission proposition were based largely on the Dutch system STEK, that is practised there since a lot of years with success.

Recovery of the gases (refrigerant)

For the decrease of emissions of fluorinated gases particularly in the refrigeration and air-conditioning these must be gathered during the maintenance and the disassembly of the plants and an arranged reuse must be provided.

Training and certification

The basic consideration of this section is, that developed personnel are supposed to deal only correspondingly with fluorinated gases. The commission must make out however minimal orders still together with experts of the member states for the training of the personnel and the member states must move these orders in quite short time, because already two years after coming into force of the regulation companies may be delivered only with fluorinated gases that possess certified personnel.

Report

Report is necessary, however the regulation refuses very consciously on a recording of the last kilogram on single company internal tier.

Marking

This clause is new and it is supposed to point out that particularly in case of widespread applications in refrigeration and air-conditioning where refrigerants are used, and guarantee a proper maintenance and disposal with that.

Limitation of the use and the introduction

The regulation plans application or marketing prohibitions in that cases, in which emissions are not to be avoided (so-called "open applications") and to stand in those ones economically acceptable alternatives exist. In the list of the prohibitions stand some substances and applications for example the application of SF6 in tires or the prohibition of fluorinated gases in aerosols for artificial snow as well as in one component foams and fire extinguishing systems.

Revision

An inspection of the effectiveness of the regulation is planned by the commission, when the data base became larger and the further evolution in a line of fields of application is to estimate better.

3 Operating Conditions of Demo Plants

The following plant has been developed and designed by Danish Technological Institute within the scope of the OSCAR-project:

Plant I: Heat pump (Chiller) System for milk cooling (SVK)

Specification of plant conditions:

- A chiller unit for milk cooling with water from ice-bank system on a farm where the heat from the condenser is used to heat tap water and water for a radiator system.
- Cooling capacity: 17-18 kW.
- Evaporating temperature (Brine inlet temperature): -5 C
- Condensing temperature (Hot inlet temperature): 42 C

Plant II: Chiller System for marine applications (Buus)

Specification of plant conditions:

- A chiller for indirect cooling in cold (or frost) storage rooms on fishing vessels.
- Cooling capacity: ≈ 10 kW
- Evaporating temperature: -21,5℃ (Brine inlet tempe rature): -18℃
- Condensing temperature: +28℃ (Sea water inlet temp erature): max. 20℃

Plant III: Ice Flake Machine for commercial market (Buus)

Specification of plant conditions:

- A compact ice flake unit (based on existing SM500UL-type for commercial market) with indirect refrigeration system including air-cooled condenser, which adds heat to its surroundings.
- Cooling capacity: 3 kW (ice capacity ≈ 500 kg / 24h)
- Evaporating temperature: -24℃ (Brine inlet tempera ture): -21℃
- Condensing temperature: +40℃

The following design conditions were defined for the Demo Plants and are basis of the computing, design und dimensioning by ILK Dresden:

Demo Plant IV: refrigerating system for cooling cells (normal cooling - NC)

Specification of the operating conditions:

- refrigerating systems for cooling cells to normal cooling with air cooled condenser and evaporator and with the refrigerant R723

- refrigerating capacity: 10 kW_o
- evaporating temperature in the cooling cell: –10℃
- condensing temperature (under ambience air conditions): +42℃

Demo Plant V: refrigerating system for cooling cells (deep freezing - DF)

Specification of the operating conditions:

- refrigerating systems for cooling cells to deep freezing with air cooled condenser and evaporator and with the refrigerant R723
- refrigerating capacity: 10 kW_o
- evaporating temperature in the cooling cell: –28℃
- condensing temperature (under ambience air conditions): +42℃

Demo Plant VI: Brine Chiller

Specification of the operating conditions:

- brine chiller as compact refrigerating aggregate with water cooled condenser and brine admitted evaporator and with the refrigerant R723
- refrigerating capacity: 20 kW
- condensing temperature in condenser: +34℃
- evaporating temperature: –14℃

The plant parameters of the demo plants are summarized in table 3.1.

Table 3.1: Parameter of plants– refrigerating capacity, evaporating- and condensing temperature

Plant	**I**	**II**	**III**
Refrigerating capacity	17-18 kW	10 kW	3 kW
Evaporating temperature	.-10℃ (t_{b_out} - 5℃)	-21℃	-24℃
Condensing temperature	.+30℃ (t_{w_out} +45℃)	28℃	+40℃
Plant	**IV**	**V**	**VI**
Refrigerating capacity	10 kW_o	10 kW_o	20 kW_o
Evaporating temperature	-10 ℃	-28℃	-14 ℃ (brine) $t_{Brine\ Outlet}$
Condensing temperature	42℃	42℃	34℃

The results of the theoretical comparison between conventional refrigerants, ammonia (R717) and the mixture of ammonia and dimethylether (R723) as well as between one- und two-stage deep freezing are represented in the Midterm-report fully. The following figures shall support the determination of the plant design described in the Midterm-report.

Figure 3.1 compares the saturation pressure for different refrigerants. R404A owns with usual condensation conditions relatively high pressures and a plant requires a higher nominal pressure stage. Then again R134a has under -26 °C evaporating temperature a saturation pressure that is under the atmospheric and at which the danger of air entry into the plant exists.

Figure 3.1: Saturation pressure of different refrigerants in dependence of saturation temperature

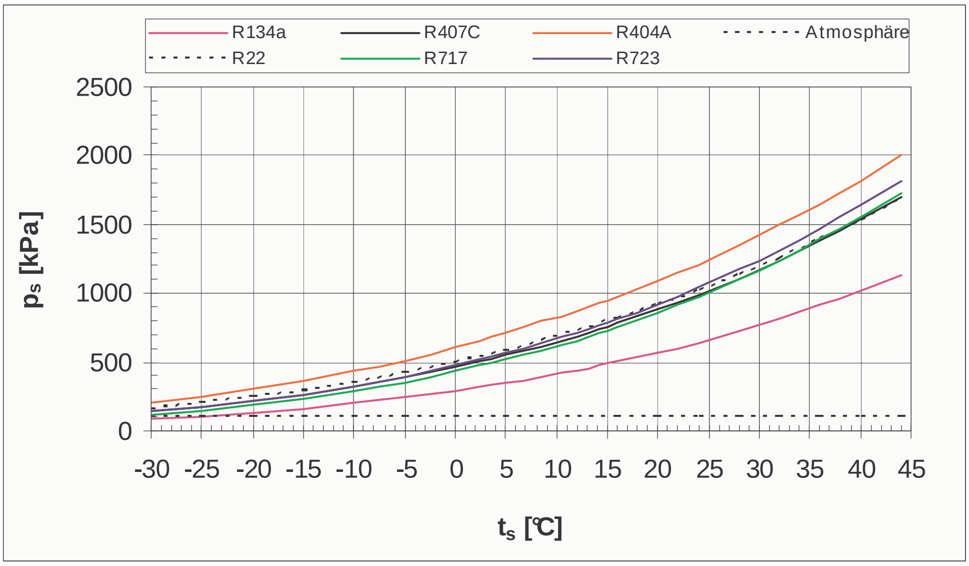


Figure 3.2: Comparison of pressure ratio and COP for R723 versus R717 in dependence of the evaporating temperature

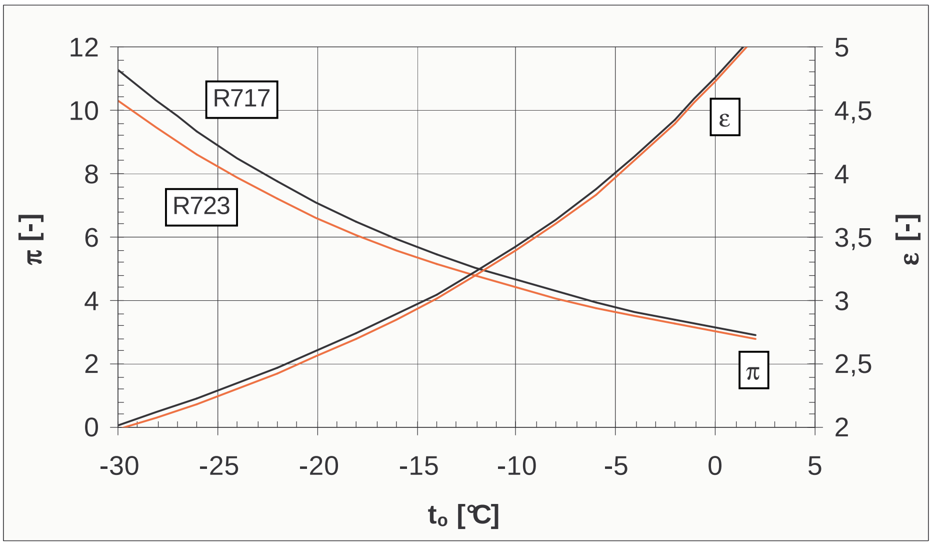
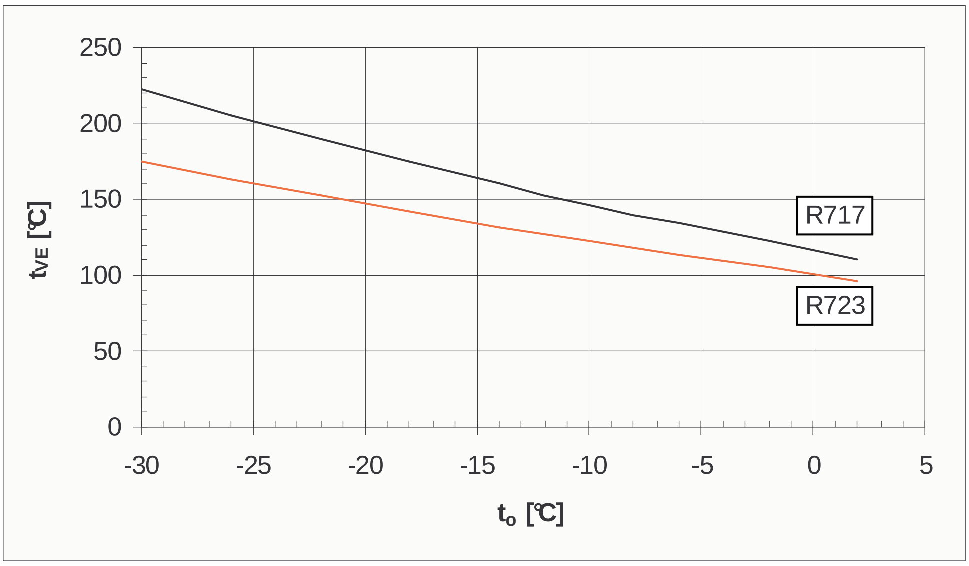


Figure 3.2 compares R717 and R723. The pressure ratio of R723 is a little under that of ammonia. From that also the small improvement of the COP for R723 results. The deciding advantage of R723 is however in the reduction of the high discharge temperatures at ammonia. From picture 3.3 it is clear that the discharge temperature can be lowered by R723 around approx. 20 to 30 K.

Figure 3.3: Comparison discharge temperature of R723 to R717 in dependence of the evaporating temperature



The superheat temperature of the refrigerant in the suction line of the compressor has also an influence on the discharge temperature but also on the COP . Figure 3.4 shows this fact for the brine chiller, figure 3.5 for the one-stage deep freezing at design conditions. Minimum superheat temperatures are to be striven.

Figure 3.4: Discharge temperature and COP for R723 and operating conditions brine chiller in dependence of the superheating temperature

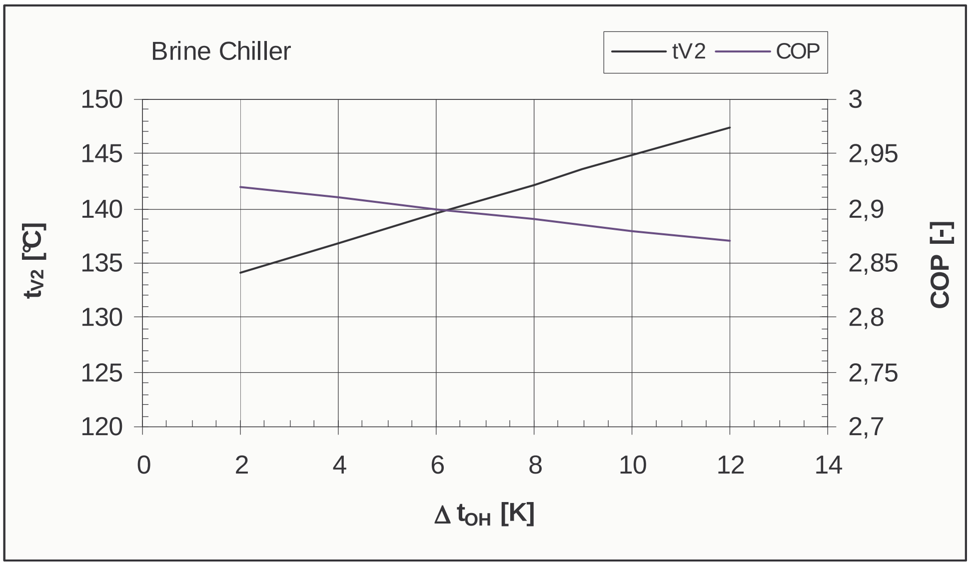
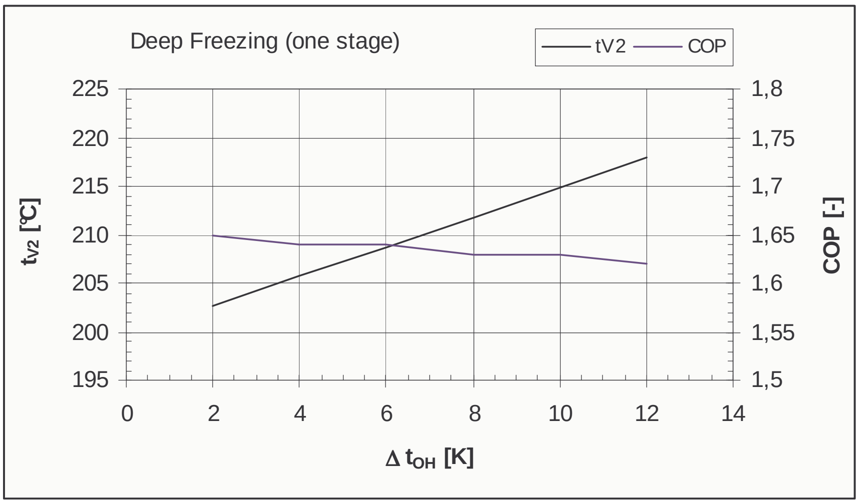
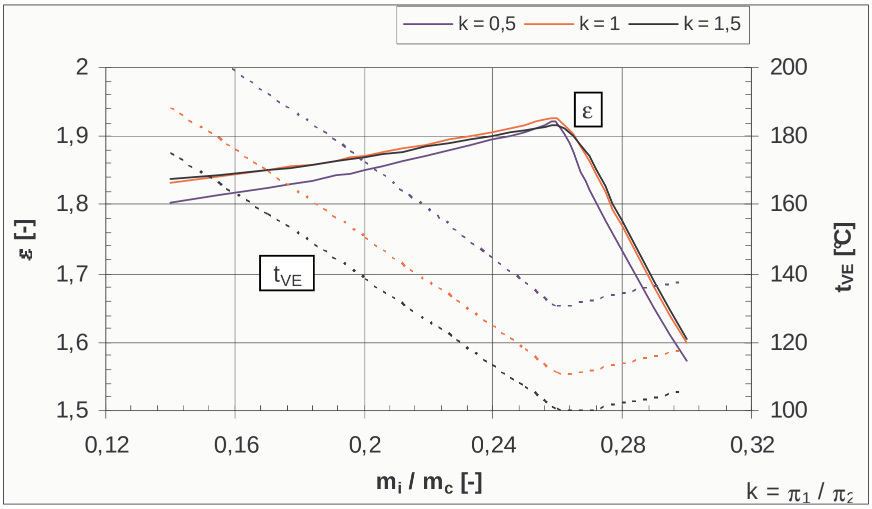


Figure 3.5: Discharge temperature and COP for R723 and operating conditions deep freezing Cell in dependence of the superheating temperature



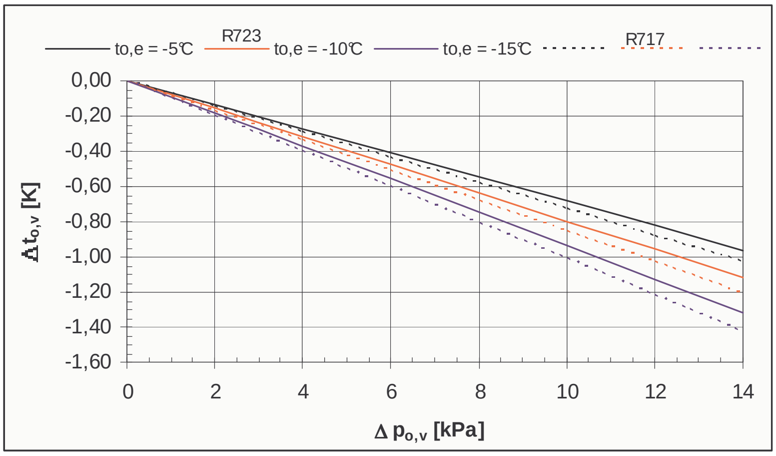
An one-stage deep freezing leads to high discharge temperatures. The compressor outlet temperature is at these conditions approx. 50 K under the discharge temperature. Within the plant measuring of the deep freezingcell the minimal possible evaporating temperature at condensing temperature of 42°C is to be determined that guarantees the admissible maximum compressor outlet temperature still. A two-stage cold generation would have an expenditure for components which would not bring any economic efficiency in comparison with current conventional plants. The expenditure would increase itself furthermore through the necessity of a intermediate cooling still. Figure 3.6 shows the influence of the injection mass flow of refrigerant to the intermediate cooling on the discharge temperature and the COP. An one-stage plant design is striven for.

Figure 3.6: COP and discharge temperature in dependence of injection massflow (economizer) at two-stage compression for the deep freezing cell



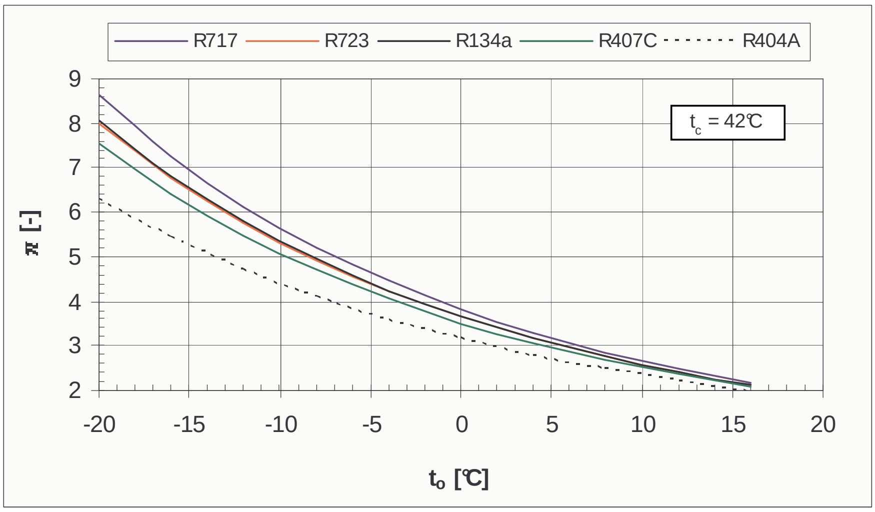
The demo plants are to be configured so that the refrigerant sided losses of pressure are small. Also in this point R723 has an little advantage opposite R717. From picture 3.7 can be taken out that evaporator losses of pressure at R723 lead only to a little smaller reduction of the evaporating temperature.

Figure 3.7: Effect of pressure loss in the evaporator on the reduction of evaporating temperature



Dimethylether (CH3-O-CH3) is not toxic, his boiling point is higher around 7 K than ammonia. Dimethylether is an inflammable medium with an flammability range in air of 3,0 to 18,6 V%. R723 has a part of Dimethylether of 40 M%. This flammability limit is to consider for the installation rooms and to compare with the amount formed out filling amount, mass percent of dimethylether and room volume. In figure 3.8 the pressure ratio which is important for the isentropic efficiency and the delivery rate is compared with different other refrigerants. R404A the is considerably more favorable opposite other refrigerants and therefore R404A are set in deep freezing applications.

Figure 3.8: Comparison of pressure ratio for different refrigerants



4 Technical Documentation of Demo Plants

The demo plants were computed thermodynamically and dimensioned according to the technical rules. The relevant safety requirements were considered and the components checked concerning their refrigerant compatibility.

4.1 Design and Computing

The design and circle computing comply with in point 3 defined and described boundary conditions.

4.1.1 Heat Pump (Chiller) System for milk cooling

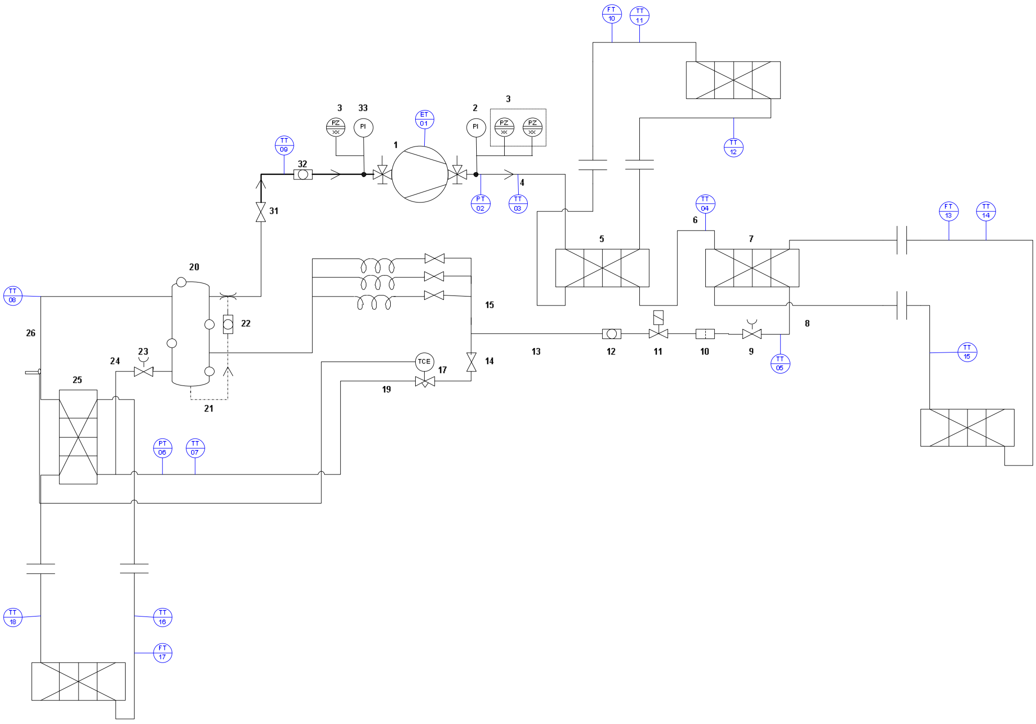
Figure 4.1 shows the calculations of the Heat Pump system at different design conditions with R723 for dimensioning and lay-out of components.

Figure 4.1: Calculations of the Heat Pump System

Dimensioning conditions		-10/42 70 Hz	-7/42 70 Hz	-7/42 50 Hz	5/30 70 Hz	5/30 50 Hz
Swept Volume	m^3/h	36,3	36,3	25,9	36,3	25,9
Superheat	K	1	1	1	1	1
Condensing temperature	℃	+42	+42	+42	+30	+30
Evaporating temperature	℃	-10	-7	-7	-5	-5
Condensing pressure	bar	17,3	17,3	17,3	12,42	12,42
Evaporating pressure	bar	3,254	3,654	3,654	5,65	5,65
Pressure difference	bar	14,046	13,64	13,64	6,77	6,77
Pressure ratio	-	5,32	4,73	4,73	2,19	2,19
Discharge temperature	℃	135,1	128,3	128,3	72,98	72, 98
Isentropic efficiency	-	0,7	0,7	0,7	0,7	0,7
Volumetric efficiency	-	0,7	0,7	0,7	0,7	0,7
Cooling capacity	kW	18,7	20,9	14,9	34,2	24,4
Power consumption	kW	6,46	6,65	4,74	4,83	3,45
COP	-	2,89	3,14	3,14	7,08	7,08
Refrigerant mass flow	kg/s	0,02501	0,02781	0,01982	0,04196	0,02994

The Piping and Instrument diagram with measuring positions is shown in figure 4.2.

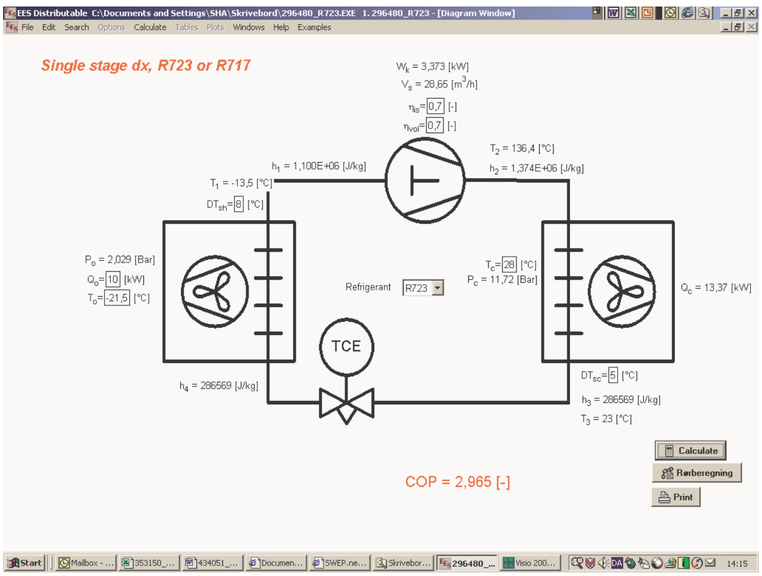
Figure 4.2: Piping and Instrument diagram for Heat Pump System



4.1.2 Chiller System for marine applications

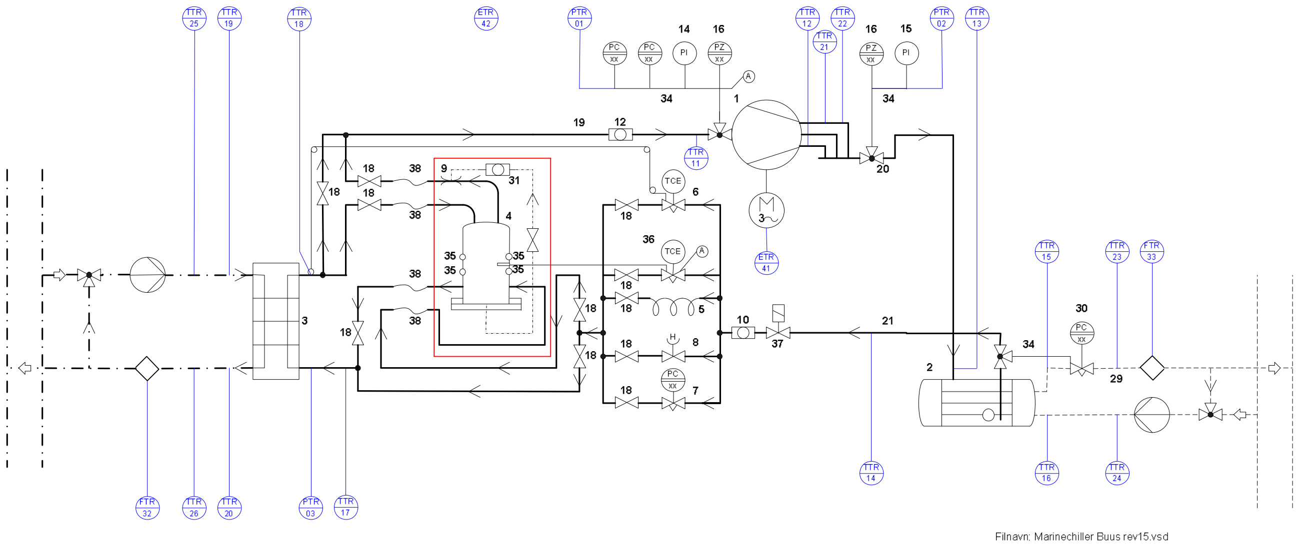
Figure 4.3 shows the calculation for the Chiller System at design condition

Figure 4.3: Calculation of the Chiller System



The Piping and Instrument diagram with measuring positions is shown in figure 4.4.

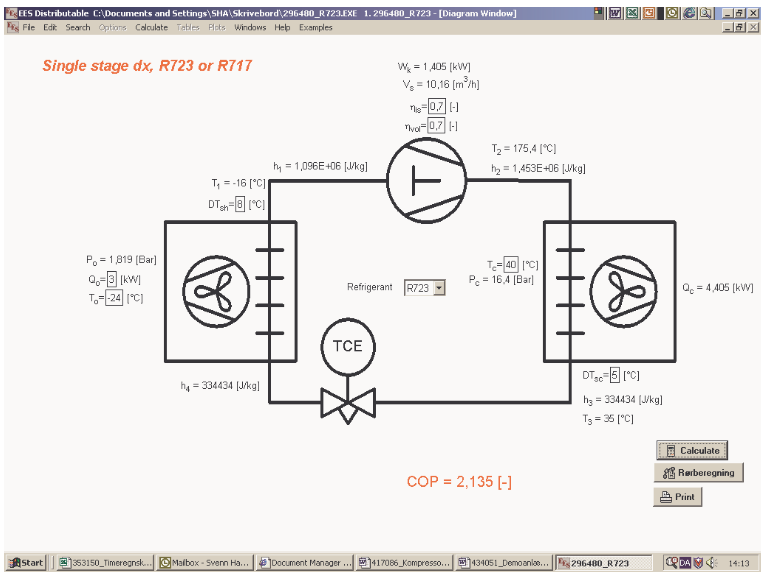
Figure 4.4: Piping and Instrument diagram for Chiller System



4.1.3 Ice Flake Machine

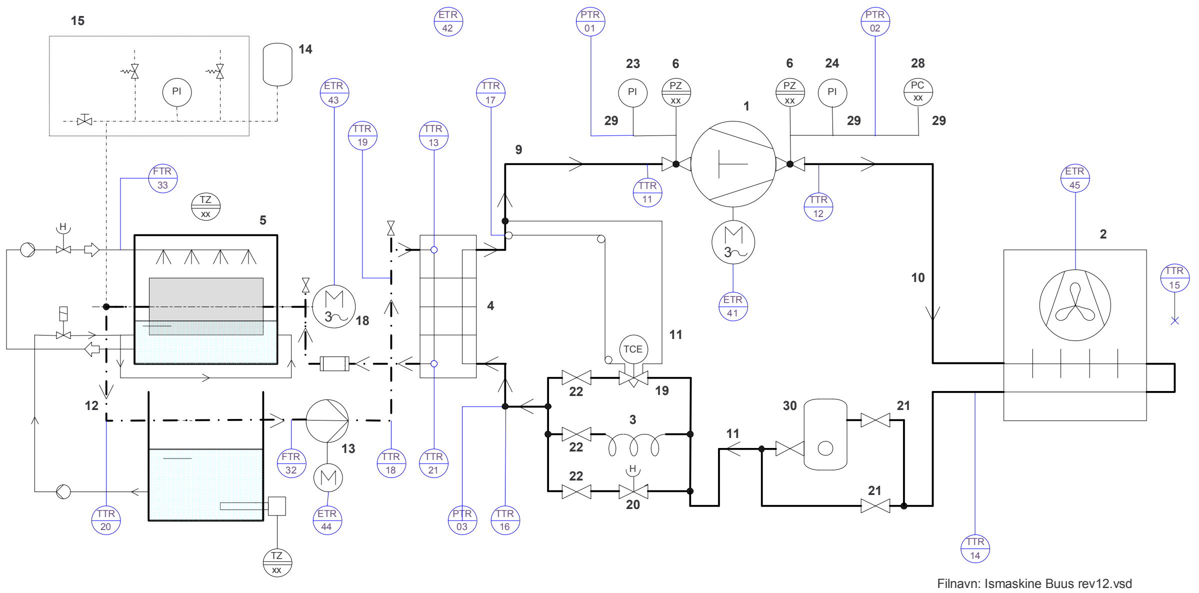
Figure 4.5 shows the calculation of the refrigeration system at design condition

Figure 4.5: Calculation of the Flake Ice Machine



The Piping and Instrument diagram with measuring positions is shown in figure 4.6.

Figure 4.6: Piping and Instrument diagram for Ice Flake Machine



4.1.4 Normal Cooling Cell

Table 4.1 contains the computing printout of the thermodynamic cycle computing. Instructions of manufacturers were considered for the isentropic efficiency and the delivery rate of compressor.

Table 4.1: Computing of cooling circle normal cooling cell

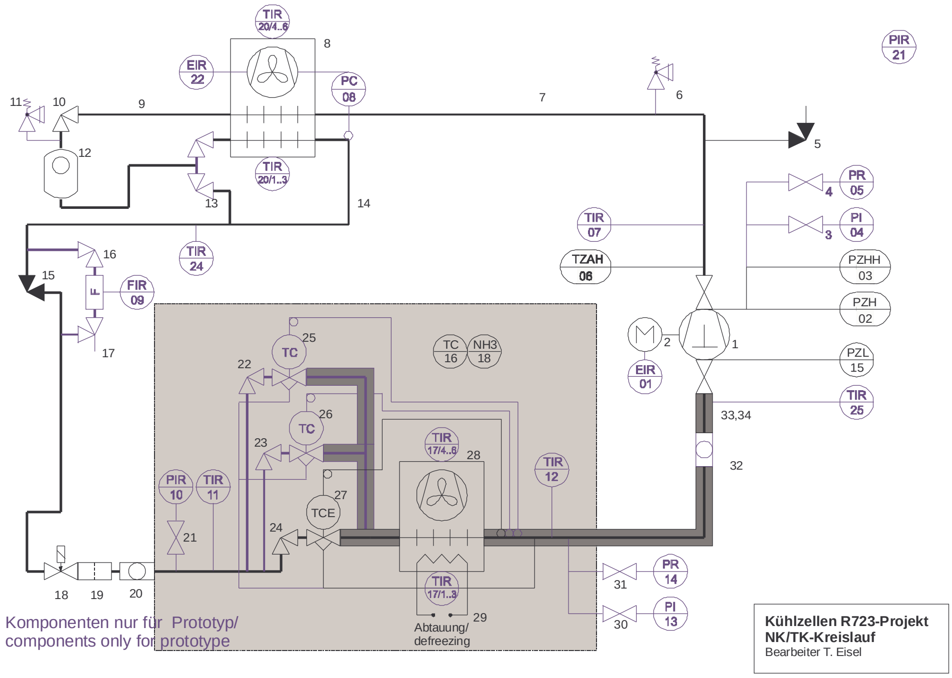
Single Stage Refrigerating Cycle

Refrigerant:		R723		Cycle data		
Evaporating temperature:	t''_0:	-10	°C	p_0 =	325,4	kPa
Superheating, evaporator:	Δt_O:	6	K	p_C =	1730,1	kPa
Superheating, suction line:	Δt_{SL}:	0	K	π =	5,32	-
Condensing temperature:	t''_C:	42	°C	q_O =	757,17	kJ/kg
Subcooling, condenser:	Δt_C:	2	K	w_t =	283,99	kJ/kg
Subcooling, liquid line:	Δt_{LL}:	0	K	ε_o =	2,67	-
Isentropic efficiency:	η_i:	0,65	-	q_{OV}=	2621,1	kJ/m³
Refrigerating capacity:	Q_0:	11,34	kW	m_O =	0,01498	kg/s
				V_O =	15,575	m³/h
				P =	4,253	kW
Displacement:	V_S:	19,47	m³/h	m_O =	0,01498	kg/s
Volumetric efficiency:	η_V:	0,8	-	V_O =	15,576	m³/h
				Q_O =	11,34	kW
				P =	4,25	kW

Cycle point		t (°C)	p (kPa)	v (m³/kg)	h (kJ/kg)	s (kJ/(kgK))	x (-)
Compressor inlet	1	-4,00	325,4	0,288870	1112,17	4,4693	
Compressor outlet, isentropic	2s	106,32	1730,1	0,073901	1296,77	4,4693	
Compressor outlet	2	150,78	1730,1	0,084940	1396,17	4,7168	
Dew point at cond. pressure	3"	42,00	1730,1	0,056277	1151,84	4,0504	
Bubble point at cond. pressure	4'	42,00	1730,1	0,001712	363,36	1,5485	
Condenser outlet	4	40,00	1730,1	0,001701	355,00	1,5224	
Expansion valve inlet	5	40,00	1730,1	0,001701	355,00	1,5224	
Bubble point at evap. pressure	6'	-10,00	325,4	0,001498	164,04	0,8668	
Evaporator inlet	6	-10,00	325,4	0,058558	355,00	1,5924	0,204
Dew point at evap. pressure	7"	-10,00	325,4	0,281440	1100,91	4,4270	
Evaporator outlet	7	-4,00	325,4	0,288870	1112,17	4,4693	

Figure 4.7 shows the scheme inclusive measuring places of refrigerating aggregate of normal cooling cell.

Figure 4.7: Scheme refrigerating aggregate normal cooling cell



4.1.5 Deep Freezing Cell

Figure 4.8 shows the scheme inclusive measuring places of refrigerating aggregate of deep freezing cell and table 4.2 the cycle computing.

Figure 4.8: Scheme refrigerating aggregate deep freezing cell

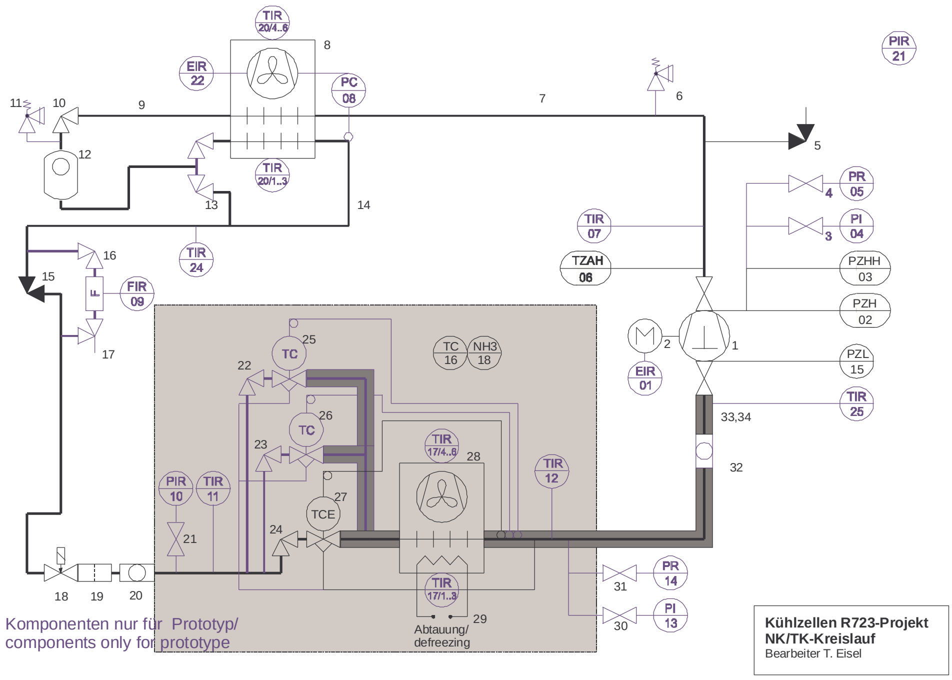


Table 4.2: Computing of cooling circle deep freezing cell

Single Stage Refrigerating Cycle

Refrigerant:		R723		Cycle data		
Evaporating temperature:	t''_0 :	-30	℃	p_0 =	138,6	kPa
Superheating, evaporator:	Δt_O:	6	K	p_C =	1730,1	kPa
Superheating, suction line:	Δt_{SL}:	0	K	π =	12,48	-
Condensing temperature:	t''_C:	42	℃	q_O =	729,15	kJ/kg
Subcooling, condenser:	Δt_C:	2	K	w_t =	445,04	kJ/kg
Subcooling, liquid line:	Δt_{LL}:	0	K	ε_o =	1,64	-
Isentropic efficiency:	η_i:	0,65	-	q_{OV}=	1137,3	kJ/m³
Refrigerating capacity:	Q_0 :	11,63	kW	m_O =	0,01595	kg/s
				V_O =	36,815	m³/h
				P =	7,098	kW
Displacement:	V_S :	46,01	m³/h	m_O =	0,01595	kg/s
Volumetric efficiency:	η_V:	0,8	-	V_O =	36,808	m³/h
				Q_O =	11,63	kW
				P =	7,10	kW

Cycle point		t	p	v	h	s	x
		℃	kPa	m³/kg	kJ/kg	kJ/(kgK)	-
Compressor inlet	1	-24,00	138,6	0,641149	1084,15	4,6626	
Compressor outlet, isentropic	2s	140,68	1730,1	0,082482	1373,43	4,6626	
Compressor outlet	2	208,74	1730,1	0,098683	1529,20	5,0108	
Dew point at cond. pressure	3"	42,00	1730,1	0,056277	1151,84	4,0504	
Bubble point at cond. pressure	4'	42,00	1730,1	0,001712	363,36	1,5485	
Condenser outlet	4	40,00	1730,1	0,001701	355,00	1,5224	
Expansion valve inlet	5	40,00	1730,1	0,001701	355,00	1,5224	
Bubble point at evap. pressure	6'	-30,00	138,6	0,001439	94,62	0,5935	
Evaporator inlet	6	-30,00	138,6	0,167176	355,00	1,6644	0,266
Dew point at evap. pressure	7"	-30,00	138,6	0,624488	1073,48	4,6192	
Evaporator outlet	7	-24,00	138,6	0,641149	1084,15	4,6626	

4.1.6 Brine Chiller

Figure 4.9: Scheme refrigerating aggregate brine chiller

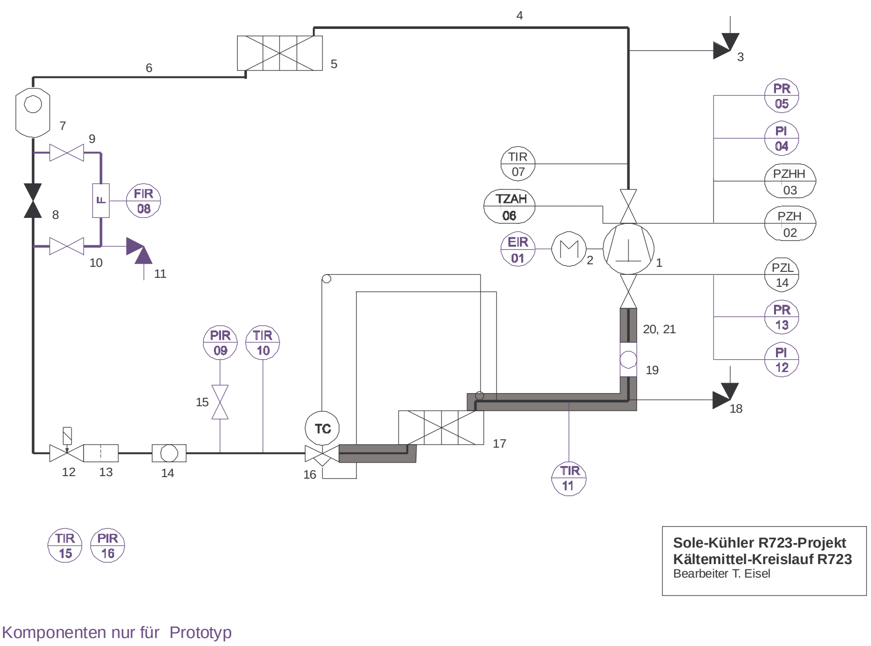


Figure 4.9 shows the scheme including the installed points of measurement of the brine chiller and table 4.3 contains the print of the thermodynamic cycle computing. The isentropic efficiency and dilevery rate of the compressor were chosen according to specifications of the manufacturer.

Table 4.3: Computing of cooling circle brine chiller

Single Stage Refrigerating Cycle

Refrigerant:		R723		Cycle data		
Evaporating temperature:	**t''_0 :**	**-14**	**℃**	**p_0 =**	**277,6**	**kPa**
Superheating, evaporator:	**Δt_O:**	**6**	**K**	**p_C =**	**1391,5**	**kPa**
Superheating, suction line:	**Δt_{SL}:**	**0**	**K**	**π =**	**5,01**	**-**
Condensing temperature:	**t''_C:**	**34**	**℃**	**q_O =**	**784,63**	**kJ/kg**
Subcooling, condenser:	**Δt_C:**	**2**	**K**	**w_t =**	**270,26**	**kJ/kg**
Subcooling, liquid line:	**Δt_{LL}:**	**0**	**K**	**ε_o =**	**2,90**	**-**
Isentropic efficiency:	**η_i:**	**0,65**	-	**q_{OV}=**	**2340,3**	**kJ/m³**
Refrigerating capacity:	**Q_0 :**	**18,43**	**kW**	**m_O =**	**0,02349**	**kg/s**
				V_O =	**28,350**	**m³/h**
				P =	**6,348**	**kW**
Displacement:	**V_S :**	**40,5**	**m³/h**	**m_O =**	**0,02349**	**kg/s**
Volumetric efficiency:	**η_V:**	**0,7**	-	**V_O =**	**28,350**	**m³/h**
				Q_O =	**18,43**	**kW**
				P =	**6,35**	**kW**

Cycle point		**t** ℃	**p** kPa	**v** m³/kg	**h** kJ/kg	**s** kJ/(kgK)	**x** -
Compressor inlet	1	-8,00	277,6	0,335269	1106,80	4,5048	
Compressor outlet, isentropic	2s	96,20	1391,5	0,090356	1282,47	4,5048	
Compressor outlet	2	139,48	1391,5	0,103411	1377,06	4,7469	
Dew point at cond. pressure	3"	34,00	1391,5	0,069983	1146,75	4,1023	
Bubble point at cond. pressure	4'	34,00	1391,5	0,001671	330,24	1,4439	
Condenser outlet	4	32,00	1391,5	0,001661	322,17	1,4180	
Expansion valve inlet	5	32,00	1391,5	0,001661	322,17	1,4180	
Bubble point at evap. pressure	6'	-14,00	277,6	0,001486	149,93	0,8130	
Evaporator inlet	6	-14,00	277,6	0,060705	322,17	1,4776	0,182
Dew point at evap. pressure	7"	-14,00	277,6	0,326653	1095,67	4,4624	
Evaporator outlet	7	-8,00	277,6	0,335269	1106,80	4,5048	

Figue 4.10: Scheme brine circle

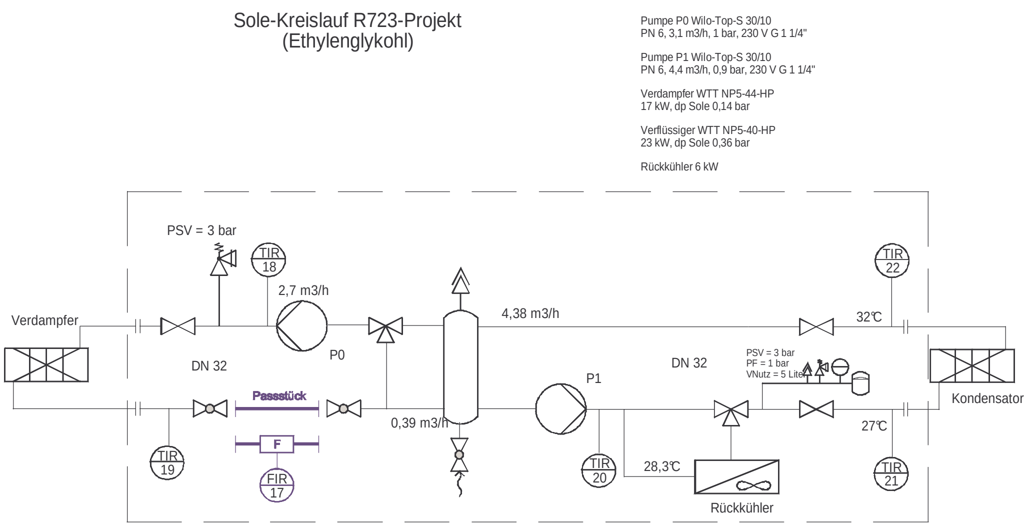


Figure 4.10 shows the brine circuit for the carrying off the condensation and supply of the evaporation heat. Cold requirement is simulated about the brine circuit when about the refrigerating capacity the prevailing part of the condensation heat is cooled return. In the circuit three way valves are installed for the adjusting of the condensation and evaporating temperature. These valves are controlled elektrothermally.

4.2 Choice of Components

In the Midterm-report (February 2004) of the project different refrigerants and different carrying out kinds of the components were compared with each other. There materials, connection technologies, compressors, evaporators, condensers, expansion valves and attachments were examined. On filter dryer, oil separator, oil return were renounced. Filter dryers have hardly any function in NH3-plants due to the molecule size of water and ammonia. From that on filter dryers were renounced. Conditional through a partial solubility of refrigerant and refrigerating oil it was estimated that an oil return is guaranteed also without additional equipment. Consequently oil separator and oil return are omitted too. Table 4.4 contains the basic design of the German demo plants IV, V and VI.

Table 4.4: Choice of components of German demo plants

Plant	**Com-pressor**	**Evapo-rator**	**Conden-sor**	**Tubes**	**Connec-tions**	**Exp.valve Refrigerant**	**Refr.-vessel**
NC-Cell (plant IV)	Frigopol Typ 19-D semiherm.	Rib HEX	Rib HEX with sub-cooling line	CuNi10	braze (LAG 72)	TEV R723	extant
DF-Cell (plant V)	Frigopol Typ 46-D semiherm.	Rib HEX	Rib HEX with sub-cooling line	CuNi10	braze (LAG 72)	TEV R723	extant
Brine Chiller (plant VI)	Bock F4NH3 open	Plate HEX	Plate HEX	St35	weld and Kugelkopf	TEV R723	extant

4.2.1 Heat Pump System

All components in the Heat Pump System are shown in following part list.

Tabel 4.5: Part list for the Heat Pump System

Pos.	Component	Type	Technical spec./ Comments	Manufacturer
1	Compressor	Maneurop SM 110-3 type 3	17 – 18 kW	Maneurop/Danfoss Nordborg
2	Gauge high pressure	Type 80 for R717	Ø80 mm, 25 bar	Tempress
3	Pressostat low pressure	KP1E 060-5300		Danfoss
	Pressostat high pressure	KP7EW 060-5304		Danfoss
4	Discharge pipe	Copper ½” x1 mm	EN 12735-1	
5	Desuperheater	GEA L25-15		DANAVEX
6	Discharge pipe	Copper ½” x1 mm	EN 12735-1	Danfoss
7	Condenser	GEA H25-50-LG-X		DANAVEX
8	Liquid pipe	Copper ½” x1 mm	EN 12735-1	
9	Stop valve	SVA 10 A223		Danfoss
10	Filter			
11	Solenoid valve	EVRE10 018F5251		Danfoss
12	Sight glass	SGN 12S ½”lodde		Danfoss
13	Liquid pipe	Copper ½” x1 mm	EN 12735-1	
14	Stop valve	SVA 10 A223		Danfoss
15	Stop valve	SVA 10 A223		Danfoss
16	Capillary tubes	2,3 mm	0,5 m; 1,0m; 2,0m	
17	Thermostatic ex-pansion valve	TUBE nozzle 6 068UXXXX		Danfoss
19	Liquid pipe	Copper ½” x1 mm	EN 12735-1	
20	Liquid separator			DTI
21	Oil return pipe	Copper ¼”	EN 12735-1	
22	Sight glass	SGN 12S ½”soldering		Danfoss
23	Stop valve	SVA 10 A223		Danfoss
24	Liquid pipe	Copper ½” x1 mm	EN 12735-1	
25	Evaporator	GEA M25-50-LG-X		DANAVEX
26	Suction pipe	Copper 7/8” x1 mm	EN 12735-1	
30	Suction pipe	Copper 7/8” x1 mm	EN 12735-1	
31	Stop valve	SVA 10 A223		Danfoss
32	Sight glass	SGN 12S ½”lodde		Danfoss
33	Gauge low pressure	Type 80 for R717	Ø80 mm, 10 bar	Tempress

The installation of the Heat Pump System are shown on the following figures.

Figure 4.11: Front picture Heat Pump System for milk cooling

4.2.2 Chiller System for marine applications

All components in the Chiller System are shown in following part list. The installation of the system are shown on the following figures (4.12, 4.13).

Figure 4.12: Front picture of Marine Chiller System

Tabel 4.6: Part list for the Chiller System

Pos.	Component	Type	Technical spec./ Comments	Manufacturer
1	Compressor	Hermetic piston MTZ72-6VI	Standard hermetic reciprocating compressor without R723 modification	Maneurop
2	Condenser	Seawater cooled K 203 HB w. 2 pass	12 kW at ΔT 5K	Bitzer
3	Evaporator	AlfaNova NS52 (H2,B2)	10 kW, t_o ÷21,5℃ V_{brine} ÷15/÷18℃ Stainless steel Soldered	Alfa-Laval
4	Multifunction vessel		Flashgas by-pass Liquid separation Suction accumulator Oil separator system	BUUS
5	Capillary tube	ø 3 mm x 1.100 m	Stainless steel	
6	Thermostatic expansion valve	TUBE orifice 6	For R290 ¼" x ½"	Danfoss
7	Electronic expansion valve	AKVA 10-3 incl. coil	Electronic type	Danfoss
8	Manual expansion valve	Metering valve	Stainless steel ¼" Gyrolok Micrometer adjustment	HOKE Inc.
9	Oil return system		Ejector	
10	Sight glass		Industrial type with steel body for welding	Danfoss Industrial Refrigeration
12	Sight glass		Industrial type with steel body for welding	Danfoss Industrial Refrigeration
13	Receiver	FS 56	Volume 5,6 l	Bitzer
14	Low-pressure gauge	Type A10	ø 80 mm ÷1 – 0 – +12 bar without R717 scale	Tempress
15	High-pressure gauge	Type A10	ø 80 mm ÷1 – 0 – +25 bar without R717 scale	Tempress
16	Pressostat	KP1E KP7EW	For high and low pressure	Danfoss
17	Stop valve	SVA 6	Straightway, welding connection, cap	Danfoss Industrial Refrigeration
18	Stop valve	SVA ST 15	Straightway, welding connection, cap	Danfoss Industrial Refrigeration
19	Suction pipe	OD 20 x 1 mm	Acid-proof welded steel pipe	
20	Discharge pipe	OD 12 x 1 mm	Acid-proof welded steel pipe	
21	Liquid pipe	OD 8 x 1 mm	Acid-proof welded steel pipe	
30	Water valve	WVS 32		Danfoss
31	Sight glass			Danfoss Industrial Refrigeration
32	Thermostat		Compressor on/off	
33	Adjusting valve		Adjustment	
34	Small pipes	Plastic hose w. fittings	-40℃/+115℃ MWPR 40 bar	Refflex
t35	Sight glass		Welded in vessel	
36	Level control valve	TEVA 20-3		Danfoss Industrial Refrigeration
37	Solenoid valve	EVRA 3	W. flange set and reel 230V, 50Hz	Danfoss Industrial Refrigeration
38	Flexible hose			
	Frequency converter	AKD 5016	Capacity regulation of compressor 30 – 80 Hz	Danfoss Drives
	Electronic control	EKC 315A	For AKVA valve	Danfoss
	Refrigerant	R723	60% Ammonia 40% Dimethylether	Schick MZ
	Secondary refrigerant	Propylene glycol	44%	
	Oil	Clavus S46	Mineral oil	Shell

Figure 4.13: Front picture of Marine Chiller System

4.2.3 Ice Flake Machine

All components in the Ice Flake Machine are shown in following part list. The installation of the system are shown on the following figures 4.14 and 4.15.

Figure 4.14: Front picture of Ice Flake Machine

Tabel 4.7: Part list for the Ice Flake Machine

Pos.	Component	Type	Technical spec./ Comments	Manufacturer
1	Compressor	LT- 28-4VI	Standard hermetic reciprocating compres-sor without R723modification	Maneurop
2B	Alternative air cooled condenser	SHVN 7/0	Copper pipes	LU-VE Contardo
3	Capillary tube	OD 3 / ID 2.2 mm t 0.4 mm	AlMn1 (HA3103)	Hydro
4	Evaporator	B25x30	Copper soldered plate exchanger	SWEP
5	Ice Flake Machine	BM500UL		Buus
6	Pressostat	KP1E KP7EW	For low and high pres-sure	Danfoss
9	Suction pipe	OD 18 / ID 15 mm t 1.5 mm	AlMn1 (HA3103)	Hydro
10	Discharge pipe	OD 10 / ID 7.5 mm t 1.25 mm	AlMn1 (HA3103)	Hydro
11	Liquid pipe	OD 10 / ID 7.5 mm t 1.25 mm	AlMn1 (HA3103)	Hydro
12	Brine pipe	20mm ABS LT-40	Plastic pipe (PN 10) Glued and screwed	GPA AB
13	Brine pump	UD 0207/ ….	IP55, 250W, 220-240V, 50 Hz	Grundfos
14	Expansion vessel	Suprex N2	2 litre	Kierulff A/S
15	“Safety kit”	Suprex D.I.Y. kit	Manometer, air escape, safety and charge valve	Kierulff A/S
18	Motor	5GU10XKB	Gearhead f. ice drum	Oriental Motor Co LTD
19	Thermostatic expansion valve	TUBE orifice 3	Stainless steel Valve and nozzle	Danfoss
20	Manual expansion valve	Metering valve	Stainless steel ¼ ” Gyrolok Micrometer adjustment	HOKE Inc
21	Ball valve	“7 Series”	Gyrolok® connection for 10 mm pipe	HOKE Inc
22	Ball valve	“7 Series”	Gyrolok® connection for 18 mm pipe	HOKE Inc
23	Low pressure gauge	Type A10	ø 80 mm ÷1 – 0 – +12 bar without R717 scale	Tempress
24	High pressure gauge	Type A10	ø 80 mm ÷1 – 0 – +25 bar without R717 scale	Tempress
28	Pressure transducer and control unit in one	Thyristor type FSX-42S	For condensing pres-sure control	Alco
29	Small pipes	Nylaflow 623 DN2 with fittings	-40℃/+125℃ MWPR 40 bar	Refflex
30	Receiver	RSV 8…	? l volume	Friga Bohn
31	Connections	Gyrolok® Tube Fittings		HOKE Inc
32	Refrigerant	R723	60% Ammonia 40% Dimethyl ether	Schick
33	Secondary refrigerant	48% Propylene glycol	®Antifrogen L	Clariant
34	Oil	Clavus G46	Mineral oil	Shell
35	Electric equipment	BUUS design		BUUS

Picture 4.15: Front picture of Ice Flake Machine

4.2.4 Normal Cooling Cell

For normal and deep freezingcell different solders were examined with regard to the suitability. The silver hard solders L - Ag 40 Sn and L - Ag 55 Sn are unsuitable for the use of R723, L - Ag 2P and L - Ag 18P are unsure. Since the L Ag 85 no more is distributed the solder L Ag 72. Table 4.8 contains only the components for the plant of the normal cooling cell that distinguish from that of the deep freezing cell. All other components of the plants are identical. The plant has a nominal pressure of PN25. As compressor the semihermetical separating hood refrigerant compressor of the Co. Frigopol with internal suction line and external pressure gas line was chosen. Lubricating oil is the PAO-oil SHC 226 of the Co. Shell.

Table 4.8: Components of the refrigerating aggregate normal cooling cell

Pos.	component	typ	technical spezification	remarks	supplier
compressor unit	compressor	19	incl. valves in- and outlet	for R723	Frigopol
compressor unit	motor			for Frigopol compressor type 19	Frigopol
expansion	TEV	TEA 20-3 (068G6004)	tc > -50℃, incl. flange and nozzle device, welding ends ½´´ in-, outlet (da/di = 21,3/16 mm), max. working pressure 19 bar (tested 28,5 bar)	for R717, gray cast iron GGG40.3	Danfoss
expansion	TUAE	TUAE Düse 6			Danfoss

The figures 4.16 and 4.17 contain views of the installed cooling aggregate and the rib evaporator in the cooling cell. Conditional through the installation of the measurement technology especially lines for the stabilization of the mass flow measurement the condenser unit was formed more freely. The air cooled condenser is with a subcooling line furnished (12 of the total 112 tubes; 10,7 %) . The refrigerant vessel is integrated between condenser and subcooling line.

Figure 4.16: View of the refrigerating aggregate cooling cell

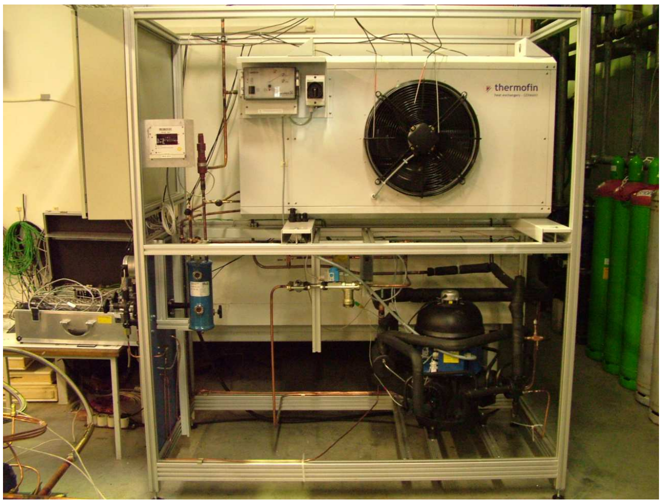


Figure 4.17: View of rib evaporator cooling cell

4.2.5 Deep Freezing Cell

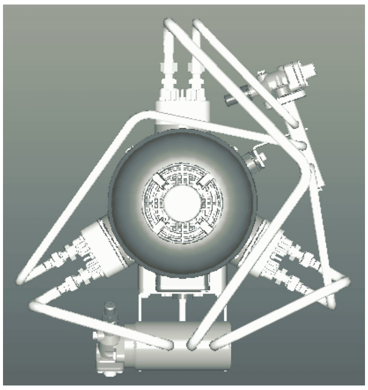
Table 4.9 contains the plant components of deep freezing aggregate. The plant is designed on PN25.

The deep freezing aggregate was built-up up analog the aggregate of the normal cooling except the components compressor and drive motor as well as the expansion valves. Material of the shutting and service valves is GGG40. As compressors the semihermetical separating hood refrigerant compressor of the Co. Frigopol type 46-D with external suction line and external pressure gas line was selected (figure 4.18). The external suction line was built return so that the standard variant with intern suction line could be measured too for comparison and analysis of the effects of the suction line kind.

Table 4.9 Components of the refrigerating aggregate deep freezing cell

component	typ	technical spezification	remarks	supplier
compressor	46	incl. valves in- and outlet	for R723	Frigopol
motor			for Frigopol compressor type 46	Frigopol
PZH & PZHH	KP7ABS	limiting refrigerant pressure of pressure line, variable 8… 32 bar inner and outer manual reset; 6x1x150mm welding end	pressure cut out for increasing pressure, for R723	Danfoss
manual shut-off valve	540 014 000 Service	angle type, femail soldering connection 12 mm, femail soldering connection 6 mm for service (dopple cone)	for R717, copper, soldered with LAG 85	AWA
air cooled fin-on-tube heat condenser incl. fan and motor	TCV 045.1-11-B-N (W5)	in-, outlet for 12x1; vertical orientation pmax = 25 bar; motor (230 V, 50 Hz, 450 W)	oxygen-free copper, not for R717, brazed with L-Ag7281	thermofin
PC	PTKE6-M (Ziehl-Abegg)		speed control incl. pressure transmitter, control depending on condensation pressure	thermofin
liquid receiver incl. sight class	s-2/12-12 (2L)	14x1 mm femail soldering connection in-, outlet, capacity 2 litres (PED class 1)	stainless steel, for R717	ESK Schulze
tube		12x1 mm	high pressure, SF-Cu semi-rigid in tubes	KME
solenoid valve	EVRA 3 (032F3103, 027L1116)	closed, complete valve with soldering flange 16 mm, solenoid 230 V 50 Hz, temperature >= -40 ℃	for R717, carbon steel	Danfoss
sight glass	16	max. 25 bar, solder flange; size 16 mm	for 16x1 mm copper tube, stainless steel	AWA
TEV	TEA 20-5	tc > -50℃, incl. flange and nozzle device, welding ends ½´´ in-, outlet (da/di = 21,3/16 mm), max. working pressure 19 bar (tested 28,5 bar)	for R717, gray cast iron GGG40.3	Danfoss
TUAE	TUAE Düse 6			Danfoss
tube		16x1 mm	SF-Cu in tubes	KME
EEV	AKVA 10-3, Code 068F3283	tc > -50 ℃, welding ends in ½´´ (da/di = 21,3/16mm), out ¾´´ (da/di = 26,9/21,6mm), max. working pressure 28 bar	for R717, stainless steel, leak of valve set is 0,02% of kv-value (22l/h), solenoid valve required?	Danfoss
insulation	AF-H-018	di = 18 mm, thickness of insulation = 13 mm		Armaflex
air admitted fin-on-tube evaporator incl. 2 x fan with motor	TEB 045.1-E-2-7-E	in-, outlet for 28x1; vertical orientation pmax = 25 bar; motor (230 V, 50 Hz, 450 W)	oxygen-free copper, not for R717, brazed with L-Ag7281	thermofin
heater for defreezing			for air admitted fin-on-tube evaporator	thermofin
tube		28 x1 mm	low pressure, SF-Cu semi-rigid in tubes	KME
insulation	AF-H-028	di = 28 mm, thickness of insulation = 13,5 mm		Armaflex
sight glass	28	max. 25 bar, solder flange; size 28 mm	for 28x1,5 mm copper tube; stainless steel	AWA

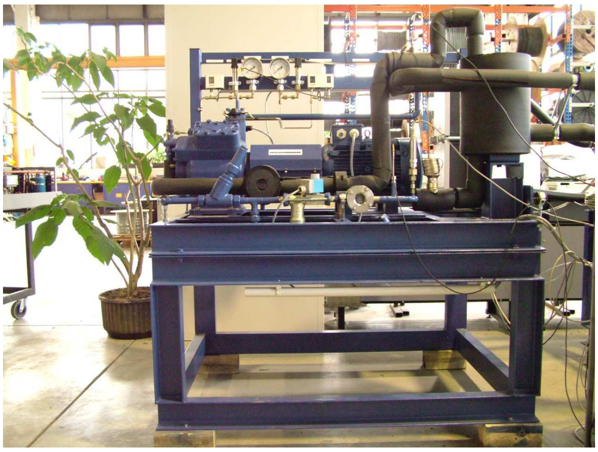
Figure 4.18: View modified separating hood refrigerant compressor with external pressure and suction gas line



4.2.6 Brine Chiller

Table 4.10 contains the main components of the brine chiller. Figure 4.19 shows views of the carried out plant. An open compressor of the Co. Bock with additional oil control rings for reduced oil shot was selected. The pistons are suction gas cooled. The working method occurs according to the countercurrent principle. A great safety is assigned to the compressor against liquid strokes. In the compressor a pressure-feed lubricating system occurs through a sense of rotation independent cogwheel oil pump. As lubricating oil the Fuchs standard oil Reniso KC 68 (2,6 l) was selected. This lubricating oil is a mineral oil and was tested already in connection with the refrigerant R723 however up to now not in plants with plate evaporators.

Figure 4.19: View brine chiller



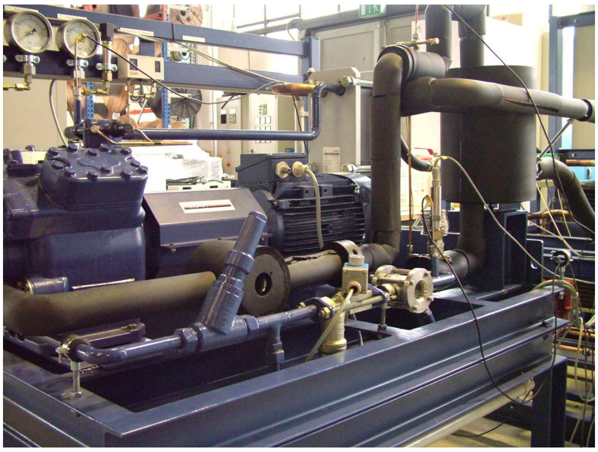


Table 4.10: Components of brine chiller

component	typ	technical spezification	remarks	supplier
compressor	FDK4 NH3	open type, incl. shutt-off valves, inlet da/di = 38/32 mm, outlet da/di = 30/24 mm welding ends	for R717	Bock
motor		7,5 kW	for Bock compressor FDK4 NH3	Bock
TZAH		thermostat with Pt 100, max. temperature 140 °C	temperature limiting device, switch of compressor	Bock
PZH & PZHH	KP7ABS	limiting refrigerant pressure of pressure line, variable 8… 32 bar inner and outer manual reset; 6x1x150mm welding end	pressure cut out for increasing pressure, for R723	Danfoss
tube	PR 06-1 VZ	6x1 mm, 1 m	measurement, carbon steel with zinc coat	Hansa Flex
tube	244821, DN 15, 1/2"	21,3x2 mm	liquid line, carbon steel black DIN/EN 2448	Sächsische Haustechnik
tube	244830	30x2,6 mm, 1 m	high pressure, carbon steel black DIN/EN 2448	Sächsische Haustechnik
plate heat condenser	NP5-30-HP	refrigerant side: welding connection inlet da/di = 30/25,4 mm outlet da/di = 33/28,7 mm; water side: male threat connection G 3/4" in-, outlet	nickel soldered stainless steel plates with pressure frame, required > 27 kW, (tc = 34 °C, inlet = 140 °C)	WTT Wilchwitz
liquid receiver incl. sight class		21,3x2 mm welding ends in-/outlet, capacity 2 litres (PED class 1)	carbon steel	Metall Halle Nord
manual shut-off valve	SVA-ST 15D 223 DIN	straight type, DN 15 (21,3x2,3 mm) welding ends DIN 2448, without service connection	for R717, carbon steel St 35N (17173)	Danfoss
sight glass	T111 DN 15	butt welding ends DIN 3239, da/di = 22/17 mm, allowed pressure 25 bar	stainless steel, for R717	HERL
solenoid valve	EVRA 3 (032F3103, 027N1115)	closed, complete valve with welding flange 1/2"	for R717	Danfoss
TEV	TEA 20-8 (068G6004)	tc > -50°C, incl. flange and nozzle device, welding ends ½´´ in-, outlet (da/di = 21,3/16 mm), max. working pressure 19 bar (tested 28,5 bar)	for R717, gray cast iron GGG40.3	Danfoss
insulation	AF-F-022	di = 22 mm, thickness of insulation = 10 mm		Armaflex
plate heat evaporator	NP5-44-HP	refrigerant side: welding connection in-, outlet da/di = 33/28,7 mm; brine side: male threat connection G 1" in-, outlet	nickel soldered stainless steel plates with pressure frame, required > 20 kW at t0= -14 °C, xin = 0,18, Dt0 = 6K	WTT Wilchwitz
tube	244833, DN 25	33,7x2,6mm, 1m	low pressure, carbon steel black DIN/EN 2448	Sächsische Haustechnik
insulation	AF-F-035	di = 35 mm, thickness of insulation = 11 mm		Armaflex
sight glass	T111 DN 25	butt welding ends DIN 3239, da/di = 34/28 mm, allowed pressure 25 bar	stainless steel, for R717	HERL

5 Measurement of Demo Plants and Evaluation

The plants were connected at cold consumer and equipped with measurement technology after build-up of the demo plants . For the plants with their experimental character no practic places and external operators could be found.

From that their linking to the cooling cell of the ILK was chosen as an experimental and measurement place for the cool cell aggregates. This had the advantage, that a duty of cold supply did not exist and modifications could be carried out rapid . The measurement of the brine chiller occurred on an experimental place in the installation hall of the Co. compact under simulation of a cold requirement.

The demo plants in Denmark were built by the enterprises and measured at the DTI on its test field.

The next points contain the measurement results.

5.1 Heat Pump System

Table 5.1 shows the measuring equipment for the test program for the Heat Pump System. All data were measured and logged with a Hewlett Pachard 3497A logger combined with DTI measuring software TI-DOP version 2.42 and 2.43.

Table 5.1: Measuring equipment for Heat Pump System

No.		Measurement of ...	Measuring equipment
01	ET	Power consumption compressor	Power meter
02	PT	Pressure of refrigerant after compressor	Pressure transducer
03	TT	Temperature of discharge pipe	Temperature transmitter
04	TT	Temperature of refrigerant after desuperheater	Temperature transmitter
05	TT	Temperature of refrigerant after condenser	Temperature transmitter
06	PT	Evaporating pressure	Pressure transmitter
07	TT	Temperature before evaporator	Temperature transmitter
08	TT	Temperature after evaporator	Temperature transmitter
09	TT	Temperature before compressor	Temperature transmitter
10	FT	Flow of hot water	Flow meter
11	TT	Temperature of hot water inlet	Temperature transmitter
12	TT	Temperature of hot water return	Temperature transmitter
13	FT	Flow of hot water low temperature	Flow meter
14	TT	Temperature of hot water inlet	Temperature transmitter
15	TT	Temperature of hot water return	Temperature transmitter
16	TT	Temperature of cooling water inlet	Temperature transmitter
17	FT	Flow transmitter	Flow meter
18	TT	Temperature of cooling water return	Temperature transmitter

Test programme

Main objectives of the test:

1: To measure efficiency during normal conditions so it can be compared with traditional systems.

2: To test if there is an improvement during operation with flooded evaporator and to test how it works.

3: To test the influence of the system during different operating conditions and at different loads.

4: To carry out an investigation of the components, where selected components are cut open and analysed for failure reasons and possible influences from the refrigerant.

Content of the test:

0: Running in

0.1: The plant was evacuated

0.2: After evacuation the plant was charged with R723. Charging was carried out at 0° degrees inlet temperature on the water side, i.e. app. -5℃ evaporation temperature and 30℃ condensation temperature. During charging the system was in operation with thermostatic expansion valve. The compressor operated at 60 Hz.

1: Efficiency measurement with thermal expansion valve

1.1: Inlet temperature on the brine was set at -5°C. The condensation temperature was kept at 42℃. The frequency on the compressor was set at 60 Hz. Operation with thermostatic expansion valve.

1.2: Inlet temperature was kept constant at -5℃. The condensing temperature was kept at 42℃. The frequency of the compressor was set at 70 Hz. Operation with thermostatic expansion valve.

2: Improvement with flooded evaporator

2.1: Inlet temperature on the brine was set at -5°C. The condensation temperature was kept at 42℃. The frequency on the compressor was set at 60 Hz. Operation with capillary tube of 2 meter. It was checked how the flooded evaporator worked.

2.2: Inlet temperature on the brine was set at -5°C. The condensation temperature was kept at 42℃. The frequency on the compressor was set at 60 Hz. Operation with capillary tube of 1 meter. It was checked how the flooded evaporator worked.

2.3: Inlet temperature on the brine was set at -5°C. The condensation temperature was kept at 42℃. The frequency on the compressor was set at 70 Hz. Operation with capillary tube of 1 meter. It was checked how the flooded evaporator worked.

2.4: Inlet temperature on the brine was set at -5°C. The condensation temperature was kept at 42℃. The frequency on the compressor was set at 50 Hz. Operation with capillary tube of 2 meter. It was checked how the flooded evaporator worked.

3: Different loads on the heat pump

3.1: Inlet temperature on the brine was set at -5°C. The condensation temperature was kept at 35℃. The frequency on the compressor was set at 50 Hz. Operation with capillary tube of 0,5 - 1 meter. It was checked how the flooded evaporator worked.

3.2: Inlet temperature on the brine was set at -5°C. The condensation temperature wad kept at 35℃. The frequency on the compressor was set at 60 Hz. Operation with capillary tube 0,5 - 1 meter. It was checked how the flooded evaporator worked.

3.3: Inlet temperature on the brine was set at -5°C. The condensation temperature was kept at 35℃. The frequency on the compressor was set at 70 Hz. Operation with capillary tube of 0,5 - 1 meter. It was checked how the flooded evaporator worked.

Table 5.2: Overview of operating parameters during tests with Heat Pump System

Parameter										
Test no.:	0	1.1	1.2	2.1	2.2	2.3	2.4	3.1	3.2	3.3
Objective	Charge	Adjusting valve	Full-load valve	Capillary tube	Capillary tube	Capillary tube	Capillary tube	Capacity	Capacity	Capacity
Test conditions	0/30	-5/42	-5/42	-5/42	-5/42	-5/42	-5/42	-5/35	-5/35	-5/35
Inlet temperature brine	0	-5	-5	-5	-5	-5	-5	-5	-5	-5
Inlet temperature hot side	30	42	42	42	42	42	42	35	35	35
Evaporation tempera-ture	-5	-10	-10	-7	-7	-7	-7	-7	-7	-7
Condenation tem-perature	30	42	42	42	42	42	42	35	35	35
Frequency	60	60	70	60	60	70	50	50	60	70
Calculated cooling performance	29	16	18,7	17,9	17,9	20,9	14,9	15,5	18,6	21,8
Calculated condensa-tion performance	33	21,3	25,3	23,6	23,6	27,5	19,6	19,6	23,5	27,6
Flow brine	5000	3400	3800	3800	3800	4300	3000	3400	3800	4700
Flow hot side	3000	2700	3200	3200	3200	3200	2500	2500	2800	3200

Table 5.3: Test results with hermetic scroll compressor in Heat Pump System

Test no.		1	2	4	5
Tests conditions		10/40	-5/30	-5/35	-5/35
Expansion valve or length capil-lary tube	[m]	TEV	TEV	1	0,5
Frequency	[Hz]	50,0	50,0	60,0	60,0
Brine outlet temp. Set point	[℃]	10,0	-5,0	-5,0	-5,0
Brine outlet temp.	[℃]	9,1	-6,0	-5,4	-4,0
Brine inlet temp.	[℃]	14,0	-3,3	-4,9	-1,9
Evaporation temp.	[℃]	1,4	-10,1	-11,0	-5,3
Superheating temp.	[℃]	10,3	-2,0	-3,8	1,9
Superheat ΔK	[K]	8,9	8,1	7,2	7,2
Compressor outlet temp.	[℃]	127,3	131,3	139,4	111,8
Condensing temp.	[℃]	38,4	31,1	32,2	34,0
Warm outlet temp. Set.	[℃]	40,0	30,0	35,0	35,0
Warm outlet temp.	[℃]	39,5	31,8	33,6	35,1
Warm inlet temp.	[℃]	29,9	23,0	23,8	23,5
Subcooling temp.	[℃]	29,2	22,9	22,4	23,6
Cooling capacity	[kW]	13,8	11,7	1,3	10,2
Heating capacity	[kW]	20,0	18,3	20,4	24,1
Power input	[kW]	3,0	3,1	3,0	3,2
COP	[-]	6,6	5,9	18,9	35,7
EER	[-]	4,6	3,8	2,6	19,9
Isentropic efficiency	[-]	0,7	0,7	0,7	0,7

Table 5.4: Test results with semihermetic reciprocating compressor in Heat Pump System

Test		1	2	3	4	5	6	7	8	9	10	11	12	13
Test no.			1.1	2.1	2.2	2.4						3.2	3.2	
Tests conditions		0/32	-5/42	-5/42	-5/42	-5/42	-10/42	-10/42	-10/50	-10/50	-10/50	-5/35	-5/35	-5/42
Expansion valve or length capillary tube	[m]	TEV	TEV	2	1	1	1	1	1	0,5	1	2	0,5	0,5
Frequency	[Hz]	70	70	70	70	60	50	70	70	70	70	70	70	70
Brine outlet temparature set point	[℃]	0	-5	-5	-5	-5	-10	-10	-10	-10	-10	-5	-5	-5
Brine outlet temperature	[℃]	-0,5	-5,6	-5,6	-5,8	-4,9	-10,1	-9,5	-9,8	-9,8	-9,8	-5,1	-5,1	-4,7
Brine inlet temperature	[℃]	3,7	-1,4	-1,4	-1,3	-0, 9	-6,4	-6,2	-6,3	-6,4	-6,3	-0,2	-0,2	-0,1
Evaporating temperature	[℃]	-3,2	-7,3	-7,8	-6,5	-5,5	-10,4	-9,7	-10,1	-10,0	-10,0	-6,5	-6,6	-6,1
Superheat temperature	[℃]	3,8	-3,2	0,6	-4,6	-3,8	- 7,6	-7,0	-8,1	-8,4	-8,5	-5,3	-5,4	-5,0
Superheat Δt	[K]	7,0	4,1	8,4	1,9	1,8	2,8	2,6	2,0	1,6	1,5	1,2	1,1	1,1
Discharge temperature.	[℃]	83,4	103,5	106,5	97,9	93,3	102,4	98,2	108,3	109,1	111,8	80,8	80,6	88,5
Condensation temperature	[℃]	32,0	41,9	42,4	41,4	40,9	40,4	35,8	45,3	46,1	47,0	34,3	34,2	40,2
Warm outlet temperature set point	[℃]	32,0	42,0	42,0	42,0	42,0	42,0	42,0	50,0	50,0	50,0	35,0	35,0	42,0
Warm outlet temperature	[℃]	32,2	42,3	42,2	42,3	41,6	41,3	40,7	46,2	47,0	47,9	35,1	35,1	41,1
Warm inlet temperature	[℃]	24,2	35,9	36,1	35,8	35,8	35,9	35,8	41,1	42,0	42,9	28,3	28,3	34,7
Subcooling temperature	[℃]	23,8	33,7	32,7	34,2	35,2	35,2	35,8	39,3	39,7	40,1	28,4	28,4	34,3
Cooling capacity	[kW]	23,5	17,2	16,7	17,9	16,1	14,6	13,2	13,8	13,5	13,7	19,8	19,8	18,6
Heating capacity	[kW]	27,9	22,0	21,5	22,5	20,1	18,8	16,8	17,9	17,3	17,9	23,6	23,4	22,4
Power input	[kW]	6,5	7,4	7,3	7,4	6,0	6,7	5,6	7,2	7,3	7,3	6,6	6,6	7,4
COP	[-]	4,3	3,0	5,6	3,0	3,3	2,8	3,0	2,5	2,4	2,4	3,6	3,5	3,0
EER	[-]	3,6	2,3	2,3	2,4	2,7	2,2	2,3	1,9	1,9	1,9	3,0	3,0	2,5
Isentropic efficiency	[-]	0,5	0,5	0,5	0,5	0,5	0,5	0,5	0,5	0,5	0,5	0,5	0,5	0,5

Figure 5.1: Heating capacity and COP, with 1 meter capillary tube and the compressor adjusted to 70 Hz. At evaporating temperature -10 ℃ and -5 ℃.

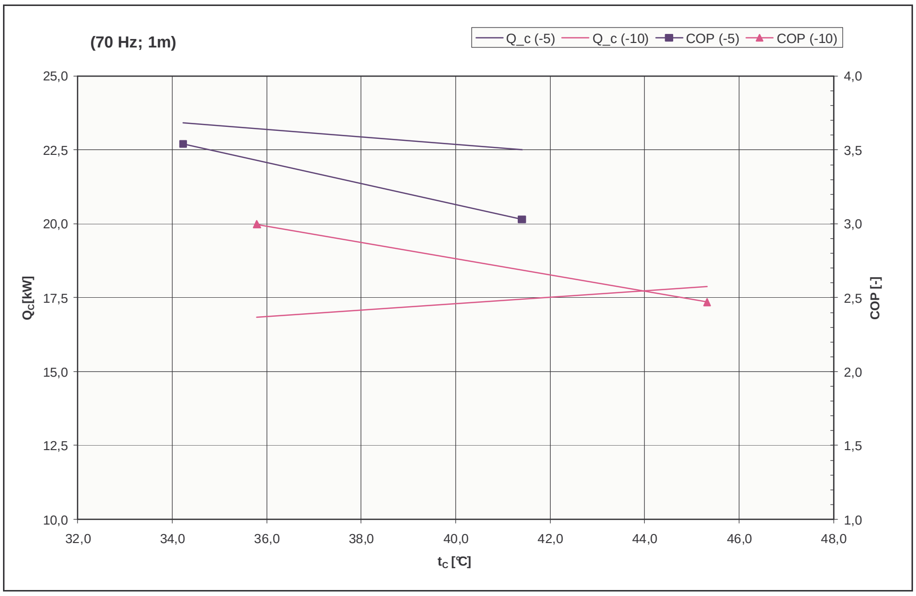


Figure 5.2: Evaporating temperature, superheat temperature and discharge temperature with 1 meter capillary tube and the compressor adjusted to 70 Hz. At evaporating temperature -10 ℃ and -5 ℃.

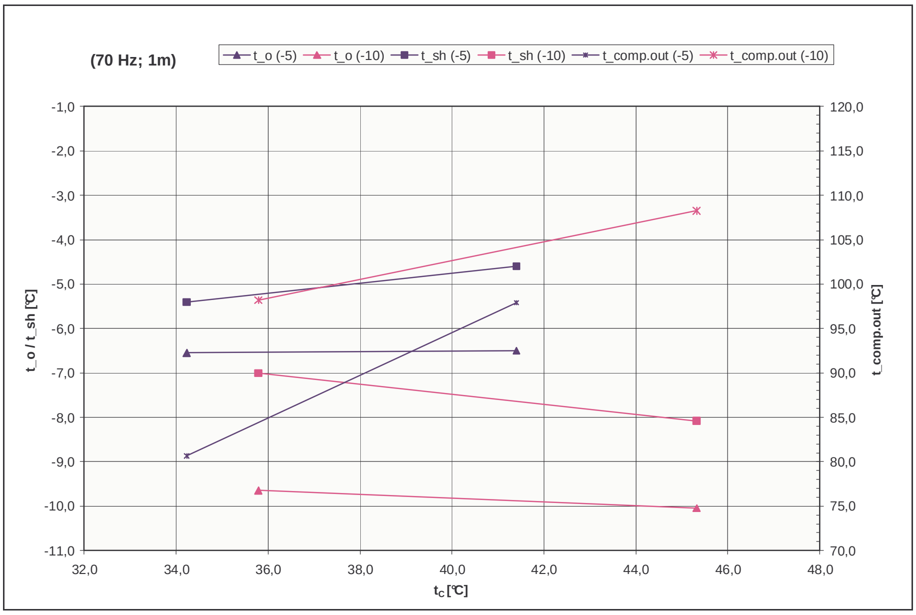


Figure 5.3: Superheat at compressor speed 70 Hz at varying operating conditions and different expansion devices; capillary tubes of 2,0 m, 1,0 m, 0,5 m and thermostatic expansion valve.

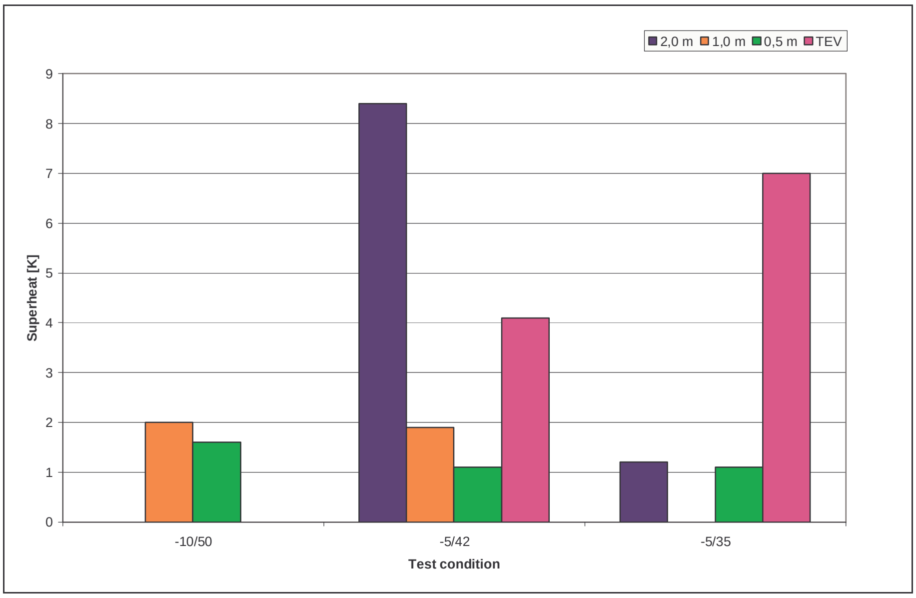


Figure 5.4: Inlet temperature difference ΔT_{in} at compressor speed 70 Hz at varying operating conditions and different expansion devices; capillary tubes of 2,0 m, 1,0 m, 0,5 m and thermostatic expansion valve.

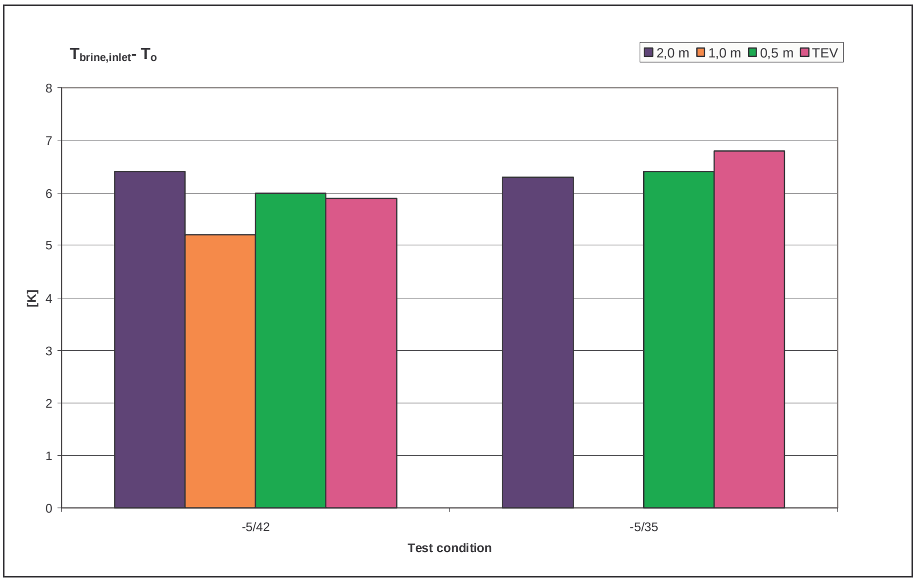


Figure 5.5: Outlet temperature difference ΔT_{out} at compressor speed 70 Hz at varying operating conditions and different expansion devices; capillary tubes of 2,0 m, 1,0 m, 0,5 m and thermostatic expansion valve.

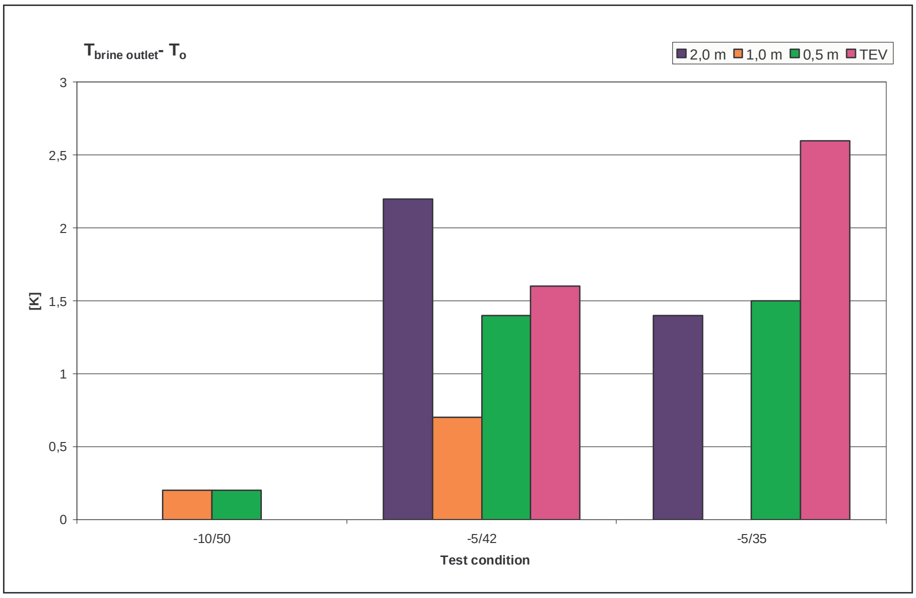


Figure 5.6: Temperatures measured during tests with thermostatic expansion valve.

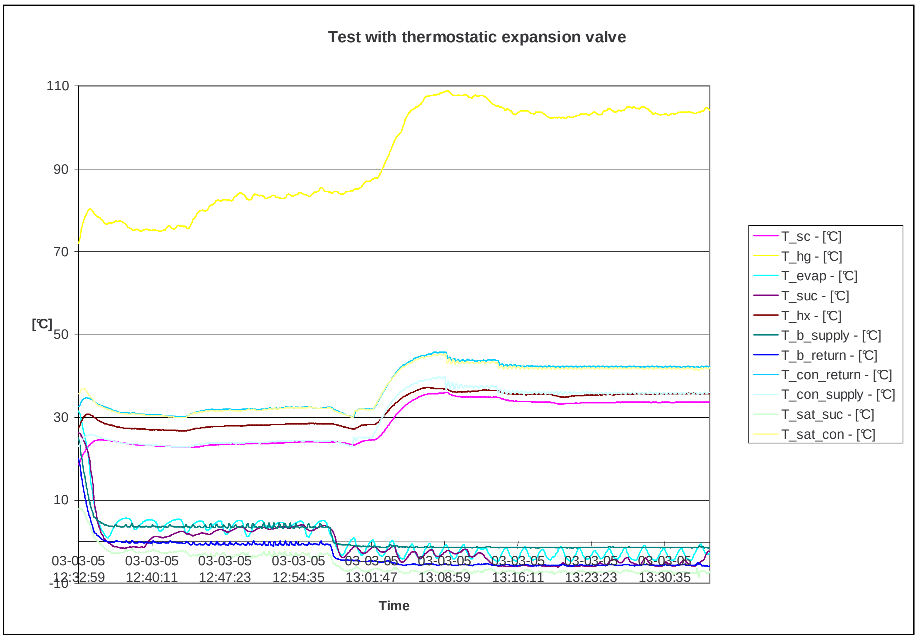
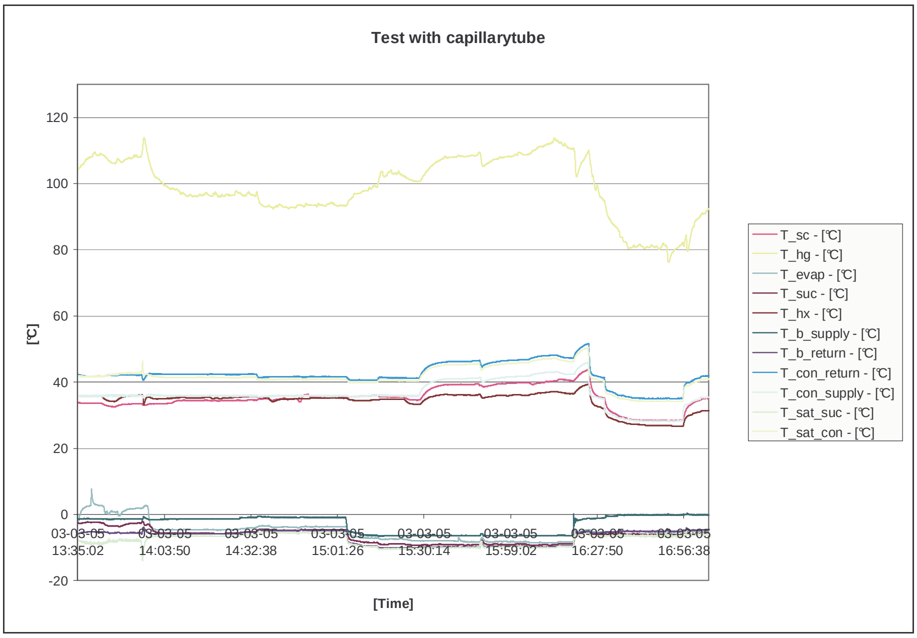


Figure 5.7: Temperatures measured during tests with capillary tubes.



Main results from tests with hermetic scroll compressor

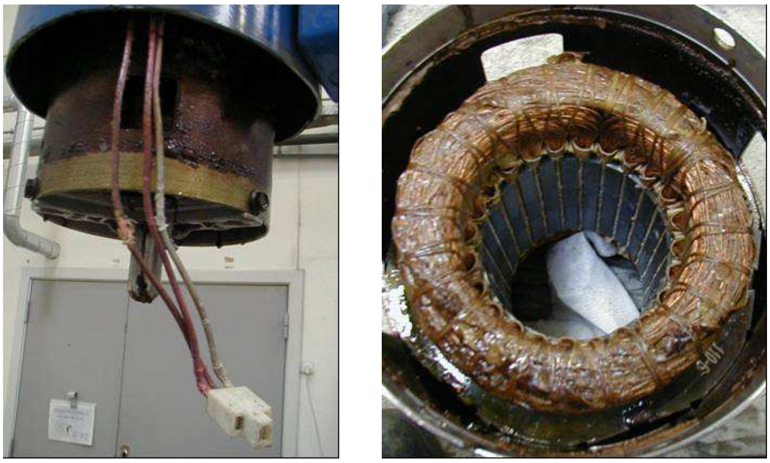
- The plant was charged with 2,5 kg R723.
- Liquid refrigerant ran through the oil return pipe to the compressor and led to frost on the outside of the compressor at start-up, and oil foamed in the sight glass of the compressor.
- The thermostatic expansion valve was adjusted to average superheat of 8,5 K with considerable fluctuations.
- The liquid level in the liquid separator was kept constant with 1 and 2 meter capillary tube.
- The hermetic scroll compressor only operated 5 hours. After 1 day with R723 there was a short circuit inside the compressor.

Main results from tests with semihermetic reciprocating compressor

- After breakdown the compressor was changed to a a semihermetic reciprocating compressor Bock AM 3/233-4 which is suitable for ammonia.
- The oil return pipe was mounted on a valve and could be shut off.
- The plant was charged with 3,2 kg R723.
- The oil return pipe was blocked and therefore the oil level in the oil sight glass remained stable with no signs of foaming.
- The thermostatic expansion valve adjusted well with an average superheating of 6 K and with fluctuations of +/- 1 K.
- With 2 meter capillary tube the liquid level in the liquid separator was low but the conditions stable. The evaporator temperature was rather low which means that the liquid was upset in the condenser and the supply of refrigerant to the evaporator was too low.
- With 1,0 meter capillary tube the liquid level was stable in the liquid separator in all tested points. No pulsations appeared.
- With 0,5 meter capillary tube there were pulsations in the liquid supply to the gas separator at -5/42 and 50 Hz. That means that the condenser was upset with liquid and afterwards it was emptied. The gas that appeared in the capillary tube when the condenser was emptied gave a considerable pressure loss in the tube and therefore the liquid in the condenser was upset and the pulsatory operation appeared.
- The right length was 1,0 meter. Compared to the thermostatic expansion valve the evaporating temperature was increased by 1 K and superheat was lowered 1 K.

- The applied construction of the liquid separator with self circulation above the level of the evaporator will make the evaporator 100% flooded and there will always be minimum superheat. The separator worked according to intention, there was good rotation of the liquid and the liquid level did not rise above half of the height of the vessel. However, the liquid separation was not optimized. It can be concluded that there must have been liquid dropslets in the suction gas because the low discharge temperature corresponds to a calculated isentropic efficiency of 1.0. A baffle plate between the liquid and gas side would optimise the gas separation and in addition there should be a demister in the outlet to the suction gas pipe.
- The oil separation did not work according to intention due to good oil solubility and refrigerant liquid was led back in the oil return pipe. Another solution has to be found.
- The evaporator was not been loaded 100% during the tests compared to the systems design conditions because the reciprocating compressor used only had a swept volume at 70 Hz of 21% lower that the basic dimensioned displacement. The principle of self circulation worked well.
- The compressor operated with an isentropic efficiency from 0.49 to 0.54 - which is low.
- The discharge temperature was low due to refrigerant liquid drops in the suction gas.
- The oil level was stable and there were no signs of foaming.

Figure 5.8: Pictures from investigation of the damaged scroll compressor



- The insulation of the inlet pipes was dissolved by the refrigerant. Dissolved varnish collected at the bottom of the stator windings. The plastic around the stator windings was dissolved.

Figure 5.9: Varnish and plastic from windings deposited at the bottom of the compressor

5.2 Chiller System for marine applications

Table 5.5 shows the measuring equipment for the test programme for the Chiller System. All data were measured and logged with a Hewlett Pachard 3497A logger combined with DTI measuring software TI-DOP version 2.42 and 2.43.

Table 5.5: Measuring equipment for Chiller System

No.		Measurement of ...	Measuring equipment
1	PTR	Pressure of refrigerant before compressor	Pressure transducer
2	PTR	Pressure of refrigerant after compressor	Pressure transducer
3	PTR	Pressure of refrigerant before compressor	Pressure transducer
11	TTR	Temperature of refrigerant before compressor	Temperature sensor
12	TTR	Temperature of refrigerant after compressor (Cyl. 1)	Temperature sensor
13	TTR	Temperature of refrigerant before condenser	Temperature sensor
14	TTR	Temperature of refrigerant liquid after condenser	Temperature sensor
15	TTR	Temperature of water supply to condenser	Temperature sensor
16	TTR	Temperature of water outlet after condenser	Temperature sensor
17	TTR	Temperature of refrigerant before evaporator	Temperature sensor
18	TTR	Temperature of refrigerant after evaporator	Temperature sensor
19	TTR	Temperature of brine before evaporator	Temperature sensor
20	TTR	Temperature of brine after evaporator	Temperature sensor
21	TTR	Temperature of refrigerant after compressor (Cyl. 2)	Temperature sensor
22	TTR	Temperature of refrigerant after compressor (Cyl. 3)	Temperature sensor
23	TTR	Temperature of water inlet to condenser (~TTR15)	Temperature sensor
24	TTR	Temperature of water outlet after condenser (~TTR16)	Temperature sensor
25	TTR	Temperature of brine before evaporator (~TTR19)	Temperature sensor
26	TTR	Temperature of brine after evaporator (~TTR20)	Temperature sensor
32	FTR	Flow of brine to evaporator	Flow meter
33	FTR	Flow of water to condenser	Water meter
42	ETR	Electric power consumption of all electric consumers, i.e. compressor, pumps, control, etc.	Electricity effect meter (3 phase)

Test programme

Main parts of the test were:

1. Mounting of Frigopol compressor instead of Maneurop compressor
2. Leakage test with N_2
3. Evacuation
4. Mounting of all measuring equipment
5. Starting and charging refrigerant R723
6. Function of the injection system and the special vessel
   - *A. Manual injection valve (dry operation)*
   - *B. Electronic thermo-valve AKVA (dry operation)*
   - *C. Capillary tubes (dry operation)*
   - *D. Thermostatic thermo-valve TUBE (dry operation)*
   - *E. Capillary tubes – w. floater in the vessel (wet operation)*
   - *D. Level adjusting valve TEVA without floater in the vessel (wet operation)*
7. Evaporator capacity and operation
8. Cooling capacity, operation and efficiency of the compressor
9. Condenser capacity and operation

Mounting of Frigopol compressor instead of Maneurop compressor

A Frigopol compressor that previously had been in operation at ILK was lent for the tests at DTI – primarily in order to obtain the results with the special vessel.

The compressor was charged with Mobil SHC 226E according to the instructions of the manufacturer.

Leakage test with N_2

The system was leakage tested with N_2 and leaks were localised and put in order.

Leaks were found at a flange, a plug, a soldering, several spindle packing boxes and several weldings.

Several leaks could not be localised with nitrogen or leakage spray (soapy water) but were found by means of ammonia smell after refrigerant charging or with sulphur stick with shut off in sections.

NB! Water absorbs ammonia and therefore soapy water is not optimal to use in connection with leak finding when there is R723 in the system.

Evacuation

A careful double evacuation with ”refrigerant charging” to minimise the moisture content was carried out to app. 1 mbar according to a precision vacuum meter.

Mounting of all measuring equipment

The plant was equipped with measuring equipment for registration of pressure, temperature, flow and electric consumption.

Heat to the compressor before start

A fan heater was installed near the compressor for heating of the oil before compressor start – to "simulate" oil heat during standstill and to avoid foaming of the refrigerant in the oil during start.

Starting and charging refrigerant R723

Charging took place in gaseous form at the service connection on the suction side of the compressor with capillary tubes as liquid injection system and with the multifunction vessel in operation at highest positionl – until charging seemed to suit the functions of the vessel.

A total of 3.28 kg R723 was charged.

Receiver is filled to app. half of the liquid level indicator when the whole charge is in the condenser.

A small green ball in the sight glass did not float on the R723 liquid!

Function of the injection systems and the multifunction vessel

The plan was to measure the operational data of the five injection systems (manual expansion valve, electronic thermo-valve, capillary tube and level adjusting valve and thermostatic expansion valve) during different operating conditions with or without multifunction vessel and the data should be used to compare and choose the most suitable system.

Figure 5.10: The principle of the injection systems and multifunction vessel

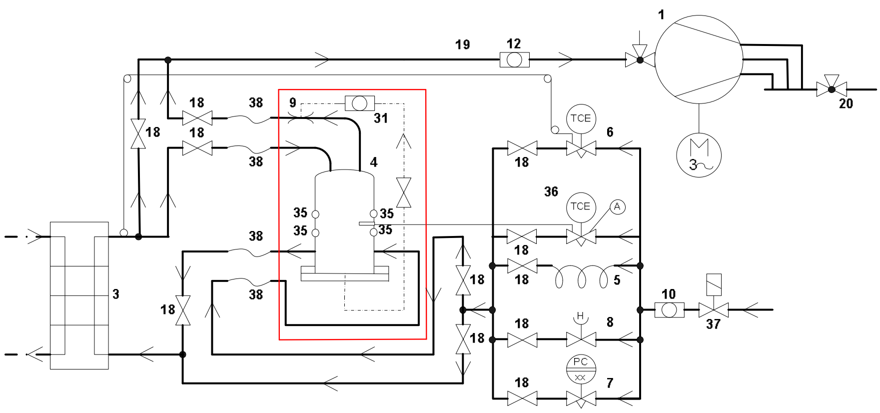


Table 5.6: Desired operating conditions in [℃] and [K]:

Parameter	Nom.	t_c ÷4	t_c ÷8	t_o ÷2	t_o ÷4	t_o ÷6	t_o ÷6 t_c ÷8
Evaporation temperature t_o	÷21,5	÷21,5	÷21,5	÷23,5	÷25,5	÷27,5	÷27,5
Brine inlet $t_{brine,ind}$	÷15	÷15	÷15	÷17	÷19	÷21	÷21
Brine outlet $t_{brine,ud}$	÷18	÷18	÷18	÷20	÷22	÷24	÷24
Condensation temperature t_c	+28	+24	+20	+28	+28	+28	+20
Cooling tower water inlet $t_{køletårn,ind}$	+20	+16	+12	+20	+20	+20	+12
Super heating	Min.	Min.	Min.	Min.	Min.	Min.	Min.

A. Manual injection valve (dry operation

The test was dropped – because it had no real objective.

B. Electronic expansion valve AKVA (dry operation)

At first we did not succeed in getting the EKC regulator and the AKV valve to work although it was possible to open the valve with the EKC controller manually, but that problem was solved after correct setting on all parameters in the control.

Is was necessary to use the multifunction vessel as suction accumulator during start-up and adjustment of regulation parameters.

It turned out, that the valve size was too small for the capacity – and the manually valve was opened in parallel i to solve that problem.

The evaporator pressure (and temperature t_o) and evaporator capacity Q_o were constantly trying to adapt to operation conditions when EKC+AKV was in operation. It typically succeeded in “capturing” liquid carry-over, but it took long time (periods of more than 10 minutes) after that before superheat again was decreased to optimal performance.

This can probably be solved by adjusting and optimizing the PID-settings in the regulator, but it could take very long time unless you get a good procedure for quick adjustment.

Picture 5.11: Thermografic picture of the plate heat exchanger during stabile operation

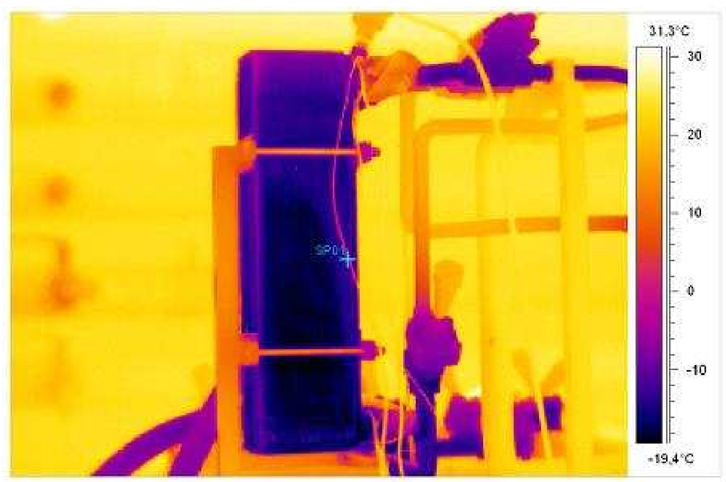
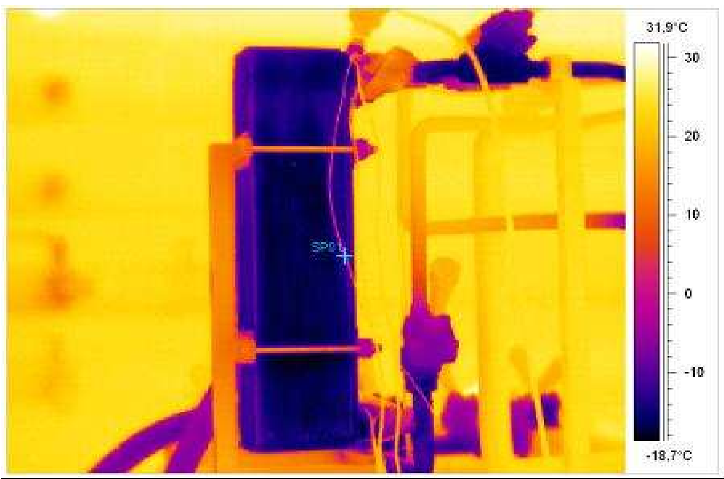


Table 5.7: Selected main data from test results (overview)

Operating condition		**nom.**	**nom.**
Suction accumulator		with	without
(TTR_19) Brine inlet	[℃]	-14,75	-14,85
(TTR_20) Brine outlet	[℃]	-17,77	-17,96
(TTR_16) Water outlet	[℃]	25,43	22,90
(TTR_15) Water inlet	[℃]	20,04	19,98
(TTR_1) Compressor inlet	[℃]	-23,53	-21,41
(TTR_2) Compressor outlet	[℃]	28,16	27,49
(TTR_3) Evaporating temperature	[℃]	-19,54	-21,10
(TTR_11) Suction temperature	[℃]	3,17	-5,27
(TTR_18) Superheat	[℃]	-11,43	-10,16
(TTR_14) Liquid	[℃]	22,41	22,43
(TTR_12) Discharge cylinder 1	[℃]	143,38	132,59
(TTR_21) Discharge cylinder 2	[℃]	135,02	121,74
(TTR_22) Discharge cylinder 3	[℃]	133,18	120,70
(TTR_13) Discharge common	[℃]	118,14	110,61
(ETR_42) Power compressor	[kW]	3,09	3,35
Evaporator capacity Q_o	[kW]	7,84	9,79
Condenser capacity Q_c	[kW]	10,58	12,79
COP_o evaporator	[-]	2,54	2,92
COP_c condenser	[-]	3,42	3,81
Evaporator			
ΔT_evaporator	[K]	4,79	6,26
ΔT_brine	[K]	3,01	3,11
ΔT_superheat	[K]	8,11	10,94
Condenser			
ΔT_condenser	[K]	7,55	6,48
ΔT_water	[K]	5,39	2,92
ΔT_subcooling	[K]	5,75	5,06

The thermografic picture shows outside temperatures of the plate heat exchanger, where it is visualized that a relatively large part of the evaporator is used for superheating. The left picture of the evaporator is taken with normal camera.

Picture 5.12: Thermografic picture of the plate heat exchanger during unstabile operation



The picture shows the same heat exchanger during unstabile operation, where it can be seen that liquid collects in the top section and leaves the evaporator:

C. Capillary tubes (dry operation)

The plant was started with the multifunction vessel as suction accumulator to avoid liquid intake at the compressor. It turned out that liquid ran into the multifunction vessel – so that the evaporator operated with wet return. It could be seen on the sight glass in the liquid line that a mixture of gas and liquid was supplied to the capillary tube from the condenser (the mixture flushed through the sight glass) at the same time as liquid splashed out of the evaporator and into the vessels upper chamber (in principle wet operation) at the same time as the liquid outlet for the evaporator was rather low (app. 6 kW cooling capacity - app. 60% of nominal capacity). The reason for the reduced output is probably the flash gas supplied through the capillary tube and the evaporator compared with an ordinary thermostatic expansion valve. In principle the liquid had been moved from the condenser side (ordinary receiver in the condenser) to the low pressure side (vessel). Throttling tests on the valve after the capillary valve – to try to avoid flash gas before the capillary tube - were no success.

The following table shows the data results from the test (with splashes out of the evaporator and liquid/gas to the capillary tube)

Table 5.8: Selected main data from test

Operating condition		**nom.**
Suction accumulator		with
(TTR_19) Brine inlet	[℃]	-14,84
(TTR_20) Brine outlet	[℃]	-17,68
(TTR_16) Water outlet	[℃]	29,12
(TTR_15) Water inlet	[℃]	20,04
(TTR_1) Compressor inlet	[℃]	-22,38
(TTR_2) Compressor outlet	[℃]	28,30
(TTR_3) Evaporating temperature	[℃]	-18,23
(TTR_11) Suction temperature	[℃]	-6,62
(TTR_18) Superheat	[℃]	-15,17
(TTR_14) Liquid	[℃]	30,34
(TTR_12) Discharge cylinder 1	[℃]	131,62
(TTR_21) Discharge cylinder 2	[℃]	120,26
(TTR_22) Discharge cylinder 3	[℃]	118,55
(TTR_13) Discharge common	[℃]	114,11
(ETR_42) Power compressor	[kW]	4,78
Evaporator capacity Q_o	[kW]	5,90
Condenser capacity Q_c	[kW]	9,04
COP_o evaporator	[-]	1,24
COP_c condenser	[-]	1,89
Evaporator		
ΔT_evaporator	[K]	3,40
ΔT_brine	[K]	2,85
ΔT_superheat	[K]	3,06
Condenser		
ΔT_condenser	[K]	7,72
ΔT_water	[K]	9,07
ΔT_subcooling	[K]	-2,04

D. Thermostatic thermo-valve TUBE (dry operation)

In spite of careful start of compressor with control of the sight glass in the suction line and a hand on an almost closed stop valve between evaporator and compressor to match operation of the stop valve quite a lot of liquid entered the compressor (that became frosted) and it had to be stopped in order to evaporate the liquid.

The multifunction vessel was connected as "suction accumulator": For quite a long time after start refrigerant liquid flow was observed to flow to the vessel from the evaporator.

At some time the brine flow disappeared because frozen brine in the evaporator! A measurement trawith a refractrometer showed that the glycol only was frost-proof to app. -21°C. Extra glycol was charged for get it additionally frost-proof.

When superheat is increased the evaporator pressure decreased - because the evaporator charge declines substantially. It presumably also has something to do with a rather poor liquid/gas mixture inlet to the evaporator because of the pipe system between the valves and the evaporator.

The thermostatic expansion valve started to work with an acceptable pendling at a superheat of app. 10 K compared to the evaporator pressure

NB! Refrigerant liquid flows out of the evaporator for several minutes when the pressure is lowered during start-up!

The pressure loss between the evaporator and the compressor almost disappeared (due to pressure loss in hoses and vessel) as the multifunction vessel was disconnected.

Pendling directly influences the evaporating pressure and the performance of the evaporator.

When trying to decrease super heating (only 2 x 1/8 rev.) liquid appeared – the limit is subtle!

Superheat once in a while submerges during adjustment, but apparently without liquid coming out of the evaporator.

The following table shows data results from the tests that were carried out. The condition (t_o - 6) could not be run due to freezing in the evaporator.

E. Capillary tubes – with floater in the vessel (wet operation)

Figure 5.13: The principle of the injection system and multifunction vessel

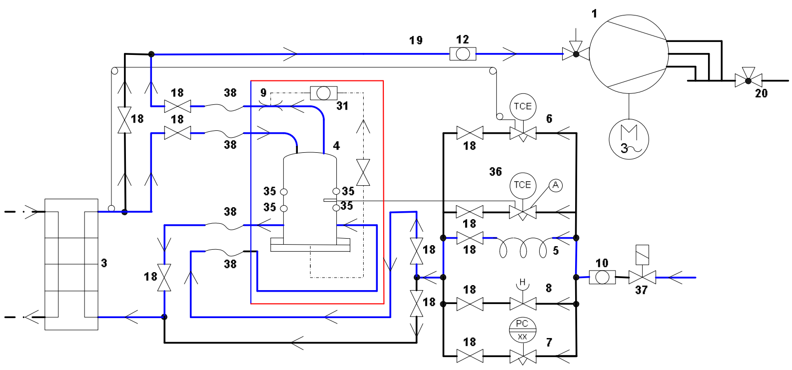


Table 5.9: Data results from tests with thermostatic expansion valve

Operating condition		**nom.**	**tc-4**	**tc-8**	**to-2**	**to-4**
Suction accumulator		without	without	without	without	without
(TTR_19) Brine inlet	[℃]	-14,89	-14,77	-14,86	-17, 08	-19,14
(TTR_20) Brine outlet	[℃]	-17,82	-17,80	-17,84	-19 ,85	-22,01
(TTR_16) Water outlet	[℃]	25,46	21,43	17,59	26,23	25,85
(TTR_15) Water inlet	[℃]	19,94	15,79	12,08	20,17	2 0,33
(TTR_1) Compressor inlet	[℃]	-22,06	-20,92	-21,57	-22,91	-24,35
(TTR_2) Compressor outlet	[℃]	27,74	23,89	20,17	28 ,29	27,40
(TTR_3) Evaporating temperature	[℃]	-21,27	-20,06	-20,74	-22,13	-23,57
(TTR_11) Suction temperature	[℃]	-5,55	-6,49	-6,48	-6,83	-7,81
(TTR_18) Superheat	[℃]	-10,38	-10,60	-10,61	-12,36	-14,13
(TTR_14) Liquid	[℃]	23,42	19,96	16,50	23,94	23,96
(TTR_12) Discharge cylinder 1	[℃]	131,08	120,75	116,6 2	133,72	133,89
(TTR_21) Discharge cylinder 2	[℃]	116,63	106,86	103,1 9	119,02	120,02
(TTR_22) Discharge cylinder 3	[℃]	119,05	109,39	105,2 2	120,78	121,06
(TTR_13) Discharge common	[℃]	115,38	107,13	103,22	1 16,54	115,55
(ETR_42) Power compressor	[kW]	5,03	4,93	4,63	4,91	4,64
Evaporator capacity Q_o	[kW]	9,73	10,73	10,56	8,99	8,20
Condenser capacity Q_c	[kW]	12,42	13,51	13,16	11,65	10,59
COP_o evaporator	[-]	1,93	2,18	2,28	1,83	1,76
COP_c condenser	[-]	2,47	2,74	2,84	2,37	2,28
Evaporator						
ΔT_evaporator	[K]	6,38	5,29	5,88	5,05	4,43
ΔT_brine	[K]	2,93	3,03	2,98	2,77	2,87
ΔT_superheat	[K]	10,89	9,46	10,13	9,77	9,44
Condenser						
ΔT_condenser	[K]	7,57	7,17	6,55	7,83	6,80
ΔT_water	[K]	5,52	5,65	5,51	6,06	5,52
ΔT_subcooling	[K]	4,33	3,93	3,66	4,35	3,43

This system was in operation during start-up and refrigerant charging where the floater started to work when enough liquid had been supplied to the plant, but after a couple of "pumps" it stopped again – probably because it again needed liquid before it again could close "the intermediate plate" and start pumping liquid to the evaporator.

Unfortunately, the liquid separator function of the vessel did not function as it was supposed to. There were some liquid splashes out of it, which entered the suction pipe to the compressor which clearly could be seen in the sight glass in the suction line.

Liquid was "blown" into the upper part of the vessel, when the liquid level approached the closing position of the ball. Probably the flash gas from below blows the liquid upwards from where it enters the suction pipe.

Besides the liquid was not able to leave the vessel before the ball closed and/or the liquid level came above the level of the pipe connection in he vessel.

The conclusion was that the liquid separator part and the liquid outlet did not work optimally and therefore constructive changes of the liquid outlet from the lower chamber of the vessel had to be carried out as well as a baffle system in the upper chamber of the vessel.

A baffle plate was built into the vessel and the liquid outlet pipe changed, so that liquid could leave form the bottom of the vessel, and 1,4 kg R723 was charged extra during start-up with changed system. Practically no oil was found in the vessel.

Unfortunately the changes was not able to solve all the problems. During the following tests liquid entered both sections in the vessel and the “valve” inside was not able to open and close to produce the desired pumping effect. Some pumping effect occurred – but it was probably due to back-flow of gas from liquid line. Liquid did not enter the evaporator in sufficient quantity for wet operation. Some of the wanted function appeared for short periods during start-up and stop.

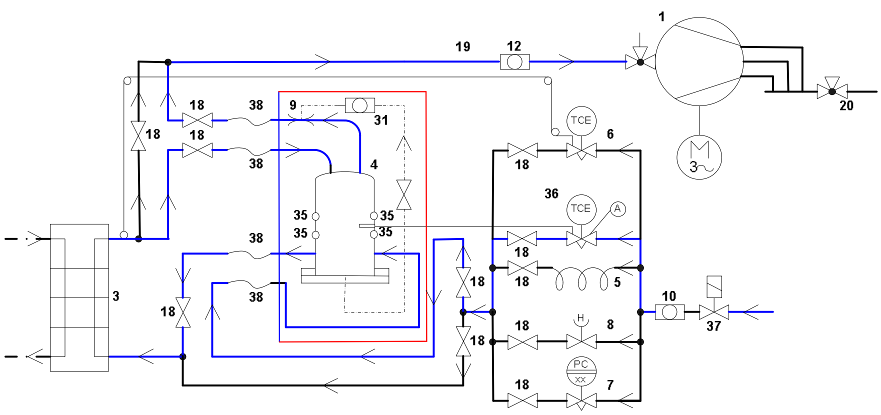
The baffle arrangement worked very well.

Perhaps the diameter of the flexible hoses and the wet return pipe are too small for wet operation, and maybe at non-return valve between vessel and evaporator can solve the problem.

The basic principle may still work, but the system has to be developed further.

D. Level adjusting valve TEVA without floater in the vessel (wet operation)

Figure 5.14: The principle of the injection system and multifunction vessel

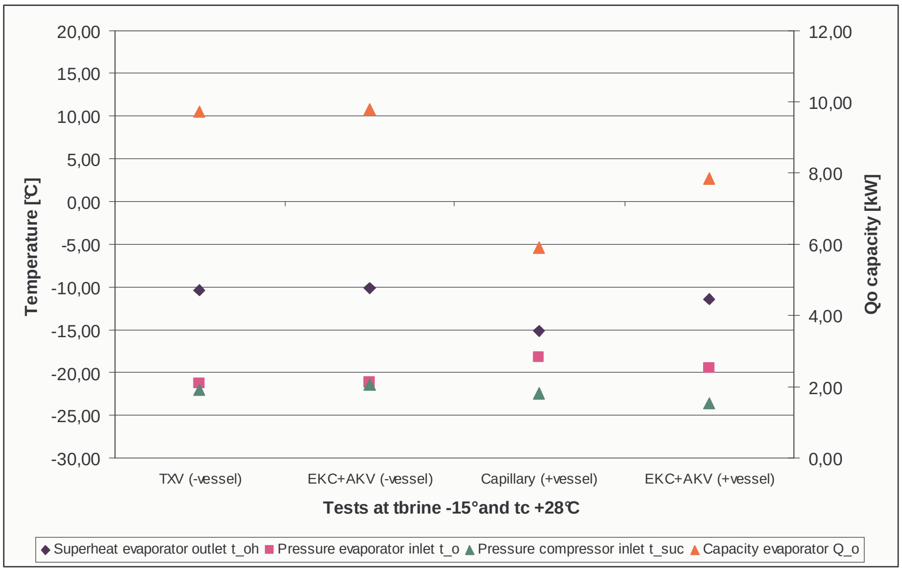


No tests were made with the TEVA-valve based on the results of the previous test with capillary tube and the special vessel. There was no reason to believe, that the system would perform satisfactory before changes had been made to the pipe connections etc. to the vessel.

Evaporator capacity and operation

Some analysis was carried out based on the results from the tests with different liquid injection systems in order to investigate the evaporators behaviour in terms of evaporator performance (Q_o), evaporator temperature (t_o) and super heating (t_{oh}), and the temperature difference between the refrigerant and the liquid ()t).

Figure 5.15: Presentation of selected evaporator data



Bases on the data and the figure one can conclude as follows:

Incoming temperature difference during tests was approx. ca. 6 K

TXV and EKC+AKV gives more or less the same results with superheat about 11 K

Operation with the capillary tube didn´t work out – liquid left the evaporator and evaporator performance were reduced to app. half.

Pressure drop in vessel and pipes (used as suction accumulator) was way too much.

Generally a suction accumulator as safety against liquid carry-over from evaporator to compressor always has to be considered – especially as safety precaution at start-up conditions.

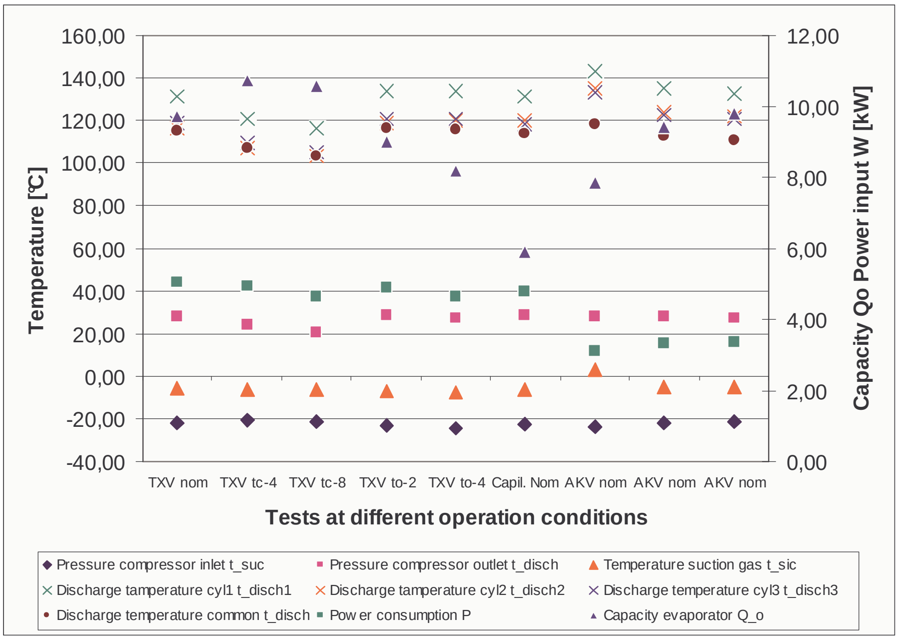
Cooling capacity, operation and efficiency of the compressor

The operation data of the compressor were measured for different operating conditions and the data used to calculate the cooling capacity (Qo) and power consumption (W) and Coefficient Of Performance (COP).

It was intended to investigate how far the suction pressure could be lowered and the condenser pressure increased before the discharge temperature became too high (the oil type would ”crack” at app. 180°C).

The compressor gives nominal design capacity at 75 Hz, and the frequency changer AKD was set at min. 35 Hz and max. 75 Hz.

Figure 5.16: Presentation of selected compressor data



Bases on the data and the figure one can conclude as follows:

There is a significant difference between the discharge temperatures of each of the 3 cylinders, where one of them (the cylinder on opposite side of suction inlet) during all test were 10-15 K above the other two.

In all tests the common discharge temperature from all 3 cylinders was below 130°C

Detailed test results and data for the compressor can be found in ILK´s section of the report.

Condenser capacity and operation

Some analysis was carried out based on the results from the tests with different liquid injection systems in order to investigate the condenser behaviour in terms of condenser capacity (Q_k), condensation temperature (t_c) and supercooling (t_{uk}), and the temperature difference between refrigerant and air ()t).

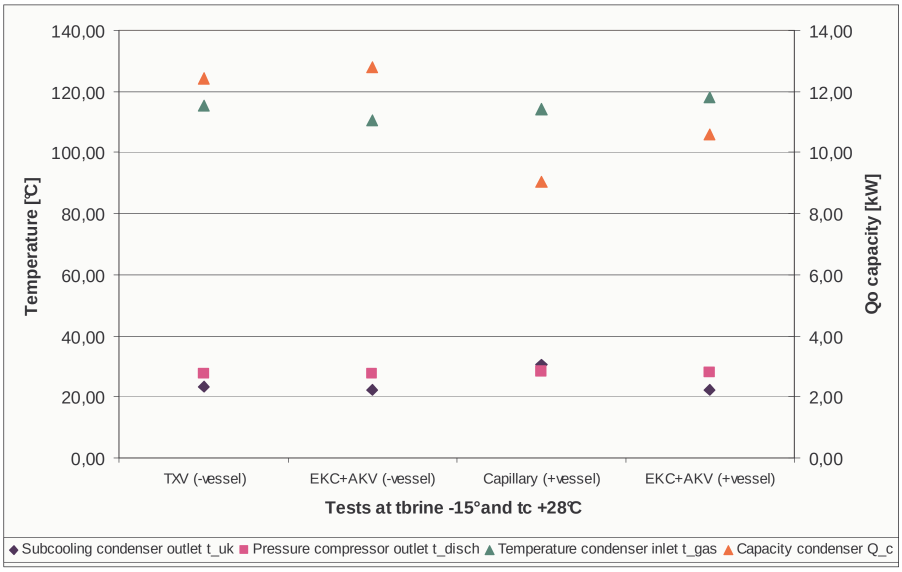
From data ad figure one can conclude:

Incoming temperature difference during the tests was 7-8 K

Subcooling in condenser (with built-in receiver) was between 3.5 and 5.5 K

No subcooling occurred during dry operation with capillary tube – because flashgas entered the capillary tube.

Figure 5.16: Presentation of selected condenser data



Main results from testprogramme with marine chiller

- Several leakages were hard to find with N_2 or leakage spray (soapy water) but easy to detect with the smell of ammonia or a sulphur stick.
- Evacuation was made down to app. 1 mbar abs.
- A total of 3,3 kg R723 was initially charged.
- A small green ball in the sightglass of the receiver did not float on R723 liquid!
- Use of the multifunction vessel as suction accumulator was necessary during start-up and adjustment of injection systems
- The electronic thermostatic valve was able to avoid situations with “liquid carry-over”, but it took very long time after each “situation” before superheat again was decreased to optimal performance.
- Thermografic pictures showed quite good liquid/vapour distribution and also showed that liquid collected in the top of the plate heat exchanger during unstable operation.
- Liquid supply by capillary was no success; flash-gas entered the tube from the condenser and too little liquid was injected to the evaporator.
- During adjustment of thermostatic expansion valve one realised the correlation between superheat, pressure, cooling performance and liquid charge of the evaporator in a very illustrative way. Adjustment was very sensitive.
- Minimum superheat with mechanical thermostatic expansion valve was app. 10 K, and pendling directly influences the pressure and the performance.
- Cooling capacity of app. 11 kW was reached compared to 10 kW in design condition.

- Unfortunately the multifunction vessel did not perform as it was supposed to with liquid supply from the capillary tube. At first liquid droplets entered the suction line and liquid outlet to evaporator was positioned to high. These two problems were solved, but the valve inside the vessel still did not open and close to give the desired pumping effect. Liquid entered the upper part and liquid supply the evaporator was inefficient, probably because of too much pressure drop in wet suction line. The basic principle may still work, but there was no time at the end of the project left to solve the problems.
- Temperature differences (incoming) for evaporator during tests were app. 6 K.
- Electronic and mechanical expansion valve reached same performance with superheat of app. 11 K.
- There were significant differences in discharge temperatures of 10-15 K from each of the Frigopol´s 3 cylinders. Cylinder opposite suction side reached highest temperature level,
- Common discharge temperature (from all 3 cylinders) reached 100 – 120℃ for suction temperatures between -20 - -24℃.
- Temperature differences (incoming) for condenser during tests were 7-8 K.
- Sub cooling form built-in receiver in condenser was between 3,5 and 5,5 K.

5.3 Ice Flake Machine

Tabel 5.10 shows the measuring equipment for the test programme for the Ice Flake Machine. All data were measured and logged with a Hewlett Pachard 3497A logger combined with DTI measuring software TI-DOP version 2.42 and 2.43.

Test programme, etc

Main parts of the test were:

1. Mounting of all measuring equipment
2. Evacuation
3. Start and charge of refrigerant R723
4. Preparation for new test
5. Mounting of new Maneurop compressor
6. Evacuation
7. New start and possible recharging of refrigerant R723
8. How the injection systems works
9. How the capillary tubes and injection system works
10. Evaporator capacity and operation
11. Cooling capacity, operation and efficiency of the compressor
12. The capacity and operation of the condenser
13. Capacity and operation of the ice machine

14. Electricity consumption of individual components

Tabel 5.10: Measuring equipment for Ice Flake Machine

No.		Measuring of ...	Measuring equipment
01	PTR	Pressure of refrigerant before compressor	Pressure transducer
02	PTR	Pressure of refrigerant after compressor	Pressure transducer
03	PTR	Pressure of refrigerant before evaporator	Pressure transducer
11	TTR	Temperature of refrigerant before compressor	Temperature sensor
12	TTR	Temperature of refrigerant after compressor	Temperature sensor
13	TTR	Temperature of brine out of evaporator (sol-dered) (.19)	Temperature sensor
14	TTR	Temperature of refrigerant after condenser	Temperature sensor
15	TTR	Temperature of air intake to condenser	Temperature sensor
16	TTR	Temperature of refrigerant before evaporator	Temperature sensor
17	TTR	Temperature of refrigerant after evaporator (.11)	Temperature sensor
18	TTR	Temperature of brine before evaporator (.21)	Temperature sensor
19	TTR	Temperature of brine after evaporator (.13)	Temperature sensor
20	TTR	Temperature of brine after ice drum	Temperature sensor
21	TTR	Temperature of brine to evaporator (soldered) (.18)	Temperature sensor
22	TTR	Temperature of water to ice drum	Temperature sensor
32	FTR	Flow of brine after ice drum	Flow meter
41	ETR	Electric power consumption of compressor	Electricity efficiency meter (3 phases)
42	ETR	Electric power consumption of all electricity consumers i.e. compressor, fan, brine pump, drum engine, control, etc.	Electricity efficiency meter (3 phases)

Mounting of all measuring equipment

The plant was equipped with measuring equipment for registration of pressure, temperatures, flow and electric power.

Evacuation

A careful and optimum double evacuation with refrigerant charge to app. 1 mbar,a was carried out to minimise the moisture content.

Start and charge of refrigerant R723

Charging was carried out with capillary tubes as liquid injection system and with closed inlet valve on the receiver (so no liquid accumulated in the receiver) until suitable superheating was obtained on the evaporator outlet (starting point app. 10K). Calculated charge amount was app. 600 g.

A total of 0,74 kg R723 was charged.

Refrigerant temperature at intake to evaporator after the capillary tube increased as more refriger-ant was charged.

After almost one week with R723 in the rubber hoses between the refrigerant bottle and the plant there was a smell of ammonia near the plant. A sulphur stick showed that the rubber hoses leaked.

The next start up of the plant – only 4 days after charging of refrigerant – was no success as the compressor had broken down. And that was the end of the first phase of the test.

Preparation for new test

The damaged compressor was demounted and sent to Maneurop for inspection and analysis.
A new one was ordered for mounting and carrying out an intensive test programme without unnecessary stops to try to obtain as many test data as possible – knowing that the next compressor also would break down.

Mounting of new Maneurop compressor

The mounting of a new Maneurop compressor was made after the oil on the compressor had been drained off (app. 700 g) and new Shell Clavus G 46 had been charged to a level of app. 2/3 in the sight glass.

Evacuation

The new compressor was evacuated with a connected crankshaft heat – and shut off from the rest of the pipe system with the stop valves from the compressor (still with R723).

New start and possible recharging of refrigerant R723

The plant was started again with the new Maneurop compressor without problems.
The plant was in operation with capillary tubes, manual valve and thermostatic expansion valve at the operating conditions that could be achieved with the room temperature in the hall (inlet temperature to the condenser) and with the ice machine (the load the system could obtain with water from the vessel with ice and water).
The control of the condenser fan was disconnected electrically and therefore the condenser fan was "force driven" together with the compressor.
A ventilator blew ambient air against the compressor to avoid the compressor housing from becoming warmer than necessary.
The oil level in the compressor was at the beginning app. ¾ of the glass.
It turned out that the plant needed to be charged and app. 450 g was recharged.

Performance of the injection systems

The operational data of the three injection systems (manual expansion valve, capillary tubes and thermostatic expansion valve) was meant to be measured under different operating conditions and the data to be used to compare and chose the most suited system. The requested result was small superheat and an acceptable liquid/gas distribution in the plate heat exchanger.

It was very difficult to adjust the manual valve - it was very sensitive – only minor changes on the scale could be made (one single partial mark) before superheating changed considerably (declined). Frost on the outside of the plate heat exchanger was not completely consistent - so the distribution was not very good. At stable superheat "peaks" on the electric power consumption disappeared.

Figure 5.17: Trend curve from operating period with the manual valve

Performance of capillary tube and injection system

The operation data of the capillary tubes and injection system were to be measured under different operating conditions. Liquid injection with the capillary tube and the manual expansion valve would probably require operation with fixed charge which had to be "trimmed" through more or less liquid in the receiver. A "suitable" superheating was to be obtained (lowest possible without liquid supply to the compressor – during start of the plant and varying conditions).

The change from manual valve to thermostatic expansion valve increased the evaporator pressure by app. 1 K.

It was difficult for the thermostatic expansion valve to get the superheat suitably stable while the electricity intake was much more stable without "peaks" compared to operation with manual valve.

Figure 5.18: Trend curve from start with the thermostatic expansion valve

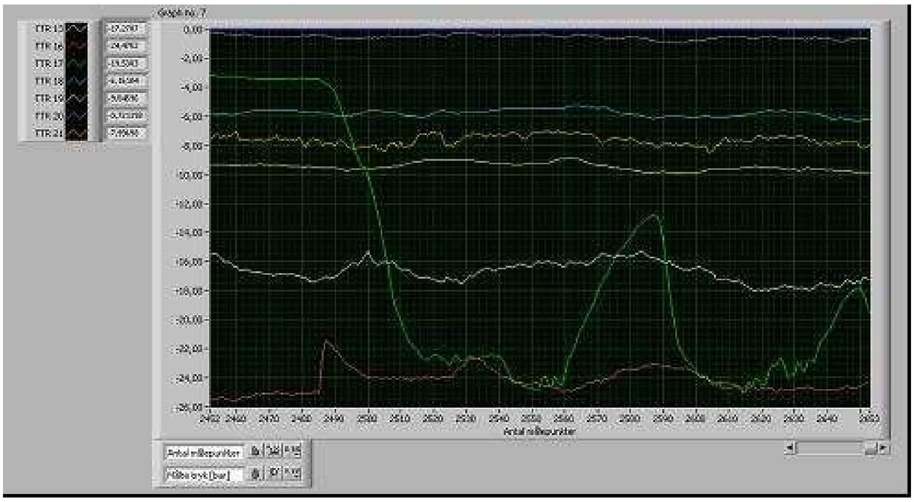


Figure 5.19: Trend curve from relatively good operating period with thermostatic expansion valve

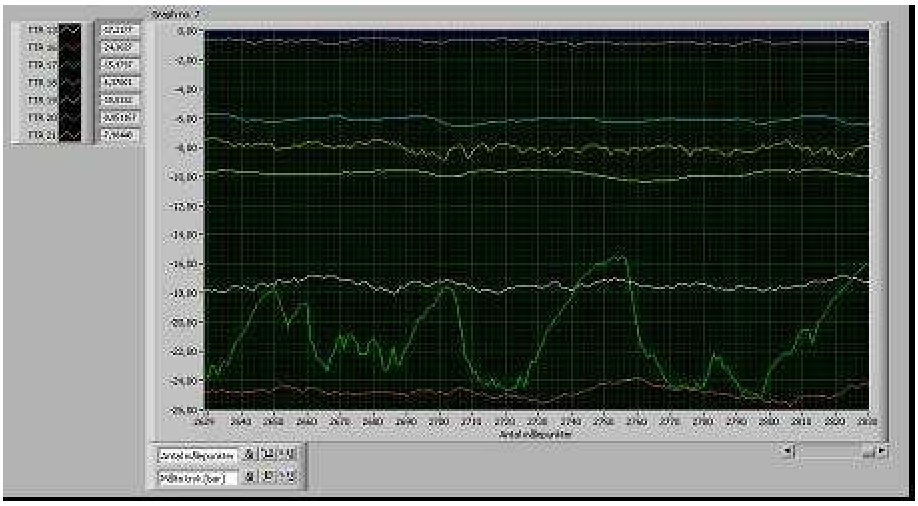
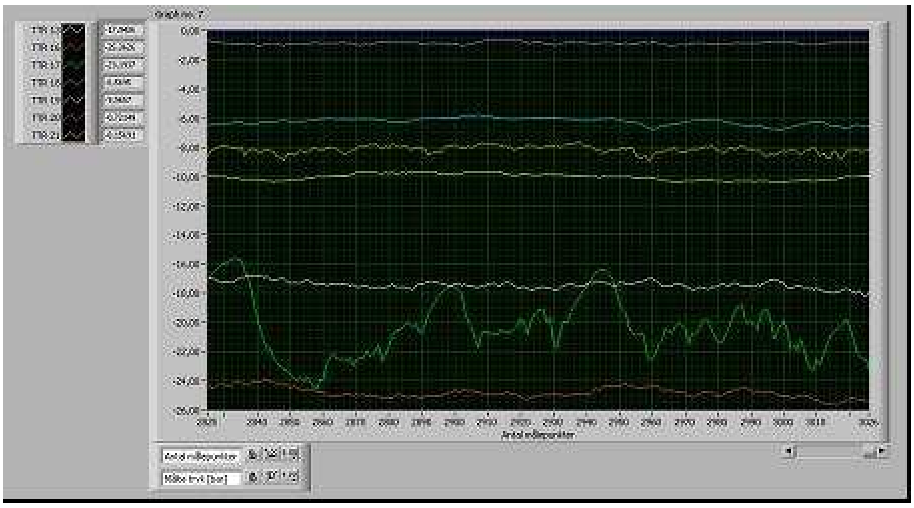


Figure 5.20: Trend curve from another good operating period with thermostatic expansion valve



On the photo it almost seems as though the thermo-valve had to be "run in".
Short time closure of the liquid by-pass across the receiver had a quick reaction with regard to pressure and temperatures! The evaporator pressure increased and superheating disappeared – because more liquid was supplied to the evaporator. Frost was formed on the compressor because it received liquid and it had to be stopped to evaporate the refrigerant.

Evaporator capacity and operation

The operational data of the evaporator was supposed to be measured for different operating conditions and the data was used to calculate the evaporator capacity, evaporator temperature and superheating and the temperature difference between refrigerant and liquid.
No relevant data existed to be analysed because of the very short test period.

Cooling capacity, operation and efficiency of the compressor

The operational data of the compressor was to be measured for different operating conditions and the data was used to calculate cooling capacity, power consumption and Coefficient Of Performance (COP).
It was to be investigated how far the suction pressure could be lowered and the condenser pressure increased before the discharge temperature becomes too high (oil type ”cracks” at app. 180℃).

Compressor 1:

The plant started up the first time without problems and produces flake ice, but the discharge temperature increased rather quickly until it was stopped at 175℃ to let a fan cool the compressor unit on the outside. Unfortunately the suction stop valve to the compressor has not been completely opened, so perhaps it had throttled a bit before the compressor!
It was not possible to start the plant at the next attempt only 4 days after refrigerant had been charged. The fuses burned on several of the phases when the power to the compressor was turned on. It turned out, that the compressor was damaged.
The compressor was evacuated because Danfoss did not want it to be exposed to more R723 before it had been opened and investigated in detail.
The oil level was below minimum level of the sight glass in connection with the evacuation and it started "to foam" when the pressure came below 1 bar,a (atmospheric pressure). Finally, some clear brown substance could be seen in the glass which looked like dissolved varnish from the windings. Behind the glass – in the bottom of the compressor house – it was possible to see the oil foam up (white-brown).
The compressor was sent to Maneurop in France to be opened and inspected. Maneurop made a “Claim Report” from which following pictures and text is taken:

ADDITIONAL PICTURES

FAILURE ANALYSIS

The using of the refrigerant called R723 (azeotropic mixture of ammonia and dimethylether) led to the damages mainly on the varnish of the stator windings, completely dissolved into the suction chambers of the compressor (see pictures).
The others effects of the refrigerant are visible onto the plastic parts: internal motor connector plug (cluster block), stator winding leads, ...
The only plastic part which has submitted no visual damage is the sheath wires made in teflon.

After having received the Claim Report further correspondence was with Maneurop, from which following parts is taken:

- The exact cause of the motor short-cut? We don't see short circuit onto the stator windings. May be that it is into the stator windings....
- The mechanical parts management of operation: All the mechanical parts have a good aspect as new parts without specific wear.
- Reason to believe that a compressor with R723-compatibel lacker on windings and R723-compatibel plastic parts can handle the job: May be. But we can say, in regard to the first results that if the laker on windings and the plastic parts are compatible with R723, the compressor will run more time that it ran…
- Any visibel evidence of the compressors operation with discharge gas temperatures up to 175℃ :
- No visible evidence of the fact that the compressor ran at high discharge temperature.
- Remark: It seems that the plastic has been diluted by R723 during stop compressor sequence because all the plastic material is located into the suction side, no plastic material into discharge side.

Compressor 2:

Danfoss delivered a new compressor of the same kind and type to replace the one that had broken down with regard to mounting a new machine and as soon as possible - without unnecessary delays – to carry out the tests before the varnish again was dissolved.

After only one day of operation the oil in the compressor house looked "brown" with varnish like substances on the inside of the sight glass above oil level.

When starting the compressor on the second day the oil in the sight glass looked strange and stiff – more or less like grease.

Short time closure of the liquid by-pass above the receiver during operation with capillary tubes resulted in liquid coming out in the suction pipe. The energy consumption increased at the same time as the discharge temperature declined. There was frost on the compressor so it was stopped for the refrigerant in the compressor to evaporate. The oil glass in the compressor was completely full after that "maltreatment".

The next morning the oil in the compressor house did not look good. It was brown with a brighter layer of "something” on top.

Unfortunately, all three fuses were blown as the compressor was about to start – so that is what happened the second time - after only two days of operation.

Some days later, there was an ammonia smell from the system when it was out of operation and a "sniffer test" followed by soapy water spray located the leaks to the union of the stop valve after the compressor.

The capacity and operation of the condenser

The operational data of the condenser was to be measured for different operating conditions and the data was to be used for calculation of the condenser capacity, condensation temperature, sub-cooling and the temperature difference between the refrigerant and the air.

Unfortunately no relevant data existed to be analysed because of the very short test period.

Capacity and operation of the ice machine

Unfortunately no relevant data existed to be analysed because of the very short test period, but the ice machine did make scale ice for a very short time as can bee seen on the picture:

Power consumption of individual components

Electricity consumption of ”electricity consumers” was measured:

Condenser fan: app. 169 W. Nominal 135 W according to plate sign.

Heating element in ice water tank: app. 2.365 W.

Small charge water pump (SDMOTOR): app. 400 W.

Brine pump: app. 350 W

Drum engine and spray engine: 120 W. Nominal 5 W according to plate sign on the spray engine.

Main results from test programme with Ice Flake Machine

- A total of app. 0,7 kg R723 was initially charged. Later app. 0,4 kg was recharged.
- After app. 1 week with R723 the rubber hoses between refrigerant vessel and system began to leak.

- The first Maneurop compressor (piston-type) had a break-down only 4 days after charging of refrigerant. It was dismantled and sent to France for inspection and analysis, and a new one was ordered. The second Maneurop compressor had a break-down after 2 days of operation.
- It was very difficult to adjust the manual injection valve – which was very sensitive.
- Evaporation pressure increased app. 1 K from manual injection valve to thermostatic expansion valve.
- It almost looked as if the thermostatic expansion valve had be "run-in". The control of the super-heat appeared to be better after a while.
- Twice a liquid carry-over from the evaporator happened during adjustment by accidents where frost was formed on the compressor. A suction accumulator would have been suitable protection for the compressor.
- The discharge temperature of the first compressor quickly reached 175°C, and it turned out later that the suction side stop valve at that time was not completely open.
- Main results from Maneurop´s compressor inspection was as follows: Varnish of stator windings (completely dissolved lacker but no electrical short-cut), damage of almost all plastic parts, mechanical parts appeared as new. The break-down was expected because the compressor the compressor was not delivered modified with components compatible with R723 as originally planned, but the damage happened much quicker than expected.
- No relevant data existed for analysis of compressor, evaporator and condenser capacity and performance because of the very short test periods.
- The scale ice machine did produce ice for very short periods.
- Electric power consumption for other electricity consumers than compressor (pumps, fan and ice drum) are relative high.

5.4 Normal Cooling Cell

Table 5.11 contains plan of the measurement points for the cooling cell aggregate. The plan of the measurement points is valid for both, for the normal cooling and deep freezing. The measuring dates were registered and stored with the registration system Medana of the firm delphin.

Following **Measuring Program** was realised:

1. Determination optimal refrigerant filling amount and adjustment HP- and LP-keeper, controller condensate pressure, cooling controller, overheating temperature
2. Dynamic measuring of cooling down
3. Measuring of plant at conditions of calculation t_o and t_c
4. Measuring with variation condensation and evaporation pressure

5. Fixing minimal achievable evaporation temperature at adherence maximal possible compressor endtemperature and t_c = 42℃
6. Measuring influence overheating temperature to evaporator (calculation 6K) on compresser endtemperature ($\Delta t_{ÜH}$ = 2K, 4K, 6K, 8K; further criterions: drops refrigerant, regulation stability refrigerant evaporator)
7. Measuring influence under cooling tube condenser
8. Analysis refrigerant concerning humidity

The results of measurement are represented in the figures 5.21 until 5.30 graphically. The heat exchanger work with good logarithmic temperature differences. The temperature of cold air outlet lies with usual superheat temperatures approx. 4 to 6 K over the evaporating temperature. The outlet temperature of the cooling air of condenser is approximately like the condensing temperature due to the cross flow character of the condenser and the influence of the refrigerant superheating.

The subcooling line in the condenser produced a subcooling of 13 K in the design point of the normal cooling cell aggregate and led to an improvement of the COP (t_o = -10 ℃; t_c = 42℃) from 2,50 to 2,69. The air temperature at the inlet of the condenser was 24,7℃ whose outlet temperature 42℃. Without subcooling line no subcooling was ach ieved.

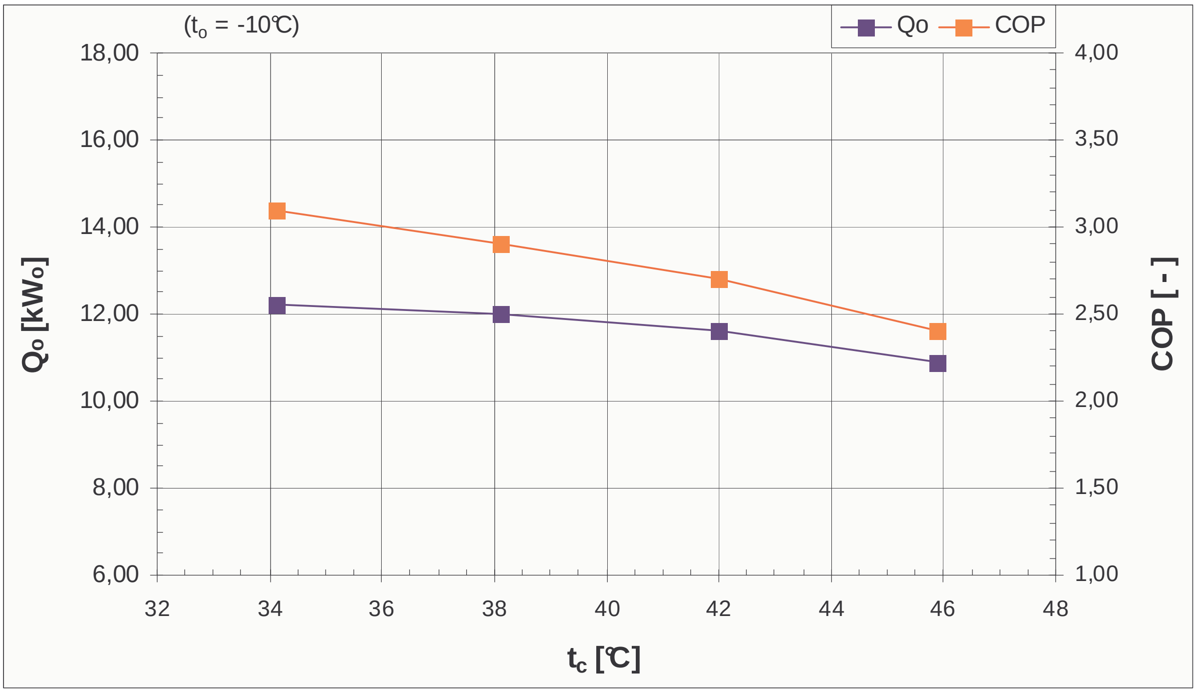
The electrical fan performance of the condenser was 414 W_{el} at maximum rotation speed that means 0,037 kW_{el}/kW_o for the design point. The values are analogous to that for the evaporator fan 1010 W_{el} and 0,090 kW_{el}/kW_o.

The figures 5.21 and 5.22 show the characteristic for refrigerating capacity and COP for different condensing and evaporating temperatures. It is clear that the measured refrigerating capacity is above the computed one.

Table 5.11: Plan of measuring places refrigerating aggregate cooling cell

name		Symbol	measuring place	note
EIR 01	kW	$P_{el,Comp}$	electric capacity compressor	instrument WT 130
PI 04	bar	p_c	condensation pressure	manometer
PIR 05	Pa	p_c	condensation pressure	pressure sensor 0…25 bar (a) PDRB Fa. Baumer
TIR 07	℃	$t_{out,comp}$	temperature outlet compressor	PT 100 – immersing feeler (60…180℃)
PC 08	bar		controller condensate pressure (adjustment p_c)	
FIR 09	kg/s	m_{Refr}	refrigerant mass flow condensate	RHM 03 Fa. Rheonik
PIR 10	bar	$p_{c,o}$	pressure in front of expansion valve	pressure sensor 0…25 bar (a) Fa. WIKA Tronic
TIR 11	℃	$t_{c,o}$	temperature condensate in front of expansion valve	PT 100 – immersing feeler (20…40℃)
TIR 12	℃	$t_{o,oh}$	temperature refrigerant outlet evaporator	PT 100 – immersing feeler (-30…10℃)
PI 13	bar	p_o	evaporation pressure	manometer
PIR 14	Pa	p_o	evaporation pressure	pressure sensor 0…10 bar (a) Fa. WIKA Tronic
TC 16	℃		cooling controller	PT 1000
TIR 17/ 1...3	℃	$t_{air,o,in,i}$	air temperature cooling cell inlet evaporator	PT 100 – air feeler, 3 x, average (-25…5℃)
TIR 17/ 4...6	℃	$t_{air,o,out,i}$	air temperature cooling cell outlet evaporator	PT 100 – air feeler, 3 x, average (-25…5℃)
TIR 20/ 1...3	℃	$t_{air,c,in,i}$	ambience air temperature inlet condenser	PT 100 – air feeler, 3 x, average (20…40℃)
TIR 20/ 4...6	℃	$t_{air,c,out,i}$	ambience air temperature outlet condenser	PT 100 – air feeler, 3 x, average (20…50℃)
PIR 21	bar	p_{air}	ambience air pressure	pressure sensor 0…25 bar (a)
EI 22	kW	$P_{el,Fan,c}$	electric capacity fan condenser	mobile electrical pliers
EI 23	kW	$P_{el,Fan,o}$	electric capacity fan evaporator	mobile electrical pliers
TIR 24	℃	$t_{c,uc}$	temperature condensate exit condenser	PT 100 – surface feeler (20…40℃)
TIR 25	℃	$t_{oh,Comp,in}$	temperature entree compressor	PT 100 - surface feeler (-20…10℃)
TIR 26	℃	$t_{out,Comp,1}$	temperature compressor outlet piston 1	thermoelement - surface feeler (-20…200℃)
TIR 27	℃	$t_{out,Comp,2}$	temperature compressor outlet piston 2	thermoelement - surface feeler (-20…200℃)
TIR 28	℃	$t_{out,Comp,3}$	temperature compressor outlet piston 3	thermoelement - surface feeler (-20…200℃)

Figure 5.21: Refrigerating capacity and COP of refrigerating aggregate normal cooling cell in dependence of condensing temperature at -10℃ evaporating temperature



In the figures 5.23 and 5.24 the compressor outlet temperature of the piston 2 is compared with the isentropic computed one and with the computed real discharge temperature as well as the isentropic efficiency of the normal cooling cell aggregate in dependence of the condensing temperature respectively the pressure ratio at -10 ℃ evaporating temperature is represented. The Co. Frigopol declares for his separating hood refigerant compressor a maximal permissible compressor outlet temperature of 130℃, without consideration of safety addition of 140℃. It is clear that this value was not reach at the warmest piston and also at 46°C condensing temperature. Since the real discharge temperature at the pressure valve of the compressor wasn´t measured this was determined with a cycle computing. The increase of the isentropic efficiency with the pressure ratio is marginal in the figure 5.24.

Figure 5.22: Refrigerating capacity and COP of refrigerating aggregate normal cooling cell in dependence of evaporating temperature at 42℃ condensing temperature

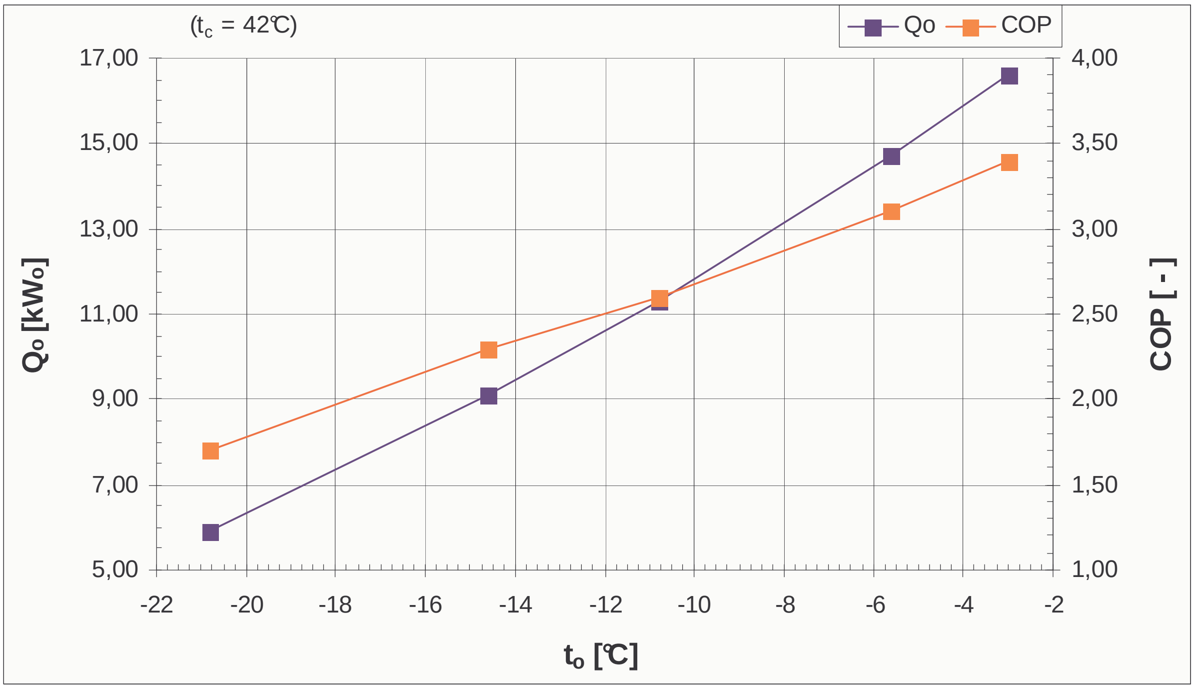


Figure 5.23: Temperature outlet piston 2, computed isentropic and real discharge temperature as well as isentropic efficiency of refrigerating aggregate normal cooling cell in dependence of condensing temperature at -10℃ evaporating temperature

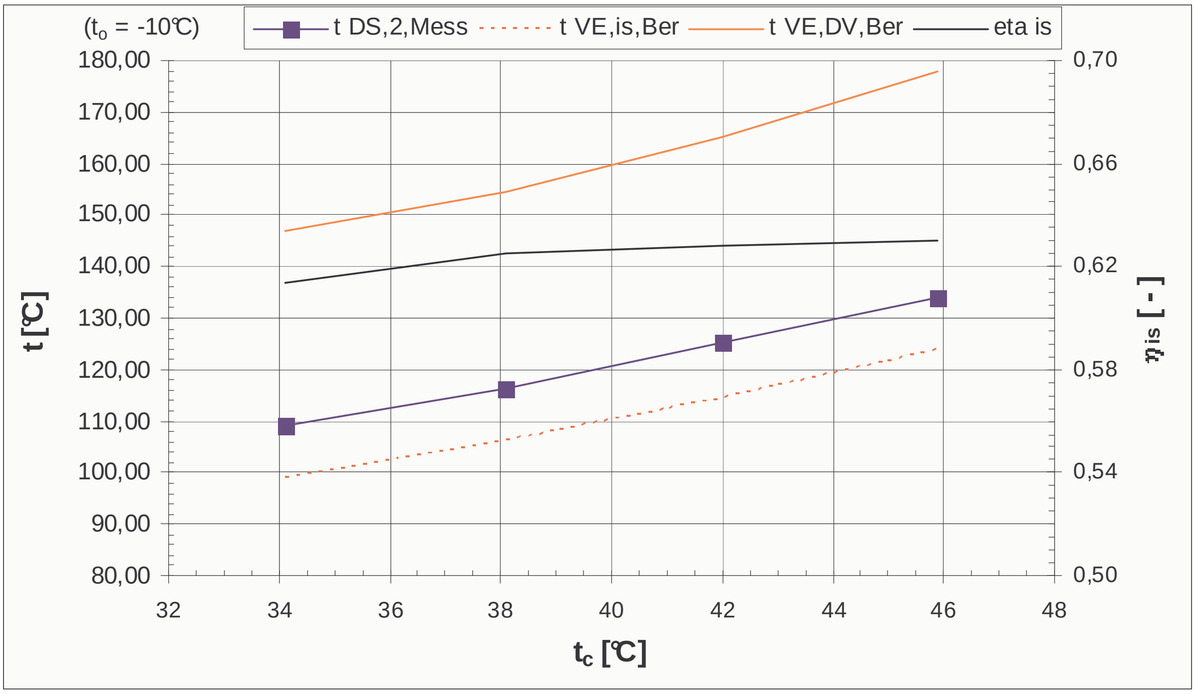
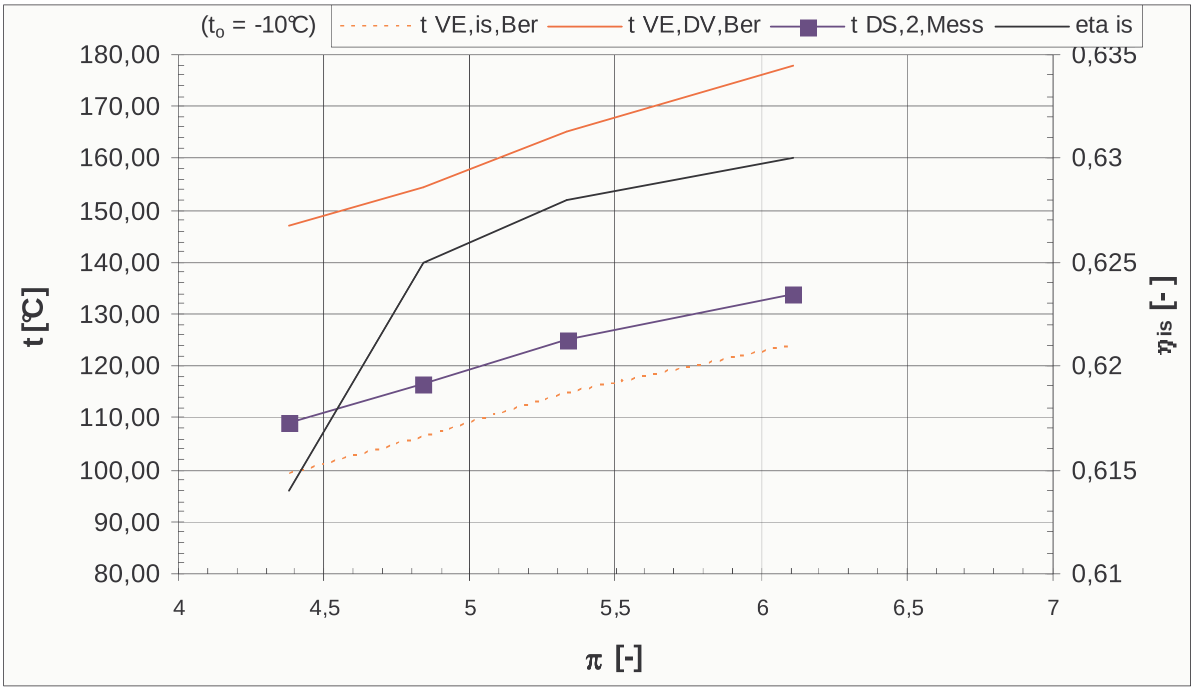
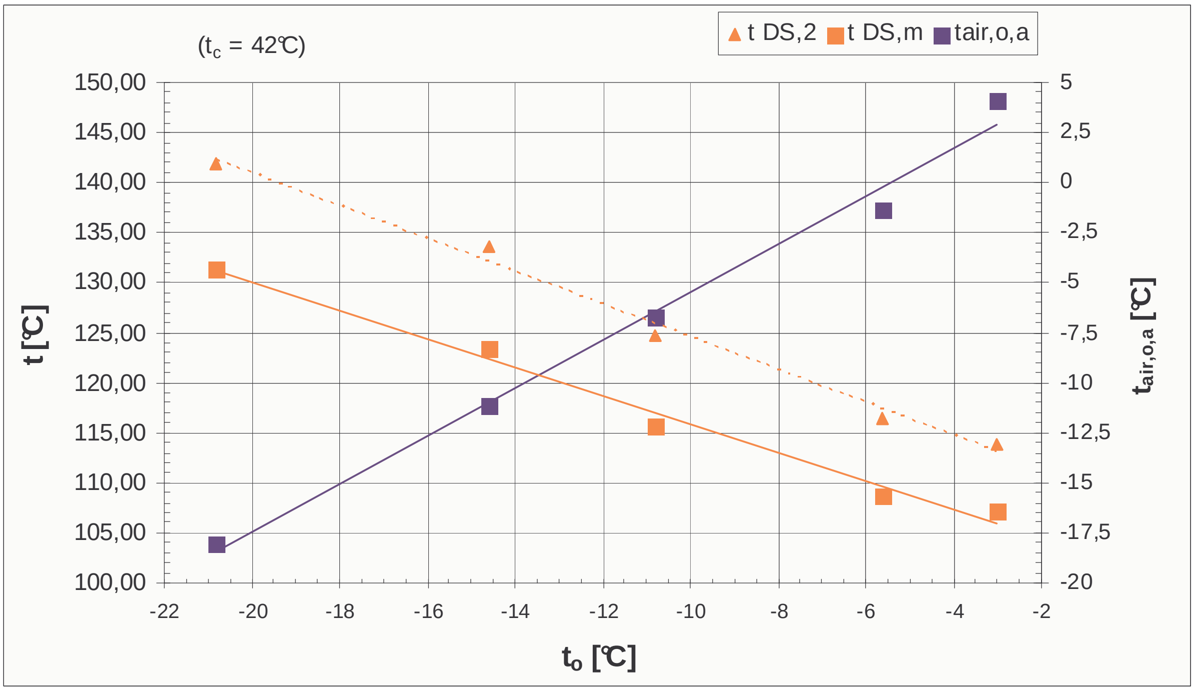


Figure 5.24: Temperature outlet piston 2, computed isentropic and real discharge temperature as well as isentropic efficiency of refrigerating aggregate normal cooling cell in dependence of pressure ratio at -10℃ evaporating tempera ture



In figure 5.25 is to see that the middle compressor outlet temperature of all 3 pistons is approx. 10 K under that of the warmest piston and the cold air outlet temperature from the evaporator is only about 4 to 6 K over the evaporating temperature.

Figure 5.25: Middle compressor outlet temperature and for piston 2 as well as outlet temperature air from the evaporator for refrigerating aggregate normal cooling cell in dependence of evaporating temperature at 42℃ condensing tempe rature



The isentropic efficiency is represented in the picture 5.26 for different conditions of the condensing and evaporating temperature in dependence of the pressure ratio, in analogical mode to that in the picture 5.27 the delivery rate of the compressor type 19.

Figure 5.26: Isentropic efficiency in dependence of pressure ratio for different conditions for refrigerating aggregate normal cooling cell

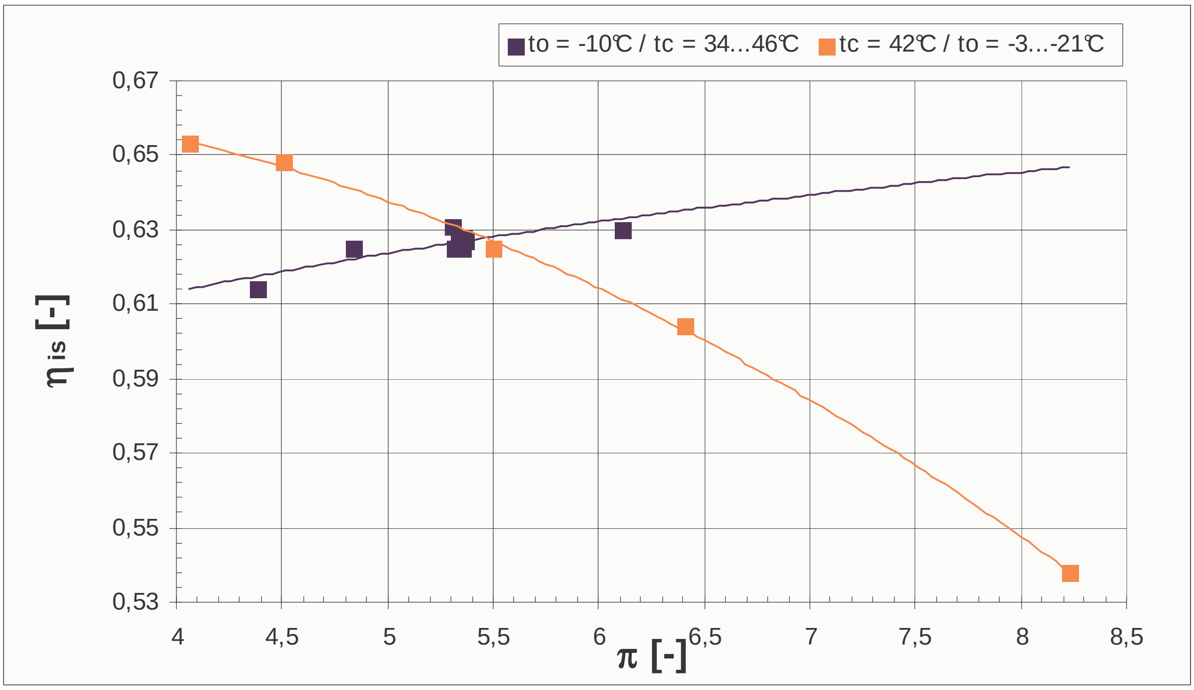
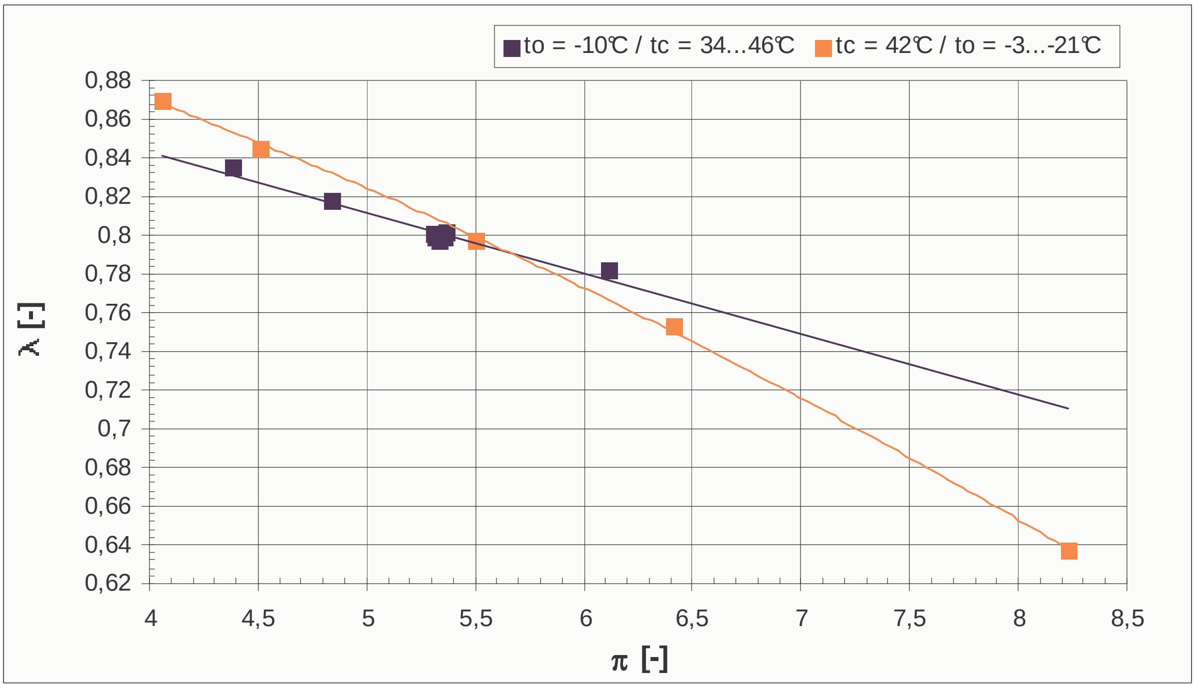
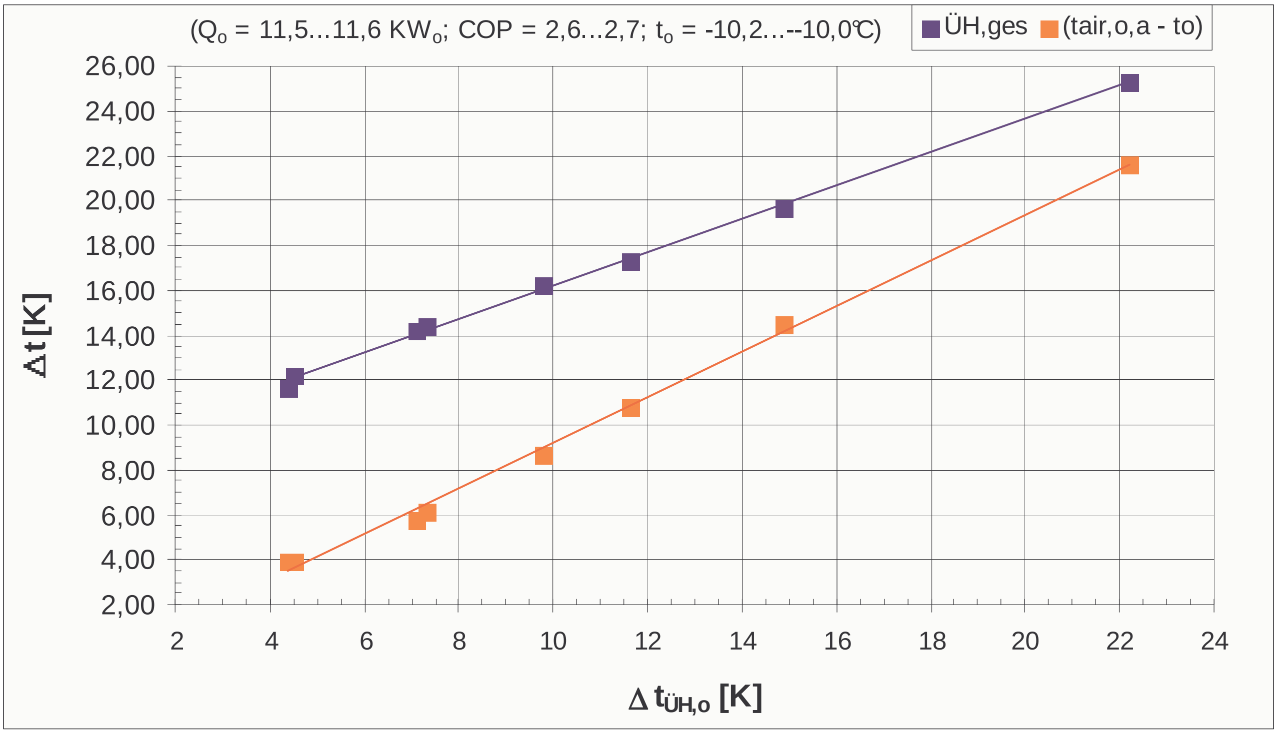


Figure 5.27: Delivery rate in dependence of pressure ratio for different conditions for refrigerating aggregate normal cooling cell



The total superheating which results from an evaporator superheat part and a suction line superheat part and the difference between evaporating and air outlet temperature from evaporator are represented in the figure 5.28. Due to the long suction line the superheat part is relatively great in that. For the evaporator a minimum superheating was necessary of 5 ... 6 K in order to be drop-free before the compressor. The superheat of the refrigerant vapour to the evaporator as well as the surface of the drops carried away in the vapour flow do not suffice in order to evaporate the refrigerant remaining liquid to the evaporator.

Figure 5.28: Total superheating and difference between evaporating and temperature of air outlet from evaporator for refrigerating aggregate normal cooling cell



The thermostatic expansion valve else authorized only for conventional refrigerants since 3 small brass components are integrated in the valve worked as reliably as the NH3-TEV. Also after 6 months integration of the valve in the ammonia atmosphere no derogation of function was detected. In case of total disassemble of the valve after the measurement period no corrosion could be analyzed at the brass parts. The electronic expansion valve (EEV) was to be adjusted heavily and would had have to be little larger designed in the conversion of the ammonia performance on R723. R723 has a higher density, however also a smaller massspecific refrigerating capacity in opposite to R717. Through that the volumetric refrigerating capacity of both is almost identical. For R723 a higher refrigerant mass flow is necessary. With the EEV very stable superheat temperatures were achieved. Since at R723 as at R717 with small superheat temperatures liquid drops are in the suction line, can the advantage of electronic expansion valves adjusting stable small superheating not to be used.

Figure 5.29 shows the courses of the refrigerant temperature to evaporator along the way to outlet of the compressor.

Figure 5.29: Course of refrigerant temperature from the evaporator to compressor outlet in refrigerating aggregate normal cooling cell

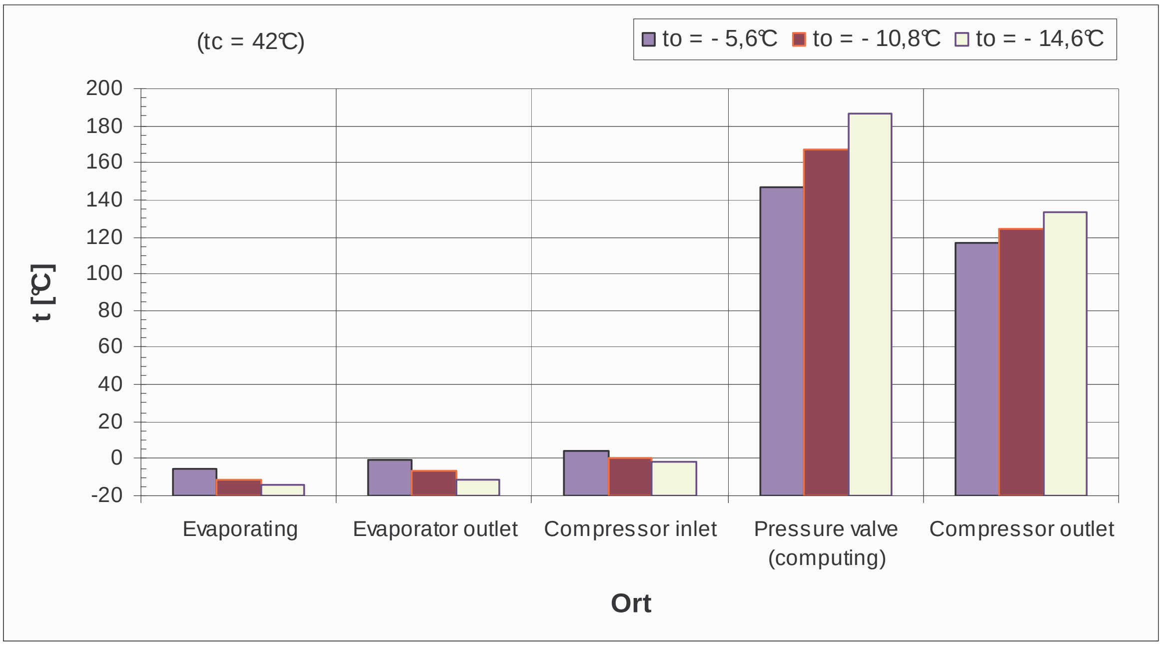
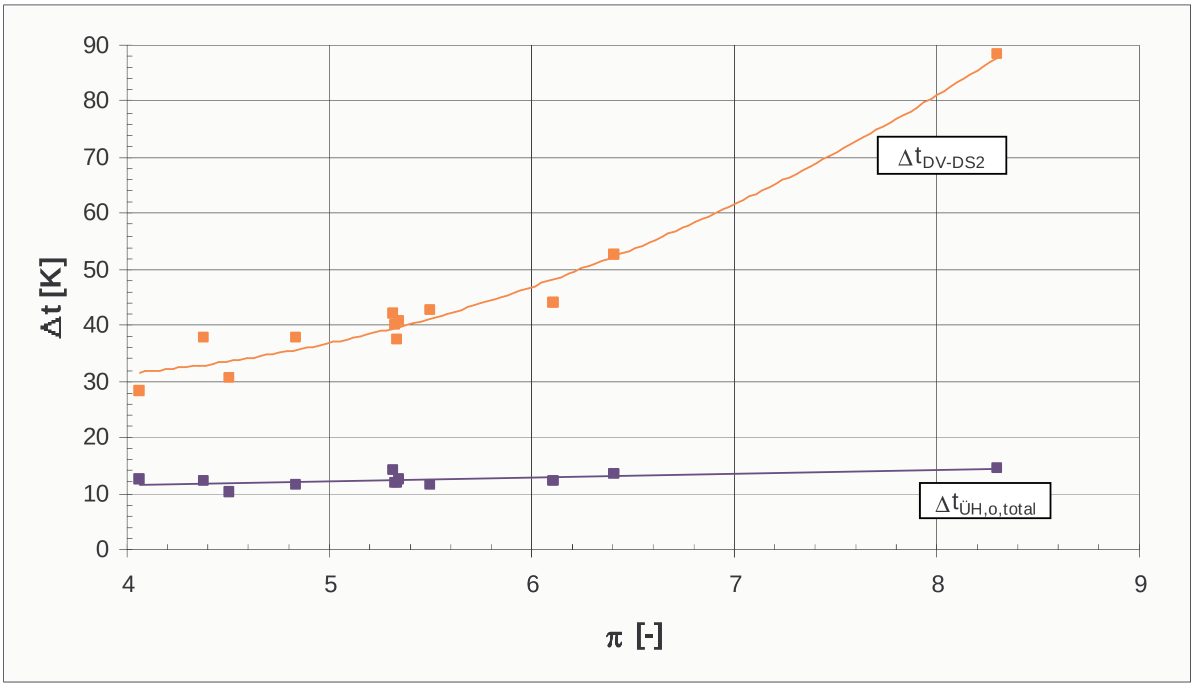


Figure 5.30: Total superheating and difference between discharge and compressor outlet temperature for refrigerating aggregate normal cooling cell in dependence of pressure ratio



The real discharge temperature lies between 30 and 60 K about the compressor outlet temperature (figure 5.30). This is reasonable in the internal heat exchange between pressure and suction gas in the compressor. The temperature difference between pressure valve and compressor outlet is dependent on the pressure ratio.

A hermetical scrolling-compressor (compressor 3 and 4), a semihermetic reciprocating compressor (compressor 2) and a hermetic reciprocating compressor (compressor 1) were used for the comparison of the isentropic efficiency, the delivery rate and the COP to the semihermetic separating hood refrigerant compressor installed in the demo plant. These compressors were measured in the past in the ILK.

Into the figure 5.31 until 5.33 the comparisons are represented graphically. The flow values and used refrigerants can be taken from these graphics. It is clear that the Frigopol compressor with R723 has a higher COP. The isentropic efficiency and the delivery rate of the compressor are also very good in this comparison.

Figure 5.31: COP comparison different compressors with different refrigerants in dependence of evaporating temperature for 42℃ condensing tempera ture

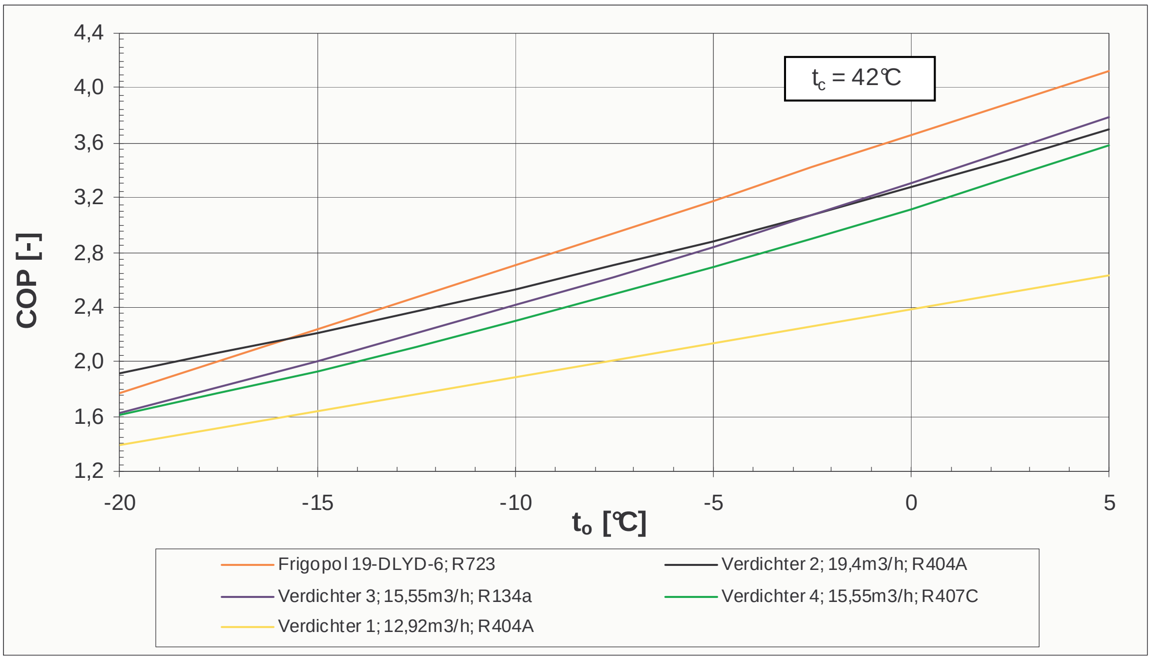


Figure 5.32: Comparison isentrope efficiency for different compressors with different refrigerants in dependence of evaporating temperature for 42℃ c ondensing temperature

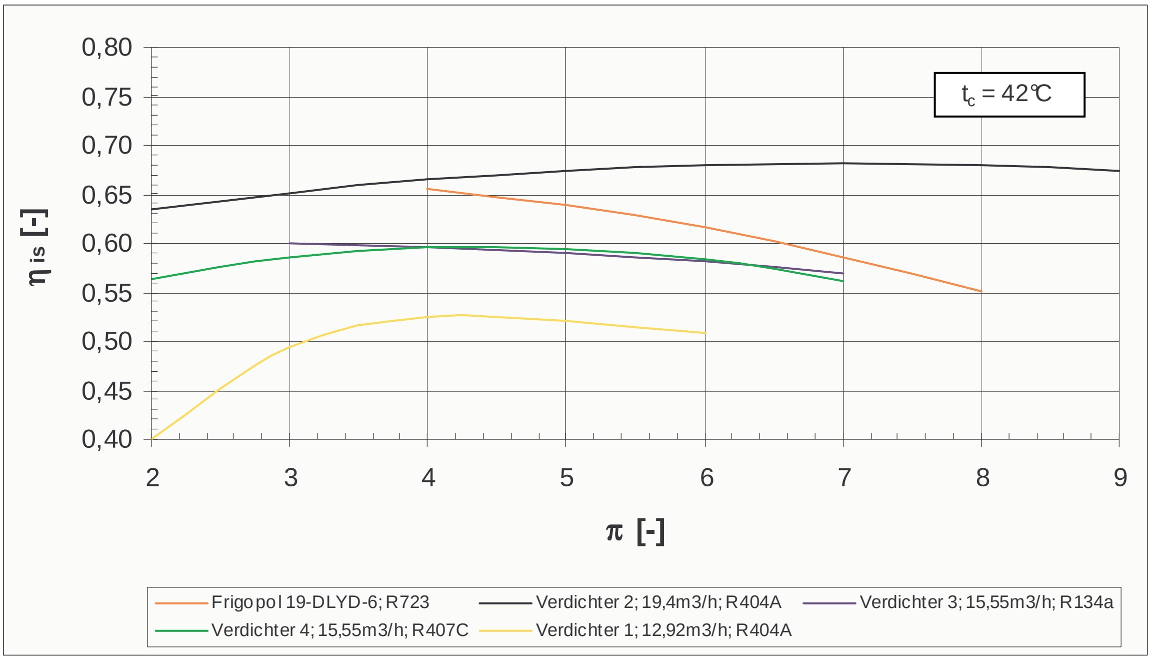
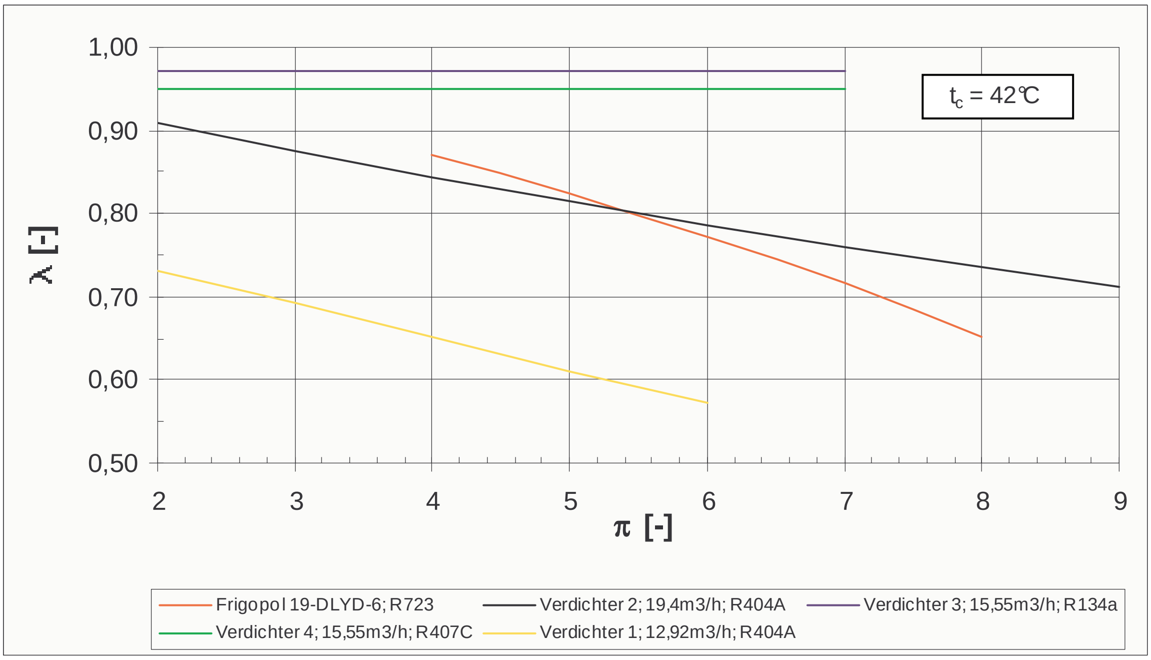


Figure 5.33: Comparison delivery rate for different compressors with different refrigerants in dependence of evaporating temperature for 42℃ conden sing temperature



After the measurement period inclusive the reconstructions a refrigerant and a refrigerating oil test were taken out the circuit and analyzed. The refrigerant investigation showed a humidity of 344 ppm which was below the permissible limit of 400 ppm with that in spite of repeated opening of the circuit. This was reasonable in after opening of the refrigerating circuit the evacuation period was sufficiently long. Also the oil investigation on completion of the test program led to the result that the oil furthermore usable and the lubricity is existent. Table 5.12 compiles the results of measurement of the oil investigation.

Table 5.12: Measuring results of the oil analysis

		Mobil SHC 226 E test plant	**Mobil SHC 226 E** delivery condition	**note.**
Colournumber	[-]	1	-	
Water content	[ppm]	168	150	nearly saturated
Kinematic viscosity 20℃	[mm ^{2}/s]	188	163	lubricity guaranteed
Kinematic viscosity 40℃	[mm ^{2}/s]	69,2	64	
Total contamination	[mg/kg]	559	-	normal (at hermetic plants about 200)
Basennumber (TBN)	[mgKOH/gÖL]	0,13	-	aging evident

Summary Results Demo Plant Normal Cooling Cell

- The refrigerating aggregate worked stable also without oil separator. The oil return was guaranteed. The performance amounted in design point 11,3 kW_o and laid form that higher as computed.
- The isentropic efficiency η_{is} (0,6...0,65) and the delivery rate λ (0,7...0,85) of Frigopol compressor Typ 19 are in comparison with conventional compressors very good.
- The COP of Frigopol compressor with the refrigerant R723 lies in comparison to conventional compressors and others refrigerants any higher, 2,7 to 2,5.
- The evaporator achieved with temperature differences between air outlet ($t_{air,o,a}$) and evaporating temperature (t_o) of 4 ... 6K good operating amounts.
- Following minimal evaporating temperatures at adherence of a maximal compressor outlet temperature were achieved:
  - $t_{o,min}$ = -20,8℃ ($t_{out,Comp}$ < 140℃) bei t_c = 42℃ ($t_{Air,out}$ = -18,1℃; Q_o = 5,9 kW_o – 52%)
  - $t_{o,min}$ = -23,4℃ ($t_{out,Comp}$ < 140℃) bei t_c = 38℃ ($t_{Air,out}$ = -21,5℃; Q_o = 5,7 kW_o – 50%)
- The subcooling tubes in condenser lead to a subcooling of 13K and enhance the refrigerating capacity by about 1 kW_o depending of the air inlet temperature in the condenser.
- The filling amount of refrigerating aggregate amounts 4,0 kg R723, that is 0,3 kg/kW_o;. The amount lies in general usual area for filling amounts but is in respect of a minimisation a little bit to high due to long condensate and suction line between evaporator and condenser unit.

- The only for conventional refrigerants approved TEV works as the NH3-TEV. The material analysis showed, that no internal corrosion of brass pieces could be found after 6 months influence of ammonia. The function ability was given.
- The electronic expansion valve brings towards the thermostatic no improvements. Little super heatings (< 6…8K) lead at R723 to liquid drops in the suction line alike at R717.
- The analysis of refrigerant and oil showed that no corrosion appeared in the system during the test time. The water content in refrigerant laid with 344 ppm under the limit value of 400 ppm.
- In future an oilheating would be practical due to the enrichment of refrigerant in refrigerating oil also at R723.

5.5 Deep Freezing Cell

The cool cell aggregate for the normal cooling cell was reconstructed. The compressor type 46 with external suction line was installed. The suction line coming from the evaporator is led in this case at first into a small separator vessel from which the suction line goes directly to the cylinder heads. At the lower vessel ground a line to the oil sump of compressor is exist for the oil return. This line was at first in 10 mm outer diameter. With deep evaporating temperatures and consequently high viscosity of the refrigerating oil the oil return was at this diameter not guaranteed anymore. This led in a operating point to an oil kick of the compressor. During the measurement of this compressor brought out that the compressor through that had taken damage in particular at the valve plates. The oil return was extended to the diameter 15 mm and built-in according to first measurements a new compressor.

The damaged compressor had as a result of the oil kick a raised oil shot. Thus too much oil reached the refrigerating circuit. The sensor of the TEV could not give stable control signals. Varying superheat temperatures and liquid injection into the suction line occurred. The compressor outlet temperature of the damaged compressor was relative low also at deep evaporating temperatures.

The figures 5.34 and 5.35 show this fact and serve as an example for a reduction of the discharge temperature through refrigerant injection into the suction line.

Figure 5.34: COP, delivery rate and isentropic efficiency for different evaporating and condensing temperatures with external suction gas line and liquid drops in the suction line for refrigerating aggregate deep freezing cell

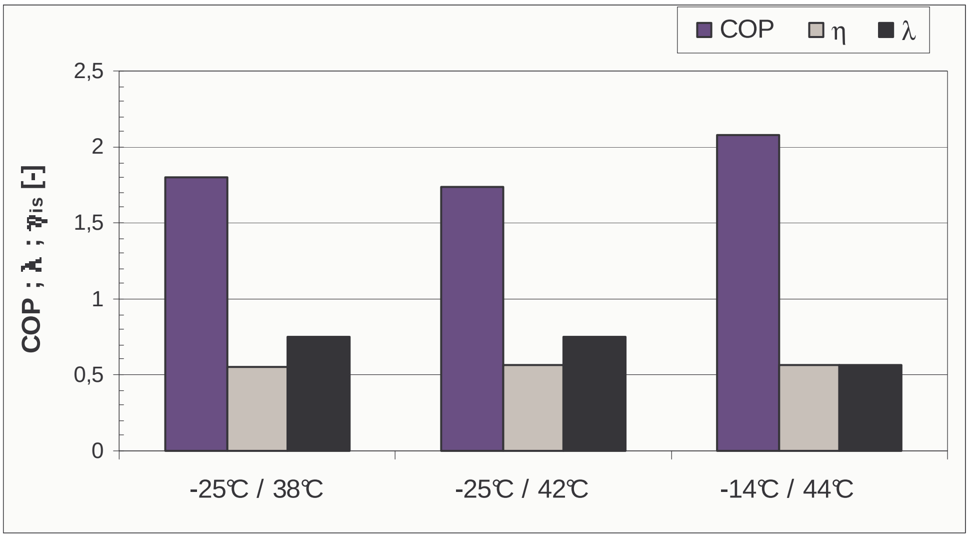
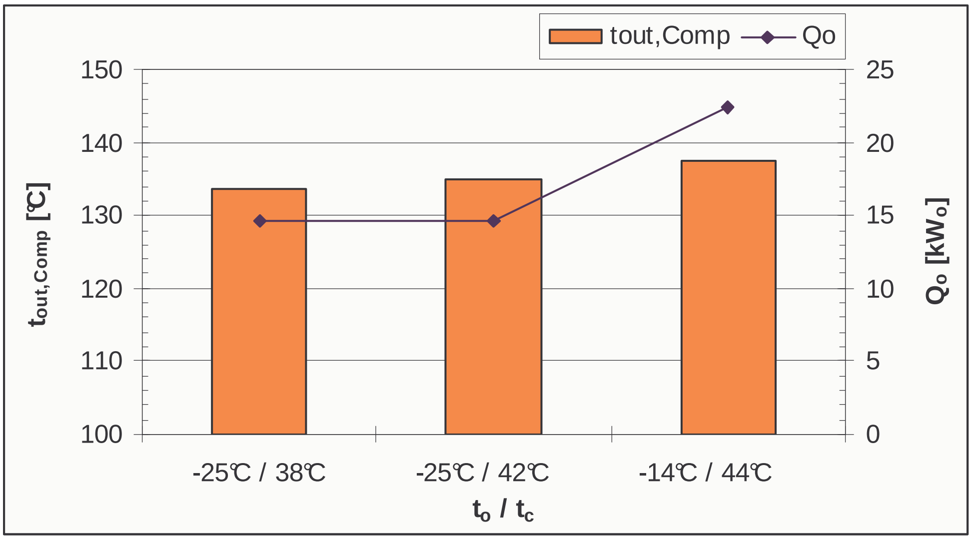


Figure 5.35: Refrigerating capacity and compressor outlet temperature for different evaporating and condensing temperatures with external suction gas line and liquid drops in the suction line for refrigerating aggregate deep freezing cell



After the installation of the new compressor higher compressor outlet temperatures were measured (figure 5.38) than with the first measurements of the first compressor. For the compliance of a compressor outlet temperature of maximally 140°C could only -15 °C evaporating temperature are gone. The minimal attainable evaporating temperature was with a condensing temperature of 34°C only -23 °C. Into the figure 5.36 and 5.37 the refr igerating capacity and the COP for different condensing temperatures are represented depending on the evaporating temperature. Since through the restriction of the compressor outlet temperature only higher evaporating temperatures could be realised in opposite to the design the refrigerating capacity was too high and consequently the condenser considerably too small. From that the characteristic curve planned in the test program could not measure for the deep freezing cell in the entirety.

Figure 5.36: Refrigerating capacity in dependence of evaporating temperature for different condensing temperatures with and without external suction gas line for refrigerating aggregate deep freezing cell

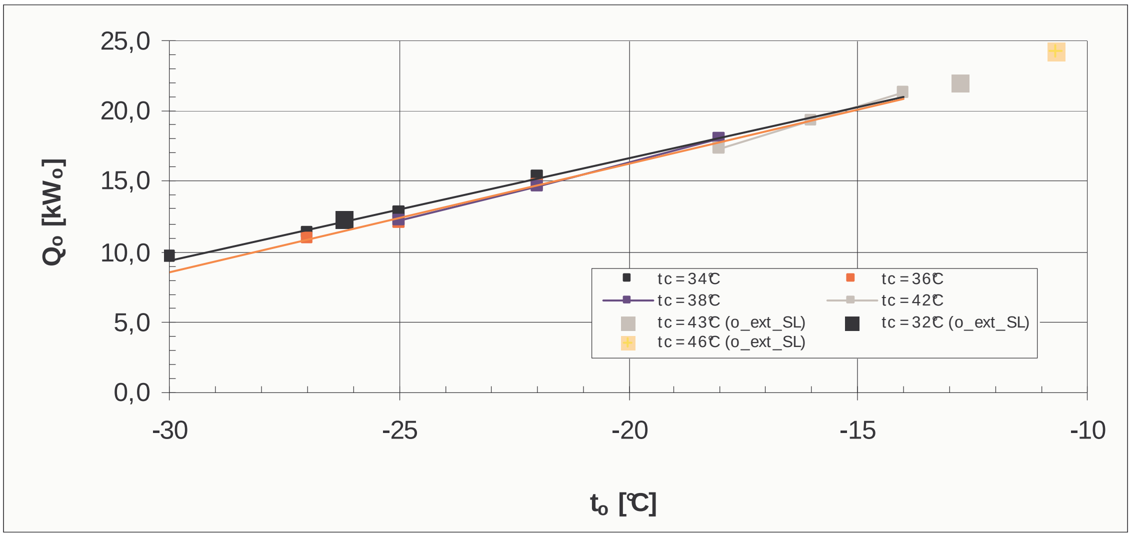
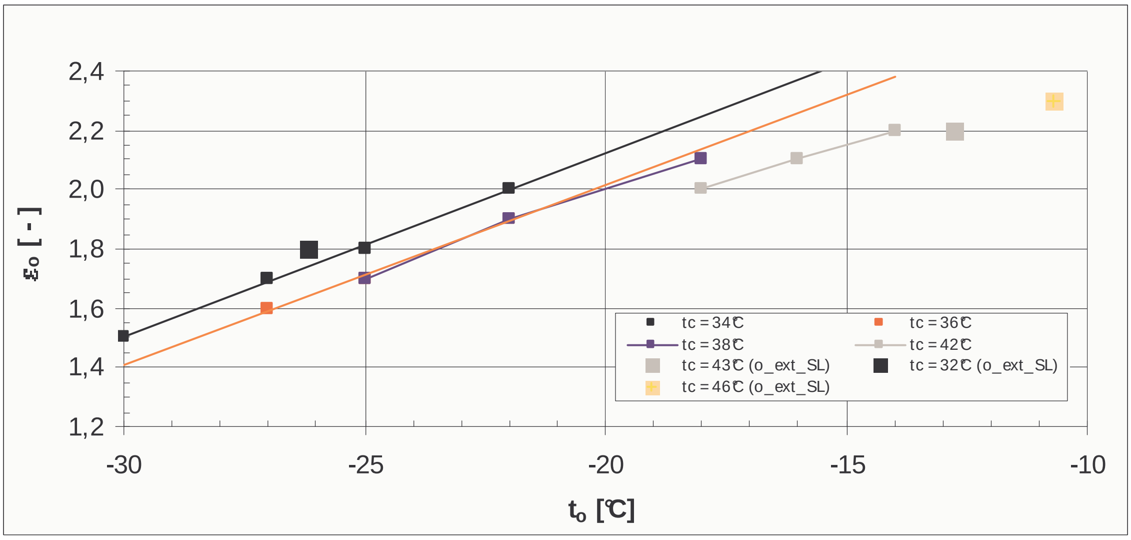
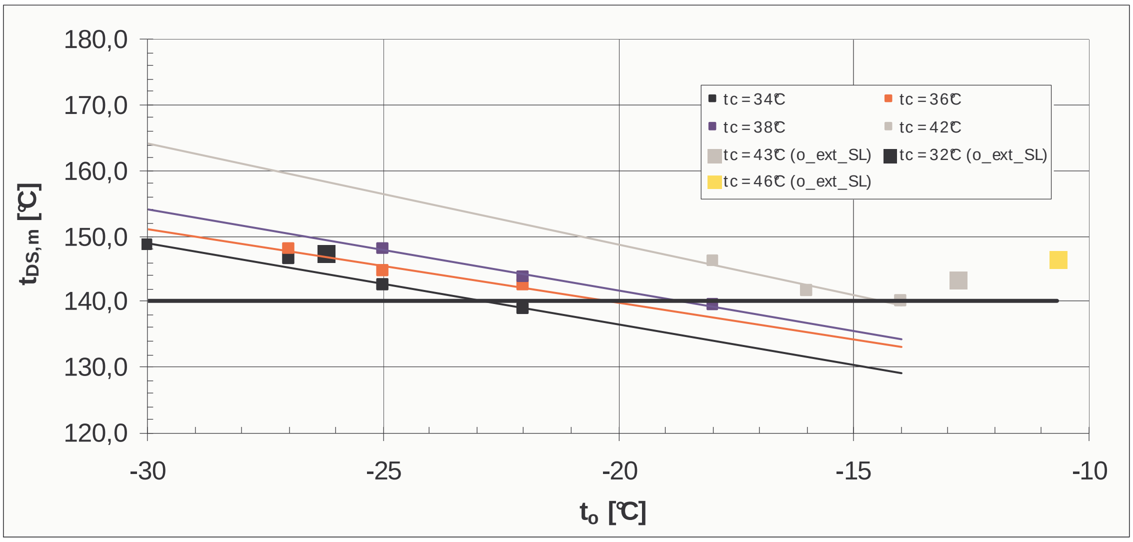


Figure 5.37: COP in dependence of evaporating temperature for different condensing temperatures with and without external suction gas line for refrigerating aggregate deep freezing cell



The measurements with internal and external suction line are represented in the following figures. The measurements led to almost same results. From figure 5.38 it is clear that the external suction line led only to a reduction of the compressor outlet temperature of approx. 5 K. The suction gas is indeed exposed at external suction line only of a smaller internal superheating happening in the compressor. But it cools no more the oil in the oil sump. Through that higher oil temperatures occur which leads in turn through the smaller lubricity of the oil to an increase of the discharge temperature. Thus can be estimated that both influences in their effects on the discharge temperature compensate.

Figure 5.38: Middle compressor outlet temperature in dependence of evaporating temperature for different condensing temperatures with and without external suction gas line for refrigerating aggregate deep freezing cell



The influence of liquid refrigerant in the suction line and consequently in the compressor inlet onto the compressor outlet temperature can be taken figure 5.39. If liquid refrigerant steps out of the evaporator the refrigerant temperature reduces at the compressor inlet and after that time-shifted the compressor outlet temperature. Yet also at the deep freezing were detected good isentropic efficiencies and delivery rates for the compressor type 46 (figure 5.40).

Figure 5.39: Time course of refrigerant temperature for evaporating, superheating as well as in- and outlet of compressor for refrigerating aggregate deep freezing cell at operating liquid refrigerant in the suction line

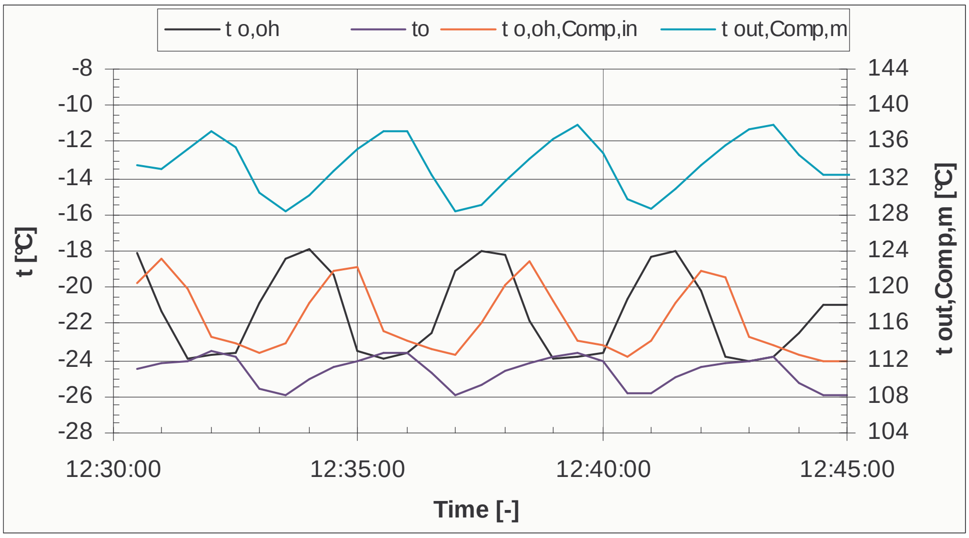
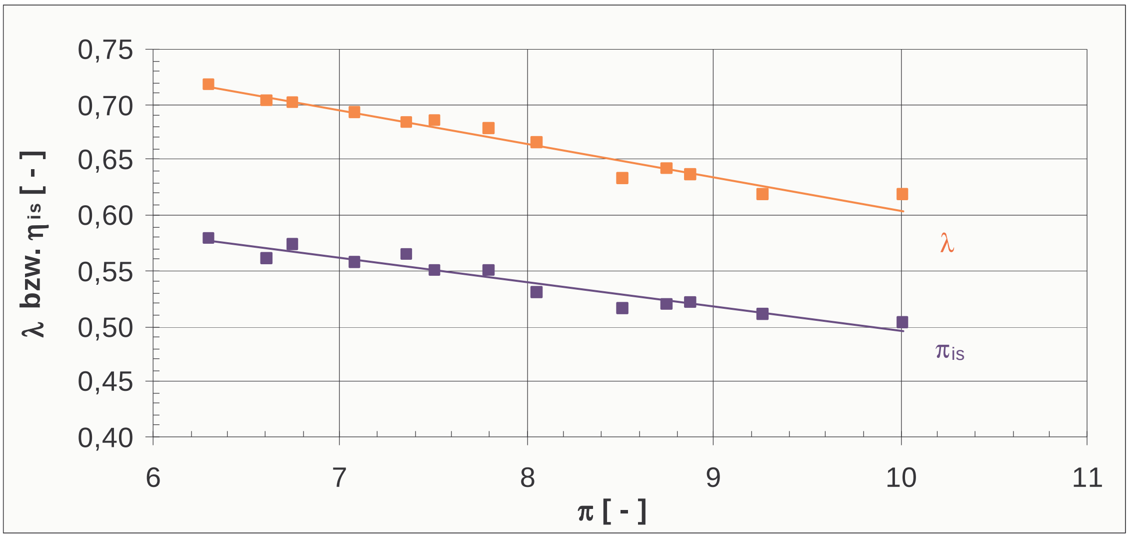


Figure 5.40: Isentrope efficiency and delivery rate for refrigerating aggregate deep freezing cell in dependence of pressure ratio



Summary Results Demo Plants Deep Freezing Cell

- A single-stage cold generation with R723 for deep freezing cells and 42℃ condensing temperature, cooling cell temperature -18℃, isn´t pos sible in cause of to high discharge temperatures.
- The design point with an evaporating temperatures of -30℃ wasn´t achieve due to higher compressor outlet temperature. The performance amounted at a condensing temperature of 36℃ and an evaporating temperature of -27℃ 11,0 k W_o.
- Following minimal evaporating temperatures at adherence of a maximal compressor outlet temperature were achieved:
  - $t_{o,min}$ = -18℃ ($t_{out,Comp}$ < 140℃) bei t_c = 42℃ ($t_{Air,out}$ = -8,8℃; Q_o = 17,3 kW_o)
  - $t_{o,min}$ = -27℃ ($t_{out,Comp}$ < 140℃) bei t_c = 36℃ ($t_{Air,out}$ = -20,2℃; Q_o = 11,0 kW_o)
- For pressure ratio $\pi > 7$ the isentropic efficiency η_{is} is with 0,5...0,57 and the delivery rate λ with 0,6...0,7 of Frigopol compressor Typ 46 in comparison with conventional compressors good.
- The COP of Frigopol compressor with the refrigerant R723 lies in comparison to conventional compressors and others refrigerants any higher.
- The subcooling tubes in condenser lead to a subcooling of 13K and enhance the refrigerating capacity by about 1 kW_o depending of the air inlet temperature in the condenser.
- The evaporator achieved with temperature differences between air outlet ($t_{air,o,a}$) and evaporating temperature (t_o) of 4 ... 6K good operating amounts.

- The filling amount of refrigerating aggregate amounts 4,0 kg R723. The amount lies in general usual area for filling amounts but is in respect of a minimisation a little bit to high due to long condensate and suction line between evaporator and condenser unit
- The only for conventional refrigerants approved TEV works as the NH3-TEV. The material analysis showed, that no internal corrosion of brass pieces could be found after 6 months influence of ammonia. The function ability was given.
- The electronic expansion valve brings towards the thermostatic no improvements. Little super heatings (< 8K) lead at R723 to liquid drops in the suction line alike at R717.
- The analysis of refrigerant and oil showed that no corrosion appeared in the system during the test time. The water content in refrigerant laid with 344 ppm under the limit value of 400 ppm. In the oil was the moisture content at 168 ppm.

5.6 Brine Chiller

Table 5.13 contains the measuring plan for the brine chiller. The measuring dates were registered and stored with the registration system Labview.

Following **Measuring Program** was realised:

- Determination optimal refrigerant filling amount and adjustment HP- and LP-keeper, overheating temperature, volume flow brine circle
- Measuring of plant at conditions of calculation t_o and t_c
- Measuring with variation condensation and evaporation pressure
- Fixing minimal achievable evaporation temperature at adherence maximal possible compressor endtemperature and t_c = 34°C
- Measuring influence overheating temperature to evaporator (calculation 6K) on compresser endtemperature ($\Delta t_{ÜH}$ = 2K, 4K, 6K, 8K; further criterions: drops refrigerant, regulation stability refrigerant evaporator)

The compressors F4 NH3 was lubricated by the mineral oil Reniso KC 68. The refrigerant distribution with injection into the bottom of the plate evaporator led to considerable problems. The causes are the distribution of the small refrigerant mass flow because of the high mass-specific refrigerating capacity at R717 and also R723 and the point, that the superheat of the refrigerant vapour does not suffice in order to evaporate liquid drops behind the evaporator.

Table 5.13: Plan of measuring places brine chiller

name		Symbol	measuring place	note
EIR 01	kW	$P_{el,Comp}$	electric capacity compressor	instrument WT 130
PI 04	bar	p_c	condensation pressure	manometer
PIR 05	Pa	p_c	condensation pressure	pressure sensor 0…16 bar (a)
TIR 07	℃	$t_{Comp,out}$	temperature inlet compressor	PT 100 – immersing feeler
FIR 08	kg/s	m_{Refr}	refrigerant mass flow condensate	RHM 06 Fa. Rheonik
PIR 09	bar	$p_{c,o}$	pressure in front of expansion valve	pressure sensor 0…16 bar (a)
TIR 10	℃	$t_{c,o}$	temperature condensate in front of expansion valve	PT 100 – immersing feeler
TIR 11	℃	$t_{o,oh}$	temperature refrigerant outlet evaporator	PT 100 – immersing feeler
PI 12	bar	p_o	evaporation pressure	manometer
PIR 13	Pa	p_o	evaporation pressure	pressure sensor 0…10 bar (a)
TIR 15	℃	t_u	ambience air temperature	PT 100 – air feeler
PIR 16	bar	p_u	ambience air pressure	pressure sensor 0…6 bar (a)
FIR 17	kg/s	m_{Brine}	brine mass flow	CMF 100 Fa. micromotion
TIR 18	℃	$t_{Br,o,in}$	brine inlet evaporator	PT 100 - immersing feeler
TIR 19	℃	$t_{Br,o,out}$	brine outlet evaporator	PT 100 - immersing feeler
TIR 20	℃	$t_{CW,BC,in}$	cooling water inlet back cooler	PT 100 - immersing feeler
TIR 21	℃	$t_{CW,c,in}$	cooling water inlet condenser	PT 100 – immersing feeler
TIR 22	℃	$t_{CW,c,out}$	cooling water outlet condenser	PT 100 - immersing feeler

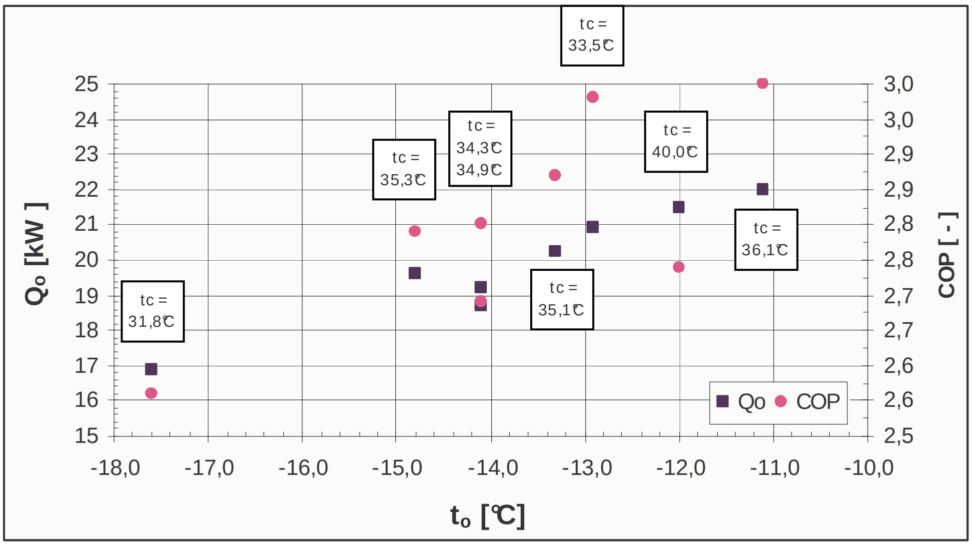
After the first experiments a conversion of system was occurred in case of which a tube plate evaporator of the Co. Vahterus was fitted. The evaporator was 4-floated designed on the refrigerant side and the injection occurred from below. In spite of the increase of the refrigerant mass flow density and the increase of the thermal length of the evaporator no stable operation could be reached also with the tube plate evaporator with justifiable superheating temperatures.

Depending on the often suction of fluid refrigerant into the compressor, the first compressor was broken down because of bearing damages, defective pressure valves and a connecting rod break. The oil necessary to the lubrication replaced by evaporating refrigerant and differences in pressure due to refrigerant vaporization influenced the oil transport. The compressor was replaced through one second identical dimension and fitted the initial plate evaporator however with a refrigerant distribution equipment again.

The refrigerant distribution equipment as a distributor lance with above installed outlet gaps for the refrigerant vapour and below for the refrigerant liquid brought improvements. However first through the incorporation of steel wool into the sectional view between expansion valve, these experiments were practised with a manual expansion valve, and distributor lance for the prevention of the separation of refrigerant vapour and liquid led to considerably more favourable circumstances. The temperature difference between brine outlet and evaporation sank from 8 K onto 4 K (Figure 5.43).

This improvement decreased with advancing working hour. It is estimated that this problem associates with a high oil throw of the compressor and a bad oil recirculation. The oil is collecting at the bottom of the evaporator and hinders the anyhow already problematic refrigerant distribution. At an estimated raised oil throw of 5 % concerning the refrigerant mass flow the oil mass flow circulating in the refrigerating circuit amounts to approx. 0.07 l/min (approx. 2 l/h). The pair of refrigerant R723 and refrigerant oil KC68 is not suitable for the application of plate-type evaporators with injection at the bottom. That means that after some time, just found in the operation, almost the entire oil was flung from the compressor. In this case by use of plate evaporators with R723, an oil deposit can not be refused on.

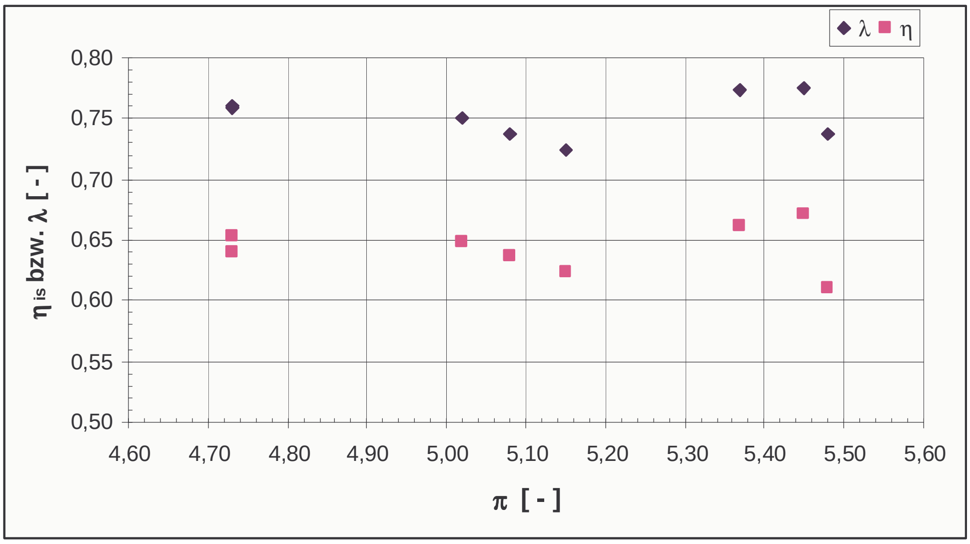
Figure 5.41: Refrigerating capacity and COP for different evaporating and condensing temperatures of the brine chiller



Without refrigerant distribution equipment the liquid freedom in front of the compressor could be reached only from superheating temperatures of >12°C. This value sank onto approx. 7 K with refrigerant distribution equipment.

With regard to the refrigerant enrichment in the refrigerant oil an oil-fired heating (80 W) and an oil differential pressure counter also at R723 are necessary for the guarantee of the compressor lubrication. The charge of the brine cooler was only 1.7 kg of R723, that is approx. 0.1 kg/kW$_o$. The refrigerating capacity was higher in the design point with 19.2 kW$_o$ than calculated (Figure 5.41). The compressor F4 NH3 of Bock company achieved with η_{is} of 0.65 and λ of 0.75 good values (Figure 5.42).

Figure 5.42: Isentropic efficiency and delivery rate of brine chiller in dependence of pressure ratio

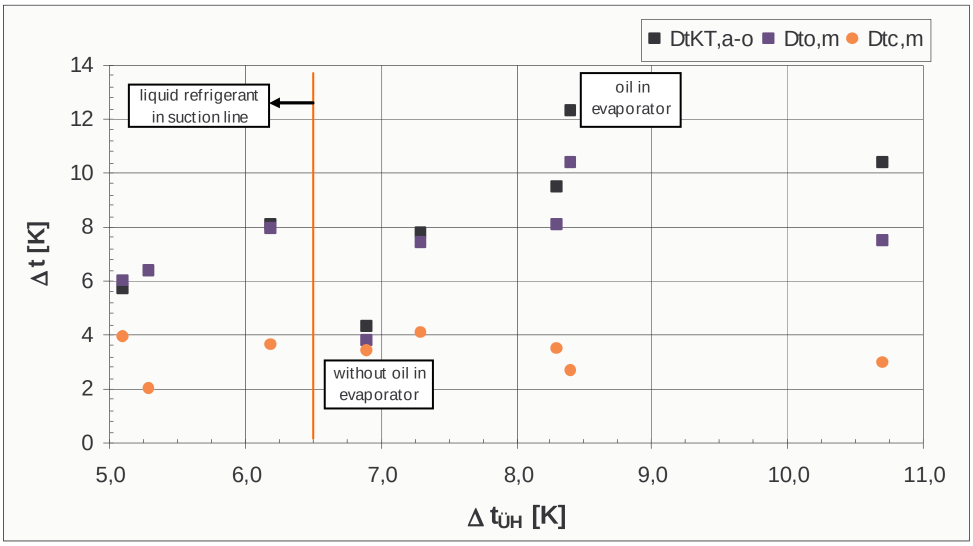


Summary Problem dry or flooded Evaporation

R723 is also not optimally for an expansion valve operation with dry evaporation. A perfect refrigerant distribution is both at shell-and-tube and at plate evaporators with expansion valve operation problematic. The unequal admission of the evaporator surface with fluid refrigerant leads to the degradation of the efficiency rate. The injection amount is very small because of opposite that one for example R22 approx. 7x larger evaporation heat, what intensifies the distribution problem besides. The superheating must be at least 8 to 10 K for a stable signal at the expansion valve sensor, that is with given cooling agent temperatures the evaporation temperature must be relatively low, which results in a higher energy consumption. In the partial load operation this problem amplifies still. In case of load alternations the average superheating is rather too great due to the post-control of the injection valve and the evaporation temperature too low, that reduces the total efficiency rate. With the application of piston compressors the necessary suction gas superheating furthermore keeps on increasing the anyhow already high pressure gas temperature. The wear increases.

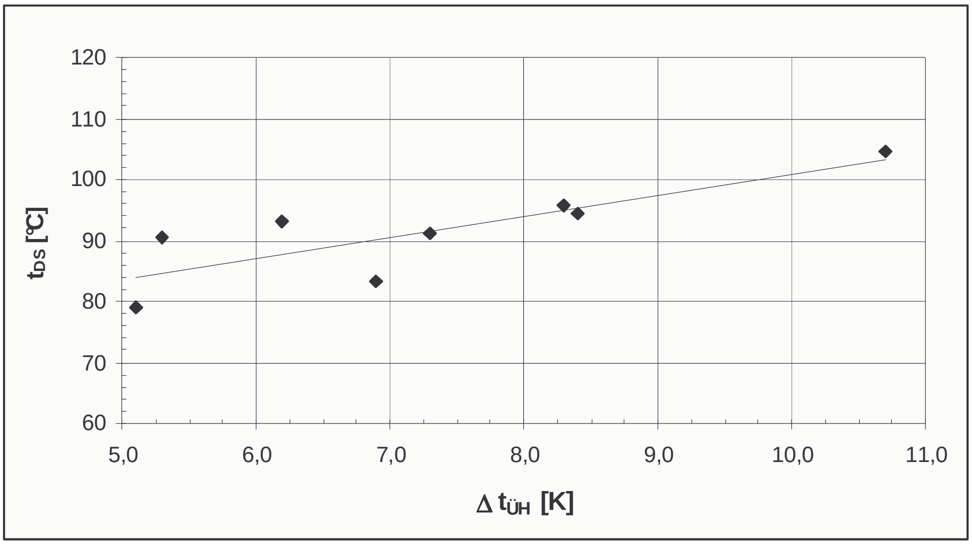
On the other hand a system with flooded evaporator, after-switched liquid separator and oil recirculation has less problems. In the task part of the Danish project partner a sampler with intern oil recirculation was developed concerning that and tested. Since no superheating has to be guaranteed, the mean temperature difference can be held in an any small way, which is advantageously in particular in the partial load operation. A smaller power consumption through higher evaporation temperatures in the whole load range is the result. A better utilization of the heat-exchanger surface brings uniform refrigerant distribution in the evaporator.

Figure 5.43: Differences of temperature brine outlet from evaporator to evaporating as well as middle logarithmic temperature differences for evaporator and condenser of brine chiller



The compressor outlet temperatures lay as expected always under the boundary values. The compressor outlet temperature shows picture 5.24 depending on the superheat temperature.

Figure 5.44: Compressor outlet temperature in dependence of superheating temperature of brine chiller



Summary Results Demo Plants Brine Chiller

- The refrigerating capacity laid in design point with 19,2 kW_o higher as computed.
- The Bock compressor F4 NH3 achieved with η_{is} of 0,65 and λ of 0,75 good amounts.
- The filling amounts of the brine chiller amounted only 1,7 kg R723, about 0,1 kg/kW_o.
- The oil shot of compressor is estimated as high in combination with this plant and the combination refrigerant and refrigerating oil.
- The combination refrigerant R723 and refrigerating oil KC68 isn´t applicable for the using of plate evaporators with under injection. In these case is an oil separator required. Here is further development demand necessary.
- Regarding the refrigerant enrichment in refrigerating oil an oilheating and an oil difference pressure switcher are necessary to the guarantee of the compressor lubrication also at R723.
- Without refrigerant distributor in plate evaporator liquid freedom could be achieved in front of the compressor only from superheating temperatures of > 12℃, with refrigerant distributor sank these amount onto about 7K.

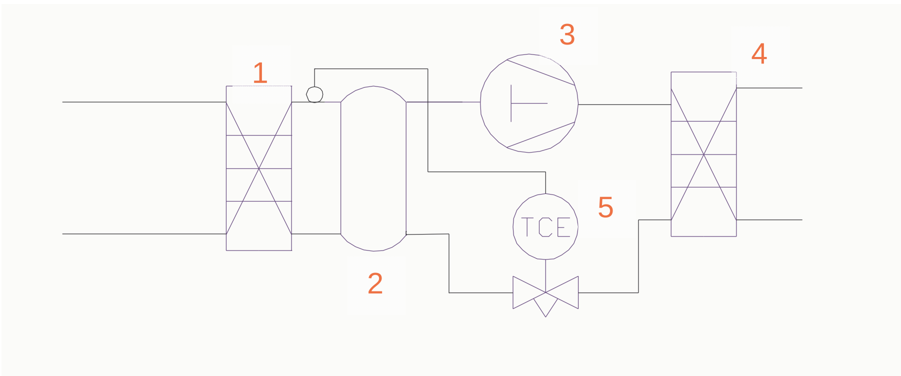
6 Results of Plant Optimation

Within the project and the measurement of the demo plants reconstructions and modifications at the demo plants were carried out. Optimizations and continuing development demands were derived from the results.

Refrigerant Distribution and Plate Evaporator

Onto the refrigerant distribution at plate evaporators was entered already in point 5.3 . A further possibility to increase the use of the plate evaporator area for the evaporation consists in separating liquid and refrigerant vapor after the expansion and admitting the evaporator only with liquid. The oil return from the simply built up separator vessel is realised via differences in pressure and injection. In figure 6.1 this equipment variant is outlined.

Figure 6.1: Basically plan liquid plate evaporator with combined separating vessel and oil return



1. Plate evaporator
2. Vessel with liquid collector, Flash Gas-Bypass and oil separator
3. Compressor
4. Plate condensor
5. TEV

In the work packages of the Danish project partner the test of these separator vessel was integrated. In the under chamber of the vessel the refrigerant vapor arising during the expansion in the expansion valve, the liquid refrigerant and the refrigerating oil were separated. The oil sinks to the ground. The swimming ball at the ground of the under chamber controls the oil level in the liquid refrigerant and gives free the oil return line to the suction line with oil accumulation. Into these the oil should be sucked through steam injection. The upper chamber of the vessel acts as a suction gas collector. The refrigerant gas is sucked from the upper field by the compressor. About the partition between upper and under chamber possible liquid refrigerant of the upper chamber can flow to the under chamber and thus to the evaporator inlet and in opposite direction refrigerant vapor flows from the under into the upper chamber. A swimming ball controls the stand of filling level of the liquid refrigerant in the under chamber.

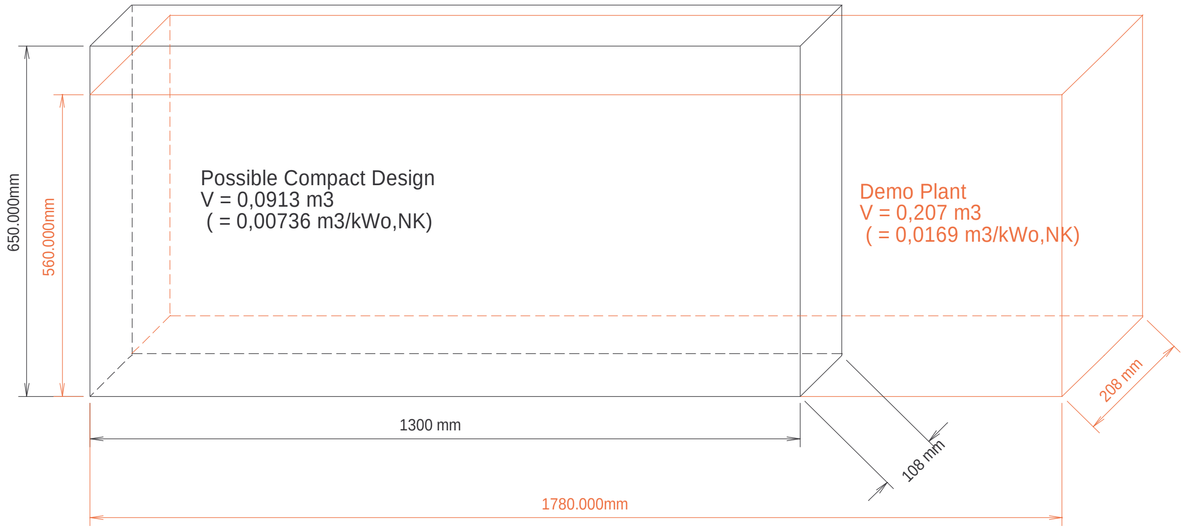
Cooling Cell Evaporator

The cool cell evaporator was optimized during the measurement by computing. The optimization computing with regard to a place and mass reduction of the cool cell evaporator for same refrigerating capacity and calculation conditions leads to the following results.

Optimation rib evaporator	**currently**	**optimized**
Rib distribution	7 mm	6 mm
Rib thickness	0,15 mm	0,2 mm
Tube distribution cross to air direction	14 tubes a 40 mm	26 tubes a 25 mm
Tube distribution in air direction	6 tubes a 34,64 mm	5 tubes a 21,65 mm
Tube diameter	12,5 mm x 0,35 mm	9,5 mm x 0,3 mm
Tube length	1,78 m	1,3 m
Heattransfer coefficient	37,2 W/m^2K	86,7 W/m^2K
Pressure loss	54,7 Pa	83,1 Pa
Fan performance	252 W	383 W
Mass	30 kg	21,5 kg
Vvolume	0,207 m^3	0,0913 m^3
Costs	1.900 €	1.600 €

The volume requirement of the evaporator can be reduced around approx. 56 % and the mass around approx. 28 %. Figure 6.2 shows the size comparison of the optimized rib evaporator of the cooling cell to the one used in the experimental plant at the present time.

Figure 6.2: Size comparison of an optimised rib evaporator for cooling cell in opposite with the evaporator used in test plant



7 Comparison of Economic Efficiency and Negotiability of Results

The NH_3 refrigeration plants have to compete with the conventional refrigeration plants, which are currently commercially available. Their specific codes of costs [€/kWo], space required [m^2/kW_o], mass [kg/kW_o] and refrigerant filling amount [g/kW_o] are target quantities of the NH_3 refrigeration plants.

By way of comparison, conventional plants for cool cell aggregates and the brine chiller today use the refrigerant R404A. R404A has a potential to greenhouse effect about GWP_{100} = 3260. Therefore, a considerable contribution to the environmental control can be reached by using coolant circuits, where the refrigerant charge amounts to 80 to 95% less than with the current direct evaporation systems. The decrease of the leakage-risks is combined with that. Next to the reduction of the refrigerant filling-amounts plants with coolant circuits offer the possibility to use refrigerants without greenhouse potential, as ammonia, for example in food markets.

For the NH_3- brine chiller a conventional plant was compared with the refrigerant R404A and a semi-hermetic compressor. This plant has with the determined operating conditions (t_o = -12 °C; t_c = 35°C) a COP of 2.57, weighs 600 kg, has a space r equired of 1.2 m^2 and costs approx. 13,000 €. With the capacity of 21.5 kW_o following specific values of 605 €/kW_o , 0.056 m^2/kW_o and 27.9 kg/kW_o arise. The specific refrigerant charge at the comparison plants amounts to approx. 400 g/kW_o (see table 7.1).

Conventional aggregates for the cool cells mainly use the refrigerant R404A. For the normal cooling costs a complete condenser unit with high performance evaporator costs approx. 4.500 €, for the deep-freezing 8.750 €. The COP of the used comparison plants amount to approx. 2,5 and to 1,45 for deep freezing. The cooling aggregates are available with hermetic or semi-hermetic compressors. The semi-hermetic compressors are approx. 10% lower with regard to COP. For the condenser unit of the deep freezing cell the following specific values of 951 €/kW_o, 0,136 m^2/kW_o and 29,5 kg/kW_o as well as 1.565 g/kW_o arise.

Table 7.1: Specific parameters of conventional refrigerating systems as comparison for the German demo plants

System	COP [-]	Costs [€]/[€/kW_o]	Place demand [m^2]/[m^2/kW_o]	Mass [kg]/[kg/kW_o]	Filling amount [kg]/[g/kW_o]
Brine Chiller, R404A W_o 21,5 kW, -12/+35°C	2,57	13.000 605	1,2 0,056	600 27,9	8,6 400
NC-Cell, R404A W_o 9,2 kW; -10/	2,1	5.100 555	0,7 0,08	220 23,9	3 330
DF-Cell, R404A W_o 9,2 kW	1,45	8.750 951	1,25 0,136	271 29,5	14,4 1.565

In table 7.2 the chances of negotiability of the individuals plants are represented summarizing.

Table 7.2: Possibilities of Negotiability of the Demo Plants

Plant	Note	Negotiability
Aggregate Normal Cooling Cell	technical realisation successful, energy saving evident, investment costs with conventional plants comparable	good negotiability after getting necessary numbers of pieces
Aggregate Deep Freezing	single-stage technical realisation with economic investment costs no achievable	no negotiability given
Brine Chiller	further development demand regarding oil return und refrigerant distribution necessary	chances of negotiability after close of optimation tasks

Since the refrigerating unit of the normal cooling cell achieved the best results technically, an efficiency comparison with units using traditional refrigerants was carried out. Table 7.3 shows the modification of the discharge volume of the necessary compressor for same refrigerating capacity as well as the modification of the refrigerating capacity at same discharge volume of the compressor for different refrigerants.

Table 7.3: Changing of volume flow of the necessary compressor for same refrigerating capacity respectively changing of refrigerating capacity at same volume flow of compressor for different refrigerants

Refr.	V_H	V_H-Enhanc.	Q_o	Q_o-Reduc.	K_I
	[m^3/h]	zu R723	[kW_o]	zu R723	[€/kW_o]
	Q_o =	11,3kW_o	V_H =	19,5m^3/h	Typ 19-D
R723	19,5	-	11,3	-	270
R717	20,8	+6,7%	10,6	-6,7%	289
R404A	21,3	+9,2%	10,4	-8,0%	253
R407C	23,8	+22,1%	9,3	-17,7%	283
R134a	37,2	+91,1%	5,9	-48,0%	446[1)] 389[2)]

1) Typ 19-D für 5,9 kW_0 2) Typ 46-D für 11,3 kW_o

For the better comparability of different refrigerating units with different refrigerants the specific capital expenditure, particular for the individual components, was formed and used in the profitability study. This is in particular advantageously since for example compressors are available only in power stages, and a comparability is falsified by the service and cost jumps.

For the profitability study in table 7.4 the capacity variations of the compressor of the Frigopol company model 19-D (design for conventional refrigerants and for R723 and/or R723) were considered at the formation of the specific costs. Also the capacity and connected design variations for the heat-exchangers like evaporator and condenser were included. The capital expenditure (capital bound costs) were converted in yearly costs with an annuity of 0.237, that means with an interest rate of 6 % p.a. and a term of 5 a. For the consumption bound costs an electric energy price was applied by 10 cent/kWh_{el}. As COPs those ones were used by real comparison compressors (item 5.1).

The capital expenditure of a R723 plant are similar with those ones of R404A- and R407C plants. The difference amounts only to about +10%. The individual component costs were considered in the capital expenditure for the comparability of plants because a R723 plant naturally has still no manufacturing numbers as the arrangement plants have. It is accepted that higher manufacturing numbers affect all plants now.

The profitability study is carried out in two variants. Variant one prescribes a fixed annual cold power demand that is covered with a refrigerating unit with a compressor model 19-D. Due to the different capacities of the unit at different refrigerants also different full use hours (a minimum of 7,210 h/a at R723 and 8,760 h/a at R407C) for the individual refrigerating units result to the cover of the cold demand. The plants with reduced capacity have an advantage through that. Onto the efficiency arrangement according to variant one was refused for R134a due to the considerably smaller refrigerating capacity opposite the other refrigerants with same piston displacement of the compressor.

Variant two considers a refrigerating unit with the same capacity (11 kW_o) to cover the cold demand for every refrigerant. Thus the full use hours are identical for all plants. However the capacity stages in the compressor offer are not considered at this way of consideration.

The profitability study points out that the operation bound costs, service, maintenance, refrigerants etc. are almost alike the arrangement plants. Cost differences with normal plants are marginal in relation to the capital and consumption bound costs.

Table 7.4: Comparison economic efficiency for different normal cooling aggregates

	investigated	refrigerants	conventional	refrigerants	
Component	**R723**	**R717**	**R404A**	**R407C**	**R134a**
			K_I [€ / kW_o]		
compressor	270	289	253	283	389
(Frigopol Typ 19-D)	(11,3kW_o)	(10,6kW_o)	(10,4W_o)	(9,3kW_o)	Typ 46 – 11,3kW_o
evaporator	146	162	134	109	154
condenser	102	116	85	70	95
	(75 €/kW_c)	(85 €/kW_c)	(60 €/kW_c)	(50 €/kW_c)	(67 €/kW_c)
TEV	27	28	10	11	17
further	120	150	100	100	100
capital expenditure-total	665	745	582	572	755
compare to R723	-	+80 (+12,0%)	-83 (-12,5%)	-93 (-14,0%)	+90 (+14,0%)
COP [-] computing	2,67	2,60	2,37	2,51	2,54
VARIANT I					
capital K_I [€/a] (at Q_o and a=0,237)	1.781	1.872	1.435	1.261	-
ΔK_I [€/a] to R723	-	+91	-346	-520	-
Q_o [kWh_o/a]	81.468	81.468	81.468	81.468	-
τ_V [h/a]	7.210	7.686	7.833	8.760	-
Q_{el} [kWh_{el}/a]	30.512	31.334	34.375	32.457	-
ΔQ_{el} [kwh_{el}/a] to R723	-	+822	+3.863	+1.945	-
ΔK_V [€/a] to R723 consumption (at 10 c/kWh_{el})	-	+82	+386	+195	-
compare ΔK_I and ΔK_V [€/a] to R723	**-**	**+173**	**+40**	**-325**	**-**
VARIANT II					
capital K_I [€/a] (at 11kW_o and a=0,237)	1.734	1.942	1.517	1.491	1.968
ΔK_I [€/a] to R723	-	+208	-217	-243	+234
Q_o [kWh_o/a]	88.000	88.000	88.000	88.000	88.000
τ_V [h/a]	8.000	8.000	8.000	8.000	8.000
Q_{el} [kWh_{el}/a]	32.959	33.846	37.131	35.060	34.646
ΔQ_{el} [kwh_{el}/a] to R723	-	+887	+4.172	+2.101	+1.687
ΔK_V [€/a] to R723 consumption (at 10 c/kWh_{el})	-	+89	+417	+210	+169
compare ΔK_I and ΔK_V [€/a] to R723	-	**+297**	**+200**	**-33**	**+403**

According to variant one the R407C plant is that one with the most low cost. The total costs of the R723 plant amounts to 4,832 € and thus past 6.7 % higher than those ones of the R407C plant. According to variant two the R723 plant (total costs of 5,030 €) is equivalent to R407C plant. The total cost difference is 0.7 %. Resulting from the cost comparison it is to conclude, that the R407C-plant and that one R723- are economically equivalent plants. The most unfavourable plant is the R134a- plant. The insignificantly higher capital expenditure are compensated through the better COP of R723 plant.

With regard to the specific values for filling amount, place demand and weight for the R723 plant are reached equivalent values as with the conventional plants. A energy consumption reduction of about 7 % is achieved. According to costs a R723 plant for normal cooling is equivalent with conventional plants and more favourable in the total costs as well as equivalent opposite R407C systems. However the efficiency can be achieved first if with sufficient number of manufacturing the component costs sink as with the conventional comparing plants.

The technical use of the brine cooler results can occur first after solving of the remaining problems. No profitability is to be achieved without refrigerant constraint for the use of natural refrigerants due to the two-staging of the R723 plant necessary for deep freezing.

Certain boundaries for the usability of the results are set, that were not to be foreseen at project beginning. There are yet no direct constraint for the use of natural refrigerants through the defusing of the refrigerant problematic caused by the EU regulation coming into force. And only the insignificantly low total costs of the R723 plants for normal cold generation for cooling cells will not cause any general market entrance of these plants yet. The capital expenditure is a little higher, the electric energy costs currently still are small for potential operators. Reservations exist opposite ammonia as well as R723 with regard to toxicity and explosion danger etc..

Development requirement consist further especially at the use of plate evaporators with regard to an optimal surface use of the heat exchanger and the development of hermetical compressors for R717 and R723. Both points are coupled at the development interest of the component manufacturers. In this respect in future further activities are undertaken.

Theoretic comparison of a heat pump demonstration plant with a traditional plant and calculation of energy savings

Circuit cycle calculations of the most frequently used refrigerants for heat pumps have been carried out. In the calculation it is assumed that the isentropic efficiency and the volumetric efficiency is 0.7.

Table 7.5: Other refrigerants compared with R723.

Operating conditions	-10/35		-10/42		-10/50		-6(1)/42		-6(1)/35	
Refriger-ant	**COP**	**Differ-ence %**	**COP**	**Differ-ence %**	**COP**	**Differ-ence**	**COP**	**Differ-ence**	**COP**	**Differ-ence**
R134	3,33	-2,92	2,74	-4,86	2,22	-7,50	3,06	-5,26	3,77	-3,58
R407C	3,25	-5,25	2,66	-7,64	2,13	-11,25	2,98	-7,74	3,69	-5,63
R717	3,40	-0,87	2,86	-0,69	2,39	-0,42	3,23	0,00	3,89	-0,51
R723	3,43	0,00	2,88	0,00	2,40	0,00	3,23	0,00	3,91	0,00
R290	3,30	-3,79	2,71	-5,90	2,20	-8,33	3,03	-6,19	3,73	-4,60

The theoretic cycle calculation shows that R723 gives an improved COP compared with the other refrigerants. If the comparison takes place with R407C the improvement is from 5.25 % to 11.25 %. A comparison of the volumetric displacement shows that R723 is the refrigerant with the lowest volumetric displacement.

Refrigerant	Volumetric dis-placement	Refrigeration capac-ity	Difference in volu-metric displacement compared with R723
	[m3/h]	**[kW]**	**%**
R723	35,25	18	0
R717	36,84	18	+4,51
R404A	42,31	18	+20,03
R407C	45,72	18	+29,70
R134A	67,35	18	+91,06
R290	46,88	18	+32,99

As R407C is the most frequently used refrigerant for heat pumps and chiller plants a price-related comparison has been made in relation to R723. The capacity in the comparison is 18 kW refrigeration capacity. In both cases, semi-hermetic compressors have been used.

	R723	R407C	R723	R407C
Component	**€**	**€**	**€/kW**	**€/kW**
Compressor	6456	3795	359	210
Condenser	377	377	21	21
Expansion valve	54	54	3	3
Liquid separator	94	0	5	0
Evaporator	377	377	21	21
Others	1200	1200	67	67
Total price	**8558**	**5803**	**476**	**322**

The comparison shows that it is 47% more expensive to build a refrigeration system withR723. The increase in price arises as the compressor is 70% more expensive. The other components cost the same.

From information obtained from the Danish company "landbrugets rådgivningscenter" it appears that the running period of a milk refrigeration system is app. 7 hours and 20 min. per day. The load is 93% of the maximum load. For a milk refrigeration system with a 18kW refrigeration system that means that there is an annual need for refrigeration of 44807 kWh. In the calculation, the electricity price is estimated at 10 cent/kWh.

		R407C	**R723**	**R723 with flooded evapo-rator**
Annual need for refrigeration	[kWh]	44808	44808	44808
COP	[-]	2,66	2,88	3,23
Power consumption	[kWh]	16845	15558	13872
Power consumption costs	[€]	1685	1556	1387

The annual operational saving amounts to 129 € when a comparison is made directly between a system with R407C and one with R723. This saving cannot compensate for the increased cost price of 2755 €.

If R723 is to be an alternative refrigerant it is necessary that the price of the compressors is brought into alignment with R407C compressors.

Marine chiller

A comparison has been carried out for R723 with R407C. In the comparison, parts have only been included for the refrigeration unit, variable speed control and control system and water circuit have not been included. Hermetic compressors have been used in the comparison.

	R723	**R407C**	**R723**	**R407C**
Component	**€**	**€**	**€/kW**	**€/kW**
Compressor	1548	2396	155	240
Condensor	1552	1553	155	155
Evaporator	942	471	94	47
Gas seperator	336	0	34	0
Thermovalve	575	87	58	9
Sight glass	67	20	7	2
Dry filter	15	15	1	1
Receiver	211	108	21	11
Manometers	135	135	13	13
Pressostates	117	117	12	12
Control AKV	667	0	67	0
Total price	**6166**	**4900**	**617**	**490**

The comparison shows that it is 26% more expensive to build a refrigeration system with R723. The increase in price is especially caused by the evaporator, gas seperator, expansion valve and control which make the system more expensive. The compressors that are used in the comparison are hermetic compressors. It should be mentioned that they have to be specially design and the price for that is unknown.

Scale ice machine

R723 has been compared with R407C and in the comparison parts have only been included for the refrigeration unit, variable speed control, control system and water circuit and the scale ice machine has not been included. Hermetic compressors have been used in the comparison.

	R723	**R407C**	**R723**	**R407C**
Component	**€**	**€**	**€/kW**	**€/kW**
Compressor	798	1032	266	344
Air cooled con-denser	673	673	224	224
Evaporator	447	447	149	149
Pressostat	117	117	39	39
Thermovalve	87	87	29	29
Manometers	135	135	45	45
Receiver	67	67	22	22
Total price	2324	2558	774	852

The comparison shows that it is 9,15% cheaper to build a refrigeration system with R723 rather than R407C. However, it should be mentioned that a suction receiver has to be used on the plant which will give a small price increase.

8 Summary

In connection with compression refrigeration plants the chemical refrigerant R134a and refrigerant mixtures such as R404A, R407A, R407C, R410A or R507 are typically used. These refrigerants do not have any ozone depletion potential but they have a global warming potential. So far, the utilisation of natural refrigerants has been limited. CO_2 and H_2O refrigeration plants are being developed and butane and propane applications are limited to very small systems and ammonia (NH_3) to larger applications. The objective of the project was the manufacturing of 6 demonstration plants, which use ammonia or the mixture of ammonia-dimethylether as natural refrigerants, which show a high COP and are economically competitive.

A reduction of the greenhouse effect can be achieved only through innovative plants that are in operation. For that the plants with natural refrigerants must be competitive compared with the conventional refrigeration plants. The energy consumption of these new plants may be not higher compared with the conventional refrigeration plants.

In the present final report the results of the development task are represented. The demo-plants was build-up, investigated and measured and regarding performance and energy consumption evaluated.

Within the project processing the EU-line for the refrigerant problematic concretised itself. The regulation proposal is represented within the final report. Because with the EU proposal the exit from the use of refrigerants with greenhouse effect was shifted, The use of R717 or R723 results less by the commitment of use of natural refrigerants than through the reduction of the energy consumption and achievement of a economic efficiency.

8.1 Summary German Part

In regard to the specific amount for filling-amount, place and mass equivalent amounts are achieved for R723- plants as for the conventional plants. Concerning of energy consumption reductions of about 7 % are achieved. The isentropic efficiency of his (0,6...0,65) and the delivery rate (0,7...0,85) of the R723- compressor lie very good in comparison with conventional compressors. The COP of the compressor with the refrigerant R723 is a a little bit higher in comparison with conventional compressors and other refrigerants, at the normal cooling 2,7 to 2,5. The refrigerating aggregate of the cooling cell was built-up with copper materials. A R723- plant for normal cooling is equivalent with conventional plants concerning the costs.

Refrigerant and oil analysis showed that no corrosion can be located in the system during the experimental time. The water content in the refrigerant was with 344 ppm under the boundary value of 400 ppm. In the oil the humidity laid on 168 ppm.

An one-stage cold generation with R723 for a deep freezing cell and 42℃ condensing temperature, cooling cell temperature -18 ℃, is not possible du e to too high discharge temperatures.

At the brine chiller further development demand is necessary. The combination refrigerant R723 and refrigerant oil KC68 is not suitable for the use of plate evaporators with under injection at the brine chiller. In this case an oil separator is necessary. With regard to the refrigerant enrichment in the refrigerant oil an oilheating and an oil difference pressure switcher are necessary for the guarantee of the compressor lubrication also at R723. Without refrigerant distribution in the plate evaporator liquid abstinence in front of the compressor is reached only from superheat temperatures of > 12℃, with refrigerant distribution this amount s ank onto approx. 7 K.

8.2 Summary Danish Part

The Danish approach was to develop and create a number of demo units as basis for further product development of competitive small refrigeration systems with ammonia. The Danish partners deliberately decided to let the demo units include as much “demonstration effect” as possible. Demo units include as much as possible of “new” technological possibilities. Danish demo units try to “extend” the technologies as much as possible. The approach have had character as “trial and error” in order to create the best and most interesting basis for further product development. The partial tasks concerned the type and circuit of the refrigeration plants, operating conditions, capacity and appropriate main components, connection techniques and the choice of materials.

Heat pump (Chiller) System for milk cooling (SVK)

The main objective goal for SVK was to build a Heat Pump System for milk cooling, which could fulfil the following conditions. The product has to be competitive with HFC-refrigeration systems, which means:

1. The operational costs shall be lower than traditional systems.
2. The efficiency shall be higher than HFC-refrigeration systems
3. The investment costs shall be at the same level as for HFC-refrigeration systems.

The partial goals were the follows:

1. To develop a refrigeration system with a natural and environmental friendly refrigerant
2. The manufacturing should not be more complicated than a HFC-refrigeration system
3. The components used should not be more expensive than traditional components.
4. The efficiency should be 10-20 % higher than HFC-refrigeration systems.

Conclusions

- The teoretical cycle calculations at the dimensioning point, shows that R723 increases the coefficient of performance COP with 5,9 % compared to R290, and with 4,9 % compared to R134 and 7,6 % compared to R407C. This means that costs to the energy consumption is decreased with the same level.
- A prototype gas separator combined with a capillary tube was developed, the price for this system is competitive to traditional expansion systems. The system increases the efficiency of the evaporator and compared to a thermal expansionvalve was the measured COP increased with 4,3 %.
- R723 will be a good alternative in heatpumps and refrigeration plants where semihermetic compressor are used. R723 is not good alternative when hermetic compressors are used, as the internal superheating of the refrigerant internal in the compressor leads to high temperatures of the refrigerant at the end of the compression.
- The manufacturing of the prototype showed that it´s possible to manufacture the heat pump as a traditional system with copper tubes. After the test were there no signs of corrosion in the copper pipes. The brass fittings used didn't either show signs of corrosion, but the brass shall not be used with R723 or ammonia because it would lead to corrosion on long terms condition.

Chiller System for marine applications (Buus)

The main objective goal for BUUS was to develop Chiller System for marine applications with a natural refrigerant, which could be competitive with HFC-refrigeration systems. As for the Ice Flake Machine the following items was specially considered important compared to alternatives in sales situations:

- Less energy consumption, same level of initial investment costs and less costs for service and maintenance leading to better Life Cycle Costs LCC.
- Better environmental ("green") profile.

Conclusions

- Experience from design and production of the demonstration plant shows that it is realistic and possible to develop a chiller system a competitive level of Life Cycle Costs compared to HFC-alternatives.
- Today it is allowed to use HFC-refrigerants and there are no signs of future regulations in that area. Further development of a Chiller System for marine applications are not very interesting as long as this is the situation on the market.

Ice Flake Machine for commercial market (Buus)

The main objective goal for BUUS was to develop an Ice Flake Machine for commercial market with a natural refrigerant, which could be competitive with HFC-refrigeration systems. Especially following items was considered important compared to alternatives in sales situations:

- Less energy consumption, same level of initial investment costs and less costs for service and maintenance leading to better Life Cycle Costs LCC.
- Better environmental (“green”) profile.

Conclusions

- Experience from design and production of the demonstration plant shows that it will be difficult to reach better level of Life Cycle Costs with very small indirect system compared to HFC-alternatives.
- Further development of an indirect system with R723 for very small ice flake machines is not very realistic. Semihermetic compressors has to be used based on the experiences from the tests with the demonstration plant with unmodified hermetic compressors. This will lead to price wise difficult situations compared to HFC-alternatives.

Commercial Refrigeration Systems with R717 or R723 in general

The project has been a success in many ways. A number of demonstration systems for R723 has been designed, manufactured, installed and tested and experience and a lot of know-how about the use of R723 to these systems have been gained. Some of the originally planned activities turned out to be impossible or very difficult to carry out. As example it was not possible to produce demonstration systems, which could be put into real field tests at potential customers and some technical aspects or problems still needs to be solved or developed further. The resources didn’t cover all the technical wishes and hopes. Based on the project work one can conclude:

Conclusions

- The European and national rules and legislations for reduction of potential greenhouse gases in the future are still not made and implemented. As long as there is uncertainty about the final results in this matter, companies are still allowed to use HFC-refrigerants and do not have to use resources on new HFC-free alternatives. At the moment refrigeration business is waiting.
- The use of R723 for small commercial systems is totally dependent of price wise competitive component – especially compressors. That was basically the reason to try hermetic compressors. Standard compressors for R507/R404A were used, when it turned out, that it was impossible to get modified test compressors compatible for R723 to the demonstration plants. The tests

showed that compressors much quicker than expected were damaged. New compatible products for R723 with external cooling have to be developed in order to reduce the discharge temperature to acceptable levels. Producers are not going to develop new products unless there are a potential market and business, which do not seem to be the case. Because of this the use of R723 will probably be limited to special medium sized applications, where already available semihermetic compressors from on the market can be used.

- Specifications for reliable and energy efficient refrigeration systems with R723 from technical point of view can be made based on all experiences from the tests with the demonstration systems, but further development has to be done. Specific and important areas for future development are: System designs for wet operation and oil return systems. Good and well proven solutions to could form a much better basis for competitive product in the future.

Symbol

COP	performance	K	costs	
Q	capacity		t	temperature
V	volume flow		Δt	temperature difference
η	efficiency		π	pressure ratio
ε	perofrmance	λ	delivery rate	
τ	time			

Indices

a	outlet	air	air
Ber	computing	c	condensing
Comp	compressor	DS	compressor outlet
DV	pressure valve	e	inlet
ext	external	ges	total
i	Injektion, Investition	is	isentrope
KT	brine	m	middle
o	evaporation, without	out	outlet
oh	overheating, superheating	s	saturation
SL	suction line	total	total
ÜH	superheating	V	loss, 100% capacity, consumption
VE,V2	discharge	2	piston 2