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TURBO WATER CHILLER WITH WATER AS REFRIGERANT

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ABSTRACT

The use of natural refrigerants in energy efficient refrigerating plants becomes more and more important for the protection of the environment against air pollution e.g. by HFCs and CO_2. The ILK Dresden is dealing with the development of turbo water chillers with water as refrigerant. The design of such chillers has to be tailor-made because of the thermodynamic properties of water. The paper describes the present situation in this ILK working field. On basis of first experiences with running of these chillers at car manufacturers like DaimlerChrysler or VW the ILK Dresden will continue its R&D activities in this field.

1. INTRODUCTION

The worldwide restrictions for the use of CFCs and HFCs caused by the proved environmental pollution due to ozone depletion and greenhouse effect call for the search for environment-friendly refrigerants. The natural refrigerants like water (R718), air, carbon dioxide (R744), ammonia (R717) and propane (R290) are some of them. The research and development activities for the use of water as refrigerant for compression refrigerating systems were started in the Dresden Institut für Luft- und Kältetechnik (ILK Dresden) in 1991. The research resulted in the two-stage turbo chillers with intercooling. Since 2000 the first of them have been proved in practice.

2. PROPERTIES OF WATER AS REFRIGERANT

The advantages of water as refrigerant are above all its non-limited availability, its low price compared to other refrigerants, its simple handling and its non-toxicity. Disadvantages are especially the low specific cooling capacity which results in high volume flows, pressure levels below ambient pressure and high pressure ratios. The comparison of the most important properties of water and ammonia is shown in Table 1.

Table 1: Comparison of refrigerants water (R718) and ammonia (R717)

Temperature	0°C		10°C		20°C	
	Water (R718)	Ammonia (R717)	Water (R718)	Ammonia (R717)	Water (R718)	Ammonia R(717)
Vapour pressure [bar]	0.006	4.3	0.012	6.2	0.023	8.6
Vapour density [g/m^3]	4.8	3,463	9.4	4,876	17.3	6,710
Specific refrigerating capacity [kJ/m^3]	12	4364	23	5966	42	7946
Volume flow at 700 kW refrigerating capacity [m^3/h]	208,000	664	108,000	466	59,000	352
Pressure ratio for 25 K temperature difference	5.2	2.3	4.6	2.2	4.1	2.1

The required temperature lift between cold and hot side results from the cold water temperature, required for the cold use, and the re-cooling conditions. Figure 1 shows the comparison between the pressure ratios as function of temperature lift for the refrigerants water, propane, ammonia and R134a. 30 K correlate with a pressure ratio of $\pi = 7$. At the same conditions the pressure ratio for ammonia amounts to 2.8 only.

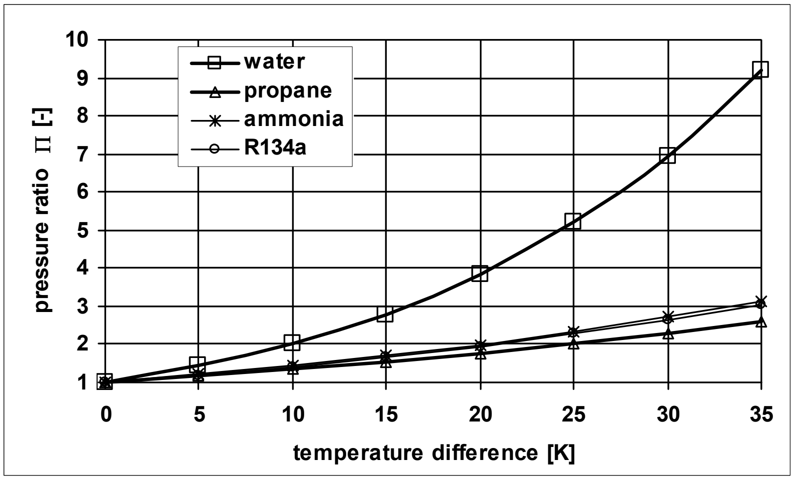

Figure 1: Pressure ratio as function of temperature lift

The required refrigerant volume flow for a given cooling capacity results from the specific cooling capacity of refrigerant. It is with 0.059 $m^3/(s\ kW)$ very low for water because of its low density (0.007 kg/m^3 @ $t_0 = 0°C$) compared to other refrigerants in spite of the high evaporation heat of water (2,490 kJ/kg @ $t_0 = 5°C$). An about 300 times higher volume flow is required compared to an ammonia system and about 200 times higher compared to an R134a system. Figure 2 shows a comparison of volume flows in % for the generation of the same cooling capacity for several refrigerants.

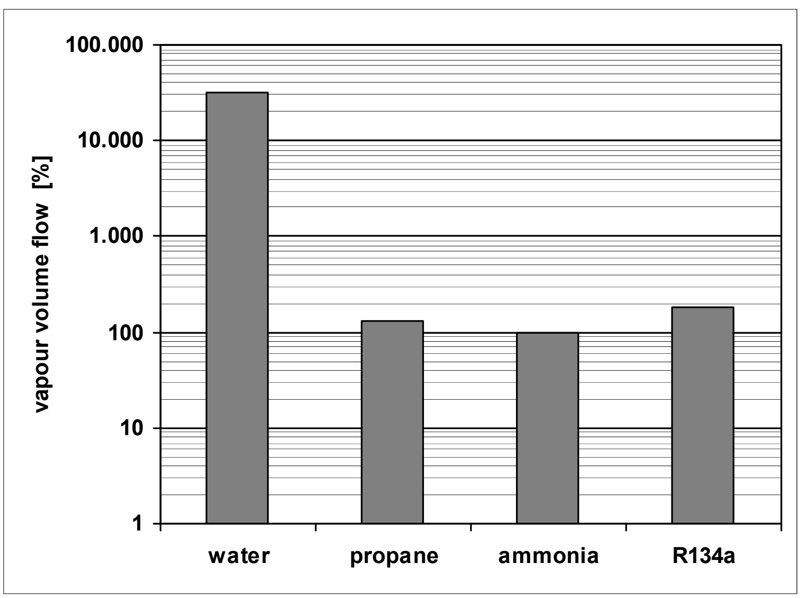

Figure 2: Comparison of volume flows at the same cooling capacity

Table 2 shows selected parameters to clarify the requirements for an R718 compression refrigerating system. The values correlate to a cooling capacity of 1 MW at an evaporation temperature of 5°C and a condensing temperature of 35°C.

Table 2: Refrigerant comparison (evaporation/condensing temperature 5°C / 35°C)

Refrigerant	ρ“[kg/m3]	p_{ev} [bar]	p_c [bar]	$\pi = p_{ev}/p_c$ [-]	r [kJ/kg]	$\dot{V}$ [m3/s]
R718	0.0068	0.0087	0.0562	6.46	2,490	59
R717 (NH3)	4	5.165	13.51	2.62	1,244	0.2
R134a	17	3.496	8.87	2.54	194	0.3

3. DESIGN AND FUNCTION OF R718 TURBO CHILLERS

Design and function of the chiller are influenced by the thermodynamic and physical properties of the refrigerant water. The high pressure ratio requires two-stage systems.

The cold water is expanded to the evaporation pressure at the evaporator inlet distributor and partially evaporated in this direct heat-exchanger. The evaporation heat which is drawn from the cold water leads to a cool down of the water. The water vapour is then compressed to a mean pressure in the first stage compressor. Superheated vapour is cooled down to nearly the saturation temperature by a partial evaporation of water through the direct contact with water. The water vapour is compressed to condensing pressure by the second stage compressor. The condensation is also realized by the direct contact of water vapour and cooling water. Because the evaporation and the condensing pressure are in the range of vacuum it is normally recommended to separate the chiller from outer water networks (cooling and cold water) by plate-heat-exchangers. Figure 3 shows the design scheme of the two-stage refrigerating system with all its main components and water as refrigerant. The two radial turbo compressors are the core of the R718 turbo chiller.

On this basis ILK Dresden developed a type series of R718 turbo chillers for a cooling capacity range of 500 to 1,000 kW. The typical cold water outlet temperatures are between 4 and 10°C. COPs of 4.6 to 7.75 are reached depending on the load conditions and water temperatures.

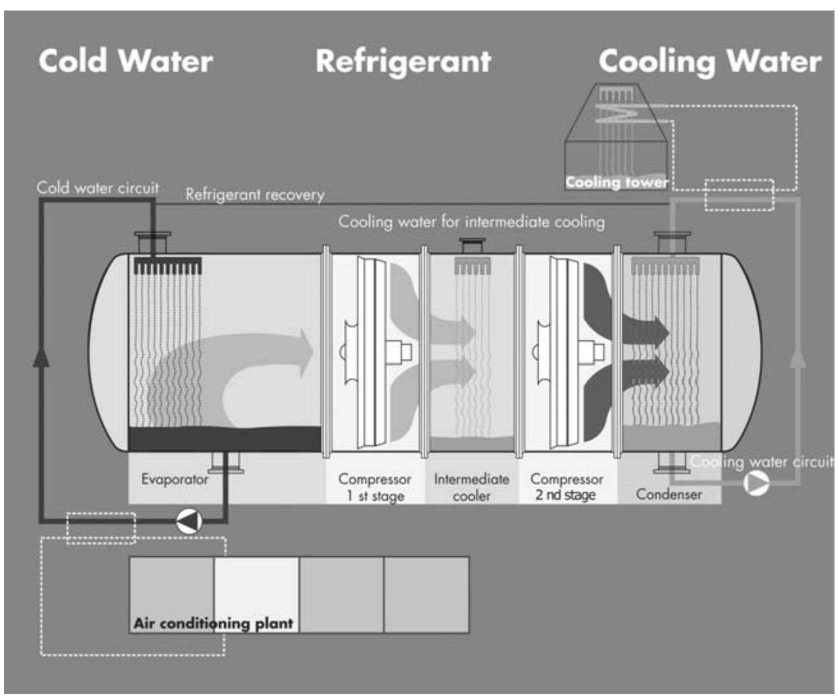

Figure 3: Design of an R718 turbo chiller

4. RADIAL TURBO COMPRESSOR

The radial turbo compressor for water as refrigerant is the main component of these chillers. The design of this compressor is influenced decisively by pressure ratio, volume flow and the medium water vapour. The obtainable pressure ratio is proportional to the impeller diameter and the square of speed. The volume flow also depends on the speed, the impeller diameter and the blade and channel width, respectively. The largest compressor of this type, developed by ILK Dresden, has an impeller diameter of 1.20 m, speed up to 9,200 rpm and generates a cooling capacity of 1,000 kW for the range of air-conditioning. The pressure ratio amounts to about 2.5 per stage and the volume flow to about 50 m^3/s.

The design of the compressor is shown in Figure 4. The intake nozzle with the entry vanes minimizes incidence losses at the impeller inlet. The impeller transfers kinetic energy to the fluid. Two tandem-arranged diffuser vanes and the following vane-less diffuser decelerate the flow, increasing the static pressure.

The medium water vapour demands and enables the application of very thin and very lightweight carbon fibre reinforced impeller blades. But these extremely lightweight blades produce themselves a centrifugal force per double-blade of 15 kN. On the other side the density of water vapour is this low at the low evaporation temperatures of 4 to 10°C that the fluid forces are negligible compared to the centrifugal forces. The blades are flat and therefore they are loaded only along their radial fibres.

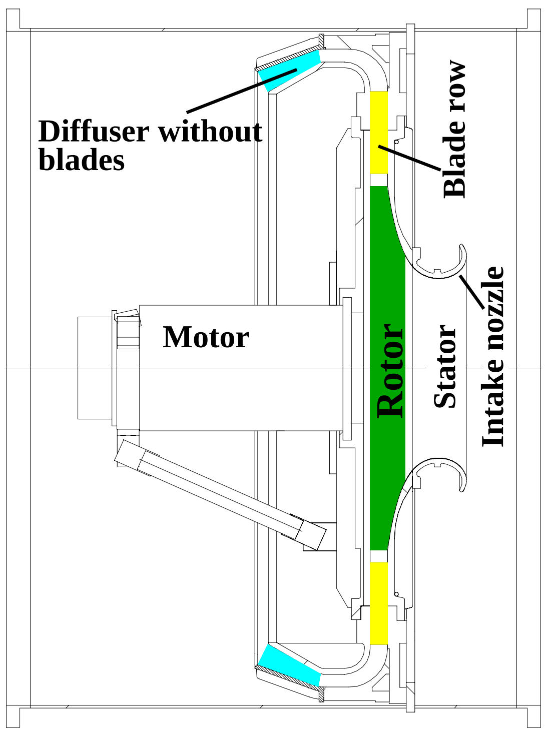

Figure 4: Compressor design scheme

At highest flow rates, the flow chokes at sonic velocities in the compressor inlet, while at low flow rates at low cooling loads surging occurs.

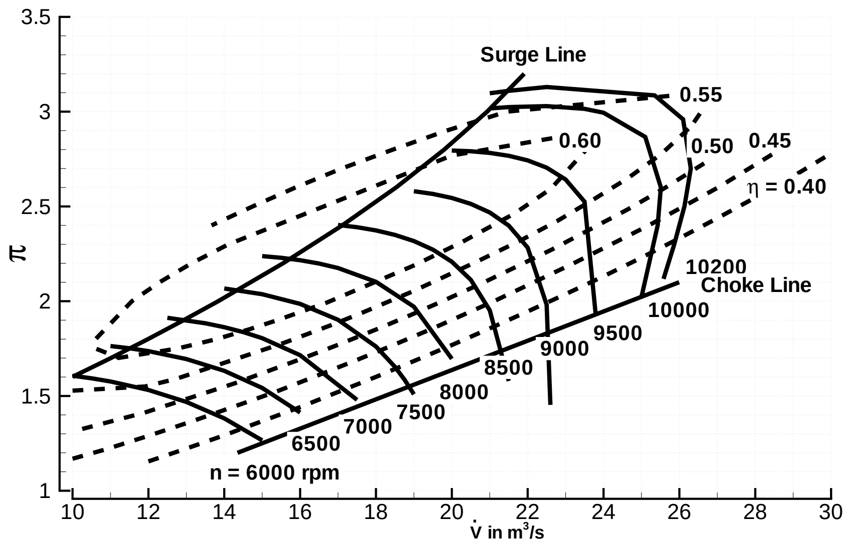

Figure 5: Compressor characteristics field with surge and choke line

The characteristics lines formed by pressure ratio over suction volume flow for constant speed of a chosen compressor are shown in Figure 5. Lines of constant efficiencies are also shown. The compressor characteristics field is limited by the surge line (on the left side) and the choke line (on the right side).

Currently ILK Dresden completes experimental and numerical tasks improving the efficiency essentially. For this goal a 2 : 1 scale compressor model which operates with air was installed at ILK. The transferability of the results is ensured because of the geometric and the dynamic similarities. The tests are carried out at identical Reynolds numbers. The Mach numbers can not be kept the same. The compressibility effects will be compensated through a correction of the measurement results. Mainly the effect of an inducer (rotating inlet guide vanes) shall be tested on this air using test rig. This inducer enables on the one hand a pre-compression in the axial part of the compressor and on the other hand a considerably improved flow turning.

Currently the numerical calculations with the CFD package “Fluent” focus on the flow through the second diffuser cascade. The goal is the determination of optimized geometric parameters like setting angle, blade length and shape. Furthermore ILK works on shaping the inner contour of the middle section. A solution is anticipated at least for moderate speeds. In this way all the compression components undergo an aerodynamic optimization.

5. HEAT-EXCHANGERS

The use of water as refrigerant enables the application of direct heat-exchangers for evaporator, intercooler and condenser. This means, there are no walls between cooling agent, heating agent and refrigerant. Hence no additional heat transfer resistance, caused by separating walls, is generated. Thus the quality of heat transfer improves and the efficiency of cold water generation increases.

5.1 Evaporator

A lot of tests resulted in a very simple and efficient evaporator design. The cold water is expanded into a free space through a pipe distributor and about 1% of the water flow is evaporated. The necessary latent heat is taken away from the remaining water flow. The separation of vapour and water are carried out mainly by gravitational separation and by a droplet separator in form of a filling material package. Minimization of the pressure loss at the vapour side was a further development criterion, since all additional pressure losses must be compensated by the turbo compressors.

The quality of a direct evaporator can be measured by the temperature difference between cold water outlet temperature and saturation temperature. The smaller this difference the more efficient is the evaporator. Currently the chosen evaporator design results in a temperature difference of only about 1 K.

5.2 Intercooler

The intercooler de-superheats the superheated vapour, which leaves the first stage compressor. Water is distributed evenly over the filling material package in the intercooler section which is shown in Figure 6. The vapour cross-flows through the filling material through which the cooling water flows downwards and is cooled down nearly to the saturation temperature. The cooling is realized only by a partial evaporation of water flow inside the intercooler. The evaporated part of cooling water is added continuously from the condenser cycle.

Figure 6: View of the opened intercooler

5.3 Condenser

The vapour which leaves the second stage compressor is de-superheated and completely condensed in the condenser. The cooling water is distributed evenly through a perforated metal sheet over the total cross section of the filling material and flows down over contact material in cross-flow with the vapour. The quality of the condenser is determined by the temperature difference between cooling water outlet and saturation temperature like for the evaporator. Currently minimum differences of about 0.5 K are obtained with this chosen solution. This value depends strongly on the inert gas concentration and therefore on the tightness of the overall system. All the entering inert gases are collected in the condenser and evacuated discontinuously from the place of the highest concentration depending on the above mentioned temperature difference. Figure 7 shows the measured values depending on the condenser capacity.

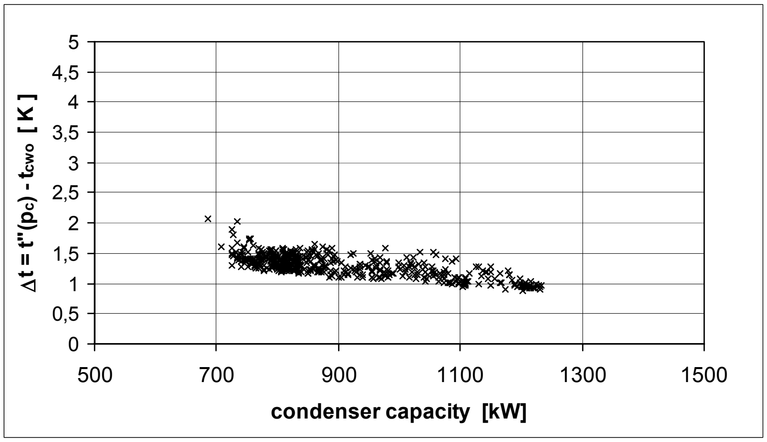

Figure 7: Temperature difference as a function of condenser capacity

5.4 Capacity control

The capacity control of the R718 turbo chillers is realized by a speed control. The mean frequency motors are driven with a frequency transformer and a following sine filter. Experimental and analytically established speed characteristics fields were implemented into the controller software ensuring that both compressors transport the same mass flows at an energetic optimum of the mean pressure. The control algorithm stipulates the relevant speed couples according to the current evaporation and condensing temperature. The control operates quasi-continuously due to the many available speed couples.

The speed control is limited, since the pressure ratio depends on the impeller speed too. So the minimum possible speed is determined by the pressure ratio at the partial load case. It is important to design applications without unnecessary reserves to realize as large as possible partial load ranges. This way it is possible to control the cooling capacity within a range from 60% to 100%.

6. CONCLUSIONS

- 1991 – start of the research and development activities for the use of water as refrigerant for compression refrigerating systems at ILK Dresden
- construction of a special test rig (small-scale) for the investigation of water as refrigerant at ILK
- development of the turbo compressor for water as refrigerant
- investigation of different components like heat-exchangers, compressor or motor
- transfer of the results to the first model of a large-scale industrial turbo water chiller
- improvement of the heat-exchangers by using of so-called direct heat-exchangers as evaporator, intercooler, and condenser
- improvement of the electrical motor for application under vacuum conditions
- since 2000 the first of chillers have been proved in practice, e.g. at the car manufacturers DaimlerChrysler and VW in Germany
- currently realization of experimental and numerical tasks to improve the efficiency essentially with a 2 : 1 scale compressor model which operates with air

NOMENCLATURE

CFD	computation of fluid dynamics	
COP	coefficient of performance	(-)
ρ''	vapour density at saturation	(kg/m3)
π	pressure ratio	(-)
r	latent heat	(kJ/kg)
$\dot{V}$	volume flow	(m3/s)
n	speed	(rpm)
p_{ev}	evaporation pressure	(bar)
p_c	condensing pressure	(bar)
t_0	evaporation temperature	(°C)
Δt	temperature difference	(K)
$t''(p_c)$	vapour saturation temperature at condenser pressure	(°C)
t_{cwo}	cooling water outlet temperature, condenser	(°C)

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