New multistage axial compressor and chiller development for water as refrigerant

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ABSTRACT

A new type of compact, low cost and highly efficient axial compressor for water vapor has been developed as a result of more than six years of research. The new compressor allows the design and development of competitive, large chillers based on water as refrigerant as an alternative to the traditional refrigerants (HFC's, ammonia, etc.). The design and testing of the compressor and the direct type heat-exchangers has been performed at Danish Technological Institute. Based on the fundamental results, a prototype commercial chiller has been designed, established and tested by Kobe Steel who has the production rights along with Johnson Controls Denmark.

1. INTRODUCTION

The world of refrigeration is dominated by the vapor compression cycle, but the choice of media or refrigerant is a pending issue. The CFC's were banned due to the ozone depleting potential, the HFC's may be phased out due to high global warming potential, and in parallel other natural alternatives such as ammonia, carbon dioxide, hydrocarbons etc. have been applied. From numerous perspectives, water is the ultimate choice of refrigerant. It is cheap and readily available; there is no contribution to global warming, no contribution to ozone depletion. It is non-flammable and non-toxic and there are no harmful atmospheric or fire breakdown products.

Water as refrigerant has been suggested in the past by various authors and technical groups, and a few full-scale plants have been established (1),(2),(3). Some of them have been based on the centrifugal compressors from IDE Technologies, Israel, with flexible blades as described for instance in (4). This technology has been further developed by ILK as described in (3) and a new generation of commercial chillers has been announced to be marketed in 2011 (5). The use of condensing wave rotors has been described (6) as a method that could potentially improve the performance of the vapor cycle. Vacuum ice production and snow making has been the focus of IDE where units have been established in South Africa, Switzerland and in Japan, among others. An axial compressor type has also been suggested for vacuum ice applications (7) and (8). Small-sized units have been suggested for instance in (9), and theoretical surveys of water as refrigerant with various conclusions have also been conducted (10), (11).

But the research and development work described in the present paper is the first time an axial compressor has been developed from scratch, specially designed for the application as chiller for industrial refrigeration and air condition, and with the potential for a commercial breakthrough for water as refrigerant.

2. FEASIBILITY STUDY

The obvious environmental advantages from using water as refrigerant were the background of a feasibility study that was completed in 1996 – 98 by the former Sabroe in collaboration with Danish Technological Institute and ConceptsNREC. The objective was to investigate whether it was technically possible to develop a chiller for the large capacity ranges from approximately 500 kW and upwards that could be competitive compared with traditional HFC and NH_3 plants on main parameters such as:

- **Size** (specifically the area or footprint to be used and the ability to ship in containers)
- **Efficiency** (it should at least be possible to achieve the same COP as with existing technologies)
- **Price** (sales price should match existing facilities, perhaps by including refrigerant cost, energy saving, safety equipment, etc.)

The so-called LEGO plant, (2), a prototype unit based on water as refrigerant, was established prior to the feasibility study, and it was in operation during the period 1995 – 2005. The LEGO plant demonstrated some of the system solutions while giving valuable operational experience. But the plant also showed that the applied generation of compressors from IDE, Israel, resulted in the plant becoming too bulky, too expensive and had too poor compressor efficiency to be commercially competitive. As a consequence, the feasibility study tried to identify and analyze alternative compressor solutions and the possibilities to reduce the physical dimensions and the total cost of the plant.

The physical properties of water cause the technology for the plants using water as refrigerant to be somewhat different from conventional plants. There are some very specific requirements and working conditions both for the plants and for the compressor. The vapor pressure of water is relatively low and for all temperatures below 100°C it is lower than atmospheric pressure, i.e. the process runs in vacuum conditions for the temperature ranges typical for chiller operation.

Figure 1 shows that the vapor pressure at a typical chiller temperature of 6-7°C is approximately 10 mbar absolute pressure while the pressure at typical condensation temperatures 25 -35°C ranges from approx. 30 to 60 mbar absolute pressure. Because of the very low pressure the specific volume for water vapor is very large as shown in Figure 1.

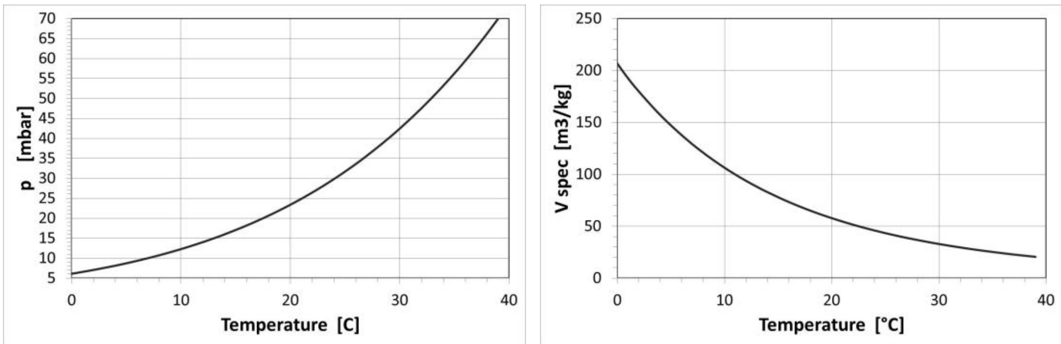

Figure 1 Left: Saturation pressure for water as a function of temperature, Right: Specific volume of water vapor.

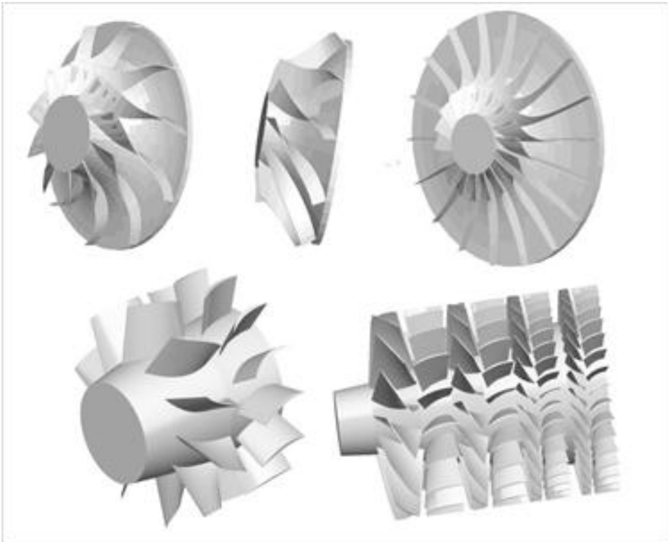

Figure 2 Various types of turbo compressors were evaluated in collaboration with ConceptsNREC, USA. The choice fell on the multistage axial compressor type (bottom right) as the most promising candidate - all things considered.

At typical chiller temperatures (6-7°C), the specific volume is in the order of 130 to 140 m^3/kg and as the latent heat for water is about 2.4 MJ/kg it means that an approx. 2 MW plant - which the feasibility study was based on - must have a compressor volume flow of approx. 100 m^3/s. Such conditions restrict the range of potential types of compressors to the relatively large turbo compressors, and a variety of candidates as shown in Figure 2 were examined on a number of parameters including production methods, material selection, flexibility etc., in addition to the mentioned three main parameters.

The conclusion of the feasibility study was positive; there was a chance to develop a system that could be competitive on all three parameters – size, cost and COP – by selecting the concept:

- The development of the axial compressor type as shown in figure 2, bottom right, in a special cost effective way.
- In combination with an optimized version of the direct contact heat exchangers.

In this way, the systems based on water as media (sketched in Figure 3) would be able to compete with traditional plants based on NH_3 or the synthetic HCF refrigerants on entirely commercial basis. The sketch shows the approximate operational conditions for the unit, the developed compressor offers sufficient margin to be able to operate on a global scale of applications.

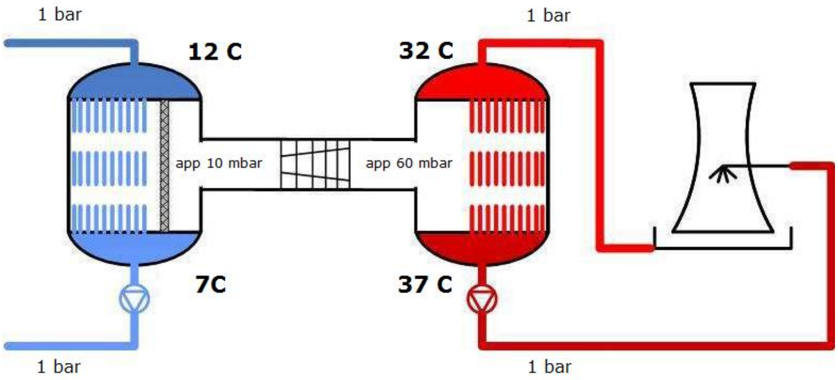

Figure 3 Sketch of refrigeration system principle with direct contact heat exchangers and axial compressor. Operational conditions are indicated.

However, in 2001 York Refrigeration (former Sabroe) decided to stop the project, despite the very promising prospects. At the time, the basic conditions had been tested either in scaled tests or in full-scale tests and a partially completed test unit was manufactured.

At the instigation of Danish Technological Institute, the project was restarted three years later with the Japanese company Kobe Steel Ltd. (Kobelco) as industrial partner in collaboration with the following four Japanese companies: The Tokyo Electric Power Company, Incorporated, Chubu Electric Power Company, Incorporated, The Kansai Electric Power Company, Incorporated and Central Research Institute of Electric Power Industry. Subsequently, the former York Refrigeration participated as a 'sleeping partner' and as part of the agreement the test rig was moved to Danish Technological Institute where continued development and testing took place.

3. DESIGN AND DEVELOPMENT OF THE AXIAL COMPRESSORS

The new project group comprising the new organization then restarted the development work where it had been stopped and the Danish Energy Agency (DEA) maintained to support the project as they had earlier. The central component was the compressor, and most efforts were employed during the 6-7 years of development of design and testing the compressors at various stages of development in different versions and under various operating conditions.

The principle of the compressor is basically the same as for all other turbo compressors. It consists of a fast rotating component (rotor) which transfers velocity to the gas (water vapor or steam in this case) and a subsequent stationary part (stators) that slows down the gas and converts the velocity to pressure. The compressor principle is shown in Figure 4, and the total required pressure ratio for the compressor is achieved by adding more stages directly after each other on the same shaft and thus with the same rotational speed.

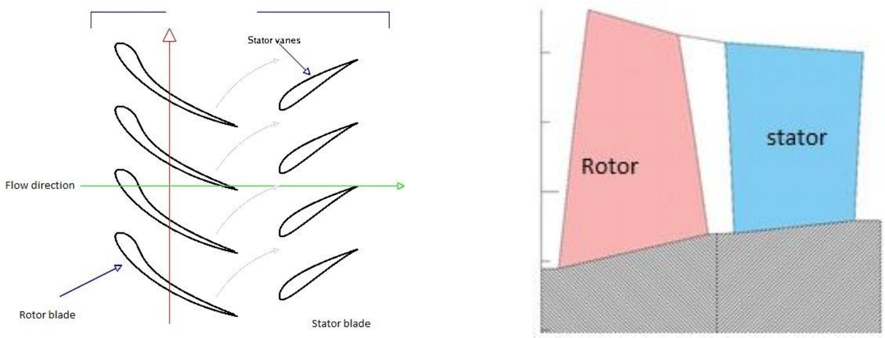

Figure 4 The principle of an axial compressor stage is that the rotors give the vapor high velocity and hence dynamic pressure after which the stationary stator blades again slow the vapor and convert velocity to pressure. [Wikipedia and AXIAL™]

The design of axial compressors is a highly specialized task with very high requirements to the computational tools and calculation accuracy as well as to the technical areas of expertise. At the same time, independent technical areas such as compressor aerodynamic optimization, selection of material and production method, optimization of stress level and natural frequencies, rotor dynamics and bearing specifications must be reconciled.

DTI has used the full software package from ConceptsNREC consisting of the 1-D meanline code called AXIAL™, the 3-D geometry generation etc. code AxCent™ with the integrated CFD solver called PushButton™. The AXIAL 1-D code offers the two loss models by Koch & Smith and Wright & Miller, respectively, (12), (13), (14) but also a very flexible system that allows the calibration of almost every parameter and loss coefficients. This feature was necessary in our case as the low pressure and thus low Reynolds numbers in combination with the relative Mach numbers in question required special corrections and a special calibration of the code based on the measured performance.

The design phase is further complicated by the fact that an axial compressor has many variables that are of importance to both the aerodynamic and the mechanical design such as number of stages, rotational speed, diameter, blade count, blade type of profile, blade height/length ratio, thrust area at front and exit, blade angle, blade camber, leading edge and trailing edge thickness, tip clearance, etc. Identification and selection of these variables affect the compressor total pressure ratio, efficiency, relative Mach number, various types of losses, etc.

The detailed 3-D design of each individual blade row followed the 1-D optimization and in particular the comprehensive CFD runs take up most of the design efforts and are rather time consuming. Figure 5 shows an example of the result of a CFD run that illustrates the flow field in one of the downstream stages, still in the transonic regime. The CFD solver could be used for multistage analysis as well, including the frozen-rotor option and various stage-stage mixing schemes.

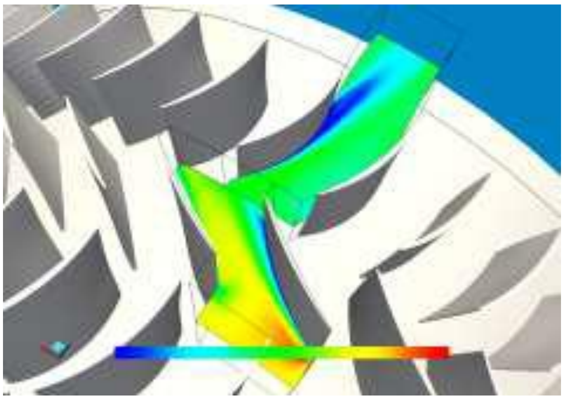

Figure 5 An example of the detailed CDF calculations that were performed for each blade row in the geometry optimization process.

The mechanical design consisted of the optimization of the peak stress level as shown in Figure 6 and an optimization of the positions of the vibration modes connected to the natural frequencies for the blade rows.

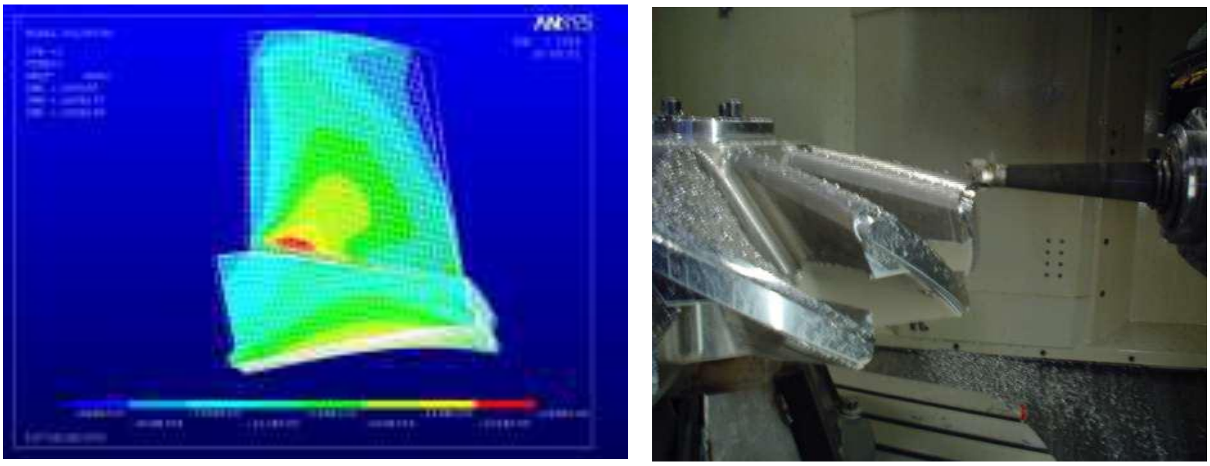

Figure 6 Left: An example of the finite element analysis of a blade in order to minimize the peak stress level. Right: the machining of one of the rotors.

The actual design procedure was typically structured as a number of iterations where the design was gradually improved and optimized by adjustments of the geometry of the previous designs.

Once the final design of a stage was established as the best compromise between partially conflicting requirements it was manufactured and the stage could subsequently be tested on the test rig.

The project developed two compressor sizes, the largest having the capacity of approximately 100 m^3/s corresponding to a chiller capacity of approximately 1.8 MW at 7°C. The smallest compressor has a volume flow of approximately 40 m^3/s, corresponding to approximately 800 kW cooling capacity.

4. TEST OF COMPRESSOR AND CHILLER

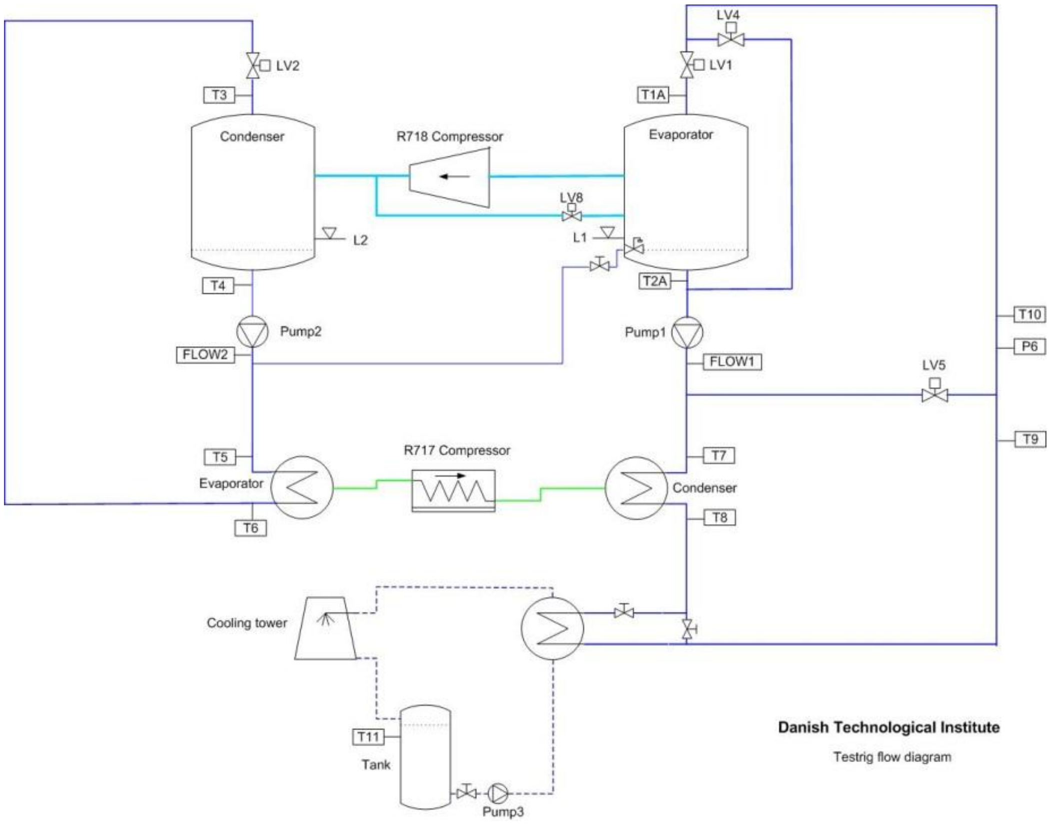

Figure 7 Outline of rig for the water vapor compressor with the NH_3 chiller acting as load.

During the test of the newly developed compressors, it was necessary to have a rig with a cooling load of approximately 2 MW available and to be able to adjust the capacity, the evaporator and the condenser temperatures as needed. Therefore, a matching NH_3 chiller was installed so the NH_3 chiller evaporator was connected to the H_2O system's condenser and vice versa. All the necessary operating conditions for test of the H_2O system could be established relatively quickly. In addition, a number of by-pass valves were arranged to help adjust the operating conditions. The entire test rig is outlined in Figure 7. The cooling towers removed the excess heat from the motors.

Figure 8 The location of proximity sensors in order to monitor bearing performance.

The compressor performance was measured by a large number of pressure and temperature sensors inside the compressor and they were compared with the detailed three-dimensional CFD flow calculations and the 1-D mean-line predictions. Such measurements were made for each complete assembly of the compressor with the actual number of stages. The measurements were also used to calibrate the software tools for better design and better prediction of the performance of the next stages.

The compressor was equipped with proximity sensors at the shaft to investigate the rotor dynamics of the system and to record data for the water-lubricated bearings, as well as vibration on the compressor casing. The measurements show that the overall vibration level is low, in the very best class for the type and size of compressors. The performance of the evaporator has also been measured and in general the value of the leaving temperature difference is less than one degree at chiller conditions. Special caution has been shown to the design of the effective drop separation system that minimizes the carry-over from the flash evaporator.

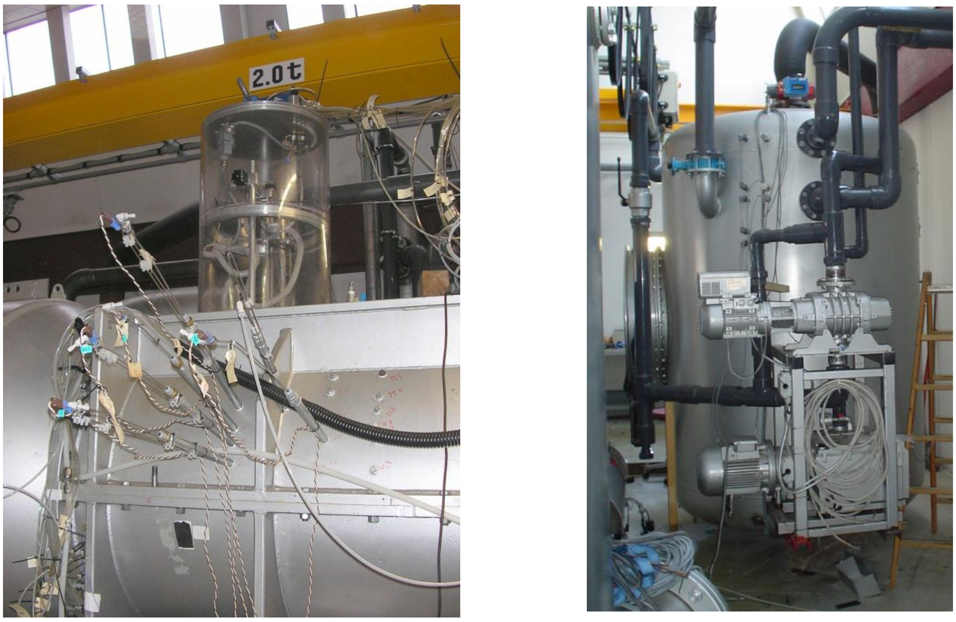

Figure 9 To the left it appears how the pressure and temperature measurements of the compressor performance were made primarily by sensors in the compressor house and partly by a robot on top. To the right it appears how vacuum is maintained by means of a Roots pump and a rotary vane pump.

Similarly, the losses in the condenser have been measured, and in general the leaving temperature difference at design conditions is less than one degree, including the contribution from the non-condensable gasses. The cooling water to the condenser is circulated through the cooling towers and will therefore inherently contain some air to be continuously removed at the vacuum conditions. That takes place in a specially designed two-stage deaerator system integrated into the condenser.

The system is very efficient and imposes a loss of only 1-2% of the total energy consumption where competing systems typically have losses in the range 5-10%. The vacuum system is shown in Figure 9 and consists of a Roots pump to the low pressure level with a rotary vane vacuum pump on top at the higher pressure level to take the non-condensable gases to atmospheric pressure. Due to various tests the piping shown in the figure on the test rig is somewhat more complicated than it will be in the upcoming commercial version.

5. EXPECTED PERFORMANCE AND CHILLER DATA

Capacity: As mentioned, two compressor sizes were provisionally developed which in turn corresponds to two different capacities. The physical properties of water make the capacity vary as a function of the outlet temperature by almost 10% per degree, which is a somewhat stronger dependency than applicable to the traditional refrigerants. Figure 10 shows how the capacity varies as a function of the outlet chiller temperature.

Physical size: The units will be approximately as compact as the traditional NH_3 and HFC systems and thus significantly smaller than the previous LEGO system. The final dimensions are not determined yet, but for the approx. 1800 kW unit, the length, width and height will be approximately 4.5 x 2.5 x 2.3 m and the smaller approximately 800 kW unit, the dimensions will be approximately 3 x 1.7 x 2.3m.

One drawback of the chiller units is the need for some inflow height to water pumps (NPSH value), typically 1.2 – 1.5 m due to the vacuum conditions in the evaporator and condenser which calls for the possibility to install pumps in the basement, alternatively submerged or alternatively the plant has to be raised.

EER: In the U.S., chillers for air conditioning are specified according to both design point EER (COP) and more informative according to the partial load ratio, defined as the weighted average value of EER at four well-defined operating points. The European Eurovent standard has taken up the method and defined conditions based on the average European climatic conditions, as shown in the table below.

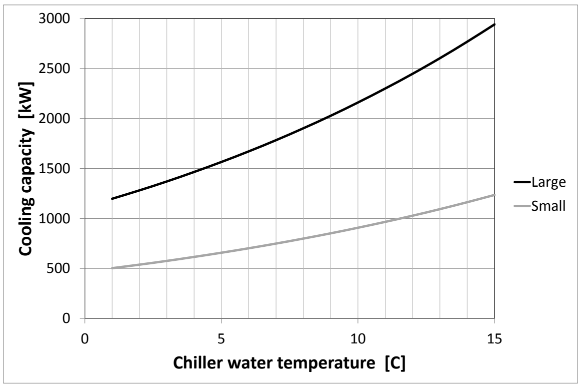

Figure 10 Cooling capacity of the chiller as a function of the outlet temperature of the cold water.

Table 1 EER values based on test rig data.

Load conditions	Eurovent - ESEER (Europa) (12/7C) cold water		ARI 550/590 - IPLV (USA) (54/44F – 12,2/6,7C)	
	Condenser cooling water inlet temperature (C)	Relative weight (%)	Condenser cooling water inlet temperature (C)	Relative weight (%)
100% load	30.0	3	29.4	1
75% load	26.0	33	23.9	42
50% load	22.0	41	18.3	45
25% load	18.0	23	18.3	12
	EER	6.00	EER	6.05
	ESEER part load	8.62	IPLV part load	9.50

The condenser water flow is defined in the European standard to be set to the value where the heating is 5.0°C in the condenser at full load; that is, heating from 30°C inlet condenser temperature to 35°C outlet temperature.

In the U.S. standard, the heating is set to be approx. 5.2°C, i.e. the outlet temperature of approx. 34.6 °C. In all cases, the calculations have been performed with the assumed electrical efficiency for the motor of 95% and all auxiliary equipment has been included.

For typical Danish conditions with the temperature conditions 27/22°C and 12/7°C, the EER is expected to be approximately 8.4 at full load.

Finally it can be mentioned that due to the high discharge gas temperature of more than 200°C the chiller may be fitted with a 100 - 200 kW hot-water system. If there is a need for clean water, the large capacity chiller can produce approx. 3 m^3/h of distilled water as a by-product during the normal process of cooling by applying the indirect condenser.

6. PERSPECTIVES

The production rights of the developed compressors and the developed direct contact heat exchangers and chiller systems are shared between Kobe Steel Ltd (Kobelco™) and Johnson Controls Denmark. Before the market introduction a commercialization phase will be needed as well as various long-term tests. Therefore, it can be expected to take 2-4 years before the plants are ready for introduction on the market, depending on size and version of the chiller.

A prototype of the Japanese commercial version - that has to meet some specific Japanese requirements - has been running various tests over the past year in Japan, including tests of specially developed indirect type of heat exchangers. An actual commercial version is under design by Kobe Steel for optimization of production and for long-term tests in Japan.

Work is underway to establish two demonstration units in Denmark. It will be a global version – that is, for global weather conditions - with direct contact heat exchangers and for an industrial cooling application as well as for air-condition application, also for product maturing, long-term tests and as a showcase. It is expected that the two units will be in operation within approximately two years.

It is one of the great advantages of the developed compressor and heat exchanger technology that it - with smaller modifications – can be designed and manufactured in many different varieties for different purposes. Possible future applications could for instance be:

- high temperature heat pumps
- more energy efficient systems for drying and concentration tasks with recompression of water vapor (VRC)
- ice production for cold storage for peak cooling demand
- ice production to take advantage of low prices during off-peak periods in the electricity supply.

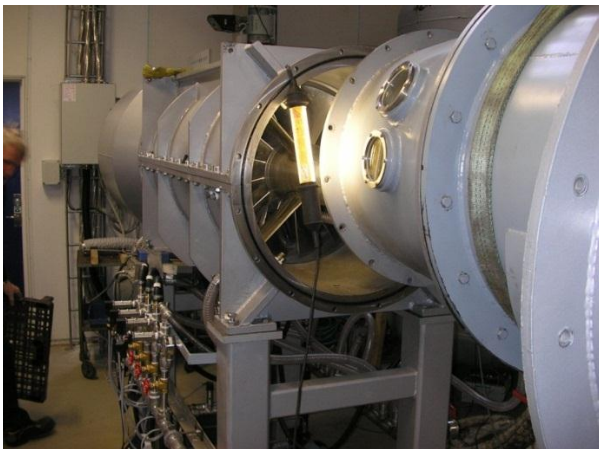

Figure 11 Left: Test rig with the larger 1.8 MW compressor. The commercial version is expected to have similar appearance, without all the instrumentation. Right: The smaller 800 kW compressor.

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