THEORETICAL DESIGN OF A HIGH-SPEED, OIL-FREE RADIAL COMPRESSOR FOR DOMESTIC HEAT PUMPS

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ABSTRACT

The development of domestic heat pump systems is mainly based on the use of volumetric oil lubricated compressors. Following the implementation of economizer based rotary compressors, which represented a major improvement step, one major opportunity to improve both efficiency and heat rate, in particular for high temperature lift heat pumps, is to use two stage cycles. However the reliability of those can be strongly impaired by oil migration, resulting in oil level unbalance when using oil lubricated compressors, unless more sophisticated auxiliaries including an oil pump are added. The need for this additional equipment, the requirement of higher vapour velocity for oil return and impediments to the efficiency of enhanced surfaces for heat exchangers are some of the incentives for the development of oil free compressor systems. As high-speed bearings and electrical motors are gradually becoming available this opens the way to consider the use of low-power, compact, oil-free and high-speed radial compressors.

This paper describes the basic design of a high-speed single stage radial compressor for high pressure ratios and a wide flow range aimed for the first stage of a two stage heat pump. A commercially available 3D viscous code has been applied to calculate the flow field through the impeller and the diffuser. The particular choice of the bearings allows using very small tip clearances resulting in low leakage losses. To complete the design a commercial 3D Finite Element code has been used for stress prediction as well as for the calculation of the blade critical frequencies and their corresponding modes. During the design process the impeller blade geometry has been continuously checked for machinability in order to ensure a feasible impeller using conventional 5 axis milling machines. This paper furthermore describes the setup of the test rig to be used for testing.

Keywords: ***turbocompressor, domestic heat pumps, high-speed, oil free***

1 INTRODUCTION

Standard oil-lubricated rotary volumetric compressors are predominantly used in domestic low-power heat pump applications. These compressors are generally characterized by a fixed built-in volume ratio, a rather average efficiency level and by the need of oil-lubrication. The built-in volume ratio limits the efficiency when facing variable pressure ratios whereas the need of lubrication results in the presence of oil in the refrigerant loop, introducing oil return constraints and decreasing the efficiency of several cycle components. A significant improvement has been the recent introduction of a scroll compressor equipped with a vapor injection port (economizer system), which is an intermediate step towards two-stage compression (Zehnder, 2002, 2004, article in ASHRAE journal). However further steps for advanced heat pump cycles will have to rely on genuine two-stage compression to improve the overall efficiency and the heat rate characteristics at variable pressure ratios. Earlier results from 2 stage units based on a mix of reciprocating and scroll compressor (Favrat 1997) or two scroll compressors (Zehnder (2004) show that, for such units to be reliable, complicated auxiliary systems would be needed to ensure an appropriate oil level in the two compressors. As high-speed gas lubricated bearings and high power density electric motors are progressively becoming more realistic the use of low-power high-speed directly driven turbocompressors represent an interesting alternative for the domestic heat pump market: such a compressor unit would be much smaller and lighter than conventional units, absolutely oil free, efficient and capable of power modulation. A feasibility study by Schiffmann et al. (2002) has shown a technical and an economic driving force for such a system: A two stage unit could increase the overall system efficiency of an air-water heat pump by as much as 20 %, could continuously meet the heat rate required (12 kW thermal) and allow high temperature lifts (T_{Air} = -12°C, T_{Water} = 60°C).

As such a system clearly has numerous advantages a proof of concept was launched to demonstrate a single stage turbocompressor for R134a with a pressure ratio Π up to 4 and a power up to 3 kW, running on gas lubricated bearings and directly driven by an electric motor. The designed compressor wheel fits the specifications of the 1st compression stage of a two-stage heat pump with a maximum heat rate of 12 kW. A two-stage compressor unit based on the same technology will follow the proof of concept unit.

This project is not the first attempt to couple high-speed turbomachinery to heat pumps. Theijse (1991) presented a work where it was tried to perform a compression and an expansion through dynamic turbomachinery also using gas bearings. The project apparently failed due to bearing problems.

2 SYSTEM LAYOUT

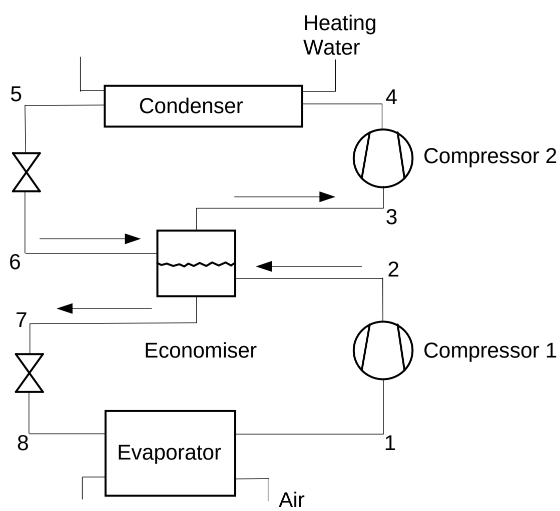
2.1 Heat Pump Cycles

The most important losses of high temperature lift heat pumps occur in the compression and the expansion processes; therefore it is of primary importance to develop advanced compressor design, which can take advantage of the cycles allowing a better recovery of the energy of the liquid refrigerant at the outlet of the condenser. In a separate project Zehnder et al. (1998) analyses several of the most promising cycles often together with experimental validation. These cycles include:

1. Single stage cycle with an additional separate cycle for upgrading some of the subcooling heat of the condensed liquid to further heat the water to be heated.
2. Two superposed separate single stage heat pump cycles. A condenser-evaporator couples the two cycles by transferring heat from the bottoming cycle to the topping cycle.
3. Cycle with a two stage expansion, an economizer phase separator and (or) an internal heat exchanger at intermediate pressure and a single stage compressor with intermediate vapor injection (Beeton and Pham, 2003).
4. The same as 3 but replacing the single stage compression by a two-stage compressor

The study has shown that the COP of solution 2 is the worst due to the internal heat transfer losses in the condenser-evaporator, whereas solutions 1 and 4 presented similar COPs. Solution 3 (which could be designated as the two-stage cycle of the poor) is an interesting short term solution even if it provides lower COPs than solutions 1 and 4. Solution 1 is not interesting for radial compressors due to the high pressure ratio needed for the main cycle, especially for high temperature lift applications where the overall pressure ratio may exceed feasibility. Solution 3 and 4 with an internal heat exchanger but no economizer-separator are of particular interest for non-azeotropic refrigerant; the internal heat exchanger instead of the economizer avoids a distillation of the refrigerant-blend. However injection of a potentially two phase refrigerant flow might be tricky to handle as high-speed radial compressors will not cope with liquid droplets. Solution 4 in combination with an economizer-separator is clearly the most simple in terms of components and control. Furthermore it allows to partly defrost the evaporator by inversing the cycle and using the economizer as an internal energy source. The two stage cycle with an intermediate economizer-separator is the solution maintained for this first project. Fig. 1 represents the schematic solution and the corresponding cycle in the T-s diagram.

Ideally the heat rate delivered by the heat pump will be modulated as a function of the external air temperature by acting on the rotational speed of the compressor. This will regulate the mass flow for a given pressure ratio, that is imposed by the outside air and by the temperature level of the heating water circuit. The intermediate pressure level will be fixed by the ratio between the mass flows of the upper and the lower cycle and can therefore be controlled through accurate impeller design. The bearings used for this high-speed application are dynamic gas bearings (the gas being the refrigerant vapor) and therefore pressure control is very important for a stable bearing operation. Even if, in air-water heat pumps, the lower and the higher pressures vary significantly during the season, the bearings will run at the intermediate pressure level, where fluctuations will be much smaller.



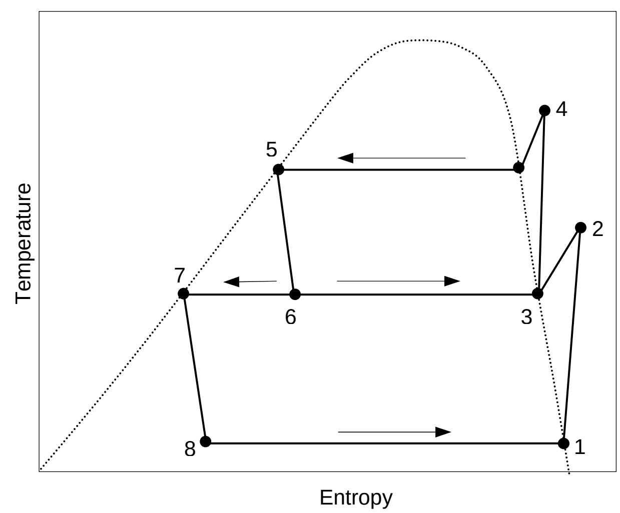


Fig. 1. Flow chart and T-s diagram of a two stage heat pump with phase separator.

2.2 Refrigerant Choice

The criteria for the choice of the refrigerant used are of ecological nature (greenhouse effect, ozone decomposition) and of thermodynamic nature. The physical properties of the refrigerant define the pressure ratios, the mass and the volumetric flows and have a considerable influence on the COP. The choice of the refrigerant also influences the size and the rotational speed of the radial compressors. The analyzed refrigerants are R134a, R407c, and some natural refrigerants like butane (R600), iso-butane (R600a), propane (R290) and ammonia (R717). The synthetic refrigerants R134a and R407C do not have any ozone decomposition potential but they have a large greenhouse effect if liberated to atmosphere. And the only alternative strategy is to swap to natural refrigerants that are either flammable or toxic. An interesting alternative would be CO_2, but it implies high pressures (100 bar), resulting in large forces on the bearings, and in very small compressor wheels.

The effects of the refrigerant choice has been calculated for the extreme point A-12W60 (Air inlet at -12°C, Water exhaust temperature at 60°C) for a nominal heat rate of 12 kW and summarized in Table 1 below. In order to estimate the rotational speed and the diameter of the two compressor wheels non-dimensional numbers have been used according to Balje (1981). The first two columns indicate the specific speed and the specific diameter of the 1st and the 2nd compressor stage according to Balje (1981). The specific speeds have been chosen in order to fit the maximum efficiencies for radial compressors. The following columns indicate the diameter, the rotational speed of the compressor, the tip speed, the speed of sound and the tangential Mach number and the volumetric flow to get a maximum COP. The last four columns represent the inlet and the outlet pressure, the pressure ratio and the COP.

The advantages of butane (R600) and iso-butane (R600a) are the high COP and the low exhaust pressures resulting in lower axial forces, and, of course the global environmental aspect. The inconvenients are the flammability, the high Mach numbers that might result in a reduction of the rotational speed and thus a reduction of efficiency. Another inconvenient is the sub-atmospheric inlet pressure with the risk of ambient air contaminating the refrigerant. Ammonia (R717) requires very high rotational speeds and very small impellers. This would not be a problem for the bearings but it certainly would for the electric motor if uncanned and for the mechanical integrity of the compressor wheels. Ammonia is extremely poisonous and therefore considered of little interest for domestic applications. The main inconvenient of R407C are the high exhaust pressure and the fact that being an azeotropic refrigerant it presents the possibility of distillation in the economizer-separator; in this case the bottoming and the topping loop might risk to operate at different refrigerant blends potentially deteriorating the compressor and cycle efficiency. However the possibility exists to use a two-stage cycle with an internal heat exchanger and only a separator instead of an economizer-separator. Such a solution is much easier in oil-free operation than in the presence of oil because boiling of the separated liquid is straight forward. The remaining refrigerants are propane (R290) and R134a. Propane has an enormous advantage in terms of low exhaust pressure but the ideal compressor wheels size is already quite small if it has to be machined on conventional milling machines. R134a allows acceptable compressor diameters, an acceptable pressure ratio and an average COP. Compared to propane (R290) the rotational speed is considerably lower, allowing relaxed specifications for the electric motor and its drive which will result in higher electric efficiency. Based on this analysis refrigerant R134a was selected as the best fluid for this proof of concept application.

Table 1. The effect of the choice of refrigerant on the overall performance of the two stage heat pump and on the size of the two impellers.

	N_S [-]	d_S [-]	d [mm]	n [krpm]	Tip Speed [m/s]	a (T_{IN}) [m/s]	M [-]	V [l/s]	P_{IN} [bar]	P_{OUT} [bar]	Π_{TOT} [-]	COP [-]
R134a	0.78	3.5	23	200	241	150	1.61	7.6	1.39	16.8	12.3	3.01
	0.62	4.5	20		209	160	1.3	2.8				
R407C	0.78	3.5	18	295	278	160	1.74	5	2.24	27.6	12.3	2.88
	0.76	3.5	13		201	170	1.18	1.7				
R600	0.78	3.5	33	195	337	198	1.7	21	0.47	6.4	13.6	3.12
	0.56	4.6	27		277	212	1.31	7.2				
R600a	0.78	3.5	28	220	322	197	1.63	14	0.75	8.68	11.6	3.07
	0.6	4.5	24		276	210	1.31	5.3				
R290	0.78	3.5	16	410	344	224	1.53	5	2.53	21.2	8.4	2.97
	0.71	3.5	13		279	235	1.19	2.3				
R717	0.78	3.5	11	1050	605	397	1.52	4.5	1.98	26.1	13.6	3.09
	0.52	4.5	9		495	428	1.16	1.7				

2.3 Simulation and Expected Performance

In order to better understand the behavior of a two stage heat pump with radial turbocompressors it was decided to generate a model of the cycle. Matlab was used for programming and the thermodynamic modeling was linked to a real gas refrigerant database (Refprop). The major input parameters of the thermodynamic model are the refrigerant, the nominal heat rate, the temperatures of the air inlet and of the heating water outlet, the pinch in the evaporator and in the condenser as well as the rotational speed of the compressor unit. It is assumed that the two compressor wheels are running on the same shaft and therefore rotating at the same speed. The model then calculates the refrigerant pressures in the heat exchangers and the intermediate pressure in the economizer according to the delivered mass flows of the two impellers. A simple impeller model (Schiffmann, 2002) allows the estimation of the efficiency and whether the compressor tends to operate in surge or in choke.

For a given operation point defined by an air inlet temperature and a water exhaust temperature the compressor will be able to run not only on one single rotational speed but over a whole range. The minimum speed is limited surge phenomena (an aerodynamic phenomenon of instability that might damage the compressor unit due to large oscillating forces). The maximum speed is limited by choke. Higher rotational speeds are possible but do not make any sense as the mass flow would not increase. The optimum COP will generally be located in between these two speed limits. This regulation of the rotational speed allows capacity modulation and therefore a better matching with the heating rate delivered by the heat pump at each operational point.

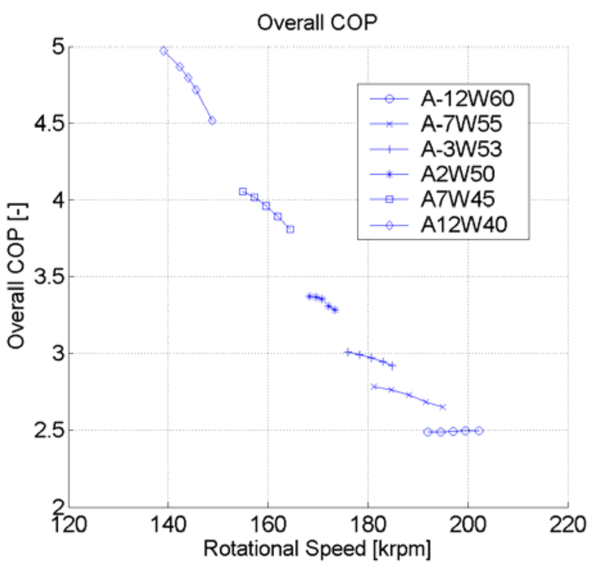


Fig. 2. External COP for six different operation points.

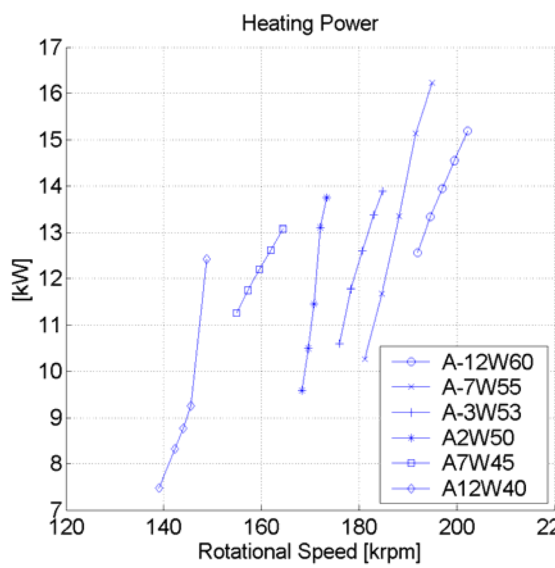


Fig. 3. Heating power delivered by the heat pump as function of the rotational speed and of the operation point.

Figs. 2 and 3 indicate the predicted external COP (defined as the ratio between the delivered heat rate and the total power consumption including the ventilator and the water pump) and the delivered heat rate as function of the rotational speed. It is interesting to observe the evolution of the heat rate delivered by the heat pump as a function of the external air temperature: it is maximum for low temperatures and minimum for higher air temperatures. This represents an inversion of the heating curve compared to conventional air-water heat pumps driven by constant speed rotary volumetric compressors. The radial compressor will therefore allow an operation with a much higher utilization coefficient, i.e. the heat pump will run in a much more continuous manner with less stop & starts than conventional systems. The expected COPs are interesting compared to commercially available heat pumps working on the same operation range. As shown by Schiffmann (2002) the annual COP of such a heat pump would be around 3.3 for a heat pump of nominal heat rate of 12 kW. This represents a significant increase compared to commercially available heat pump systems (single stage scroll compressor cycle with intermediate liquid injection).

3 SYSTEM DESIGN

3.1 Specifications

The final aim of the present project is the construction of a complete two stage compressor unit. A feasibility study (Schiffmann, 2002) has shown that the bearings running at high Mach numbers, the small impeller size and the power density of the electric motor represent high technological risks and are therefore important bottlenecks for the success of this system. It was therefore decided to build a single stage compressor serving as proof of concept before stepping directly to the two stage compressor.

This single stage unit should obviously be built so that a second impeller wheel can be added on the rotor in order to obtain a two stage unit. Therefore it was decided to design the compressor wheel using the specifications of the 1^{st} compressor stage; although the pressure ratios have to be of similar value between the two stages the net axial force is considerably lower on the 1^{st} stage.

As already discussed in section 2.1 the intermediate pressure will remain nearly constant over the operating range of the heat pump. Therefore the R134a bearings are designed to run at an average pressure of 0.58 MPa. The compressor will need a maximum rotational speed of around 220 krpm. Therefore the bearing system is required to support the rotor in the stable way up to a speed of 250 krpm. The electric motor and the power electronics need to be able to drive the rotor up to this speed with a corresponding power of 6 kW at the maximum speed. Only 2.5 kW are required for the single stage compressor but the 6kW will be needed for the two stage application.

The wheel specifications have been obtained with the thermodynamic model described above. Table 2 indicates the inlet conditions, the pressure ratio and the required mass flows for five operation points:

Table 2. The specifications for the 1^{st} compressor stage.

Operation Point		1	2	3	4	5
T_{AIR}	[°C]	-12	-7	2	7	12
P_{IN}	[Mpa]	0.14	0.17	0.24	0.29	0.35
ρ_{IN}	[Kg/m^3]	6.97	8.4	11.9	13.9	16.7
μ	[μPas]	10.2	10.4	10.7	10.9	11.1
Π	[-]	4.2	3.4	2.4	2	1.7

3.2 Compressor Design

In order to gain an overview of the optimal impeller geometry the non-dimensional method by Balje (1981) has been used at each operating point. The method allows the definition of a specific speed for optimum efficiency corresponding to each couple of parameters (pressure ratio and volume flow). Efficiency drops at too low rotational speeds due to the increased aerodynamic friction in the longer channels of the impeller, whereas for too high rotational speed it drops due to the higher losses linked to the high relative speeds of the gas in the impeller channels. This non-dimensional method applied to the five operation points results in significantly different geometries and rotational speeds for each point. Table 3 illustrates this divergence: the optimal impeller for the point A-12W60 has a diameter of 24 mm and a corresponding rotational speed of 186 krpm. The optimum wheel for the point A12W40 has a diameter of 7 mm and a rotational speed of 458 krpm. It is therefore evident that optimum efficiency cannot be achieved over the whole operating range using the same impeller.

Table 3. Optimum geometry and rotational speed for each operating point according to the non-dimensional speeds and diameters (Balje, 1981).

		A-12W60	**A-7W55**	**A2W50**	**A7W45**	**A12W40**
n_s	**[-]**	0.73	0.74	0.755	0.762	0.767
d_s	**[-]**	4	4	4	4	4
N	**[krpm]**	186	205	258	299	458
D_2	**[mm]**	24	18	13	10	7

The yearly overall efficiency of the heat pump corresponds to the integration of the COP (coefficient of performance) over the heating season. The heat pump compressor providing the highest yearly performance coefficient has to be primarily optimized (in terms of efficiency) for the most frequent operation point. The most frequent air temperature during the heating season in middle Europe is often of the order to 2°C (operation point 3 in Tables 1 and 2 above). For this point the impeller should therefore have a maximum efficiency at the pressure ratio of 2.4 but should be capable of pressure ratios up to 4.2. Furthermore the size of the impeller and the rotational speed should allow a safe operation in terms of rotor-dynamic stability; the size of the compressor should be at least such that it can be machined on a conventional 5 axis milling machine. This latter specification limits the minimum diameter and consequently also limits the maximum rotational speed. Based on scaling of known wheel geometries and investigations regarding complex surface machining it was decided to limit the minimum diameter to 20 mm.

A detailed description of the aerodynamic impeller design is presented elsewhere by Schiffmann and Favrat (2004). The impeller was developed using a 1D meanline analysis followed by a 2D inviscid code coupled with loss correlations for preliminary design, and the fine tuning of the blade geometry has been performed using the 3D Navier-Stokes solver (Concepts NREC software). The preliminary design includes the modeling of several physical effects like diffusion, efficiency, slip and losses using empirical parameters (Japikse, 1996). These parameters have previously been tuned on impellers of sizes much larger than the wheel considered here and as no measured data were available for wheels of such a small size, these calculated results have been treated with care.

The resulting impeller has a diameter of 20 mm, with 9 main and 9 splitter blades, a backsweep angle of 50°, a channel exit height of 1 mm and an operating running speed between 110 and 220 krpm. The 15 micrometer tip clearance chosen (between impeller and shroud) is consistent with the use of gas bearings. Fig. 4 represents the predicted compressor map for the main operation points, the predicted efficiency and the corresponding rotational speeds. The maximum mass flow for the highest pressure ratio is slightly below the specifications. This results from a compromise between maximizing the mass flow at high pressure ratio and allowing continuous operation at the minimum mass flow corresponding to the point A2W50. As a result, the designed impeller allows a continuous operation for air temperatures lower than 2°C. At outside air temperatures exceeding 2°C an intermittent mode of operation will be necessary. The maximum impeller efficiency is expected to be in the range of 80%.

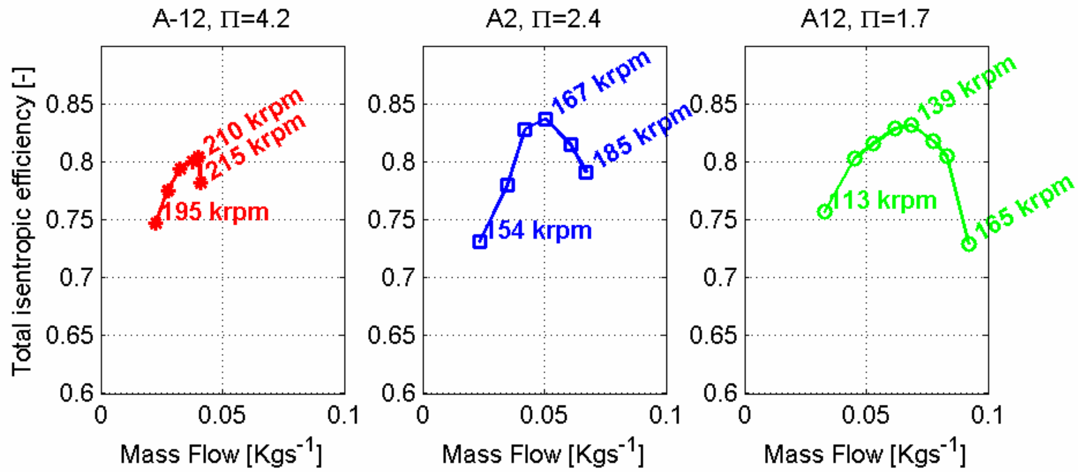


Fig. 4. Compressor map for the three main operation points A-12W60, A2W50 and A12W40 indicating the predicted efficiency and the rotational speeds.

A mechanical analysis performed with a finite element method allowed the prediction of the mechanical stress as well as the critical blade frequencies and their corresponding modes. The tangential speed being relatively low, mechanical stress is low enough to allow the use of an aluminum alloy, thus facilitating the machining compared to a titanium alloy. The maximum stress is found on the trailing edge of the splitter blade close to the hub surface. The stress level is around 100 Mpa and very concentrated, therefore causing no concern regarding the mechanical integrity of the wheel. The first critical bending mode of the blades is higher than 40 kHz, which in the absence of inlet guide vanes and diffuser vanes (vaneless type), cannot be excited in the operating range of the compressor. The deformation of the wheel due to the centrifugal forces is such that the blades are being bent towards the hub surface away from the shroud and therefore present no risk of touching at high rotational speeds.

The compressor wheel has been machined by a specialist workshop with experience in machining of complex surfaces. First the envelope of the wheel has been turned on a numeric lathe and channels then machined on a CNC 5 axis milling machine with a high-speed spindle. The final touch has been performed with a 0.5 mm diameter mill. Fig. 5 shows the compressor wheel in comparison with a 1 Euro coin:

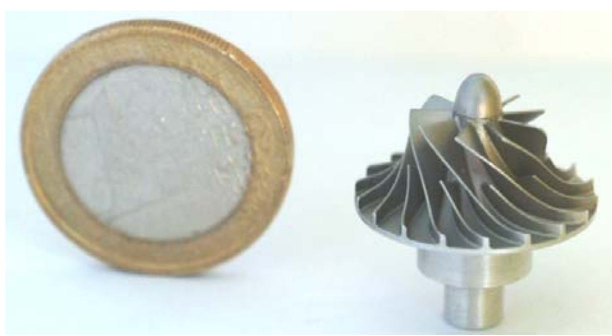


Fig. 5. The machined compressor wheel in comparison with a 1Euro coin.

3.3 Rotor and Bearing Design

The final rotor will be composed of one (later two) radial compressor wheels, an electric motor, two dynamic radial gas bearings and two dynamic axial gas bearings. Several rotor layouts are possible presenting different advantages and inconvenients. The criteria for choosing the most suitable layout version include a first critical bending speed sufficiently above the maximum operating speed, easiness for production and assembly as well as cost minimization. Fig. 6 shows one possible rotor layout. On the left hand side the two impellers are mounted overhang back to back. On the right hand side the motor is placed overhung as well. The middle section is occupied by the bearing system. This layout might seem astonishing, but it allows machining the complete rotor in one part only and therefore considerably reduces the manufacturing difficulties and allows a very easy assembly procedure:

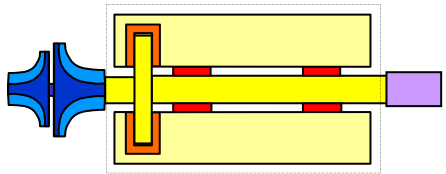


Fig. 6. Basic layout of the rotor the two compressor wheels back to back on the left and the overhung electric motor on the right. The middle section is occupied by the bearing system.

The bearings used in this application are dynamic gas bearings. This type of bearings presents the advantage of not requiring the need for oil lubrication and therefore allowing very high rotational speeds with low mechanical losses. By their nature they present very low damping coefficients and have highly cross coupled stiffness. Therefore their design is an integral part of the complete rotor design in terms of rotor dynamic stability. A model of gas bearing supported rotors has been generated and the bearing design together with the rotor optimized for maximum stability. Fig. 7 represents the critical speeds of the rigid modes and the corresponding stability (logarithmic decrement) as a function of the rotational speed of the optimal rotor design. The rotor presents 4 rigid critical speeds, 2 cylindrical and 2 conical modes, each with both forward and backward modes. The conical modes are between 20 and 50 krpm whereas the cylindrical ones are situated between 50 and 100 krpm. The 4 critical speeds are highly damped and are considerably below the operating range of the compressor. Experience has shown that these rigid critical modes can be crossed without encountering any trouble. The first critical bending speed is around 420 krpm offering a large margin relative to the maximum rotational speed.

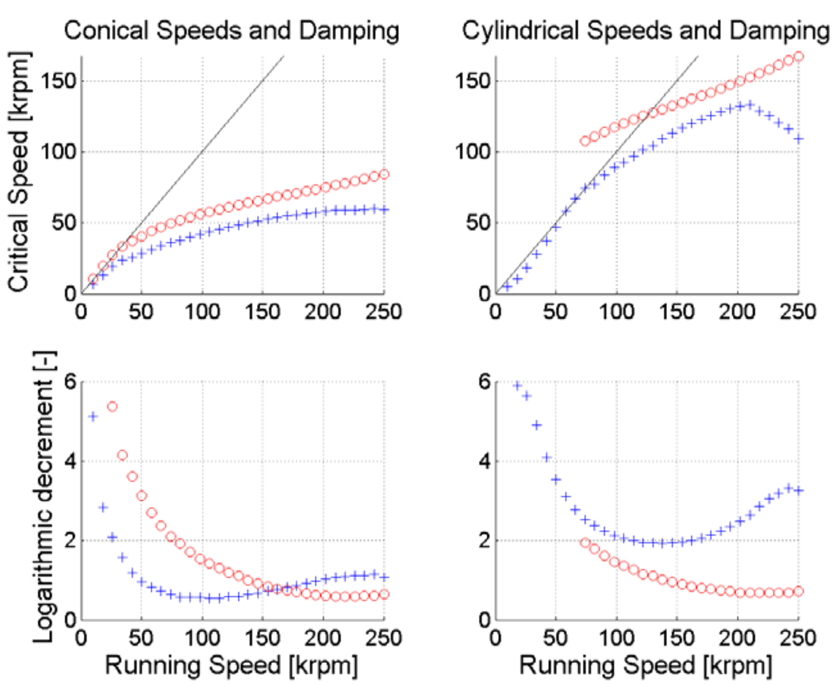


Fig. 7. Rotor dynamic characteristics of the designed rotor representing the evolution of the rigid modes and the corresponding stability.

3.4 Motor Design

The electric motor used in this application is unique in terms of the required rotational speed, the power, the efficiency and the size. The motor's specifications are the following:

- 6kW of mechanical power
- A maximum rotational speed of 250 krpm
- Gas and liquid refrigerant R134a is available for cooling
- Need to be as small and short as possible to comply with rotor dynamic considerations.

The synchronous motor chosen has two poles with permanent magnets and is driven by three phases. The overall diameter of the electric motor is 50 mm. An efficiency higher than 90% is predicted. The motor and the bearings are operated in a 5.8 bar R134a environment.

3.5 Unit Design

The small size of the compressor unit components, the impeller, the electric motor and the bearing system allows the design of a unit with an overall volume not much larger than a 0.3 l size can. The demonstration unit will be able to reach a pressure ratio of 4.2 allowing temperature lifts from -12°C to 30°C. The final two stage unit is expected to allow a total pressure ratio slightly higher than 16 for a temperature lift from -12°C to 60°C with a total compressor power of 6kW. The unit needs two additional circuits: a clean gas supply for the gas bearings, and a liquid loop for stator cooling.

4 TESTRIG DESIGN

The aim of the testrig is to allow the measurement of the turbocompressor performance over the complete operating range. As the first unit tested will correspond to the 1st compression stage planned to work at a pressure around 0.58 MPa, the tests will be run with a fixed exhaust pressure of 0.58 MPa and only the inlet pressure will be regulated in a range between 0.14 and 0.35 MPa. The rig is going to operate in a closed loop with refrigerant R134a. Two possible rigs could be envisaged: A real single stage heat pump cycle with a condenser and an evaporator where the compressor inlet and exhaust pressures are regulated through the temperature levels of the cooling and the heating circuit. The second possibility would be a vapor circuit, where the test cycle operates only on the right side of the saturation line, i.e. vapor is compressed, expanded, cooled and led to the compressor inlet again. Due to a shorter response time to temperature commands it was decided to design a vapor test rig as such a rig would allow shorter test sequences.

Fig. 8 represents a schematic view of the test rig as designed and built. The main refrigerant circuit is composed of the compression unit, a mass-flow meter, a controllable expansion valve, a cooling heat exchanger and a separator. The latter is to avoid that small debris or liquid droplets will flow into the compressor inlet and damage it. The heat exchanger dissipates the heat generated by the compressor through an external cooling system. Liquid and vapor refrigerant will be present in the heat exchanger; the liquid will stabilize the pressure of the vapor sucked by the compressor. The compression unit can be isolated by means of two valves from the main circuit for accessibility purpose.

The bearing gas circuit is composed of a pressure expansion valve for stabilizing the ambient bearing pressure, of a filter avoiding contamination of the bearings and of a microvalve allowing the regulation of the cooling mass flow. The gas is bled from the compressor exhaust through the bearings into the low pressure zone of the main refrigerant circuit. The high pressure gas can be tapped either before or after the mass flow meter. The stator of the electric motor is cooled through an external chiller for the initial tests. Of course, the final twin stage compression unit will have all the cooling circuits directly integrated into the heat pump cycle and will therefore not need any of these auxiliary systems.

The instrumentation of the test rig allows the acquisition of all relevant data for the determination of the compressor performance and for real time monitoring of the bearings, of the electric motor and of the operation of the complete testing loop. Two separate acquisition cards are being used: one for rapid sampling of the bearing position as well as compressor and bearing pressures, and a slower one for temperature acquisition. The two boards are piloted through a VI generated on LabView.

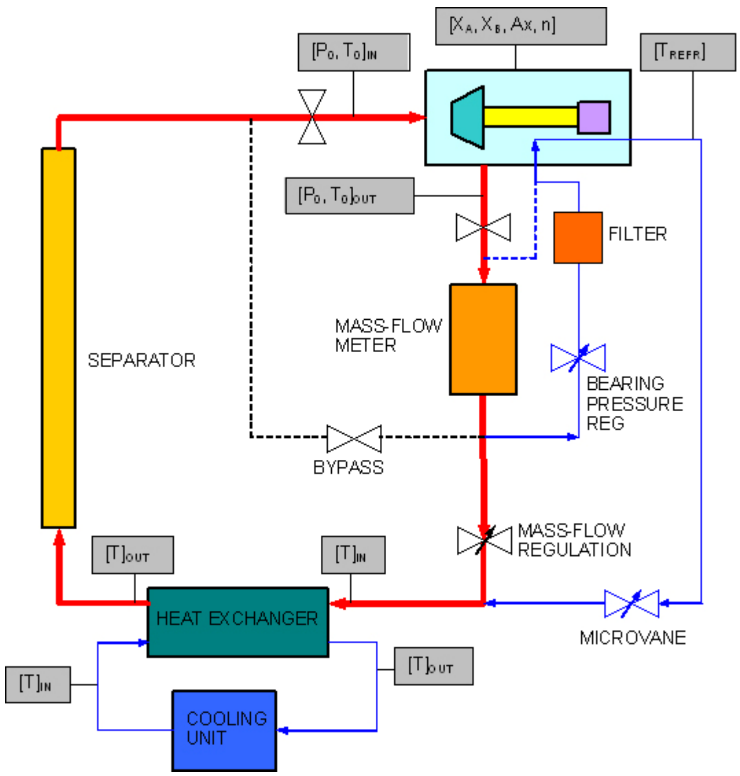


Fig. 8. Schematic view of the testrig illustrating the main cycle and the motor cooling cycle.

5 CONCLUSION

Dynamic oil-free refrigerant compressors are expected to boost the heat pump performance and relax piping constraints, in particular in advanced multi-stage applications. The procedure of the design process of a proof of concept high-speed low-power radial compressor has been described from the specifications to the final design. The unit is aimed at the application of domestic high temperature lift heat pumps. Some of the elements composing the unit are already available and testing should start in the near future.

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