1996

Development of Air Cycle Technology for Transport Refrigeration

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ABSTRACT

Developments for air cycle systems with respect to special requirements concerning the use of compression and expansion machines for transport refrigeration are discussed in this paper. Historic aspects are mentioned briefly and the present situation in Germany is sketched in the example of an air cycle air–conditioning system for the new generation of the German high speed train ICE 2.2. In order to develop air cycle systems for transport refrigeration, it is a challenge to select adequate machine concepts. This is one of the most important points to be considered because of the significant influence of the isentropic efficiencies with respect to the coefficient of performance. Especially the efficiency of the expansion machine has to be considered. Two fundamentally different concepts are described. The first concept consists of a motor–compressor–expander–unit using turbo machines. Characteristic data of the machines are used to simulate the full and part load behavior of this concept within an air cycle unit for use in transport refrigeration. The second concept consists of a pressure wave machine. Measurement results are shown and difficulties in obtaining serious efficiency data are discussed. In addition to this two finite difference schemes for fluid dynamics which have been developed and applied to evaluate fluid dynamic properties for the pressure wave machine are introduced.

1 INTRODUCTION

Concerning the use of HFCs as substitutes for CFCs and HCFCs a major concern in applications with remarkable leakages of refrigerant is its contribution to global warming. Therefore, the attention has been drawn early to the air cycle using a totally benign working fluid [1],[2].

In the early days of mechanical refrigeration air cycle technology has already been used for several applications like on board of ships for chilled meat transport. The use of air cycle systems continued for many years until ammonia and carbon dioxide for vapor compression cycles were applied. After CFCs had been created, they were the main refrigerants also for transport refrigeration and air conditioning, like R22 in ships and R12 in containers, railways and cars. With the development of gas turbine engines for aircrafts the air cycle was applied for their cabin cooling mainly because of having pressurised air already available to be used for the cycle. Today's systems in aircrafts are very complex in order to fulfill all requirements in any altitude and at any outside temperature. Open air cycle systems are now being used for about 50 years in aircrafts and there they have proven their reliability. Since their energy efficiency is rather low, for other applications mainly CFCs have been used in the past because of their better energy efficiency. Today's discussion about environmentally benign working fluids brought back into discussion the use of air cycle technology for other applications than for aircraft cooling when the leakage of refrigerants with high GWP would override the advantages of lower energy consumption concerning the Total Equivalent Warming Impact $TEWI$.

2 RESEARCH AND DEVELOPMENT PROJECTS IN HANNOVER

Since 1985 different projects related to air cycle systems have been conducted in Hannover. The main topics of these projects were:

- Basic analytical and experimental investigations about the performance of air cycles using different types of compression and expansion machines [3].

- A proprietary project for a leading German manufacturer in the railway sector related to railway and tram applications.
- A proprietary project for a leading German manufacturer in the area of high speed electrical drives related to high speed–motor–compressor–expander units.
- An overview about available compression and expansion machines, mainly for applications with small to medium capacities, including experimental results concerning capacity data and a flexible simulation program for air cycle systems, based on real characteristic data for machines and heat exchangers [4].
- The development of an air cycle system for transport refrigeration [5].

3 PRESENT SITUATION IN GERMANY

3.1 German High Speed Train ICE

Based on the evaluation of alternative air conditioning cycles two different companies were asked by the 'Deutsche Bundesbahn AG' to build air cycle systems for air conditioning in the German high speed train ICE. Both systems have been running successfully since spring 1995 and therefore it is planned to use these systems in the new generation of the train, the ICE 2.2 starting on the new high speed track between Hannover and Berlin in 1998. One manufacturer had chosen an open cycle unit which was derived from an air conditioning system for aircrafts. The other system is a new development using a closed air cycle. Fig. 1 shows the closed air cycle compact unit to be installed in the roof of the train.

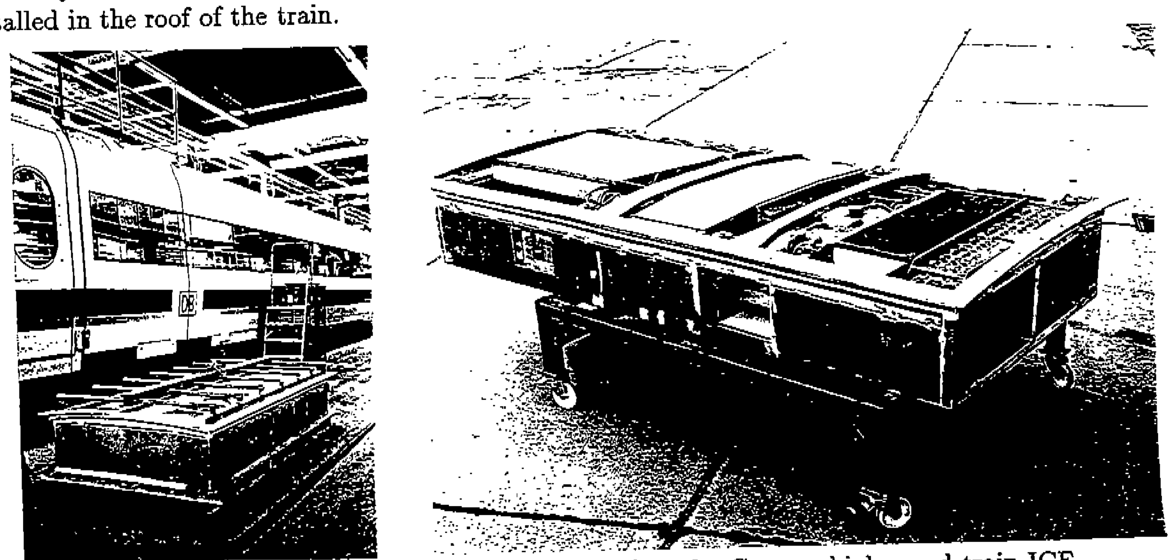

Figure 1: Air cycle compact unit as installed in the German high speed train ICE.

In comparison to vapor compression cycles this machine has got the same dimensions, nearly the same price and more or less the same weight. The relative energy consumption in comparison to the currently used R12 split system and to a modern R134a system is shown in Table 1 (R134a is set to have an energy consumption of 1.0) [6].

Table 1: A comparison of the relative energy consumption and the $TEWI$–number.

Train	System	Energy (factor)	TEWI kg CO_2 /Year [1]
ICE 1	R12 split system	1.2	111700
ICE 2	R134a system	1.0	31200
ICE 2.2	air cycle unit	1.2	26400

As can be seen, the developed air cycle system has the same energy consumption as the currently used R12 system. The new R134a system shows an improvement of roughly 20%. Comparing vapor compression sytems with air cycle systems other important aspects can be considered. An important advantage is that the maintenance work for an air cycle system is less cost and time intensive than for other systems. Taking into account that currently about 60% of the refrigerant annually charged into the atmosphere due to leakage and maintenance, the total equivalent warming impact $TEWI$ for the air cycle system is even better than for the modern R134a system (refer also to Table 1).

This example can prove that air cycle technology for other applications than for aircraft air–conditioning is not just a research project, but has already reached the application stage. This is of rather great interest as it is one possibility for future refrigeration systems, especially for applications where remarkable leakages are to be expected.

3.2 Transport Cooling

Like railway air conditioning, also transport refrigeration came out to be one of the interesting applications for air cycle systems. FKW is currently developing a prototype system for use in small distribution vehicles. The general concept is an open air cycle system with an internal heat exchanger. The system was already discussed in another paper [7], therefore this paper will focus on the machine part.

In comparison to railway air conditioning, where about 32 kW cooling capacity is required, only 5 kW and less capacity is needed for transport refrigeration. This means especially for the selection of machines that small sized machines, e.g. turbo chargers, have to be used, whereas equipment from the aircraft industry cannot be used.

Looking to all parameters influencing the COP of the cycle one finds that the efficiencies of the compression and especially the expansion machine are most significant. The equation to calculate the COP as a function of efficiencies results out of an energy balance and can be given as follows:

$$COP = \frac{\dot{Q}_0}{P} = \frac{t_1 - t_3 - \eta_{is,e}(t_3 - t_{4s})}{\frac{t_{2s}-t_1}{\eta_{is,c}} - \eta_{is,e}(t_3 - t_{4s})}, \tag{1}$$

where t_1 is the compressor inlet temperature, t_{2s} the isentropic outlet temperature of the compressor, t_3 is the expansion machine inlet temperature and t_{4s} the isentropic outlet temperature of the expansion machine. $\dot{Q}_0$ is the cooling capacity, P the energy consumption and the efficiencies are $\eta_{is,c}$ for the compressor and $\eta_{is,e}$ for the expansion machine. As can be seen, the efficiency for the expansion machine influences the COP twice: the cooling capacity (numerator) and, secondly, the rejected energy to the compressor (denominator), whereas the efficiency of the compressor influences the COP only due to the power consumption (denominator). For this reason, special attention must be given to the expansion machine.

First experimental investigations have been conducted using a mechanically driven scroll compressor and a free running turbo compressor–expander unit (exhaust gas turbo charger) in a so–called bootstrap cycle with internal heat exchanger (refer to Fig. 2).

The efficiencies of both machines are in the order of 0.6, which is rather low for a good COP of the air cycle [3]. Refering to the working principle of super charging systems, remarkable increases in efficiencies cannot be expected. This means that a machine concept must consist of special highly efficient machines like adequate turbo machines or pressure wave machines.

3.3 Machine Concept Using Turbo Machines

In cooperation with a manufacturer of supercharger systems, a compact unit, a turbo compressor, a turbine and an electrically driven asynchronous motor, was developed. The motor is designed to provide a rotational speed of up to 100000 rev/min which allows to reach rather high efficiencies. This compact unit is sketched in Fig. 3.

In order to investigate the influence of the efficiencies of the compressor–expander–unit under different load conditions a theoretical study using a simulation program was conducted. The simulation program was developed within the EC project [4] to meet the following main requirements:

- flexibility with respect to the simulation of a variety of different cycle configurations,
- accurate results for all implemented modules for components at full and part load conditions.

All different components of an air cycle system, compressors, expanders, heat exchangers, tubes, cold storage rooms, ambients and humidity separating devices are implemented as modules. Each module can be connected to

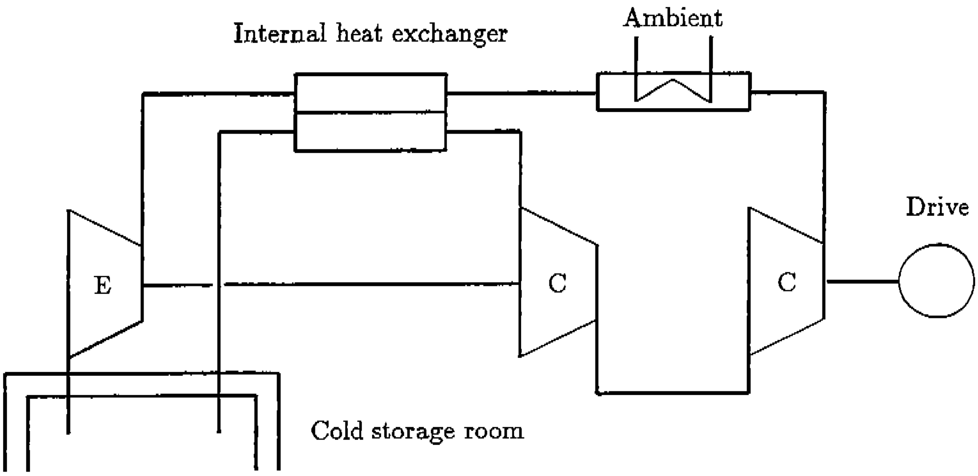

Figure 2: Bootstrap open cycle configuration with internal heat exchanger.

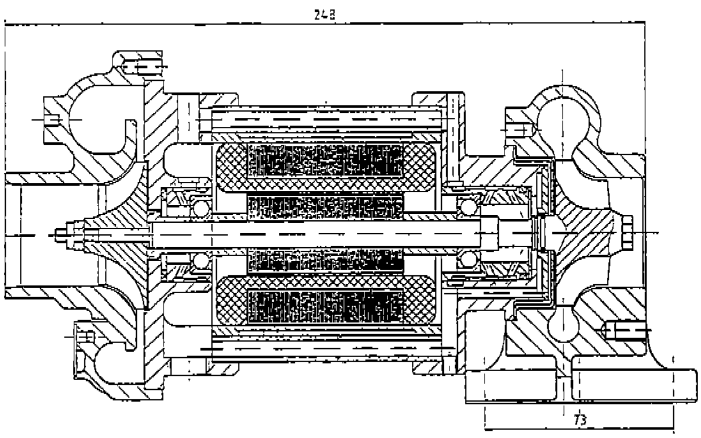

Figure 3: Compact unit for use in an air cycle refrigeration system.

another module by a node. The connections must be defined by the user, depending on the configuration of the cycle. Out of all connections the program itself sets up a set of equations which are solved using a slightly modified iteration method which is based on the Newton–Raphson–scheme. To perform accurate modules for the compressor and the expander, characteristic diagrams for performance data at different load conditions from the manufacturer were used to implement polynomial functions for the isentropic efficiency, the mass flow rate, the rotational speed and the power consumption. Data for heat exchangers were taken from measurement results and polynomial functions were established to calculate the pressure drop and the heat transfer capacity. Pressure drops in tubes are calculated depending on the Reynolds number and boundary conditions were set to meet the requirements for transport refrigeration (30 $°C$ ambient temperature and +5 $°C$ in the cold storage room). Simulation results have shown that the concept, using the compact unit, is of great promise. Fig. 4 (left hand side) shows simulated COP results for both $+5°C$ temperature and $-30°C$ in the cold storage room as a function of the rotational speed.

The calculated efficiencies for the expansion machine are based on characteristic data available for use with $600°C$ exhaust gas. From experience with the test rig the efficiencies for use with lower temperatures are about 10% to 15 % lower. The efficiency for the compressor is accurate because the available characteristic data meets similar conditions as there are at the test rig. Fig. 4 (right hand side) shows simulated results for the COP–value with respect to different efficiencies for the expander while using real characteristic data for the compressor and the heat exchanger. As can be seen, efficiencies in the order of 0.8 are required in order to get a resonable COP.

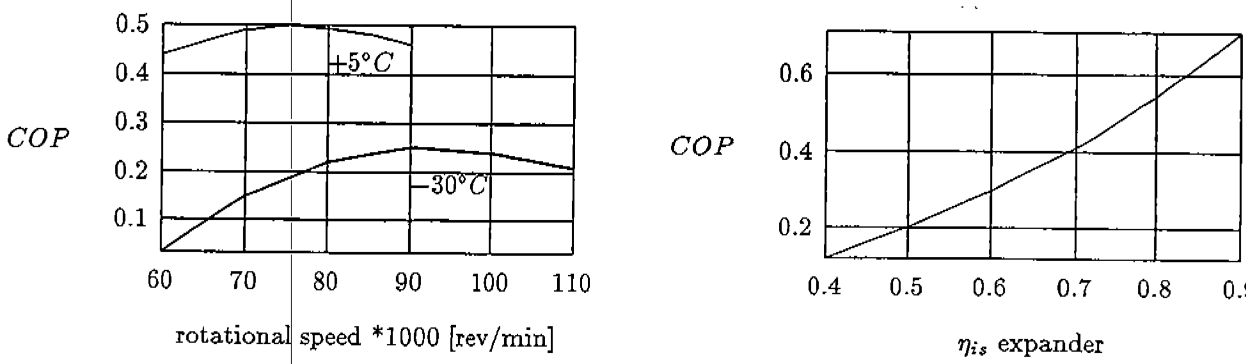

Figure 4: Left: Simulated COP results for $+5°C$ and $-30°C$ in the cold storage room. Right:Simulated COP results as a function of the efficiency of the expander.

The difference between the influence of the compressor efficiency and the influence of the expander efficiency can be precised by evaluating the partial derivative of the COP with respect to the efficiencies. This results in:

$$\frac{\partial(COP)}{\partial\eta_{is,e}} = 1.13 \qquad and \qquad \frac{\partial(COP)}{\partial\eta_{is,c}} = 0.88.$$

These values are calculated using the same conditions as given above and assuming that both efficiencies are exactly 0.7. The result shows that the efficiency of the expansion machine is more important than the efficiency of the compressor as already explained with Eqn. 1.

The following Table 2 compares values for the coefficient of performance COP for a standard R22 vapor compression system with results from the test rig, obtained with a scroll compressor, and the results for the turbo-unit.

Table 2: A comparison of COP for different systems.

R22 vapor compression cycle	2.5
Turbo–compressor–expander–motor–unit	0.5
Test rig (scroll compressor with turbo supercharger)	0.2

As can bee seen, there is a significant increase in COP when using the compact unit instead of using the scroll compressor. There remains a significant difference to the R22–systems. This is because these COP numbers do not take into account that R22–systems need additional energy, e.g. to drive a blower, for defrosting and for capacity control. Comparing the R22 system with respect to extra energy consumption to the R12 system for the German ICE 1, it can be said that the overall COP is in the order of 1.2 and so much lower than the stated value of 2.5.

Further increases in COP for air cycles using turbo machinery can be achieved by selecting better combinations for compressor and turbine, which will meet the requirements of a cold air cycle system better than the current configuration, which is designed for supercharging system. Maximum efficiencies of about 0.8 can be expected for turbo machines in this capacity range when applying adequate design work in this field.

3.4 Machine Concept Using Pressure Wave Machines

In order to get highest efficiencies, especially for the important expansion machines, the pressure wave machine can be recommended. This machine is a combined compressor and expander. Therefore it has to be integrated either into a bootstrap unit or into a cycle using an additional compressor, working parallel. The machine has a rotor which consists of small chambers that are located concentrically around the shaft. The housing on the left hand side (see Fig.5). represents the expander and the other housing represents the compressor. The housings have inlet and outlet lines for the air flow. Compressed air, entering the rotor by inlet line 3 and leaving by outlet 4 is expanded by giving its energy to the air that is compressed, entering at inlet line 1 and leaving at outlet 2. The

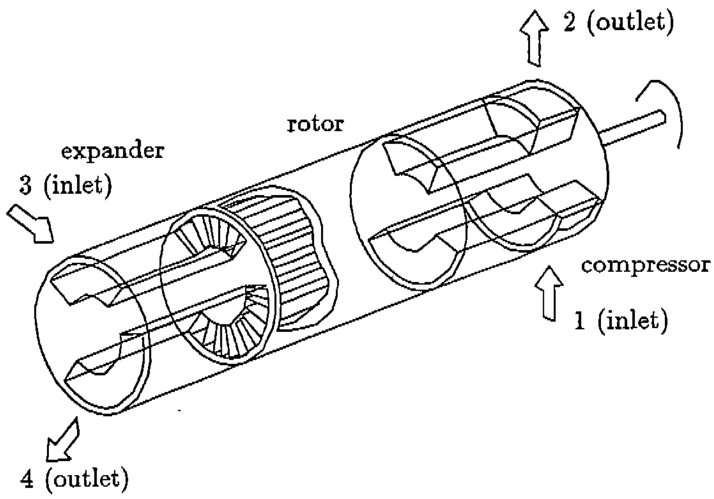

Figure 5: Sketch of a pressure wave machine.

energy transmission between both fluid flows occurs by pressure and expansion waves which results in very few losses and high efficiencies for compression and expansion.

Due to the fact that there is a direct contact between both fluid flows there is a difficulty in calculating efficiencies. The air flow entering via inlet line 3 must not necessarily exit completely at outlet line 4. Depending on the boundary working conditions which are pressure ratios as well as rotational speed there can be a leakage mass flow rate from inlet 3 to outlet 2 or from inlet 1 to outlet 4 (ref. Fig. 6). For this reason all different mass

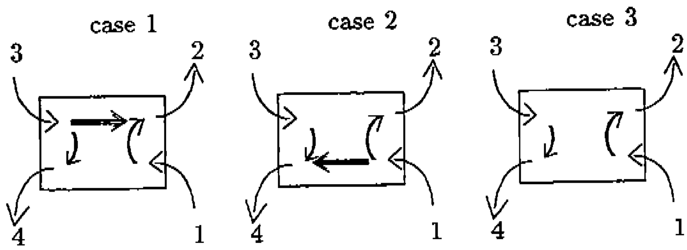

Figure 6: PWM: Definitions for efficiencies

flow rates have to be considered for evaluating efficiencies. Applying an energy balance to this system the following equations apply in three different cases:

Case 1:

$$\dot{m}_3 > \dot{m}_4 \qquad \eta_{is,c} = \frac{T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}} - 1\right]}{\frac{\dot{m}_2}{\dot{m}_1}(T_2 - T_3) + T_3 - T_1} \qquad \eta_{is,e} = \frac{T_3 - T_4}{T_3\left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{\kappa-1}{\kappa}}\right]} \tag{2}$$

Case 2:

$$\dot{m}_1 > \dot{m}_2 \qquad \eta_{is,c} = \frac{T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}} - 1\right]}{T_2 - T_1} \qquad \eta_{is,e} = \frac{T_3 - \frac{\dot{m}_4}{\dot{m}_3}(T_4 - T_1) - T_1}{T_3\left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{\kappa-1}{\kappa}}\right]} \tag{3}$$

Case 3:

$$\dot{m}_3 = \dot{m}_4; \;\; \dot{m}_1 = \dot{m}_2 \qquad \eta_{is,c} = \frac{T_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{\kappa-1}{\kappa}} - 1\right]}{T_2 - T_1} \qquad \eta_{is,e} = \frac{T_3 - T_4}{T_3\left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{\kappa-1}{\kappa}}\right]} \tag{4}$$

On a second test rig in Hannover, the pressure wave machine was investigated in order to study the characteristic behavior of this type of machinery as well as to get results for the efficiencies. Fig. 7 shows the resulting efficiencies for the compression side $\eta_{is,c}$ and for the expansion side $\eta_{is,e}$. The results were obtained at the test rig with an original pressure wave machine that was designed to work as a supercharger. The boundary working conditions were 21°C compressor inlet temperature, 30°C expander inlet temperature and 1.4:1 as the pressure ratio.

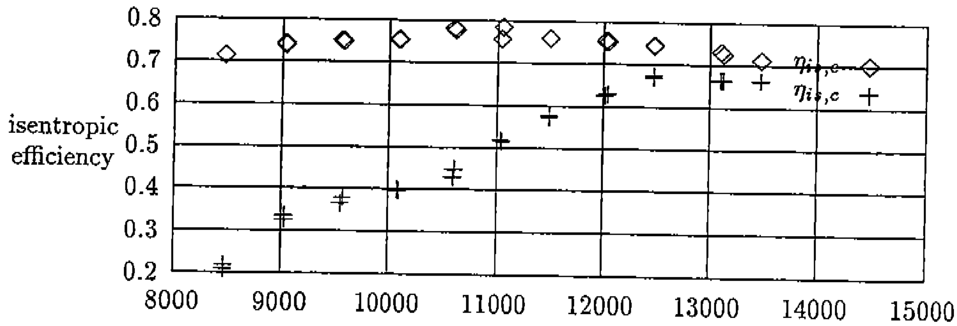

Figure 7: Measured efficiencies for the pressure wave machine.

Further increases in efficiencies are possible by using computer simulation tools to figure out the optimum geometry for the inlet and outlet lines. Such a simulation program was recently developed and is currently checked for accuracy. The program is based on two finite difference schemes, a one–dimensional to predict fluid dynamic properties inside the rotor and a two–dimensional to predict the fluid dynamic properties in the inlet and outlet lines. Both schemes are basically solving the conservation laws for mass–(5), momentum–(6) and energy–(7) which are (in the two–dimensional case) given by:

$$\frac{\partial \varrho u}{\partial x} + \frac{\partial \varrho v}{\partial y} = 0 \tag{5}$$

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\varrho}\frac{\partial p}{\partial x}; \qquad u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} = -\frac{1}{\varrho}\frac{\partial p}{\partial y} \tag{6}$$

$$h_{tot} = c_p T + \frac{u^2 + v^2}{2} \tag{7}$$

At least inside the rotor there are shock waves present which results in discontinuities for the variables velocity u, v, temperature T and pressure p. For this reason the program uses the mass, momentum and the energy as variables, which are constant even when shock waves are present. A two–step variant of the Lax–Wendroff–scheme [8], using a predictor and a corrector step (Mac Cormack, 1969), was implemented in two ways. The first calculates an unsteady state, incompressible flow with friction and the second one calculates the same flow (using the Euler–equations (6)) without friction. The differences of both are not too big, but disregarding the friction results in faster running programs. The results are confirmed applying the algorithms to a shock tube problem. Concerning flow velocity, speed of the shock wave, temperature and pressure, the results are accurate within a tolerance of about 2% comparing theoretic results with the simulation results.

In order to get finite differences a mesh was used to get fixed geometric places in two dimensions for the inlet and outlet lines and in one dimension for the rotor. Both systems are connected via boundary conditions saying that the fluid dynamic properties are the same where rotor and inlet/outlet–lines are connected. On the outer side of the inlet/outlet lines the pressure is given as boundary condition and for both inlet lines (line 1 and 3) temperatures are given as well. The program calculates first one revolution for the rotor, assuming start values as boundary conditions, and then applying the same time period to calculate the properties of all inlet/outlet lines. The results are used to calculate better boundary conditions for the rotor as well as for the inner part of the inlet/outlet lines. Repeating these steps results in constant conditions over the whole machine after calculating about 30 revolutions.

The results of the program can be used to produce figures containing streamlines, temperatures and pressures at each point inside the machine. Especially the streamline diagram can be used to fit the geometric position of the inlet and outlet lines to an optimum position. The current state is that the results are confirmed with measured data to make sure that this tool is delivering accurate results.

4 CONCLUSIONS

An environmentally benign working fluid is a fluid which consists only of ingredients that are already in the atmosphere. There is mainly one fluid which can fulfill this: air. This paper gives an impression of some historical aspects and an idea of the current state of the art with respect to compression and expansion machines for use in air cycle systems for low to medium capacities.

The use of standard equipment from different supercharger systems cannot be the final solution for air cycle technology, but it is from the current point of view one possibility to build up systems which are working with reasonable performance. Increases in efficiencies for turbo machines can be achieved by developing compression and expansion machines especially for air cycle applications, as already done for aircraft systems and for the new German high speed train. Taking into account that the price is always an argument, the needed machines for low capacities in other applications like transport cooling can be developed on the basis of standard exhaust gas turbo chargers. Significant changes in the design have to be made for the turbine to increase the efficiency.

It is further recommended to use the pressure wave machine as compressor and expander instead of a turbo machine, especially for small to medium mass flow rates. This machine has high efficiencies and a very simple design resulting in low prices for such machines. One needs accurate design tools, which are presented in this paper, in order to get optimum performance results. Currently, a complex simulation program is verified using measured data. First results are accurate with respect to the prediction of efficiencies and temperatures.

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