REVERSIBLE R744 (CO_2) HEAT PUMPS APPLIED IN PUBLIC TRAINS

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ABSTRACT

This paper presents opportunity for use of R744 (CO_2) as refrigerant in the air conditioning system in public trains. The R744 system shall provide cooling in the summer and heating in the winter. Today 75% of the air conditioning systems in trains use HFC-134a as refrigerant. The GWP of HFC-134a is 1410 while R744 is 1(0). A replacement from HFC-134a to R744 gives possibilities of large environmental savings. Different technical system solutions of the heat pump are presented, each with its own method of providing cooling and heating.

Solution I; changes between cooling and heating mode are performed by changing the direction of the refrigerant flow through the system.
Solution II changes between cooling and heating mode are done by changing the configuration of the air streams (movable hatches) through the heat exchangers.
In Solution III the refrigerant flow direction and the configurations of the air streams is always the same. The whole heat pump is placed on a turntable unit and the change between cooling and heating is done by rotating the entire heat pump 180°.

In all the three technical solutions there are separated heat exchangers for fresh and exhaust air. This gives an energy efficient system which recovers heat from the exhaust air. Computer simulation shows that a system solution with one evaporation pressure and one stage compression is problematic for low ambient temperatures; the system must stand temperatures to -40 °C. A system solution with two levels on the evaporation pressure and a two stage compression showed to improves the COP from 1,7 to 3,2 when the ambient temperature at -40 °C.

Norway is a country with cold climate. Weather statistics show that a train which drives in Oslo every day from 0600 to 1800 throughout a year will need cooling 3% of the time and heating 83% the time. This heating should be done by a heat pump and not with electrical heating as today. Results of the computer simulation shows that the annual energy consumption of heating the train will be reduced by 78 % if the designed R744 heat pump is used in stead of electrical heating.

1. INTRODUCTION

Vehicles on the roads are not the only one which contributes to global HFC emissions, as shown in a survey "Final Report Maritime, Rail, and Aircraft sector" by *Schwarz &Rhiemeier* (2007)

In 2006 the rolling stock of the railway, tram and metro operators consisted in the EU of 175.000 units, 65.000 of them equipped with air conditioning systems. 75 % of these HVACs applied R134a as refrigerant and 25% used HFC-407C. The total charge (stock) of refrigerant in the railway sector is 1.605 tonnes CO2 - eq. This is 26% more than in the Maritime sector. However, the HFC (& HCFC) emissions from the Maritime sector are much higher than for the Railway units, mainly due to poor service and high leakage rates of maritime units. The estimated annual leakage rate is around 40%, even when applying indirect systems 20% of the refrigerant is lost every year. In the Railway Sector the leakage rate is 5% for the majority of the vehicles

Estimations of the emissions in 2020 with a business-as-usual assumption show that the HFC emissions from the railway sector will double to 174 tonnes CO2 eq while the emissions from the maritime sector will increase to the threefold to 1 141 tonnes CO2 eq.

Morgenstern & Ebinger (2008) perform research on future railway vehicle air conditioning systems and how the energy consumption can be reduced. R744 and HFO-1234yf are two evaluated refrigerants.
If HFO-1234yf is applicable in the public transport sector business as usual can be practised, beyond safety issues known technology will be applied. However, concerns are shown about this new fluid due to flammability issues and the deadly toxic decomposition products. Neither are all thermodynamic and technical properties of this refrigerant known, understood nor published.

R744 HVAC systems have to stand much higher working pressures (high pressure side ca 80-120 bar). A completely new technology has to be applied, which may result in higher initial costs of the first HVAC units. As commonly known, carbon dioxide applied as working fluid has the advantage that it is natural, environmentally friendly, globally available at low costs and there are no unknown effects which will ban this fluid applied as refrigerant in the future.

Annual energy savings compared with conventional R134a system for air conditioning of passenger coaches are measured by *Morgenstern & Ebinger*, (2008). Show results of an experiment were annual energy demands for different HVAC system in railway passenger coaches are compared to an existing R134a system (100%). In this experiment a prototype CO2 system had lower energy demands (~52%) than an optimized R134a system (~64%).

2. MOBILE HVAC SYSTEM LAYOUT

There are many ways to design the HVAC system which are able to perform both heating and cooling, as shown by *Hafner (1998; 2003).* Different system layouts for mobile HVAC systems will be presented and discussed.

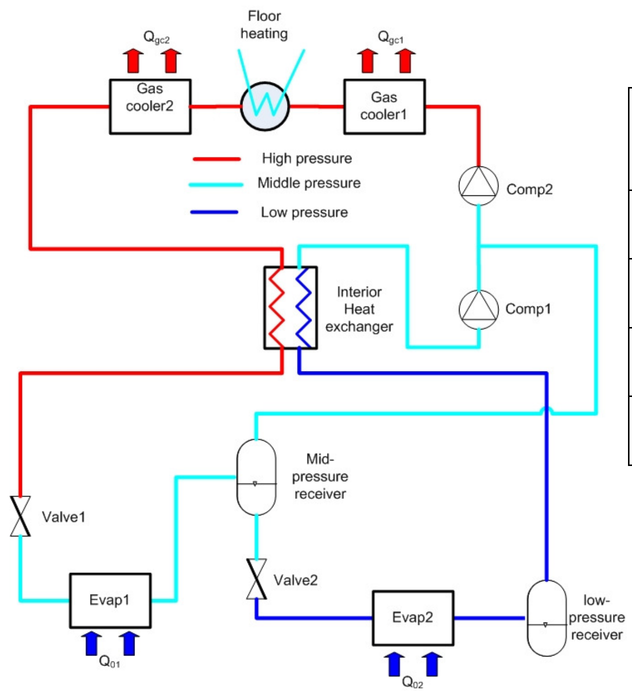
Solution I. The heat pump system changes from cooling mode to heating mode by reversing the direction of the refrigerant flow through the unit. In this solution the heat exchangers have to be designed to operate both as evaporator and gascooler. (not part of this study)

Solution II. The switch between heating and cooling mode is done by redirecting the air stream through the HVAC system. The refrigerant flow inside the heat pump is not re-directed, i.e. it flows in the same direction and the air heat exchangers do not change function from gas cooler to evaporator when the system mode is changed.

Solution III. The heat pump is located on a turntable unit and changes from cooling to heating mode are done by rotating the table by 180°. The CO2 always flow in the same direction and the configuration of the different air streams is also similar in the two operating modes.

A system design where fresh/ambient and exhaust air is mixed in the first place and then sent to a heat exchanger is not preferable since some of the potential energy/exergy in the exhaust air is lost. Therefore separated evaporators and gas coolers are proposed for fresh and exhaust air. All the system solutions are designed to contain four air to refrigerant heat exchangers, one refrigerant to glycol heat rejecting exchanger, to supply heat to a possible floor heating circuit and an internal suction line heat exchanger.

Due to different temperature levels of the inlet, outlet, re-circulated and ambient air, two gas coolers and two evaporators are applied in air ducts of theses HVAC systems. The fresh/ambient air has its temperature level and a defined volume flow rate through the evaporator before entering the passenger compartment and the heat rejecting gascooler which rejects the heat to the ambient during the season when cooling of the compartment is required. The interior / passenger compartment air is already treated and has a temperature level close to the desired compartment temperature, i.e. in cooling mode exhaust air, which leaves the passenger compartment, has most of the time a lower temperature than ambient, which allows for a further temperature reduction of the refrigerant downstream of the first gascooler.



	Cooling mode ($T_{\_amb}$ > $T_{\_exhaust}$)	Heating mode ($T_{\_amb}$ < $T_{\_exhaust}$)
GC 1	Ambient air	Comp. air (recycled)
GC 2	Air which exits compartment	Ambient air (enters comp.)
EV 1	Ambient air (enter comp.)	Air which exits compartment
EV 2	Comp. air (recycled)	Ambient air

Figure 1 Circuit of R744 HVAC cycle (example) and configuration table of the air stream directions.

The glycol heat exchanger (located between the gascoolers) is applied for floor heating and is only in use during heating mode.

In both heating and cooling mode the coldest air crosses gas cooler 2, thereby the lowest possible gascooler refrigerant side exit temperature can be achieved, which can be of temperature levels below ambient temperature.

From a thermodynamic point of view system designs containing two evaporators in series and therefore operating at similar evaporation pressure are not an optimal solution, when the air side temperatures are mostly quite different. In addition, refrigerant distribution at the inlet of the second evaporator might be challenging.
There are many possibilities to handle multi evaporation pressure levels:

- Expansion to lowest suction pressure downstream of the evaporator (not preferable),
- Multi stage compression
- Expansion via ejectors
- Etc.

As shown in Figure 1 two stage compression can be applied as an example, while a second expansion device reduces the pressure of the liquid refrigerant downstream of the first accumulator. In this way the two evaporators can handle different evaporation temperatures.

During the heating period the exhaust air is already heated, however, it has to be partly replaced by fresh air. Heat from air which leaves the passenger compartment is partly recovered in evaporator1 while the ambient air at a lower temperature is the main heat source to be cooled in evaporator 2.

Air reversing solution (II)

Solution II uses movable hatches to redirect the air flow through the HVAC system when a mode change is required. This system requires an air duct and fan system that controls the airstreams and changes them according to the requirements of the two modes. The evaporators do not have to be dimensioned to stand high pressure side since they always operates as evaporators. The system circuit is simple and there are no three way valves to control direction of refrigerant flow. This will reduce the chance of failure and leakages in the system circuit.

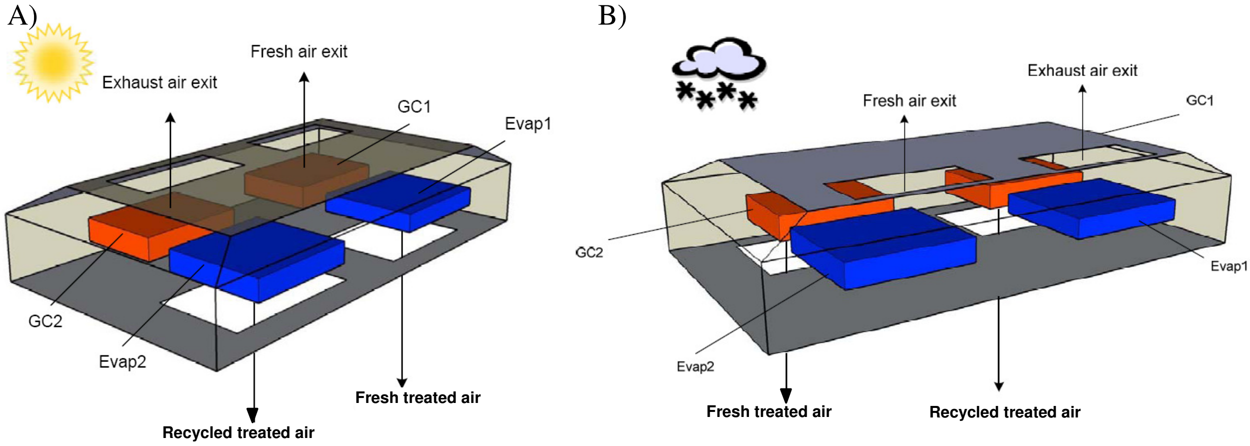


Figure 2 Configuration of the air reversing HVAC system at cooling (A) and heating (B) mode

During cooling mode, re-circulated air from the passenger compartment enters Evap2, while ambient air is cooled by Evap 1.
On the gas cooler side there are two holes in the unit roof to let the air exit the system. Ambient air on the gas cooler side first has to flow down to meet the face area of GC1 on the bottom side and than through the heat exchanger to exit the system. Air from the compartment enters the unit and exchange heat with GC2 then leaving the HVAC system.

As the figure shows the heat exchangers are the same as in cooling mode, however, the air streams changes diagonal. The holes in the floor have also moved from the evaporator side to the gas cooler side, since during the cold season the heated air has to enter the coach. The holes in the roof are moved from above the gas coolers to above the evaporator, since it is the cooled air from either the compartment (energy recovery) or the ambient air (main heat source) which has to exit the system.

Turntable solution (III)

In this solution the heat pump is located on a turntable device. In Solution II all the air streams changes diagonal when to mode change from cooling to heating. When the heat pump rotates 180° the heat exchangers position changes diagonally. Solution III have a circuit where the refrigerant flows in the same direction as in Solution II, and the air streams are similar in cooling and heating mode as shown

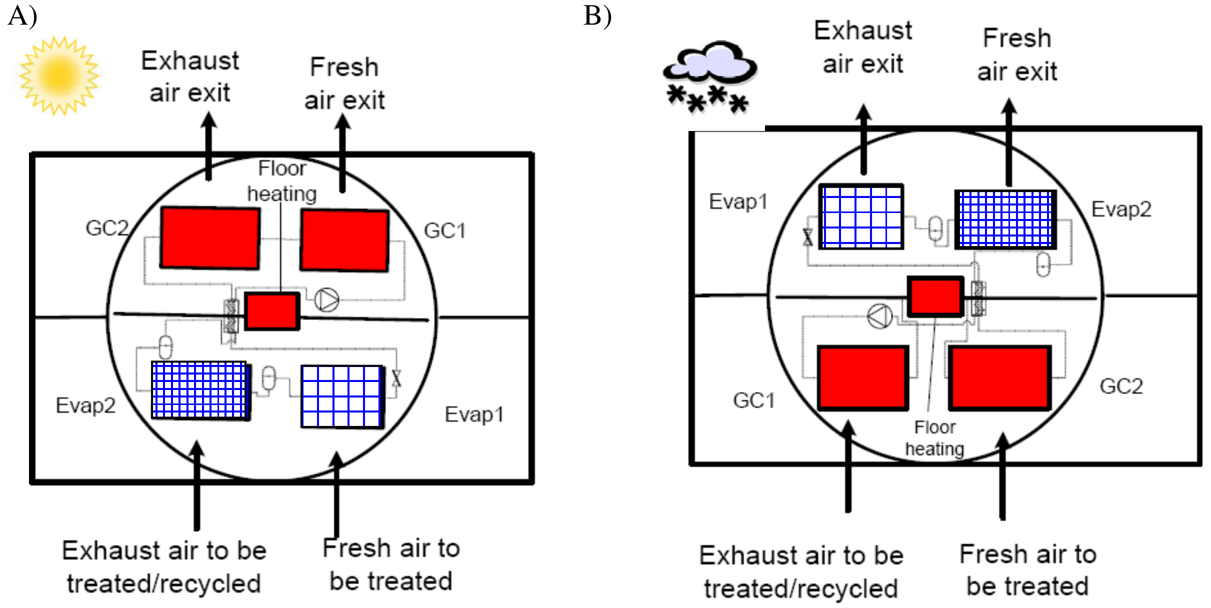


Figure 3 Configuration of the turntable HVAC system at cooling (A) and heating (B) mode

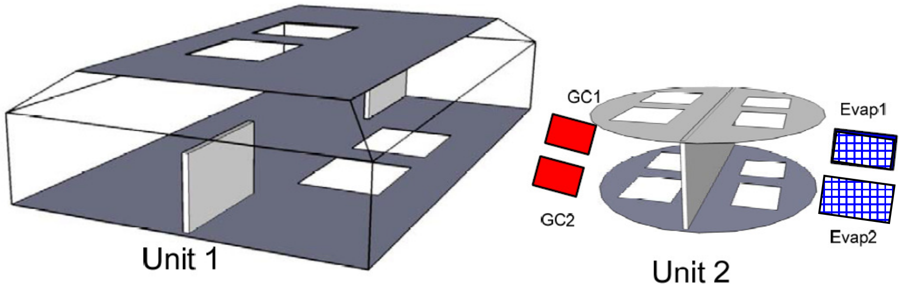


Figure 4 Cover of turntable HVAC system (Unit 1) and turntable (Unit 2)

The cover of the rooftop HVAC system (unit1) includes the turntable device (unit2) where the heat pump is located**.** Unit1 has a wall at each side and unit 2 has a centre wall, together these walls separate the hot and cold side of the HVAC module.

Unit1 has to air connections in the bottom where the treated air is distributed into the duct system of the passenger compartment. The two holes in the roof are the connection to ambient air inlet and where the exit air leaves the system.

The turntable device has two holes in the floor and the roof on both sides.
During cooling mode the treated air leaves the evaporator side and enters the treated air duct through the holes in unit2 and unit1. The holes in the roof above the evaporators on unit2 right hand side are closed by the roof on unit1. In the same way the holes in the floor of unit2 left hand side are closed by the unit1 floor. In the summer season both exhaust and fresh air exchange heat with the gas cooler side flow through the holes in the roof of unit2 and unit1 and than exit the system.

When the system changes from cooling to heating unit2 is rotated 180 °. Than the holes in the floor fit for the gas cooler side to enter the passenger compartment and the holes in the roof fits for the air flow rate which has to be released via the evaporator, while the holes in the floor on the evaporator side are closed as well as the holes in the roof on the gas cooler side.

3. SIMULATION RESULTS

The computer simulation of the system circuit as shown in Figure 1 is performed by applying and combining the following tools: *RnLib, HXsim* and *ProII.*
- *RnLib* is a refrigerant properties library developed by *NTNU-SINTEF*. It calculates thermodynamic data and transport properties of refrigerants and refrigerant mixtures.
- *HXsim* is a simulation programme developed by *SINTEF*. It is for dimensioning and performance calculations of heat exchangers, i.e. real heat exchangers models are applied during the calculation.
- *PROII/PROVISION* is a simulation programmes which simulates a whole system circuits with heat exchangers, compressors, receivers, expansion valves.

The following assumptions are made

Cooling mode

- Temperature of air leaving the passenger compartment: 25°C
- Temperature of air leaving the evaporator: 15°C (t_o = 5°C)
- Gascooler 1 refrigerant exit temperature = t_{amb} + 5K
- Gascooler 2 refrigerant exit temperature = $t_{air\ coupè\ 2}$ + 2K

Heating mode

- Temperature of air leaving the passenger compartment: 20°C
- Temperature of air entering the compartment (heated + re-circulated air): 30°C
- Evaporating temperature = t_{amb} - 10K
- Floor heating (glycol): 25 to 30 °C; ~ 3 kW
- Gascooler 2: Pinch temperature is set to 5 K. When the cycle is transcritical the pinch point occur in the outlet. When the cycle is subcritical the pinch occurs inside the exchanger.

The applied and required fresh air ratios depending on the ambient temperature are shown in Figure 5. Depending on the fresh air ratio, the cooling capacity varies as shown in Figure 6.

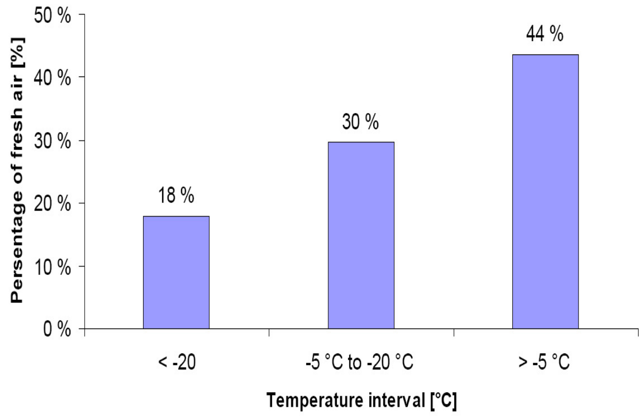


Figure 5 Required amount of fresh air per coach is dependent on the ambient temperature (NS-EN 13129-1, 2002)

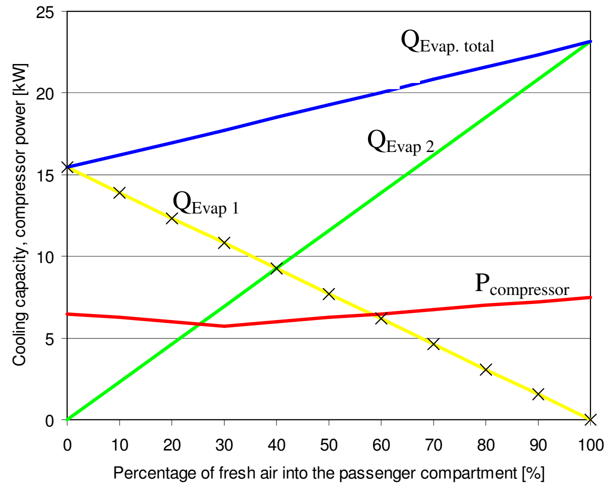


Figure 6 Cooling capacity and compressor power as a function of air re-circulation ratio.

Figure 7 shows the simulated HVAC system efficiencies at the entire temperature range where trains are operated in Europe. Depending on relative humidity of the ambient air a heating COP between 3 and 6 can be expected while during the cooling season efficiencies between 4.5 and 2 can be obtained.

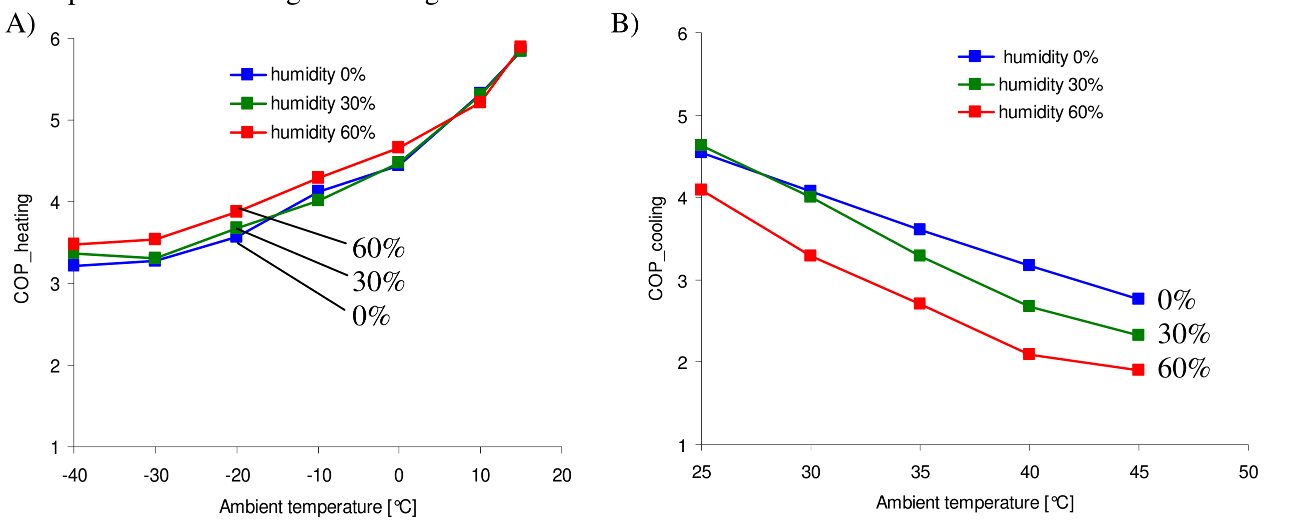


Figure 7 Simulated system efficiencies for heating (A) and cooling (B) mode of the HVAC systems.

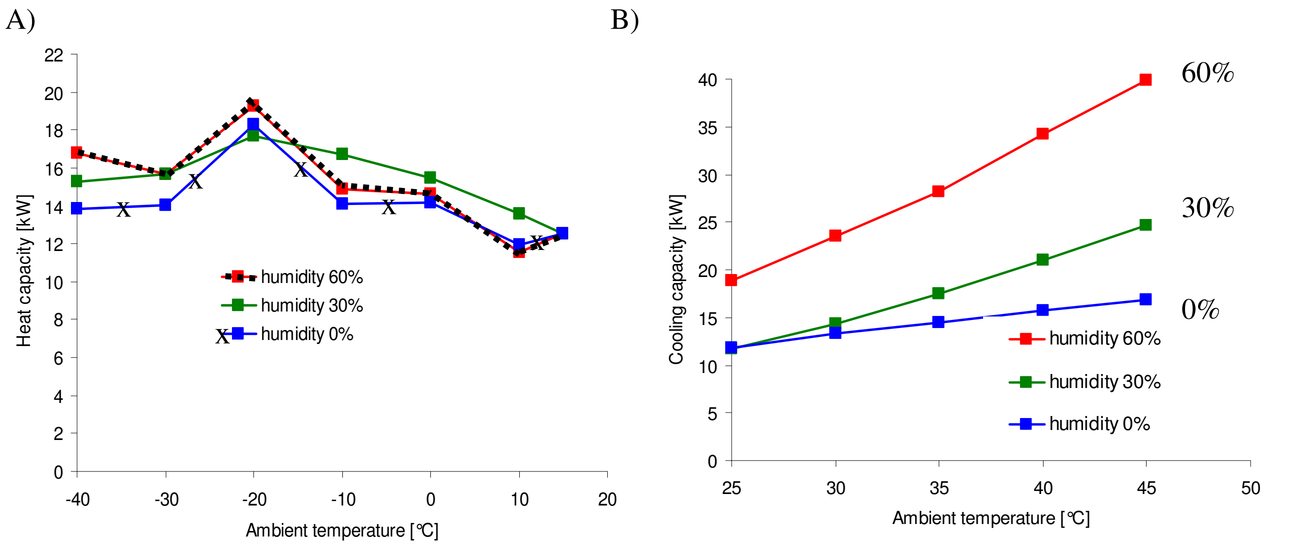


Figure 8 Simulated system capacities for heating (A) and cooling (B) mode of the HVAC systems.

Figure 8 shows the simulated HVAC system heating and cooling capacities at the entire temperature range where trains are operated in Europe. The heating capacity of the applied system configuration is in the range of 18 to 12 kW while the cooling capacity ranges from 12-19 kW at low ambient temperatures to 15-40 kW at high ambient temperature conditions.

These values are applied to calculate the energy saving potential for coaches operated in three different climate zones of Europe.

4. ENERGY SAVING POTENTIALS

Weather data from the METEONORM (2005) library as shown in Figure 9(A) are applied to calculate the energy consumption for a typical coach, in duty during the daytime from 0600 to 1800.In this case study Norway (Oslo) represents the cold European climate while Frankfurt climate was chosen as a Central European location with moderate ambient temperatures. The high ambient temperatures of Athens require a large portion of compressor energy for AC as shown in part B of Figure 9

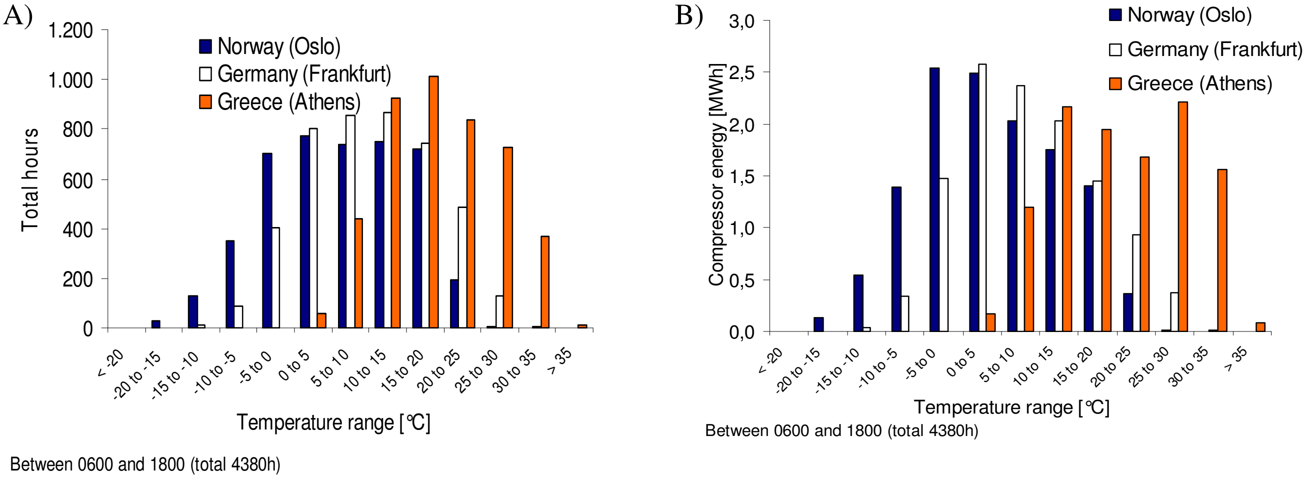


Figure 9 Applied weather data/temperature bin (A) and calculated energy consumption of the compressor of the HVAC system (B).

For the energy calculation it is assumed that a railway coach requires cooling when the ambient temperature is above 20 °C and heating below 15 °C. Part A of Figure 10 shows the different heating and cooling demands in Northern and Southern Europe. Due to the cold climate in Scandinavia, 96% of the energy is required for heating of the passenger compartment, while only 4% is applied for cooling. Even in Central Europe a small share of the energy is used to cool the coach to comfort levels. However, in Southern Europe about 60% of the Energy is applied for cooling of the passenger compartment.

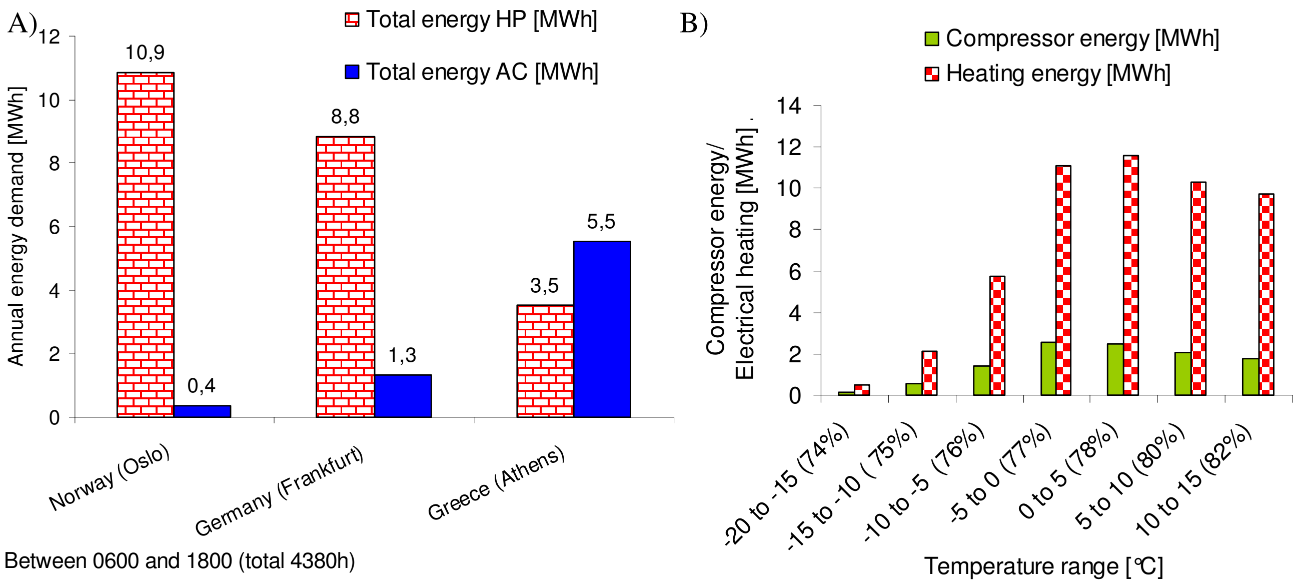


Figure 10 Annual energy demand per HVAC system (A) and Share of compressor energy versus total heating energy (B) for a coach applied in Oslo.

Figure 10 (part B) shows how much energy is required to heat a coach in Oslo at different ambient temperatures. When applying a heat pump up to 82 % of the electrical power can be saved since only the compressor energy has to be provided, while current systems require 100% electrical power to heat the coaches.

GHG emissions
About 65 000 of 175.000 railway-, tram- and metro units are equipped with air conditioning systems in the EU. The total refrigerant charge of these vehicles is ~1.180 metric tons. The average annual leakage rate is 5% in the railway sector. Based on this data each unit has approximately 18 kg refrigerant charge. Norwegian trains consist on average of 5 coaches which give each train a refrigerant charge of 90 kg. The calculations show that leakage from a train with the average annual leakage of 5% contributes to an annual Green House Gas emission of 63 metric tonnes CO_2 eq.

5. DISCUSSION / CONCLUSION

The energy saving potential could be further improved if the requirements for fresh air into the compartment would be based on air quality sensors inside the passenger compartment. The more people are travelling with a coach the more compartment air has to be replaced by the HVAC system.

A further improvement of the system performance is expected when work recovery devices like ejectors or expanders are applied to recover partly the unutilized expansion work in R744 HVAC devices.

Applying such compact R744 HVAC units for trains will contribute to a significant reduction of energy required to keep the passenger compartments at comfort level during all seasons and in most global climate zones.

7. ACKNOWLEDGEMENTS

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8. REFERENCES

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