All-year vehicle simulation with analysis of capacity control of R-134a and R-744 piston compressors for coach HVAC systems

Nicholas Carsten Lemke
TLK-Thermo GmbH, Germany, n.lemke@tlk-thermo.com

Christian Kaiser
TLK-Thermo GmbH, Germany, c.kaiser@tlk-thermo.com

Wilhelm Tegethoff
TLK-Thermo GmbH, Germany, w.tegethoff@tlk-thermo.com

Jürgen Köhler
Technische Universität Braunschweig, Germany, j.koehler@tu-braunschweig.de

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Christian KAISER[1]*, Nicholas LEMKE[1], Wilhelm TEGETHOFF[1], Jürgen KÖHLER[2]

[1]TLK-Thermo GmbH,
Braunschweig, Germany
c.kaiser@tlk-thermo.com, n.lemke@tlk-thermo.com, w.tegethoff@tlk-thermo.com

[2]Technische Universität Carolo-Wilhelmina zu Braunschweig, Institut für Thermodynamik,
Braunschweig, Germany
juergen.koehler@tu-braunschweig.de

*Corresponding Author

ABSTRACT

The state of the art for controlling air-side cooling capacity in conventional omnibus HVAC systems is the reheat of the cooled air. This method is pragmatic, although energetically highly inefficient. A more efficient way is to control the cooling capacity depending on actual cooling capacity demand. This is commonly achieved by adjusting the capacity of the refrigerant compressor. Various methods and technical solutions are feasible for adapting the capacity of the typical refrigerant compressor used in omnibus HVAC systems. For this purpose, speed control with a continuously variable transmission (CVT) or speed control by means of an innovative two-speed pulley gearbox based on a planetary gearbox as well as cylinder bank shutdown by suction gas interlock can be used. Also feasible is the novel method of combining speed control by means of a two-speed pulley gearbox and cylinder bank shutdown by suction gas interlock. The main object of the present study is to identify potential fuel savings and opportunities to improve cooling capacity through the use of the refrigerant compressor capacity adaption methods and techniques mentioned above. In respect thereof, this study looks at an R-134a and an R-744 coach air conditioning system. For this purpose, a vehicle simulation model was developed and three climatically different, realistic driving route scenarios (Germany, Portugal/Spain and India) are considered within this vehicle simulation. Based on these three driving route scenarios, virtual driving scenarios are performed on every 15th of the month for a representative year to determine average annual fuel consumption.

1. INTRODUCTION

Reciprocating compressors with constant displacement are typically used in air conditioning systems for conventional buses. The compressor is usually driven directly by the internal combustion engine, which is realized through the use of a belt drive and a magnetic clutch. Due to the speed-synchronous mechanical linkage to the engine and the constant compressor displacement, different control techniques and methods are necessary to realize variable cooling capacity for efficient use of the air conditioning system. This leads to the main objectives in the present study: Identifying energetically efficient methods to adapt the capacity of reciprocating piston compressors for air conditioning systems of conventional coaches and identifying energy savings potential as well as opportunities to improve the cooling capacity for different refrigerants and application scenarios. For this purpose, feasible techniques and methods for implementing a capacity adaption of the refrigerant compressor in omnibus air conditioning systems are described. This includes speed control by continuously variable transmission (CVT) as well as cylinder bank shutdown by suction gas interlock. Furthermore, the innovative speed control by means of a two-speed pulley gearbox based on a planetary gearbox as well as the novel method of speed control by pulley gearbox and cylinder bank shutdown by suction gas interlock are described. Our study of these methods is done by means of a detailed vehicle simulation of a coach. The vehicle model includes an R-134a and an R-744 air conditioning system, which is briefly described. Further on, three climatically different driving route scenarios for the comparative study are introduced. Following this, the numerical simulation results of the specified compressor capacity adaptation methods and techniques are presented depending on the three climatically different driving route scenarios.

2. COMPRESSOR CAPACITY CONTROL METHODS AND TECHNIQUES

In the following, different speed control methods and techniques such as speed control by continuously variable transmission (CVT) and by an innovative pulley gearbox based on a planetary gearbox as well as cylinder bank shutdown by suction gas interlock are described and specified for the subsequent study. Further, the novel method of combining speed control by pulley gearbox and cylinder bank shutdown by suction gas interlock is specified for the study.

2.1 Speed Control by Continuously Variable Transmission (CVT)

Our consideration of speed control by continuously variable transmission is conceptually based on a cone disc embracing gear. The cone disc embracing gear transmits speed and torque from the drive to the driven shaft by the force and friction connection of a V-belt or chain. One cone disc from each cone disc pair can axially displaced from their axes of rotation. An axial change in distance between a cone pair changes the contact radius of the V-belt or chain on the drive or driven side. For the forces acting in the circumferential direction, the effective radius changes and both speed and torque are converted (see Figure 1).

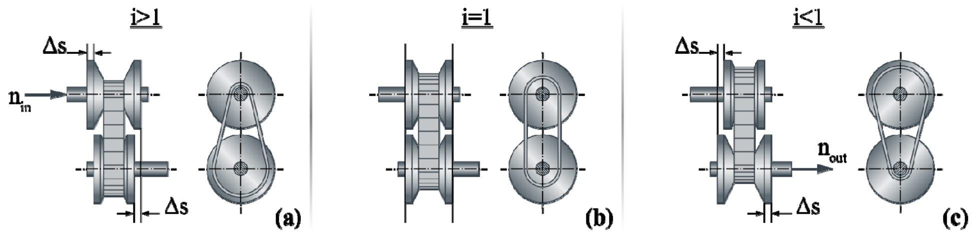

Figure 1: Operating principle of continuously variable transmission with cone disc embracing gear; (a) transmission into lower speed, (b) no speed transmission, (c) transmission into higher speed.

The study with CVT application will be evaluated for an ideal scenario and based on typical average efficiency. The ideal assumption without frictional power losses demonstrates the technical limit potential of the variable compressor speed control. For a realistically evaluated CVT, the calculation of typical mean frictional power losses based on an average efficiency of $\eta=0.89$, which was determined from a variety of specific research studies, e.g. Tenberge (1986), Sattler (1999), Kruse (2013). For the study of the ideal as well as the more realistic CVT application within the vehicle simulation, the refrigerant compressor speed limits in accordance with the manufacturer's specifications is considered when adjusting the CVT transmission ratio (see Table 1). The CVT transmission ratio is obtained by controlling the interior temperature (reflects the actual cooling demand) with a conventional PI controller.

Table 1: Refrigerant compressor speed limits as given in manufacturer's specifications.

Refrigerant application	Min. compressor speed	Max. compressor speed
R-134a	500 min^{-1}	3500 min^{-1}
R-744	500 min^{-1}	3000 min^{-1}

2.2 Speed Control by Pulley Gearbox

The compressor speed control with pulley gearbox considered here is based on a planetary gearbox integrated in a compressor belt pulley, which was presented for automotive application by Baumgart *et al*. (2006). With this integrated planetary gearbox, two transmission ratios ($i<1$ and $i=1$) can be implemented (Baumgart and Tenberge, 2010). Figure 2 shows the design, schematic and operating principle of the pulley gearbox. Insofar as the brake is closed and the clutch is released (switching position I), the gearing between the ring gear and the sun gear generates a transmission ratio into higher speed ($i<1$). If the brake is released and the clutch is closed, the gear unit rotates as one part (switching position II, $i=1$) and the gearbox runs without any friction losses. In this case, the refrigerant compressor is driven only by the transmission ratio of the belt drive. If both the brake and the clutch are released, the planetary gearbox is under-determined and decouples the refrigerant compressor from the belt drive (switch position III), thus enabling the refrigerant compressor to be disconnected from the drive as before with a conventional magnetic clutch. For the study within the vehicle simulation, Table 2 shows the selected transmission ratios for the refrigerant compressor drive with the two-speed pulley gearbox application depending on the different operating conditions of the considered climatically different driving route scenarios. For the selection of the compressor drive

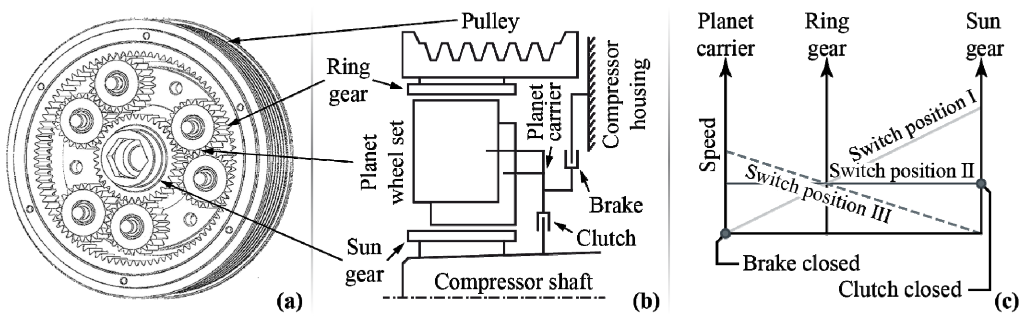

Figure 2: Design (a), schematic (b) and operating principle (c) of two-speed pulley gearbox.

total transmission ratios of belt drive and planetary gearbox, the conventional belt drive transmission ratio should be maintained. For the selection of the second additionally transmission ratio, the resulting transmission ratios of the CVT application were evaluated. Based on this, the second transmission ratio was selected in such a way that frequent shift operations of the two-speed gearbox during normal operational speed changes of the internal combustion engine in general driving can be avoided. This is also intended to prevent the superheat control from oscillating. For the Germany and Portugal/Spain driving route scenario, switch position I represents the maintained conventional belt transmission ratio, and for the India driving route scenario switch position II does so (see Table 2). The other transmission ratios in Table 2 are the newly selected transmission ratios. To adapt the refrigerant compressor speed, the two-speed pulley gearbox is controlled depending on the interior temperature (reflects the actual cooling demand). If the interior temperature reaches or exceeds the upper value of $t_{Set}+0.5K$, where t_{Set} is the interior set temperature, the two-speed pulley gearbox shifts into switch position I. If the interior temperature reaches or falls below the lower value t_{Set}, the two-speed pulley gearbox shifts into switch position II. Friction power losses with an active two-speed pulley gearbox in switch position I are calculated with a gear box efficiency of $\eta=0.96$ according to Baumgart (2010).

Table 2: Transmission ratios for refrigerant compressor drive by two-speed pulley gearbox application.

Refrigerant application	Driving route scenario: Germany and Portugal/Spain		Driving route scenario: India	
	Switch position I	Switch position II	Switch position I	Switch position II
R-134a	0.654	1.471	0.464	0.654
R-744	0.654	1.471	0.540	0.654

2.3 Cylinder Bank Shutdown by Suction Gas Interlock

The cylinder bank shutdown by suction gas interlock is usually installed in the cylinder head of one cylinder or cylinder pair of the refrigerant compressor. Figure 3 illustrates the schematic and operating principle of the suction gas interlock. Insofar as no voltage is applied to the solenoid valve, the high pressure pass to the locking valve is closed and the spring pushes the locking valve into the upper valve seat. The connection between the suction chamber and suction gas line is open and the refrigerant compressor operates at its full capacity. When the solenoid valve is actuated, the access of the high pressure pass to the locking valve is opened, high pressure refrigerant flows above the locking valve and presses it into the lower valve seat. As a result, the connection between the suction chamber and suction gas line is blocked and the refrigerant compressor operates at reduced capacity.

To adapt the refrigerant compressor capacity, the suction gas interlock is controlled depending on the interior temperature (reflects the actual cooling demand). If the interior temperature reaches or falls below the lower value t_{Set}, where t_{Set} is the interior set temperature, the suction gas interlock application is activated. If the interior temperature

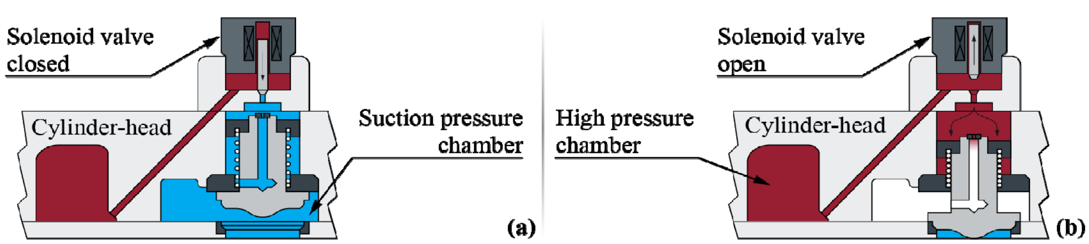

Figure 3: Schematic and operating principle of suction gas interlock: (a) inactive interlock, (b) active interlock.

reaches or exceeds the upper value of t_{Set}*+0.5K*, the suction gas interlock application is deactivated. For the study of cylinder bank shutdown by suction gas interlock within the vehicle simulation, the refrigerant compressor displacement volume can be controlled between *50%* and *100%*.

2.4 Combination of Pulley Gearbox and Suction Gas Interlock

Combining the two-speed pulley gearbox and suction gas interlock described above can potentially improve the performance of the refrigerant compressor capacity control compared to their individual application. Table 3 shows the selected transmission ratios for the two-speed pulley gearbox application for the specific combination with the suction gas interlock application. Compared to the selected transmission ratios for the isolated pulley gearbox application in Table 2, the presented transmission ratios are modified for the Germany and Portugal/Spain driving route scenarios based on a preliminary study (see Kaiser *et al.*, 2013). With this adaption of the additional transmission ratio, frequent shift operations of the two-speed gearbox and activation of the suction gas interlock during normal operational speed changes of the internal combustion engine in general driving mode can be avoided. This is also intended to prevent the superheat control from oscillating. For the India driving route scenario, the combination of two-speed pulley gearbox and suction gas interlock has less influence to each other, so the transmission ratios are identical as shown in Table 2. For the study within the vehicle simulation, the refrigerant compressor capacity control by means of the two-speed pulley gearbox application and the suction gas interlock application are used as follows: First, the refrigerant compressor speed is adapted by the two-speed pulley gearbox. Afterwards the cylinder shutdown by suction gas interlock can be activated. Use of the two-speed pulley gearbox and suction gas interlock is still controlled depending on the interior temperature as described earlier.

Table 3: Transmission ratios for refrigerant compressor drive when combining two-speed pulley gearbox and cylinder bank shutdown by suction gas interlock.

Refrigerant application	**Driving route scenario: Germany and Portugal/Spain**		**Driving route scenario: India**	
	Switch position I	**Switch position II**	**Switch position I**	**Switch position II**
R-134a	0.654	1.0	0.464	0.654
R-744	0.654	1.0	0.540	0.654

3. VEHICLE SIMULATION MODEL AND DRIVING ROUTE SCENARIOS

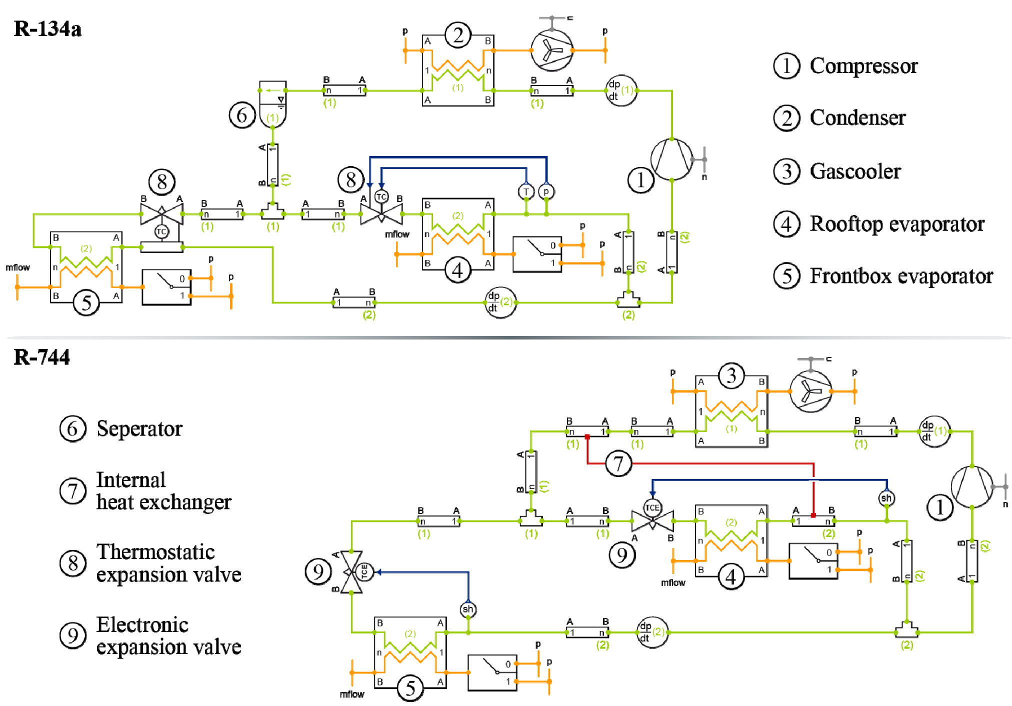

Figure 4: Principal structure and simulation model of R-134a and R-744 refrigerant circuit for bus air conditioning system: R-134a-based system (top), R-744-based system (bottom).

A complete physical vehicle model of a coach was developed and validated for research issues in the realm of air conditioning systems in buses, see Kaiser (2018). The overall model includes the following subsystems: driving and ambient conditions as boundary conditions, longitudinal driving dynamics, interior of the bus, refrigeration cycle, climate controller, electrical system as well as the engine cooling and heating cycle. Special emphasis was put on detailed models with modeling of all fundamentally relevant heat transfers and pressure losses for two air conditioning systems based on refrigerant R-134a and R-744. These R-134a and R-744 refrigerant circuits for bus air conditioning are shown in Figure 4.

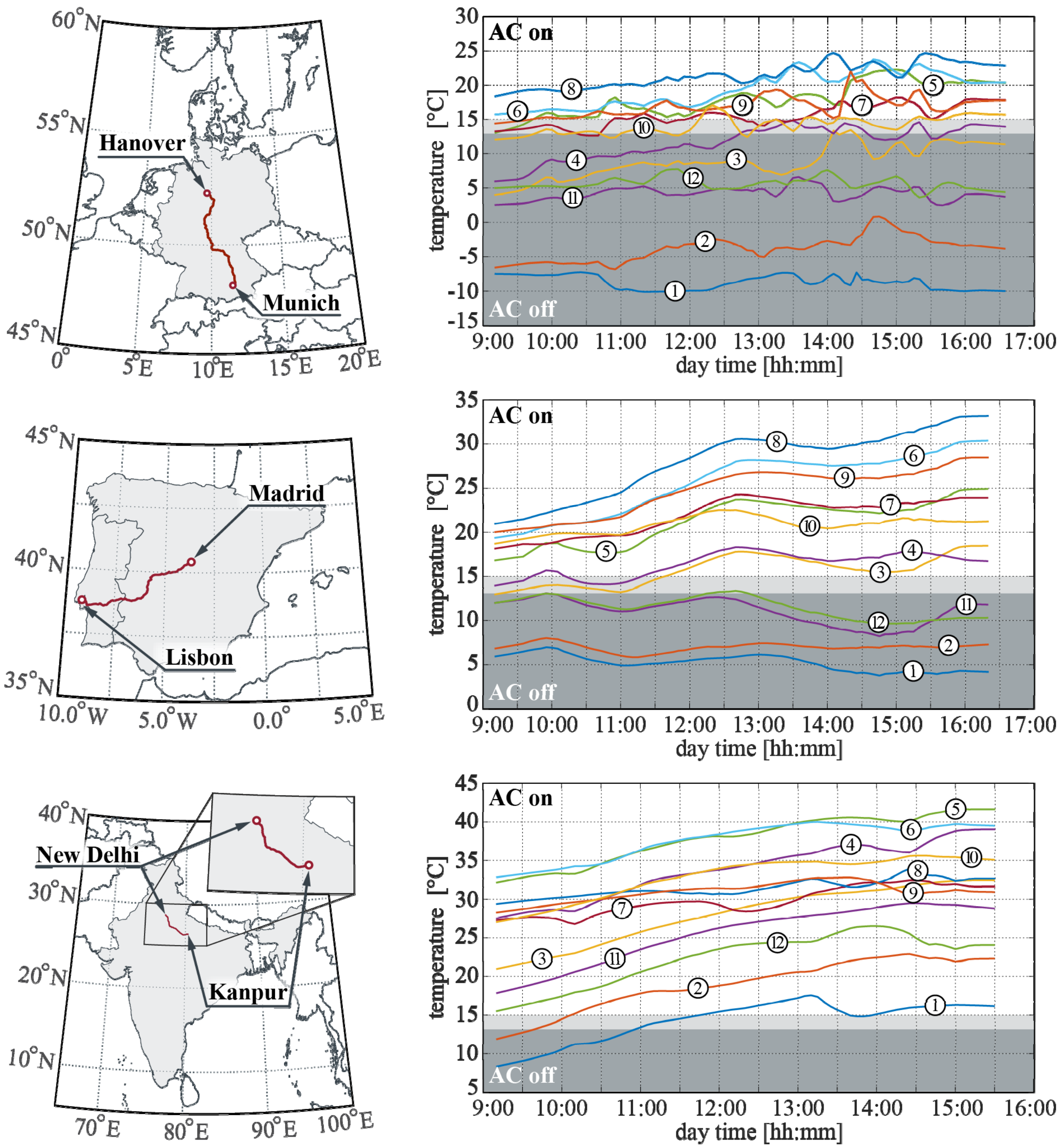

Figure 5: Climatically different, realistically modeled driving route scenarios and corresponding transient ambient temperatures for every 15th of the month in a representative year based on meteorological database.

Driving route scenarios: Hanover to Munich (top), Lisbon to Madrid (middle), New Delhi to Kanpur (bottom); numbered temperature curves: (1) represents 15th of January … (12) represents 15th of December; gray temperature areas: t_{amb}≤13°C refrigerant circuit is off, t_{amb}≥15°C refrigerant circuit is on.

Three climatically different driving scenarios were realistically modelled for the addressed research issues in the realm of bus air conditioning systems. For this, Figure 5 shows the selected driving routes in their respective map sections, which are dynamically driven through with the vehicle model. Based on the geographic coordinates of these three driving scenarios, individual velocity and elevation profiles were calculated to describe the target state for the vehicle model driving simulation. Depending on the defined velocity profile and the geographic position, time-dependent representative ambient conditions were calculated based on a meteorological database (Remund *et al.*, 2013) for the three driving route scenarios (see Kaiser, 2018 for more details). For the presentation of an annual cross-section, transient ambient conditions are calculated for every 15th of the month in a representative year. Thus the ambient conditions include ambient air temperature, ambient air pressure, ambient relative humidity as well as direct and diffuse ambient solar radiation. Figure 5 shows the calculated ambient air temperature curves for the three driving route scenarios as an example. The numbered ambient air temperature curves represent the 15th of each numbered month. The background areas in gray represent the ambient temperature range in which the refrigerant circuit is not active with respect to the climate controller algorithms implemented. In this process, the refrigerant circuit is automatically switched off at $t_{amb} \leq 13°C$ and automatically switched on at $t_{amb} \geq 15°C$.

4. RESULTS

This section presents the numerical results of the vehicle simulation with application of the refrigerant compressor capacity control methods described above depending on the three climatically different driving scenarios. The presented results are shown in relation to the reference system simulations with R-134a-based and R-744-based air conditioning systems without refrigerant compressor capacity control. Figure 6 shows the numerical results for the R-134a system and Figure 7 shows the numerical results for the R-744 system with the individual application of refrigerant compressor capacity controls presented above. Figure 8 shows the numerical results for the R-134a and R-744 system with the novel combination of two-speed pulley gearbox and suction gas interlock. In addition to the relative change in fuel consumption ΔB_S, the figures show the interior air temperature at the driver's workplace t_{Driver} as well as the average passenger compartment air temperature t_{PC}. Furthermore, the figures for the relative change in fuel consumption also include the theoretical limit potential of the possible fuel savings through operation of the air conditioning system (black boxes). This theoretical limit potential is calculated within an addition reference vehicle simulation where the use of the air conditioning system does not consume any energy.

The results in Figure 6 and Figure 7 show continuous reduction of fuel consumption in the driving route scenarios Hanover-Munich and Lisbon-Madrid. In these driving scenarios, the cooling capacity produced by the R-134a and R-744 reference air conditioning exceeds the actual cooling demand. As a result, the applications with CVT, suction gas interlock and two-speed pulley gearbox reduce the refrigerant compressor capacity. In this manner, the cooling capacity is more closely matched to the actual cooling demand with a further advantage being reduced fuel consumption. In contrast, the results of the driving route scenario New Delhi-Kanpur show the opposite behavior in more than half the results due to the extreme ambient conditions. In these driving scenarios, the performance limit of the R-134a and the R-774 air conditioning system is reached. Consequently, speed control by CVT attempts to increase cooling capacity by increasing the refrigerant compressor speed. In the same case, the suction gas interlock remains completely inactive. Comparing the temperatures of the driver's workplace t_{Driver} and passenger compartment t_{PC} to the temperatures of the reference systems, it can be seen that the performance of the R-134a and R-744 air conditioning refrigeration circuit is only slightly improved by increasing the compressor speed. In addition, the coefficient of performance decreases considerably by the disproportionate increase in compressor shaft power compared to cooling capacity, resulting in a significant increase in fuel consumption. For these cases, other solutions for improving the cooling capacity of the R-134a and R-744 air conditioning system have to be developed, e.g. improved air recirculation mode, refrigerant subcooling and the use of an ejector as shown by Kaiser (2018).

The comparison of the suction gas interlock and two-speed pulley gearbox shows that capacity control is more efficiently achieved by the two-speed pulley gearbox. For instance, in the Hanover-Munich driving scenario as well as in the Lisbon-Madrid scenario, the speed-dependent friction power loss in the refrigerant compressor is simultaneously reduced in addition to the capacity adaption due to compressor speed reduction by the two-speed pulley gearbox. As a result, the refrigerant compressor shaft power and fuel consumption are reduced more compared to the suction gas interlock.

The numerical results in Figure 6 and Figure 7 show that continuous capacity control by the CVT is considerably more efficient than individual application of the two-speed pulley gearbox and suction gas interlock. A similar version to the CVT was already used in buses (see Krieg, 1989 and Buhler, 2000). However, that application was discarded due to the high design effort and additional required installation space. In contrast to this is the suction gas interlock, which is already being used in part, and the innovative pulley gearbox, which is novel in bus application and can save space when integrated in the belt pulley of the refrigerant compressor. Figure 8 shows the numerical results by combining the two-speed pulley gearbox and the suction gas interlock. A comparison of the fuel savings between the driving route scenarios Hanover-Munich and Lisbon-Madrid shows the following: In the study with individual application of the two-speed pulley gearbox and suction gas interlock, further improvements have been achieved. In addition, the combination of suction gas interlock and two-speed pulley gearbox approximates the cal-

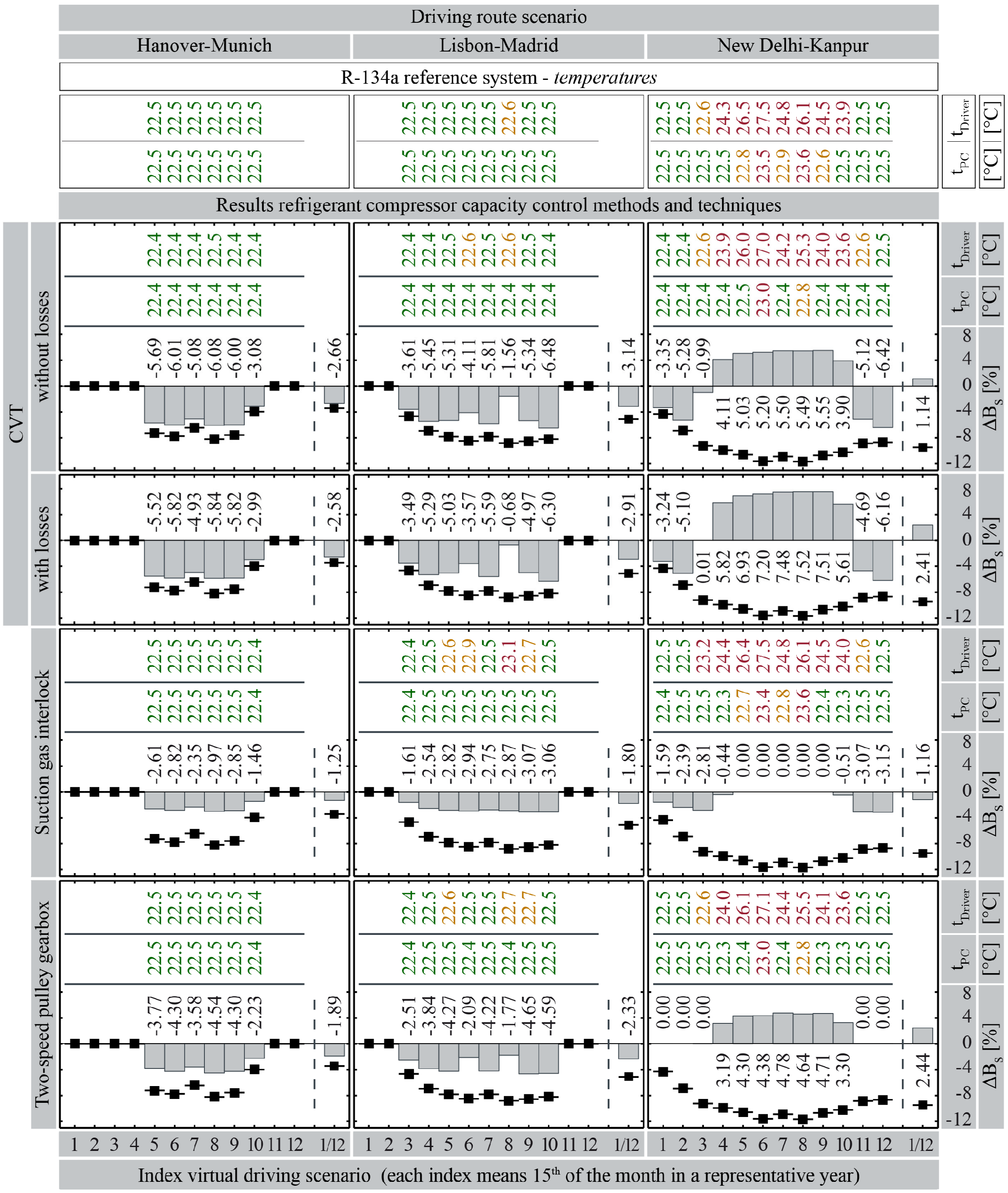

Figure 6: Numerical results of individual application of the ideal and realistic CVT as well as of the suction gas interlock and the two-speed pulley gearbox for an R-134a-based air conditioning system. Shows the fuel consumption of each monthly driving scenario and, in the last column of each diagram, the average annual fuel consumption (1/12). Black boxes present the theoretical limit potential of fuel savings through operation of the air conditioning system.

culated fuel savings with the realistic scenario of a CVT with losses. Compared to the considered ideal CVT, the combined concept of two-speed pulley gearbox and suction gas interlock achieves *78-81%* of the technical limit potential in the R-134a air conditioning system and *88-91%* of the technical limit potential in the R-744 air conditioning system.

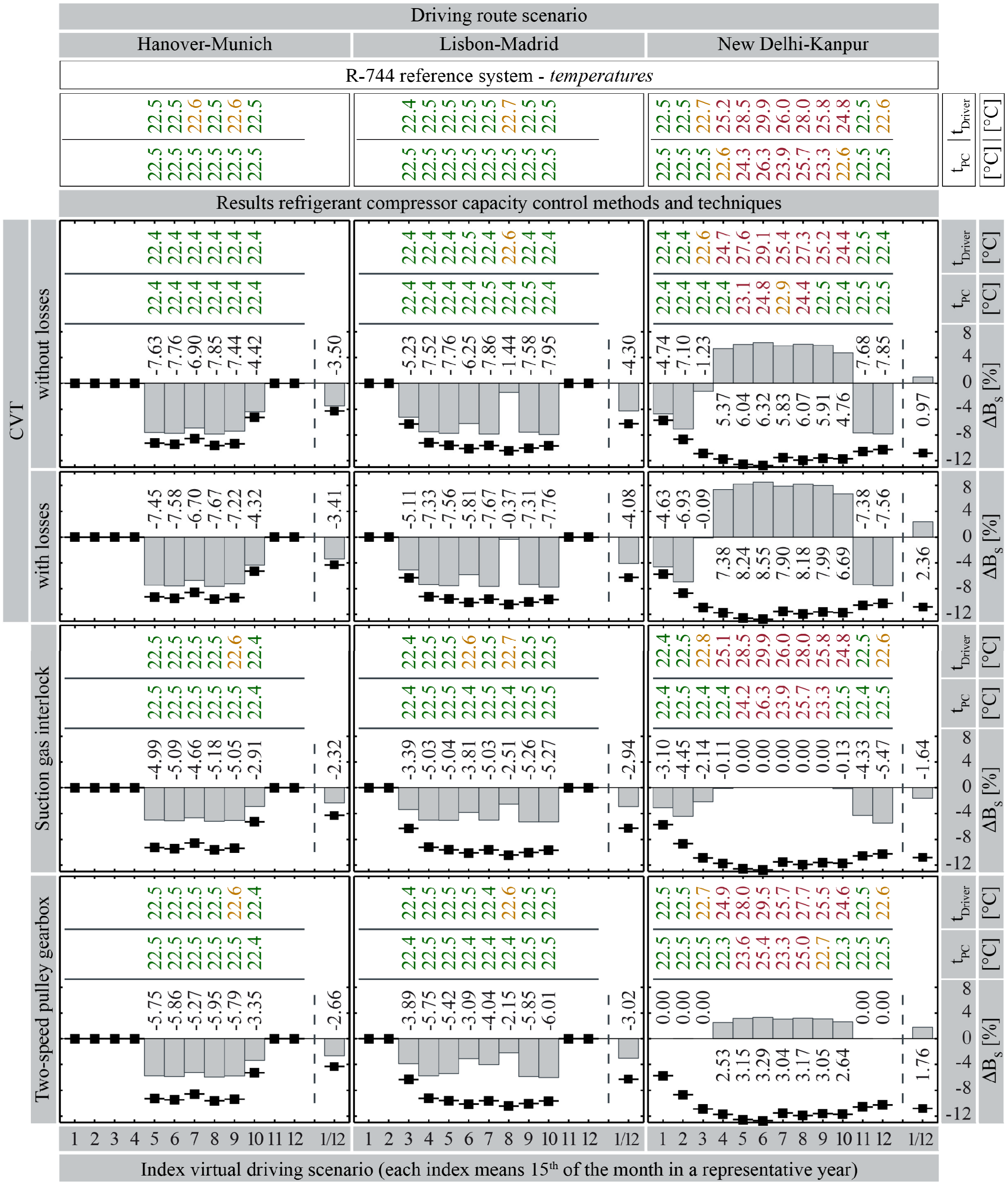

Figure 7: Numerical results of individual application of the ideal and more realistic CVT as well as of the suction gas interlock and the two-speed pulley gearbox for an R-744-based air conditioning system. Shows the fuel consumption of each monthly driving scenario and, in the last column of each diagram, the average annual fuel consumption (1/12). Black boxes present the theoretical limit potential of fuel savings through operation of the air conditioning system.

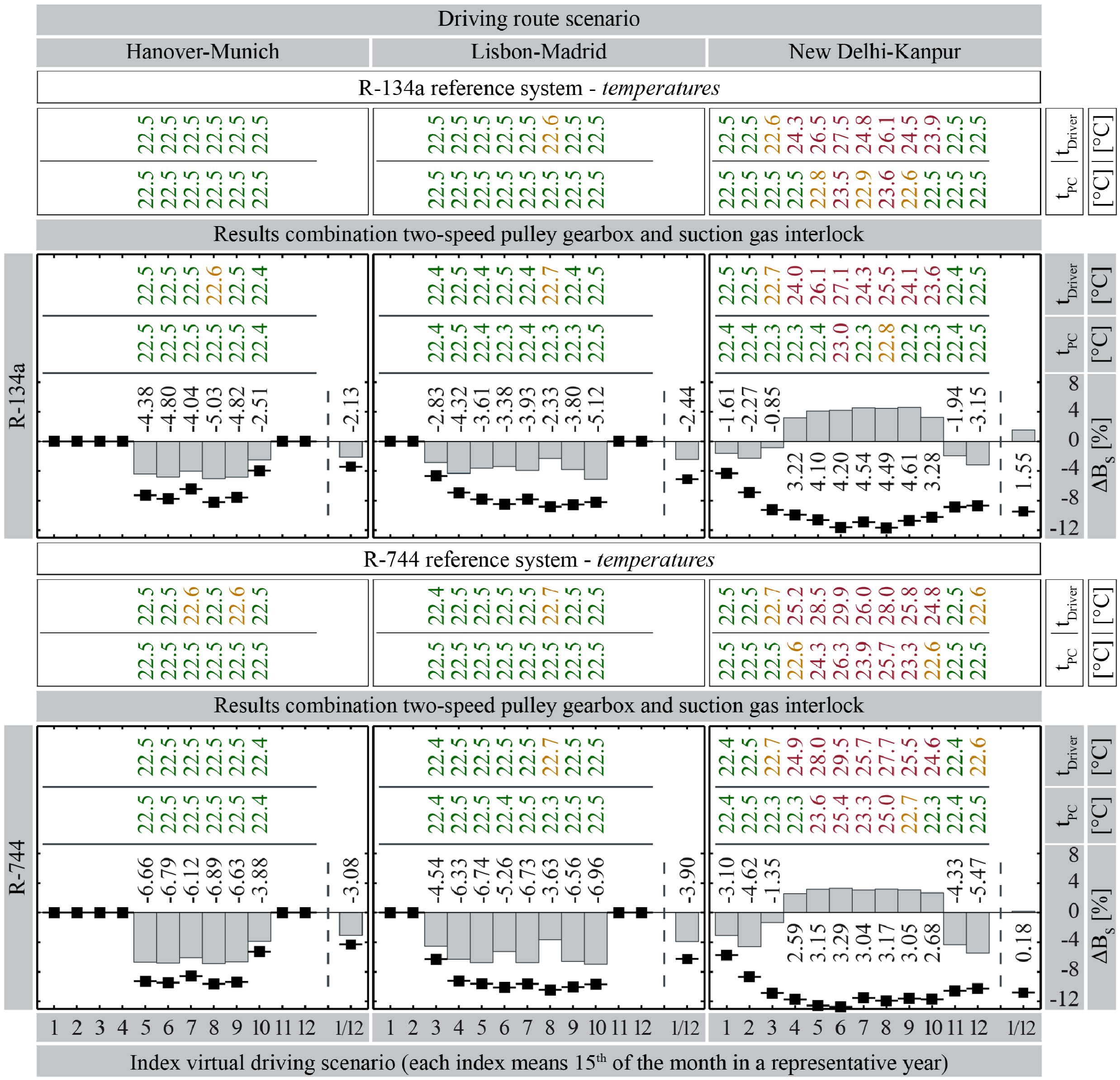

Figure 8: Numerical results of application combination of two-speed pulley gearbox and suction gas interlock for R-134a-based and R-744-based air conditioning system. Shows the fuel consumption of each monthly driving scenario and, in the last column of each diagram, the average annual fuel consumption (1/12). Black boxes present the theoretical limit potential of fuel savings through operation of the air conditioning system.

5. CONCLUSIONS

Our analysis of the ideal lossless CVT shows the possible limit potential in order to improve cooling capacity control and fuel consumption. In the Hanover-Munich and Lisbon-Madrid driving route scenario, the ideal CVT application shows average fuel savings related to the entire vehicle of *4.7-5.3%* for the R-134a air conditioning system and *6.5-7.0%* for the R-744 air conditioning system, based on the driving scenarios with active refrigerant cycles. Compared to the CVT, the more compact and novel combination of two-speed pulley gearbox and suction gas interlock still achieves good fuel savings. With this concept, the technical limit potential is achieved with an average of

78-81% in the R-134a air conditioning system and with an average of *88-90%* in the R-744 air conditioning system. With the extreme ambient conditions of the New Delhi-Kanpur driving route scenario, the increased speed of the refrigerant compressor alone is not sufficient to improve cooling capacity. The installed system capacity of the R-134a and R-744 refrigerant system is not primarily designed for application under the extreme ambient conditions of the New Delhi-Kanpur driving route scenario. As a consequence, these systems are operating at their performance limits. Other solutions have to be developed for this hot climate application in order to increase cooling capacity efficiently. For this purpose, Kaiser (2018) shows and investigates some possible solutions to increase cooling capacity with very minor structural changes.

NOMENCLATURE

η	Efficiency	(-)
i	Transmission ratio	(-)
t_{amb}	Ambient air temperature	(°C)
t_{Set}	Set value of interior air temperature	(°C)
t_{Driver}	Air temperature at driver's workplace	(°C)
t_{PC}	Air temperature of passenger compartment	(°C)
ΔB_S	Relative difference of fuel consumption	(%)

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