

2008

Reducing the Fuel Consumption by Speed Control of the Air Conditioning Compressor

Rico Baumgart

Chemnitz University of Technology

Peter Tenberge

Chemnitz University of Technology

Follow this and additional works at: <http://docs.lib.purdue.edu/icec>

Baumgart, Rico and Tenberge, Peter, "Reducing the Fuel Consumption by Speed Control of the Air Conditioning Compressor" (2008). *International Compressor Engineering Conference*. Paper 1841.
<http://docs.lib.purdue.edu/icec/1841>

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.

Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at <https://engineering.purdue.edu/Herrick/Events/orderlit.html>

Reducing the Fuel Consumption by Speed Control of the Air Conditioning Compressor

Rico BAUMGART^{1*}, Peter TENBERGE²

¹TU Chemnitz, Institute of Engineering Design and Drive Technology,
Chemnitz, Germany
rico.baumgart@mb.tu-chemnitz.de

²TU Chemnitz, Institute of Engineering Design and Drive Technology,
Chemnitz, Germany
peter.tenberge@mb.tu-chemnitz.de

*Indicate Corresponding Author

ABSTRACT

For compliance with future emission regulations, fuel consumption of vehicles must be reduced. This is only possible by an improvement of each car component, for example the air conditioning system.

At the Chemnitz University of Technology, computer programs for detailed simulation of the whole refrigeration cycle and the thermal behavior in the cabin were developed. With these, several modifications concerning the fuel saving potential can be analyzed and evaluated.

Today's air conditioning compressors are driven by a constant transmission ratio by the crankshaft. A gearbox which is integrated in the compressor pulley offers a possibility for reducing fuel consumption. Depending on the engine speed and the refrigerating capacity, the compressor could be run at an operating point with lower driving power. This speed control was analyzed with the simulation programs mentioned above. The results are shown in the article.

1. INTRODUCTION

In order to meet ever-tightening emission regulations, fuel consumption and CO₂ discharge of vehicles have to be reduced drastically. Due to the rising environmental awareness the ecological aspects become increasingly important purchase criteria. Therefore, fuel saving gains particular importance in the development of automobiles. The driving comfort and safety, however, must not be affected. These various demands lead to a conflict of interests.

Today, most cars are equipped with an air conditioning system. The higher comfort affects fuel consumption. Different modifications, for example an improvement of

- the cabin, e.g. by thermo glass and
- the compressor, e.g. by a better valve design

offer potential for fuel saving.

Up to now, the air conditioning compressors are driven by a constant transmission ratio by the crankshaft. Modern compressors make it possible to adjust the displacement to the refrigerating capacity demand. By decreasing the displacement the driving power is declined. The compressor efficiency is impaired, though. Moreover, an increasing compressor speed leads to higher friction losses. It is more efficient to run the compressor at lower speed and higher displacement to achieve a particular refrigerating capacity. As a result, the driving power can be reduced. One possible solution is the integration of a gearbox in the compressor pulley.

The automotive industry always critically examines such modifications by comparing the achievable improvements to the investment costs. To enable an exact analyzing and objective evaluation of these optimizations, detailed simulation programs for

- the compressor,
- the heat exchangers and
- the cabin

were investigated at the Chemnitz University of Technology. With the aid of these, the whole refrigeration cycle and the thermal behavior in the cabin can be simulated.

The models will be described in the following.

2. COMPRESSOR MODEL

The model is based on a conventional swash plate axial piston compressor which is controlled externally. The dynamic behavior of the discharge and suction valves has a high impact on the compressor properties. That is why the mathematical describing of the valves was strongly focused during the modeling process. The actual valves were substituted by theoretical systems (Figure 1a, 1b). As Figure 1b shows, different forces act on the valve plate, such as

- spring force F_s ,
- inertia force F_m ,
- damping force F_d and
- pressure force F_p .

These forces depend on the time and the operating point.

With regard to the discharge valve it has to be considered that the lamella will attach continuously on the limiter while the valve lift rises. During this process the spring stiffness changes. In the case of the inlet valve an edge in the cylinder bore acts as a valve lift limiter (Figure 1a). When the lamella contacts this edge, the spring stiffness changes as well. For describing the dynamic valve behavior, the spring stiffness is often accepted as a constant value. This assumption can only be applied if there is no limiter or the contour of the limiter is identical with the bending line. Therefore, an algorithm which allows the determination of the spring stiffness depending on the valve lift was developed. It works perfectly without expensive and time-consuming FEM-models. However, a good agreement with FEM-results could be achieved (Figure 1c).

Individual approaches were chosen for inertia-, damping- and pressure forces.

From the balance of the valve forces

$$F_d + F_s + F_m - F_p = 0 \quad (1)$$

explicit differential equations second order for describing the dynamic behavior of the discharge and the suction valve can be derived.

Furthermore, it is possible to develop mathematical relations for the mass flow on the valves and the temperature variation of the refrigerant gas in the cylinder. The real gas equations, the density and temperature of the gas allow the calculation of the cylinder pressure.

Finally, a system of differential equations can be formulated, that can be solved numerical.

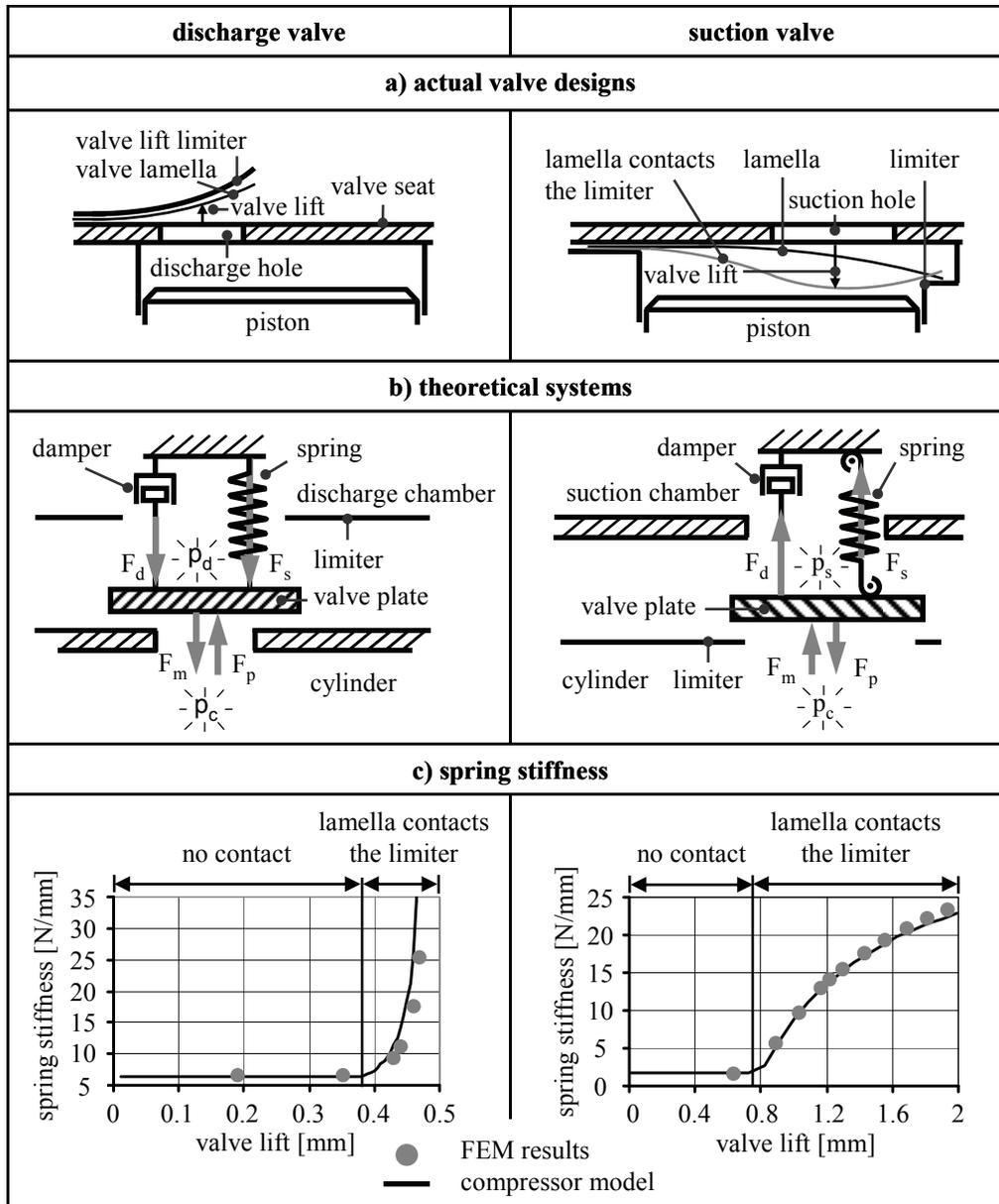


Figure 1: Valve models

Figure 2 shows the simulation results for an exemplary operating point. At the rotation angle $\varphi = 290^\circ$ the valves are closed (Figure 2d). The piston is in the compression phase and the cylinder pressure increases (Figure 2e). The latter exceeds the pressure in the discharge chamber at $\varphi = 310^\circ$. As a result, the outlet valve opens and the gas is emitted. The opening movement is retarded by damping effects and inertia. Figure 2d clearly illustrates that the discharge valve shows no lift motion in the range from $\varphi = 316^\circ$ to $\varphi = 330^\circ$ because the lamella fits completely on the limiter. The valve closes at $\varphi = 330^\circ$. It is remarkable that the gas mass in the cylinder increases after the piston has reached the top dead center (Figure 2b). This is caused by the retarding valve movement which prevents the complete valve closing at the beginning of the expansion phase. The so called back flowing effects occur. At only $\varphi = 365^\circ$ the discharge valve is completely closed. From then on, the remaining gas in the dead volume needs to be expanded. At $\varphi = 408^\circ$ the suction valve opens and the refrigerant gas flows into the cylinder. The inlet valve shows the typical high frequency oscillation of valves without a pronounced limiter. At $\varphi = 550^\circ$ the inflowing process is finished. Back flow effects also occur between $\varphi = 550^\circ$ and $\varphi = 570^\circ$ (Figure 2c). The compression phase begins at $\varphi = 570^\circ$.

The mass and temperature of the gas in the cylinder have to be identical at the beginning and the end of the process at a constant operating point.

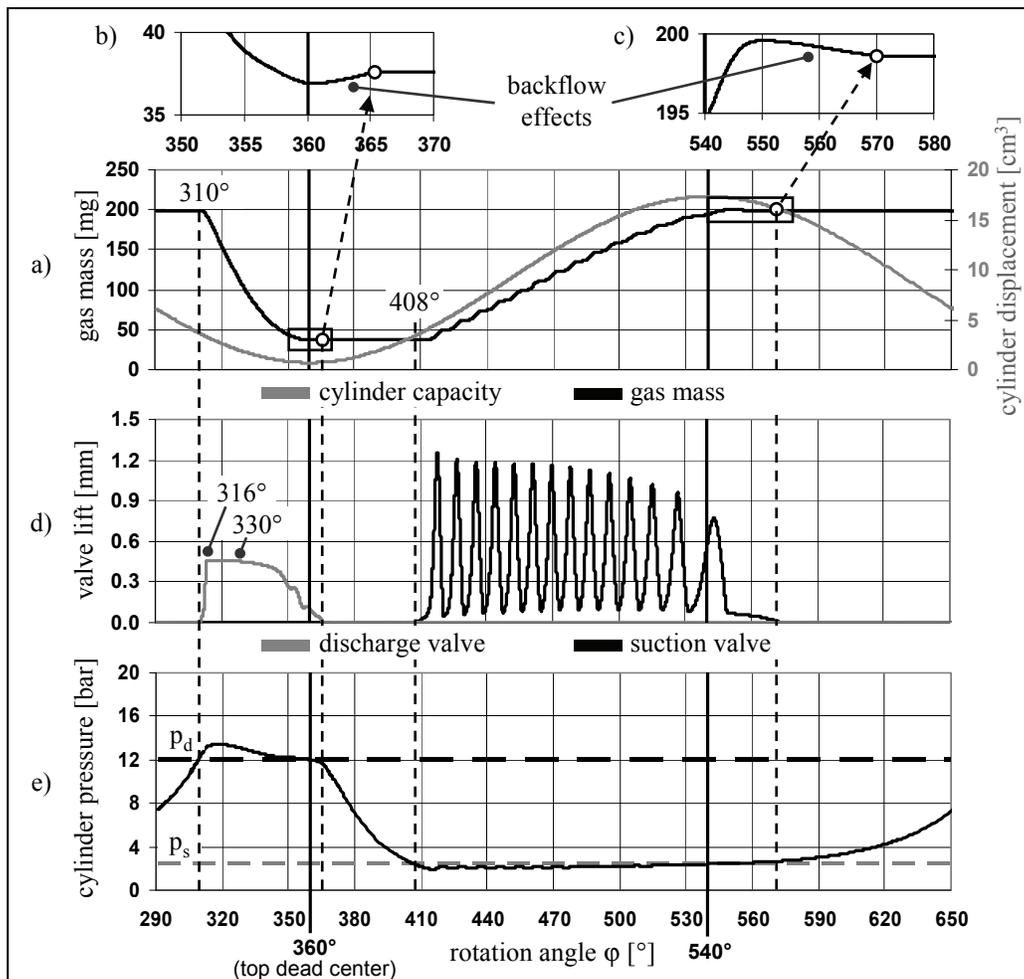


Figure 2: Procedures in the compressor

The model gives a detailed insight in compressor procedures. In this way the compressor can be modified specifically, for example to achieve a higher efficiency. Furthermore, it is possible to exactly determine parameters such as the delivered mass flow and the required input power for each operating point.

3. HEAT EXCHANGER MODELS

The condenser is integrated behind the front bumper and the radiator grill. Thus, on the condenser plate an inhomogeneous flow field occurs during the drive. These flow effects are simulated at the Chemnitz University of Technology by CFD programs.

The condenser and evaporator are subdivided in several elements (Figure 3). The simulated local air flow speed is assigned to each element. In a stationary state, the heat flows on the air side and the refrigerant side are identical. In order to determine the state at the heat exchanger outlet, the iterative calculation of the output state needs to be done sequentially for each element. This fact requires that the input state of the air and the refrigerant is known. A general differentiation between gas-, steam- and fluid flows is necessary. In terms of the evaporator the outlet state of the air has to be computed.

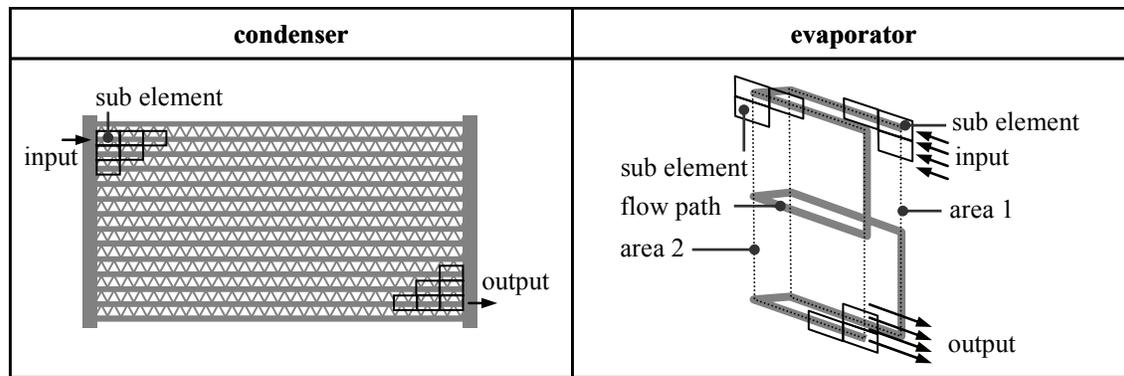


Figure 3: Sub elements of the heat exchangers

4. CABIN MODEL

For the mathematical description of the thermal processes in the cabin, the vehicle geometry is represented by flat polygons (Figure 4).

Generally, all components of the cabin can exchange heat by electromagnetic waves. In the case of two randomly positioned surfaces, a part of emitted heat radiation may not reach the receiver surface. This is expressed by so-called radiation numbers. An analytical calculation of these numbers would be either laborious or impossible. Additionally, it is complicated by elements between the corresponding surfaces, which interrupt the radiation exchange. For these reasons the radiation numbers are calculated numerically in the model.

The solar radiation could heavily influence the thermal behavior in the cabin. Therefore, it is considered depending on the date, the time and the global position. Regarding the solar radiation within the cabin, the shadowing effects are very significant.

Convection is another important heat transfer mechanism, which can be included by commonly accepted relationships (VDI Wärmeatlas, 1994).

At warm surfaces a buoyancy-driven airflow develops, which reaches upper air zones. On cold surfaces an air mass flows downwards lower air zones in the same way. Due to this phenomenon the temperature in the head zone is always higher than in the foot zone after parking in the sun. Because of the non-uniform temperature distribution, the cabin has to be subdivided into several air zones (Figure 4). These zones correspond by a mass exchange, which is caused by the fan and the thermal flow effects mentioned above.

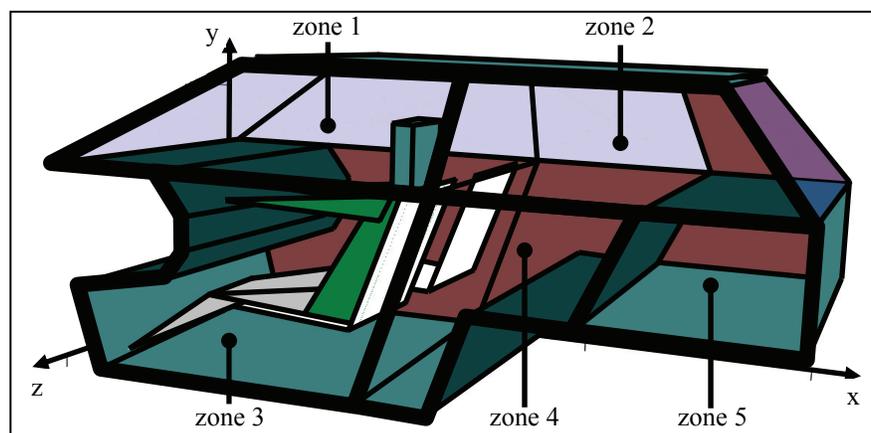


Figure 4: Cabin model

Convection and radiation occur at the driver's surface, too. Furthermore, the human body is able to emit heat by transpiration and breathing. This leads to increasing air humidity in the cabin. That is why the cabin-program includes an algorithm for the driver.

5. RESULTS

The combination of the described models for the compressor and the heat exchangers allows the simulation of the procedures in the whole refrigeration cycle.

Figure 5 shows characteristic diagrams for the air conditioning system for an exemplary vehicle speed. In Figure 5a an operating point for a refrigerating capacity of 2.6 kW is drawn in. The compressor speed is 3500 rpm and the relative displacement is 20 %. At this point the compressor requires an input power of 1.15 kW (Figure 5b). If the compressor speed is reduced to 1200 rpm, for example by a gearbox, the same refrigerating capacity will be achieved with a relative displacement of 50 %. Consequently, the input power could be decreased to 0.75 kW, which is the same as a reduction of 35 %.

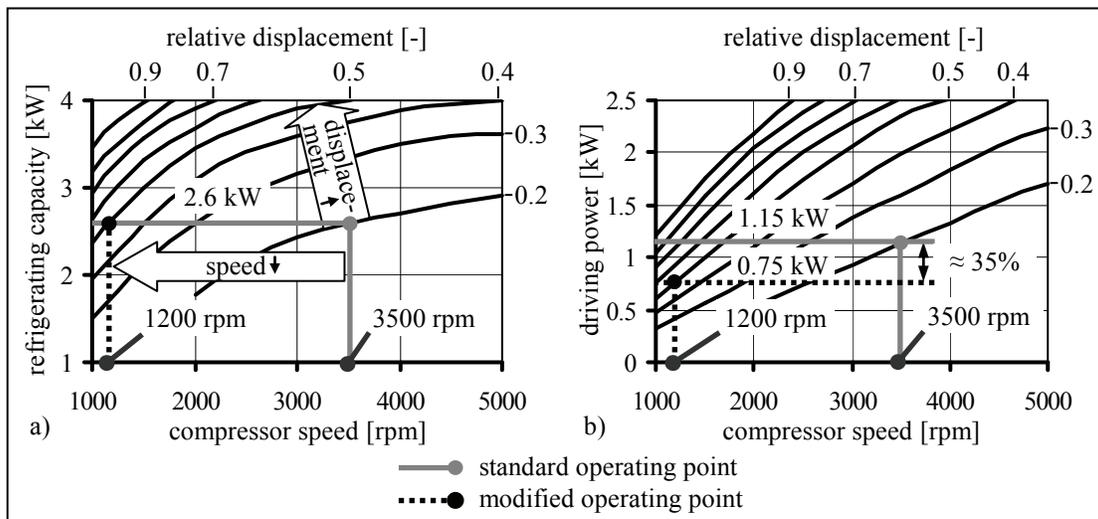


Figure 5: Exemplary compressor operating point

For an urban and an extra-urban driving cycle, refrigerating capacities were calculated which are necessary to achieve a constant comfort temperature of 21.5 °C in the cabin head zone (Figure 6). As a basic environmental condition, a summer day in Chemnitz (Germany) was assumed. Further on, it was taken as a basis that, before the cycles were started, the car had been parked for one hour in the sun. Another assumption was that the same amount of air mass flew into the zones 1 and 3 during the drive (see Figure 4).

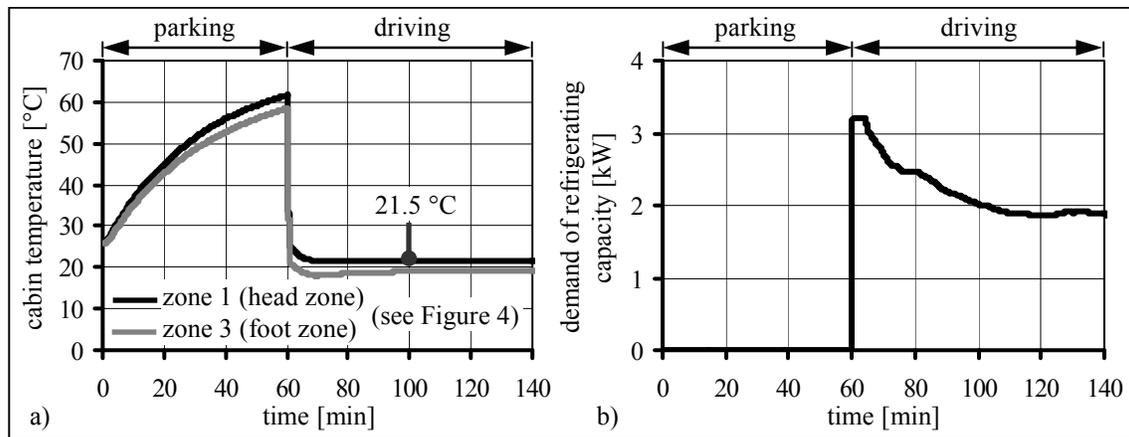


Figure 6: Cabin temperature and demand of refrigerating capacity in the urban driving cycle

It would always be practicable to run the compressor independent of the engine speed, at an optimal operating point with a minimum driving power by using a continuously variable transmission (CVT).

In order to determine the maximum energy saving potential by speed control of the compressor, a perfect CVT was taken as a theoretical basis. A minimum compressor speed of 500 rpm was set to avoid poor lubrication.

CVTs have been developed for the speed control of ancillary aggregates. With regard to engineering and financial effort, two-speed planetary gearboxes with the gear ratios $i = 1$ and $i > 1$ are more appropriate for this application. At a gear ratio of $i = 1$ the compressor is driven at conventional speed and at $i > 1$ the compressor speed is reduced. The energy saving, that can be achieved in the driving cycles, was calculated for several two-speed gearboxes (Figure 7).

As was to be expected, with the theoretical CVT the most effective energy saving can be achieved. Among all two-speed gearboxes, the one with the gear ratios $i = 1$ and $i = 2.3$ shows the highest saving potential.

Normally, the standard compressor speed in the extra-urban driving cycle is higher than in the urban cycle. Thus, the saving potential on the highway is always above the saving potential in the city.

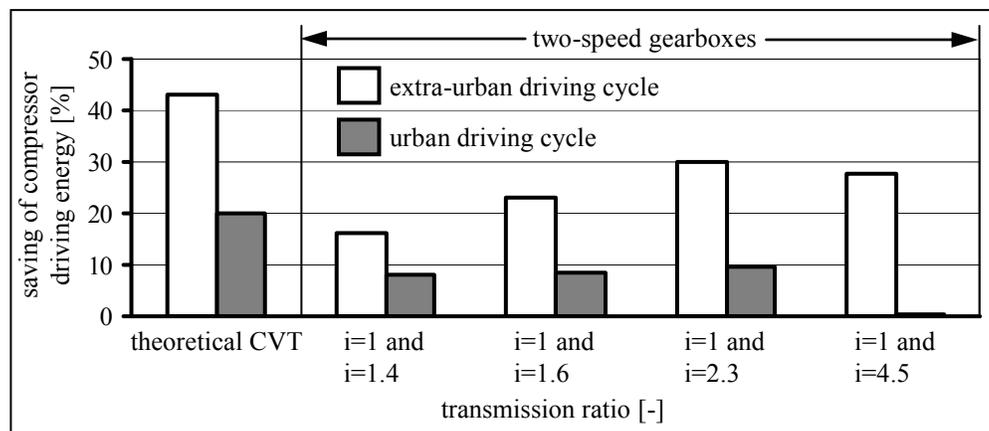


Figure 7: Saving of compressor driving energy

6. CONCLUSION

The presented simulation models showed the energy saving potentials achieved by speed control of the compressor.

The air conditioning is not only used in the summer. In cold seasons, it serves as a dehumidifier of inflowing air and therefore prevents windows from fogging. For that reason, the analysis must not be limited to summery environmental conditions. Future investigations have the aim to develop an optimal control strategy and suitable gear ratios for achieving a minimization of the annual fuel consumption. Additional modifications, for example a better valve design, will also be analyzed and compared to the speed control.

REFERENCES

VDI Wärmeatlas, 1994, *Berechnungsblätter für den Wärmeübergang*, VDI Verlag, Düsseldorf: p. F-G.

ACKNOWLEDGEMENT

The authors are very grateful to

Petra Baumgart,
Diana Lohse,
Rene Buchmann and
Anke Großmann

for supporting this work.