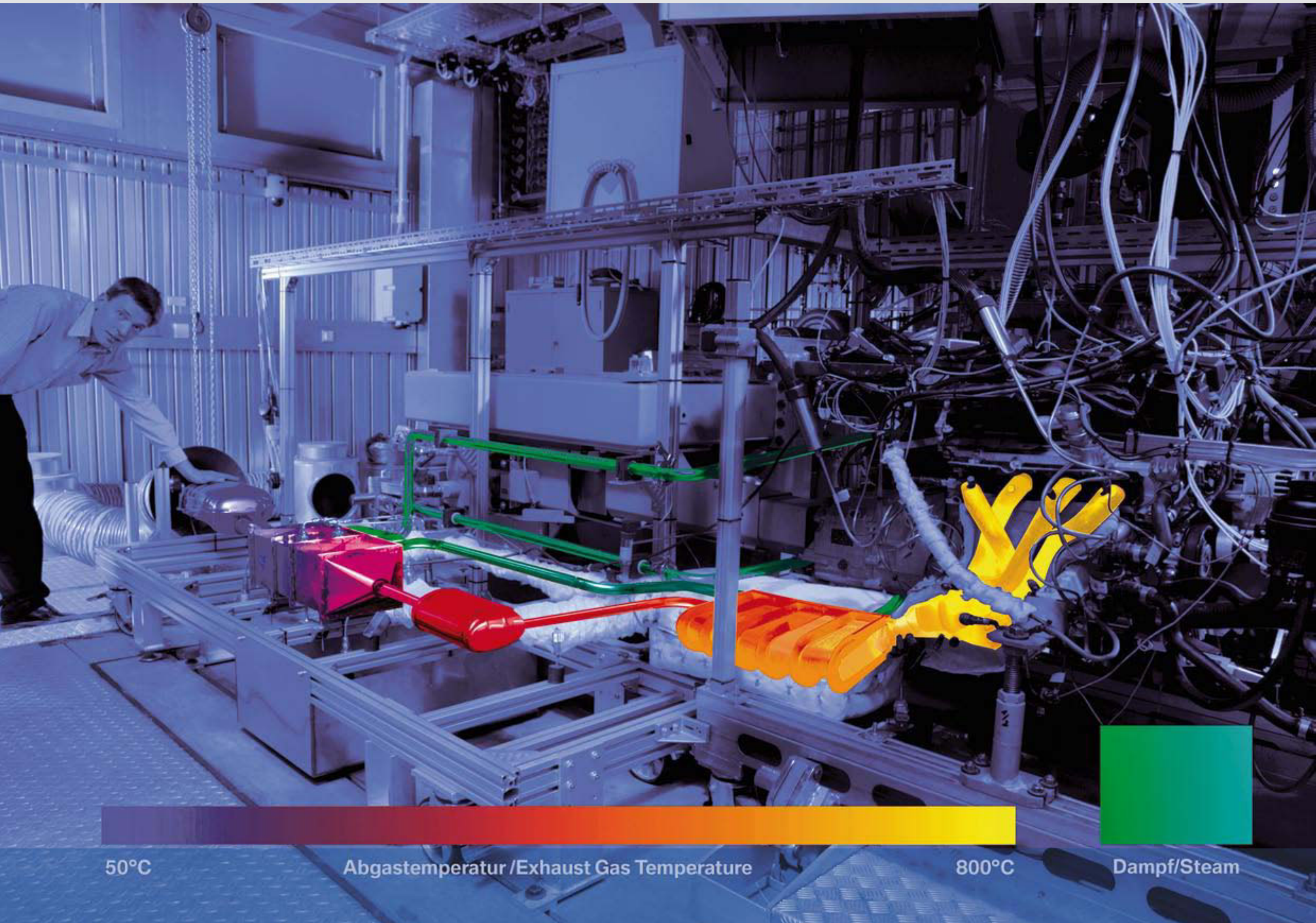

The Turbosteamer: A System Introducing the Principle of Cogeneration in Automotive Applications

BMW's turbosteamer is the first system to use the potential of waste heat in order to enhance the efficiency of the combustion process in the automobile. Applying the principle of cogeneration, fuel consumption can be reduced in a four-cylinder spark ignition engine by more than 10 per cent or, conversely, power and performance may be increased accordingly under relevant stationary operating conditions.

1 Introduction

The progress achieved in the development of the combustion engine in the last 120 years is truly gigantic. While the Benz Motor Carriage in 1886 developed maximum output of 0.55 kW at 400 rpm, engine output of 350 kW at 8,000 rpm is nothing special today in a modern sports car. But whenever the driver wishes to use this 350 kW by pressing down the accelerator pedal, the combustion engine requires a supply of fuel three times as great as the power actually generated (that is more than 1,000 kW!). This is because the combustion engine, with its maximum efficiency of approximately 40 per cent, converts only about one-third of the energy it receives by way of fuel into mechanical energy, even under favourable operating conditions. Two-thirds of the energy in the fuel, in turn, is converted into heat within the combustion engine. This energy referred to as „heat loss“ is discharged „into the environment“, as it were, via the engine‘s cooling system and the car‘s exhaust, without being used in the slightest for any practical purpose.

Considering the limited supply of fossil energy reserves, it is essential in our modern day and age to capitalise on the fuel energy used for powering the automobile. Taking the conventional approach, the focus of research and development activities in enhancing the efficiency of the combustion engine is to optimise the process of fuel/air mixture preparation and the combustion process. A relatively new perspective, by comparison, is to take a closer look at the „periphery“ around the engine with all its units for converting energy, seeking in this way to reduce fuel consumption through the general improvement of energy management.

This paper presents a system serving to recuperate heat losses and convert such heat into mechanical energy. It is shown that this technology is able on a four-cylinder spark ignition engine to reduce fuel consumption in the range of 10-15 per cent or, corresponding, to increase power output at relevant, stationary operating points. Cogeneration therefore offers a significant potential in solving the conflict of interests between fuel economy, on the one hand, and the performance of a vehicle, on the other. It is therefore an important element and, indeed, the next logical step in the “EfficientDynamics” approach taken by BMW [1].

2 Efficiency of the Combustion Engine

The constant-volume cycle is the ideal approach in describing the thermodynamic processes to be observed in all piston engines with periodical combustion and generation of mechanical energy. As shown by **Figure 1** in the T,s diagram, the constant-volume cycle is based on a cycle process made up of reversible adiabatic

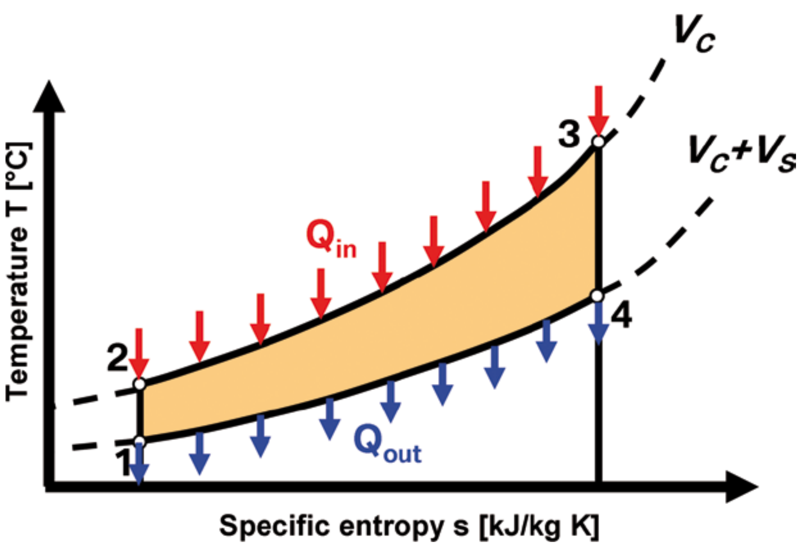

Figure 1: Constant-volume process in T,s diagram

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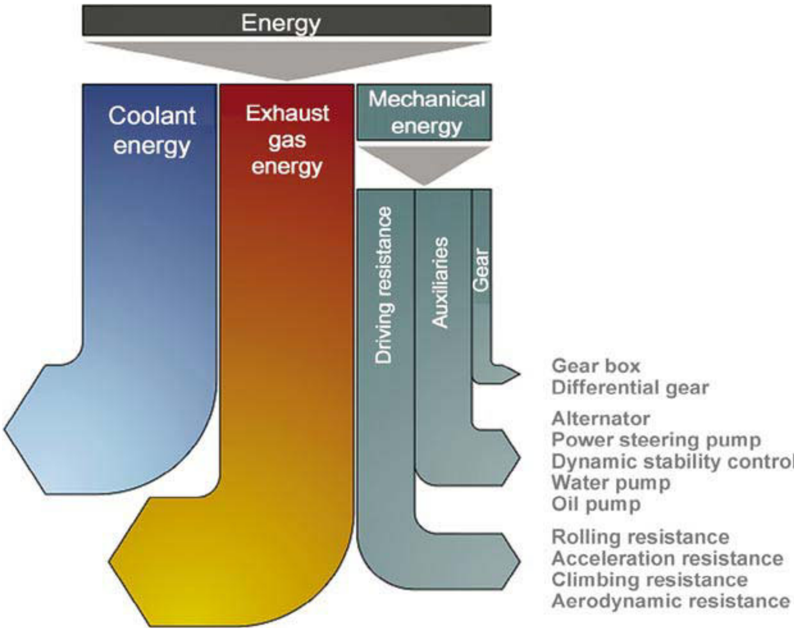

Figure 2: Energy conversion in a vehicle

(isentropic) compression (1→2), the isochoric supply of heat (2→3), isentropic expansion (3→4), and the isochoric transfer of heat (4→1).

According to [2], this cycle process shows an efficiency of

$$\eta_{ideal} = \frac{W}{Q_{in}} = \frac{(Q_{in} - Q_{out})}{Q_{in}} = 1 - \frac{Q_{out}}{Q_{in}} = 1 - \frac{T_l}{T_u} = 1 - \varepsilon^{(1-\kappa)} \qquad \text{Eq. (1)}$$

In this context Q_{in} defines the heat fed into the process (2→3), Q_{out} the heat transfer (4→1), W the work performed by the system (3→4), T_l, T_u the lower and, respectively, upper temperature level of the cycle process, κ the isentropic exponent of the ideal gas. ε, finally, stands for the compression ratio resulting as follows from compression volume V_C and stroke volume V_S:

$$\varepsilon = \frac{(V_C + V_S)}{V_C} \qquad \text{Eq. (2)}$$

In its (maximum) value, the efficiency of the constant-volume cycle thus depends exclusively on the compression ratio ε and the isentropic exponent κ. Eq. (1) shows that due to $Q_{out} > 0$, the power unit can never reach a level of efficiency of 100 per cent, not even under ideal conditions.

It goes without saying that in the real process of operating a (reciprocating-piston) combustion engine the heat loss is far greater and the efficiency far lower than in the ideal constant-volume process. Again following [2], therefore, the effective efficiency of a combustion engine may quite generally be defined as follows:

$$\eta_{effective} = \frac{W}{E_{Fuel}} = \eta_{ideal} - \Delta\eta_{rcharge} - \Delta\eta_{icomb} - \Delta\eta_{rcomb} - \Delta\eta_{lwall} - \Delta\eta_{leakage} - \Delta\eta_{lcharge} \qquad \text{Eq. (3)}$$

In this context, W defines the mechanical energy delivered by the power unit within a specific period and E_{fuel} the fuel energy used for this purpose. The individual efficiency terms in the actual, real-life process describe the influence of the current state of charge, the conversion losses resulting from incomplete combustion, combustion losses as a result of the real combustion process, thermal losses due to heat transfer on the walls of the combustion chamber, leakage losses resulting from blow-by effects, and charge cycle losses.

As shown by **Figure 2**, only a fraction of the chemical energy contained in the fuel is actually converted into mechanical energy in a motor vehicle. Making a very optimistic assumption, one may state that in the real operation of a motor vehicle not more than one-third of the energy content in a given amount of fuel is actually converted into mechanical energy under realistic operating conditions, with one-third being discharged into the coolant and one-third leaving the vehicle through the exhaust system.

If it were possible to recuperate this lost thermal energy at least in part, converting it into mechanical energy, the overall degree of system efficiency could be increased accordingly. Assuming the efficiency $\eta_{coolant}$ for the conversion of coolant heat $Q_{coolant}$ into mechanical energy $W_{coolant}$ and the efficiency $\eta_{exhaust\ gas}$ for the conversion of thermal exhaust energy $Q_{exhaust\ gas}$ into mechanical energy $W_{exhaust\ gas}$, we obtain the following equations:

$$W_{coolant} = \eta_{coolant} \cdot Q_{coolant} \qquad \text{Eq. (4)}$$

$$W_{exhaust\ gas} = \eta_{exhaust\ gas} \cdot Q_{exhaust\ gas} \qquad \text{Eq. (5)}$$

This leads to the following equation describing the effective degree of efficiency $\eta'_{effective}$ of a power unit with a cogeneration system operating downstream:

$$\eta'_{effective} = \frac{W}{E_{fuel}} = \eta_{effective} + \frac{(\eta_{coolant} \cdot Q_{coolant} + \eta_{exhaust\ gas} \cdot Q_{exhaust\ gas})}{E_{fuel}} \qquad \text{Eq. (6)}$$

Applying

$$\Delta\eta_{coolant} = \eta_{coolant} \cdot \frac{Q_{coolant}}{E_{fuel}} \qquad \text{Eq. (7)}$$

$$\Delta\eta_{exhaust\ gas} = \eta_{exhaust\ gas} \cdot \frac{Q_{exhaust\ gas}}{E_{fuel}} \qquad \text{Eq. (8)}$$

the following results from Eq. (6):

$$\eta'_{effective} = \frac{W}{E_{fuel}} = \eta_{effective} + \Delta\eta_{coolant} + \Delta\eta_{exhaust\ gas} \qquad \text{Eq. (9)}$$

The following chapter examines the magnitude of $\Delta\eta_{coolant}$ and $\Delta\eta_{exhaust\ gas}$.

3 Cogeneration

The system of cogeneration examined here is based on the Clausius-Rankine steam process [3] presented in the T,s diagram of **Figure 3**. The underlying ideal process involves the isentropic compression of a liquid medium (1→2), isobaric heating, evaporation and superheating of the medium (2→3) by supplying the thermal loss energy Q_{in} of the power unit, the isentropic expansion of the vaporous medium (3→4) in an expansion machine with a supply of mechanical energy, and isobaric condensation of the medium back into the liquid phase (4→1), together with the dissipation of heat Q_{out} into the environment.

Assuming an ideal process, the efficiency of the Clausius-Rankine process is as follows:

$$\eta_{CR,ideal} = \frac{(W_{expansion} - W_{pump})}{Q_{in}} \qquad \text{Eq. (10)}$$

whereby $W_{expansion}$ is the mechanical expansion work of a machine operating free of losses generated within a specific period (3→4), W_{pump} is the work used as applied by a pump operating free of losses (1→2), Q_{in} is the heat transferred completely to the process (2→3), as recuperated from the loss energy of the engine unit.

The values for the expansion and pumping work shown in Eq. (10) can be calculated from the differences in the enthalpic values entered in Figure 3. Applying Eq. (10), this results in the following equation:

$$\eta_{CR,ideal} = \frac{((h_3 - h_4) - (h_2 - h_1))}{(h_3 - h_2)} = 1 - \frac{(h_4 - h_1)}{(h_3 - h_2)} = 1 - \frac{Q_{out}}{Q_{in}} \qquad \text{Eq. (11)}$$

Under real process conditions, it is important to consider that the existing heat losses cannot be recuperated in full – some will be lost, for example, in the heat transfer process on account of radiation or convection. There will also be losses in the system due to throttle effects, flow and mechanical friction between the various components. And ultimately expansion of the medium in its steam phase inevitably also involves losses.

Taking all these losses into account, we obtain the effective degree of efficiency in a real-life Clausius-Rankine process:

$$\eta_{CR,effective} = \eta_{CR,ideal} \cdot \eta_{heat\ transfer} \cdot \eta_{flow,friction} \cdot \eta_{expansion} \qquad \text{Eq. (12)}$$

The similarity of Eq. (11) and Eq. (1) in the ideal process in terms of the ratio between heat input and heat output is quite amazing. Accordingly, we see also in the Clausius-Rankine process that the best efficiency is achieved when (particularly in the wet steam phase) heat input is kept to the highest possible and heat output to the lowest possible temperature level.

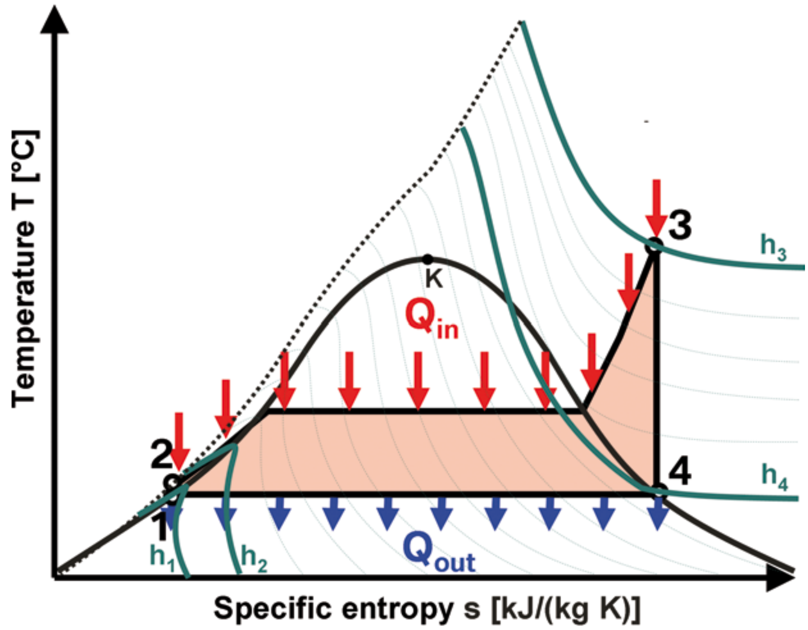

Figure 3: Clausius-Rankine steam process in T,s diagram

This point is significant since, when applying the cogeneration principle in a combustion engine, we have to consider that the coolant and the exhaust gases, that is the two media removing heat from the system, show a very different temperature level: While coolant temperature is generally between 100 and 115 °C, the temperature of the exhaust gases may well be up to 900 °C. Given these significant differences in temperature, it is appropriate, taking a „maximum approach in the recuperation of heat loss" to consider a dual-circuit system – a low-temperature cycle for recuperating heat from the coolant and a high-temperature cycle for recuperating thermal exhaust energy.

To assess the potentials available with such a concept, we will focus, by way of example, on the high-temperature cycle in the following. Since in this case the input of heat comes from hot exhaust gases, we may proceed from the following realistic temperature levels in process management: evaporation temperature = 200 °C, superheating temperature = 400 °C, condensation temperature = 100 °C. Under the assumption that the medium used in the cycle process is water, we then, applying the data well-known from steam vapour tables [4], obtain the following degree of efficiency in accordance with Eq. (11):

$$\eta_{CR,ideal} = 1 - \frac{(2363 - 417)\,\frac{kJ}{kg}}{(3255 - 419)\,\frac{kJ}{kg}} = 0{,}22 \qquad \text{Eq. (13)}$$

We may also proceed from the following realistic figures for losses in an actual system under real-life conditions: $\eta_{heat\ transfer} = 0.8$, $\eta_{flow,friction} = 0.9$ and $\eta_{expansion} = 0.7$. Applying Eq. (12), this shows the following effective degree of efficiency in the high-temperature cycle:

$$\eta_{CR,effective} = 0{,}22 \cdot 0{,}8 \cdot 0{,}9 \cdot 0{,}7 = 0{,}11 \qquad \text{Eq. (14)}$$

By definition, this factor is identical to the efficiency $\eta_{exhaust\ gas}$ shown in Eq. (5). Assuming a ratio of 1/3 between $Q_{exhaust\ gas}/E_{fuel}$, the increase of efficiency in the combustion engine on account of the recuperation of thermal exhaust energy is as follows, in accordance with Eq. (8):

$$\Delta\eta_{exhaust\ gas} = 0{,}11 \cdot \frac{1}{3} = 0{,}037 \qquad \text{Eq. (15)}$$

That is 3.7 per cent.

Without going into further detail, it is important to note in this context that, based on similar considerations, the effective degree of efficiency in a low-temperature cycle serving to recuperate coolant heat, using ethanol as the medium, is $\eta_{CR,effective} = \eta_{coolant} = 0.06$.

Again assuming a ratio of 1/3 for $Q_{coolant}/E_{fuel}$, Eq. (7) shows the following increase in the degree of efficiency in the combustion engine on account of the recuperation of coolant heat:

$$\Delta\eta_{coolant} = 0{,}06 \cdot \frac{1}{3} = 0{,}020 \qquad \text{Eq. (16)}$$

That is 2.0 per cent.

Proceeding from these considerations and in accordance with Eq. (9), one may therefore assume that the degree of efficiency of a combustion engine will increase in absolute terms by 5-6 per cent when recuperating thermal energy in both the coolant and the exhaust gas, that is using both sources together. Applying a result of only 5 per cent, this means – given a maximum efficiency in the combustion engine at its best operating point of approximately 40 per cent – a relative improvement of the degree of efficiency of at least 12 per cent within the operating range of the combustion engine.

4 The Turbosteamer

4.1 Development of the System

A large number of fundamental questions has to be addressed in determining the experimental set-up used to verify the cogeneration concept. Without going into the details of the many considerations involved, where the focus was not only on technical features, but also on the effort involved and deadlines, only the most important decisions are presented in the following:

1. Realisation of a dual-circuit system serving to use the full potential of thermal recuperation (maximum approach).
2. Use of water as the medium in the high-temperature and ethanol as the medium in the low-temperature loop.
3. Choice of a four-cylinder spark-ignition engine for the fundamental examinations conducted. Considering the smaller absolute heat loss of a four-cylinder versus a six-/eight- or even a 12-cylinder power unit, this was seen as the worst case for a heat recuperation system.
4. Use of vane-cell expansion machines in both the high-temperature and low-temperature loop. The main reason for this choice was the very simple configuration and construction of an expander system of this type.
5. Application of expander performance with the focus on the optimum efficiency of the combustion engine and not on its maximum power.
6. Use of various types of heat exchangers in accordance with the operating conditions in each case in terms of heat exchanger performance, temperature and pressure levels, as well as volume (overall package).

Proceeding from these initial, basic decisions, the next step was to develop the process diagram of a possible cogeneration system presented in **Figure 4** [5]. This diagram shows the vane-cell expanders with a certain similarity – at least in graphic terms – with a turbo-

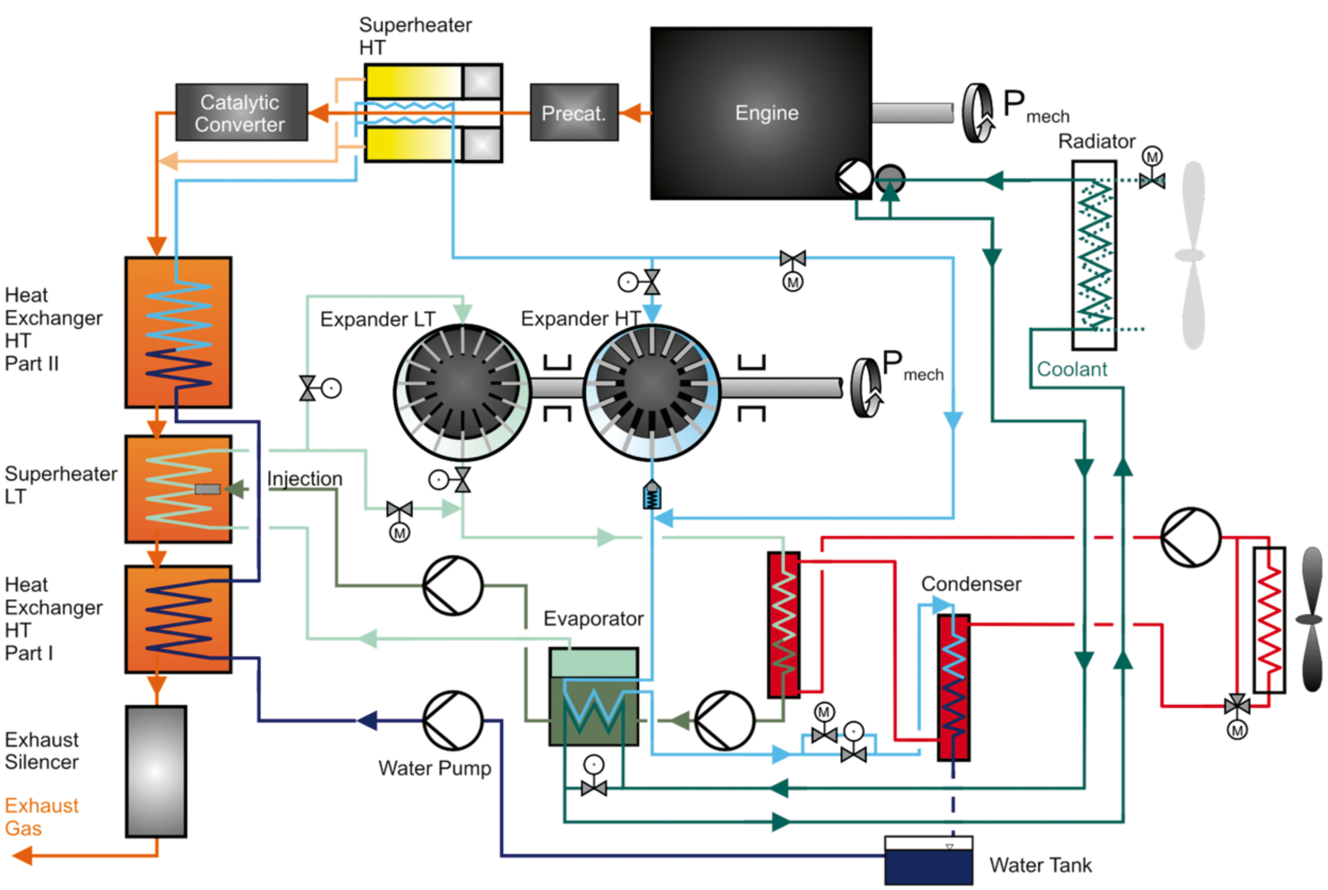

Figure 4: Process diagram of the Turbosteamer

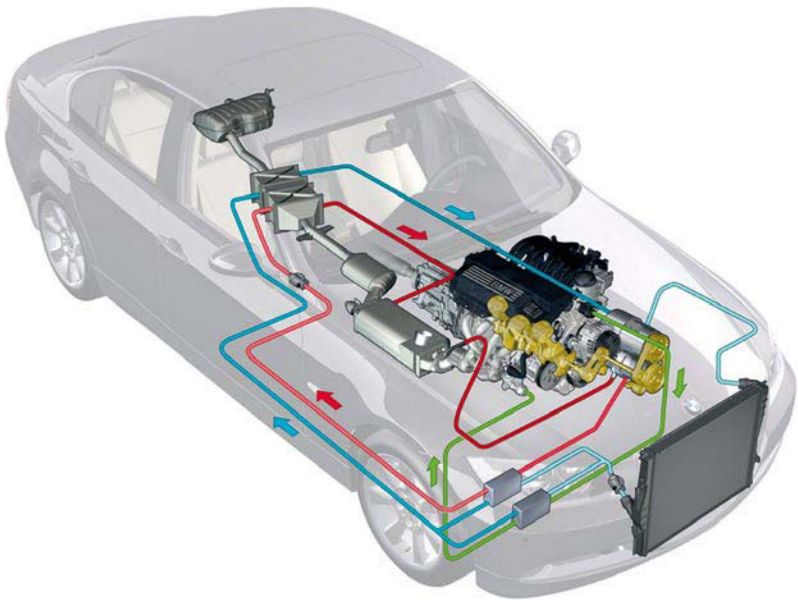

Figure 5: Turbosteamer in a BMW 3 Series

charger, which therefore led to the name „Turbosteamer“ given to the recuperation system.

After development of the appropriate hardware, all components were tested individually for their qualities. The results achieved in this process fulfilled and – with one exception – even outperformed the specifications established in advance. Only the tests on the vane-cell expanders were disappointing. With the objective being to develop a fully functioning overall system at an early point in time, the decision was taken not to seek further optimisation for the time being, but rather to replace the vane-cell expanders by axial piston machines.

Applying an appropriate CAx process, we were able to develop a technically sophisticated recuperation system featuring the right kind of package within a relatively short period. As is to be seen in **Figure 5**, the system largely fulfills the requirements for full integration in a BMW 3 Series. The structure, layout and configuration of the dual-circuit system developed on the basis of Figure 5 is described in the following:

The high-temperature loop (HT loop) consists of the pump, steam generator, superheater, expander, and condenser. Apart from its function as a heat exchanger, the steam generator also serves as a silencer in the exhaust system. Further issues such as exhaust gas back-pressure were also taken into account in the design, configuration and integration of this component. To avoid disadvantages in cold start emission behaviour, the superheater is positioned behind the precatalyst, its steam pipes forming part of the overall exhaust system. The condenser in the HT loop, on the other hand, fulfils a dual function, acting at the same time as the evaporator for the low-temperature (LT) loop.

In analogy to the HT loop, the LT loop likewise features the system components typically used in the Clausius-Rankine process – the pump, evaporator, superheater, expander, and condenser. Apart from condensation heat from the HT loop, the coolant heat from the engine is also fed into the medium in the LT evaporator. The medium is superheated in the exhaust gas heat exchanger downstream from the evaporator of the HT loop.

The expander machines of both cycles feed their power into the system – with an appropriate transmission ratio – mechanically on the crankshaft of the combustion engine.

4.2 Experimental Examinations

The objective of these first experimental examinations was to confirm the theoretical context presented in Chapter 3 and, in particular, to confirm the fuel savings predicted. Hence, the test bench configuration of the Turbosteamer, **Figure 6**, is laid out specifically to provide such confirmation and verify the theoretical assumptions made. The configuration thus provides precise information on the operation of the overall system consisting of the combustion engine with or without the Turbosteamer connected to the crankshaft, running at stationary operating points. In addition, this test array allows the development of initial strategies for controlling the dynamic behaviour of the system.

The very first practical examinations confirmed that the amount of heat loss is by all means significant, as is to be

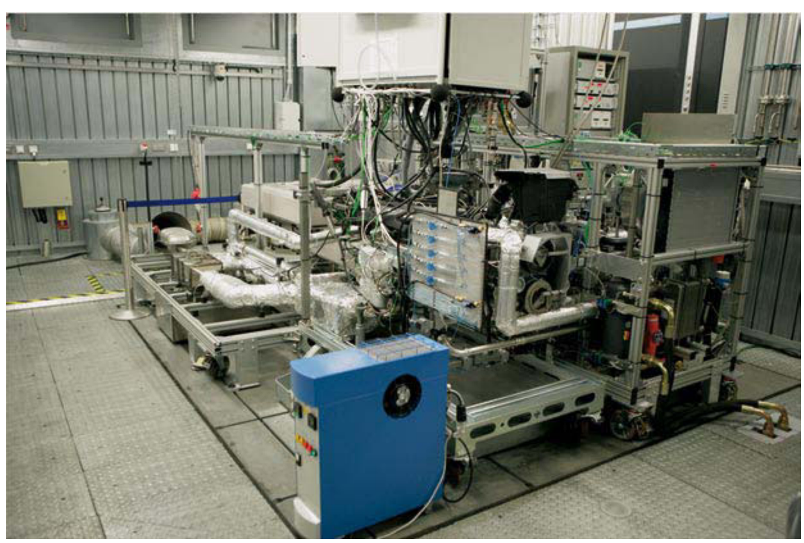

Figure 6: Test bench configuration of the Turbosteamer

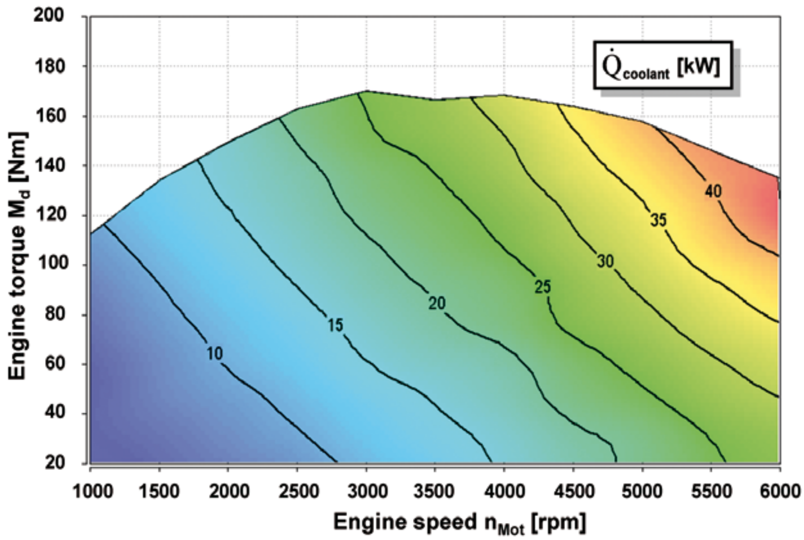

Figure 7: Heat power in coolant

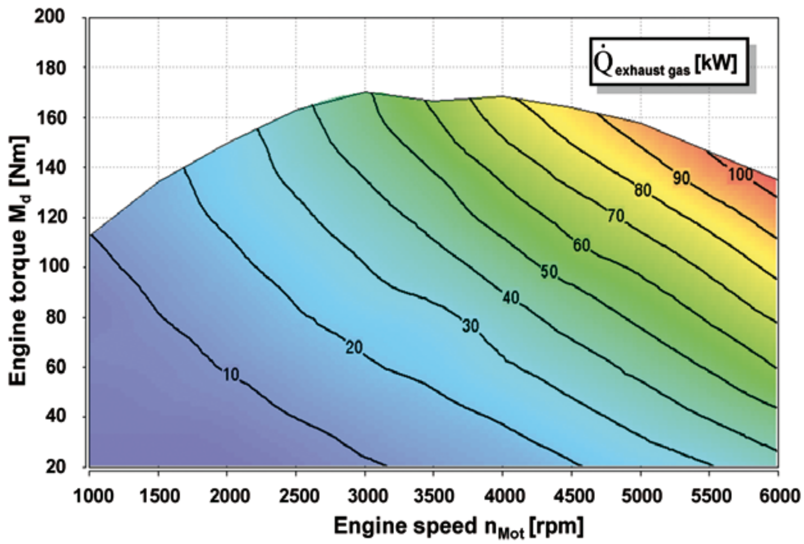

Figure 8: Heat power in exhaust gas

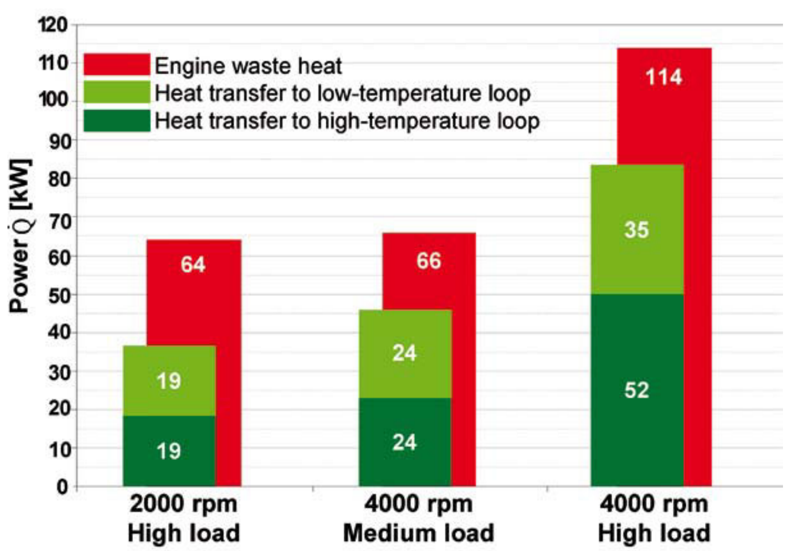

Figure 9: Heat transfer to the loops

seen in **Figure 7** and **Figure 8**. Given the similar magnitude of coolant and exhaust heat losses, it is advisable to take both heat flows into account in developing a Turbosteamer.

By and large, the potential of a Turbosteamer system in converting thermal energy loss into mechanical energy relates directly to the following two questions:

1. How good is the transfer of heat loss from the combustion engine into the steam cycles on the Turbosteamer?
2. To what extent can the heat within the loops be converted into mechanical energy?

To answer the first question, **Figure 9**, by way of example, presents the heat transfer into the LT and HT loops determined by way of experiment at various operating points in the engine. The result obtained is positive, since the coolant and exhaust gas heat available allows the generation of high-grade steam. The examinations have shown quite generally that the transfer of heat into the steam cycles does not present a problem. Indeed, it is fair to assume that approximately 80 per cent of the heat loss may be recuperated thermally. An impressive result in this context is also the reduction of the exhaust gas temperature within the system, **Figure 10**.

The second question does not allow such a direct, straightforward answer. The factors to be considered in this case are, first, the general management of the Clausius-Rankine process and, second, the efficiency of the expansion machine. Reference is deliberately not made here to the – many – options for the process management. On the other hand the tests conducted were able to provide a very clear picture of the behaviour of the axial piston machines used in terms of their efficiency: Throughout a broad range on the control map, the degree of efficiency established was 55 per cent. Taking the specific design and construction of the prototype into account, it is furthermore fair to assume that the potential of this component was not yet exhausted in full and that further optimisation should allow a degree of efficiency of $\eta_{expansion}$ of up to 70 per cent (in accordance with Eq. (12)).

While there is still significant potential for improvement within the test con-

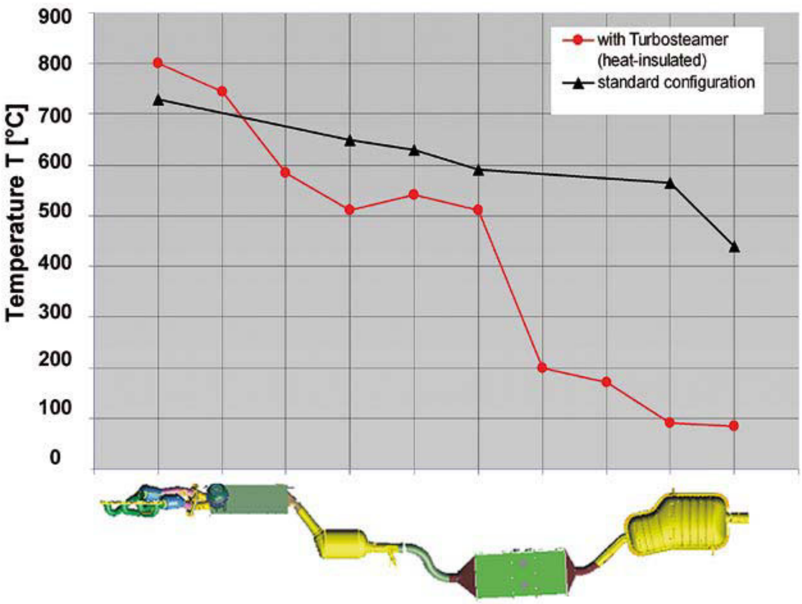

Figure 10: Reduction of the exhaust gas temperature in the exhaust system

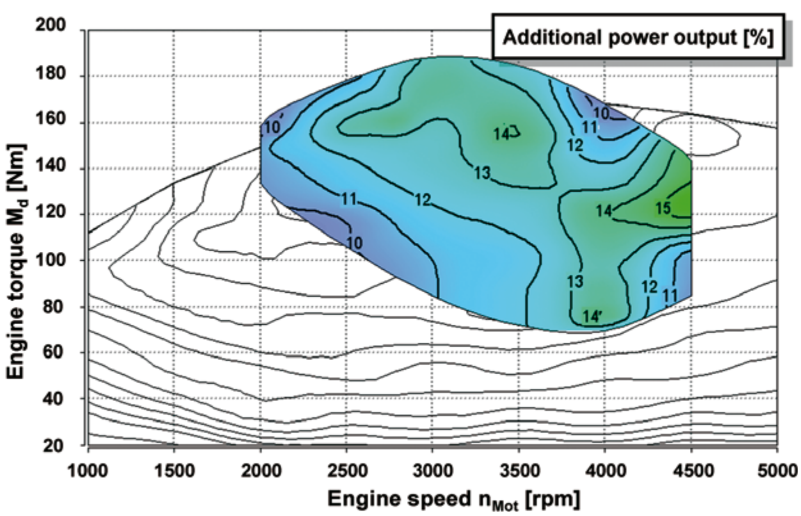

Figure 11: Relative additional power output of the Turbosteamer in the engine map

figuration used, the prototype tested already revealed the potential of a cogeneration system comparable in its magnitude to the theoretical forecast. As is to be seen in **Figure 11**, therefore, the Turbosteamer provides significant mechanical recuperation performance throughout an essential scope of the engine operating range.

5 Summary and Outlook

By way of experiment, the Turbosteamer for the first time proves the potential of a cogeneration system in enhancing the fuel economy of a motor vehicle. The thermodynamic estimates and calculations for configuring a Turbosteamer system based on theoretical parameters were confirmed for the first time in the tests conducted, which also showed that such a system can be appropriately accommodated in a motor vehicle in the right package.

The investigations currently being conducted focus on the reduction of the Turbosteamer's complexity, the reduction of its size, weight and cost, and the implementation of transient and highly dynamic driving cycles on the test bench.

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