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MASTER DEGREE PROGRAM

Development Engineer of Mechanical Engineering

Potential of charged combustion engines for the implementation in motorcycles

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SUPERVISED BY

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ABSTRACT

For the last few years, the charging technology has been playing a more and more significant role in modern combustion engine development. This thesis delivers an introduction into the thematic of charged combustion engines and shows how this technology can be used for the implementation in motorcycles. At the beginning, the thermodynamic fundamentals of the charging technique are worked out. One of the main objectives will be the comparison of the different charging systems in terms of the most effective implementation in motorcycle engines. In the next step, attention will be drawn to the implementation of charge air cooling systems in motorcycles. At the end, the potential for using the charging technique for motorcycles will be examined.

TABLE OF CONTENTS

SYMBOLS

h, h_1, h_2	specific enthalpy/ specific enthalpy inlet/ specific enthalpy outlet	$\frac{J}{kg}$
c_1, c_2	gas speed inlet/ gas speed outlet	$\frac{m}{s}$
q_a	specific heat flow (above system boundary)	$\frac{J}{kg}$
p_1, p_4	total external gas pressure	Pa
p_2	total pressure after dynamic compressor or total pressure before positive displacement compressor	Pa
p_3	total pressure after positive displacement compressor or total pressure before turbine	Pa
π_e, π_c	compression ratio engine/ compression ratio compressor	$-$
T_1	external temperature	K
T_2	temperature after dynamic compressor or temperature before positive displacement compressor	K
ρ_1, ρ_2	external density/ density after dynamic compressor or density before positive displacement compressor	$\frac{kg}{m^3}$
cyl	number of cylinder	$-$
V_h	cylinder capacity	m^3
L_{st}	stoichiometric fuel rate	$\frac{kg_{air}}{kg_{\ gas}}$
λ_l	volumetric efficiency	$-$
m_{cyl}	real mass of gas which remains in cylinder after suction stroke	kg
m_{th}	theoretical mass of gas which remains in cylinder after suction stroke	kg
$\dot{m}_c$	mass flow compressor	$\frac{kg}{s}$
$\dot{V}_{suck}$	from combustion engine sucked air flow	$\frac{m^3}{s}$

n, n_e, n_c	speed / engine speed / compressor speed	$\frac{1}{s}$
H_u	lower heating value	$\frac{J}{kg}$
κ	isentropic exponent	$-$
R	gas constant	$\frac{J}{kgK}$
w, $w_e w_i$	specific work / specific effective work /specific inner work	$\frac{J}{kg}$
$w_{i\text{-}c}$	specific inner work input compressor	$\frac{J}{kg}$
$w_{s\text{-}pdc}$	specific work by isentropic state change positive displacement compressor	$\frac{J}{kg}$
$w_{s\text{-}dc}$	specific work by isentropic state change dynamic compressor	$\frac{J}{kg}$
$w_{i\text{-}ce}$	specific inner work output combustion engine	$\frac{J}{kg}$
η_s	isentropic efficiency coefficient	$-$
$\eta_{s\text{-}c}$, $\eta_{s\text{-}dc}$	isentropic efficiency compressor/ isentropic efficiency dynamic compressor	$-$
$\eta_{s\text{-}pdc}$	isentropic efficiency positive displacement compressor	$-$
η_m, $\eta_{m\text{-}c}$	mechanical efficiency / mechanical efficiency compressor	$-$
η_p	process efficiency	$-$
M_i	inner engine torque output	Nm
P_i	inner engine power output	W
$P_{e\text{-}c}$	effective power input compressor	W

1 INTRODUCTION

1.1 Preface

For the last few years, the charging technology has been playing a more and more significant role in modern combustion engine development. From the early beginnings and applications in aircraft and sport car engines, this technology has highly enhanced over the years. Nowadays it is a state of the art technique for the use in combustion engines of trucks and vehicles. Because of this, the potential of the charging technique for the use in prospective motorcycle engines will be analyzed.

1.2 Scope of work

The following work gives a general understanding of the charging technique and its potential for the use in combustion engines. Especially positive aspects for the application in motorcycles are outlined. It should be noted that in the present paper only four stroke engines which are operated with gasoline are covered. The influence on the endurance strength of a combustion engine will also not be treated.

At the beginning, a brief summary is given of the thermodynamic fundamentals for compressors and combustion engines. Furthermore, important correlations between the engine and the charger are worked out. Further on, essential equations are written down for a better understanding of the present paper. However, it is not the scope of the work to cover the whole thermodynamic basics and therefore a basic knowledge is presumed in the field of thermodynamics.

Based on the fundamentals, a comparison of the different charger systems is made and it is shown how they can be used for applications in combustion engines. More precisely the operational characteristics of the different charger systems are worked out and possible linkages between the compressor and the engine are shown. Furthermore, possibilities to

adjust the compressor for different engine operating conditions are discussed and systems which can be used for this aim are presented.
Attention is also paid to the effect of the increased air temperature, which occurs in real compression processes. Therefor different systems to compensate this negative effect are introduced.
Finally, the potential of using a charger for the application in a motorcycle is analyzed and the operation performances of the different charging systems are compared critically.
At the end a brief conclusion about the charging technique and the potential for its use in motorcycles is given. Difficulties which may occur are recapped and suggestions to solve these problems are offered. An outlook for prospective trends is presented as well.

2 FUNDAMENTALS

2.1 Thermodynamics

In the following section the thermodynamic fundamentals, which are needed for the understanding of the present paper, are worked out. It should be noted that this is not a complete repetition of the thermodynamic basics. Those are not discussed in this section and for further information see e.g. [1,2]

According to [1] the work of a stationary flow through an open system can be calculated with the first law of thermodynamics for open systems with:

$$w_i = h_2 - h_1 - q_a + \frac{c_2^2}{2} - \frac{c_1^2}{2} + g(z_2 - z_1) \tag{1}$$

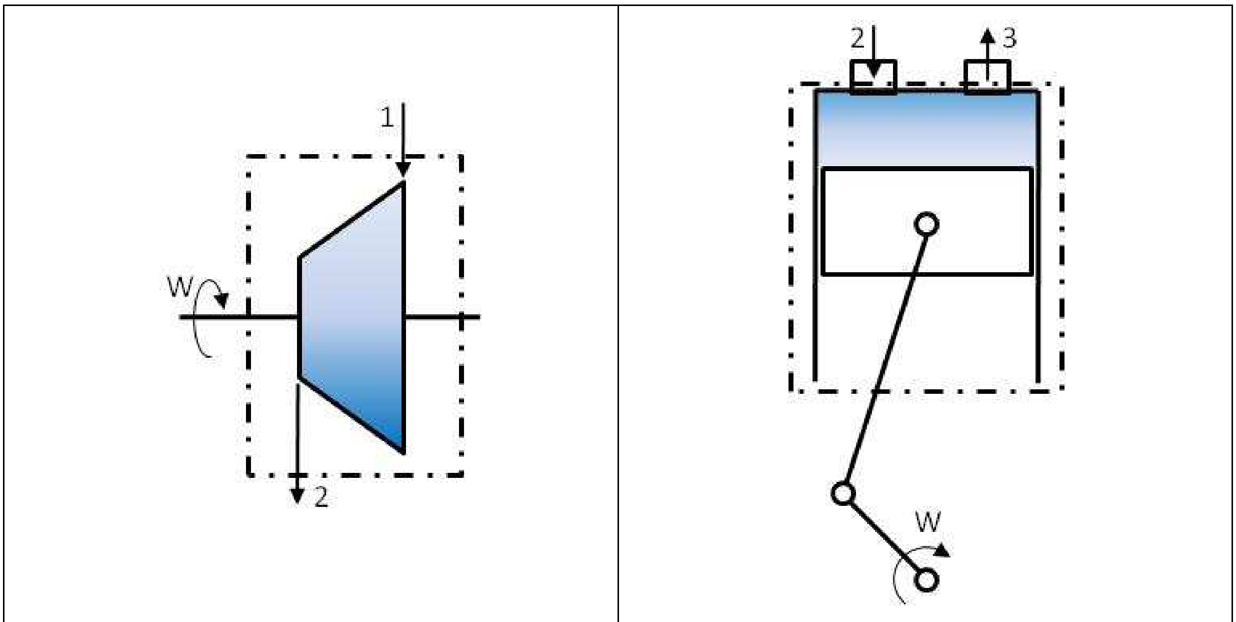
Usually the kinetic energy and the potential energy (for a gaseous flow medium) can be neglected. This leads to:

$$w_i = h_2 - h_1 - q_a \tag{2}$$

This equation is generally valid and can be used for the evaluation of the inner work from the compressor or the turbine. The process losses of the compression work are therefore included in the enthalpy flows and it should be stated that Eq. (2)contains no driving shaft friction or other outside losses. To consider this, a mechanical efficiency η_m is added, which results in the effective work:

$$w_e = \frac{w_i}{\eta_m} \tag{3}$$

On the basis of these formulas the work of the charging system can be derived. Figure 1shows the different charging systems. For a dynamic compressor1denotes the inlet of the medium and2marks the outlet, while for the positive displacement compressor the inlet is named 2 and the outlet 3. The work which has to be imported into the thermodynamic system is named W.



dynamic compressor	positive displacement compressor

Figure 1: schematic picture from different type of compressors

The work of the compressors can be modeled with an isentropic state change. According to [1] the compression work of a dynamic compressor can be calculated with (Eq. 4) and the positive displacement compressor with (Eq. 5).

$$w_{s-dc} = \frac{\kappa}{\kappa-1} p_1 v_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \tag{4}$$

$$w_{s-pdc} = \frac{\kappa}{\kappa-1} p_2 v_2 \left[\left(\frac{p_3}{p_2} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \tag{5}$$

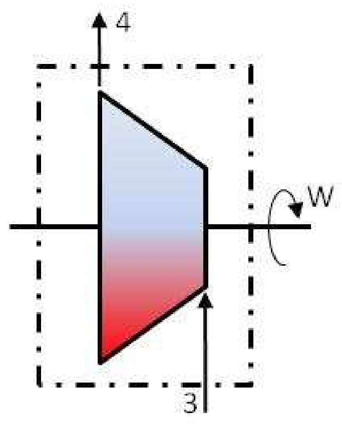
However there are no friction losses during the compression work, which are modeled with Eq. (4,5). Hence an efficiency coefficient has to be used to derive the inner work of the compressor. For this Eq. (4) is multiplied with the isentropic efficiency coefficient of the dynamic compressor and Eq. (5) with the isentropic efficiency coefficient of the positive displacement compressor, which results in the inner work of the dynamic compressor Eq. (6) and the inner work of the positive displacement compressor Eq. (7).

$$w_{i-dc} = w_{s-dc} \cdot \frac{1}{\eta_{s-dc}} \tag{6}$$

$$w_{i-pdc} = w_{s-pdc} \cdot \frac{1}{\eta_{s-pdc}} \tag{7}$$

To get the effective work of the compressor, Eq.(6,7) has to be inserted into Eq. (3) with the mechanical efficiency of the compressor.

In Figure 2 a turbine is shown and the inlet of the medium is written with 3 and the outlet with 4.



turbine

Figure 2: schematic picture of a turbine

For the computation of the turbine work Eq. (2) can be applied. The only difference to the compressor formula is that the work is going outside the thermodynamic system. According to [1] the work of the turbine, is calculated with an isentropic state change:

$$w_{s-t} = \frac{\kappa - 1}{\kappa} p_3 v_3 \left[1 - \left(\frac{p_4}{p_3}\right)^{\frac{\kappa-1}{\kappa}}\right] \tag{8}$$

Multiplied with the isentropic efficiency of the turbine we find the inner work:

$$w_{i-t} = w_{s-t} \cdot \frac{1}{\eta_{s-t}} \tag{9}$$

The effective work of the turbine is then derived with Eq. (3).

Cf. [1]

2.2 Combustion engine

In the subsequent passage a brief introduction of some important combustion engine characteristics is given. For additional information see e.g. [1,2,7].

The goal of the combustion engine is to convert the chemical energy of the fuel into mechanical work. For this process a specific ratio of oxygen(provided by air)and fuel has to be fulfilled to reach a complete combustion. Therefor the stoichiometric fuel rate for gasoline engines is introduced:

$$L_{st} = 14{,}7 \left[\frac{kg_{air}}{kg_{\ gas}}\right] \tag{10}$$

The engine can be seen as a positive displacement compressor (Figure 1), where the amount of air, which has to be sucked by the engine, can be calculated(according to [2]) with Eq. (12). This formula is simplified and only valid for four stroke engines. The influence of the valve overlap is not considered with this equation. But for this operation the volumetric efficiency (obtained from [7]) Eq. (11) is needed. That quotient describes the ratio of the real amount of gas which remains after the suction stroke in the cylinder, to the theoretical remaining gas in the cylinder. Furthermore the volumetric efficiency strongly influences the torque Eq. (14)and the power Eq. (15)of the combustion engine.

$$\lambda_l = \frac{m_{cyl}}{m_{th}} \tag{11}$$

$$\dot{V}_{suck} = \frac{V_h \cdot n}{2} \cdot \frac{p_2}{p_1} \cdot \frac{T_1}{T_2} \cdot \lambda_l \tag{12}$$

According to [2], the inner work of the combustion engine can now be derived from:

$$W_{i-ce} = \frac{V_h \cdot H_u \cdot \eta_p}{L_{st}} \cdot \rho_2 \tag{13}$$

For convenience the volumetric efficiency is not included in Eq. (13).

It can be seen that there is a proportional dependency between the density of the gas and the inner work of the combustion engine. Hence, with a precompressing of the gas it is possible to increase the inner work of the engine proportional to its density.
With the previous equations and according to [5] the inner torque of the engine can be written with Eq. (14) and the inner power with Eq (15).

$$M_i = \frac{cyl}{4\pi} \cdot \frac{V_h \cdot H_u \cdot \eta_p}{L_{st}} \cdot \rho_2 \tag{14}$$

$$P_i = \frac{cyl \cdot n}{2} \cdot \frac{V_h \cdot H_u \cdot \eta_p}{L_{st}} \cdot \rho_2 \tag{15}$$

To derive the effective torque and power from the engine, a mechanical efficiency has to be attached to the formulas Eq.(14) and Eq. (15).
An important factor is to determine the operating performance between the engine and the compressor. This can be studied with pressure air flow maps. In these diagrams the compression ratio of the engine for different engine speeds is plotted over the air flow capacity. Corresponding to [2], the compression ratio of the engine is the quotient of the total pressure in the cylinder (before the agglomeration stroke begins) and the external total pressure:

$$\pi_e = \frac{p_2}{p_1} \tag{16}$$

A schematic diagram of the engine pressure ratio plotted over the air flow capacity for a combustion engine without valve overlapping is shown in Figure 3.

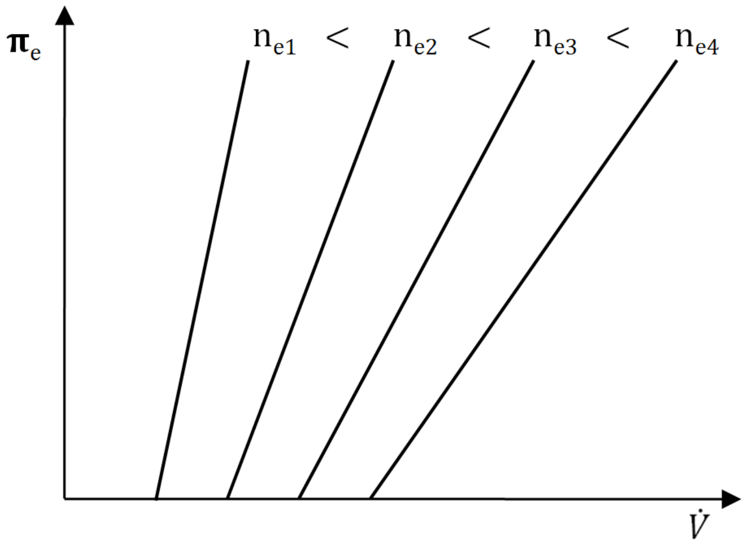


Figure 3: Engine pressure ratio over the air flow capacity of a combustion engine for different engine speeds

It can be seen that for a constant engine speed with increasing pressure ratio the sucked air flow rises. This effect of the rising air flow is becoming more significant for higher engine speeds.
Cf. [1,2,4,5]

2.3 Charging system

Charging is used to precompress the air and thereby to raise the air density which is sucked from the engine into the combustion chamber. This effect increases the inner work (Eq.(13)) and consequently the inner torque (Eq.(14)) and inner power (Eq.(15)) of the engine. The density of an ideal gas can be calculated with:

$$\rho = \frac{p}{R \cdot T} \tag{17}$$

With the above equation it can be seen that the gas density can be raised by increasing the pressure of the gas (precompression) or by reducing the temperature of the gas. However, a precompression of a gas always leads to a raise of temperature. Hence, the elevation of temperature partly compensates the effect of the higher air pressure.

This increased air temperature induced by the compression can be calculated corresponding to [2] with:

$$T_2 = T_1 \left[1 + \frac{1}{\eta_{s-c}} \cdot \left[\left(\frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} - 1 \right] \right] \tag{18}$$

For further information it is referred to the section of the air cooling systems.

An important factor for the characterization of compressors is the compressor ratio, which is defined by the quotient between the total pressure after and before the compressor:

$$\pi_c = \frac{p_2}{p_1} \tag{19}$$

Further on, this ratio is needed for the comparison of the different charging systems.

The effective power which is needed to compress the air by a compressor can now (according to [2]) be derived from:

$$P_{e-c} = \frac{1}{\eta_{m-c}} \cdot \dot{m}_c \cdot w_{i-c} \tag{20}$$

Cf. [1, 2, 4]

3 CHARGING SYSTEMS

The following chapter gives an general overview of the operation characteristics of different charging systems. It should be noted that not every available charging system can be discussed in this paper. Only systems which can be used for the application in motorcycles are treated in this work. For further information see e.g.[2,3,4].

3.1 Operation characteristics

Based on the theoretical background of the charger systems in section 2,their operational characteristics will now be examined. For this purpose, the pressure ratio is plotted over the air flow. (see e.g. Figure 4, Figure 5)

A schematic pressure flow map for a positive displacement compressor is shown in Figure 4. The diagram indicates the relation between the compressor ratio and the air flow with lines of constant compressor speed. The maximum reachable air flow is limited due to the maximum speed of the compressor which is stated with $n_{c\text{-}max}$. The dotted lines show the contour lines of the isentropic compressor efficiency. The area with the green color indicates the max. efficiency and the red color the minimum.
If this characteristic map is compared to the combustion engine (Figure 3),it can be seen that the constant speed lines for higher compressor ratios move in the opposite direction. This means that the air flow decreases with higher compressor ratios. Furthermore, the map indicates that a high pressure ratio can already be reached with a low compressor speed. It is also an advantage that there is no instable area which can be reached with this compressor. Finally, it is stated that the efficiency in Figure 4 is only schematic. The efficiency of a positive displacement compressor strongly depends on the layout of the compressor. The maximum efficiency is reached at the minimal leakage, choke and the friction losses.

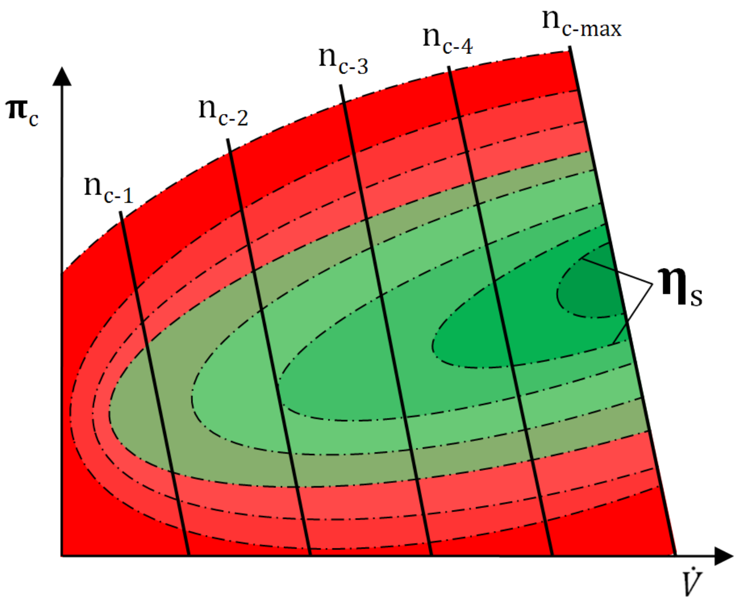


Figure 4: schematic map of pressure ratio over air flow from apositive displacement compressor

The second type of charger is the dynamic compressor (Figure 1 left). For the discussion of the operation characteristics the pressure ratio is plotted over the air flow (illustrated in Figure 5).

In this diagram, the lines for constant impeller speed are presented with continuous lines. The maximum possible impeller speed is marked with n_{c-max}. The dotted lines show the isentropic efficiency of the compressor for different operating conditions. Thereby the green color is the area of high compressor efficiency and the red one of low efficiency. In contrast to the positive displacement compressor, the dynamic compressor shows an instable area for high compressor ratios with small air flows. In this instable part of the map the compressor should not be operated otherwise the impeller is going to pump. For this purpose the surge line is introduced which separates the instable left part of the map from the usable one. But this line is not only compressor specific. It strongly depends on the suction pipes and boxes before the impeller inlet.

On the right side the compressor map is limited by the choke line. After this line the air flow could reach supersonic speed and cannotbe raised anymore by reducing the pressure ratio.

In contrast to the positive displacement compressor the dynamic cannot reach high pressure ratios for low charger speed. This means that for higher pressure ratios the speed has to be increased.

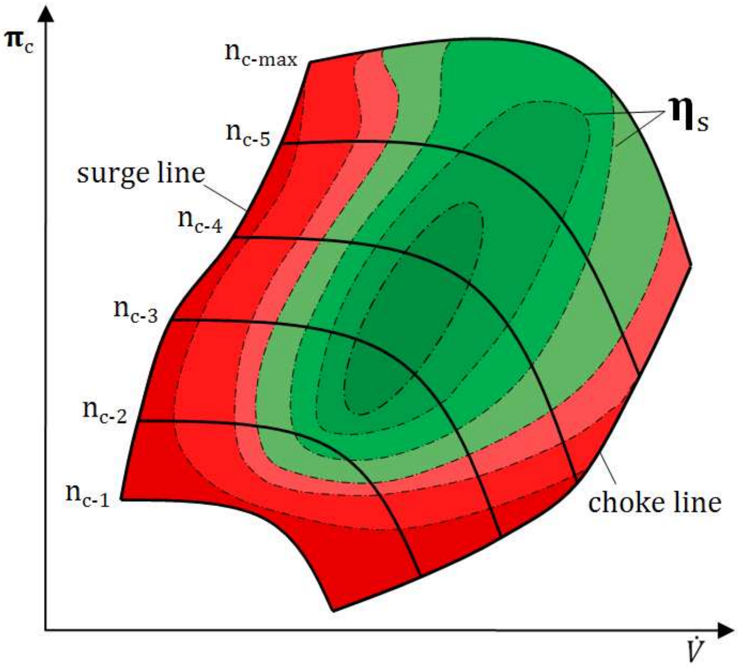


Figure 5: schematic map of pressure ratio over air flowfrom a dynamic compressor

3.2 Compressor drive

As we have seen so far, there are two types of compressors which are used for boosting and precompressing a medium. Furthermore, they can be differentiated by how they are driven.

One possibility is to drive them with a belt or a gear wheel drive, which is then named mechanical charger. Or for dynamic compressors it is also possible to use a turbine (Figure 2) for driving the impeller. This system is named "rigid geometry exhaust turbo charger" when the geometry of the turbine inlet is unchangeable. Or "exhaust turbo charger with variable turbine geometry (short VTG)" for a changeable turbine inlet. Additional systems which are propelled by electric motors or with an auxiliary hydraulic system exist. But these will not be covered here.

3.3 Engine load control

For gas engines the adjusting of the loading has to be done by varying the flow of the fuel air mixture in the engine. For this reason, the possibility to adjust the flow of the charger for the different loading conditions of the engine has to be considered. There are a few systems which can be used for this task(Figure 6). In the following, different control systems will be introduced and discussed for the application with the positive displacement or dynamic compressor.

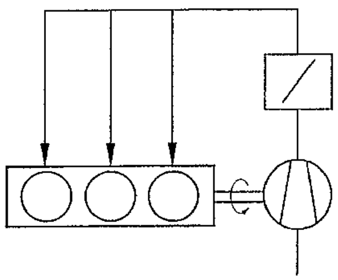
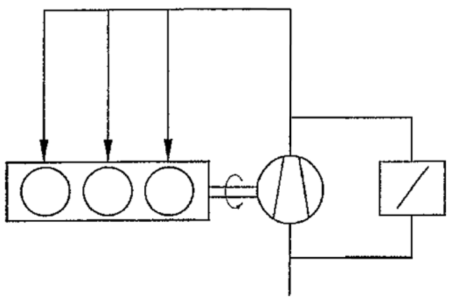
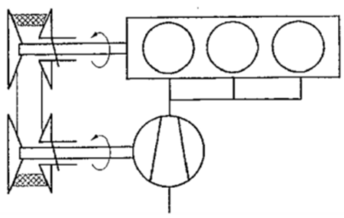
		
Throttle valve	Backflow valve	Variable gear drive

Figure 6: systems for controlling the engine load [4]

One possibility of controlling the flow is a throttle valve which can be plugged in before or after the compressor. However, only the system with the throttle after the compressor is discussed. For full load conditions the throttle valve is fully opened. Only if partial load conditions are required the valve is closed partly and the pressure ratio after the impeller increases. Hence, the compressor load point moves along the line of constant compressor speed to the left and the air flow in the engine decreases (shown in Figure 7). Therefore, less air is pumped into the combustion chamber and the engine load is reduced. The increasing pressure ratio leads to a decrease of the air flow. As a consequence, the positive effect of the reduced air flow is decompensated with the higher compressor ratio. This is the reason why the compressor driving power is not positively affected by this. However this is not an optimal solution in order to minimize the fuel consumption in engine part load conditions.

When using dynamic compressors, attention also should be paid to not moving the operational point into the unstable area of the compressor map.

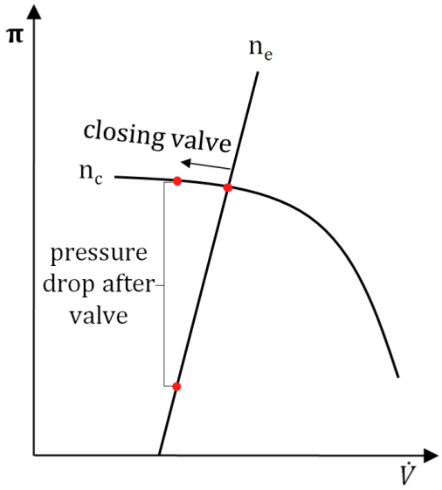


Figure 7: controlling the engine load with a closing valve

Another possibility for controlling the load is to blow a part of the boosted air back in front of the compressor and in this way reduce the flow into the engine. For this purpose a piping system that connects the outlet and the inlet pipe of the impeller is installed. The amount of air can then be controlled by a valve which is mounted in the backflow pipe. For part load condition the valve is opened and a part of the air is flowing back to the inlet of the compressor which leads to a decreasing compressor ratio. Therefore the compressor load point moves along the constant compressor speed line to the right (shown in Figure 8). The engine load is then reduced because of the lower air flow into the cylinder. For this system the effect of the compressor driving power at part load is similar to the throttle valve system. The positive effect of the pressure ratio decrease is decompensated by the increased air flow. So there is in fact no positive effect in the compressor driving power by operating the system in part load.

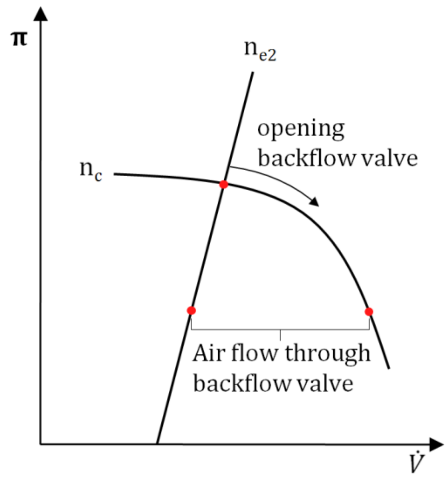


Figure 8: controlling the engine load with a backflow valve

An alternative system for controlling the engine load is to vary the speed of the compressor.
For mechanical chargers this can be done with a variable gear system between the engine and compressor. With this system the compressor can be optimally adapted to the operating conditions of the combustion engine. Especially for operating the compressor in engine part- load this system can reduce the compressor driving power drastically. Therefore it has a big efficiency advantage in part- load conditions in contrast to the previously discussed systems.
If a VTG turbo charger is used, the speed of the impeller can be varied with guide vanes which are adjusting the angle of the inlet flow into the turbine blades. This type of exhaust charger can achieve a higher pressure ratio already for lower exhaust flows than similar rigid geometry exhaust chargers. Additionally, this exhaust charger is easier to adapt for different engine operating conditions. For further information see e.g. [2, 4, 6]

Cf. [2, 3, 4, 6]

4 CHARGE AIR COOLING

In the following section a brief introduction to air cooling systems will be given. For further information see e.g. [2,3,6]

As shown in Eq. (18), a precompressioning of a gas in a compressor leads to an increase in the gas temperature. However this influences the density of the gas in a negative way, which can be seen in Eq. (17) where an increasing of the air temperature reduces the density of the gas. Therefore, the positive effect of a precompression is decompensated due to a higher gas temperature.

Figure 9showsthe influenceof re-cooling the gas back to the temperature before the compressor. In this diagram different values for constant density ratios (Eq. (21)) are plotted against the compressor ratio (y-axis) and the isentropic efficiency (x-axis).

$$density\,ratio = \frac{\rho_2}{\rho_1} \tag{21}$$

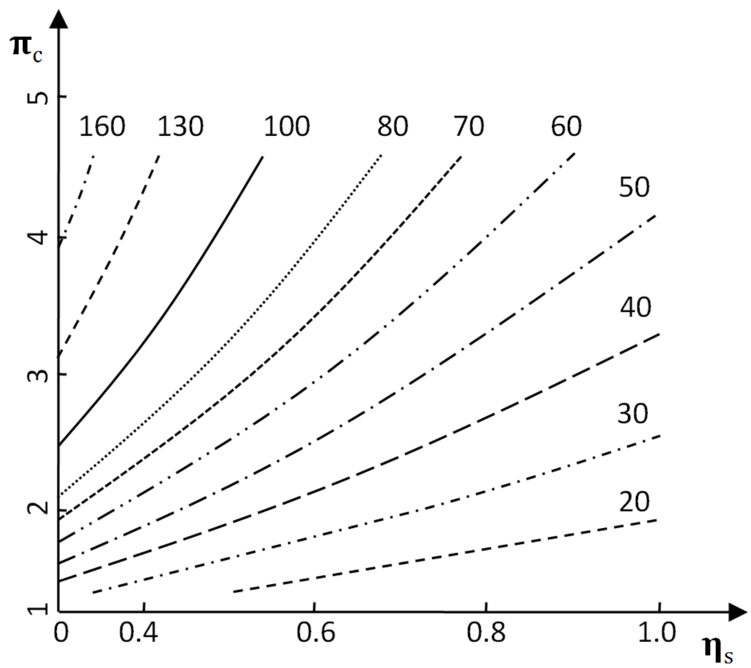


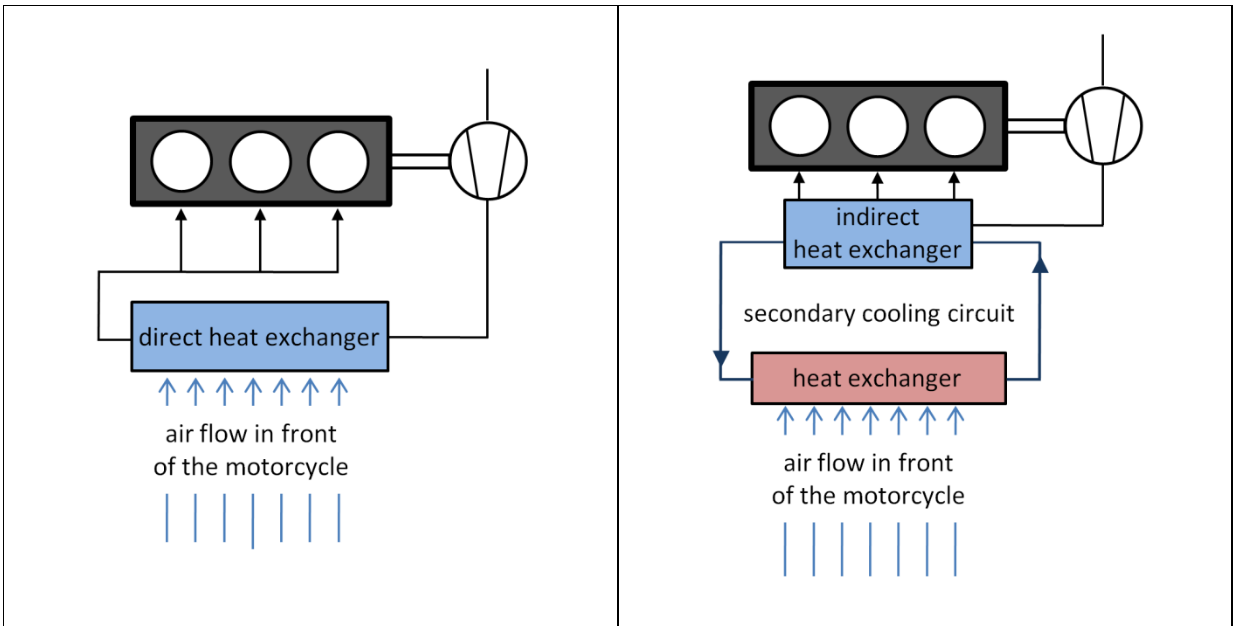
Figure 9: percentage of air density increase when gas is cooled back to the temperature before the compressor [4]

Moreover, this diagram illustrates how much potential exists in cooling back the charged air for different isentropic efficiencies and pressure ratios. According to Eq. (14,15),the density of the air is directly connected with the power and the torque of the engine. Therefore the re-cooling is

highly important for the operating performance of an engine. Additionally, the emissions and the knocking combustion (only for gas- engines) are positively affected by reducing the air temperature.
Subsequently, two systems for cooling the charged air will be introduced.

4.1 Air cooling systems

The cooling of the air can be achieved with different types of systems. Nowadays, primarily systems which use heat exchangers for cooling the charged air are used and are "state of the art". Other systems will not be discussed in this work.
Among these, it can be distinguished between systems in which the charged air is cooled directly (called direct air cooling). Or systems in which the air is cooled indirectly with a secondary cooling circuit (called indirect air cooling). For the application in motor vehicles the direct heat exchanger is cooled with the airflow in front of the vehicle and the indirect system with the cooling water of the secondary cooling circuit. The chill water of the secondary system is tempered with a heat exchanger where the airflow in front of the vehicle is used as cooling medium. The layout of the two systems is illustrated in Figure 10.



direct air cooling system	indirect air cooling system

Figure 10: different charge air cooling systems

A big advantage of the indirect system is that the charge air heat exchanger is smaller than the heat exchanger of the direct system. Furthermore, there are no charge air pipes needed for the layout that go from the compressor to the heat exchanger and from the heat exchanger to the intake air plenum. In fact the head loss of the indirect system is much lower than the one of the direct system. Nevertheless, the complexity and the weight of the system increases by using the indirect charge air cooling layout. But for motorcycle applications, where the available space is rare, the indirect system is favored.

Cf. [2, 3,4]

5 CHARGING SYSTEMS FOR THE USE IN MOTORCYCLES

After the essential features and properties of the charging technique have been worked out in the previous sections, now the possible potential to use this technique for motorcycle engines with the help of an available full load characteristic of a street bike will be discussed. However, it should be noted that finding an optimum full load curve to improve the drivability of the motorcycle without making it undrivable is not an easy question. For further information see e.g.[7].

Then, in a further step, the operation performance of different charging systems for motorcycle applications will be analyzed.

5.1 Expected potential of charged motorcycles

On the basis of the engine full load characteristics of a street bike the potential of using the charging technique for motorcycles will be discussed. A schematic torque curve of a street bike with a cylinder capacity of 600ccm is shown in Figure 11. For the data source see [8].

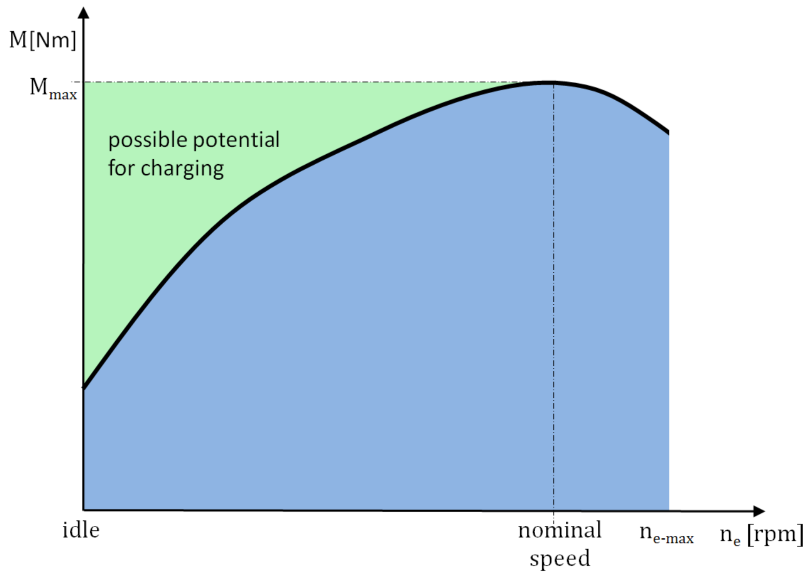


Figure 11: schematic full load characteristicof a motorcycle with a cylinder capacity of 600ccm, in which the green area shows apossible potential for the charging technique

In the diagram, it can be seen that the maximum torque of the engine occurs relatively late at a rather high engine speed. With lower engine speeds, this leads to a lazy start up behavior so that the engine has to be kept on higher speeds in order not to lose the ability for a fast acceleration.

In this lower speed area therefore, there exists a potential for increasing the moment with a charger. This would lead to a better start up behavior of the motorcycle. Furthermore, a charger with a fast response time can positively affect the start up behavior at part- load operating points. However, it is not discussed how this full load curve shall look like.

5.2 Examination of the operation performance of charged motorcycle engines

In the following, the operation performance of motorcycle engines with mechanical chargers for engine full- load is analyzed. For this purpose, schematic diagrams, of the pressure ratio over the air flow, for combustion engines (Figure 3) and for the different compressor systems (Figure 4, Figure 5) are used.

With the help of these diagrams, the operation performance of the positive displacement compressor (Figure 12left) and the dynamic compressor (Figure 12 right) for an application with a combustion engine can be studied.

For both diagrams the compressor is linked to the engine with a fixed gear ratio. In doing so, two different values of gear ratios are used, which are shown with continuous lines for the first gear ratio and with dotted lines for the second gear ratio. The two operating curves are the result of the intersection of the constant speed lines with the constant compressor lines.

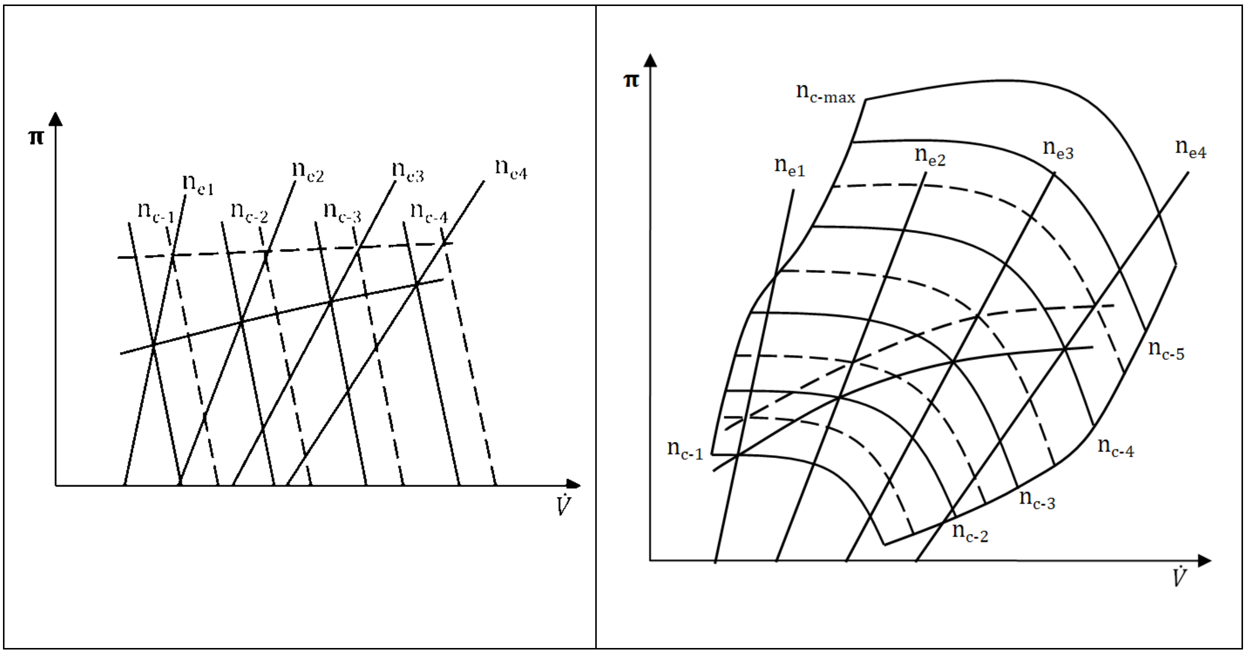


Figure 12: schematic diagram which shows the interaction between the positive displacement compressor and the combustion engine (left) and the dynamic compressor with the combustion engine (right)

The diagram shows, that for the application with the positive displacement compressor, an adequate compressor ratio can be achieved for the whole speed range.

But the dynamic compressor cannot reach an optimum compressor ratio at a low speed level. Therefore the gear ratio has to be increased. For lower flows, however this leads to problems when the operating point

moves into the instable area, or for higher flows when the choke line or the maximum allowed compressor speed is reached.
However, the optimum solution would be to drive the compressor with a variable gear drive. In order to do so, an optimal torque curve at full load conditions can be modeled and the positive effects for controlling the engine load (Section 4.3) can be used.

Cf. [2, 4, 7, 8]

6 CONCLUSION

The charging technique has become indispensable in modern engine development and extends over a big field of applications from the use in highly charged sport engines to efficient series engines. However, for the use in motorcycles this technique has not been established until now.
When looking on at the full load characteristic of a motorcycle it can be seen that in comparison to the nominal speed torque, for lower engine speeds only a marginal value of torque is available. Therefore, a possible potential for increasing the torque with a charger exists. Now the question occurs how this full load curve should look like in order to increase the start up behavior at lower engine speeds while not making the motorcycle undrivable. This question was not covered in detail and should be subject of further research.
It is however, certain that if a specific full load torque curve is modeled by the use of a compressor, it is advantageous to drive the compressor with a variable gear ratio. This variable gear drive is not absolutely necessary when modeling the full load torque curve with a positive displacement compressor, but for dynamic compressors it is mandatory. However, for an efficient operating of the compressor in engine part- load conditions, a variable gear drive is absolutely necessary.
Another important factor when charging a combustion engine is the charge air cooling, which plays a significant role especially for higher compressor ratios with lower isentropic compressor efficiencies. Because

of the little space available in motorcycles, it is recommended to use the indirect charge air cooling system. Another positive aspect of the indirect system in contrast to the direct system is the lower head loss, which occurs along the charge air cooling system.

Due to the rapid evolution of the charging systems, it is likely that in the future new systems with better operating conditions will be developed. As a result, new possibilities for charging motorcycle engines will arise, and the sense of charging motorcycle engines will be questioned once again. This work shall therefore be used as a basis for this question.

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