CZECH TECHNICAL UNIVERSITY IN PRAGUE

Faculty of Mechanical Engineering

Department of Automotive, Combustion Engine and Railway Engineering

HYBRID VEHICLE POWERTRAIN WITH RANGE EXTENDER

DOCTORAL THESIS

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Doctoral study programme:	Mechanical Engineering
Branch of study:	Machines and Equipment for Transportation
Doctoral thesis tutor:	**prof. Ing. Jan Macek, DrSc.**

Prague, 2021

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DECLARATION

Ph.D. student's name:
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Title of the doctoral thesis:
Hybrid Vehicle Powertrain with Range Extender

I hereby declare that this doctoral thesis is my own work and effort written under the guidance of the tutor **prof. Ing. Jan Macek, DrSc.**

All sources and other materials used have been quoted in the list of references.

The doctoral thesis was written in connection with research on the project:

• **SGS13/184/OHK2/3T/12** Electromobility and Utilization of Internal Combustion Engine as a Range Extender;

• **SGS16/213/OHK2/3T/12** Energy Consumption and Emission Reduction of Modern Vehicles in a Real Drive Cycle.

Prague, 10th July 2021
(added to 27th May 2022)

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Signature

Abstrakt

Disertační práce se zabývá problematikou pohonných jednotek s prodlužovačem dojezdu pro hybridní vozidla. Zaměřuje se zejména na spalovací motory využívané v těchto jednotkách a jejích návrhem. Vzhledem k tomu, že bateriová elektrická vozidla stále používají těžké, objemné a drahé baterie, které nabízejí velmi proměnlivý, v některých případech nedostatečný dojezd, představují pohonné jednotky s prodlužovačem dojezdu rozumné řešení těchto problémů. Prodlužovače dojezdu jsou pomocné pohonné jednotky, které přeměňují energii ukrytou v palivu na energii elektrickou. Pro pohon elektrického generátoru se používá většinou specifický typ spalovacího motoru. Prodlužovač dojezdu umožňuje ne jenom prodloužit dojezd vozidla, ale také zmenšit kapacitu baterie (tedy i její velikost a hmotnost), tím že dodává systému dodatečně elektrickou energii. Vzhledem k tomu, že spalovací motor není ve většině případů přímo spojen s koly vozidla, je možné jej provozovat v bodech (oblasti) s optimální účinností a spotřebou paliva. Tím, že spalovací motor bude pracovat pouze v jednom nebo dvou provozních bodech, lze také zkoumat nová a inovativní konstrukční řešení. Toto naopak vyžaduje vhodné návrhové nástroje, metody a přístupy, které umožní rychle a efektivně dosáhnout stanovených cílů. Disertační práce se zaměřuje také na tuto problematiku a předkládá možnou metodu navrhování s využitím analytických výpočtových modelů a parametrických modelovacích technik v CAD.

Abstract

The doctoral dissertation examines issues related to range extender units for hybrid vehicle powertrains. In particular, it is focused on the internal combustion engines utilised in these units and their design. Because battery electric vehicles still use heavy, bulky and expensive batteries, that provide a varying, and in some cases, insufficient driving range, the powertrains with range extender present a reasonable solution to these issues. The range extenders are auxiliary power units that convert the energy hidden in fuel into electric energy. They use a distinctive type of IC engine to drive an electric generator. The range extender also allows us to reduce the capacity of the battery, thus its size and weight, by supplying the system with electricity. Since in most cases, the internal combustion engine is not connected directly to the wheels of the vehicle, it can be run at points (operating range) with optimum efficiency and fuel consumption. Operating the IC engine at only one or two makes it possible to explore new and innovative design solutions. This in turn requires suitable design tools, methods, and approaches, which allow us to achieve the objectives quickly and effectively. The doctoral dissertation also focuses on this topic and presents a possible design method using analytical calculation models and parametric CAD modelling techniques.

Dedication

To my family.

Acknowledgment

The author warmly thanks his tutor **prof. Ing. Jan Macek, DrSc**. for his professional guidance, inspiring consultations and cooperation in the working over the Doctoral Dissertation. He also thanks his colleagues **Ing. Antonín Mikulec, Ing. Jiří Kubíček** and **Škoda Auto a.s.** for their support.

Contents

List of Used Abbreviations

APU	Auxiliary Power Unit
BEV	Battery electric vehicle
BDC	Bottom Dead Centre
BMEP	Brake mean effective pressure
BSFC	Brake specific fuel consumption
BSG	Belt-driven starter generator
CA	Crank angle
CAD	Computer-aided design
CAE	Computer-aided engineering
CAM	Computer-aided manufacturing
CI	Compression ignition
CMV	Conventional motor vehicle
CR	Compression ratio
EREV	Extended-range electric vehicle
EV	Electric vehicle
EM	Electric motor
FCEV	Fuel cell electric vehicles
FEA	Finite element method
FMEP	Friction mean effective pressure
HEV	Hybrid electric vehicle
HV	Hybrid vehicle
IC	Internal combustion
ICE	Internal combustion engine
IMEP	Indicated mean effective pressure
ISG	Integrated starter generator
MV	Motor vehicles
NA	Naturally aspirated
NEDC	New European Driving Cycle
NVH	Noise, vibration, and harshness
OHC	Overhead camshaft
OHV	Overhead valve
PHEV	Plug-in hybrid vehicles
R&D	Research and development
RE	Range extender
REEV	Range-extended electric vehicle
RICE	Reciprocating internal combustion engine
SI	Spark-ignition
SoC	State-of-charge
TDC	Top Dead Centre
WLTC	Worldwide harmonized Light vehicles Test Cycles
WLTP	Worldwide Harmonised Light Vehicle Test Procedure
(x)	Source of figures and graphs
[x]	Bibliography source

1. General Introduction and Overview

1.1. Current State of Contemporary Transport System

Transport has become a major sector of the national economy and plays an essential role in the contemporary world, which depends completely on it. In its traditional meaning, the current transport system relies on energy derived from fossil fuels. Without doubt, one of the most significant achievements of modern science is the invention and subsequent development and innovation of the internal combustion (IC) engine as a primary power source for all different kinds of vehicles. In particular, motor vehicles make possible the free movement of people and goods all around the world, and therefore contribute to the social development and economic growth of modern society. [27, 50]

The human population is growing at a fast rate, which is also being reflected in the growth of the number of passenger vehicles. According to the latest statistical data, there were more than 1,32 billion motor vehicles (passenger and commercial) in 2016 [53] worldwide, and another 92,8 million motor vehicles in 2019, 77,9 million in 2020, and 79,1 in 2021 [74] were newly built (e.g., 21,8 million in 2019, 17,1 million in 2020 in the EU). These numbers are still rising quickly with the continual process of world urbanisation and globalisation. On the other hand, this significant number of motor vehicles has a negative impact on the natural environment and human life and leads to many problems. The main reason is hidden in the high reliance of the present transport system on fossil fuels. The atmospheric pollution caused by emissions from burned petroleum products leads to poor air quality (air pollution), changes in the Earth’s climate, global warming or exhaustion of crude oil resources.

1.1.1. Energy Supply and Demand

Our planet is confronting the worldwide growing demand for energy and supplies. As it is generally known, the petroleum (crude oil) reserves are limited and they are drawn excessively. The future use of this type of energy source fully depends on the finding of new oil reserves, which is getting more and more difficult. As a result, the rate of new oil reserves is rising slowly.

Nowadays the global consumption of oil averages around 100 million barrels per day, and at the same time, around 1700 billion barrels of crude oil are proven as oil reserves. If the rates of oil consumption and oil extraction (the term "oil" includes petroleum or crude oil, other petroleum liquids, and biofuels) remain similar, the prediction of the period of global oil depletion is around 45 years. [12] However, global energy demand rises much faster than the rate of finding new oil reserves. The latest studies show that the amount of oil consumed by passenger transport is estimated at 61 % of total world transport energy consumption (2012). [64] Based on that fact, it is consequent that the reduction of oil consumption in passenger transport (i.e., improving the efficiency and fuel economy of vehicles) is crucial for the future of oil supply and achieving energy and environmental sustainability.

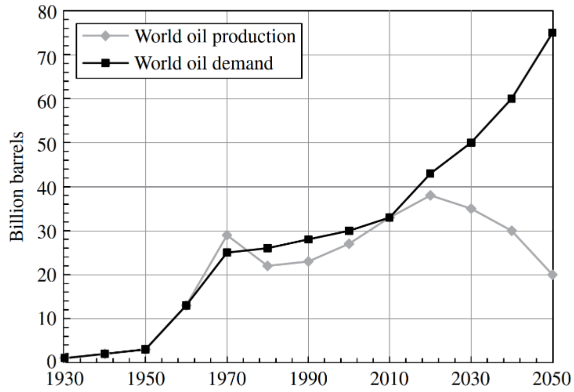


Fig. 1 World oil demand and depletion history and projections (1)

The conventional fuels used in transport are produced from petroleum, an energy source (fossil fuel) that emerged from the decomposition of living matter (plants and animals) deposited in geologically stable layers on the Earth in the past. Millions of years are necessary to complete the whole process of transformation and build up new deposits. That is why the available fossil fuel resources are limited and classified as a non-renewable energy source.

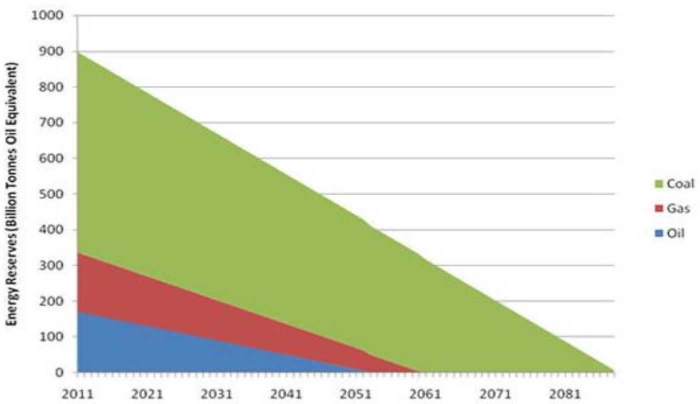


Fig. 2 Energy reserves (coal, gas and oil) – prediction in time (2)

1.1.2. Atmospheric Pollution and Global Climate Change

Global climate change is one of the biggest environmental issues that humanity is facing nowadays. The greenhouse effect, leading to global warming and extreme weather conditions, is mostly caused by increased concentrations of carbon dioxide (CO_2), methane, and other gases released by human activities and accumulated in the atmosphere. By high concentrations of greenhouse gases (GHG), solar radiation (sunlight) passes through the Earth's atmosphere. The infrared part of the radiation is reflected from the Earth's surface, but it is not allowed to leave the atmosphere. The greenhouse gases trap (absorb) the infrared radiation, and the heat energy that the radiation carries stays in the atmosphere and increases the temperature. In the long-term, a higher global average temperature of the Earth leads to many ecological issues, natural disasters, ecosystem unbalance, and the rising sea level.

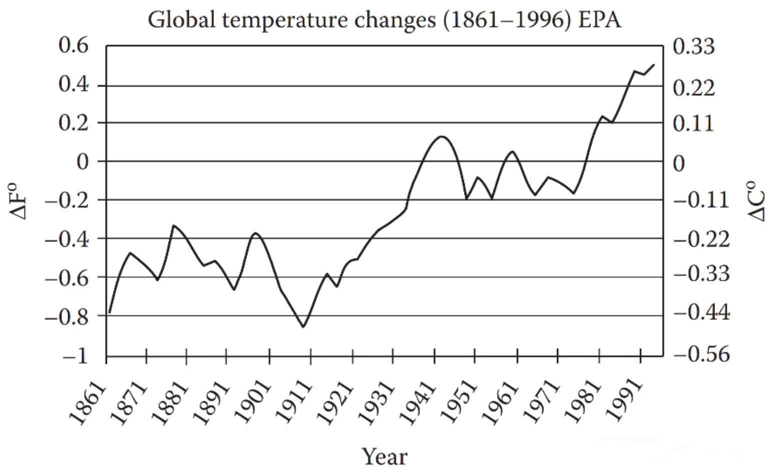


Fig. 3 Global Earth atmospheric temperature (3)

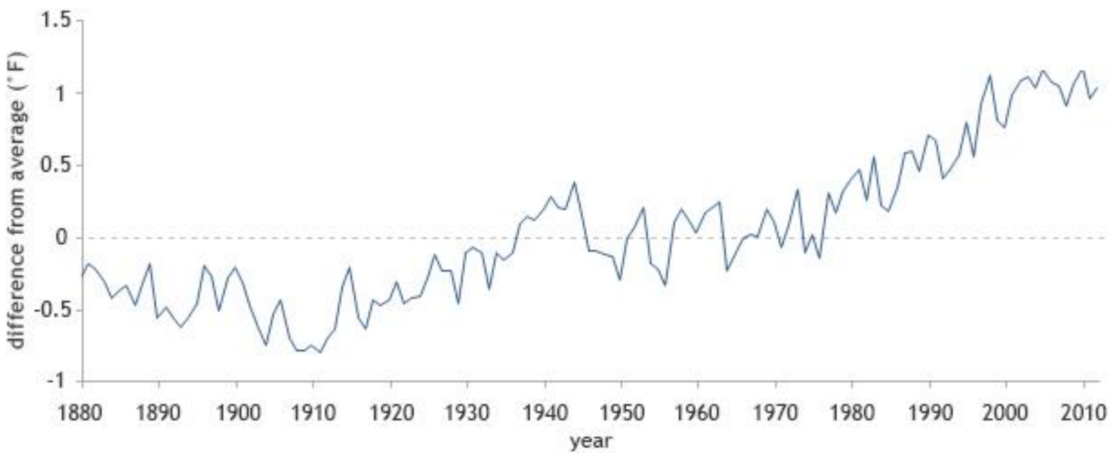


Fig. 4 Yearly surface temperature anomalies since 1880 (4)

Vehicle Emissions

Currently, all motor vehicles use a propelling system driven by chemical energy stored in hydrocarbon fuels. This energy is released from the fuel during the combustion process (an exothermic chemical reaction between the fuel and the air) inside the IC engine cylinder in the form of heat accompanied by burning products. The heat energy is transformed into useful mechanical work by the engine crank train, while the waste products are discharged into the environment. Hydrocarbons are organic chemical compounds composed of hydrogen and carbon atoms. The complete combustion process is an oxidation reaction of the hydrocarbons that produces only carbon dioxide (CO_2) and water (H_2O), which are not harmful to the environment (e.g., green plants absorb CO_2 during the photosyndissertation process). Motor vehicles equipped with an ICE (i.e., transport) are one of the significant human CO_2 emitters.

However, the combustion process of a hydrocarbon fuel inside the IC engine cylinder can never be perfect. That is why there are other “unwanted” products emitted through the burning of fossil fuels as well. In addition to carbon dioxide (CO_2) and water (H_2O), a specific amount of nitrogen oxides (NO and NO_2), carbon monoxide (CO), hydrocarbons from unburned vaporised fuel, sulphur dioxide (SO_2), and soot (particulate matter) can be present in the exhaust gases as products of the incomplete combustion process. All these emitted chemical substances are harmful to human and animal health and pollute the atmosphere.

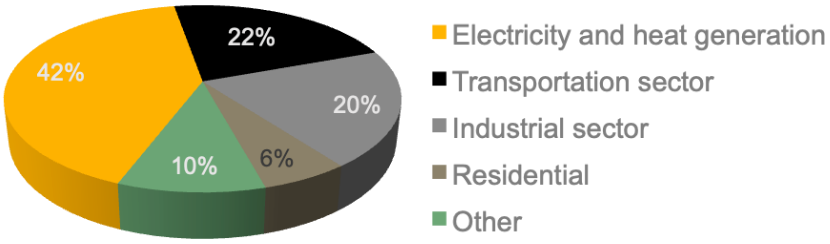


Fig. 5 Sources of CO_2 emissions from fossil fuels (5)

All current motor vehicles are equipped with advanced aftertreatment systems in order to minimise harmful exhaust emissions. However, to start operating effectively, the catalytic converter has to be warmed up by the hot exhaust gases at least to 300 °C after a cold engine start (usually one or two minutes). Based on this fact, around 80% of all emissions from the vehicle are produced during the cold engine start (for a considered driving cycle), as illustrated in Fig. 7. They are also inevitably a subject of further improvement.

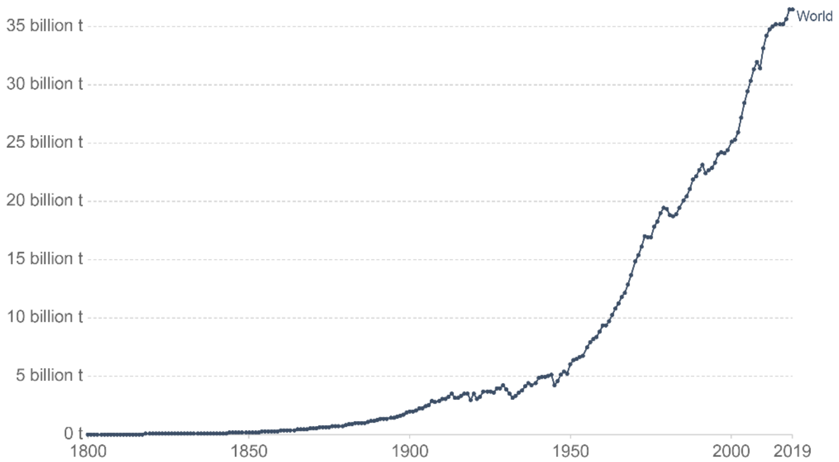


Fig. 6 Annual CO_2 emissions mainly from burning fossil fuels (1900 till present) (6)

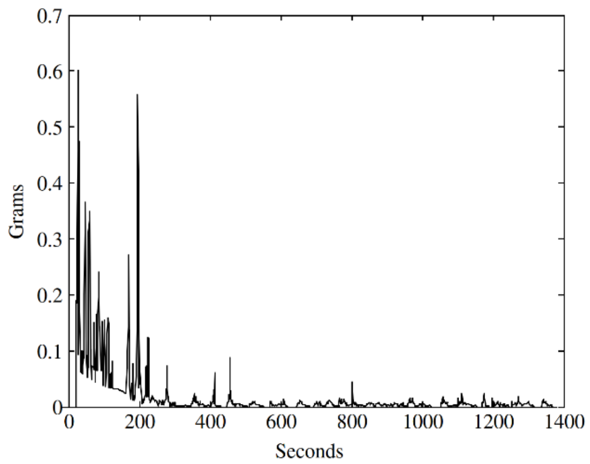


Fig. 7 Vehicle emissions during a cold start (7)

Assessment Tools for Vehicle Emissions [73]

It is appropriate to mention the need for a general method for comparing vehicle powertrain systems using specific measurement tools, for example, Well-to-Tank (WTT), Tank-to-Wheels (TTW), Well-to-Wheels (WTW), or Life Cycle Assessment (LCA). All these analyses are used for an accurate assessment and comparison of the GHG emission contributions of particular vehicle powertrain solutions and used power sources (fuels).

Fig. 8 Scheme of Well-to-Wheel analysis (8)

A significant benefit of switching to a more comprehensive measuring approach (WTW or LCA) over a simple exhaust CO2 measurement (TTW) is the ability to realise that there are different solutions available (electrification is not the only one) which can affect the effort and process of GHG emission reduction.

The WTW analysis evaluates the environmental impact of vehicle architectures by combining the values measured directly at the vehicle exhaust (TTW, on-board fuel usage) and those ones associated with the use of a specific energy source (fuel or/and electricity), i.e. CO2 emissions generated during production and supply (WTT - transport and filling the fuel tank/charging the electric battery). The figure below presents a WTW analysis of several different concepts of medium passenger vehicles (one car using different energy sources).

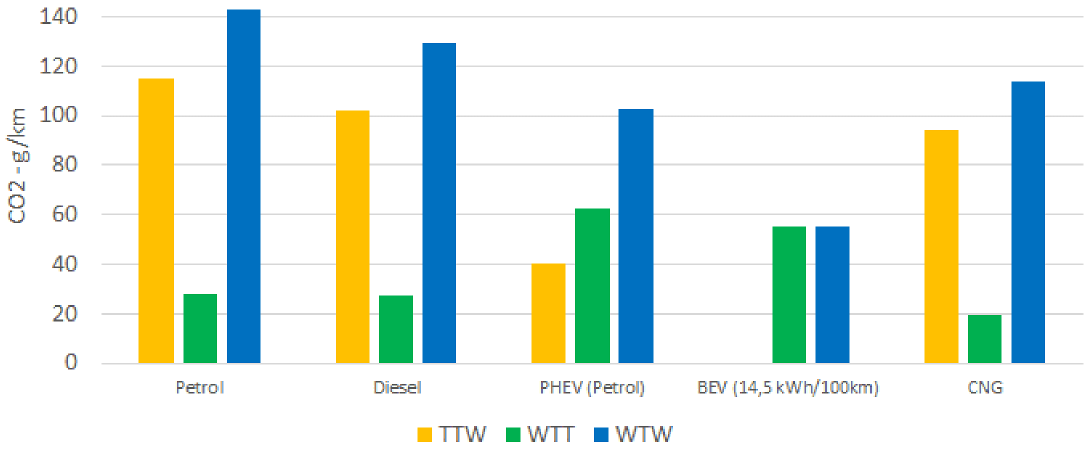


Fig. 9 Comparison of WTW emissions, WLTP, same vehicle (9)

As is obvious, BEVs offer the best performance in TTW analysis simply due to the lack of CO2 emissions produced directly by vehicles (no exhaust emissions; from this point of view, they seem zero-emission). This also leads to a better performance of BEVs in the WTW analysis compared with conventional fuels. However, the analysis shows their poor performance on a WTT basis, i.e., CO2 emissions are roughly two times greater (production and transmission of electricity considering the current EU energy mix of 106 g CO2/MJ).

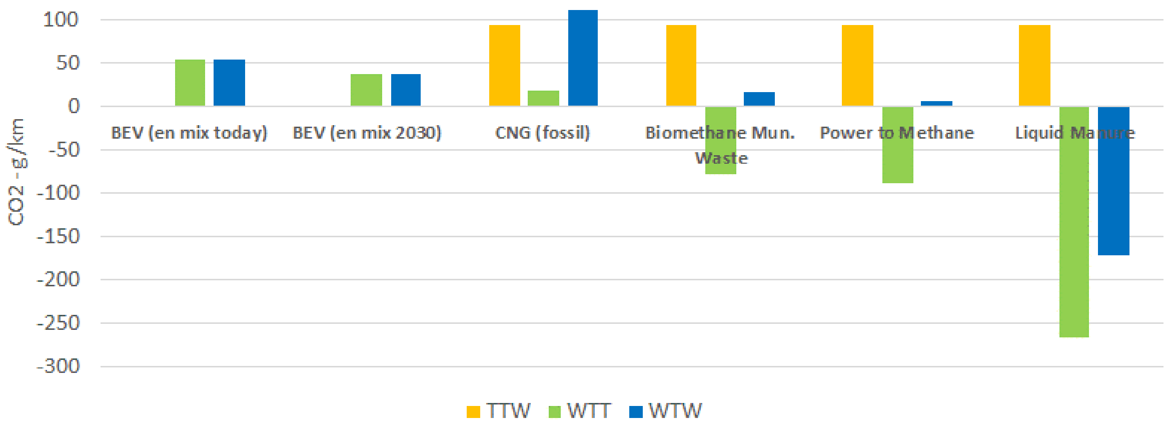


Fig. 10 Comparison of WTW emissions considering renewable energy sources, WLTP (10)

It is interesting to point out the effect of the use of natural and renewable gas fuels. Powering vehicles with natural gas can reduce the exhaust emissions by approximately 25% compared to conventional fuels (petrol, diesel). Renewable gas from biomass (biomethane) can reduce CO2 emissions even below BEV levels. The use of renewable power sources is able to make vehicles cleaner, so the transport system can continue to use IC engines in different powertrain concepts for propulsion.

A complex way to evaluate the environmental impact of transport is to analyse the whole life of vehicles. This involves the use of LCA methodology (Life Cycle Assessment or Analysis), also known as a Cradle-to-Grave (C2G) study. It includes measurement and evaluation of vehicle sustainability, performing a more complete assessment of the emissions arising not only from the use of the vehicle itself (WTW) but also within its whole lifetime, starting from the manufacturing process and ending with disposal and material recycling (end of life). According to statistics, the average lifespan of a personal passenger vehicle in the European Union is 10,5 years and 160 000 km (an average annual mileage of 15 000 km).

The conventional transport technologies (IC engines powertrains and drivetrains) use very similar processes from manufacturing to the end of the life of vehicles. The introduction of electrified solutions, on the other hand, requires a different approach because the manufacturing process of electric batteries is very energy and raw material intensive, and the subsequent recycling process is also very demanding (and needed due to the scarcity of rare materials such as lithium, cobalt, or nickel).

According to the latest studies, the largest amount of CO2 from electric vehicles occurs during their manufacturing, especially of the electric batteries (e.g., lithium-ion) – e.g., assuming an average value of 90 kg CO2/kWh, 4500 kg CO2 for a 50 kWh battery system.

For the same medium-sized passenger car considered above, the expected exhaust emissions produced by the vehicle (TTW) are within the limits as follows: petrol - 115 g/km, diesel - 102 g/km, hybrid electric vehicle (HEV) with petrol engine - 85 g/km, plug-in hybrid electric vehicle (PHEV) with petrol engine - 40 g/km and CNG - 94 g/km. For BEV, the energy consumption is 14,5 kWh/100 km, assuming the current EU energy mix of 106 g CO2/MJ.

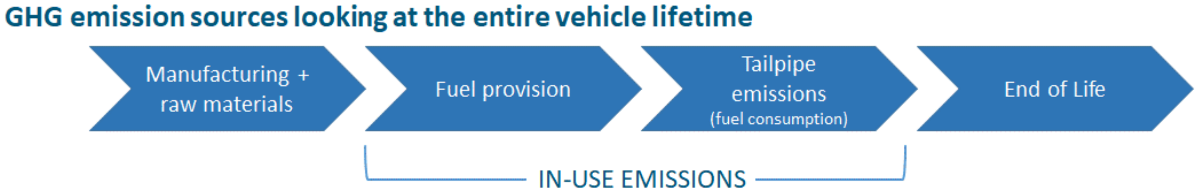


Fig. 11 GHG emission sources (steps) in Life Cycle Analysis (11)

The graph below shows results from such an analysis, including all emissions from the vehicle manufacturing process, fuel/energy production and supply (WTT), use of the fuel (TTW), and end of vehicle life. This analysis allows a complete comparison of the vehicles in terms of their emissions to be done. BEVs do not have any emissions because they do not burn fuel during propulsion (no TTW emissions - energy comes from the electric battery). However, this does not mean they are zero-emission vehicles. They look like zero-emission when considering exhaust emissions only, so switching to WTW and LCA will give a more accurate idea of the BEVs. For example, a life-cycle analysis shows that manufacturing such a vehicle results in nearly twice the emissions in comparison with conventional vehicles.

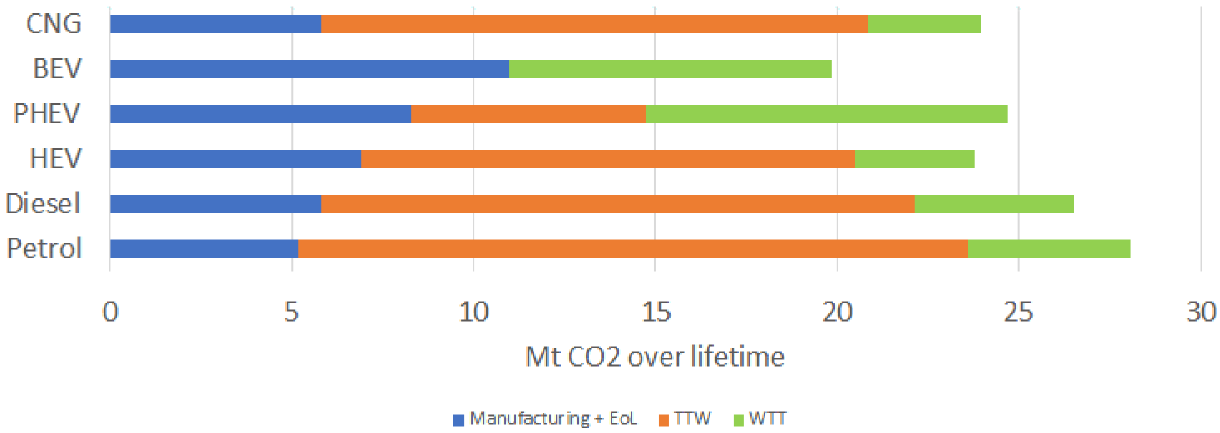


Fig. 12 Comparison of LCA GHG emissions (lifetime) (12)

The next chart (Fig. 13.) presents an extended comparison considering renewable energy careers and sources. The use of renewable gas fuels presents a simple and effective way to significant reduction the life-cycle emissions of vehicles (due to negative emissions).

The overall results from these particular studies are shown in Fig. 14. Some interesting observations follow. Considering the battery manufacturing footprint, PHEV is similar to HEV. If there is no need to replace the battery over the vehicle's entire life, BEVs can be expected to produce half the emissions of conventional vehicles (2030 energy mix). The emissions of bio-CNG (synthetic gas, gas from waste) vehicles can be compared with those of BEVs considering operation on electricity from wind or solar energy. In some cases, the use of biofuels appears to be a very effective way to produce negative GHG emissions (i.e., gas from liquid manure). Fossil CNG vehicles have similar emissions as petrol HEVs and PHEVs.

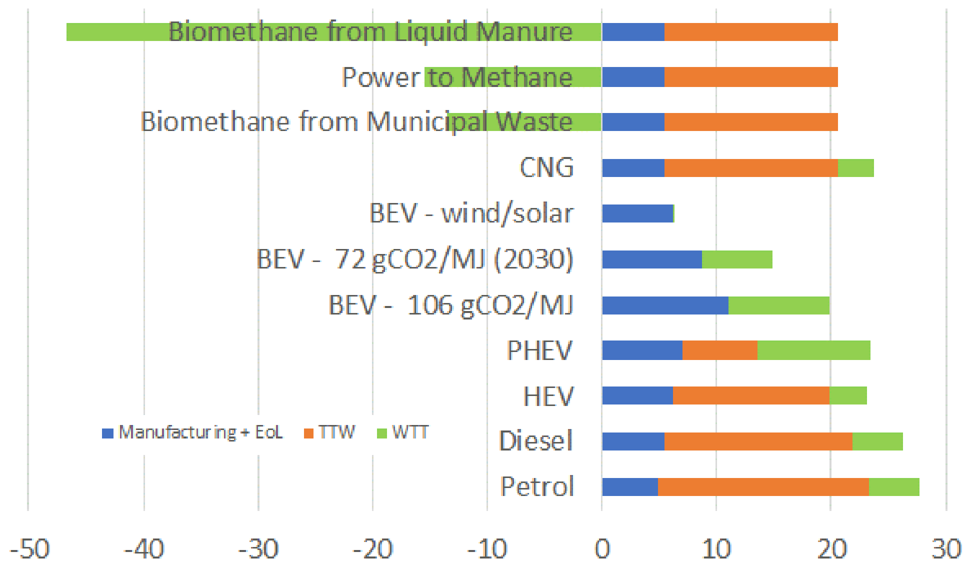


Fig. 13 Comparison of LCA GHG (lifetime) emissions considering renewable sources (13)

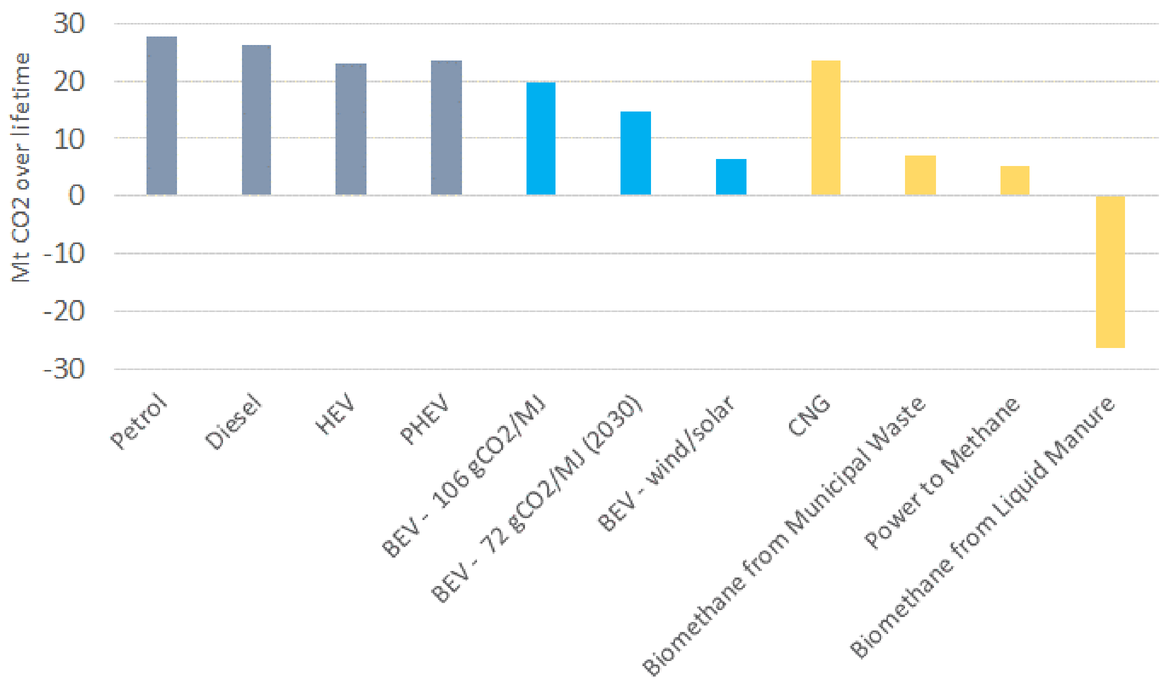


Fig. 14 Overall comparison of LCA GHG emissions (lifetime - 160 000 km) (14)

In conclusion, from this brief outline of the carbon footprint assessment of the vehicles, it is important to emphasise that this issue should be considered thoroughly from all different aspects, otherwise it can result in misleading conclusions (e.g., as it can be seen, a simple TTW analysis is simply not enough) and lead to the favouring of one solution over the others. Clearly, this kind of analysis should serve as a starting point in the creation of future transport decarbonisation plans, policies, legislation, and strategies for sustainability and mobility. [73]

1.1.3. New Strategy for Low-Emission Mobility

Nowadays, modern human society is striving for a sustainable future. Transport and mobility are essential for anticipated sustainability. However, in the view of a longer-term perspective, the current system for personal transport does not correspond to this idea. The reasons for that have been already mentioned, i.e. the total energy supply for the sector comes mainly (approx. 90%) from fossil fuels (limited source) and the harmful emissions from burning them. That is why a new, more favourable for economic development and more environmentally friendly transport model has to be developed and introduced. It is necessary to focus on efficient, safe, clean, and smart mobility and, at the same time, outsmart future obstacles like oil scarcity, traffic congestion, reduction of CO_2, pollutant emissions, etc.

It is considered that transport contributes nearly a quarter of all greenhouse gas emissions in Europe and they are one of the biggest pollutants. The data indicate that since 1990, the emissions from transport have not decreased gradually as has happened in the other segments. The first improvement in transport emissions appeared after 2007. However, today's values of emissions are still higher than those in 1990. It is known that road transport is one of the biggest greenhouse gas emitters (around 70% of all emissions from transport in Europe in 2014). In the EU, road transport produces around 21% of its total CO_2 emissions (2017).

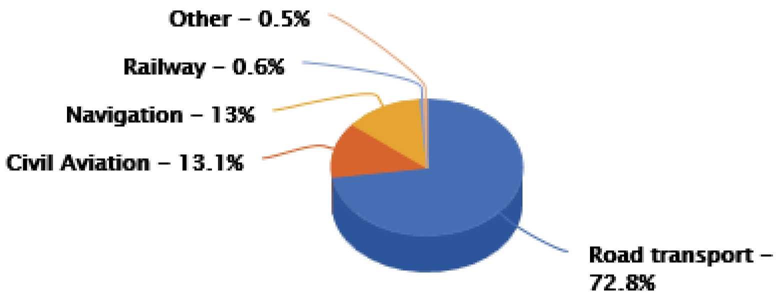


Fig. 15 Greenhouse gas emissions from transport in 2014 (15)

The new European strategy for low-emission mobility was approved in July 2016 in order to ensure Europe's competitiveness and responsiveness to the incessantly changing transport and mobility demands. The new EU legislation is an important step forward in the effort to successfully reduce GHG emissions and the total energy consumption. It is very important to develop new technologies to improve the fuel efficiency of conventional vehicles.

The European response to the ongoing competition to cut carbon emissions from transport is to switch to low-emission mobility. The first stated target is to reach a minimum 60% lower greenhouse gas emissions from transport by 2050 (in comparison with 1990). The new strategy aims to provide better air quality, lower levels of noise and less traffic congestion around Europe. Other awaited advantages are the introduction of highly efficient vehicles with lower energy demands, improved accessibility of alternative fuels, and better safety, etc. [63]

The first attempt aims to reduce the use of fossil fuels and carbon emissions through the use of clean and safe energy. In recent years, research and development in the automotive industry has been focused on efforts to improve transportation efficiency, safety, and cleanliness. Different solutions were proposed as alternatives or more likely as a supplement to conventional motor vehicles, such as hybrid and electric vehicles.

During the last few years, the vehicle research and development process has been oriented toward high-efficiency, clean, and safe transportation. Electric vehicles (EV), hybrid electric vehicles (HEV) and fuel cell vehicles are suggested as a replacement or more likely as a supplement to conventional motor vehicles in the foreseeable future.

In order to propose a sustainable transport model, it is important to overview the methods of producing energy and also the ways in which vehicles are powered. The energy available today comes from three different types of energy sources: non-renewable, renewable and nuclear. The non-renewable energy sources are based on fossil fuels, which are finite in nature (oil, natural gas, and coal). Renewable energy (usually electric) comes from alternative Earth's natural sources that are not exhaustible, such as water, solar, wind, geothermal, biomass, etc. For example, energy from biomass is placed among the renewable sources for its derivation from plant and animal materials such as wood, crops, waste from forests and landfills, cellulose, etc. However, there is a lot of nuclear power, this source is not renewable because of the limited natural resources of radioactive elements. Moreover, people have to answer the global question about the nuclear safety and treatment of nuclear waste.

The relations among different types of energy sources, energy carriers, and vehicles are graphically presented in Fig. 16. The common fuels (petrol, diesel, petroleum, and natural gas), which represent energy carriers with fossil origin, power the conventional types of motor vehicles. On the other hand, modern hybrid vehicles offer better efficiency and lower fuel consumption in comparison with the conventional ones, but they remain fossil-fuel dependent (primary energy source). That is why conventional vehicles, together with hybrid ones, are not adequate solutions for future sustainable mobility. Electric vehicles (EVs) and fuel cell electric vehicles (FCEVs) are respectively powered by electricity and hydrogen, which represent secondary forms of energy and can be produced from a different range of renewable and non-renewable primary energy sources. In case only renewable energy sources are used, both types of vehicles could be sustainable. In contrast, plug-in hybrid vehicles (PHEVs) are still not completely sustainable. However, they offer the advantages of conventional vehicles and EVs at once. Plug-in hybrid vehicles (PHEVs) can suppress the use of fossil fuels by using only energy from the electric grid. This type of vehicle is not a perfect solution for sustainable transport, but these vehicles present an intermediate step on the way to the next level stage.

1.1.4. New Regulations for Vehicle Emissions

On September 1st, 2017, the new Worldwide Harmonised Light Vehicle Test Procedure (WLTP) was introduced. It marked the start of a new era in automotive engineering. The WLTP is a new laboratory test procedure for measuring fuel consumption, CO_2 and gas pollutant emissions from passenger cars. It replaces the previous test procedure, the New European Driving Cycle (NEDC), which was introduced in the 1980s. In recent years, automotive technology and driving conditions have substantially changed, and the NEDC has become insufficient. This led to the development of a new test procedure that is designed to obtain more accurate test results and better reflect real-world driving conditions and vehicle behaviour. The test conditions of the NEDC are based on a theoretical driving profile, unlike those of the WLTP cycle, which are based on real driving data and better represent current driving conditions. A change in regulations could not be introduced overnight, it has been implemented gradually in successive steps. A comparison of driving cycles is shown in Fig. 17 and 18. [21]

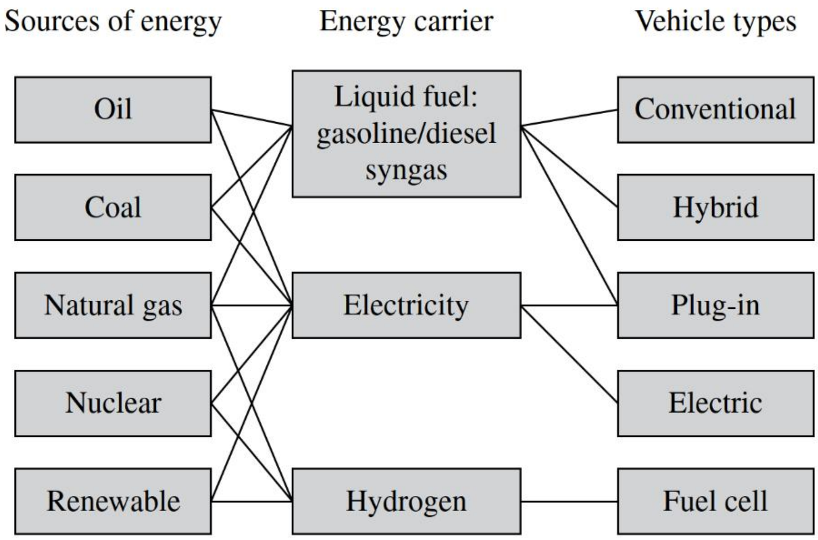


Fig. 16 Relations among different types of energy sources, energy carriers, and vehicles (16)

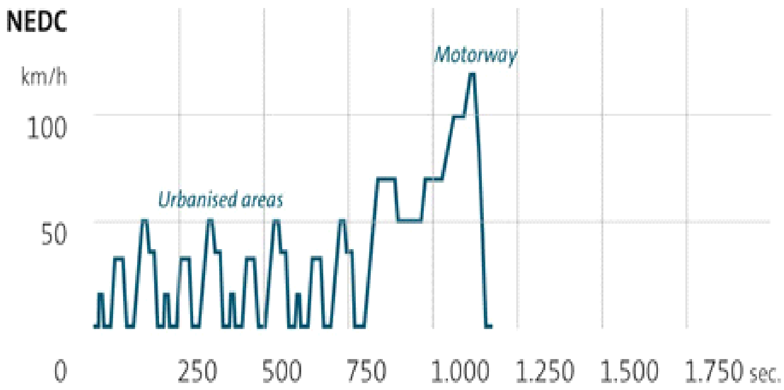


Fig. 17 The NEDC driving cycle profile (17)

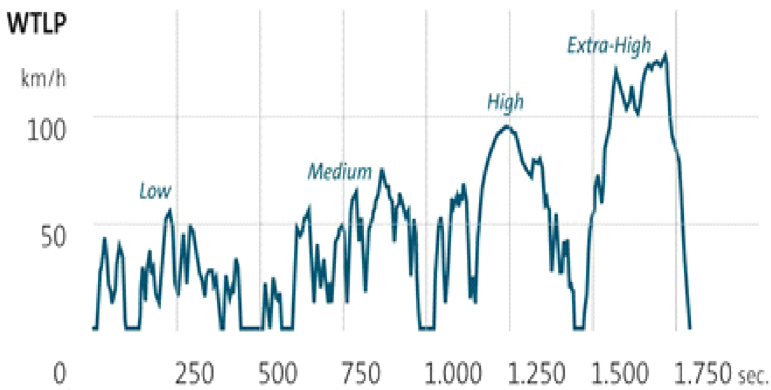


Fig. 18 The WLTP driving cycle profile (18)

1.2. Summary and Global Aims of the Dissertation

The internal combustion engine is the most commonly used power source in motor vehicles, and the situation probably will not change significantly in the near future. However, further measures are needed to increase the overall efficiency and fulfil new, stricter and more demanding emission standards.

It is essential to push forward the research process of engine and powertrain systems and to continue the search, design, and development of new solutions for a wide range of applications ready to achieve new goals. An important step is the development of innovative, functional, and reliable technologies for engine aftertreatment, fuel systems, air/fuel mixture (cylinder charge) formation, etc. The development process has to be flexible and adapt quickly to continuously changing requirements and demands. This can be achieved by the utilisation of the experience and knowledge gained in the design and development of IC engines in combination with a wide range of advanced technologies, available resources and development capabilities, as well as advanced and efficient computer-aided design approaches and methods based on analysis and simulation. This will help to develop new innovative IC engines and powertrain concepts for the future, such as alternative fuels and hybrid powertrains.

The continual process of electrification of vehicle powertrain systems leads to the implementation of different hybrid-electric technologies. In modern hybrid electric vehicles (HEVs), which are becoming more and more popular, IC engines are also the main option for a primary power source. In these vehicles, the IC engine is joined together with an electric generator, an electric battery, and an electric motor. The utilisation of the IC engine in automotive hybrid systems requires a different manner of design, control, and operation in comparison with that of conventional motor vehicles (MVs). The IC engine can run in a high-efficiency range for a longer time period (steady-state) and usually there is no need for a fast (transient) change of operating mode. Generally, these IC engines are not designed and developed separately for this application. They are often based on conventional IC engine series. However, it is desirable to continue with research and development in order to achieve an improvement in the overall efficiency. All effort is focused on the transition to fully EVs. In any case, vehicle electrification still presents a huge technical challenge with a lot of questions and unknowns, and the future course of this process remains, for now, uncertain. [27]

2. Current State of Knowledge

2.1. Electrification and Hybridisation as Alternative to Conventional ICE-powered Motor Vehicles

2.1.1. Some Facts from the History

At some point in the 19^{th} century, electrically driven vehicles appeared for the first time. It is not possible to give merit on this occasion to a specific inventor or country. The idea of electric propulsion has arisen after a series of discoveries and inventions – from electricity, through electric batteries, to electric motors. The first electric-powered vehicles were trialled successfully in the second half of the 19^{th} century. At that time, electricity was not the only one new source of energy. Alongside the strength of horses, which was still remaining the main source, steam engines and IC engines powered by fossil fuels were already introduced.

At that time when, the Industrial Revolution was in full swing, people strived to improve the options for transport and, as a result, many different kinds of motor vehicles were invented. Steam engines had already been proven as a reliable source of power for the industry, rail and maritime transport. Naturally, there were also steam-powered vehicles. However, this kind of engine was not very practical for passenger vehicles, mainly because of the frequent need for water and coal refilling and the long time needed to get them into operation. On the other hand, the first internal combustion engines were also not without flaws. They were very loud, the exhaust gases smelled unpleasant, and the vehicles at that time were hard to operate (hand starting with a crank, rough gear changing, etc.).

In contrast, electric-powered vehicles looked very promising and soon became popular. They did not suffer from any of the above-mentioned disadvantages of the heat engines. They were very quiet, easy to drive, and did not emit exhaust gases.

Vehicle technology developed and improved very fast. However, electric vehicles, which disappeared in the 1930s, were based on a few facts. They were more expensive in comparison with the vehicles with an IC engine, the electric grid was sparse and electricity was not available outside the big cities. In addition, large discoveries of petroleum reserves around the world led to petrol price reductions and changed petrol availability.

After that, electric vehicles appeared once in a while during the 20^{th} century on different occasions like petrol shortages or restrictions, technology improvement, etc. However, their application was again very limited (low speed, limited range).

Nowadays, EVs are appearing again and becoming popular for similar reasons to which they were popular more than a century ago, but also for many others. This trend started at the beginning of the 1990s when questions about ecology and environmental protection became an everyday occurrence. The collective effort results in various restrictions and regulations on all human activities, including transport, forcing sustainable development.

It is obvious that widely enforced electromobility has plenty of potential to affect in a positive way the human effort to achieve environmental sustainability. Nevertheless, electrified vehicles are still in their childhood, so it is difficult to predict precisely their future evolution. It is clear that only new technologies (like better battery technologies) and time will show the future position of electrified vehicles. [48]

2.1.2. Modern Electrified (Electric and Hybrid) Vehicles

2.1.2.1. Definition and Classification of Electrified Vehicles

It is appropriate to start with a brief overview of some essential terms and definitions of electrified vehicles, such as pure electric (EV) and hybrid electric vehicles (HEV), as well as their powertrain architecture and hybrid concepts. The terms vehicle electrification and vehicle hybridisation are used for a wide range of technologies for propelling vehicles with electrical energy or with more than one energy source. [30]

The most common conventional motor vehicles (MV or CMV) powered by an internal combustion engine (ICE) rely on a highly energy-dense petroleum-based (fossil) fuel as an energy carrier. It offers excellent performance together with a great operating range (mileage or travelling distance). Despite this fact, the CMV has, from today's point of view, some significant weaknesses and disadvantages, such as insufficient fuel economy and a not negligible environmental impact from harmful exhaust emissions. [27]

In contrast, electrified propulsion systems are supported by electrical energy coming from electrochemical or electrostatic energy sources (electric storage batteries). They use a minimum of one electric machine as a traction motor for full or partial propulsion of the vehicle. The most common alternative type of vehicle, i.e., electric vehicles, offers some advantages over the CMV. In particular, there is higher energy efficiency and a lower environmental impact (measured on a local scale, see 1.1.2). In comparison with CMV, the overall performance of the EVs is much more limited, especially the operating range, time needed for energy renewal (battery recharge), overall weight and cost. The reason for this is hidden in the significantly lower energy density of bulky and heavy electric batteries. [27]

Hybrid electric vehicles (HEV) combine both the mentioned power sources (petroleum fuel and electric batteries) and take advantage of both CMV (high energy and power density) and EV (zero local emissions) while at the same time resolving their disadvantages.

It is important to take a look at the essential and fundamental ideas and definition of hybrid vehicles (HVs), their powertrain architecture, how they work, and how they differ from conventional vehicles because the principles are important for subsequent research. Here, the term “engine” refers to all kinds of IC engines (petrol, diesel); the term “motor” refers to the electric motor (EM) or electric motor/generator (M/G).

A hybrid vehicle (HV) is one that has more than one powertrain and uses a combination of two different types of energy sources for propulsion (no more than two, due to the rising complexity). Different feasible concepts are available, but the most common are solutions using IC engines and electric motors or fuel cells and electric motors. According to the particular powertrain architecture, both power sources can supply one or two independent propulsion systems (e.g., series vs. parallel HV).

An alternative definition of HV is based on operating modes. The HV operates at least in two different propulsion modes (pure electric or pure ICE mode). Otherwise, it is not a HV (e.g., diesel-electric powertrain without an electric energy storage system). A specific case, in harmony with the definition of a hybrid vehicle (HV), is the hybrid electric vehicle (HEV). It joins together the components of the ICE-powered CMV and the pure EV.

It is essential for any type of vehicle powertrain to produce adequate power to comply with the vehicle’s needs in terms of performance, efficiency, pollutants, driving range, etc. Because hybrid powertrains offer far better flexibility and allow operation with high efficiency, significant overall improvement can be achieved here by proper control and adaptation to special operating conditions. It is important for the overall vehicle’s performance to run the powertrain in high-efficiency areas. [30]

2.1.2.2. Electric Propulsion Systems

A pure electric vehicle (EV, battery electric vehicle – BEV, or all-electric vehicle – AEV) is a vehicle powered completely by electrical energy. A scheme of a modern electric powertrain is shown in Fig. 19. The EV propulsion system is usually composed of the following subsystems:

– electrical energy storage system (energy source – electric battery, power electronics and energy management system);

– electric motor drive (electric motor, transmission);

– auxiliary devices (steering, heating, etc.) and

– control unit.

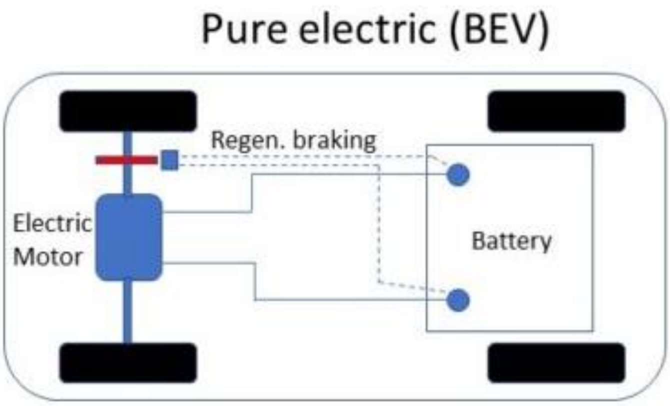


Fig. 19 A scheme of a modern electric powertrain (19)

BEVs face different problems, such as significant costs (development, materials, and production), limited driving range, long charging time and sparse charging infrastructure. The battery needs to be recharged by the electrical grid, which takes a much longer time in comparison with the standard time for fuel tank refilling. The real driving autonomy of these vehicles is insufficient (up to 400 km) due to the significantly lower energy density of the electric battery (around 200 Wh/kg) and their dependency on weather and driving conditions. [34]

Recent EVs have mostly been built on the basis of existing conventional ICE-powered vehicles. Making such a transition conversion, i.e., replacing the conventional powertrain (ICE, transmission, fuel system) with an electric one (EM, battery system, control system), while keeping all other systems the same, we obtain an EV which is weighty, less flexible and weakly performing (e.g., Fiat 500e, 2013 or Volkswagen e-Golf, 2015). In contrast, the most current EVs are specially designed with their own unique body and powertrain structure, making the electric propulsion more flexible (e.g., Volkswagen ID.3, 2019 or Fiat 500e, 2020). [27]

2.1.2.3. Hybrid Electric Propulsion Systems

Hybrid electric vehicles (HEV) provide a reasonable solution to some of the EV problems by combining an IC engine with an electric drive system to propel the vehicle. HEVs still run on fossil fuels, but without the possibility to recharge the battery externally. Joining together both powertrain systems into one makes it possible to utilise the best characteristics of both – the high energy and power density from ICE-powered vehicles and zero local emissions from electric vehicles. They can achieve considerably lower fuel consumption than conventional vehicles by multiple measures. The most common of them are:

– downsizing the IC engine without affecting the maximum power supply;

– recuperation of energy during vehicle braking;

– optimal possible power distribution between both systems and

– stop/start mode of the IC engine in order to minimise fuel consumption.

On the other hand, the newly emerged hybrid powertrain is heavier than IC engine-powered vehicles (up to 30%), which can have the opposite effect. [34]

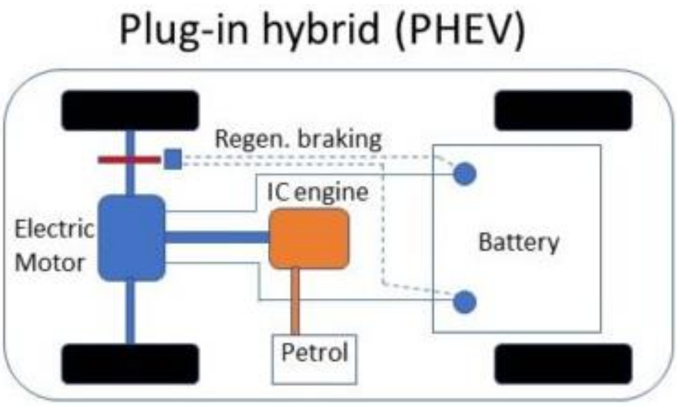


Fig. 20 A scheme of a modern electric powertrain (20)

A **plug-in hybrid electric vehicle (PHEV)** is a vehicle similar to an HEV, but it is capable of recharging the battery from the power grid. Furthermore, HEVs typically have a larger battery in combination with a more powerful electric motor and can operate in an all-electric mode (charge depletion CD) with a limited range (around 60 km), which is ideal for daily driving. When the battery is completely depleted, the vehicle runs in the same way as a standard HEV (charge-sustain (CS) mode or extended-range mode). [50]

2.1.2.4. Powertrain Architecture of Hybrid Electric Vehicles

Generally, the HEV architecture is specified as a relationship between all powertrain components. A brief description **of the powertrain systems of hybrid electric vehicles** based on their design (architecture) reveals they can be sorted into four separate groups:

– **series**,

– **parallel,**

– **series-parallel** and

– **complex hybrid**.

This classification is not academically very accurate and can sometimes be confusing. [27]

Series Hybrid Electric Vehicle (S-HEV)

What are the benefits of the series HEV architecture? The concept of the series hybrid electric powertrain has evolved from that of pure electric vehicles. In comparison with the CMV, the EV benefits from higher energy efficiency and zero local emissions, but it suffers from a limited operation range due to the low energy-dense, large volume, and heavy electric storage batteries and a long recharging time. So, the original idea of creating the series HEV was to extend the travel range of a pure EV by involving an engine-generator (an auxiliary power unit, or APU) to recharge the batteries on the road. The engine’s power output is converted into electrical energy in order to recharge the battery or drive the EM directly. In this case, the ICE output shaft, as a source of mechanical energy, is connected mechanically only to an EG.

Therefore, the IC engine's operation is independent of the vehicle's power requirements (actual vehicle speed and traction power demand), i.e. the engine can run under optimum conditions (speed and throttle position) in order to maximise efficiency and to achieve the lowest possible fuel consumption. Furthermore, this approach simplifies the control of the IC engine and makes it easier to reduce the vehicle’s exhaust emissions. Since the whole unit containing the IC engine and the generator is not connected to the vehicle’s driveshaft (mechanically decoupled), eliminating the need for a mechanical clutch and transmission, it can be installed anywhere in the vehicle. The vehicle is purely electric-driven, and it is propelled by an electric traction motor, powered by two electric power sources – the battery and/or the engine-generator unit. The traction electric motor can operate in both motoring and generating mode in order to drive the vehicle and to regenerate some energy back to the electric battery during braking. The powertrain system consists of three main parts: an IC engine, an electric generator, and an electric traction motor, just as in a pure EV. The last one needs to be dimensioned to cover the vehicle's maximum power demand, i.e. to deliver adequate power for sufficient vehicle performance, because the performance of the S-HEV is fully dependent on the parameters of the traction electric motor.

On the other hand, the series hybrid electric powertrain also brings some issues. The energy, coming from the IC engine, required to drive the wheels of the vehicle has to be transformed two times. All mechanical energy is transformed to electric energy (in a generator), which is subsequently converted back to mechanical energy (a traction electric motor), and every single conversion involves relevant losses. So the total losses in the series powertrain depend on the efficiency of all machines (IC engine, generator, motor, power electronics, etc.).

Conversion of energy: Mechanical → Electrical → Mechanical

All mechanical power is converted to electric and then back to mechanical. Each conversion has associated losses.

Energy flow: IC engine → El. generator → Electrical motor → Differential / Wheel

The electric machine, which serves as a MG, is relatively large and weighty, but it benefits from the high power. The power capacity of the engine-generator (APU) together with the battery power capacity corresponds to the total power demand of the vehicle. [27]

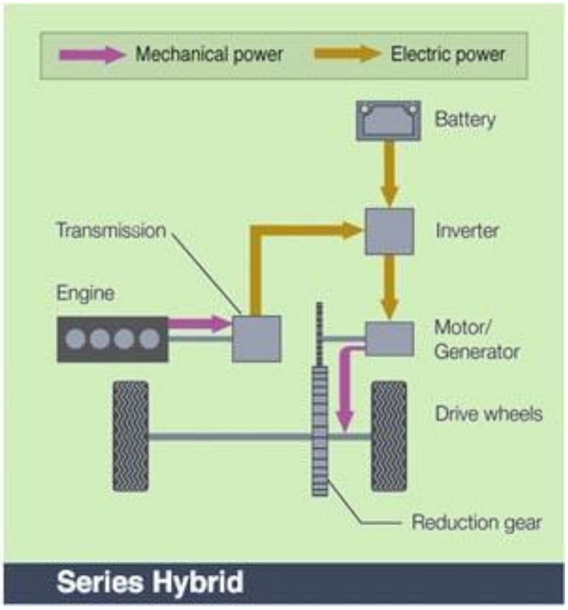


Fig. 21 Energy flow diagram of series hybrid system (21)

Parallel Hybrid Electric Vehicle (P-HEV)

Unlike series hybrids, which look more like an EV equipped with an engine-generator, parallel hybrids look more like an IC engine-powered conventional vehicle assisted by an electrical part or equipment (electric motor). A parallel hybrid drivetrain is one in which the engine supplies its mechanical power directly to the drive wheels in a manner similar to a conventional

IC engine vehicle. In P-HEVs, both the IC engine and the electric motor are directly mechanically connected (mechanical coupling) to the wheels, and both can drive the vehicle either individually or together at the same time. Moreover, the link between the engine part and the electric part is mechanical. This gives more freedom to optimise the distribution of the traction power between two separate parallel lines in order to satisfy the vehicle's power requirements. However, the system needs a complex mechanical transmission. The IC engine can directly power the wheels and can be assisted by the electric motor when high power is needed (accelerations). This allows us to divide the amount of maximum power needed by the vehicle between both machines during the design process.

A traction motor is an electric machine that can operate as a motor to drive the vehicle or as a generator in order to charge the battery by regenerative braking or directly through the IC engine. It is more compact and lighter in comparison with the series layout. Moreover, an additional electric generator is not required. The vehicles are able to operate in a pure electric mode that is essential for enhancing the overall fuel efficiency, mainly in city stop-and-go traffic. A significant advantage of the parallel arrangement is the single transformation of energy for both lines, and as a result, energy losses may be reduced:

IC engine mechanical energy → Drive shaft mechanical energy

EM electrical energy → Drive shaft mechanical energy.

However, because the IC engine is mechanically connected to the wheels, it is not possible to optimise its operation and run it in a few operation points, so the IC engine should be controlled in a similar way as in a conventional vehicle. Furthermore, the engine throttle setting varies, causing difficulties in controlling emissions. Varying engine throttle setting causes engine operation at higher specific fuel consumption. To a certain extent, that leads to a certain overall system complexity and control demands.

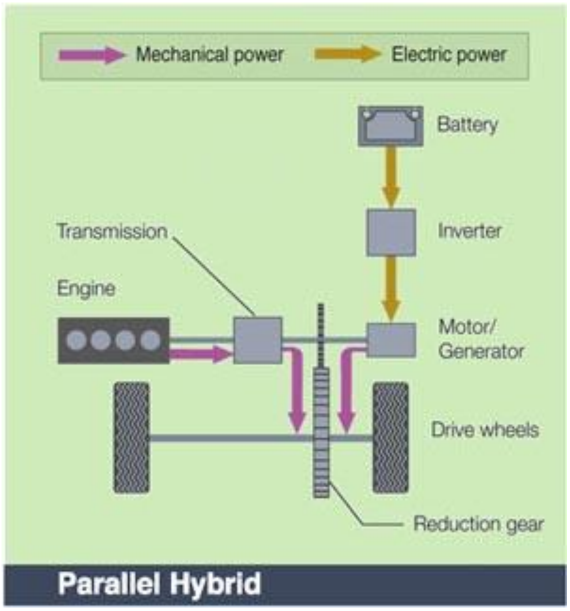


Fig. 22 Energy flow diagram of parallel hybrid system (22)

The higher weight of the vehicle can have a negative effect on the overall efficiency, but all applied measures make it better in comparison with the ICE-powered vehicle. The mechanical connection with the engine's driving shaft limits the installation location of the engine. [34]

Series-Parallel (Mixed or Combined) and Complex Hybrid Vehicles

The hybrids from this group merge features from both series and parallel hybrids. They are similar to the P-HEVs but offer some features of the S-HEVs. They use two separate electric machines with both mechanical and electrical connections. One of them operates as a traction motor with an option for regenerative braking (as in parallel layouts), whereas the other one operates as an electric generator used for recharging the battery by the IC engine (as in series layouts). A power split device makes possible operation in different modes: an IC engine-only mode, an electric mode, battery recharging, regenerative braking, and power assist modes. These hybrid systems are much more complex and more expensive. Of course, there are many different hybrid proposals and concepts that do not belong to any of these categories, and they are usually classified as **complex hybrids**. [34]

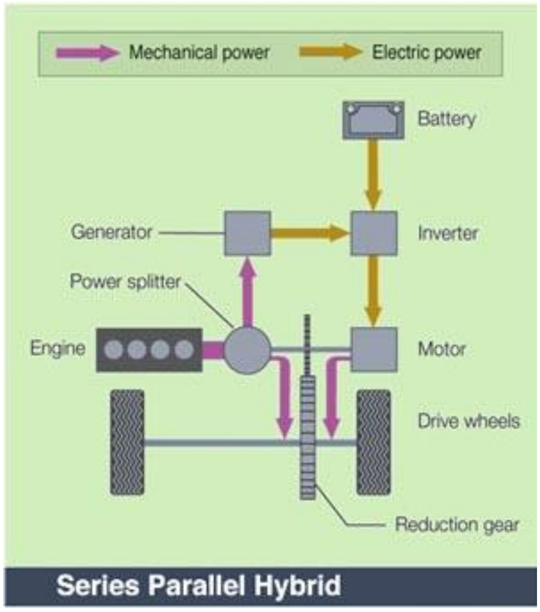


Fig. 23 Energy flow diagram of series-parallel hybrid system (23)

Selection of Hybrid Layout

With the development of hybrid technologies, pure series or parallel layouts are rarely used alone as they do not often fulfil the requirements. Today, the most commercial hybrid vehicles use a mixed series-parallel architecture, which is more flexible. The selection of a specific hybrid layout depends on the vehicle's type, its purpose, and the resulting requirements. [30].

2.1.2.5. Functional Classification of Hybrid Vehicles

Hybrid systems can also be classified according to their degree of hybridisation, independently of their physical architecture. The degree of hybridisation (level of hybridisation or hybridness) is a parameter (usually an independent variable) that characterises the overall nature of the hybrid systems regardless of the physical structure. It is defined as the ratio of the rated electric power (one or more electric machines) to the total power of the vehicle.

$$DoH = \frac{Rated\ el.power}{Rated\ el.power + Max.ICE\ power}$$

Using this parameter, the following categories of hybrid vehicles can be defined.

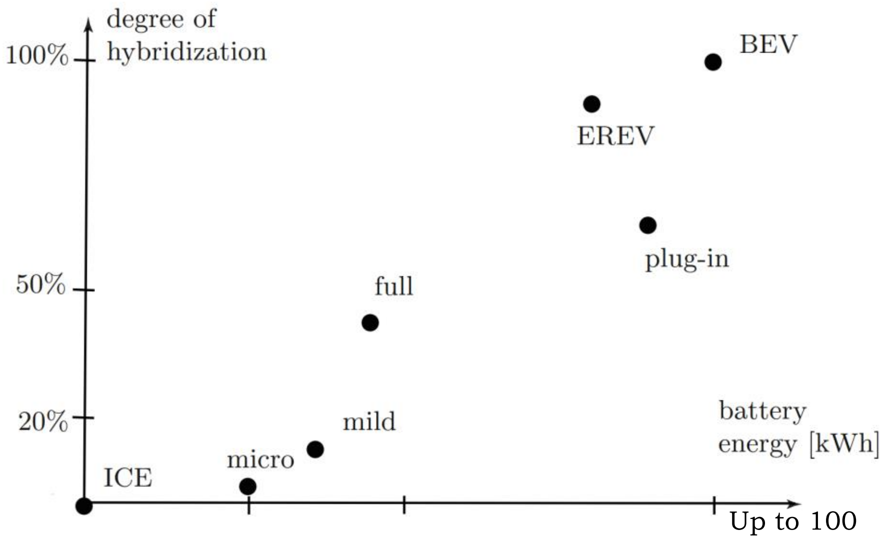


Fig. 24 Classification of HEV according to the DoH and battery capacity (typical values) (24)

Micro hybrid is the most basic hybrid concept (parallel). It presents a conventional ICE-powered powertrain supported by a small electric motor. It uses the standard electric system with a small capacity battery. In the simplest case, the electric motor is a modified starter motor, which supports the automatic stop-start ICE operation. Another solution uses a modified electric alternator, which also maintains the ICE stop-start operation. This solution is termed as an **integrated starter generator (ISG)** or **belt-driven starter generator (BSG).**

Mild hybrids are more advanced hybrid systems where the electric part (e.g., ISG/BSG) is used to assist the IC engine during moving off, acceleration, or when power is needed, and to recuperate energy during braking. However, they do not offer an entirely standalone pure electric driving mode.

Full hybrids feature all possible operating modes: power assistance, energy recuperation, and entirely electric operation. For that aim, they are equipped with a high-voltage system, which provides a much higher level of electric power. With a higher level of hybridisation, the reduction in energy consumption becomes more significant, but the extra cost of the hybrid solution also rises.

Plug-in hybrids have a larger-capacity battery storage system, so they can operate in pure electric mode with a range that is comparable to BEVs. The batteries of plug-in hybrids can be recharged externally from the electric power grid. Extended-range electric vehicles are recharged from the grid. For longer trips, they utilised the REx unit to extend the range. [34]

2.1.3. Issues of Hybrid Powertrain Units with Range Extender

2.1.3.1. Definition of Range Extender, Reasons for Introduction, Concept Description

In the beginning, it is essential to provide a solid definition of the range extender and its concepts, as well as to specify the individual components and some of their features.

As it has already been mentioned, these days the interest in alternative fuel vehicles is growing rapidly. In particular, battery electric vehicles (BEVs), which also belong to this category, are becoming again more and more popular for different reasons. Since the very beginning, more than a century ago, the available technology, manufacturing processes, and human knowledge have changed significantly. Thanks to these prerequisites today, it is possible to develop and produce far better and more reliable electric-powered vehicles with significant improvements in performance, efficiency, and travel (driving) range.

Anyway, the range autonomy of pure EVs (i.e. how far they can drive) is still one of the major disadvantages and obstacles to great success, together with the higher price. BEVs have a limited range, i.e. they can travel less distance in comparison with conventional vehicles (on one fuel tank). It does not depend only on the vehicle's speed, trip distance, track profile, weather conditions, etc., but also on the usage of heating and climate control, lighting, and any other vehicle equipment on-board powered by the electric battery. In addition to the insufficient strength of the battery systems, the electric infrastructure for charging EVs is very sparse and needs a lot of improvement. The next reasonable step should be oriented toward a refinement of the battery technologies to lower the price, reduce the weight, and increase the capacity. However, this is a long-term process requiring further research and time.

An interesting and more accessible solution to the issue of outsmarting the limited driving range can be achieved by a discreet modification of the pure battery-electric powertrain. The electric vehicle is supplemented additionally with an auxiliary power unit (APU, a secondary power source), commonly called a Range Extender (REx), which is used exactly to extend the reachable travel range to a more acceptable limit by supplying the battery system with electricity on the road. This solution allows reducing the total capacity of the electric battery to the minimum needed for a specific use (e.g., urban operation) and thus reducing the price and weight of an electric vehicle. The remaining energy needs (e.g., occasional extra-urban operation) of the vehicle are covered by the range extender. Thus, the newly emerged concept of an electric vehicle with an extended travel range represents a distinctive type of hybrid powertrain. The main difference compared to the conventional types of hybrid vehicles is the primary energy source, which here is the electric battery, and the propulsion is purely electric.

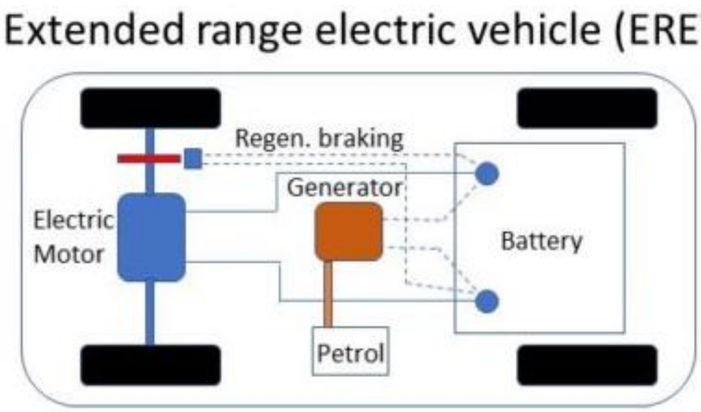


Fig. 25 A scheme of a modern electric powertrain (25)

This auxiliary power unit is usually labelled as a Ranger Extender (REx). It consists of a secondary fuel-based power source (usually a small heat engine as a fuel convertor) that drives a mechanically connected electric machine (operating as an electric generator – EG) in order to supply (charge) the battery system of the vehicle and the electric motor with electricity. The unit transforms the chemical energy from the fuel, which is released as heat energy during combustion, into mechanical energy that is afterwards transformed into electrical energy by the generator. There is a variety of different types of thermal engines, but the present doctoral dissertation is focused on reciprocating internal combustion engines. Since there is no mechanical link between the ICE and wheels, it can be installed anywhere in the vehicle. [34]

2.1.3.2. Description of Operating Principles of EV Powertrain with REx

The vehicle architecture (layout) is very similar to that one of the series and plug-in hybrids with all its advantages and disadvantages. Now, it is clear the role of the range extender unit is to increase the EV travel range to a more acceptable level when it is necessary. It runs only when the battery has reached a certain critical level of charge, and then it helps the vehicle to get to a charging station for recharging the battery. However, in most situations, this device serves only as a safety measure to eliminate the range anxiety (i.e. to stay broken down on the road with a flat battery). This kind of hybrid vehicle can be found under several other names: extended-range electric vehicle (EREV), range-extended electric vehicle (REEV) or battery electric vehicle with a range extender (BEVx).

In fact, the BEVx is an electric vehicle, but at the same time, it is also a hybrid electric vehicle based on the series powertrain layout. The primary power source (primary mover) is the traction electric motor and electric battery, whereas the secondary is the auxiliary power unit (APU) – the range extender, which means the pair of IC engine-electric generator. If there is enough energy stored in the battery system, the BEVx operates as a pure electric vehicle. The APU is then activated in a situation when the battery is almost flat (depleted). It is reasonable to operate the vehicle mostly in the electric mode, which requires a regular battery recharge from an external source (e.g., a charging station or wall power plug) at a time when the vehicle is not in use (at night, during work day, etc.). The battery system must be designed to provide power for daily driving (a range of ca. 50 km), but at a price as possible.

According to the latest data, the daily travel distance in European countries averages from 40 km (in the UK) to 80 km (in Poland or Spain). This yields a suitable BEVx configuration – a small vehicle with a small storage battery (allowing lowering the cost, as well as the weight) with a capacity sufficient to cover daily driving in pure electric mode. The APU, a backup energy source, will be involved in case of emergency to reach a charging station or to ensure occasional longer journeys. It is clear that it presents an excessive weight at the most of the time. That is why the REx unit should be small, light, and affordable. Most of the time the vehicle runs in a pure electric mode and the operation is very quiet and comfortable. That is why a significant emphasis is also placed on the noise and vibration properties. [34, 52, 64].

2.1.3.3. General Requirements for Range Extender Unit

A competitive range extender unit for an electric vehicle should meet all requirements and demands. One of the common issues related to BEVs is the travel range, i.e. how far it is possible to drive without recharging and the related anxiety of staying on the road with a flat battery. This is a much more complex issue than it appears because it is not just about driving. The electric energy from the storage covers all the vehicle needs – lighting, heating, cooling, on-board electric equipment, etc. As a result, the vehicle range is limited and varies significantly. The use of a range extender unit with an IC engine, which allows us to extend the range of autonomy almost independently of the battery, is a promising solution. The affordable small battery electric vehicles on the market are suitable for urban and suburban operations. The range extender unit can significantly affect the range of vehicles. A competitive range extender engine for an electric vehicle should meet all special requirements and demands.

***Specific Requirements and Characteristics for Range Extender Unit*:**

• **size and dimensions** – small and compact, allowing optimal installation in electric vehicle; mostly single and twin configurations, but also three-cylinders; inline, V-type, flat; displacement up to 1,0 litre;

• **weight** – as low as possible, usually around 50 to 60 kg, since most of the time it is only an unnecessary weight. Depending on the EREV concept, the vehicle is equipped with a REx to extend its range without changing the battery system, or the capacity of the battery is reduced and a REx unit is installed, when the total weight (smaller battery + REx unit including cooling and controlling system) should be lower that of the initial BEV battery;

• **cost** – low as possible, for development and production – up to 1000 Euro per unit; simple and proven technologies;

• **output performance** – in general limited – narrow speed range, but optimal for REx application, around 20 kW to 35 kW at a constant speed to ensure battery recharge and travel range extension; (maximum output performance is provided by the electric drive only); easy start end stop;

• **efficiency** – as high as possible; optimised with regard to required power, low fuel consumption and emissions;

• **emissions** – depending on the concept, the REx is not expected to operate frequently in start-and-stop conditions (during real-world or legislated driving cycles). Therefore, proper thermal management of the engine and exhaust system is needed to control the emissions;

• **superior NVH characteristics** – quiet operation, low vibrations (balance of inertia forces), noise insulation and dampening;

• **optional equipment** – selected at a client's request. [29, 31, 39, 60, 61]

2.1.3.4. Power Requirements for Range Extender Unit

The REx unit is usually not connected directly to the wheels, i.e. there is no mechanical link. It is only used to generate electricity for recharging. The operating conditions and power demands for the ICE differ from those of conventional powertrains. There is no need to deal with part-load and transient operating regimes. The operation is completely independent of wheels and vehicle speed and can be designed and tweaked for better efficiency. Of course, it also depends on the characteristics of the electric equipment (optimum at a similar point to the engine). Actually, the REx unit should only generate the average power needed to drive the vehicle. A simulation and analysis of the driving dynamics (driving cycles, track profile, etc.) of the complete vehicle and its subsystems will identify the power and size of the REx unit. The peak power of the REx unit is determined according to the required maximum speed. Fig. 26 shows the relationship between engine power and vehicle speed for various vehicle types. At higher speeds, the parallel powertrain offers better efficiency and needs less power in comparison with the series hybrid (e.g., 26 kW vs. 35 kW at 130 km/h). [23]

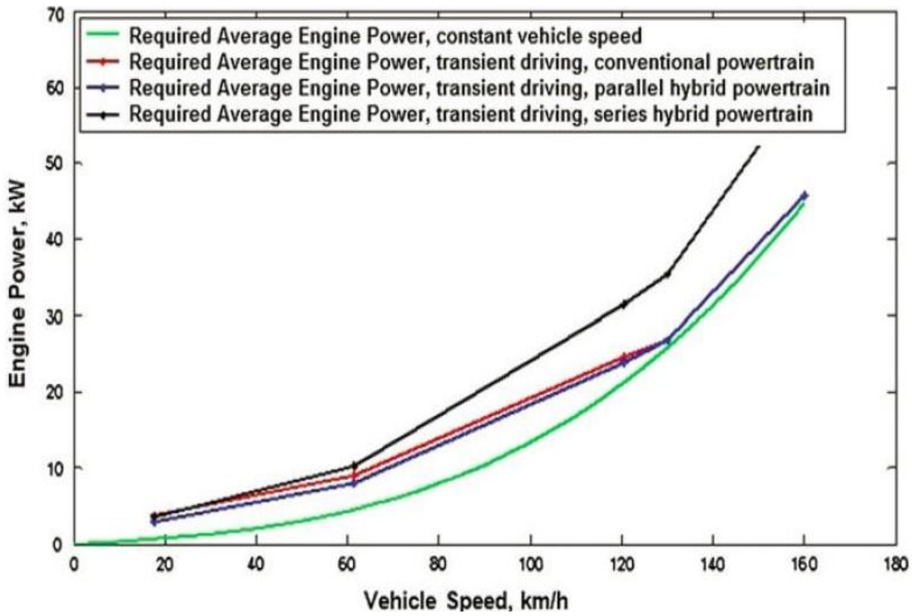


Fig. 26 Relationship between the engine power and vehicle speed for various vehicle types (26)

2.1.3.5. Size of IC Engine for Range Extender Unit

The displacement volume of the reciprocating IC engine utilised in the REx unit can be found in different ways, which are discussed further. It can be done by an analysis of the relation of the brake power to the displacement. For example, for steady motorway driving, when the speed is 4000 min^{-1} to generate 30 kW of power, the displacement is 800 cm^3. [23, 34, 45].

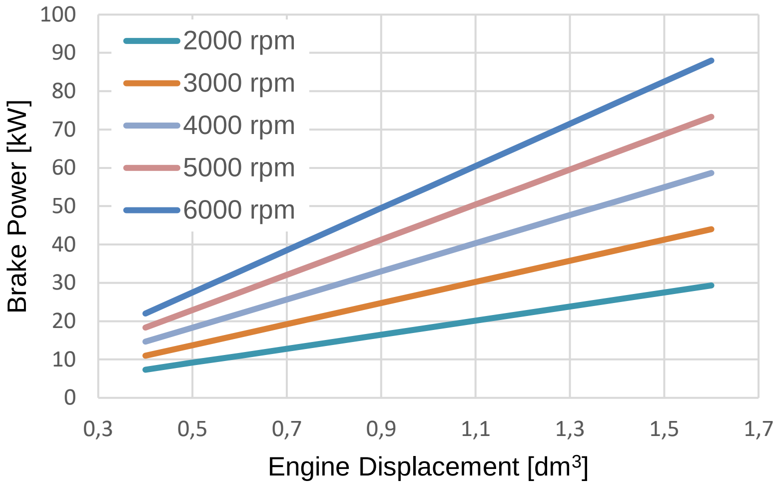


Fig. 27 Brake power related to displacement for constant BMEP (27)

2.1.3.6. Range Extender Engine Balancing

Engine balance is a very important characteristic since there is a strong emphasis on the NVH of the REx unit for EV. For a specific engine, its balance depends on the number of cylinders and their configuration. They, in turn, are selected with respect to the engine displacement and have a significant effect on the engine's characteristics – especially overall dimensions and package volume, weight, costs, vibrations (NVH), etc.

Various studies and assessments of different possible engine concepts have been carried out in order to evaluate their advantages and applicability to these specific operating requirements and to suggest the most suitable one. A wide range of different engine concepts and operating cycles have been considered. Since the engine is intended for mass production, the main focus is on the reciprocating piston internal combustion engine, even though other concepts, such as rotary engines and micro-turbines, have characteristics that are advantageous for a range extender unit. The spark-ignition four-stroke engine has been selected as a primary solution since it provides some certainty in terms of complexity, efficiency, NVH, pollutant emissions, aftertreatment and final costs.

For example, the company MAHLE Powertrain Ltd. has conducted a study of various engine concepts, such as inline-twin, V-twin, and flat-twin (boxer) engines, and compared them with the basis three-cylinder engine. Using parametric CAD models, the engine configurations have been compared in terms of package volume and size, weight, and cost. Using different benchmarks and assessment methods, the inline two-cylinder configuration has been selected as the optimum, due to the smallest package volume, weight, and production costs. [46, 77]

There are several different configurations of the inline twin-cylinder engine. An important factor in lowering the production costs is the use of a simple cranktrain design. This raises the idea that this can be achieved by eliminating the need for additional balancing. However, the standard 360° and 270° inline-twin configurations use one or two balance shafts to compensate for the primary forces. The balance shafts involve additional weight, design complexity, friction loss, costs, and noise (usually driven by gears). This leads in some applications to the use of different engine configurations, e.g., the 90° V-twin, which benefits from the absence of first-order imbalance.

On the other hand, the engine has a second cylinder head and a complex valvetrain and needs more space. An alternative solution is the utilisation of a 180° engine configuration, which has no primary force imbalance, but there is a contra-rotating primary moment (as in an inline three-cylinder). Usually, this could be a problem, but in this case, the range extender unit with the engine can be mounted in a way that allows free movement about the centre of gravity within specific limits. The second option is the use of partial balancing using an external imbalance of the flywheel and pulley, which is used, for example, in many three-cylinder engines. However, the twin-cylinder engine suffers from vibrations more than the three-cylinders. Nevertheless, it benefits from the lower weight, size, and complexity. More information can be found in some current scientific outputs. [38]

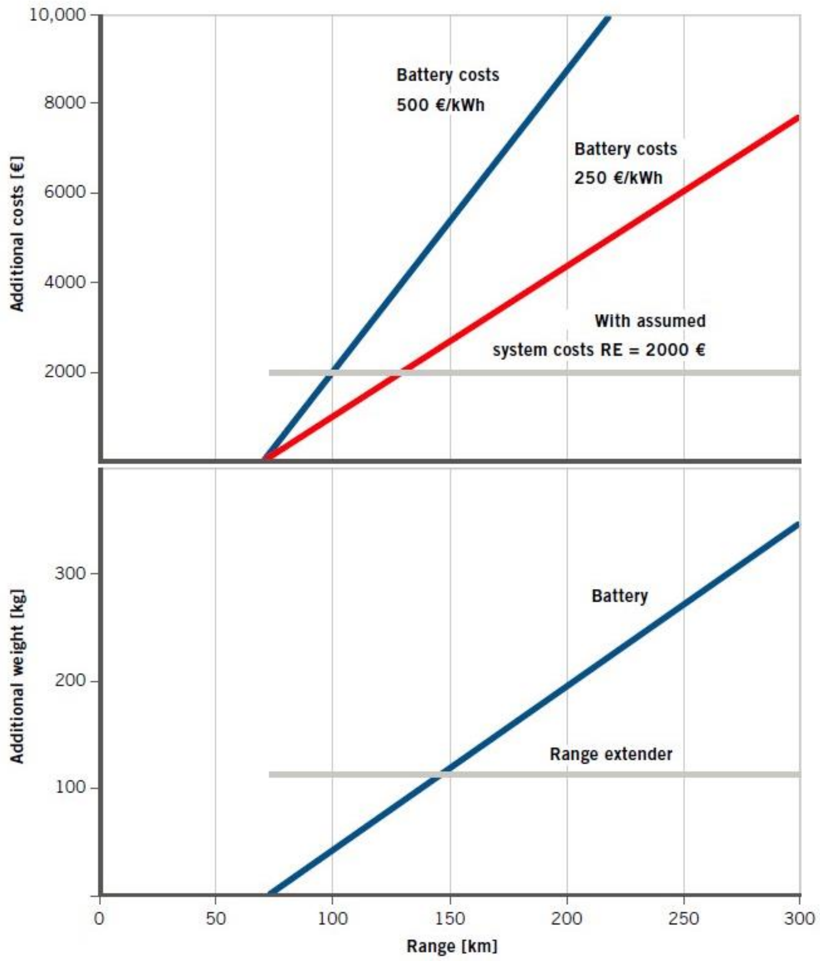


Fig. 28 Price and weight comparison of automobile concepts EREV and EV (28)

2.1.3.7. Economical Overview

Various research studies are estimating the cost, performance, and weight of the BEV in comparison with the EREV. In the example below, a pure electric range of 75 km is assumed. For a range of 100 km (battery costs 500 Euro/kWh) or 130 km (battery costs 250 Euro/kWh), the REx unit is very favourable, since the costs remain constant, whereas the costs for batteries grow linearly. The same can be observed in the comparison of weights, where the intersection point shows when the REx starts to be more favourable (150 km range). [23, 34].

2.1.3.8. Operating of the Range Extender Unit

Choosing the operating mode for the REx unit can be done in two ways. The **first option** is to run the engine in the optimum area with the best efficiency and the lowest fuel consumption. If there is enough energy or power demand is low, the engine is turned off completely.

The engine runs at a constant speed, which is a disadvantage in terms of acoustic comfort. Moreover, the engine sound cannot be masked by the wheel and aerodynamic noise at low vehicle speeds (increasing with speed).

Using this method may result in a high demand for engine cooling. The engine can operate under high loads and engine speeds even if the vehicle speed is low. This can cause cooling issues because there is not enough airflow to keep the engine cool. Installing an additional cooling fan will reduce the system’s efficiency. [23]

The **second approach** involves a dependency of the engine speed on the vehicle speed, i.e., the engine speed is controlled at several levels according to actual power demand and vehicle speed. The high losses in energy storage are minimised and the airflow is sufficient for cooling the engine. The engine runs at high speeds at higher vehicle speeds, which somehow masks the noise during travel. Maximum power is limited by the acceptable level of noise. [23]

- **NVH of Range Extender Unit.** The noise produced by the movement of the vehicle (aerodynamic, wheels and suspension, etc.) is comparable for both vehicle types. However, the electric powertrain generates much less noise in comparison with conventional vehicles equipped with an IC engine and gearbox, and the noise is masked on a global scale. This involves a requirement for the quietness of the REx unit because noise from the engine can be disturbing. In general, the NVH characteristics of the REx unit, the system of installation and mounting, noise insulation, etc. are very important in order to limit noise distribution.

- **ICE Noise and Vibration.** In general, the most noticeable noise from the IC engine comes from its intake and exhaust systems (the sound of the combustion process is not so distinct). The main focus is on the design of those systems by optimising the length the cross-section of the pipelines. The vibrations from the engine (frequency range up to 1 kHz) are perceived by the steering wheel and seats. They are caused by the acting gas pressure forces and emerging inertial forces and moments and their imbalance. The inertial forces depend on the square of engine speed and dominate approx. after 3000 min^{-1}. Multi-cylinder engines (four and above) are advantageous in terms of overall behaviour. But they are unsuitable for this application, where small and low-cylinder engines are preferred, which has a negative effect on the vibration level (which increases as the number of cylinders decreases). [31]

• **Principles of Controlling.** Depending on the particular approach to the control of the hybrid system, at least one or more operation points of the IC engine have to be determined with regard to the required power, optimum engine speed, and efficiency. For example, the efficiency curves of the range extender engine from MAHLE are presented in Fig. 29 There is one operating point in quite a wide optimum efficiency area. So, the exact placing of the operating points in the engine map involves a complex analysis of all engine characteristics (torque, NVH, emissions, etc.). [57]

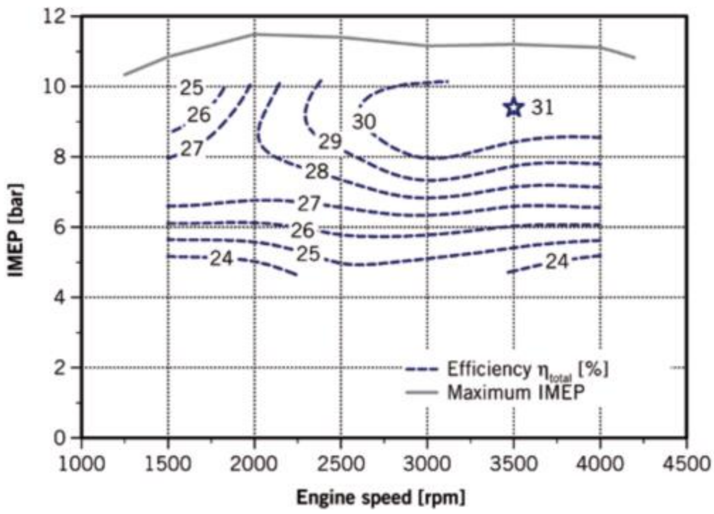


Fig. 29 MAHLE's engine efficiency (29)

Control According to State-of-Charge

The state-of-charge (SoC) parameter is an important variable that is used for the energy management of the vehicle. It expresses the amount of energy remaining in the vehicle's batteries. It is defined as the ratio between the remaining energy E_{remain} contained in the battery and the maximum energy E_{total} when the battery is fully charged.

$$SoC = \frac{E_{remain}}{E_{total}} \ [\%]$$

In order to achieve the highest possible operating efficiency, it is desirable to use as much energy as possible from the vehicle's battery (charge-depleting). Once the SoC reaches a certain limit, the range extender is put into operation (charge-sustaining). Usually, it is not possible to consume all the energy from the battery. That is why it is necessary to keep a certain amount of energy in the batteries as a backup (e.g., 15 - 20 %). It is used to cover peak power demands that exceed the maximum power of the internal combustion engine. This presents the most common SoC-driven control strategy: the battery is discharged to this energy level, then the ICE starts charging to a specific higher level and stops, and the cycle repeats.

More advanced control strategies include analysis of driving conditions (vehicle speed, power demand, energy consumption), data from digital maps (road profile, motorway driving, hill climbing, etc.), or traffic in order to predict the need for recharging (see next).

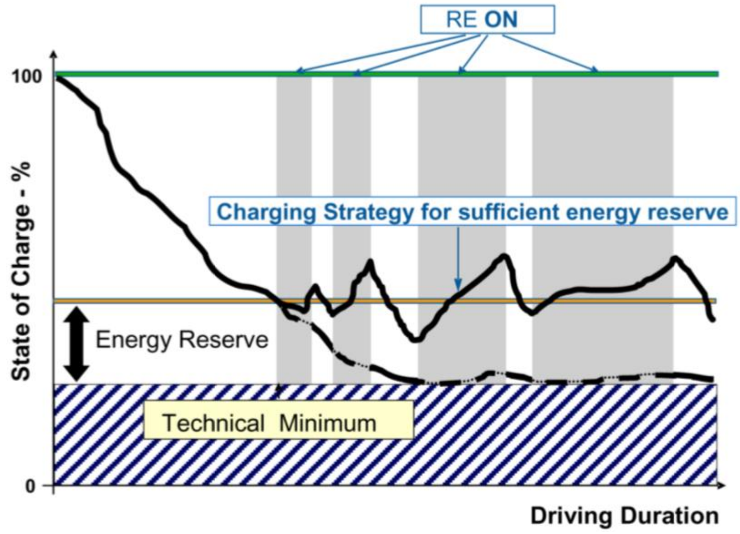


Fig. 30 Course of battery SoC and range extender operation (30)

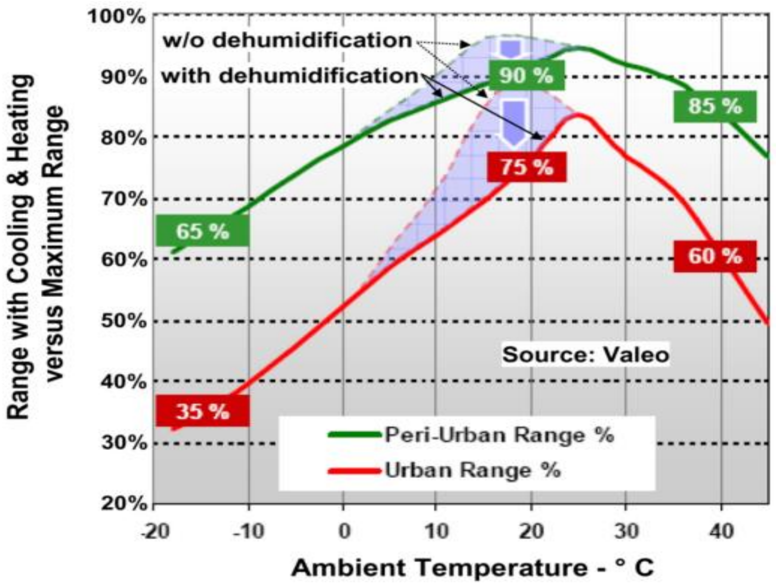


Fig. 31 Range according to the temperature of the environment urban and extra-urban driving (31)

Control According to the Route Profile

Another way of controlling and planning energy efficiency consumption is to use route profile data from detailed maps and GPS data. After entering the final destination into the navigation system and calculating the optimal route, the vehicle control unit can optimise the use of both power sources. The aim is to minimise the total energy consumption, mainly by full use of the battery energy with support from the IC engine. The task gets more complicated if a specific route is not selected in advance. The hybrid control unit must continuously calculate and predict different possible routes according to the current position of the vehicle in order to plan the energy sources. Other factors aimed to consider and predict are the driving style, vehicle load, usage of vehicle electrical equipment (lighting, heating, cooling, entertainment, etc.), the ambient temperature and weather conditions (impact on the battery state), traffic conditions, etc. These complex real-time calculations are essential in the effort to optimise the operation of the hybrid system and to minimise the need to involve the IC engine.

The graph shows the vehicle electric range of the car as a percentage of the maximum theoretical value as a function of the ambient temperature. It can be seen that the range can be limited to 35% of the maximum value in urban traffic at low temperatures. [57]

• Selecting Range Extender and Battery Parameters

Depending on the BEVx concept, different design options are available in terms of selecting the size of the battery (capacity) and IC engine (power) for the REx unit. In every case, it is all about a trade-off between the specific requirements and useful properties for cost, travel range, weight, energy consumption, charging and refilling time, etc. Since the IC engine is not used directly for vehicle propulsion, which is ensured by the electric motor, it should not be designed to deliver high power. As a result, a small, lightweight, low-cost engine (up to three-cylinders, up to 1,0 L) can be used. The size of the fuel tank depends on the required additional range.

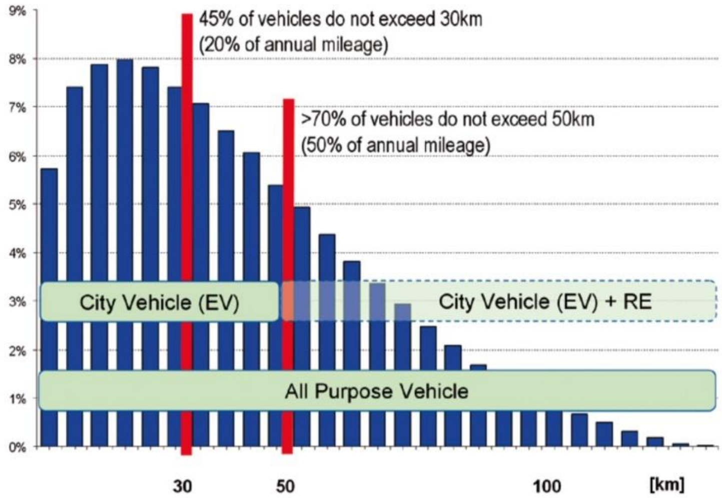


Fig. 32 Average daily distance driven in Germany (32)

The chart shows that the average daily range of 70% of vehicles does not exceed 50 km. It can also be seen that about 85% of vehicles travel up to 80 km per day. [29]

One common concept in the design of hybrid vehicles is to optimise the capacity of the battery to cover, in a pure electric mode, a certain average driving range. The vehicle will operate as an ordinary EV, requiring a regular recharge of the battery from the power grid. The engine will be utilised in the case of long-distance trips.

In general, the production cost depends on the desired vehicle range. Requiring a higher capacity of the battery to extend the electric range linearly increases the price and the weight of the vehicle. In contrast, using a REx unit, it is possible to increase the range at a reasonable constant price. To provide a large travel range, the vehicle should be equipped with a heavy, high capacity battery, which increases the cost and weight. That is why the concept of smaller batteries is also popular. [57] Therefore, it is advantageous to have a relatively low electric range, enough to cover daily needs, and to add additional range by using a range extender. The energy density of fossil fuels is significantly higher than that of electric batteries. For example, the energy density of petrol (42,5 MJ/kg) is up to 100 times higher than that of a lithium-ion battery. [55]

So, the right choice of the operating range and points for the range extender engine is a trade-off of different parameters. A further and deeper IC engine study and optimisation are necessary to ensure that the particular engine meets all requirements. The example above is only an illustration because the parameters of the engine: power and specific fuel consumption are very basic. There are a lot of other viewpoints that should be taken into account, like the engine exhaust emissions, the engine weight, placing and mounting, etc.

2.2. Internal Combustion Engine for Range Extender Unit

Since the fuel converter in a REx unit can operate in its optimal efficiency area, independently of vehicle speed (due to the lack of mechanical connection with the wheel), almost all heat engines, even those which were historically unsuitable for vehicle propulsion (as prime mover, due to transient/warm-up operations issues), can be utilised. Various engine concepts have been considered and discussed, such as piston engines, rotary engines, gas turbines, fuel cells, etc. However, considering all the criteria and requirements, the most suitable option for the REx unit for EVs appears to be the reciprocating internal combustion engine, more precisely, the conventional naturally aspirated four-stroke spark-ignition (petrol) ICEs. It offers higher specific power (up to 40 kW/dm^3) and low specific weight (up to 3 kg/kW) but higher specific fuel consumption even at optimum operating conditions due to the lower compression ratios required to ensure adequate resistance to knock (spontaneous ignition) and other losses due to the imperfect combustion process. The composition of the exhaust gases is not favourable, but aftertreatment by a three-way catalytic converter is very effective (for $\lambda = 1$). The particle content is low due to the burning of a prepared-mixture. The efficiency achieved is around 35%. Moreover, this type of ICEs can operate on renewable fuels (i.e. E85). It is undemanding, simple in design and cheap.

The limitations coming from the general requirements, especially the development and manufacturing costs, do not allow the use of the latest advanced technologies applied generally in the automotive industry in terms of improving efficiency, e.g., downsizing, downspeeding, diesel or alternative fuel engines, direct fuel injection, optimisation of the combustion process and charge exchange, friction reduction etc. In any case, the implementation of this type of IC engine implies a number of issues and challenges.

Because the range extender serves only as a backup power source, it is only used when the battery is low on charge or the vehicle requires higher power, e.g., on the motorway. The IC engine runs all the time in a steady mode (high load at a constant engine speed), which brings some potential risks. Special attention is given during the design to the engine starting up, heating up, lubricating, and cooling systems. According to the specific control strategy of the REx unit, it should be prepared to start up the engine and run it from idle to high load (issues of engine preheat and lubrication).

Since the IC engine is controlled by the vehicle's on-board control unit (i.e., not by the driver) and runs in a narrow speed range, some engine components can be dimensioned and optimised more accurately in terms of weight, strength, and durability. [6, 61]

2.3. Concepts of Range Extenders with Internal Combustion Engine

Various concepts of small IC engines are suitable for applications in REx units. A lot of them are more or less explored, and the results and conclusions are presented in research and design studies, articles, and reports from different automotive and engineering companies. [5, 6, 23, 38, 39, 41, 46, 66]. A lot of different engine and vehicle concepts have already been created. Some of them are commercially available on the market or are prepared for production, like the BMW i3 REx, Chevrolet Volt, Mazda MX-30, or the Nissan Note e-POWER. A brief overview of some existing range extender concepts shows specific features and issues related to the design of the range extender units with internal combustion engines.

• **BMW.** The BMW i3 REx electric vehicle is equipped with a range extender unit. It is powered by a twin-cylinder inline, four-stroke, naturally aspirated spark-ignition engine W20K06U0 with a displacement of 647 cm^3, which produces a power of 28 kW at 5000 min^{-1} and torque of 55 Nm at 4500 min^{-1} with a compression ratio of 11,6:1. The cranktrain uses a configuration with crank pins at 270° and two balancing shafts. [57, 58, 75]

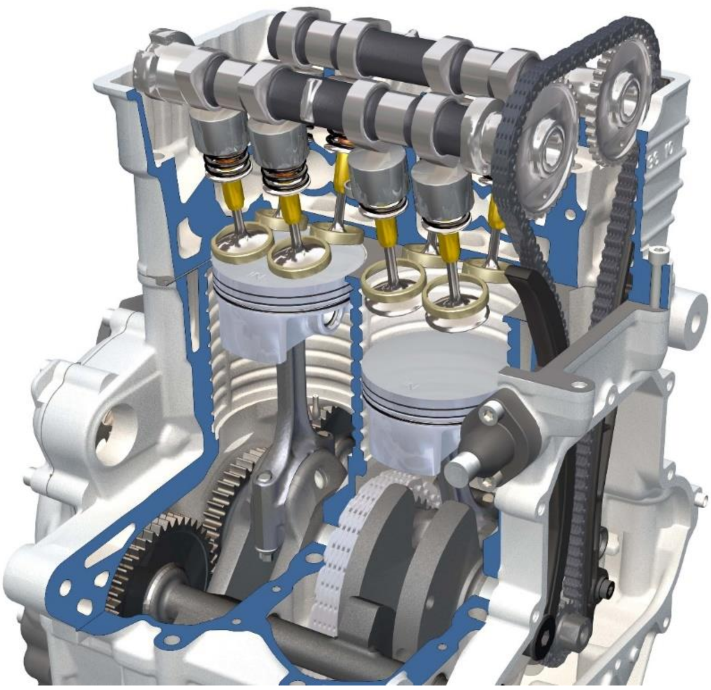


Fig. 33 BMW range extender engine (33)

• **Lotus.** The engineering team from Lotus Engineering has done extensive research with simulations and design studies, and as a result, it has developed a petrol SI engine designed specifically for hybrid powertrains. It is an all-aluminium, naturally aspirated, petrol, 3-cylinder inline engine with a displacement of 1,2 litres. The engine has 2 valves per cylinder, port fuel injection and generates an output of 15 kW at 1500 min^{-1} (for urban driving) and 35 kW at 3500 min-1 (for suburban driving with a speed up to 120 km/h). The ICE is designed with efficiency, lightweight, and cost in mind. The total weight is 56 kg. The ICE uses a TWC ($\lambda = 1$). [66]

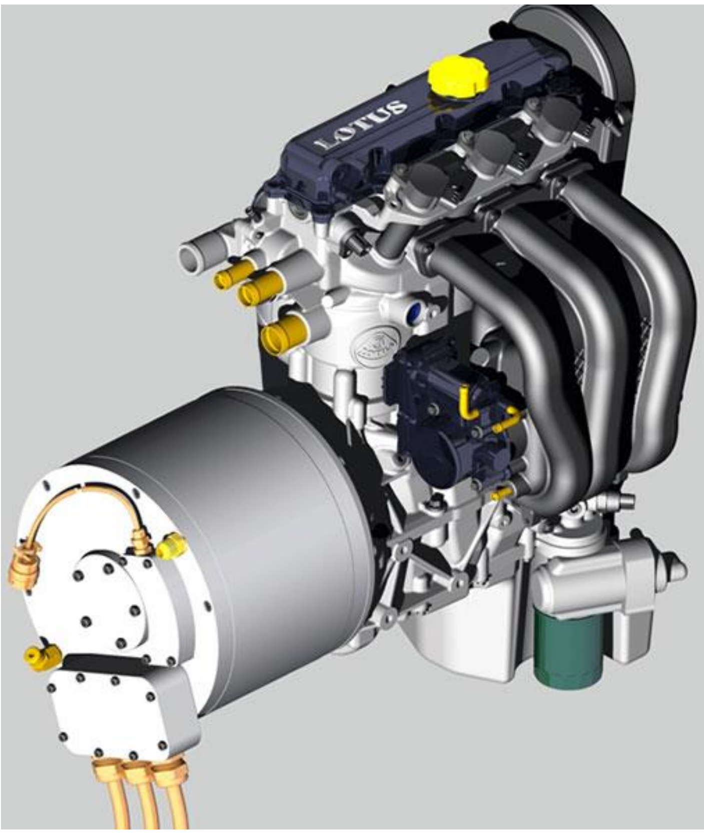


Fig. 34 Lotus range extender engine (34)

• **MAHLE.** The company MAHLE GmbH has done extensive research and has developed a twin-cylinder spark-ignition (petrol) four-stroke engine with a displacement of 900 cm^3, which has a maximum power of 30 kW at 4000 min^{-1} with a focus on efficiency and compactness. [5, 6, 7, 47] The overall dry weight of the unit, including the generator, is 70 kg. The IC engine has a crankshaft with a 180° angular offset between crankpins. This design configuration makes it possible to omit the additional balancing shaft used to balance the first-order inertial forces of the reciprocating moving masses, which is favourable to the reduction of the size, weight, and cost of the engine. On the other hand, the 180° crankshaft configuration increases the unevenness in the engine's running since during one revolution both cylinders are fired. The unevenness in the engine's running is characterised by engine speed irregularity, which leads to more noise and vibrations. This disadvantage is to some extent compensated by the use of the electric generator torque control, with no losses in efficiency and output power. Thanks to a redesigned oil pump and oil system, the engine can be mounted flexibly, practically in any position, with any inclination angle between vertical and horizontal. [77]

Fig. 35 MAHLE powertrain compact range extender engine (35)

Fig. 36 MAHLE powertrain compact range extender engine (36)

• **NISSAN.** The Nissan company has developed its three-cylinder engine HR12DE with a displacement of 1,2 litres and an output of 58 kW for its range extender electric powertrain technology e-POWER, introduced in the Nissan Note. The engine itself has been redesigned (combustion chamber, piston crown shape) in order to ensure better combustion and thermal efficiency. External balancing is involved to reduce the noise and vibrations of the three-cylinder engine. This is done by adding weights to the flywheel and crankshaft pulley. This hybrid solution is more of a series hybrid because the battery pack is small. However, it is interesting in terms of engine design and development. [57]

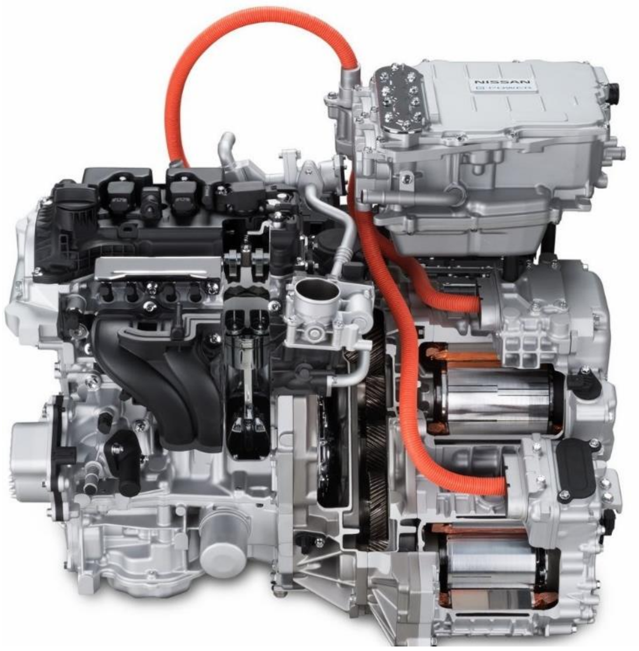


Fig. 37 Nissan range extender (37)

• **Rheinmetall Automotive AG**. The automotive suppliers Rheinmetall Automotive AG (formerly KSPG or Kolbenschmidt Pierburg) and FEV Europe GmbH have developed together a concept for a compact 30 kW 2-cylinder REx unit for BEVs. This REx has been focused on and designed for small passenger vehicles (A-segment) and was first introduced in 2012 in the Italian vehicle Fiat 500. The engine is a petrol V-twin with a displacement of 800 cm^3 and a 90° angle between the cylinders. It is a naturally aspirated engine with port fuel injection and 2 valves per cylinder. The power output is 30 kW at 4500 min^{-1}. The overall weight of the unit (engine + generators) is 60 kg. According to the research studies, a power output of 26 kW should be sufficient to drive the vehicle at a speed of 100 km/h on a track with a gradient of 3%. To minimise the overall height of the entire unit, the IC engine and the generators are mounted in a vertical position. The engine's crankshaft drives, by gear wheels, two electric generators with a power output of 15 kW each. Among other things, these EG are used to reduce the irregularity of operation (balancing) of the V-twin layout. [40, 41, 54]

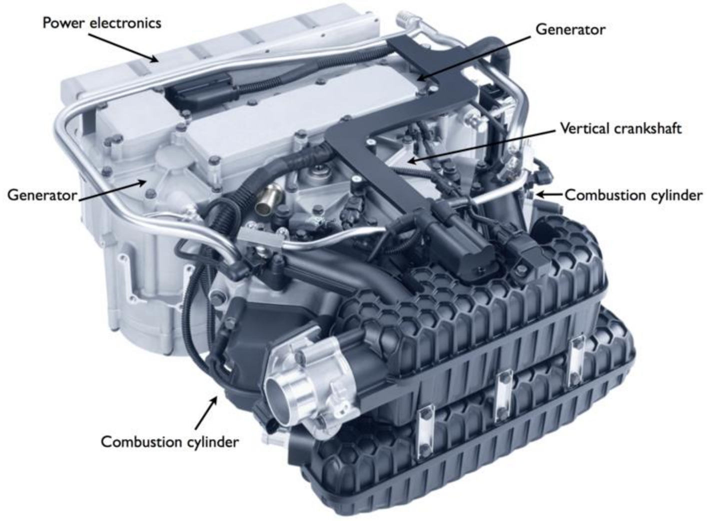


Fig. 38 Rheinmetall Automotive AG range extender engine (38)

Fig. 39 Range extender engine installed in a vehicle (39)

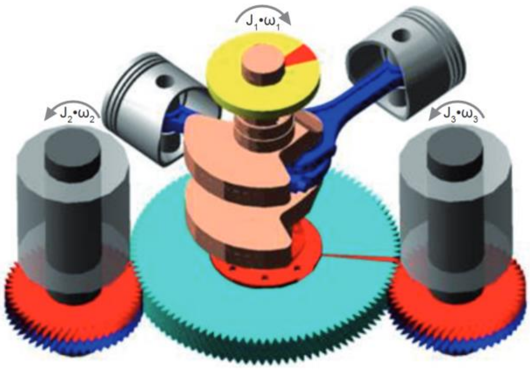


Fig. 40 CAD model of a REx engine of Rheinmetall Automotive AG (40)

• **ŠKODA AUTO a. s.** For research, development, and testing purposes, as a part of a project from the programme Josef Božek National Centre – NCK1, the Czech automobile manufacturer ŠKODA AUTO a. s. has been creating an inexpensive experimental engine with two cylinders suitable for REx application. Because it is built from and inside a standard three-cylinder engine from the EA211 engine family, basically, it is not a real twin-cylinder engine (fake twin). Specifically, the series engine 1,0 MPI, which has been produced in the town of Mladá Boleslav since 2012, is a spark-ignition, naturally aspirated, three-cylinder engine with multi-point port fuel injection. The objective of this conversion is to keep as many serial parts as possible, including auxiliary engine systems, which results in significant cost savings and a reduction in the time to build a functional prototype. Since only the second and third cylinders are utilised, the excessive components, such as the piston group, connecting rod, and rocker arms from the first cylinder, are omitted. The camshafts and camshaft carrier (cover) are newly designed by ŠKODA AUTO. In cooperation with partners from Brno University of Technology, a new optimised crankshaft with two crank pins spaced at 180° has been designed. Since the serial three-cylinder engines do not have a balancer shaft, it is necessary to modify the flywheel and damper pulley unbalance in order to achieve an external balancing of the twin-cylinder configuration. The engine has a displacement of 800 cm^3 and it is designated as 0,8 MPI REx.

Because the engine is expected to operate at lower speeds (2000 to 4000 min^{-1}) than the standard SI engine, the piston stroke is increased from the standard value of 76,4 mm to 91,6 mm. By retaining the cylinder bore at the same value of 74,5 mm, we get a distinctly longer stroke engine with a B/S ratio of 0,813 (original configuration – 0,975). The objective is to improve low-speed performance, fuel economy, and exhaust emissions. Many of the used components remain the same as in the serial engine, so they are somewhat oversized for the REx application. This is adequate in order to perform measurements and testing of the engine parameters at this stage. However, further modification and optimisation are still needed. Due to the proper function of the subsystems, the mounting position of the REx unit inside the engine compartment remains, for now, the same as for all the EA211 engines.

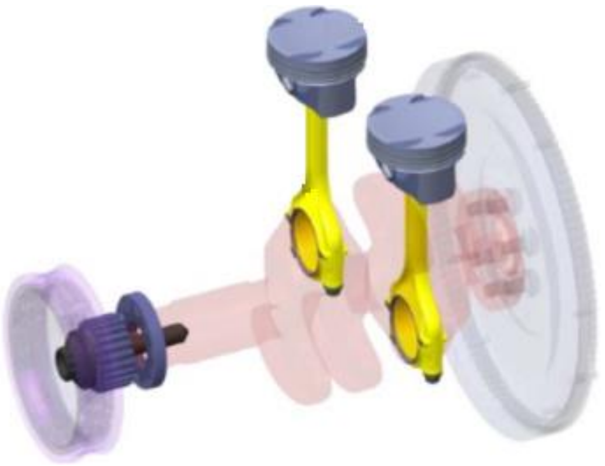


Fig. 41 Twin-cylinder range extender of ŠKODA AUTO a. s. (41)

In the following step, a virtual CAD prototype of a real twin-cylinder engine has been prepared by direct modification of the 3-cylinder engine model. The final design of the engine, on the other hand, must be optimised in order to simplify the entire system and to adapt it to the REx requirements (weight, cost, serial production), which is a complex and difficult task. After the functional sample can be tested, the basic parts of the engine can be modified for eventual series production. [25, 26, 38] Specifications of the engine for the REx engine:

	Unit	1,0 MPI evo	0,8 MPI REx
Number of cylinders	[–]	3	2
Displacement	[dm^3]	1,0	0,8
Cylinder displacement	[cm^3]	333,0	399,3
Bore	[mm]	74,5	74,5
Stroke	[mm]	76,4	91,6
B/S ratio	[–]	0,975	0,813
Max. power / speed	[kW at min^{-1}]	59 / 6200 min^{-1}	13-30 / 2000-4000 min^{-1}
Torque	[Nm at min^{-1}]	95 / 3000-4200 min^{-1}	63-72 / 2000-4000 min^{-1}
Compression ratio	[–]	10,5:1	12:1
Weight	[kg]	97	63

Tab. 1 Specifications of ŠKODA AUTO engine for range extender engine (42)

Summarising and comparing the preliminary results, the twin-cylinder engine from ŠKODA AUTO does not lag behind the competitors' engines mentioned above. According to expectations, the ICE will offer a sufficient power, but the weight and size could be optimised.

According to the above-mentioned overview, it is obvious that every automotive manufacturer has used its own approach to the subject of range extenders. Anyway, they reached several common analogies. All engines use a simple and proven technique. They are naturally aspirated and use simple port fuel injection and valvetrain systems with two valves per cylinder (except for Škoda engine, which uses four valves per cylinder). The power outputs are (except for Nissan) in the range of up to around 40 kW. All concepts strive to achieve similar objectives – minimise the fuel consumption, friction, noise, and vibration from the engine and, if possible, to eliminate/minimise the need for additional balancer shafts (employing external cranktrain balancing, electric generator torque control, two generators instead of one, and specialised ignition control).

Producer:	**BMW**	**Lotus**	**MAHLE**	**Nissan**	**Rheinmetall**	**Škoda Auto**
Layout, № of cyl. [-]	Inline-2	Inline-3	Inline-2	Inline-3	V-2 90°	Inline-2
Brake Power [kW]	25	35	30	59	30	30
ICE Displacement [cm^3]	647	1192	898	1198	799	799
Bore [mm]	79,0	75,0	83,0	78,0	80,0	74,5
Stroke [mm]	66,0	90,0	83,0	83,6	79,5	91,6
B/S Ratio (S/B Ratio)	1,197 (0,835)	0,833 (1,200)	1,000 (1,000)	0,933 (1,072)	1,006 (0,994)	0,813 (1,230)
Engine speed [min^{-1}]	4300	3500	4000	5400	4500	4000
Mean piston speed	9,46	10,50	11,07	15,05	11,93	12,21
Compression Ratio [-]	10,6:1	10:1	9,8:1	12:1	-	12:1

Tab. 2 Comparison of range extender solutions (43)

Most of the projects focused on IC engines for a range extender unit and the range extender itself have been conducted within research and development activities as studies aimed at exploration and definition of the issues and presenting interesting solutions to them. These proposals have been made according to the knowledge and experience of the individual manufacturers. Since there are a lot of requirements, most of the proposals use simple and proven technologies in order to find an optimal solution for this specific task. This in turn excludes solutions and technologies targeted at improving the overall efficiency of the engine, which are financially expensive. The final solution implies compromises between all requirements, leading to one or more optimum operating points. The final solution implies compromises between all requirements, leading to one or more optimum operating points.

2.4. Summary

The IC engine intended for REx units presents a specific type of engine. It differs from other types of IC engines used for the propulsion of conventional motor vehicles in design, size, operation, and control. Therefore, it has to be adapted to a lot of diverse requirements. A reasonable solution for the operation of the REx concept can be achieved using one or two optimised working points (constant speed, wide-open throttle, efficiency) in which the IC engine runs. As a modular unit connected in series, the REx unit can be used in various types of electric vehicle types, improving their operating range. By reducing some disadvantages, this solution should lead to a better acceptance of electric vehicles from the customer side.

This solution paves the way for pure electromobility, using all the current advantages of electric vehicles while supporting them with proven and viable IC engines in order to minimise the disadvantages. In the meantime, it will be interesting to see if the battery technology will evolve progressively and thus offer a more attractive ratio between capacity, weight, and price, which will make this solution meaningless.

3. Goals and Methods

As known, the engineering design process (not only in the automotive industry) is a complex, demanding, and challenging process that constantly asks for new solutions. Therefore, to ensure higher productivity and competitiveness, it is important to explore and continuously find new ways to improve every step of the design process - from the idea and initial design proposal, through prototyping and testing, to the production and market launch of the product.

Often, the proposal and design of a new product do not start from scratch. The development process is usually based on the design and engineering knowledge and experience gathered over the years. The process starts with analysis and research aiming to understand the current state of engineering and technology, requirements and legislation, the market, customers, and their needs. The ideas and demands for the product originate from a variety of incentives, such as new research and development findings, results, and conclusions, new technologies, market needs, environmental considerations and concerns, competitors (analysis, reverse engineering), and strategic positioning. The selection of suitable ideas always involves further analyses, proposals, feasibility studies, decisions, and conclusions based on technical, economic, and environmental requirements.

Designing a new internal combustion engine is not a simple task, even one for a range extender unit. It may look easy because it is just a small engine, but the opposite is true. This is due to quite different requirements and operating conditions, which open new possibilities in design and operation. In this case, it is the ability to run the engine at a single or two operating points. The design of an IC engine requires a much more complex analysis and design approach, sometimes even asking for completely new design solutions, almost from scratch. In many cases, however, the IC engines found in the REx units use proven solutions from series production with particular additional adaptations.

3.1. Goals of Doctoral Dissertation

Considering all circumstances, this leads to the choice of a general focus of the doctoral dissertation on the design of internal combustion engines for range extender units and, in particular, on the modifications of available cylinder units. It aims to find an appropriate solution, offering an acceptable trade-off between the operating characteristics and efficiency and the engine package and design (size, weight). This can be accomplished, for example, by using different methods for:

– mechanical design of IC engine components and subsystems, aided by parametric CAD modelling techniques, thermodynamic analyses of the engine cycle, structural analysis (calculation of stresses and strains), and so on;

– reducing (optimisation) the weight, size and volume of ICE unit;

– optimisation of design and operating properties of REx units and HEV layouts.

Regarding these general goals, **the particular practical objectives of the present doctoral dissertation are defined as follows**:

– examine, identify, or determine the power demands for an EREV using a suitable approach (measurement, calculation, or simulation);

– examine, identify, or determine the power of a REx IC engine suitable for this vehicle using a suitable approach (measurement, calculation, or simulation);

– suggest a suitable design workflow for the initial phase of the design process, aiming at an increase in productivity using the tool from the previous step;

– suggest a suitable approach, method, model, or tool for quick initial (conceptual) design of the IC engine, its mechanical systems and components during the first phase of development;

– create (develop) and implement the model or tool using the software systems and equipment available;

– test (validate) the developed method and workflow on the IC engine design;

– provide a preliminary design proposal of the IC engine, including initial analysis and verification, with respect to the specific requirements, which will serve as a basis for further design stages.

3.2. Methods

Nowadays, in the field of mechanical engineering design (but not only), it is possible to observe a distinction between traditional analytical calculation (analysis) methods and modern computer-aided simulation and computation (numerical) methods (CAx technologies). The general view is that traditional analytical calculations are becoming less important in the design of mechanical components since the available modern computer-aided methods for designing and analysing seem to offer more options and accuracy while requiring less time and work. Mostly, this is true, but it is still necessary to retain a basic understanding of the engineering and design problems (physical nature) and their solutions, which usually analytical methods provide the best. It might seem that the analytical methods are less accurate for some tasks, or in some cases, they may seem a little obsolete, but they have their merits. This provides prerequisites for the implementation of new design approaches by combining (integrating) analytical calculation methods with virtual numerical methods and models directly in the CAx environment in order to obtain a more flexible solution for design.

That is why the doctoral dissertation aims to propose and provide a suitable approach for solving design problems in the field of mechanical engineering using traditional calculation methods in combination with parametric CAD modelling techniques, thus connecting the design intent and engineering knowledge.

Because the main focus is directed at the ICE for REx units, the suggested method will be applied (tested) to the mechanical design of the IC engine, and in particular to the design of cranktrain components. However, this approach is practically intended to be versatile and applicable in the design process of other mechanical systems as well. The method strives to conduct critical engineering calculations relatively quickly with sufficient accuracy and precision and to provide direct feedback on the design, aiming to increase productivity in the initial stage of the development process. It can be considered as a preliminary stage before the use of more advanced simulation-based design methods (simulation-driven design).

Therefore, the **suggested approach** involves:

– preparation of a model for the preliminary calculation of IC engine parameters (main dimensions, kinematic and dynamic properties, etc.);

– preparation of a model for the initial design proposal of the IC engine cranktrain components by applying traditional analytical and empirical calculation methods and models for designing and dimensioning (incl. initial strength analysis, etc.);

– preparation of parametric three-dimensional CAD models of the IC engine cranktrain components using modern parametric CAx modelling techniques.

An integration of all these prepared models is used to:

– gain the best from both worlds, including a feedback between them (from calculation to design and vice versa),

– design and optimise the parameters of the IC engine and its cranktrain, and

– obtain a preliminary design proposal of the IC engine cranktrain, providing a basis for further consideration, decision-making or detailed design of units.

Following the considered design workflow and using the proposed designing approach, which is supported (assisted) by the engine cycle simulation, with the pre-prepared models of the IC engine and its components, and collecting the main (input) parameters of the designed engine, it is possible to complete the initial design steps, obtaining its initial design proposal as a result.

4. Internal Combustion Engine Design

4.1. Internal Combustion Engine

The conventional reciprocating internal combustion engine (RICE) remains the most popular power source used all around the world in all kinds of motor vehicles and, in the near future, the situation will obviously not change significantly. After more than 130 years of evolution, there is a widespread belief that internal combustion engines have already reached their limits and there is a little room for future development, but the opposite is true. The ICE, their design, processes, operation, as well as all individual technologies, have been continuously improving over the years. However, extra measures are needed in order to increase the overall efficiency and fulfil new emission standards, which are becoming more and more strict and complex.

It is necessary to search for new solutions to these challenges effectively in order to achieve significant enhancements in performance, efficiency, power density, weight and size, as well as gas emissions of the ICEs. These development objectives can be achieved by the implementation of new modern technologies, materials, processes, and design methods while still using the gained experience and knowledge. For example, advanced knowledge-based, analysis and simulation-based design approaches, methods and techniques assist the further improvement and optimisation of the ICEs during the entire development process. [27, 36, 50]

The continual process of electrification of automotive powertrain systems, which has already been carried out, has led to the implementation of different hybrid-electric technologies. In modern hybrid electric vehicles (HEVs), which are becoming more and more popular, ICEs are also the main option for a primary power source. In these vehicles, the IC engine is joined together with an electric generator, an electric battery and an electric motor. The next stage in vehicle drivetrain electrification is the utilisation of a larger-capacity energy storage electric battery together with external recharging from the electric power grid, which involves the use of plug-in hybrid (PHEV) technologies. The use of ICEs in automotive HEV and PHEV propulsion systems requires a different manner of control and operation in comparison with that of conventional motor vehicles (MVs). In these hybrid systems, the IC engines can operate in a high-efficiency mode for a longer period of time (steady-state), and thus, they do not need to change the operating mode fast and frequently. Generally, these ICEs are not designed and developed specifically for this application. They are frequently based on the conventional series-production ICE with appropriate modifications. However, additional research and development are still required in order to achieve an improvement in the overall efficiency. The pure electric vehicle is the final stage in vehicle electrification.

Vehicle electrification still presents a huge technical challenge with a lot of questions and unknowns, and the future course of this process remains, at this point, uncertain. [27, 36, 50]

4.2. Basic IC Engine Characteristics and Measures

The ICEs are complex systems described by different parameters and measures. The most common of them are briefly presented in the overview given below. Sources used for this overview are [8, 19, 35, 36, 45, 68].

Engine Torque and Power. Two common measures that evaluate the performance of the ICE are useful work and power. The useful work (or energy output) of the engine is usually expressed by the torque output. It is also essential to evaluation, not only how much work is done, but also at what rate the work is done. The time rate at which the engine does work is defined by the second measure – the engine power. It is given as a product of the engine torque and shaft rotational speed.

$$P_e = M_t.\omega = M_t.\frac{2.\pi.n}{60}.\frac{1}{1000} = M_t.\frac{\pi.n}{30000} = M_t.n.0{,}1047.10^{-3} = M_t.\frac{n}{9549}$$

$$M_t = 9549.\frac{P_e}{n},$$

where P_e *– engine brake power [kW];* M_t *– engine torque [Nm];*
n *– engine speed [min*$^{-1}$*];* ω *– angular velocity [s*$^{-1}$*].* [35, 36, 45]

Mean Effective Pressure presents the specific work of the cycle. It can be defined as a hypothetical average (constant) pressure that is assumed to act on the piston during the entire power stroke, causing the same change in volume change (from TDC to BDC) and would do the same amount of work as the actual engine cycle. The MEP can be found by dividing the engine's work output by the displacement volume. Using the indicated work (determined from the pressure-volume p-V diagram)), the result shows the indicated mean effective pressure (IMEP), but more common is to use the brake work output, which gives the brake mean effective pressure p_e (BMEP). It can be calculated by the relation:

$$p_e = \frac{P_e.30.\tau}{V_z.n}$$

where p_e*- break mean effective pressure [MPa];* P_e *–brake power [kW];* n *– engine speed [min*$^{-1}$*];*
V_Z *– engine displacement [dm*^{3}*];* τ *– number of strokes per cycle [–].* [36, 45]

Brake Specific Fuel Consumption. An essential measure of IC engine performance is its thermal efficiency, i.e. how efficiently the energy contained in the fuel can be transformed into a useful work (torque). The thermal efficiency can be estimated directly as follows:

$$\eta_e = \frac{P_e}{H_u.\dot{m}_p}$$

where η_e *– brake engine efficiency [–];* P_e *– brake power [W];* $\dot{m}_p$ *– fuel mass flow rate [kg/s];*
H_u *– lower heat value [J.kg*$^{-1}$*].* [36, 45]

The thermal efficiency is therefore the rate at which work is done (power) divided by the rate at which energy from fuel is supplied. The definition of brake specific fuel consumption m_p (SFC) is practically relevant. It is expressed by the mass flow rate of fuel divided by the power output. Using the brake engine power, the relation gives, as a result, the brake specific fuel consumption, m_{pe} or BSFC:

$$m_{pe} = \frac{\dot{m}_p}{P_e} = \frac{1000.\dot{m}_p.3600.1000}{P_e} = \frac{\dot{m}_p.3{,}6.10^9}{P_e}$$

where m_{pe} – *brake specific fuel consumption [g.kW⁻¹.h⁻¹];* P_e – *brake power [kW];* $\dot{m}_p$ – *fuel mass flow rate [kg/s].* [36, 45]

Volumetric Efficiency. The work produced by the engine is limited by the amount of supplied air (oxygen) needed for the combustion of the fuel. Volumetric efficiency is an essential measure that shows how well the displacement of the cylinder can be filled with air during the intake stroke (for a given speed), i.e., how effectively the engine displacement is used. Actually, this designation is not very precise since the measure is a ratio between the actual (real) air mass flow rate that entered the cylinder and the ideal (theoretical) mass flow rate (if the cylinder displacement could be filled entirely with air):

$$\eta_{vol} = \frac{\dot{m}_{actual=real}}{\dot{m}_{ideal=theor}} = \frac{\dot{m}_{actual}}{m_{ideal}.\frac{n}{2}} = \frac{\dot{m}_{actual}}{\rho_{ref}.V_z.\frac{n}{2}}\ [-]$$

where $\dot{\boldsymbol{m}}$ – *mass flow rate [kg/s]*, $\boldsymbol{n}$ – *engine speed [min⁻¹],* $\boldsymbol{\rho_{ref}}$ – *air density [kg/m³].*

Both mass flow rates include only the air, which is valid even for SI with external mixture formation (mass of fuel is not included). This is important because it measurably impacts the resulting efficiency. It is essential to specify the reference conditions for estimation of air density (for NA engines, usually ambient pressure and temperature). [36]

4.3. Initial ICE Design – Introduction and Design Process Workflow

4.3.1. Main Parameters – Initial Description and Preliminary Proposal

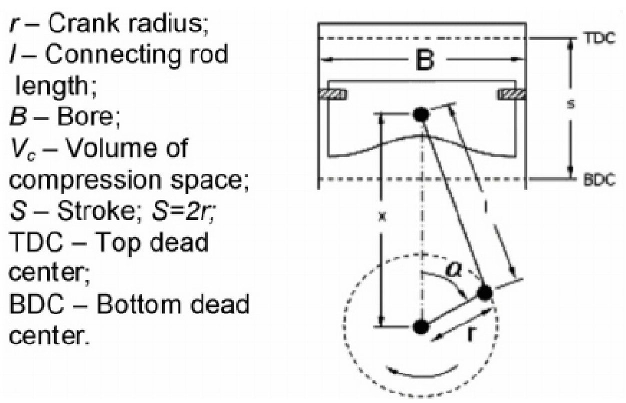


Fig. 42 The main geometric properties of a reciprocation internal combustion engine (44)

The cylinder bore B, the piston stroke S, the crank radius R and the connecting rod length L are common parameters outlining every reciprocating ICE. Since the relationships (proportion) between them predetermine the overall character of the engine, they have to be considered during the design process from the beginning. [13, 14, 15, 16, 17, 18, 36, 45, 80].

The considered process for a preliminary (initial) design proposal of a new IC engine and its mechanical subsystems begins with the resolution of four essential engine specifications, which predetermine its character and behaviour. They have a significant effect on the design, size, performance, and operating properties of the engine. All these parameters depend on the particular engine type and application.

Firstly, it is found out what value of total **displacement volume** the IC engine needs in order to generate a specific amount of power. The displacement presents the volume of the engine cylinders swept (displaced) by the pistons (without the volume of combustion chambers). Secondly, it is necessary to specify a finite **number of cylinders** into which the total volume will be distributed. After that, the dimensions of the cylinders have to be determined. The geometry of each engine cylinder is defined by two values, which are the bore B – the diameter of the cylinder; and the stroke S – the distance travelled by the piston between its two end positions (TDC and BDC). Finally, the arrangement of the cylinders in the engine frame (block) is specified by selecting a suitable engine layout (configuration). [17, 18, 36, 45]

Engine Displacement. The easiest way to calculate the total engine displacement is to use the common relation for engine power:

$$P_e = \frac{i_{cyl}.V_{Z1}.p_e.n}{30.\tau},$$

where P_e – *brake power [kW];* i_{cyl} – *number of cylinders [–];*
V_{Z1} – *cylinder displacement [dm³];* p_e – *brake mean effective pressure [MPa];*
n – *engine speed [rpm, min⁻¹];* τ – *number of strokes per cycle [–].* [36, 45]

Using the values for the desired maximum power delivered at a specific engine speed, and considering a certain mean effective pressure for the particular engine type, the total displacement can be easily obtained:

$$V_Z = \frac{120.P_e}{p_e.n}\ [dm^3].$$

The dependency of the brake power P_e [kW] at different engine speed [min^{-1}] on the engine displacement V_Z (dm^3, litres) can be graphically presented, as shown in Fig. 27. [39, 45]

Another approach is to consider the engine as a positive displacement air pump. The specification of four key input parameters – the required maximum power output [kW] of the engine, the engine speed [min^{-1}, rpm] at which to generate this power, together with the assumed specific fuel consumption and the volumetric efficiency in this operation mode, is necessary. The specification of the engine displacement is a reverse process. To produce a certain amount of work (power at a specific engine speed), the engine has to burn an appropriate quantity of fuel. To ensure complete combustion of the fuel, a sufficient amount of air has to be delivered into the engine.

So, after a rough outline of the initial expectations about the engine and its performance (power, torque, operating speed), and a prediction of the specific fuel consumption (m_{pe}, BSFC), the mass fuel flow needed can be calculated as follows:

$$\dot{m}_{fuel} = \dot{m}_p = \frac{m_{pe}.P_e}{3{,}6.10^6}$$

Knowing the air to fuel mass ratio (A/F ratio, AFR) for the particular engine, the mass air flow needed can be obtained:

$$A/F = \frac{\dot{m}_{air}}{\dot{m}_{fuel}} \Rightarrow \dot{m}_{air} = \dot{m}_{fuel}.A/F = m_{pe}.P_e.A/F$$

where m_{pe} – *specific fuel consumption [g.kW⁻¹.s⁻¹]*; $\dot{m}_{fuel} = \dot{m}_p$ – *mass fuel flow [kg.s⁻¹]*; $\dot{m}_{air}$ – *mass air flow [kg.s⁻¹]*; P_e – *effective power [kW]*; A/F – *air to fuel ratio [–]*. [36]

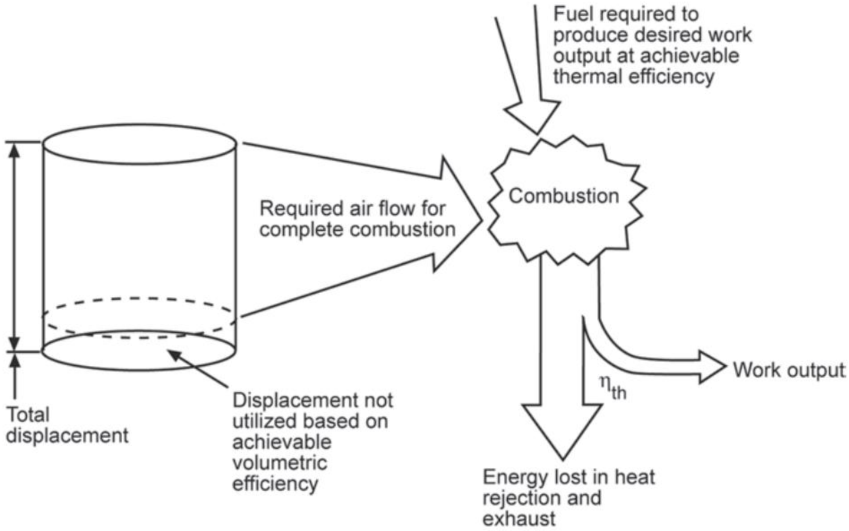


Fig. 43 Determination of engine displacement according combustion and output requirements (45)

In naturally aspirated spark-ignition engines, the output power is in general limited by its displacement across the overall speed range. The **air to fuel (A/F) ratio** of the charge mixture has a direct effect on the engine's power output. Generally, the maximum power is obtained by using a richer mixture, i.e. for most liquid fuels, an A/F ratio within 11,5 to 12,5 to 1. However, for proper management of the exhaust gas aftertreatment (proper operation of a three-way catalytic converter), the A/F ratio is nowadays very precisely adjusted to a stoichiometric mixture (chemically correct) in order to ensure complete combustion of the fuel, i.e. for petrol, close to 14,7:1 across all operating conditions. On the other hand, the best fuel consumption is achieved by a lean mixture, so the usage of a stoichiometric mixture ($\lambda = 1$) for the entire speed range involves some trade-offs. [17, 18, 36]

The actual displacement volume required in order to supply the needed air will be the ideal volume divided by the volumetric efficiency:

$$\dot{m}_{actual} = \frac{\dot{m}_{ideal}}{\eta_{vol}}$$

A displacement rate can now be calculated:

$$Displ.Rate\ \left[\frac{l}{min}\right] = Displ.\frac{n}{2}.\eta_{vol}$$

By rearranging this equation it is possible to calculate the displacement required to supply the necessary air flow rate at the given engine speed:

$$V_d = \frac{\dot{m}_{air}}{\rho_{air} \cdot \left(\frac{n}{2}\right) \cdot \eta_{vol}}$$

where ρ – *the air density at the inlet [kg/m^3]; η_{vol} – the expected volumetric efficiency [kg/min]*.

Detailed discussion on displacement volume determination can be found in reference. [36]

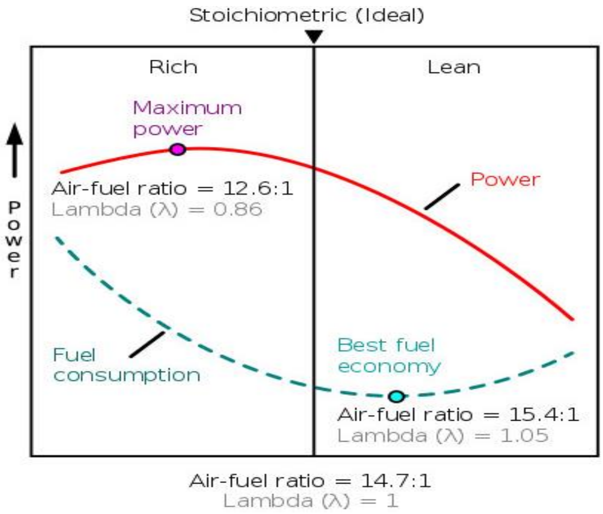


Fig. 44 Engine power vs. Air-fuel ration (46)

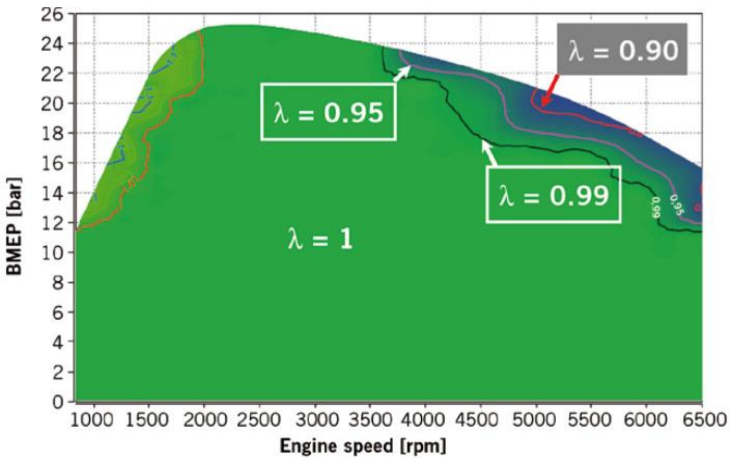


Fig. 45 Map of engine BMEP (47)

4.3.2. Number of Cylinders and Engine Layout

When the determined engine displacement exceeds 500 cm^3, a multi-cylinder configuration is usually more suitable. Because the number of cylinders and their arrangement (position and orientation in the engine block) are key design elements, the next step is to choose the engine layout. Various factors have to be considered during this design stage, such as engine complexity and cost, reciprocating and rotating masses, the balance of mechanical forces, vibrations, the surface-to-volume ratio of the combustion chamber, engine speed, mean piston speed, pumping losses, engine dimensions (size, weight, packaging), etc. The mechanical forces and moments acting inside the cranktrain and the resulting engine balance present another key element in favouring a specific engine layout. [36, 45]

Depending on the specific engine application and its subsequently resolved displacement, it is important to choose a reasonable number of cylinders into which the total volume will be distributed. Typical values of the cylinder displacement of various engine types are presented in Tab 3.

The *mechanical complexity* of the engine and the resulting costs are usually proportional to the number of cylinders. The higher the number of cylinders, the higher the complexity (number of components) and the higher the costs (for design, materials, manufacturing, assembly, testing). The difficulties associated with increasing the number of cylinders and engine displacement can be overcome by the use of forced induction in a smaller engine with a smaller number of cylinders (2,0L 4-cyl. turbocharged vs. 3,0L 6-cyl. NA engine). However, this also leads to an increase in the overall complexity and cost of the power unit by adding additional components, such as a supercharger (a mechanically or turbine-driven compressor), charge cooler (intercooler), intake manifold, etc. The final decision is a trade-off between all the requirements for engine performance, size, complexity, and cost. [17, 18, 36]

Engine type	Cylinder Displacement (cm^3)
Small utility engine	140 - 500
Sport motorcycle engine	100 - 350
Cruiser motorcycle engine	375 - 1000
Automotive SI (petrol) engine	350 - 850
Automotive CI (turbo-diesel) engine	475 - 840
Truck CI (turbo-diesel) engine	840 - 2660

Tab. 3 Cylinder volume for different engines (48)

On the other hand, for a specific engine displacement volume, the reciprocating masses increase by reducing the number of cylinders. An engine with a higher number of cylinders will have a smaller cylinder bore (diameter) and therefore smaller and lighter piston groups and connecting rods, and therefore lower reciprocating masses. The loads inside the engine components are caused by the reciprocating moving masses and instant acceleration, which is proportional to the engine's operating speed. That is why a higher number of small-bore cylinders with lighter piston groups is suitable for high-speed engines in order to keep the acting forces within a reasonable limit for each cylinder (mass distribution). [36, 45]

There are different variations in the cylinder arrangements: inline (straight), V (Vee) and W-shaped, flat (horizontally opposed), rotary (Wankel), and radial. The decisions in this context are focused on the determination of the engine size, including the basic dimensions – the length, height and width of the engine, including the layout of the intake, exhaust and cooling system, engine mounting system, etc. The inline engine configuration is the most common and has cylinders arranged aligned in one row (cylinder bank). Since it is the simplest one, all the systems of the engine are easy to set up. Other engine configurations have more than one cylinder bank. This involves the necessity to double some of the engine components, e.g., the cylinder head with valvetrain, inlet and exhaust manifolds, etc. The overall height of the engine is an important parameter and should be in harmony with the size of the engine compartment. The same is valid for the length of the engine, which depends on the number of cylinders. In some cases, the installation of ICE into the vehicle's engine compartment can be challenging and this involves a mounting orientation other than vertical, e.g., inclined or horizontal. [36]

The lower height of other possible engine layouts is compensated by the increasing width. In a V-engine the cylinders are arranged into two separate banks and this allows us to reduce both height and length at the expense of width. However, the design of cooling, intake, and exhaust systems is more complicated. Some difficulties exist in the area of main and connecting rod bearings, where space is somewhat limited (connect two connecting rods to a single pin instead of one rod to a single pin as in the straight engine). [36]

The flat (horizontally opposed) engine benefits from a very low height and length, similar to a V engine, but it is much wider. A significant reduction in the engine length can be achieved by switching from an inline cylinder configuration to a V-type or a boxer.

4.3.3. Determining the Cylinder Bore-to-Stroke Ratio

After specifying the engine displacement volume, the number of cylinders and their arrangement in the engine block, the cylinder and combustion chamber have to be shaped appropriately. To achieve this goal, one more essential proportion has to be precisely defined – the ratio between the cylinder bore B (the diameter of the cylinder) and piston stroke S (distance between the TDC and the BDC). In theory, for a specific cylinder displacement, an endless number of bore-to-stroke ratios is possible. The B/S ratio has a significant effect on overall engine size (dimensions, e.g., block length vs. deck height, connecting rod length vs. piston compression height) and weight, engine speed (mean piston speed), the design of components (e.g., crankshaft, connecting rod), the compression ratio, the size (area) of intake and exhaust valves, the engine balancing and vibrations, the cylinder surface area, etc. Moreover, it has a significant impact on the following factors: engine performance (power output and speed), thermal and combustion efficiency (heat transfer losses, surface-to-volume ratio at TDC), pumping and mechanical friction losses (piston speed, valve size and flow area), and also on the valvetrain arrangement etc. These parameters should be optimised together with the specification B/S ratio. The ratio is given by the following equation:

$$\mathrm{B/S} = \frac{\text{Cylinder Bore Diameter}}{\text{Piston Stroke Length}} = \frac{\mathrm{B}}{\mathrm{S}}$$

Values for the B/S ratio for different types of engines and vehicles are presented in Fig. 46. The B/S ratio has a significant impact on the engine's performance. The size of the cylinder (B vs. S) and valvetrain characteristics (B vs. valve diameters, timing) have to be adjusted to optimise the engine's performance output. An engine configuration with equal bore and stroke (B = S) is termed as a square (square engine). An over-square or short-stroke engine has a ratio higher than 1 (B > S – larger bore, shorter stroke). This allows reaching higher engine operating speeds but also using larger valves to optimise the inlet air velocity and charge exchange at those speeds, resulting in better power density (high-speed performance). The engine can also be more compact, the piston speed will be slower and the vibrations of the engine will be lower. A higher value of the engine bore will ensure more space for valves and the cylinder wall area will be smaller. And vice versa, a configuration with a B/S ratio lower than 1 (B < S), called an under-square or long-stroke engine, allows the peak power to be achieved at lower engine speeds (i.e. better low-speed performance). [17, 36]

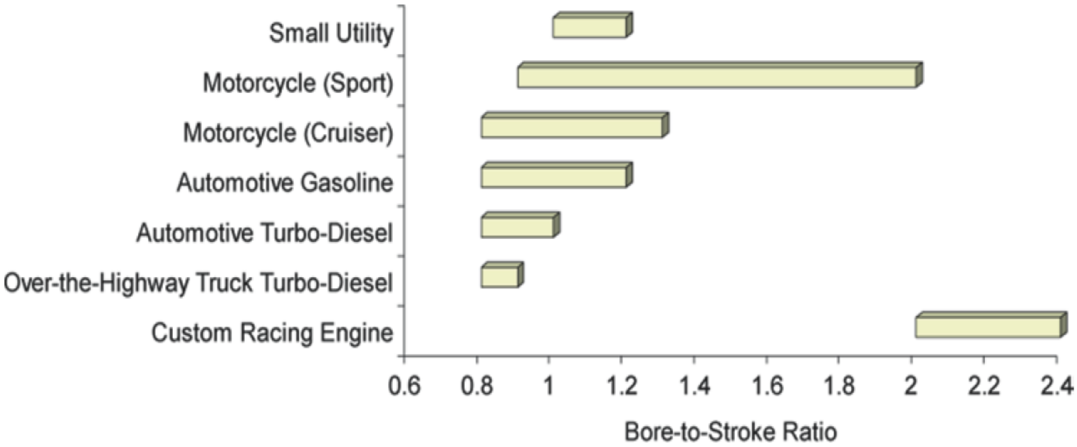


Fig. 46 Bore-to-Stroke Ratio for different types of engines (49)

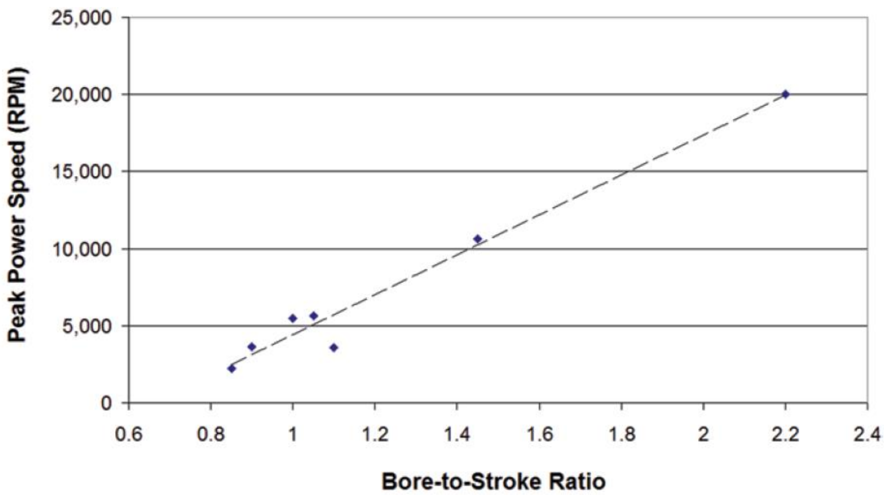


Fig. 47 Peak power speed as a function of B/S ratio (50)

The pistons, which move in a reciprocating manner, continuously change their velocity and acceleration rates during every stroke. The piston velocity is zero at TDC, and the piston accelerates to its maximum velocity around the middle of the stroke, and then decelerates and stops at BDC. The maximal values of the piston velocity and acceleration depend on the length of the piston stroke and the engine speed. The rate of acceleration affects the inertial forces acting on engine components and impacts their mechanical stresses. A practical method to characterise and compare the engine design and structural limits, and to compare different engines, is to use the measure of the mean piston speed.

$$c_s = \frac{2.S.n}{60} = \frac{S.n}{30}$$

where c_s *– mean piston speed [m/s];* S *– piston stroke [m];* n *– engine speed [rpm, min*$^{-1}$*].*

For specific (fixed) values of mean piston speed and cylinder bore, increasing the stroke lowers the engine speed. A higher mean piston speed increases friction and wearing, but it increases also the load forces in the connecting rod. In some cases, it is possible to reduce the stress in the connecting rod by using a longer connecting rod in a combination with a shorter and lighter piston. [15, 36]

The valvetrain system ensures cylinder charge exchange by precisely controlling the intake and exhaust timing events. The design of the valvetrain is a different theme, but there are a few key points to note. The valvetrain is a fine mechanism that should be optimised with regard to the kinematic and dynamic characteristics without exceeding the material and stress limits of the components while staying within the component's material and stress limits. A rough approximation of the inertia forces acting on the valves can be obtained by the relation:

$$F_v \approx m_v.h.n^2,$$

where F_v *– inertia force on valve [N];* m_v *– mass of the valve [kg];*
h – maximum lift (stroke) of valve [m]; n *– engine speed [s*$^{-1}$*]*. [36]

The value of the cylinder bore affects the size (diameter) of the intake and exhaust valves. For example, for large bores, the intake valve also needs to become larger to ensure a sufficient quantity of air at higher engine speeds. However, this negatively affects the weight of the valve. The combination of higher valve weight and higher engine speed (higher valve acceleration) impacts the inertia forces. Currently, the overhead camshaft (OHC) system is preferred over the overhead valve (OHV) system. It allows us to use a system of 4-valves per cylinder and to distribute the weight among smaller valves, optimising the charge exchange, inertia properties, and stresses in the mechanism.

An overview of typical ranges for common engine parameters is shown in Tab. 4. These ranges are constantly improved during the development process in order to reduce fuel consumption and create cleaner exhaust emissions. [36]

Engine type	B/S ratio	Peak power speed [min^{-1}]	Mean piston speed [m/s]	BMEP at peak torque [MPa]
Small utility engine	1,0 - 1,2	3 600	8-10	0,8 - 1,0
Sport motorcycle engine	0,9 - 2,0	6 800 - 14 500	16-24	1,0 - 1,5
Cruiser motorcycle engine	0,8 - 1,3	4 250 - 7 000	13-23	0,9 - 1,2
Automotive SI (petrol) engine	0,8 - 1,2	4 000 - 7 000	13-24	1,1 - 1,3
Automotive CI (turbo-diesel) engine	0,8 - 1,0	3 300 - 4 000	9-13	1,3 - 2,1
Truck CI (turbo-diesel) engine	0,8 - 0,9	1 800 - 2 600	10-14	1,9 - 2,3
Custom racing engine	2,0 - 2,4	20 000	22-33	1,6

Tab. 4 Peak power speed as a function of B/S ratio (51)

4.3.4. Surface Area to Volume Ratio

Another important characteristic/feature of the engine cylinder is the ratio of the surface area of the combustion chamber to its volume.

$$S/V = \frac{A_{ch}}{V_{ch}}$$

The **surface area to volume ratio** (S/A or SA/V ratio) has a notable effect on both combustion efficiency and heat transfer. As the piston moves in a reciprocating manner during engine operation, the current value of the S/V ratio changes with the change in the cylinder volume. At BDC, the S/V ratio of the cylinder is almost independent of the B/S ratio, and vice versa – the B/S ratio heavily impacts the S/V ratio at and around TDC. When comparing different engines with an equal cylinder displacement, at TDC, for an engine with a higher B/S ratio (larger bore, shorter stroke) the S/V is also higher, and thus more heat is rejected from the cylinder into the walls. This is very important, especially at TDC when the heat energy is quickly released during the combustion. Regarding engine performance and efficiency, the heat rejection into the wall surfaces of the combustion chamber should be minimised. [17, 18, 36]

The number of cylinders as well as the B/S ratio of the engine have an effect on the S/V ratio. Changing the number of cylinders for a fixed engine displacement increases or decreases proportionally the S/V ratio (2 vs. 4). A bigger cylinder results in a lower value of the S/V ratio.

The B/S ratio significantly influences the heat transfer process and **combustion efficiency**. For different reasons, the design and optimisation of the combustion chamber shape with a rising bore are getting more and more complicated. A rising bore diameter will change the aspect ratio of the combustion chamber (to a flat disc). For a SI engine, larger B results in a longer flame travel distance and negatively affects the combustion process and heat energy release. [36]

The maximal achievable value of the **compression ratio** (CR) of the engine is also limited by the B/S ratio. In general, the effort is focused on increasing the value of the compression ratio in order to enhance the thermal efficiency as well as the fuel efficiency of the engine. It is not easy to reach a higher compression ratio for an engine (for constant displacement) in the case of higher B/S ratios (when the bore is large, and the stroke is reduced). In this case, the combustion chamber becomes narrow and a collision between the valves and the piston at TDC can occur. The compression ratio is defined as a ratio between maximal and minimal volume inside the cylinder:

$$\varepsilon = \frac{Max.volume}{Min.volume} = \frac{V_k + V_Z}{V_k} = \frac{V_{cc} + V_{cyl}}{V_{cc}}$$

The total length of the pipelines of the intake and exhaust systems required to fill the cylinders evenly with air and guide out the exhaust gases into the environment is also determined by the number of cylinders. The longer length of the intake and exhaust manifolds, as well as the smaller diameter of the pipelines, affect the engine **pumping losses** negatively.

This side effect can be minimised by selecting appropriate pipeline lengths in order to optimise the filling and emptying of the cylinder using dynamic pressure waves at a given engine speed. Another issue arises as the number of cylinders increases, resulting in a smaller cylinder bore, which limits the valves' diameters. [17, 18, 36]

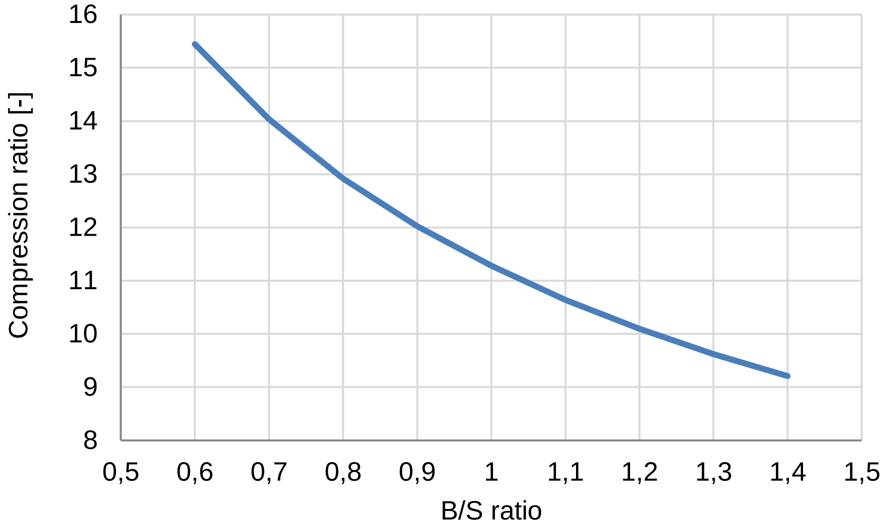


Fig. 48 CR for equal combustion chamber height (52)

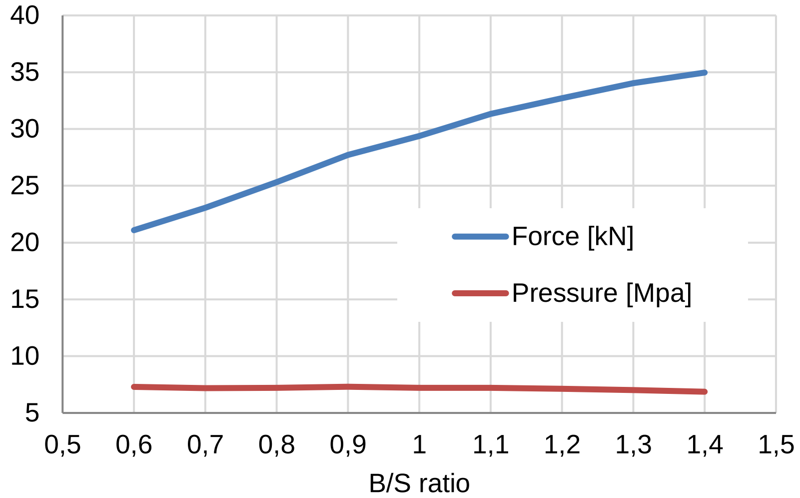


Fig. 49 Force on piston vs. cylinder peak pressure at 4000 rpm, CR 12 (53)

The B/S ratio slightly affects the **mechanical frictional losses** of the engine. The use of a higher B/S ratio (B/S > 0, i.e. B > S) results in a higher load on the engine components and bearings. The reason for this is that a similar gas pressure will act on a larger surface area of the piston crown. That is necessary to be taken into account during the design and optimisation of all separate engine components. This, in turn, involves the need for bearings with a higher load capacity, but with higher hydrodynamic losses. The growing mass of the piston group (function of the bore) will negatively impact the inertial forces and components and the bearing load. In contrast, a longer piston stroke (B/S < 0; i.e. B < S) will result in a higher mean piston speed c_s, resulting in increased friction between the piston and piston rings and the cylinder wall (all other conditions remain the same).

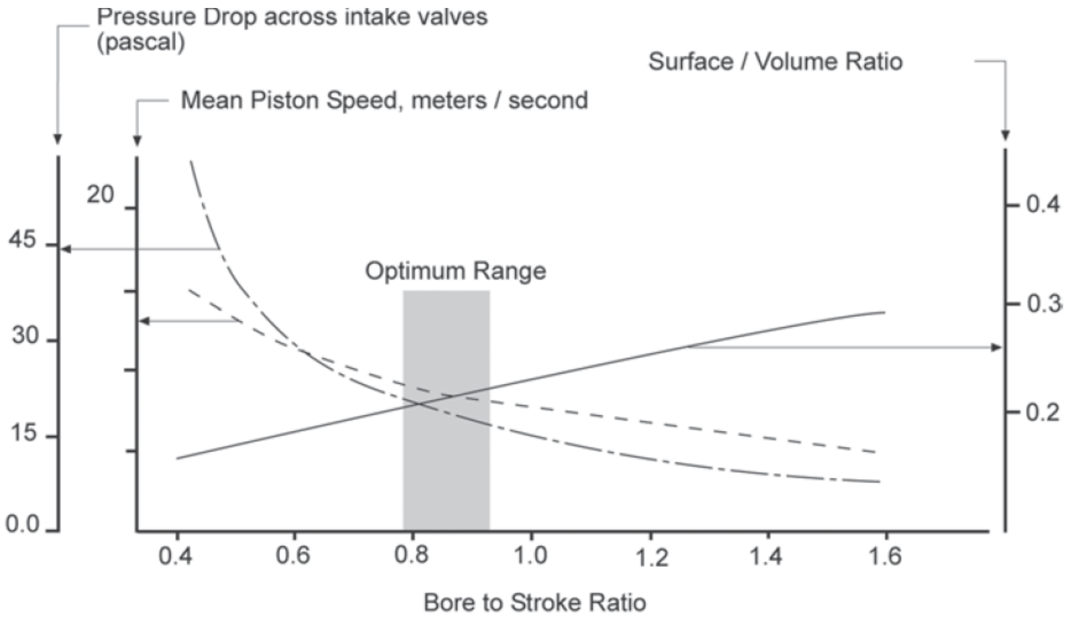


Fig. 50 Determination of bore-to-stroke ratio (54)

Since all the discussed parameters have a combined effect, they should be considered in a complex manner. A trade-off between all these parameters should be made to determine the optimal B/S ratio. The cylinder bore (and stroke) must be appropriately sized in terms of the design limits for mean piston speed and minimisation of pressure drop across the valves (which increases rapidly with reduction of valve area). As a result, the bore value is not selected to be larger than the one necessary to meet these two requirements. This is made with one goal – to minimise heat loss from the cylinder. For every engine, the specific design criteria and limitations will affect the resulting value of the B/S ratio. In order to achieve the optimum balance of performance, fuel efficiency, and emissions, the B/S ratio for automotive engines is usually under 1,0. When high specific power is required (high-performance engines), the B/S ratio tends to be higher in order to optimise the charge exchange and operate at acceptable piston speeds at high engine speeds. [17, 18, 36, 45]

Existing Engine. There is also another common task in engine development that needs to be mentioned here, i.e., how to improve the performance of an already developed or existing engine (e.g., by 10 – 20 %). This can be achieved in a few ways. Only a slight improvement can be achieved in terms of volumetric efficiency and specific fuel consumption with a **fine adjustment** of the current state of the designed engine.

A different technique for increasing the engine power is to raise the operating **speed of the engine**. It is a capable solution, but it always leads to worsened fuel economy (due to higher mechanical friction and pumping losses). Moreover, high revolutions of the engine result in higher inertia forces and stresses in the engine components. [17, 18, 36]

Supercharging by a mechanical or turbo-compressor (forced induction) is another option for increasing the power of an engine. However, the engine components should be modified substantially in order to withstand higher peak pressures and loads.

Usually, the only possible way to improve the power output of the conventional spark-ignition engine (naturally aspirated) is to **expand its displacement volume**. Earlier, it was relatively easy to achieve. Anyway, today the development is focused on minimising engine weight and size, which in turn involves some other challenges related to the increase in the values of engine bore or stroke in terms of some critical dimensions and design trade-offs.

4.3.5. Connecting Rod

The connecting rod is one of the main IC engine's components, and it has an indisputable position in the engine cranktrain. It connects the piston to the crankshaft physically, thus also linking their motions – reciprocating and rotational. The properties of the connecting rod, such as weight and length, affect the magnitude of the resulting forces in the cranktrain, so it is important to design the connecting rod properly. In terms of mechanical properties (strength), a well-designed connecting rod should handle the loads (withstand the compression and tensile forces) in a wide range of operation conditions. In this context, an important moment is the appropriate design and optimisation of the piston with respect to its weight (which affects the reciprocating mass), but also the right choice of the engine operating speed range (affects the inertial forces). The main effort is directed towards the design of enough light (to improve the engine response) and a strong connecting rod for the particular IC engine. In some situations, a decent modification of the connecting rod length only (piston stroke remains the same) will allow us to tweak slightly the engine performance or lifespan. A brief study shows what the real benefit of altering the connecting rod length is. [17, 18, 36]

Connecting rod ratio: the length of the connecting rod is involved in an essential engine parameter called the connecting rod ratio λ. This parameter represents some geometrical proportions of the engine's cranktrain and its values affect the working cycle, behaviour and lifespan of the engine. It is given as a relation between the connecting rod length L and the crank radius R (half of the piston stroke S):

Fig. 51 Connecting rods with different design and length (55)

$$\lambda = \frac{R}{L} = \frac{S}{2.L}\ [-].$$

The length of the connecting rod L is defined as the distance between the axis of the crank pin and the axis of the gudgeon (piston/wrist pin). The crank radius is defined as the distance between the axis of the crank pin and the axis of the main journal of the crankshaft. Usually, there is a limited range of values for this parameter used for most conventional IC engines. This is derived from design basics – the connecting rod must be longer than the engine stroke (it is not possible to be equal) and should not be too long (a length of twice the stroke will make the engine very high). Common values of the ratio λ for automotive engines are between 0,23 (long rod) and 0,33 (short rod), which means that the length of the connecting rod is approximately 1,5 to 2,2 times longer than the engine stroke.

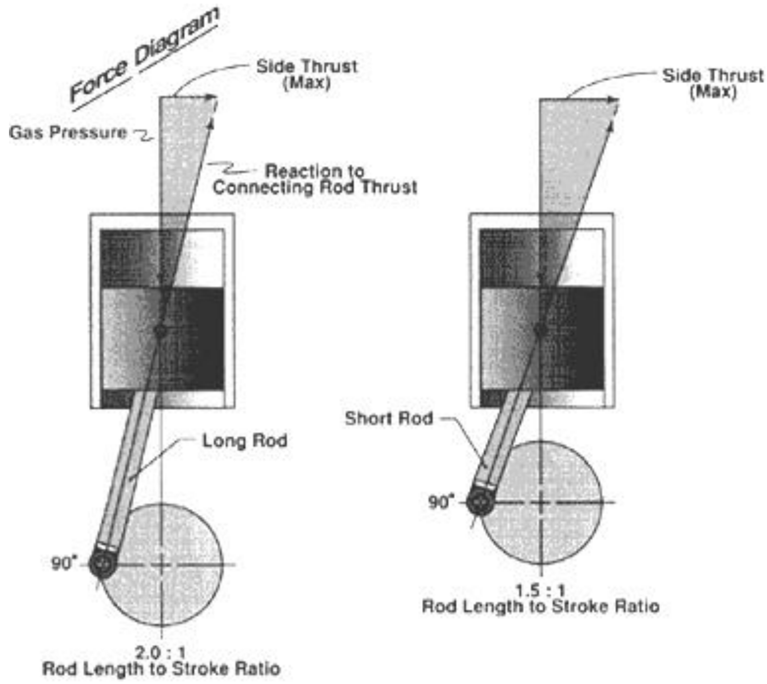


Fig. 52 Comparison of force load in cranktrain (56)

The connecting rod length L (for a constant value of engine stroke) alters the angle β between its axis and the axis of the cylinder and affects the kinematic properties of the cranktrain. This angle determines the forces acting on the piston skirt and cylinder wall (which impacts the friction) as well as on the connecting rod. For example, a shorter connecting rod (a bigger value of λ) increases the angle β and thus, bigger side forces will appear. This will increase the friction (and wear) between the piston skirt and the cylinder wall, which in turn can lead to a higher engine and oil temperature. In contrast, the shorter connecting rod results in a more compact engine design – a lower and lighter engine block and lower pistons. Moreover, the engine will offer better throttle response and low-speed performance, which are suitable for more comfortable city driving and vice versa.

The charge exchange process and mixture formation are affected due to the differences in the piston motion at and around TDC (during intake). In addition, ignition advance (for SI engines) should be optimised, in order to reduce engine knocking and optimise combustion and heat release. [17, 18, 36]

Fig. 53 Comparison of connecting rod length and piston height (57)

Long connecting rod: A longer connecting rod causes a longer stay of the piston at and around TDC. The combustion chamber volume remains for a longer time at the same small value and retains the compression state longer, which leads to better and faster combustion. The engine performs better (torque and power) at middle and higher speeds. The engine friction is reduced due to the smaller connecting rod angle (smaller side force between the piston and cylinder wall). A lighter and shorter piston can be used, which balances the slightly higher connecting rod weight. A worsening of the engine filling (volumetric efficiency) occurs due to reduced airflow velocities, mainly at low engine speeds. The piston velocity after the TDC will be slightly reduced, and the maximum value will be shifted further. The low-end torque and throttle response will be worsen. [17, 18, 36]

Short connecting rod: A shorter connecting rod will improve the airflow velocity during strokes at lower engine speeds. It will result in better filling of the cylinder at the beginning of the intake stroke, better mixture formation (in terms of homogeneity), and, in turn, it will improve the performance in this operating range. The piston will move faster after the TDC and the combustion chamber volume will increase faster, which will shift the position of the maximum cylinder pressure. On the other hand, a higher piston velocity after TDC negatively affects the combustion process. At higher engine speeds, it can lower the total cylinder pressure. The time at which the piston stays at and around the TDC is shorter. The piston will go down faster and the pressure and temperature in the cylinder will also decrease faster. [17, 18, 36]

4.4. Engine Component Design

4.4.1. Key Points in Design and Construction of Internal Combustion Engines

To design a new IC engine, it is desirable to use reasonably all the experience and knowledge obtained in engine building over the years. Usually, IC engines of one type are characterised by similar typical design features as a result of the objective way of evolution. For this aim, it is necessary to study all the characteristic features of the particular engine type or to select an appropriate existing engine as a sample and basis for designing the new one.

Before starting the design process, it is useful to collect all the available relevant data: parameters and properties of the designed ICE in order to have a better idea about the requirements, as well as of the sample ICE in order to have stable input data for initial design activities and thermodynamic analysis. The most common of them are listed below:

– type of engine – spark-ignition (petrol), compression-ignition (diesel);

– type of cylinder filling (charging) – natural aspiration, supercharging (forced induction);

– type of mixture preparation – external (carburettor, port injection) or internal (direct injection) for petrol, pre-chamber or direct injection for diesel;

– compression ratio – ε [–];

– excess air coefficient – λ or α;

– maximum (rated or peak) power – P_e [kW];

– rated engine speed – n [min^{-1}];

– number of cylinders – i;

– arrangement of cylinders;

– cylinder bore (diameter) – B or D [mm];

– piston stroke – S [mm];

– bore-to-stroke ratio – $x = B/S$ [–];

– connecting rod length – L [mm];

– crank radius – R [mm];

– design ratio – $\lambda = R/L$, [–];

– engine displacement volume – V_z [dm^3];

– mean piston speed – c_s [dm^3];

– mean effective pressure – p_e [MPa];

– power per litre volume – P_e/V_z [kW/dm^3].

However, mean piston speed c_s and mean effective pressure p_e are specific parameters essential for the evaluation of the sample engine. The mean piston speed c_s is the path of the two strokes that the piston travels during the rotation of the crankshaft as a function of time t = 1/n. They can be determined from already known parameters using the equations:

$$c_s = \frac{2.S.n}{60} = \frac{S.n}{30} = \frac{2.R.n.\pi}{30.\pi} = \frac{2}{\pi}.R.\omega$$

where $\boldsymbol{c_s}$ – *mean piston velocity [m/s];* $\boldsymbol{S}$ – *piston stroke [m];* $\boldsymbol{n}$ – *engine speed, [min^{-1}],* $\boldsymbol{R}$ – *crank radius [m];* $\boldsymbol{\omega}$ – *angular velocity [1/s].*

$$p_e = \frac{30.\tau.P_e}{V_z.n}$$

where $\boldsymbol{p_e}$ – *mean effective pressure [MPa];* $\boldsymbol{P_e}$ – *brake power [kW];* $\boldsymbol{V_z}$ – *engine displacement volume [dm^3];* $\boldsymbol{n}$ – *engine speed [rpm, min^{-1}];* $\boldsymbol{\tau}$ – *number of strokes per cycle [–], for four-stroke engines = 4.*

4.4.2. Specification of Basic Design Parameters of the Designed Engine

Piston Stroke S and Cylinder Bore B. The displacement volume of the IC engine required to achieve a specific power at a specific mean effective pressure can be obtained from a preliminary calculation (as it was presented above), engine proposal or thermodynamic analysis. The ratio of the cylinder bore to the piston stroke x and the number of engine cylinders i are derived from the sample engine or depend on further design considerations. From these initial data, the following calculations can be made:

– displacement volume of the cylinder V_{z1}:

$$V_{z1} = \frac{V_z}{i}$$

where V_z – *engine displacement volume [dm³]*; V_{z1} – *cylinder displacement volume [dm³]*, i – *number of cylinders [–].*

– bore (diameter) of the engine cylinder B (or D):

$$V_{z1} = \frac{\pi . B^2}{4} . S = \frac{\pi . B^3}{4 . x}$$

where $S = B/x$

$$B = \sqrt[3]{\frac{4 . x . V_{z1}}{\pi}}$$

The obtained value is usually rounded to the nearest whole number or the one decimal place.

– piston stroke – with the final value for the cylinder bore found above, the piston stroke can be determined as follows.

$$S = \frac{B}{x}$$

Also, the value of the piston stroke is usually rounded to a whole number or one decimal place. Now the final displacement volume of the new engine can be recalculated with these values for the basic engine dimensions.

$$V_z = \frac{\pi . B^2}{4} . S . i$$

A refinement of the final performance characteristics of the proposed engine can be calculated by the newly obtained (rounded).

$$P_e = \frac{V_z . p_e . n}{30 . \tau} \qquad M_t = 9549 . \frac{P_e}{n}$$

4.4.3. Design and Construction of Cranktrain Components

An overview of some simple but common approaches to the design of mechanical components of the IC engine cranktrain (in a first approximation) is presented below and used afterwards in the suggested method for a preliminary design proposal. Most of the design dimensions of the components can be determined according to the statistical data and practical experience to a first approximation.

Piston. The design of the piston and its strength calculation and optimisation, in order to withstand high gas pressure and temperature, is a complex, demanding and challenging task. However, for the initial design stage, a conceptual design proposal of the piston is necessary to be created. For this aim, statistical data or data from the specimen engine should be used. The basic dimensions of the piston are shown in Fig. 54 and Tab 5.

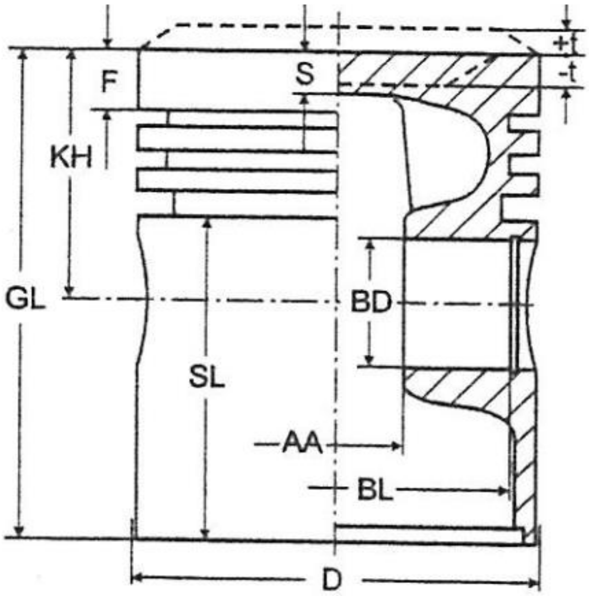


Fig. 54 Basic dimensions of piston (58)

	Name	Petrol	Diesel
D	Cylinder bore (diameter) [mm]	65-105	75-180
GL	Total piston height	(0,6-0,7).D	(0,9-1,3).D
KH	Compression height	(0,3-0,45).D	(0,5-0,8).D
SL	Piston skirt	(0,4-0,5).D	(0,5-0,9).D
F	Top land height	(0,02-0,12).D	(0,04-0,23).D
S	Piston crown height	(0,07-0,10).D	(0,10-0,15).D
BD	Gudgeon pin diameter	(0,2-0,26).D	(0,3-0,4).D
BL	Gudgeon pin length (floating)	(0,78-0,88).D	(0,8-0,9).D
AA	Distance between piston bosses	(0,2-0,35).D	(0,2-0,35).D
+t/-t	Convex head / Concave head (chamber)		

Tab. 5. Relative dimensions of piston for four-stroke engines (58)

For a correct selection of the above-listed dimensions, it is necessary to specify the number and dimensions (height, radial width) of the piston rings and the dimensions of the piston (gudgeon) pin (diameter and length). Data are usually retrieved from the specimen ICE. The final refinement of the piston pin dimensions is made after its strength calculation.

Important dimensions for the further design are the compression height, the overall height of the piston, the eccentricity of the piston pin axis, as well as the dimensions and volume of the combustion chamber inside the piston crown (if present).

Crankshaft. The crankshaft is the most structurally complex and the most stressed component of the engine, receiving alternating loads from gas and inertia forces and their moments. The design process usually proceeds in two stages. The initial design proposal of the crankshaft is based either on design considerations or data from the specimen engine. The basic dimensions, their designations and approximate values for automotive engines needed at this stage are in Fig. 55 and Tab. 6. After a complete calculation of strength and optimisation using a specific methodology, nowadays computing, the final dimensions are obtained.

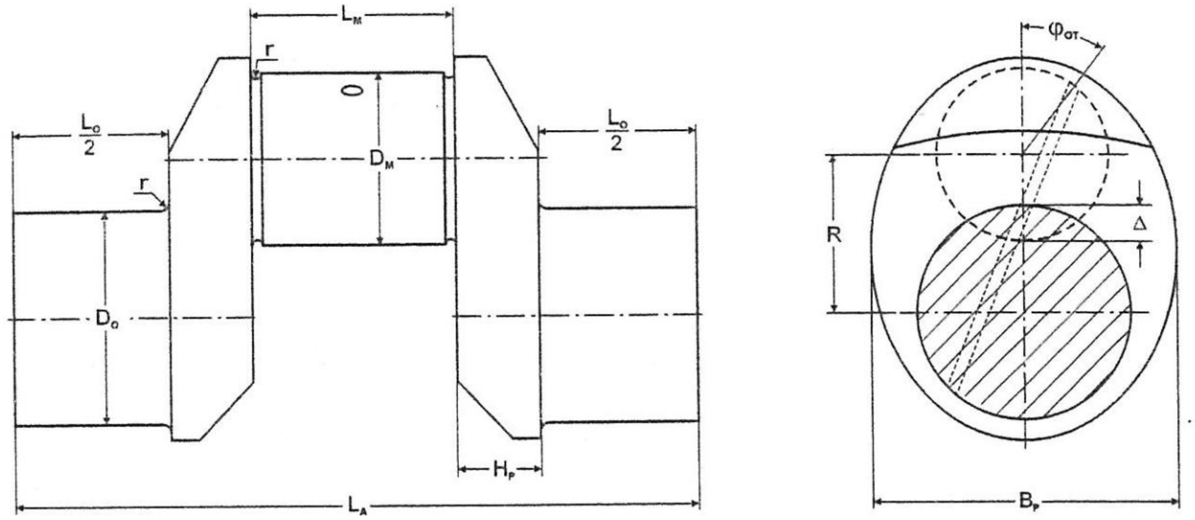


Fig. 55 Basic dimensions of crank throw (60)

	Name	Spark-ignition engines	
		Inline	**V**
D_o	Main journal diameter	(0,60-0,73).D	(0,75-0,85).D
L_o	Main journal length	(0,28-0,36).D	(0,28-0,32).D
D_M	Crank pin diameter	(0,53-0,60).D	(0,55-0,75).D
L_M	Crank pin length	(0,26-0,33).D	(0,25-0,55).D
H_p	Crank web thickness	(0,21-0,26).D	(0,13-0,20).D
B_p	Crank web width	(0,8-1,1).D	(0,9-1,2).D
L_A	Crank throw length	(1,07-1,16).D	(1,1-1,6).D
r	Filler radius	(0,06-0,09).(D_M; D_o), min. 2 mm	

Tab. 6. Relative dimensions of crank for four-stroke engines (61)

For the design of a conventional crankshaft (two main journals for one crankpin), it is enough to observe and design (size) in the first approximation of only one crank. The distance between the axes of two adjacent cylinders (i.e. the bore spacing) has to be taken into account, assuming that the length of the crank throw equals the same distance (bore spacing).

The length of the main journals and connecting rod journals (crank pins) includes the rounds (fillets). The length of the final main journals may differ from that of the intermediate journals. The diameters of both main journals and crankpins have to be determined with respect to the condition of non-zero overlapping Δ. The size of the journal overlap can be calculated by the following formula:

$$\Delta = \frac{D_o + D_M}{2} - R\ [mm], \qquad R = S/2.$$

The outlet of the oil hole in the connecting rod journal must be located in the least loaded area of the journal surface. The position is defined by angle φ_{OT} measured from the vertical axis of the crankshaft and usually is between 30°-50°.

The final dimensioning of the crankshaft is made after the strength calculation. The dimensions of a single crank shown above are enough to carry out a strength calculation using the method of stress concentration factors. Substantial changes are not needed if the selected dimensions are optimal and the crankshaft will be able to withstand the working loads. Otherwise, if the strength is insufficient, the shaft dimensions should be modified (enlarged), respecting the design requirements and logic. Today a variety of computational and FEA methods are used for the estimation of the strength and lifespan of the crankshaft.

Connecting Rod and Connecting Rod Bolts. The connecting rod is an important component of the engine cranktrain and plays a significant role in the kinematics of the mechanism. Its shape and dimensions are largely determined by the size of the components it connects – the piston and the crankshaft. The basic dimensions of the connecting rod, which should be determined at the initial stage are shown in Fig. 56.

The small end of the connecting rod has a cylindrical shape and, in most cases, is monolithic. The diameter of the hole is equal to the diameter of the gudgeon pin, whereas the width of the small end is about 1-2 mm shorter than the distance between the piston bosses. In the case of a floating type piston pin, an aluminium alloy or bronze bushing is pressed into the small end hole. Otherwise, when the piston pin is stationary (fixed) by press-fit into the small end of the connecting rod, the bushing is not necessary. In highly loaded engines, a trapezoidal shape of the small end is used to reduce mean bearing pressures. The joint between the piston pin and the connecting rod is modified by increasing the cross-section of the lower part of the small end and narrowing the upper part.

The shank (stem) of the connecting rod links the small end on the top with the big end on the bottom. The cross-section has an I-shape (or double T, or H-shape opened in a perpendicular direction to the I-shape). The longitudinal axis of the cross-section lies in the plane of swinging of the connecting rod. The final dimensions of the shank are determined by strength calculations.

The bottom (big) end of the connecting rod is usually cut (separated) into two parts, which allows mounting in engines with the most common monolithic crankshafts. Both parts are tightened by connecting rod bolts. In order to preserve the shape of the hole during the reassembling, the connecting rod cap is fixed with precise screws, stud bolts, bolts and nuts, or by cracking the big end, which allows fixing of the cap by cracked fracture surfaces.

The width of the big end is determined by the length of the connecting rod journal and the bearing width. It is usually 0,3-0,6 mm shorter than the length of the journal.

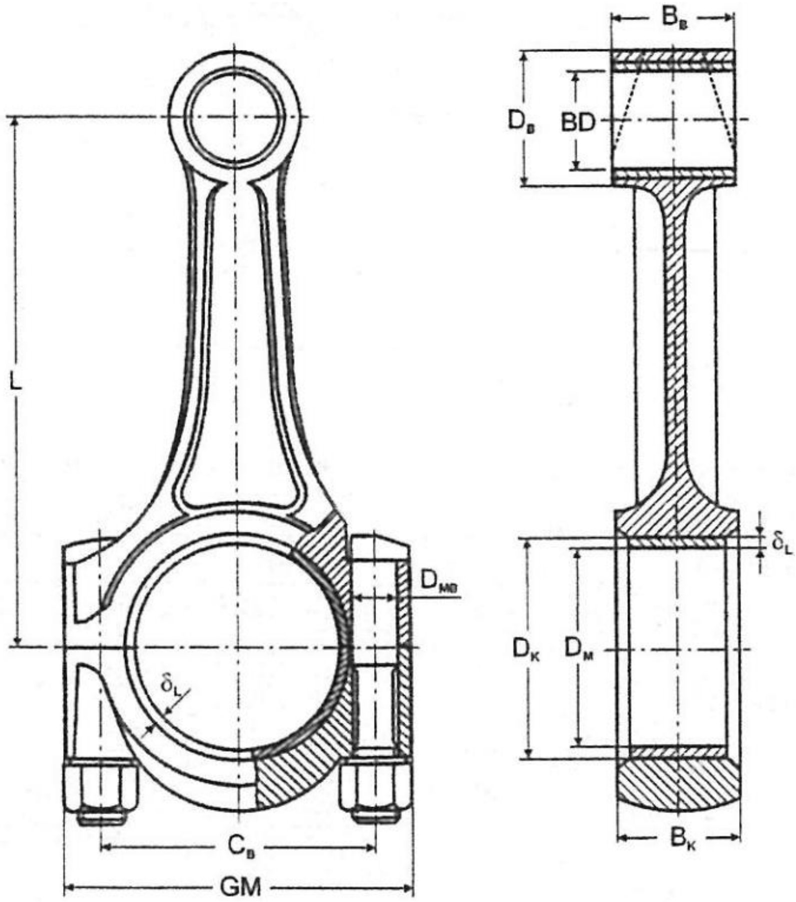


Fig. 56 Basic dimensions of connecting rod: L – connecting rod length; B_B – small end width; D_B – small end diameter; BD – gudgeon (piston) pin diameter; D_k – dig end hole diameter; D_M – connecting rod journal (crankpin) diameter; B_k – big end width; δ_L – connecting rod bearing thickness; D_{MB} – connecting rod bolt diameter; C_B – connecting rod bolts spacing; GM – big end length (62)

The piston is assembled to the connecting rod and then installed in the cylinder in engines with a monolithic cylinder block-crankcase (common cylinder block and upper crankcase). For the connecting rod, in order to pass through the cylinder, it is required to have an outer dimension of the big end smaller than the cylinder bore (diameter). The difference is usually 2-5 mm. This requirement relates to the individual dimensions of the big end very strictly. The diameter of the big end hole is specified by the diameter of the connecting rod journal of the crankshaft and the thickness of the bearing shell. For automotive engines, the thickness of the bearing shells is 1,5-2 mm. The diameter of connecting rod bolts is 8-10 mm.

The hole diameter of the big end and connecting rod bolts can be increased without changing the overall size and width of the connecting rod by chamfering (rounding) the side surfaces of the big end.

In some highly loaded engines, it is necessary to use a larger and more robust connecting rod, which must be split in an inclined plane (usually 45° from the vertical axes) in order to pass through the engine cylinder. This measure reduces the forces in the connecting rod bolts, so their size can be reduced. If this approach is not satisfactory and the dimensions of the connecting rod still do not allow it to pass through the engine cylinder, the piston and connecting rod must be mounted from the underside of the cylinder block, which is additionally split into two parts. Or a three-part connecting rod (marine head) has to be used.

The *length of the connecting rod* is essential for the kinematic characteristics of the engine and its overall height. For example, increasing the length leads to a reduction of the second-order inertial forces but, on the other hand, to an increase in the height and mass of the engine, and vice versa. The length of the connecting rod is initially determined by the chosen value $\lambda = R/L$ coefficient (ratio), which takes into account the finite length of the connecting rod in the calculation of the kinematic properties. The value of this ratio is for automotive engines in the range of 0,23÷0,33. Higher values mean a shorter connecting rod, and in these cases, the length has to be modified with respect to the design and the shape of the cylinder liners and engine crankcase in order to avoid possible collisions. If it is necessary, the length is increased and the ratio λ is corrected in order to calculate engine kinematics.

Design and Construction of Engine Body Components

Engine Cylinders – dimensions, arrangement, construction. In the construction and design of the engine cylinder block, the most important parameters are the dimensions of the cylinder bores, their spacing from each other (bore pitch, bore spacing, or cylinder spacing), and the distance from the top plane (deck) of the block to the crankshaft axis (deck height). The other construction details are specified at a later stage.

The size (diameter) of the cylinder bore is specified during the initial determination of the basic engine parameters and is not usually subject to change during the design process. The length of the engine cylinder depends on the piston dimensions, engine stroke and the requirements for the position of the piston at the top and bottom dead centre. The position of the piston at TDC is determined according to the required volume of the combustion chamber (compression volume).

In modern automotive diesel engines, the combustion chamber is formed mostly in the piston. That means that the distance between the piston crown and the flat mating plane of the cylinder head must be minimal so that there is no contact between them (less than 1 mm). The presence of a cylinder head gasket with a thickness of 1,2÷1,7 mm implies some piston protrusion at the TDC above the top edge of the cylinder. Considering the deformation of the gasket (0,05÷0,07 mm) during tightening of the head, this protrusion is 0,5÷1,2 mm.

In petrol engines, the combustion chamber is usually dome-shaped and formed in the cylinder head (which is not flat-faced). A part of the combustion chamber is sometimes formed also in the piston crown. In all cases, the piston protrusion is $p = 0$, so it is not taken into account during determining the cylinder length.

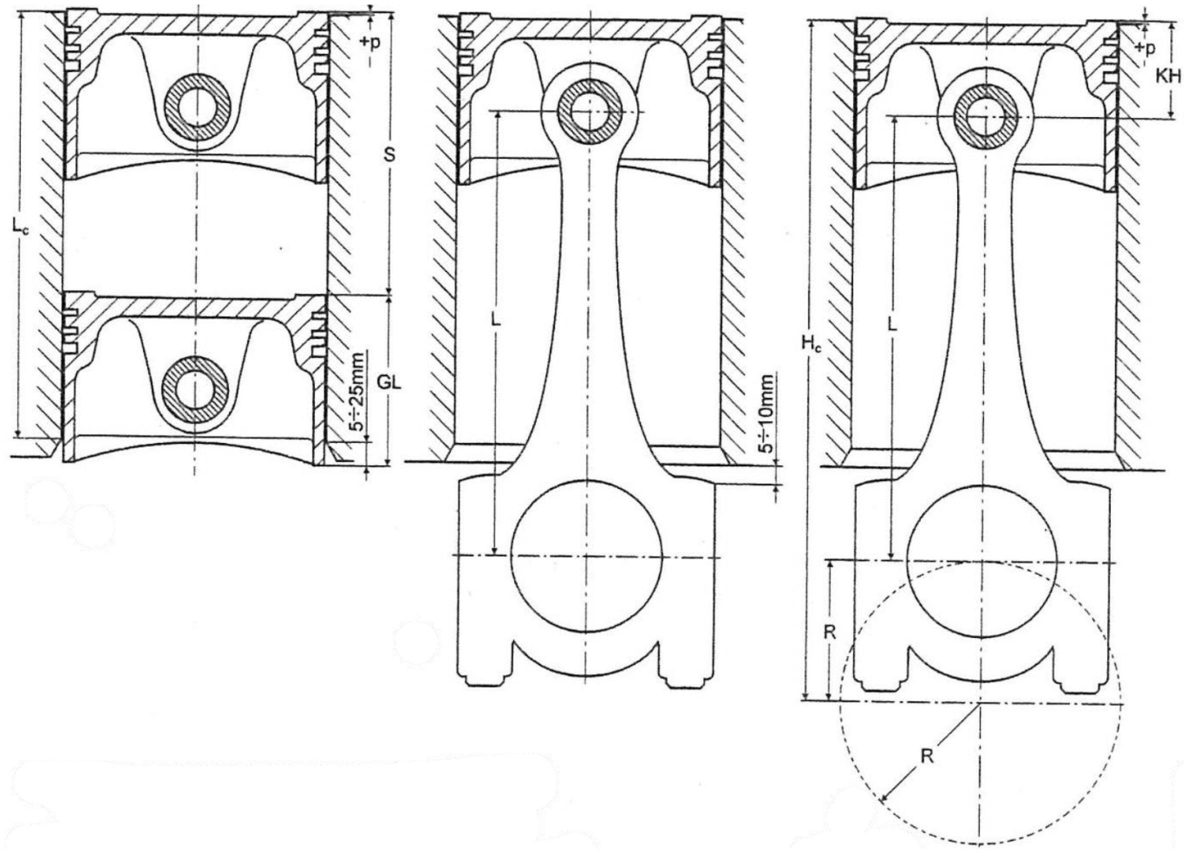


Fig. 57 Determination of main dimension of cranktrain (63)

At the BDC, it is possible for the piston skirt to protrude under the cylinder bore by 5÷25 mm. This is an opportunity to reduce the engine height. An additional rule for the position of the cylinder bottom edge is the requirement for clearance of 5÷10 mm between the edge and the connecting rod big end. This implies a modification of the connecting rod length. The lower end of the cylinder bore does not always match with the edge of the cylinder since there is often a chamfer or round in order to allow free smooth movement of the connecting rod shank (stem). Using these constraints, the length of the cylinder is calculated:

$$L_C = S + GL - p - (5 \div 25)\ [mm].$$

The vertical position of the cylinder is given as the distance from of its upper edge to the axis of rotation of the crankshaft, which is determined by the formula:

$$H_c = KH - p + L + R\ [mm].$$

The cylinder spacing depends on the type and construction of the cylinder liners (sleeves). There are different cylinder sleeve designs (dry or wet) and different variants of installation in the engine block (cast-in-fit, force-fit) and sealing. Some important dimensions and ranges for engines with cylinder bore in the range of 75÷140 mm are presented in Fig. 58.

The distance between the cylinder bores depends on the wall thickness and the size of the cooling space (water jacket). It is usually in a range of 8÷20 mm. However, it is possible to go beyond these limits in some specific design cases because the spacing between the cylinder axes is directly related to the crankshaft (crank throw) length, i.e. the main and connecting rod journals and crank webs must be fitted within this dimension. In some engines with small cylinder diameters, where the cylinder spacing is also small, all the journals with appropriate sizes cannot be fitted. This leads to the use of non-fully (partially) supported crankshafts, which allows the cylinder spacing not to be equal, e.g., to be larger between the cylinders where the main journal is located.

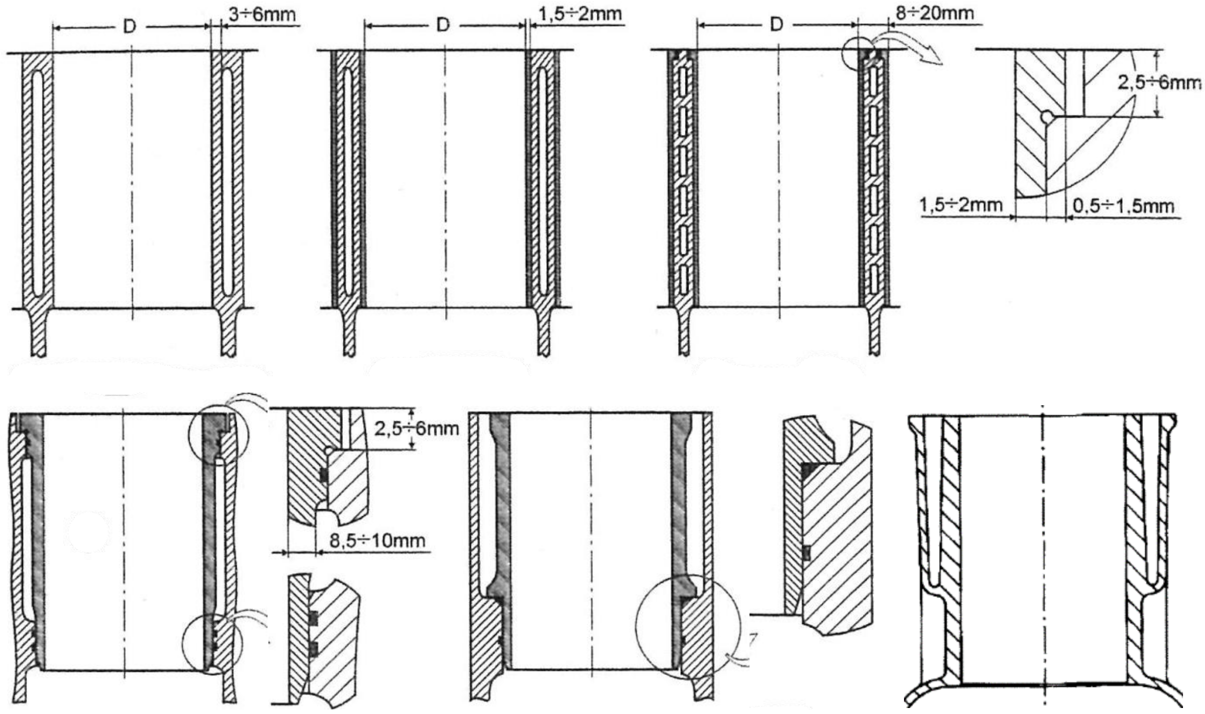


Fig. 58 Different design of cylinder liners (sleeves) (64)

Shape and dimensions of the engine crankcase. The shape of the engine crankcase is determined by the outer contour (envelope) of the connecting rod trajectory, which the outermost points outline during the movement. The minimum distance (clearance) between the crankcase walls and the trajectory contour of the points is between 4÷10 mm.

In order to plot the connecting rod trajectory, it is necessary to know the following: the length of the cylinder and its position relative to the axis of rotation of the crankshaft, the length and shape of the connecting rod and the radius of the crankshaft.

Firstly, the sizing of the crankcase starts with outlining the cylinder bore with its dimensions. Secondly, the position of the centre of the bore of the connecting rod small end at TDC and BDC is determined, and this gives the line in which the small end moves. The centre of the big end bore moves along the circle defined by the radius of the crankshaft and its centre of rotation.

The trajectory of the connecting rod is obtained by placing its silhouette in successive positions defined by the rules of its motion. The centre of the small end is located on the axis of the cylinder in the line between the two dead centres, while the centre of the big end is located at the corresponding position on a circle with radius R. The envelope curve of the moving connecting rod (given by the multiple silhouettes) is the trajectory of its outermost points. In practice, this can be done using graphical computer programs.

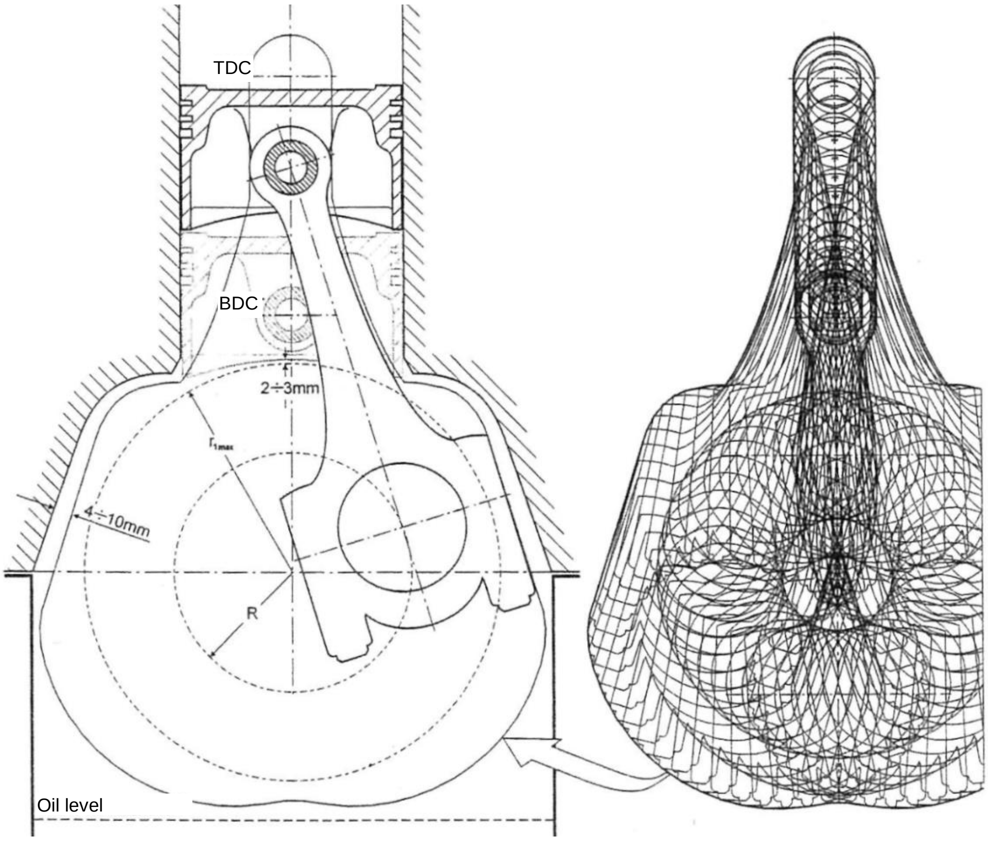


Fig. 59 Conrod trajectory, outer envelope and determination of shape and size of crankcase (65)

After plotting the trajectory of the connecting rod and its outer envelope, the walls of the upper crankcase are formed, respecting the need to ensure sufficient clearance so that the connecting rod does not touch the crankcase. The lower crankcase, which is usually the oil pan, is outlined (formed and sized) with respect to the necessary volume of stored lubricating oil. The position of the oil level is clearly defined since the connecting rod should not dip into the oil as it moves.

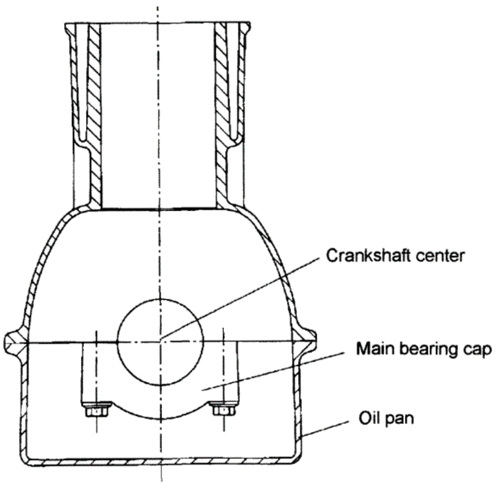


Fig. 60 Engine block design 1 (66)

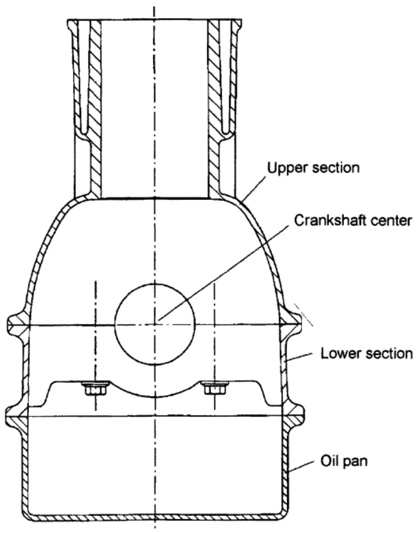


Fig. 61 Engine block design 2 (67)

When the shape and dimensions of the crankcase and piston are already known, and also considering the position of the piston at the BDC the maximum possible radius r_{1max} of the counterweights can be determined. If necessary, this radius can be further increased by cutting the bottom part of the piston in order to ensure divergence from the counterweights during the motion with a minimum distance of 2÷3 mm.

Determination of Masses of Crankshaft Components

The size of the masses of the cranktrain components is important and necessary to determine the inertial forces and moments acting in the mechanism. At this stage of the project, these masses can be defined only theoretically using two approaches.

The first one considers the engine components as a collection of simple three-dimensional bodies with easily determined volumes. After determining the total volume of the component [m^3], the mass can be obtained by multiplying this volume by the density of the material of which the component is made. E.g. the average densities of commonly used metals in engine building are: aluminium – 2700 kg/m^3, cast iron – 7200 kg/m^3, steel – 7850 kg/m^3.

In the second approach, statistical data about the relative masses of components are used for the estimation of the masses. Most often, they are related to the piston surface area in m^2. The values of relative masses for the components of the cranktrain are shown below.

IC engine cranktrain component	Relative masses kg/m^2	
	Petrol ICE D = 60-100 mm	Diesel ICE D = 80-120 mm
Piston group (piston, piston rings, piston pin) – m_{GB}:		
– aluminium alloy pistons	80 - 130	130 - 200
– cast iron pistons	150 - 250	250 - 400
Connecting rod – m_M	100 - 200	200 - 400
Unbalanced masses of single crank w/t CW – m_{KA}		
– steel crankshaft with solid journals	150 - 200	200 - 400
– cast iron crankshaft with hollow journals	100 - 180	150 - 300

Tab. 7. Relative masses of cranktrain components (lower values apply to automotive engines) (68)

In order to determine the mass of a given component, it is necessary to multiply the appropriate value from the table by the piston crown surface area of the specific engine. However, when using values from corresponding ranges, it is still necessary to consider the design features of the particular engine and the relevant components. For example, for greater values of the B/S ratio (short-stroke), the unbalanced masses of the crankshaft are reduced, and therefore smaller values should be used. Or, for lower values of $\lambda = R/L$ (long connecting rod), the larger values should be used, etc.

At this moment, it is possible to carry out an analysis of the kinematics and dynamics properties of the cranktrain, which is presented in Appendix 1. There is also a brief description of the methods for estimating of the friction in the engine.

4.5. General Engine Development Process

The development of a new IC engine is a long and demanding procedure. In automotive engineering, the whole engine development process is divided into many individual successive steps, starting with the definition of the engine specifications and ending with the release of the engine into mass production. A general representation of the development process and a brief overview of the separate processes are necessary to specify the area of interest on which the proposed method is focused. The following description is based on a development flowchart published by the independent research institute AVL List GmbH. However, the own development process of individual automotive manufacturers can differ at some points according to their workflow, specifics, experience, and know-how. Development of a new IC engine starts in the first stage with a concept study. The second stage focuses on the creation of a prototype of an engine based on the initial requirements. At the next stage, the prototype is tested and verified (pre-production activities). And the entire process concludes with the preparation for serial production. The area of interest is the first stage, but other stages are listed briefly for completeness.

I. Design of Engine Concept Study

1. Selection of Basic Engine Concept and Analysis of Needs. The first step is oriented towards the selection of the basic concept of the new IC engine. It is very important to consider the purpose and the usage of the engine, but also to realise all the facts before the start of the design process. First of all, it is necessary to analyse the application of the engine (e.g., what type of vehicle, different vehicle configurations, etc.). Next, according to the particular type of vehicle, it is important to estimate the expected performance (e.g., the engine's power and torque characteristics, operating speed (min^{-1}), engine transient response, etc.). Other factors, such as engine weight and size, engine durability, fuel consumption, maintenance intervals, and requirements, have to be considered. And finally, everything has to be in line with the cost targets in order to make the product competitive.

2. Initial Thermodynamic Calculation. The answers to all questions about the overall engine concept will provide input data for the next step when an initial thermodynamic analysis of the engine is conducted. A decision about the **engine type and its ignition system** is made (spark-ignition vs. compression-ignition) according to the particular vehicle application, performance expectations, target market, available fuel types and their prices, and current valid emission standards. A common spark-ignition engine, i.e., is simpler in design, more lightweight, and thus cheaper, but it produces a reasonable amount of power. It also meets the emission standards more easily. Next, an **initial engine analysis** is performed in order to determine the engine displacement needed to generate a specific amount of power at a specific engine speed according to particular requirements. These activities are very closely related to the next ones – the engine layout design. This is a critical part of the early-stage design, and the conclusions affect both the mechanical design and the performance characteristics of the engine.

3. Initial Mechanical Calculation. As stated above, the engine layout and the performance analysis are inseparable from each other. The determination of the cylinder bore and piston stroke together with the configuration of the cylinders resolve the layout of the engine. Another important early-stage design decision is about the **engine layout**. The distribution of the engine displacement to a specific number of cylinders, their arrangement (position and orientation) and size (bore and stroke of each cylinder) are essential for the mechanical design of the engine, as well as for its performance characteristics. In any case, the estimation of these parameters has a significant effect on engine design and complexity, operation speed range, force and moment balance, final costs, etc. The preliminary layout design has to be carried out at the same time. It is clear that most decisions depend on the particular application of the designed engine.

4. Fixing Engine Displacement and Configuration. This step follows the initial analysis of the engine performance and layout. It presents a basis for the next steps of the engine design but also for the preparation of the tooling design process.

II. Development of Engine Prototype

5. Engine Concept Design. At this phase of the development process, a few essential dimensions which affect different aspects of the overall engine design, operating loads and their distribution, durability, etc. (e.g., the cylinder-to-cylinder spacing, the deck height, camshaft configuration, etc.) are resolved. At this moment in the layout design, the size of the engine unit is refined (outer package dimensions vs. vehicle engine compartment space). For the next design decisions, it is important to select suitable materials for the main components (cylinder head and engine block). This will affect the overall structure and rigidity of the engine.

6. Engine Thermodynamic Layout. Many different computational and simulation tools are used for initial engine modelling, design, and analysis before physical engine hardware building. It is necessary to use these tools, together with prior experience and knowledge, to design and optimise the engine, e.g., operating cycle (combustion process, heat transfer, efficiency, etc.), cranktrain system, valvetrain system (valve timing, cam profiles), intake and exhaust system characteristics (runner lengths, gas motion), etc., in order to shorten the time needed for development. The stumbling block to the utilisation of these tools could be their previous precise validation and calibration from experimental data – there are many different parameters and a mixture of empirical and physically-based models. It is also possible to conduct additional modifications to the engine design in the future with these tools and models.

7. Engine Mechanical Analysis. At this stage, many different simulation tools are also used for structural analysis of the design, evaluation of the fatigue limits and component durability under mechanical and thermal loading before physical testing. Simulations are also used for the evaluation of the dynamic characteristics of the engine cranktrain and valvetrain. The fluid dynamics of the engine cooling and lubrication systems are also evaluated and optimised by suitable simulation tools. Also, these models have to be verified, calibrated and updated by the measurement data from engine testing in order to increase the precision of the models for a further design modification.

8. Acoustic Analysis. The study of noise, vibration, and harshness (NVH) properties is very important, especially in vehicular applications. These properties depend on many different factors, such as the engine's component design, the engine assembly itself, and its mounting inside the vehicle. The selected engine configuration (number of cylinders and their position) specifies the acting mechanical forces and moments inside and outside the engine, their balancing, but also remaining unbalanced forces and moments (magnitude and frequency) are transmitted inside the vehicle through the engine mounts. The critical noise and vibration frequencies are found by modal analysis. After that, some changes to the design of the components can be made in order to minimise vibration amplitudes and to move the resonance frequency range outside the engine operating speeds.

9. Construction of Production Tooling. The separate design and construction process of the manufacturing tooling starts when all crucial dimensions of the engine are fixed during the initial process of engine layout design.

10. Detailed Design and Cost Estimation. A detailed proposal of individual engine components which provides the final dimensions and tolerances is conducted according to the initial design layout. Everything is prepared for manufacturing – technical drawings and documentation, material selection and tests, specification of the manufacturing process, etc.

11. Engine Breadboard Testing. Despite continuously improving techniques for the analysis of engine performance, it is still essential to supply the analytical studies with experimental data in order to validate and improve them. At this early stage of development, when production-like prototypes are not available yet, breadboard testing is the most suitable way to start. This testing consists of a modification of the current engine using prototype components in order to emulate the new one as closely as possible.

12. Rig Testing. For verification of durability, each engine component is proven on a test rig. The engine components are fixed according to the boundary conditions and loaded in the same manner as in the engine and are trialled according to a particular methodology (set of required tests and their evaluation). As a result, the components are well optimised before being testing in the engine.

13. Prototype Builds. After finishing the engine layout design and initial analysis, it is time to assemble the first several engine prototypes from prototype components. They will serve as samples for advanced tests and analyses of performance and mechanical characteristics in an engine testing laboratory or a particular vehicle.

14. Engine Performance and Emission Development Resting. According to the precision and scope of the initial analysis of the engine, the combustion process and performance are examined and optimised. The engine is prepared for emission certification. Testing and optimisation of the engine are performed on engine and chassis dynamometers, as well as in real driving conditions.

15. Function and durability testing. The next step in engine testing is engine validation, based on structural analyses and rig trials done earlier. Usually, there are specified methodologies for validation and qualification testing (sequence of various steady-state and dynamic tests that evaluate the durability of the components).

16. NVH testing and analysis. An important point in testing and development is the NVH characteristics of the prototype engine itself, but also of the vehicle where the engine is installed in practice. The properties of the system are analysed both by simulation models and physical testing.

17. Completion of manufacturing tooling. At this stage, the final construction and qualification of the manufacturing tooling are done and it is ready for release into production.

III. Pre-production Development

18. Pre-Production Engine Builds. Building a small pre-production series of engines with the production tooling for further testing, evaluation and certification.

IV. Validation for Series Production

19. Validation and Release into Production. This is the final major step when the IC engine is prepared and released to series production.

4.6. Summary

As it is obvious, the design and development of a new IC engine is an extremely complex and demanding task. In essence, it is a multidisciplinary process that connects almost all fields and disciplines of mechanical engineering, such as statics, kinematics, dynamics, heat transfer, thermodynamics, fluid dynamics, material science, strength of materials, mechanical design, acoustics, manufacturing, etc. However, the doctoral dissertation and the considered design workflow focus on the first stages of the development process – the ICE concept study and design – including the selection of basic conception, initial thermodynamic calculation, initial mechanical calculation, determination of engine displacement and configuration, and preliminary conceptual design.

5. Simulation of Electric Vehicle with Range Extender

Analysis of the entire vehicle is a quite complex and demanding task. It involves advanced modelling and simulation of the vehicle systems from the vehicle concept, its dynamics and control, powertrain and drivetrain, etc. The aim is to study, propose, evaluate, and optimise vehicle driving characteristics, performance (acceleration, deceleration, etc.), fuel economy and emissions (driving cycles, real-drive cycles), transmission dynamics (gears, shifting, control), drivetrain dynamics (torsional vibrations, weight distribution, tyre traction, etc.), and engine and powertrain control systems and strategy. [32, 33, 42, 68]

There is a wide range of simulation software that can be used to perform various vehicle analyses and studies at different system levels during the design process. In the following example, it is GT-SUITE from Gamma Technologies, which is a versatile multi-physics modelling and simulation tool with many capabilities focused on the automotive industry. An initial analysis and comparison between a traditional pure battery electric vehicle (BEV) and an electric vehicle equipped with a small range extender (REx) has been carried out, using a simple simulation model to explore the vehicles' properties. [16]

The main task of this study is to find out what power is needed (approx.) for the REx engine. All vehicle tests are carried out under the conditions of both the driving cycles (NEDC and WLTP), and allow us to compare how vehicles perform under various circumstances.

Vehicle model description. There are several possible layouts for an EV, but only the most common one is considered and explored by the selected simulation model. The vehicle is equipped with an electric powertrain, including a battery system for supplying electricity and a single traction electric motor that drives the wheels through a drive shaft, gears, and differential. The powertrain is next extended with a REx unit, consisting of an IC engine, an electric generator, and a controlling unit. The layout is similar to that of series hybrids.

Main vehicle parameters have been retrieved from the **BMW i3** series electric car. This is due to the general availability of this electric vehicle, the existence of both purely electric version and electric one with a range extender, as well as the availability of numerous independent research, measurements, and data. The simulation models are adjusted roughly to correspond to the characteristics of the BMW i3 vehicle. Because the simulation and optimisation of EVs is not a main goal of the doctoral dissertation, the simulation model used is not calibrated or verified in any way, and the results are only provided as an example. [58, 75, 76] The BMW model i3 is the most popular and widely available electric vehicle, having been produced since 2013 in various versions. There are two main options: a pure electric vehicle (i3) and an electric vehicle with a range extender (i3 REx). They are powered by an electric motor with a maximum power of 125 kW or 135 kW (for Sport variants), supported by an electric battery with a usable capacity of 18,8 kWh (60 Ah), 27,2 kWh (94 Ah), and 37,9 kWh (120 Ah). The overall real-driving range varies from approximately 130 km to 260 km for electric versions, and 240 to 330 km for the REx versions. The average energy consumption based on real driving data is around 17 kWh/100 km (13 kWh/100 km in WLTP). [57]

The BMW i3 REx is equipped with a range extender unit. The electric generator, with a peak power of 26,6 kW at 5000 min^{-1} is driven by a twin-cylinder, four-stroke, naturally aspirated spark-ignition engine W20K06U0 with a displacement of 647 cm^3 which produces a power of 28 kW at 5000 min^{-1} and torque of 55 Nm at 4500 min^{-1}. The compression ratio of the engine is 11,6. The vehicle's range can be increased by 120 – 150 km by using a fuel tank with a capacity of 9 litres of petrol. The total weight of the REx unit is 120 kg. [57]

The vehicle runs in pure electric mode until a specific state-of-charge of the electric battery is reached (charge-depleting mode). Then the REx starts operating in order to recharge the battery (charge-sustaining mode). Due to the lower system power of the REx, the vehicle has limited performance, e.g., it is not capable of high-speed operation (on the motorway). [57]

Using a simplified simulation model, three different configurations have been examined:

– an electric vehicle with a battery capacity of 60 Ah battery;

– the same electric vehicle equipped with a range extender installed;

– the same vehicle with a higher capacity battery of 94 Ah.

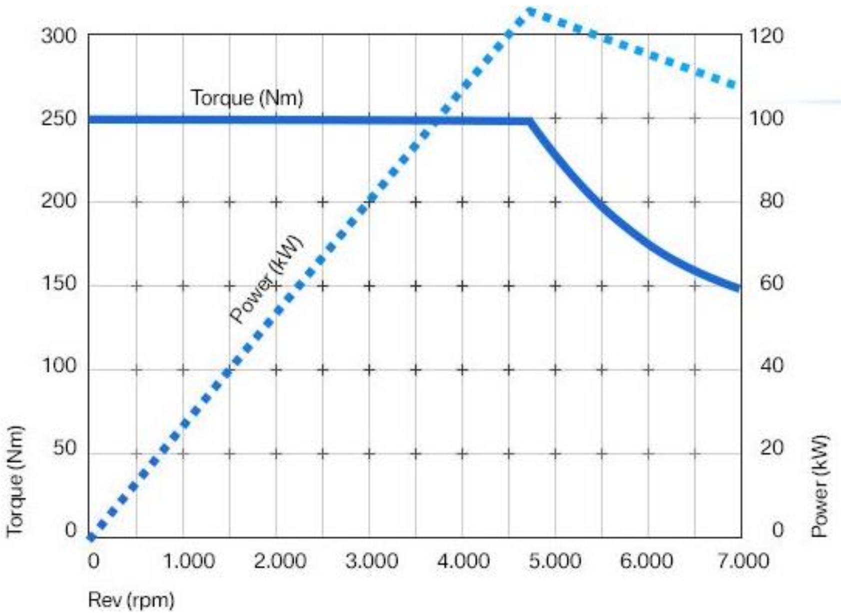


Fig. 62 BMW i3 electric motor power and torque curves (69)

The power and torque curves of the i3 electric motor are presented in Fig. 62. The considered simulation model in GT-SUITE is shown in Fig. 63. It consists of different components that define the electric vehicle and its subsystems: the battery, electric traction motor, IC engine, and electric generator, control management system, etc.

	BEV 60 Ah	BEV 94 Ah	Rex 60 Ah
Vehicle weight [kg]	1270	1320	1390
Battery weight [kg]	230	280	230
REx weight [kg]	-	-	120
Electric motor power [kW/rpm]	125/4800	125/4800	125/4800

Tab. 8. The specification of the considered electric vehicles (70)

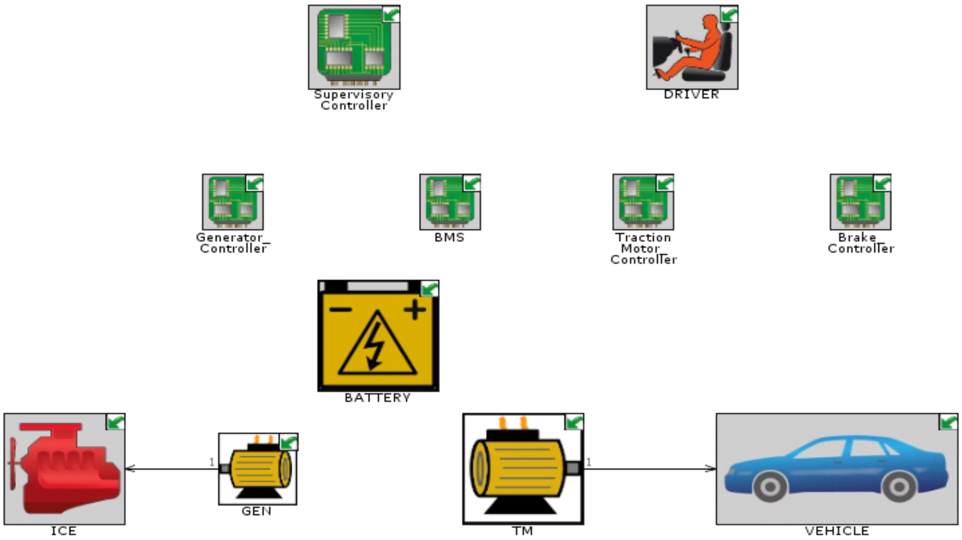
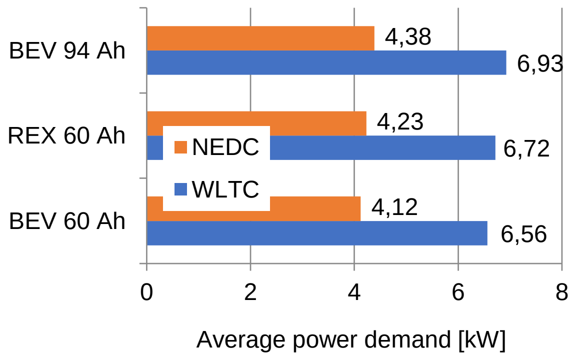


Fig. 63 A vehicle simulation model in GT-SUITE (71)

Firstly, the model is used to calculate the traction power demand to run a specified driving cycle. The computed results show that the average power that a BEV with a 60 Ah battery needs to travel on the NEDC cycle is 4,1 kW and 6,6 kW on the WLTP cycle. For the BEV with a 94 Ah battery, which is 50 kg heavier, it is 4,2 kW and 6,7 kW, respectively. The BEVx equipped with a 60 Ah battery and range extender (an additional 120 kg) requires on average 4,4 kW in NEDC and 6,9 kW in WLTP. The peak power demand varies from 34 to 43 kW.



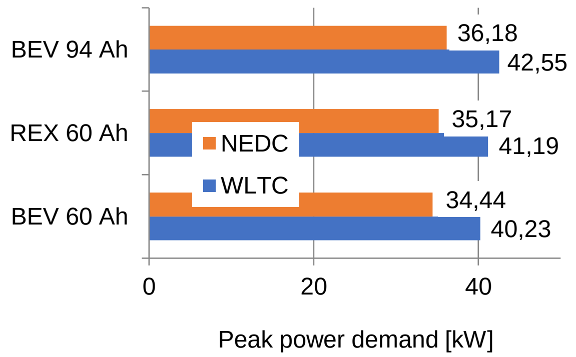


Fig. 64 The average traction power demand (72) **Fig. 65** The peak traction power demand (73)

The analysis shows the differences in reachable electric range just by changing the size of the battery and the contribution of using a range extender instead of increasing the battery size. This quick study also reveals some preliminary parameters of the range extender unit. For simplicity, a single simulation model is used to perform all analyses. The vehicle parameters are modified in each case. In the pure EV, the IC engine is turned off. [16]

The overall EV range is estimated by performing the specific driving cycle a few times in a sequence until the electric battery is fully discharged. The results show that by increasing the capacity of the battery from 18,8 kWh (60 Ah) to 27,2 kWh (94 Ah), i.e., adding an additional 50 kg of weight, the EV can drive 65/53 km more in the NEDC/WLTP accordingly (i.e., +53 %).

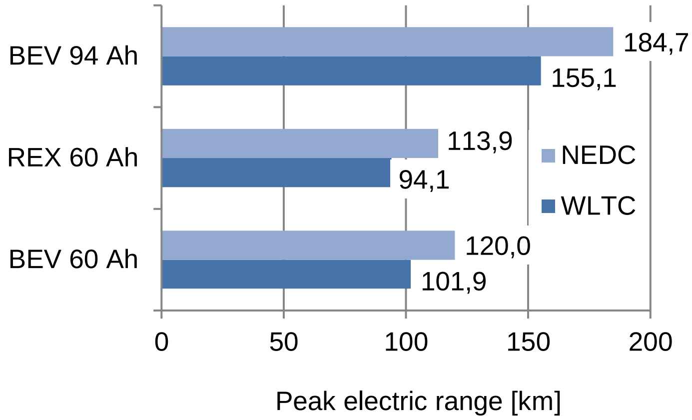


Fig. 66 Vehicle electric range (74)

One more important question appears here: what amount of power does the IC engine in the REx unit have to generate and how the engine operation should be controlled: when to start/stop it in order to guarantee an appropriate extension of the range. The preliminary results indicate that an engine with an output power of about 20 kW to 30 kW has to be enough to ensure an acceptable extension for the vehicle segment. For this model example, only one engine operating point is used (power of 20 kW). Obviously, it is important to develop a more advanced control strategy for the range extender (e.g., more than one operating point).

Since the parameters and maps of the IC engine are unknown, the simulation model is simplified, and the IC engine is represented by simple mechanical components that simulate its function just by applying torque and speed to a rotating shaft. In the electric mode, the state-of-charge (SoC) of the battery is set to 100% (fully charged). The simulation runs the vehicle in the specified driving cycle a few times until the battery is completely depleted (fully discharged). In the REx mode, the REx starts when a specific battery SoC is reached. The total vehicle range is presented in the graph below.

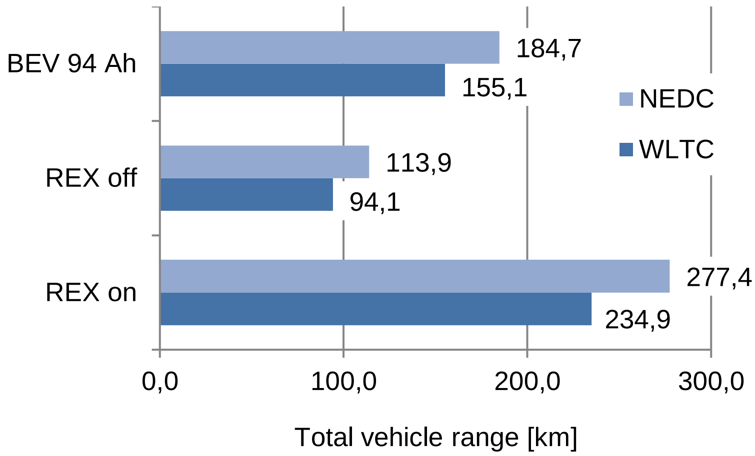


Fig. 67 Comparison of the total vehicle range (75)

The model example shows roughly what amount of power is needed to extend the vehicle's range. A 20 kW IC engine can provide an approximately 50% longer travel range in both NEDC (+93 km) and WLTP (+80 km) compared to the BEV 94 Ah.

An easy way for extending the range of an EREV while keeping the rest of the system the same is to increase the capacity of the fuel tank. This solution benefits from low added weight and volume. Another way to improve the range is the enhancement of the engine fuel efficiency. These and other related issues, such as model calibration and testing, estimation of energy consumption and vehicle range, variation of battery capacity, battery ageing effect, optimisation of the drivetrain by using a two-speed gearbox, and so on, are discussed in more detail in [57]. Different control approaches are discussed in the author's paper [16].

Conclusions. As is well known, the motor vehicle is a very complex machine. Its proposal and design include many multidisciplinary tasks across scientific disciplines. That is why there is a wide variety of simulation tools used for vehicle analysis. Since the main object of the doctoral dissertation is not the whole vehicle but rather the IC engine for the range extender unit, this chapter presents briefly a model example of an electric vehicle with a range extender. This essential task aims at the determination of the power demands of the vehicle powertrain as a part of the development vehicle process and design workflow. A REx unit with a 20 to 30 kW IC engine can ensure a sufficient range extension. The result corresponds with the power produced by the REx units available so far (see Chapter 2).

Fig. 68 BEV vs. EREV Range Comparison (76)

The results from the study confirm the fact that EVs with REx have great potential, and it is noteworthy to pay more attention to this viable solution for the future of personal mobility. A big question (or challenge) can be observed in this context. An important factor for further decision-making is the relationship between the vehicle range and vehicle weight. It depends on many design and technological factors. It is important to decide at the beginning if it is worth it to go for a higher capacity (heavier) battery or for a REx unit solution (additional finite weight) in terms of added range, weight, and cost. This implies, for example, a requirement for the complete REx unit (IC engine, electric generator, and control electronics) to weigh as little as possible. Or at least, the weight of the small battery and REx unit should be lower than that of the higher capacity (bigger) battery. The optimisation of the weight and cost of every solution is challenging. Of course, there is always a trade-off between the available means and particular requirements. In Fig. 68, the situation of the model example is presented graphically. Fig. 69 presents the result from the analysis by KSPG. [3]

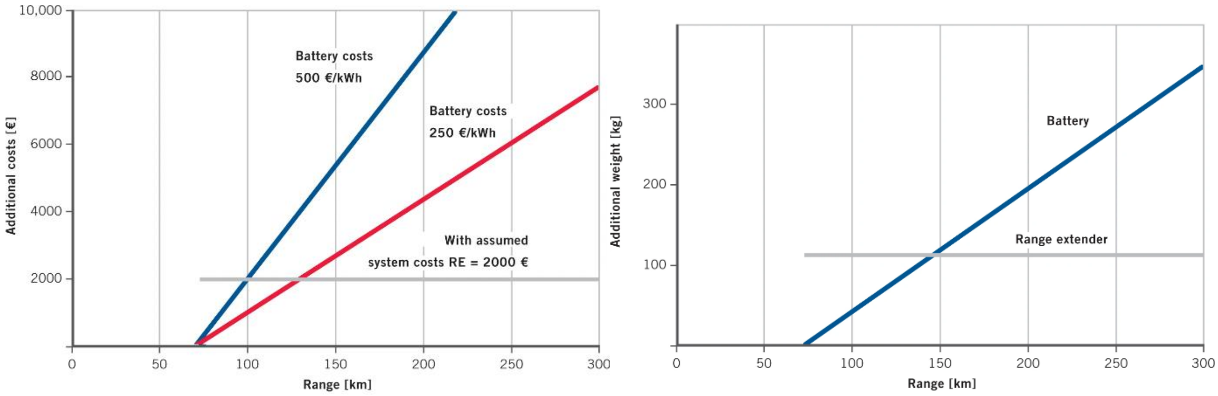


Fig. 69 BEV vs. EREV Range Comparison (71)

6. Simulation of IC Engine for Range Extender

When designing a new internal combustion engine, it is essential to obtain an overall view of its performance. Considering all the data about friction, heat transfer, and combustion, the effects of various factors (parameters) on the engine's performance can be explored in the initial phase of design. These parameters are the engine size (displacement), number of cylinders and their geometry (bore and stroke), piston speed, engine load, compression ratio, valve and ignition timing. Proper dimensioning of an ICE is important with respect to its particular utilisation. Therefore, it requires a comprehensive study of a particular combustion engine itself and the propulsion system in which it is installed.

For a newly designed ICE, various simulation studies are carried out using specialised software. These computer programs use known theory, gained knowledge and experience, together with validated mathematical models, to describe the processes inside the engine. The aim of simulations is, first of all, to predict the gas flow (dynamics), combustion process, heat transfer, and overall engine performance, allowing their evaluation before the development of a real prototype engine that can be tested and measured. Since the ICE is a very sophisticated machine, computer simulation can be an extremely complex and demanding assignment. Modern simulation systems (e.g., Gamma Technologies GT-SUITE, AVL Simulation Suite, Lotus Engine Simulation) are already very complex and accurate enough, offering capabilities and functionalities for ICE development. The simulation is not only limited to the analysis of the engine thermodynamics, combustion process, and performance but can also include analyses of emissions formation and aftertreatment, friction, lubrication, and cooling, as well as mechanical systems (cranktrain, valvetrain and timing, balancing), acoustics, etc. With the use of an appropriate calibration of the simulation models, the results could correspond with reality (mathematical models are experimentally verified on a wide range of production ICE).

Following the steps of the intended design workflow, the simulations at this stage of the design process are used to obtain results from an initial engine analysis. These results are essential for the initiation of the mechanical design of the engine and its components, which is the next step. The aim in our case is to find preliminary optimum operating points of the ICE with regard to its use in a REx unit (no direct connection to the wheels) and performance expectations, and to predict the course of cylinder pressure and loading forces. Since this is an intermediate step in the workflow, the doctoral dissertation is not primarily focused on advanced simulations and optimisation of engine parameters. The particular work builds on extensive experience with parametric engine design, simulation, and optimisation studies at the Czech Technical University in Prague, e.g., the study of engine downsizing limits [9], or the study of the thermodynamic potential of a turbocharged small engine [69]. Furthermore, a comprehensive thermodynamic study of the parameters of ICEs intended for a REx unit has been performed and presented with relevant conclusions in 2018 and 2021. This study of the engine parameters was followed by multi-parameter and multi-objective optimisation, which searched for optimum parameters with a high degree of freedom. [60, 61, 62] The motivation for this was the fact that the research on ICEs for REx units done so far has been focused in most cases on the engine configuration and design in terms of simplicity, NVH properties, and overall costs (Chapter 2).

6.1. Basic Principles of Modelling the IC Engine Cycle

The history of numerical calculations and simulations of the ICE operating cycle began more than half a century ago with the progress in computing science and technology, as well as with the research and improvement of knowledge about the individual processes inside the engine. This has led to the creation of sophisticated simulation tools widely used in practice, offering multidisciplinary features (combustion, chemistry, heat transfer, friction, etc.).

Simulation of the ICE thermodynamic cycle is the most common type because it uses a 0D/1D modelling approach and is quite sufficient for our case. This kind of simulation can provide decent results in a short time. Therefore, it is used for general thermodynamic studies exploring a wide range of engine parameters. The simulation models focus on in-cylinder processes. The thermodynamic system (cylinder content) is described by the first law of thermodynamics and from it derived expressions (time derivatives) for the pressure, temperature, volume, and mass as functions of time (crank angle) respecting the engine design variables, operating conditions, and submodel parameters. Depending on the formulation of the thermodynamic relations (a set of ordinary differential equations), the simulation models are one, two, or multi-zone models. In order to solve the obtained system of ordinary differential equations, the thermodynamic properties, cylinder volume and rate of its change, surface areas, combustion process, cylinder heat transfer, mass flow rates, and friction are described with additional submodels. These submodels (mentioned below) are sufficient to describe mathematically the IC engine and perform an essential analysis of its parameters and performance (power, torque, BMEP, BSFC, etc.).

Main Engine Geometry. The geometric parameters of the reciprocating IC engines related to their size have already been presented (Chapter 4.3.). These parameters influence the overall engine nature, the geometry of the combustion chamber, and the resulting engine power and torque. As mentioned, the cylinder bore B and the piston stroke S physically define the engine cylinder, its displacement volume, and its compression ratio. The stroke and mean piston speed define the engine's operating speed [min^{-1}]. The actual volume of the cylinder and the rate of its change can be found from the engine kinematics (app. 1) together with the actual cylinder surface area. As a first approximation, various empirical or experimental relations are commonly used to relate the engine parameters to the cylinder bore.

Combustion Process and Heat Transfer: The simulation model considers an idealised spark-ignition combustion process, which can be described by a single or two-zone heat release model. The heat release diagram is an S-shaped curve represented by a single Wiebe function (for SI engines). [32, 42, 71] It defines the heat release (burn) rate (mass fraction of fuel/mixture burned) as a function of crank angle. The simulation program works with this curve by setting values for two parameters: combustion duration and phasing angle (in crank angles).

Cylinder Heat Transfer: The process of heat transfer is a very complex phenomenon, especially in the engine cylinder where the piston moves. The modelling of the heat transfer process inside the engine cylinder (convective) is described by empirically derived correlations. The most common is the correlation of Woschni, which estimates the heat transfer coefficient between the gas inside the cylinder and its walls. A part of heat transfer modelling involves the specification of surface areas and temperatures of the components. More sophisticated submodels use predictive functions and simplified geometry models of cylinder surfaces.

Mass Flow Rates: The gas flow submodel is considered quite simplified in comparison with the real gas flow. The flow is assumed to be one-dimensional, quasi-steady and reversible. Hence, the flow relations are corrected by empirical flow (discharge) coefficients (typically C_D), defined as a ratio between the actual and ideal mass flow rates. The flow coefficients for the inlet and exhaust valves depend on the ratio between the valves' lift and the valves' diameter.

Friction: The estimation of engine friction losses is necessary for the determination of the engine output (brake) quantities. Simulation programs usually use various empirical equations, numerical models, or predictive modelling algorithms and techniques. They depend on engine parameters such as operating speed, piston speed and/or peak cylinder pressure. A common one is the Chen-Flynn friction model (chap. 9.2). [20] The numerical model Vyvaž is an example of an in-house friction model from CTU, which is also used with success in the REx IC engine simulation studies. [44, 60]

Additional Simulation Submodels: Other submodels can be used to extend the capabilities of the ICE simulation to include additional features (e.g. exhaust gas composition, exhaust gas recirculation, knocking phenomenon (reaction kinetics), etc.).

6.2. Study of Twin-cylinder IC Engine for Range Extender Unit

As previously stated, the range extender unit has to meet a variety of requirements. Most of its characteristics depend directly on the IC engine itself, such as efficiency, fuel consumption, emissions, cost, assembly size and weight, NVH properties, etc. This calls for a compact and highly efficient ICE, which in turn contradicts its complexity and cost. Various research studies have shown that the most suitable engine for this particular application is the spark-ignition naturally aspirated engine because of its simplicity and undemanding, low-cost design. As a result, the following simulation study of a model example of this type of engine was performed to bring us closer to the problems and provide a better understanding. Since there are many unknown parameters at the beginning of the design stage, it is necessary to make some assumptions, approximations, or simplifications, or even use already available data to complete the initial analysis. Their detailed proposal is a part of the subsequent design stages. The aim of this chapter is to generalise findings from ICE for REx simulation studies.

Let us first create the assignment for the model example. The objective is to propose a new SI engine suitable for a REx unit using existing cylinder units. Considering the design workflow, the following input data are used for this task:

- required system power - obtained as a result of the vehicle analysis (chap. 5.),

- engine displacement needed to generate the desired amount of power – can be initially determined using the described approach (chap. 4.),

- main engine data (parameters) were taken over from a sample engine – the twin-cylinder REx engine designed at ŠKODA AUTO a. s. (Ch. 2.3.), presented in Tab. 9. [25, 38]

Parameter	Unit	Value
Engine type	[-]	Naturally aspired, spark-ignition
Engine layout	[-]	Inline-twin
Fuel type	[-]	Petrol
Injection type	[-]	Multi-point port injection
Ignition interval	[°]	0 - 180
Bore, B	[mm]	74,5
Stroke, S	[mm]	91,6 (original 76,4)
Bore/Stroke ratio, ξ	[-]	0,813 (1,230)
Number of cylinders	[-]	2
Cylinder displacement	[cm^3]	399,3
Engine displacement	[cm^3]	798,6
Crank radius, R	[mm]	45,8
Compression ratio	[-]	12
Connecting rod length	[mm]	137
Connecting rod ratio	[-]	0,33

Tab. 9 Main parameters of inline twin-cylinder engine designed in Škoda Auto (72)

Considering all the data and assumptions, the newly proposed engine has to deliver around 30 kW of power at mid-range engine speed, ca. 4000 min^{-1}. From the physical point of view, a more important quantity is the mean piston speed c_s, which in this case equals to 12,2 m/s. It can be assumed that the specific fuel consumption at that operation regime could be around 240 g/kW^{-1}.h^{-1} and the engine volumetric efficiency of around 90%. These requirements, data and assumptions give an estimation of the displacement volume of 800 cm^3. The model example corresponds with the sample engine.

<u>**Preparation of Simulation Models:**</u> The simulation models used in this study represent the particular type of engine by following the indicated specifications. The models are used to perform some initial sensitivity analyses of the engine parameters and to examine

the effects of some of them on the engine's performance and operating characteristics. A relatively simple but useful tool is used to build the simulation models, i.e., the Concept Builder tool from the Lotus Engine Simulation package. This tool allows the user to quickly understand and set the initial values for the main parameters according to the requirements and to prepare a simulation model. It includes cylinder geometry, intake and exhaust systems, gas flow parameters, etc. Additional submodels describe processes inside the engine. The program is based on known engine theory and experience and could therefore serve as a good starting point for the engine development process. Three essential parameters are needed to prepare the simulation models using this tool: the number of cylinders, the engine displacement needed to generate the desired (target) power, and the particular engine speed. [32, 42, 43]

Simulation study of twin-cylinder IC engine for REx: Since the engine displacement needed to generate a desired amount of power is already known (roughly), the purpose of the following ICE simulation study is to analyse three design cases. Following the intended design workflow and considering the reference REx engine from Škoda, this study compares several engine configurations in terms of cylinder geometry, focusing on a range of operating and design parameters, such as fuel consumption or efficiency.

Engine Parameters and Operating Conditions. For the design and proposal of a new ICE, respecting the physical limitations (constraints), it is advantageous to apply gained experience, generalised by the use of similarity rules. It is common to use quantities (i.e., BMEP) independent of the ICE size (i.e., engine displacement), which facilitates the comparison of different engine variants. Geometric similarity considers identical geometric parameters (compression ratio, bore/stroke ratio, connecting rod ratio, valve timing, etc.) and allows the ICE to be characterised by a single parameter, typically the cylinder bore B, while other geometric parameters are related to it. Kinematic similarity of ICEs is usually fulfilled by observing the value of mean piston speed c_s (which depends on the ICE configuration itself, i.e., the motion of the piston) and ensures similar inertia forces and friction wear. Thermodynamic similarity is ensured by identical brake mean effective pressure p_e (specific volumetric work of the cycle). Similar ICEs are also assumed to be made of materials with identical density and strength, which results in similar mechanical stresses on the engine components (from gas and inertial forces). For similar engines and identical mechanical stresses on components, the following conditions apply:

$$p_e = const, \quad c_s = const, \quad \frac{B}{S}(\xi) = const \quad and \quad \lambda = const$$

$$(hence \quad B.n = const \quad and \quad c_s.p_e = const)$$

Similar engines are also assumed to have identical types of fuel, combustion systems, and charge exchange systems. The considered ICEs are intended as two-cylinder, spark-ignition, naturally aspirated, with four valves per cylinder. The compression ratio is set to a higher value of 12:1 in order to improve the engine's efficiency and performance. Increasing the compression ratio leads to a decrease in brake-specific fuel consumption. The higher the compression ratio, the better the fuel efficiency. However, the CR compression ratio cannot be increased unlimitedly. The higher compression ratio negatively affects the heat and friction losses. The ignition timing (spark advance) has to be set for optimum efficiency (equivalence factor λ = 1). Optimum compression ratios (ca. 18:1) for SI ICE are never reached in order to avoid the knock phenomenon, and they are limited to 12–13:1.

Common values of brake mean effective pressure vary for the spark-ignition naturally aspirated ICE in the range of 1,0-1,2 MPa. The mean piston speed c_s is an important parameter. It has to be carefully selected and constrained (upper value) with regard to the expected life of the engine and to avoid possible issues associated with the wear of the sliding surfaces, dynamic force loading of the cranktrain (i.e., higher attention on engine balance, mechanical losses, torsional vibrations, etc. is needed for higher values of c_s), the effect on thermal loads and stresses (heat transfer coefficients), as well as the negative effect on the specific fuel consumption and hydraulic losses (flow velocities) in the manifolds (affecting the cylinder charge exchange and engine performance). Values for naturally aspirated automotive engines used today are up to 15 m/s. Assuming an equal value of mean piston speed c_s for similar engines will result in identical inertial forces and an identical travel distance of the moving parts, which can roughly ensure equal wear (e.g. pistons, cylinder walls, piston rings).

In general, an improvement in fuel consumption at constant mean piston speed c_s can be achieved by switching to longer-stoke configurations (B/S < 0, ξ > 0), especially for diesel engines (Design Case 2). However, for spark-ignition engines, in which we are interested, short-stroke configurations are common and, in principle, much more suitable (B/S = 1 to 1,6 / ξ = 0,6 to 1). These configurations allow us, e.g., to increase the valve diameter and thus improve the cylinder charge filling and charge exchange (Design Case 3).

Another important geometric parameter of the ICE is the **ratio λ between** the crank radius and the connecting rod length. Together with the bore-to-stroke ratio, it affects the overall height of the ICE. Larger values (shorter connecting rods) increase the normal forces between the piston and the cylinder wall and negatively affect the engine balance. Typical values are 0,2 to 0,3.

As already stated, the effective power P_e of an ICE is given by the first relation below. Another important parameter is the specific volumetric power (power density) of the ICE P_V (power related to the engine displacement volume), which can be used as an indicator for power concentration regarding the total engine volume and weight:

$$P_e = \frac{i_v.V_{z1}.p_e.n}{30.\tau} \qquad P_V = \frac{P_e}{V_z} = \frac{P_e}{i_v.V_{z1}} = \frac{p_e.n}{30.\tau}$$

A common task in engine development is to improve the ICE performance. This is why the overall power and power density (specific power) of ICEs are continuously increasing. From this, it is evident, how the ICE parameters affect the performance and what the ways to improve the engine performance are – by increasing any or all of these parameters: cylinder displacement V_z, number of cylinders n (which both increase the weight and dimensions of the engine), and operating speed or/and mean effective pressure on the piston. Already knowing the relation for the mean piston speed c_s and cylinder displacement V_z:

$$c_s = \frac{S.n}{30} \quad \left[\frac{m}{s}\right], \qquad V_z = \frac{\pi.B^2}{4}.S \quad [dm^3], \quad c_s.p_e = const,$$

the ICE power can be expressed as:

$$P_e = \frac{2{,}5.i_v.\pi.B^2.c_s.p_e}{\tau} \quad [kW].$$

From this relation, the cylinder bore can be found as:

$$B^2 = \frac{4.P_e}{2{,}5.i_v.\pi.c_s.p_e} \qquad B = \sqrt{\frac{4.P_e}{2{,}5.i_v.\pi.c_s.p_e}} \quad [dm]$$

Considering the sample engine and expected performance and operating conditions as: power target of 30 kW, number of cylinders i_v = 2, mean piston speed c_s = 12,2 m/s and BMEP p_e = 1,1 MPa, we can obtain a bore size of 75,4 mm, which corresponds with the specifications of the sample engine.

Another step is represented by the decision on how many cylinders i_v to distribute the displacement volume, which goes hand in hand with the determination of the cylinder bore B. The number of cylinders and their arrangement have a serious impact on engine complexity and cost, weight of components, engine speed, acting mechanical forces and moments, their balancing, the surface-to-volume ratio, losses, etc. Different studies show the advantages and disadvantages of many different engine layouts [36, 45]. Since the focus is on the utilisation of available engine units, the decision is simple and parameters are also taken over the sample engine – inline twin-cylinder. The cheapest engines have the smallest number of 1-2 cylinders.

The combustion process in the considered spark-ignition engine is described by a single heat release (Wiebe) function and controlled by two parameters. The first of them, the anchor (phase) angle, defines the position at which 50% of the fuel has been burnt (CA50, MFB50, 50% mass fraction burned point). The second one, the combustion duration, defines the crank period in degrees in which fuel from 10% to 90% has been burnt (MFB10-90, 10 – 90% mass fraction burnt). Typical values for the phase angle (CA50) are in the range of 5° to 12° after TDC, whereas those for the duration are in the range of 25° to 35° [32]. Tuning these parameters, e.g., for lower fuel consumption results in the shifting of the CA50 angle to higher values of 12° – 15° after TDC (later start of combustion) in order to avoid knocking, whereas the burning period is reduced even further to values from the range of 10° – 20° in order to improve the overall performance (thermal efficiency, BMEP). The burn duration tends to increase with the rise of B/S ratio, i.e., the shorter-stroke variants need a longer crank period to complete the combustion process, probably due to the increasing operation speed. However, these trends and results look somewhat optimistic. In order to maintain a certain level of reliability, the CA50 was set to 10° and the duration was around 20°. However, a revision and further refinement of the submodel settings are needed.

The engine's valvetrain is an essential subsystem. Since this is a complex mechanical system, the main design and operating parameters needed for the simulation are, in simple terms, the dimensions of the inlet and exhaust valves, their timing, and lift data. Empirical relations are used to determine the lift and dimensions of the valves, which are commonly related to the size of the cylinder bore. For example, 0,34.B for inlet valves and 0,28.B for exhaust valves, or 0,3.B for the valve lift. In this case, the valve lift data can be adapted from the 1,5 MPI serial engine from Škoda [56]. The valve timing events are set in common ranges: inlet valves open (IVO) at 10° – 40° before TDC, inlet valves close (IVC) at 40° – 60° after BDC, exhaust valves open at 40° – 60° before BDC, and exhaust valves close (EVC) at 10° – 30° after TDC. The tuning of valve timing resulted, with respect to the target to minimise fuel consumption, in quite late exhaust valve closing and a large valve overlapping period (up to 50 degrees CA), which probably led to the reduced volumetric efficacy. The idealised flow through the valves is corrected by the discharge coefficient.

The lengths of inlet and exhaust manifold runners are very essential parameters that require correct selection in order to ensure a proper charge exchange process with regard to output power and torque, as well as fuel and volumetric efficiency. The dimensions of pipe systems depend on the particular engine type, application, and operating speed range. Typically, the geometry of the runners is fixed, which places the best performance of the engine at a specific operating (speed) range. This could be advantageous because the operation of the engine is limited to one or two operating points. Previous simulation studies have shown that the overall optimisation of the engine parameters is less sensitive (not so dependent) on the runners' lengths in comparison with other parameters. [60, 61, 62] In general, the optimisation process shortens the runners' lengths at higher operating speeds and vice versa. In this study, the optimisation tool tends to reduce the lengths for both modes, which is related to the fact that shorter runners will reduce the pressure losses in the system and improve cylinder filling. A more in-depth analysis of both engine systems, which will consider their further design in terms of the engine parameters and operating characteristics, is required.

In the following simulation studies, various operating and design parameters of the ICE are considered and tuned, aiming to find preliminary favourable settings or identify some trends in performance, efficiency and design. An objective could be a reduction in fuel consumption, which could be important for frequent operation of the REx unit (although it is not a priority parameter). A reasonable steady-state operating point is selected with respect to the preliminary power requirement of 30 kW. The operating conditions in this model example are constrained by the mean piston speed and mean effective pressure. The main engine parameters were tuned roughly to improve fuel consumption. A further complex optimisation of parameters is still needed.

For the considered model example of an ICE intended for a REx unit, several ICE design case studies can be examined. The aim is to highlight and summarise some important findings from this initial (theoretical) part of the design process, which are important for further decision-making. The design case studies considered include studies of:

1. of small ICEs for REx – the effect of the engine size (displacement volume)
2. of small ICEs – effect of the piston stroke size (by identical cylinder bore)
3. of small ICEs with identical displacement volume (with different B/S ratios).

Although the size of ICE for a particular application can be roughly determined at the beginning, a further analysis is always necessary. That is why **Design Case Study 1** presents a comparative study of three similar small ICEs with different sizes of displacement volume (V1, V2 and V3). For simplicity, in this case, all three engine variants are considered as "square", i.e., the cylinder bore is equal to the piston stroke (B/S ratio = 1). The mean piston speed c_s is set to 9,9 m/s (an equivalent to 4000 min^{-1} for 74,5 mm stroke), whilst the connecting rod ratio λ to 0,33. All the conditions for similarity between the engines are fulfilled, and they can be compared correctly. A summary of some parameters at full load is presented in the table below.

	Unit	**V1**	**V2**	**V3**
ICE Total Displacement	[cm^3]	421,5	649,5	947,7
ICE Cylinder Displacement	[cm^3]	210,8	324,8	473,9
B/S (ξ) Ratio	[-]	1	1	1
Bore - B	[mm]	64,5	74,5	84,5
Stroke - S	[mm]	64,5	74,5	84,5
Mean piston speed	[m/s]	9,93	9,93	9,93
Operating Speed	[min^{-1}]	4620	4000	3527
Power - Pe	[kW]	21,1	28,0	35,3
Specific Power - Pv	[kW/dm^3]	50,1	41,1	37,3
FMEP	[bar]	1,44	1,36	1,31
BMEP	[bar]	13	12,9	12,7
BSFC	[g/kWh]	240,1	237,0	235,4
Brake Thermal Efficiency	[%]	34,9	35,3	35,6
Mechanical Efficiency	[%]	90	90,5	90,7

Tab. 10 Main parameter of engine configurations (full load) (73)

For identical mean piston speed and brake mean effective pressure (i.e., $c_s.p_e$ = const.), all three engine variants will be loaded similarly by mechanical forces, whereas the thermal loads will rise with the size of the cylinder bore. The power density (specific volumetric power) of ICEs, however, decreases with the bore (displacement) size. It can be seen that with the growth of engine displacement, i.e., with cylinder bore, both the fuel consumption (BSFC) and friction losses (FMEP) tend to decline. The decreasing operating speed, favoured by the longer stroke (whose length itself, however, negatively affects the FMEP), also positively contributes to this trend. This is all reflected in slightly better thermal and mechanical efficiency.

A comparison of all engine variants operating under conditions of equal output and mean piston speed (as a single operating point for the REx application) should be much more telling. The results coming from the same engine setups as in the previous step are presented in the Tab. 11. Since not all variants can fulfil the power target of 30 kW (most likely due to the limited c_s and lower operation speeds), they are tested at a lower power target of 20 kW.

	Spec. power P_v [kW/dm^3]	**Speed n** [min^{-1}]	**FMEP** [bar]	**BMEP** [bar]	**BSFC** [g/kWh]	**Br. Therm. Eff.** [%]	**Mech. Eff.** [%]	**Vol. Eff.** [%]
V1	47,5	4620	1,44	12,3	243,3	34,4	89,6	97,8
V2	30,8	4000	1,36	9,2	255,3	32,8	87,1	76,9
V3	21,1	3527	1,31	7,2	268,0	31,2	84,6	62,7

Tab. 11 Main parameters of considered ICE configuration, target 20 kW (74)

In general, the larger displacement results in a higher amount of power that an ICE can generate. The higher the displacement, the lower the speed needed to reach the selected power target. In these circumstances, to achieve lower power targets for the REx (i.e., produce less power), the engine variants have to be run in partial (low) load areas. This handicaps the overall engine effectiveness due to non-optimum operation conditions. This is obvious from the results: the volumetric efficiency drops significantly with the displacement, which is caused by engine regulation (throttling). That is why the overall performance in terms of fuel consumption and efficiency gets worse with the increase in displacement. On the other hand, as the engine displacement rises, FMEP drops due to the increasing size of the bore and decreasing operating speed (despite the increasing stroke). This can favour the NVH properties of ICE. Larger engines seem not to be very suitable for REx applications (power vs. size vs. weight ratio). We should remember that for the initial power target (30 kW), the situation will probably turn around (considering the results from full load analysis) and larger engines could perform better. However, it should be noted that an ICE with a displacement of around 1000 cm^3 could already be a full-featured vehicle engine, which is common nowadays in small and mid-size vehicles, which probably no longer makes sense.

Clearly, the proper selection of the ICE size for a specific propulsion system is essential. It depends on many aspects and has a significant impact on the behaviour, performance, and efficiency of the ICE. For the REx, it is important that the ICE displacement is set appropriately with regard to the power target, engine load, NVH, efficiency, lightness, and compactness.

As noted, considering the similarities between ICEs allows their characterisation by a single parameter, commonly by the value of the cylinder bore B. Therefore, **Design Case Study 2** is focused on a comparative study of a few small ICEs which differ by the size of the piston stroke. For this analysis, all engine variants have an identical value for cylinder bore (74,5 mm), and the size of the stroke is determined from B/S ratios from a selected range of

0,6 to 1,4. The mean piston speed is set to 12,2 m/s (equivalent to 4000 min^{-1} for 91,6 mm stroke) and the connecting rod ratio λ is 0,33. Parameters are summarised in Tab. 12.

	Unit	S1	S2	S3	S4	S5
ICE Total Displacement	[cm^3]	464	541	666	799	1083
ICE Cylinder Displacement	[cm^3]	232	271	333	399	541
B/S (ξ) Ratio	[-]	1,4 (0,714)	1,2 (0,833)	0,975 (1,026)	0,813 (1,230)	0,6 (1,667)
Bore - B	[mm]	74,5	74,5	74,5	74,5	74,5
Stroke - S	[mm]	53,2	62,1	76,4	91,6	124,2
Connecting Rod Length - L	[mm]	79,8	93,1	114,6	137,4	186,2
Mean piston speed	[m/s]	12,2	12,2	12,2	12,2	12,2
Operating Speed	[min^{-1}]	6878	5895	4791	3996	2948
Power - Pe	[kW]	30,8	31,25	32,7	32,9	33,4
Specific Power - Pv	[kW/dm^3]	66,5	57,8	49,1	41,1	30,8
FMEP	[bar]	1,77	1,65	1,53	1,43	1,31
BMEP	[bar]	11,6	11,75	12,3	12,4	12,6
BSFC	[g/kWh]	252,3	250,1	246,2	244,4	242,2
Brake Thermal Efficiency	[%]	33,37	33,47	34,01	34,26	34,58
Mechanical Efficiency	[%]	87,20	87,65	88,95	89,60	90,55

Tab. 12 Main parameter of engine configurations (full load) (75)

The engine displacement volume gets smaller with the increase of the B/S ratio (going from long to short stroke). As the stroke of the piston shortens, the operating speed of the engine variants rises (respecting the fixed mean piston speed), which negatively impacts the friction losses and fuel efficiency and can cause serious NVH issues. Considering the specifications of the engine variants, in this particular case (same bore size) and the mathematical relations between the main parameters, it is obvious that the operating conditions will be identical, and a similar power target can be easily achieved.

$$P_e = \frac{2,5.\, i_v.\, \pi.\, B^2.\, c_s.\, p_e}{\tau} \quad [kW].$$

Tab 13. shows the results under these similar conditions (single point) – power target of 30kW, mean piston speed of 12,2 m/s, bore of 74,5 and BMEP of 11,3 bar.

	Spec. power P_v [kW/dm^3]	**Speed n** [min^{-1}]	**FMEP** [bar]	**BMEP** [bar]	**BSFC** [g/kWh]	**Br. Therm. Eff.** [%]	**Mech. Eff.** [%]	**Vol. Eff.** [%]
S1	64,7	6878	1,77	11,3	254,7	32,9	86,5	93,7
S2	55,5	5895	1,65	11,3	252,6	33,1	87,2	92,9
S3	45,0	4791	1,53	11,3	250,1	33,5	88,2	92,0
S4	37,5	3996	1,43	11,3	249,4	33,6	88,7	91,7
S5	27,7	2948	1,31	11,3	247,8	33,8	89,6	91,1

Tab. 13 Main parameters of considered ICE configuration, target 30 kW (76)

Again, in this case, it is clear that all the common trends in ICE design are present as well (similar to Case 1). The power density decreases with the displacement size. The shorter the length of the piston stroke, the higher the operating speed (for the given piston speed), which has a negative effect on efficiency. Obviously, with the rising piston stroke (and displacement), i.e., with the decreasing operating speed, fuel consumption (BSFC) and friction losses (FMEP) improve (reflected in thermal and mechanical efficiency). All engine variants have to withstand similar mechanical forces because of identical mean piston speed and BMEP ($c_s.p_e$ = const). The thermal loads will also be similar because of the constant value of the cylinder bore. So, the proper selection of the piston stroke during the design stage is a very common task. Considering the bore size as identical, the stroke can significantly affect the engine's behaviour and performance but also the engine size in terms of displacement volume, weight, and overall dimensions. The ICE size and weight are crucial for the REx unit system, so a proper decision on the piston stroke length has to be made. Again, the decision is a trade-off between target power, engine load, NVH, efficiency, lightness, and compactness. For this purpose, it is advisable to support the simulations with parametric CAD studies.

	Unit	**B/S 1**	**B/S 2**	**B/S 3**	**B/S 4**	**B/S 5**
B/S (ξ) Ratio	[-]	0,600 (1,667)	0,813 (1,230)	1,000 (1,000)	1,199 (0,834)	1,400 (0,714)
ICE Cylinder Displacement	[cm^3]	399,1	399,3	399,1	399,3	399,6
ICE Total Displacement	[cm^3]	798,3	798,6	798,2	798,6	799,2
Bore - B	[mm]	67,3	74,5	79,8	84,8	89,3
Stroke - S	[mm]	112,2	91,6	79,8	70,7	63,8
Operating Speed	[min^{-1}]	3265	4000	4590	5181	5741
Piston speed c_s	[m/s]	12,2	12,2	12,2	12,2	12,2
Power - Pe	[kW]	25,6	31,8	36,0	38,7	42,3
Specific Power - Pv	[kW/dm^3]	32,0	39,8	45,1	48,5	52,9
FMEP	[bar]	1,35	1,43	1,50	1,57	1,64
BMEP	[bar]	11,8	12,0	11,8	11,2	11,1
BSFC	[g/kWh]	238,4	238,6	240,4	245,5	248,3
Brake Thermal Efficiency	[%]	35,1	35,1	34,8	34,1	33,7
Mechanical Efficiency	[%]	89,8	89,3	88,7	87,8	87,1
Volumetric Efficiency	[%]	91,2	92,4	91,9	89,2	88,7

Tab. 14 Main parameter of engine configurations (full load) (77)

Since the engine displacement needed to generate a desired amount of power is already known (or can be roughly determined), the purpose of the next simulation study is to compare several engine configurations by cylinder geometry and observe its effect on the engine's nature and performance. **Design Case Study 3** presents a comparative study of five small ICEs with an identical size of cylinder displacement but a different cylinder shape. The displacement is maintained at a constant value while the geometry of the cylinders (i.e., the values of the cylinder bore and the piston stroke) varies according to B/S ratios from a selected range (0,6 to 1,4). The operating conditions remain similar. Although in this case it is not possible to comply with all similarity conditions because the B/S ratio is not retained the same (geometric similarity), the quantities of mean piston speed c_s and brake mean effective pressure p_e (kinematic and thermodynamic similarity) could be kept at constant values. The operating speeds are determined according to the B/S ratios (Fig. 70). A review of the ICE variants and some parameters at full load is shown in Tab. 14.

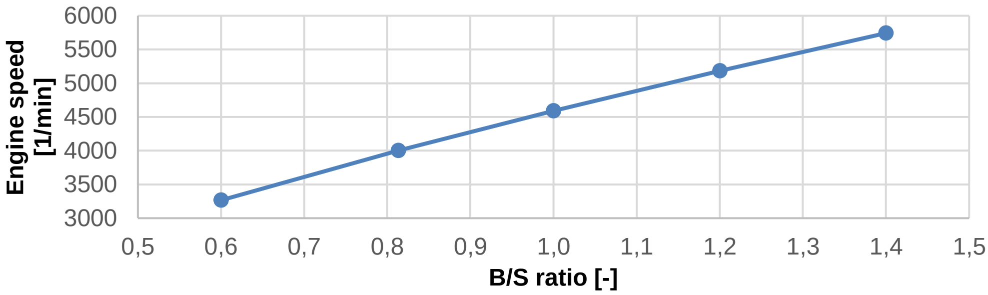


Fig. 70 Change of engine speed according B/S ratio, c_s = 12,2 m/s (78)

It is evident that, under these circumstances (full load), an ICE with the specified displacement and any of the selected configurations of the cylinder geometry can roughly meet the desired power requirement. Clearly, the shorter-stroke engines ($B/S > 1$) operate at higher speeds, which would lead to higher mechanical friction losses, wear or NVH issues. The results show that a higher B/S ratio (shorter S, bigger B) results in a better overall performance in terms of brake power, brake torque, fuel consumption, and volumetric efficiency even at these higher-speed operating ranges. It can be assumed that this is caused by the increased valve diameters (due to the larger bore), which favours the cylinder charge exchange, volumetric efficiency, and final performance. The higher value of bore also favours the time for charge formation and combustion near TDC. However, the larger-bore (shorter-stroke) configurations would lead to higher thermal loads. From this point of view, the short-stroke variants seem to perform overall better and to be more suitable for this application.

B/S (S/B) ratio	Speed n [min^{-1}]	Br. power [kW]	IMEP [bar]	FMEP [bar]	BMEP [bar]	BSFC [g/kWh]	Vol. Eff [%]
0,600 (1,667)	3265	25,6	13,15	1,35	11,8	238,4	91,2
0,813 (1,230)	4000	30,0	12,73	1,43	11,3	243,7	89,6
1,000 (1,000)	4590	30,0	11,3	1,50	9,8	249,9	80
1,199 (0,834)	5181	30,0	10,27	1,57	8,7	257,2	72,9
1,400 (0,714)	5741	30,0	9,54	1,64	7,9	275,0	70,3

Tab. 15 Results for single point, Pe = 30 kW, c_s = 12,2 m/s (79)

However, the following observations can be made by focusing on the single operating point (30 kW REx target) and considering similar operating conditions for all variants (a results summary in Tab. 15). The considered ICE configurations produce an average brake power of 29,1 kW, which roughly conforms to the desired power target. The achieved average specific fuel consumption is 252,8 g/kWh. It gets better with the increase of piston stroke and the decrease of operating speed. The friction losses are negatively affected by the operating speed. The average mechanical efficiency achieved is around 87%, whereas the brake thermal efficiency is 33%. [36, 61, 62]

It is clear that all engine variants (excl. B/S1) are capable of achieving the power target. In this case study, it is also obvious that the higher operating speeds (considering constant c_s) will result in a much higher amount of power (for shorter-stroke variants). Therefore, to reach the required power target, these configurations should operate under part-load (non-optimal) conditions, leading to a worsening of characteristics (BMEP, efficiency, fuel consumption, etc.).

The result has shown that reciprocating IC engines with similar displacement, but different cylinder geometry will differ in their way of operation, which comes from their nature. Here again, it is evident that the cylinder bore and piston stroke are essential parameters that can change significantly the ICE behaviour. The shorter-stroke ICEs don't seem very suitable.

The prediction of **peak cylinder pressure** is another important point during the engine simulation. This quantity determines the cranktrain loading conditions (load force, inertia forces, and moments). Higher peak pressures mean higher loads on the engine components. These loads have to be considered during the subsequent design proposal and optimisation of engine components (piston, connecting rod, crank shaft, bearings, etc.). It is therefore necessary at this design stage to limit the cylinder pressure values to a reasonable range. Obviously, short-stroke engine variants operate under higher loads, which increases demands on the components' structure. [15]

Another point worth mentioning is the effect of the **connecting rod length** on the engine design and performance. This parameter affects the force distribution inside the engine's cranktrain and the maximum values of the acting forces. A shorter connecting rod will reduce the load forces acting on it and the crankshaft over the entire speed range, but will increase the side (lateral) force acting between the piston and the cylinder wall. This has a negative effect on the friction and wear of the engine components. In general, there is an effort to keep the connecting rod length as short as possible regarding the particular engine configuration, engine block design, and overall height. Further analysis is needed. [14, 15, 16]

Conclusions. This chapter does not aim to offer a comprehensive solution for a range extender IC engine. First, because the engine's complete design and development depend on many different aspects, which require various additional evaluations and decision-making; second, because such a task exceeds the objectives and scope of the doctoral dissertation. It tries to offer rather a brief overview or guideline generalising, to a certain extent, the findings of the REx engine's design and parameters (research, simulation results, etc.). It could serve as a basis for further research and design. The purpose of the chapter is also to apply a suitable analysis approach or tool that fits the considered design workflow for the engine initial proposal, which is the ICE mathematical simulations. The presented ICE simulation studies focus on the analysis of a model example of an ICE suitable for a REx unit (twin-cylinder, NA, SI, based on the current state of technology). The aim is to explore the ICE parameters and find preliminary favourable settings. This is a necessary step before the beginning of the physical design.

Since this is a simple computer simulation of a virtual engine at an early stage of design, many of the parameters have to be appropriately selected or modified to match the desired characteristic features of the particular engine. To a certain extent, the simulation models of the SI engine are idealised. It is advisable to start with the initial (default) settings for the specific engine type, or to use available values from similar engines already examined (previous experience), and then to search for better settings according to the specific requirements. Hence, many parameters need further refinement (optimisation). The presented studies are more of a specific sensitivity analysis of the engine size (displacement) and cylinder shape (bore, stroke) to the performance, rather than an ordinary engine cycle simulation. Naturally, the presented results are unique for the considered REx ICE model example and design case studies, particular conditions and assumptions. On the other hand, the outlined trends in the effects of the parameters on the ICE nature and performance are fairly common in theory and practice.

The preliminary results from the ICE simulation and the initial tune-up of the model parameters successfully identified some trends and operating areas. It can be concluded that the shorter-stroke or square engine variants should be more suitable for higher required output powers (and higher loads) when they can operate with better efficiency. And vice-versa, the longer-stroke variants benefit from lower power output targets. However, it is good to remember that engine efficiency and fuel consumption (which were emphasised) do not have the highest priority during the design of a REx IC engine. So, it is all about a trade-off between all engine characteristics, vehicle type and architecture. If a shorter-stroke engine comes out better in terms of package dimensions or weight, it could be preferred to a longer-stroke configuration at the expense of efficiency and fuel consumption. In any case, a further optimisation of ICE configurations for these particular operation conditions (higher or lower operating speed, high or partial-load) could improve the situation a little bit in terms of overall efficiency and reduce the difference between the engine variants. So, a square or a shorter-stroke configuration with a displacement of around 600 cm^3 can be pointed out as a suggestion for a starting point. In any case, the results are in line with the REx engines already available.

Suggestions and Recommendations for Future Research. A more complex multi-objective and multi-parameter optimisation is desirable in order to refine the discussed parameters of the REx IC engine with respect to not only overall efficiency, performance, and operating characteristics, but also to the physical design of its subsystems and components. To refine the simulation results, it is appropriate to extend the applied simulation models with more advanced modelling features and techniques with an emphasis, first of all, on predictive modelling capabilities (models for combustion, knocking, friction, etc.). Also, proper calibration of the models, at least at some level, with real engine data is very essential. This will provide more realistic, accurate, and trustworthy results.

It should not be forgotten that the overall size and weight of the IC engine for REx are decisive parameters that could limit the choice of parameters and decision-making process and thus affect the final solution, i.e., the engine's character and performance. Therefore, a significant opportunity for the creation and providing of a conceptual IC engine proposal is the application of a complex modelling approach which involves an interconnection (coupling) of fast 0D/1D cycle simulations directly with parametric 3D design and FEA methods and also aims to provide feedback from the simulation to the design and vice versa. This will improve, for example, the heat transfer analyses and modelling by obtaining a more accurate cylinder thermal field and wall temperatures using 3D geometry and finite element models, or by transferring the results from engine cycle optimisation directly into the parametric 3D design proposal of the engine, thus maintaining the links between individual design stages.

7. Practical Application of the Suggested Approach

Chapter 4 presents a basic overview of some important moments in the design of the ICEs. The computer simulation of the vehicle’s drivetrain and the model example of an EREV described in Chapter 5 give some further information about the IC engine and the REx unit. First of all, it is important to determine the power needed to ensure a desired (or decent) extension of the vehicle’s range. The ICE simulation shown in Chapter 6 gives an idea of the REx engine’s behaviour (performance) based on a model example. At that moment, it can be said that there is enough information about the engine in order to start the components' design.

As mentioned before, the design and development process of a machine like ICE is a complex, demanding, and challenging process that constantly asks for new engineering solutions. Therefore, it is important to search for possible ways how to enhance the effectiveness of the individual design processes (stages): from the idea and design proposal, through prototyping and testing, to the launching of the product, aiming at better productivity.

The proposal and design of a new IC engine usually (or rarely) do not start from scratch, but always use the current state of technology, knowledge, and experience. Therefore, an objective of the present doctoral dissertation is also to suggest or develop a suitable method, model, or tool for quick initial (conceptual) design of mechanical systems at an early stage of design. It should strive to connect design intent and engineering knowledge in order to use, maintain, extend, and share intellectual property. Since the main focus is the ICE for a REx unit for hybrid powertrains, the suggested method is applied (adapted) to this field, in particular to the design of the engine cranktrain. However, the approach is intended to be versatile and to be used in the design of other mechanical systems as well. The aim of the tool is to provide a complete preliminary design proposal, using intuitively all knowledge and experience gathered over the years of development, including initial analyses or verifications with respect to specific requirements. This proposal will serve as a basis for the next design activities.

In the field of mechanical engineering, it is possible to observe a distinction (often artificial) between traditional analytical design methods and modern advanced computer-aided technologies (CAx). The general view is that analytical design approaches are becoming less important. In addition, it seems that modern computer-aided design and analysis methods (approaches) offer more options and accuracy, requiring less time and effort. In most cases, it is true, but it is still necessary to retain a basic understanding of the specific problems and their solutions, which usually analytical methods provide the best, thus avoiding the use of the trial and error approach for solving them. This provides a prerequisite for the implementation of approaches that combine and integrate analytical calculation methods directly into the CAx environment in order to obtain more flexible and faster solutions to the design problem. It might seem that the traditional analytical methods are not very accurate or suitable for some tasks, or in some cases, they may seem a little obsolete, but they have their merits.

The suggested method strives to make it possible to perform critical engineering calculations at the beginning of the design process relatively quickly with sufficient accuracy and to obtain direct feedback on the design in a CAx environment, for which the use of traditional models can significantly contribute. The method aims to increase the productivity of early design steps, to prepare a basis for the subsequent design activities, design decisions, and changes in harmony with the collected experience and knowledge.

7.1. Parametric Modelling Approach in CAD

Over the last few decades, computer-aided technologies (CAx), like computer-aided design (CAD), computer-aided engineering (CAE), or computer-aided manufacturing (CAM), have become an integral part of the development process in all fields of mechanical engineering. They involve a wide application of computer-based design tools, aiming to support and assist designers and engineers in every task of the development process. CAx systems are focused on simplifying, optimizing, and facilitating design process workflows with the goal of increasing productivity and improving design and manufacturing quality and accuracy. Lying at the heart of virtual product development, these software design tools allow the creation of a complete virtual three-dimensional model (digital prototype or digital twin) of the designed mechanical component or system. They use advanced design, modelling, and analysis techniques while reducing time and cost for completion due to a more efficient and optimised workflow.

The utilisation of parametric CAD modelling techniques in virtual design is very common nowadays, and many capabilities are implemented in modern CAD software systems. A parametric CAD model consists of a set of geometrical features, dimensional and geometric constraints, and is controlled by a set of parameters, expressions, and relations. Using parametric models during early-stage development is essential for preparing a flexible representation of the designed entities (components and subsystems), including their physical characteristics (material properties), size (dimensions), and shape (geometry). Furthermore, using these representations, it is possible to conduct a preliminary estimation of the mass and inertia properties, which are crucial for the overall design and operating characteristics of the designed unit. Parametric models make it possible to respond quickly to the design requirements and to make the right decisions and modifications directly and more easily.

In order to improve the speed and flexibility of the suggested designing method, it is necessary to work with geometrically optimised (simplified) CAD models. This requires removing complicated geometric elements from components and replacing them with more simple shapes, while retaining all the important cross-sections. Using simplified CAD models means considering fewer details, but this also allows much easier parameterisation and modification of the components, making the model lighter and faster to regenerate. Because the desired output from this stage is a preliminary design concept of the ICE cranktrain, the parametric modelling approach has to achieve a sufficient ratio between accuracy and speed. The parametric CAD models of the individual ICE components and their assemblies have to provide an initial idea of themselves and the engine subsystems, while also being able to respond to design changes during the initial design phase.

In contrast, by using traditional CAD modelling techniques, the designer's effort is focused on creating more detailed (accurate) design proposals, including different technological and manufacturing features. Due to the complexity, this prevents the use of parametrisation and makes greater demands on making changes and improvements to the design. Two-dimensional drawings of the ICE unit and its individual components reflecting the design changes can also be generated from the parametric three-dimensional CAD models.

7.2. Practical Implementation of Method

As stated, the doctoral dissertation aims to prepare a tool or method for solving mechanical design problems. The suggested method for initial (conceptual) design of ICE and its components is based on the application of traditional calculation models (analytical, empirical) in combination with parametric 3D modelling CAD techniques. Then, with his assistance, using available input data and pre-prepared calculation and CAD models of the engine subsystems (components), it is possible to obtain a preliminary design proposal of the ICE and its components. Analytical models for solving and dimensioning of IC engine components can be found in available technical literature, for example in [8, 19].

For a practical implementation of the intended design method, the individual relations and equations can be programmed, depending on the available SW equipment, either directly into parametric CAD models or into a utility tool. The aim is to link the analytical design calculations with the CAD models in order to control the geometry. Once the basic input data is entered, the analytical model performs a quick calculation, providing a solution, which is transferred directly to the CAD model. So, combining analytical design approaches with parametric CAD techniques allows the creation of CAD models of individual components, which are updated according to the computations and thus reflect the design changes.

The individual parametric CAD models of the engine components and their assembly are created using PTC Creo Parametric 4.0. Some relations between the parameters are inserted directly into the CAD models, but the main calculation program is prepared using MS Excel spreadsheets. Both software systems allow seamless cooperation and interaction.

The proposed analysis-driven method for parametric design can be divided into a few individual steps. The first one is to define the design intent, which means describing the problem to be solved and defining the design parameters and constraints. The parameters of components are sorted into two groups: independent (quantities that control the design) and dependent (quantities that are calculated and optimised) parameters (variables). During the second step, the initial design proposal is prepared using pre-prepared analytical calculations and parametric CAD models. Then, in the third step, this proposal is optimised according to the requirements, and as a result, a final preliminary proposal is obtained. A next (fourth) step can be introduced (if needed) for verification and refinement of the proposed CAD model with numerical computer simulation.

7.3. Description of Workflow and Model

The considered designing process starts with a specification of the design intent, including a summary of available data for the newly designed ICE and clarification of the specific application with relevant requirements and demands. Once an idea of the performance expectations is obtained, the engine displacement needed to generate the required amount of power can be roughly estimated. The process continues with the decisions about the number of cylinders, engine layout, bore-to-stroke ratio, and connecting rod length.

The ICE calculation model, which uses mathematical relations and equations, can be filled with already known data and used to perform initial design activities. The calculations are done in a spreadsheet in MS Excel (a common, widely available, and intuitive program, which can plot graphs). However, it is also possible to use other programs for engineering calculations (i.e., Matlab, Mathcad). The ICE model includes a summary of all parameters, a preliminary estimation of the masses of the cranktrain components, a calculation of kinematic and dynamic properties, etc. The model can also contain calculations for initial strength analysis of the components (analytical submodels). After entering all the needed data, all the output quantities are calculated. Then the data can be transferred (updated) to a prepared CAD model of the ICE cranktrain (allowed by the Excel analysis in Creo Parametric).

Thereafter, the first design proposal obtained is optimised in several steps, aiming to minimise the weight of components while the stresses in critical areas are kept within reasonable limits. For simplicity of the entire process, optimisation is carried out step by step. For example, it starts with the piston pin, which has to transmit the gas pressure load from the piston to the connecting rod while withstanding the load. The next part is crank throw, which as the most complicated component has to withstand load forces, moments, and torques. Once the diameters of the piston pin and the crank pin are known, the connecting rod is optimised, etc. The entire process runs inside Creo Parametric using the Optimisation feature and the connection to the calculation model in Excel and exchanging data between the both programs.

The final result is a preliminary design proposal of the ICE and its cranktrain, optimised in a first approximation by means of analytical calculations. Thus, the designers have at their disposal a complex solution in which they can find important data and context for the newly designed engine. This initial design proposal can serve as a basis for the subsequent detailed design of the individual components, for package (size, weight) and installation analyses, etc.

The proposed analysis-driven design method is applied to the model example of a twin-cylinder inline ICE intended for a REx unit. It shows that the chosen approach has good potential for application in practice. The performed computations show that the effectiveness of the designing method depends on the particular way of implementation and applied SW tools, their flexibility and "speed". For the used utilities, it can be concluded that the model computations run fast enough (in Excel), but the regeneration of CAD models in Creo during the optimisation process is quite time-consuming. So, using an alternative CAD system or method of building models may yield better results. Therefore, it is necessary to find a better practical way to implement the method in order to get the best of it.

In summary, the considered analysis-driven design approach allows us to prepare a preliminary design (conceptual) proposal of the ICE components (at first approximation) by applying analytical calculation models proven over the years. As a suggestion for enhancement of the considered method, an advanced numerical (FEA) simulation aimed to verify or refine the initial (analytical) design solution can also be performed as an intermediate step. This will make the design process a two-step process with two independent approaches.

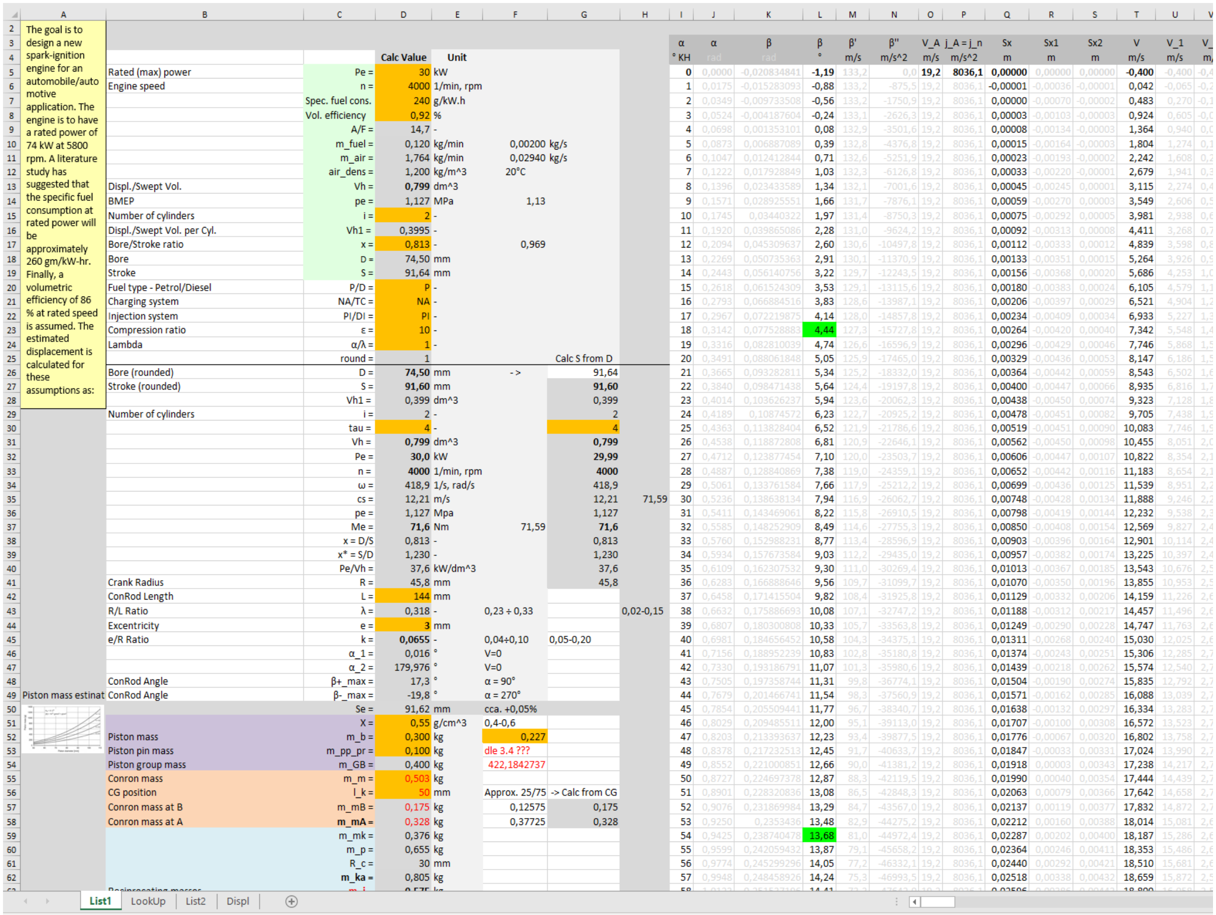


Fig. 71 Engine calculation model build in Excel (80)

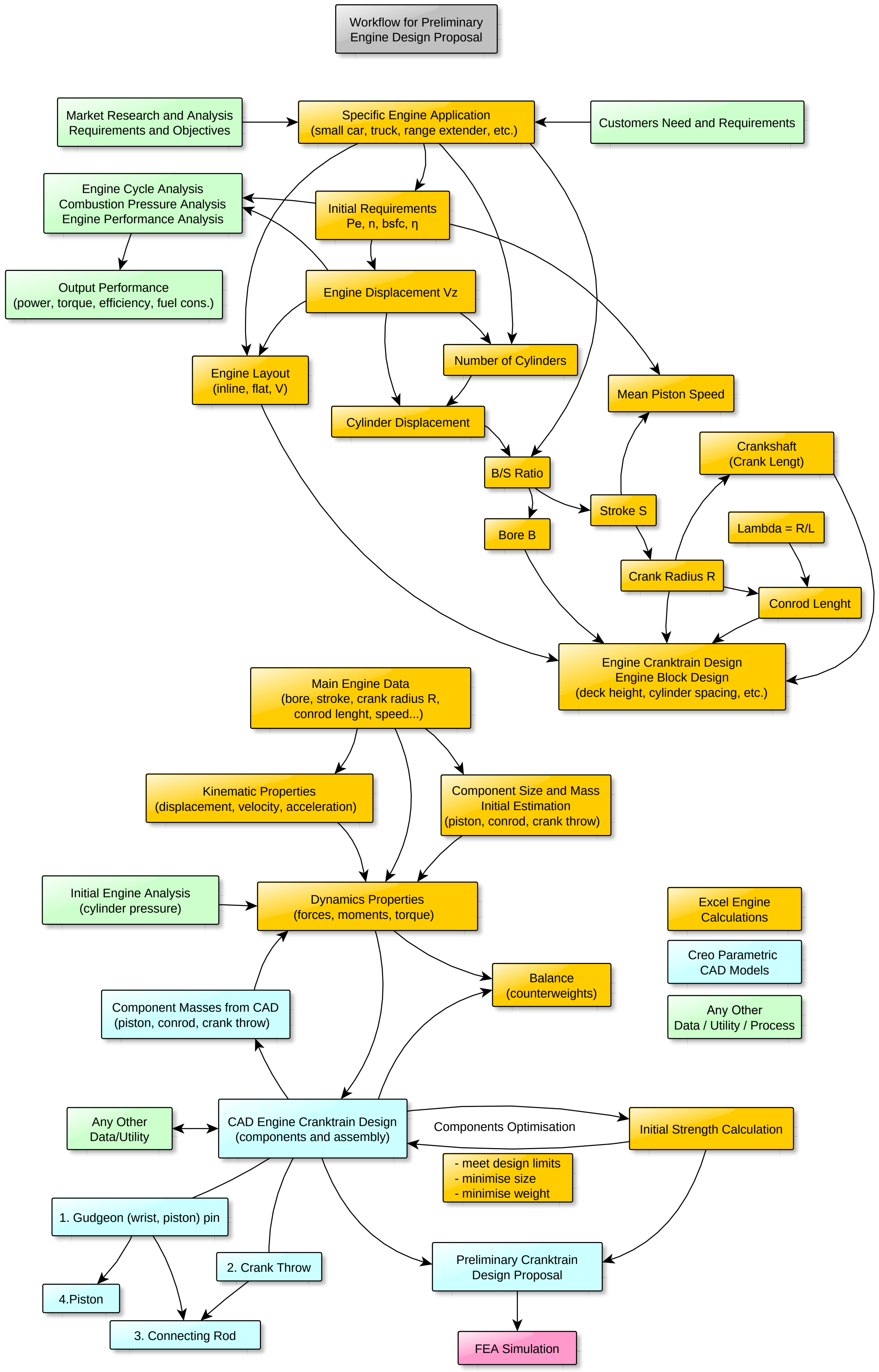


Fig. 72 Overview of suggested workflow for preliminary cranktrain design proposal (81)

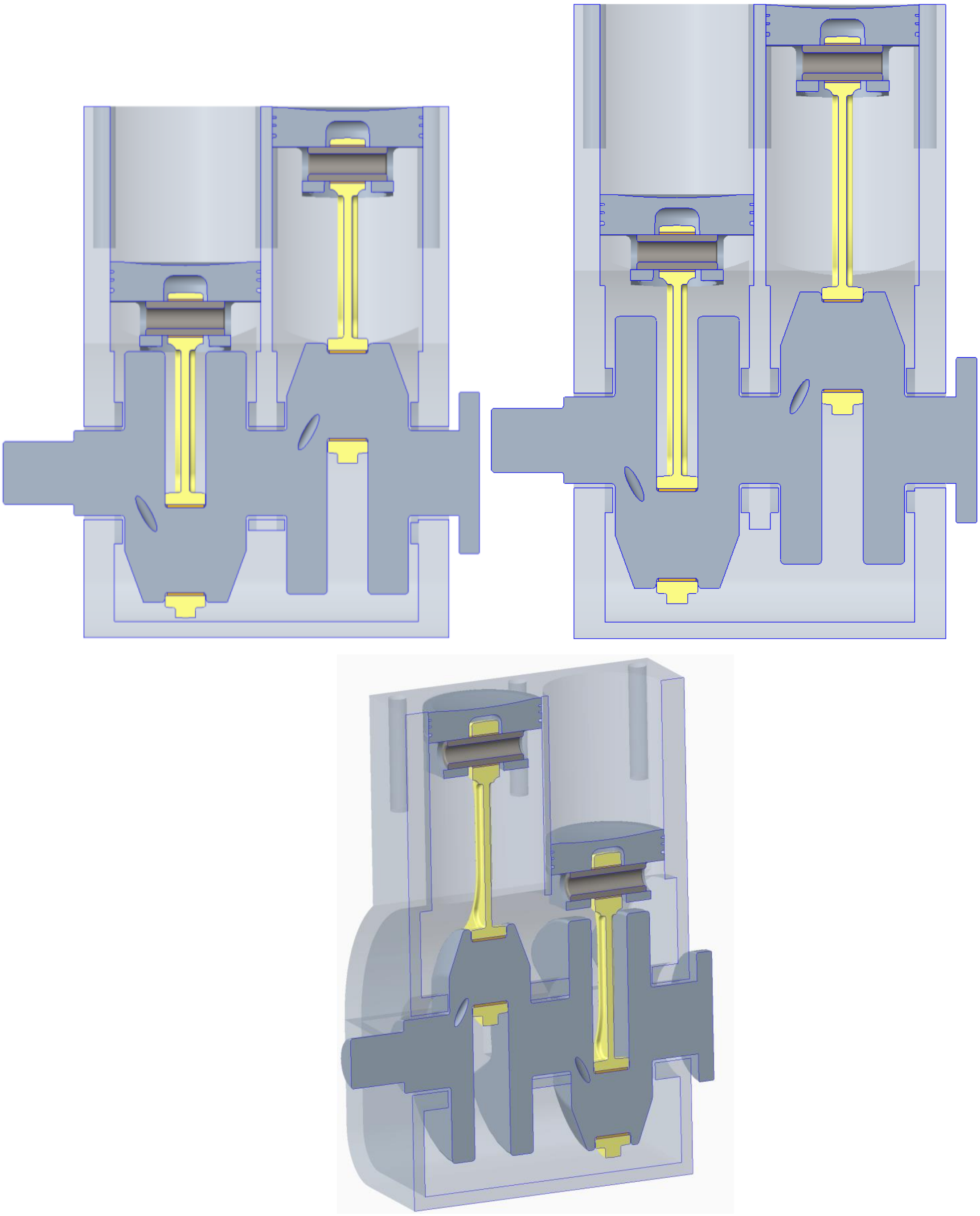


Fig. 73 Illustration of parametric 3D CAD model of twin-cylinder engine (82)

Various design studies can be performed using the pre-prepared parametric CAD models of the particular internal combustion engine and its components. Depending on the specific design level, the CAD model can be extended to the entire range extender unit. The package analysis is the most important, as it can provide a better idea of the IC engine or REx unit dimensions, weight and mounting space requirements. Considering the results of Design Case Study 2, which presents a comparison of IC engines with constant values of cylinder bore and different piston strokes, parametric design studies can be performed. In this case, it is certain that the upper part of the engine (cylinder head, valvetrain, cover, intake and exhaust manifold) could be the same for all variants, so the cranktrain can be analysed. A comparison of the sizes of the cranktrain is presented in Fig. 73 as a demonstration (S3 vs. S4).

8. Final Thoughts

8.1. Conclusions, achievement of objectives and contribution

Modern society and the transport system are evolving very quickly. This brings a lot of issues and questions. The main reasons for that are the growing human population, the increasing number of vehicles, growing energy demand, as well as the dependence of transport on fossil fuels, whose resources are limited in nature, and last but not least, the effect of transport on air pollution and CO_2 emissions.

Modern human society is striving for a sustainable future, for which mobility is essential. However, from a longer-term perspective, it appears that the current system for passenger transportation is not leading to the desired target. This is the reason why a new, more environmentally friendly transport model has to be introduced, aimed at efficient, safe, clean, and smart mobility, which at the same time outsmarts current and future obstacles like oil scarcity, traffic congestion, reduction of CO2, and pollutant emissions.

Various attempts aim to reduce dependence on fossil fuels and carbon emissions by involving the use of cleaner and safer energy. In recent years, research and development in the automotive industry have focused on improving the efficiency, safety and cleanliness of transport. Many different solutions have been proposed as an alternative or as a supplement to conventional motor vehicles, such as hybrid and electric vehicles.

The development process should be more flexible and adapt quickly to requirements and demands. This can be achieved by using the experience and knowledge gained in a combination of a wide range of latest advanced design technologies as well as modern computer-aided approaches. This will help to develop new innovative engines and powertrain concepts for the future.

Nevertheless, the internal combustion engine is the most commonly used power source in transport, and this situation probably will not change significantly in the near future. In any case, further measures are needed to increase efficiency and fulfil new and stricter emission standards. It is therefore essential to push forward the research, design, and development of ICE and powertrain systems and to continue the search for new technical solutions.

The electrification of vehicles has led to the implementation of various hybrid-electric powertrain solutions in which the ICEs are also the main option for the primary power source. However, their utilisation requires different ways of design, control, and operation. All efforts are focused on the transition to fully electric vehicles, but vehicle electrification still presents a huge technical challenge, and the future of this process remains uncertain.

Compared with conventional vehicles, the limited autonomy range of pure electric vehicles still remains one of the major disadvantages and obstacles to great success, together with the higher price. Refining the battery technologies for higher capacity, lower cost, and weight is a long-term process that will take more time and research.

The range extender unit presents an accessible solution to these issues. It involves a discreet modification of the pure electric powertrain, which is equipped with an auxiliary power unit used to extend the travel range to a more acceptable limit by supplying the EV system with electricity. This solution, e.g., allows reducing the total capacity (thus weight and cost) of the electric battery. Other energy needs are covered by the range extender. The dissertation presents a suitable approach for quick identification of the power demands of an EREV, i.e., vehicle simulation [16]. A comprehensive simulation study of the parameters of the IC engine for REx, which successfully identified the areas of optimum values, is presented [15, 17, 61].

The design of an ICE for a REx unit for EV is not a simple task due to completely different requirements, design and operating conditions. The overview of available REx ICE shows that in some cases, they use proven solutions from series production, while in others they require a more complex design approach, even asking for quite new design solutions.

Development is a complex, demanding, and challenging process that constantly asks for new findings. The design of a new ICE usually does not start from scratch but is always based on the current state of knowledge and experience. It is necessary to search for new ways how to enhance the individual design phases, aiming at better performance. A goal of the dissertation is to determine the potential of a suitable designing method aiming at an increase in productivity. Next goal involves suggestion and description of an appropriate design workflow, which is to be applied at the beginning, covering every single step.

That is why the doctoral dissertation aims to explore ways to enhance the individual steps of the design process and to improve productivity, thus obtaining the desired results faster. Traditional design approaches were chosen as an instrument to meet the goals.

This doctoral dissertation suggests and presents a theoretical and practical development of a suitable method (or a model) for quick preparation of conceptual design proposals of mechanical components and systems during the early phase of development. The aim of this tool, in our case, is to provide a preliminary design proposal of ICE components and subsystems, using intuitively all the knowledge and experience gathered over the years, including an initial analysis or verification, while respecting the specific requirements and demands. It can serve as a basis for further design stages. The designing method also involves the need for a suitable workflow to carry out initial engine design tasks with better productivity.

The main interest of the doctoral dissertation is focused on the IC engine for the range extender unit. Hence, the suggested design method is applied to the proposal of the engine cranktrain and its components. A significant contribution of the approach is the intention to be versatile and to be applied in the design process of other mechanical systems as well. It strives to perform critical engineering calculations relatively quickly with sufficient accuracy and to provide direct feedback on the design, aiming to increase productivity in the initial stage of the development process. It presents a step before advanced simulation-based design methods.

The suggested approach for an initial conceptual design of the IC engine is based on the application of traditional analytical and empirical calculation models used for designing and dimensioning the IC engine components in combination with parametric three-dimensional CAx modelling techniques and approaches. Using an ICE calculation model and virtual 3D models of the IC engine components, it is possible to complete the initial design tasks and obtain a conceptual design proposal. Today, it might seem that the analytical methods are not very accurate for some tasks, or in some cases, they may seem a little obsolete, but they still have their merits (retain a basic understanding of the specific problems and their solutions).

The designing method has been successfully tested on a model example of an IC engine suitable for a range extender unit. The design study of the IC cranktrain presented shows that the suggested design method has good potential for application in engineering practice. Moreover, it can be used for educational purposes. Through the use of an integration of both analytical and parametric CAx models, a preliminary virtual design solution can be prepared while maintaining an awareness of the physical nature of issues. The effectiveness of this design tool, however, depends on the method of implementation and the software tools used (here MS Excel and PTC Creo). The prepared calculation model can run fast enough, but the regeneration (update) of CAx models during the optimisation process is rather time-consuming. Thus, an alternative way of implementation is desirable in order to improve the speed and flexibility of the designing tool.

8.2. Suggestion for Further Research

The following themes, directions, or expansions can be considered as further areas (topics) for research in the field of internal combustion engines for range extender units for EV:

– complex multi-objective and multi-parameter parametric studies, sensitivity analyses and optimisation of the IC engine parameters and thermodynamic cycle; calibration of models;

– application of advanced predictive simulation models and approaches; FEM models;

– enhancement and extension of the suggested method with FEM numerical methods;

– methods for assessment of the mounting system of the range extender units in the vehicle, dynamics and vibrations of REx units;

– methods for optimising hybrid vehicle layout – design, optimisation, and control;

– thermal management of the range extender unit, including analysis and assessment of emission system and vehicle heating systems;

– methods for assessment of IC engine start/stop (on/off) operation aimed at fast-changing load and speed in order to reach quickly the optimum operating point;

– considering the low-cycle fatigue and wear of the cold engine.

9. Appendix I

9.1. Kinematics and Dynamics of Engine Cranktrain

In the reciprocating internal combustion engine, the straight-line reciprocating motion of the piston is converted to the rotational motion of the crankshaft by means of a slider-crank mechanism (engine crank mechanism or engine cranktrain). Sources used for this overview are [8, 19, 35, 36, 45, 68].

The engine crank mechanism in common automotive IC engines consists of the following components: pistons with piston rings, gudgeon (wrist, piston) pins, connecting rods, crankshaft with counterweights, bearings (connecting rod bushings, connecting rod bearings and main bearings).

Fig. 74 Engine cranktrain of four-cylinder engine (83)

9.1.1. Kinematic Analysis of Engine Slider-Crank Mechanism

9.1.1.1. Kinematics of Specific Points

The kinematics of the crank mechanism is determined by the equations of motion (kinematic equations), which describe the behaviours of the components by estimating their displacement (position), velocity, and acceleration. The analysis of the kinematic properties of the mechanism is important, especially for the determination of the inertial forces which, together with the forces from the gas pressure, load the mechanism but also for the analysis of the engine operating cycle, the design of the valve train and engine timing, etc.

From the theory of mechanisms, it is known that the motion of an object can be described by the motion of one or more of its points. In order to analyse the kinematic properties of the slider-crank mechanism of a real engine, it is necessary for the mechanism to be replaced by a simplified kinematic model. The process of transformation (reduction) into a kinematic model is shown in Fig. 75.

The slider-crank mechanism performs a planar motion, and a maximum of two points is enough to describe the motion of any of its components. In the kinematic diagram, the crank (crank throw) of the crankshaft and connecting rod are replaced by straight lines connecting these two specific points, and the piston is replaced by a single point.

The axis of rotation of the crank mechanism, which coincides with the axis of the main journals of the crankshaft, is represented by the point O. The axis of the crankpin (connecting rod journal) of the crankshaft and the centre of the rod big end bore are located at point A. The piston is represented by the point B, which coincides with the axis of the piston pin and the bore of the rod small end. The distance OA is the radius of the crankshaft R, and AB is the length of the connecting rod L.

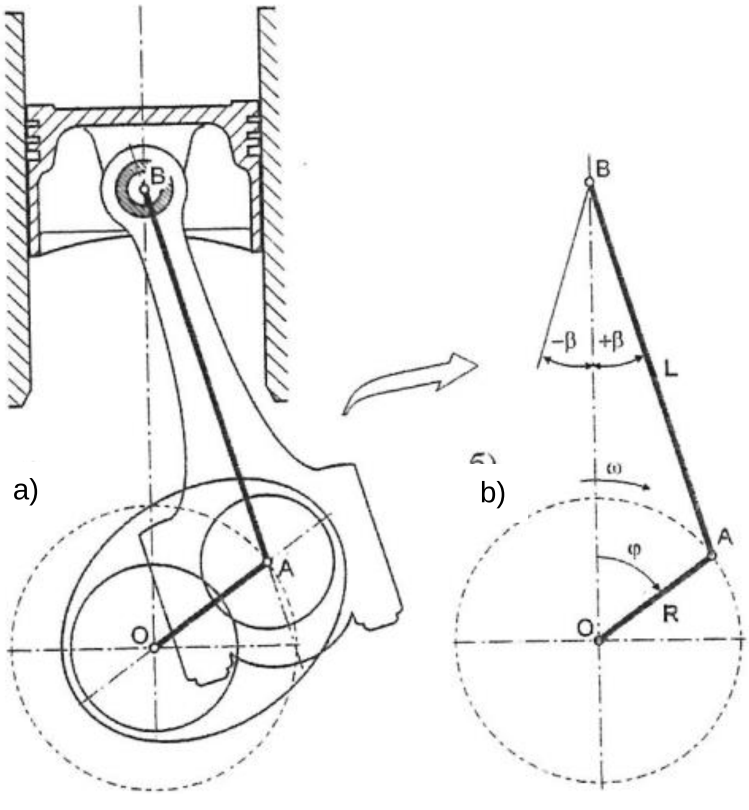


Fig. 75 Kinematic diagram of engine slider-crank mechanism (84)

The piston performs a simple straight-line, reciprocating (oscillating) motion inside the cylinder and along its axis and can be described by any of its points. The crankshaft performs a simple rotational motion, and one point is enough to describe it. The connecting rod executes a complex planar motion (the small end performs straight-line, reciprocating motion together with the piston; the big end – rotational with the crankshaft, the whole rod is swinging in the plane of crank rotation) and two points are necessary to describe it.

To study the kinematic properties of the mechanism, it is sufficient to analyse the motion (determine the path, velocity, and acceleration) of points A (for piston) and B (for crank), and both points together (for connecting rod).

The main parameter of the slide-crank mechanism, which has been already presented, the ratio $\boldsymbol{\lambda = R/L}$ takes into account the finite length of the connecting rod. This ratio is the main criterion for the similarity of different crank mechanisms. For modern automotive engines, it varies in the range of 0,23 to 0,33.

During a single rotation of the crankshaft, the piston moves down from the upper position (TDC – top dead centre) to the lower (BDC – bottom dead centre) position and then returns back to the initial point. Thus, it performs two strokes S. Every single movement of the mechanism's components can be described by the angle of rotation φ/α of the crankshaft, which in its turn simplifies the analysis since all kinematic and dynamic quantities are defined as a function of this angle.

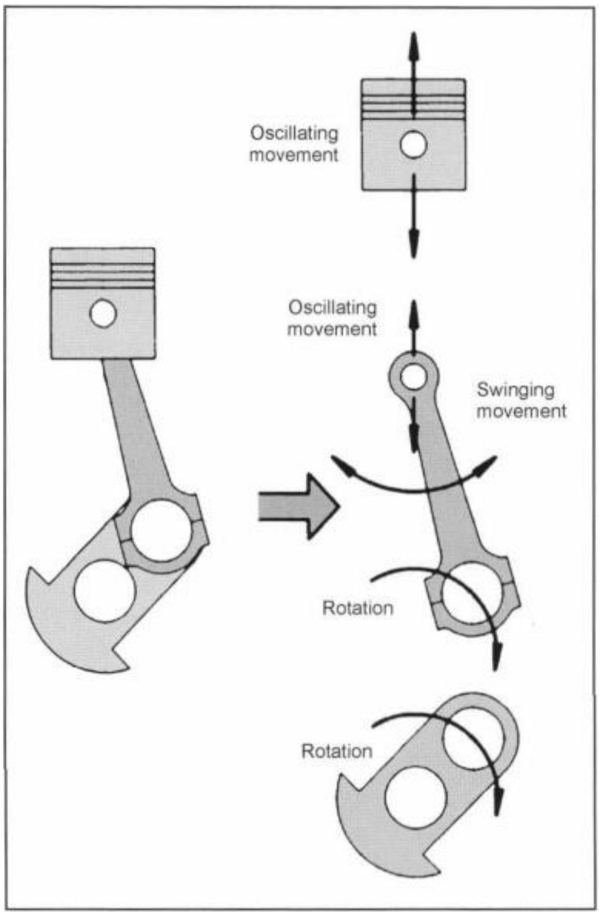


Fig. 76 Movement of components (85)

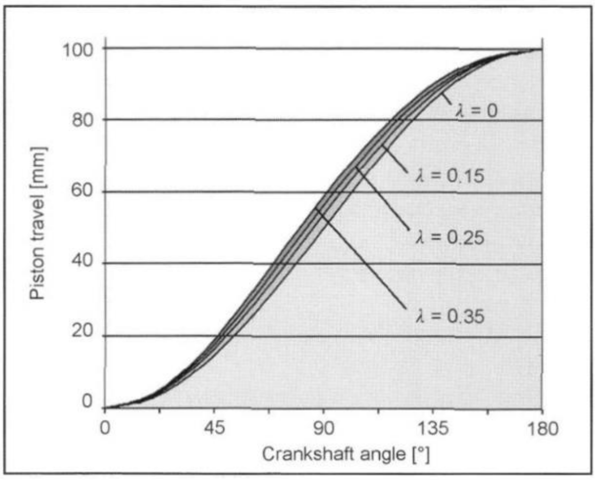


Fig. 77 Piston travel as a function of the crankshaft angle for different conrod ratios (86)

This angle, also designated as °CA, varies in the range of 0° to 360°, which is also the angular period of variation of the kinematic and dynamic parameters, which have cyclic character. That is why, at a constant angular velocity ω, i.e. by a uniform motion, the crank angle of rotation α is an independent variable directly proportional to time t:

$$\alpha\ [rad] = 2.\pi.n_s.t = \frac{2.\pi.n.t}{60} = \frac{\pi.n.t}{30} = \omega.t$$

$$\alpha\ [°] = 360.n_s.t = \frac{360.n.t}{60} = 6.n.t$$

where α – *crank angle [°, rad];* n_s, n – *engine speed [s^{-1}, min^{-1}];*
t – *time [s];* ω – *angular velocity [rad/s]*.

The angular velocity ω of the rotating crankshaft is a key element in the kinematic calculations. Within a single rotation of the crankshaft, the angular velocity ω varies due to the changing nature of the gas pressure forces (and the rising torque) and crankshaft deformations. However, these variations are small enough to be neglected, so for a given operation speed, the angular velocity can usually be assumed as a constant, $\omega = const$. For a given rotational speed of the engine n [min^{-1}], the angular velocity is calculated using the formula:

$$\omega = 2.\pi.n_s = \frac{2.\pi.\mathrm{n}}{60} = \frac{\pi.n}{30}$$

where ω – *angular velocity [rad/s];* n_s, n – *engine speed [s^{-1}, min^{-1}]*.

Thanks to this assumption and the fact that point O is stationary, the relations of the kinematic quantities are simplified and they can be presented as a function only of the angle of rotation α, i.e. the variation of the kinematic parameters is a function of the position of the crankshaft.

The pivot angle of the connecting rod, at which the power is transmitted to the crankshaft, is marked with β varies within the limits $-\beta_{max}$ and $+\beta_{max}$. For automotive engines, the angle values are between 13° and 18°.

9.1.1.2. Types of Cranktrain Mechanism

There are two types of cranktrain (slider-crank) mechanism. The first one is the so-called inline mechanism. In this type, the axis of the cylinder, which is also the axis of motion of point B, crosses the axis of the crankshaft. The other one is the offset cranktrain mechanism, where the axis of motion of point B is shifted and does not lie in the same plane as the crankshaft axis. An offset cranktrain can be obtained by shifting the cylinder axis relative to the crankshaft axis or with shifting (deaxising) only of the piston pin axis. The kinematics of both mechanisms can be described by the same equations.

For clarity, the shifting has a sign. It is positive when shifting is in the direction of the minor thrust (unloaded, counter pressure) side of the cylinder or in the direction of rotation. In the relation in which the offset is involved, it is taken with its sign.

An additional measure (criterion) to the characterisation of the offset cranktrain mechanisms is the coefficient of relative offset (shifting) $k = e/L$, which can also have a positive or negative value. For automotive ICE, its absolute value is in the range of 0,04-0,10.

The offset cranktrains have some advantages. Considering the action of the gas forces, during the power (expansion) stroke of the load on the cylinder wall is reduced at the expense of an increased load during the compression stroke, i.e. by an offset cranktrain with a positive offset (displacement), the load difference between both sides of the cylinder is reduced and more even (uniform) wear of the cylinder wall is generally achieved.

Another advantage is the holding of the piston around the TDC, which increases the time for the combustion process. This is very useful, especially for diesel engines. Shifting the cylinder axes is the only design option for arranging the cylinders in a V-type engine with a very small angle between the cylinders.

A feature of the cranktrains with positive displacement is that the crankshaft rotation angle during movement of the piston from the TDC to the BDC is bigger than that one during the return movement.

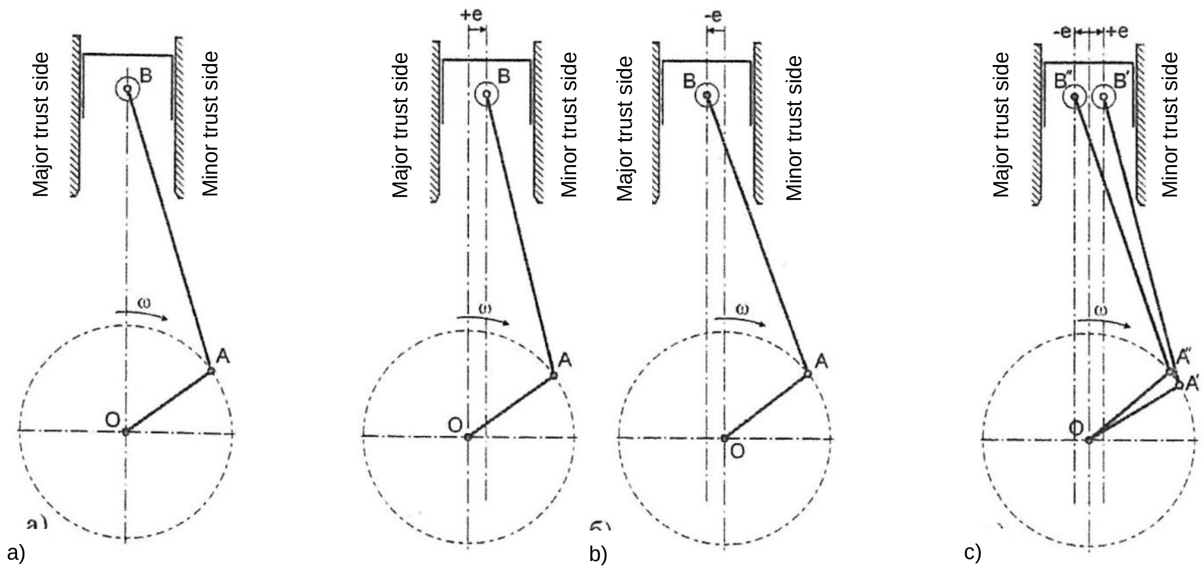


Fig. 78 Types of cranktrain: a) inline, b) cylinder axis offset, c) piston pin offset (87)

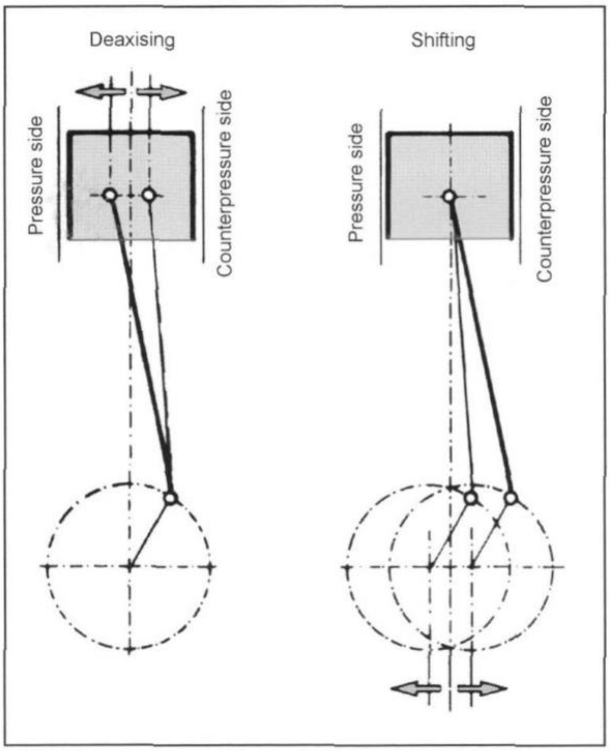


Fig. 79 Offset cranktrains (88)

When all other parameters are the same, the piston stroke of an offset mechanism is longer than that of an inline mechanism. Along with this, there is a minor increase in the accelerations and inertial forces from the reciprocating masses. However, these differences are so little that they can be neglected during the determination of the compression ratio and in strength calculations.

9.1.1.3. Calculation of Kinematic Parameters – Inline Cranktrain

This description best describes a simpler coaxial crank mechanism, where the crankshaft axis intersects the cylinder axis. It is characterised by simpler kinematic relations and the diagram shown above.

Kinematics of Rotating Crank

The crank (crank throw) is a part of the engine crankshaft, which is defined by the axis of rotation – point O and the axis of the crankpin (connecting rod journal) – point A. The distance between points O and A is the crank radius R. Since it is assumed that $\omega = const$, within a single rotation the point A crankshaft executes a steady (uniform) rotational motion.

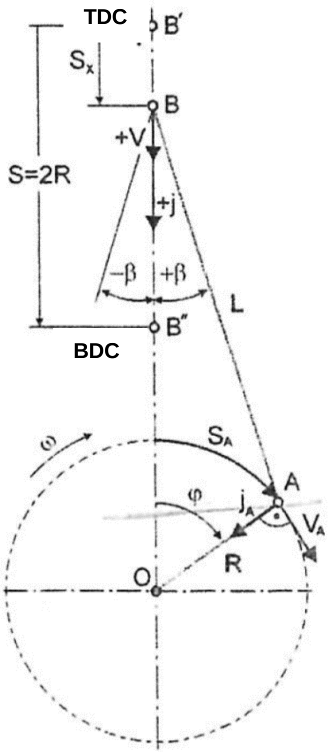


Fig. 80 Kinematics of inline cranktrain (89)

Description of the diagram: α/φ – angle of rotation of crankshaft [°, rad], TDC at point B' by $\alpha = 0°$, BDC at point B'' by $\alpha = 180°$; $R = \overline{OA} = 2.R$ – crank radius [m]; $L = \overline{AB}$ – connecting rod length; β – angle of displacement of connecting rod from the cylinder axis; S_x – total displacement (trajectory) of piston [m], $S_x = 0$ at TDC, $S_x = \overline{B'B''} = 2.R$ at BDC; point B – axis of piston pin; point A – axis of crankpin; point O – axis of the crankshaft = the axis of rotation.

The displacement (position) of the crankpin (p. A) is uniquely determined by the angle of rotation of the crankshaft α/φ. It moves along a circular path with radius R and travels distance (length of the arc) S_A from the initial position:

$$S_A = \varphi.R\ [m], \varphi\ [rad], R[m].$$

The velocity of point A has a constant magnitude and is determined by the angular velocity of rotation ω. Since it is a circumferential velocity, its direction corresponds is with the direction of movement and perpendicular to the radius of the crank.

$$V_A = \omega.R\ [m/s], \omega\ [1/s], R[m].$$

The acceleration of a point moving in a circular path has two components – normal and tangential. Since $\omega = const$ (uniform rotation), the tangential acceleration equals to zero. Therefore, the acceleration of point A has constant magnitude and equals to the normal (centripetal) acceleration, which is always directed toward the centre of rotation. This acceleration causes the centrifugal inertial force, which loads the crank mechanism.

$$j_A = j_n = V_A.\omega = \omega^2.R\ [m/s^2].$$

The motion of any point of the crankshaft can be described by these relations, where it is enough to substitute the radius from the specific point to the centre of rotation O.

Kinematics of Piston

The piston, represented by point B, performs a straight-line reciprocating motion. During this movement, the piston changes the velocity, accelerates and decelerates. The actual position is determined by the distance $\boldsymbol{S_x}$ (line $\overline{B'B}$) travelled from the reference point B', which is TDC.

The displacement (trajectory) of the piston can be expressed (from geometric dependency) as:

$$S_x = R + L - R.\cos\varphi - L.\cos\beta = R.\left[(1-\cos\varphi) + \frac{1}{\lambda}.(1-\cos\beta)\right]\ [\mathrm{m}].$$

This is a function of two angles α and β. However, the pivot angle β of the connecting rod depends uniquely on the angle of rotation φ:

$$\sin\beta = \frac{R}{L}.\sin\varphi = \lambda.\sin\varphi.$$

It is more convenient to use an expression in which the piston trajectory is only a function of angle φ. So, after some mathematical operations, with a negligible inaccuracy, the trajectory can be obtained by:

$$S_x = R.\left[(1-\cos\varphi) + \frac{\lambda}{4}.(1-\cos 2\varphi)\right]\ [\mathrm{m}]$$

At TDC by $\varphi = 0°$ -> $\mathrm{S_x} = 0$; at BDC by $\varphi = 180°$ -> $\mathrm{S_x} = \mathrm{S} = 2.\mathrm{R}$

Since the velocity of the piston is a varying quantity, it changes both magnitude and direction. The equation is obtained as a time derivative of the expression for piston trajectory, also as a function of the angle of rotation φ:

$$V = R.\omega.\left(\sin\varphi + \frac{\lambda}{2}.\sin 2.\varphi\right)\ [\mathrm{m/s}].$$

At TDC by $\varphi = 0°$ and at BDC by $\varphi = 180°$ the velocity is $V = 0$, because at these points the piston changes the direction of movement. The maximum value of piston velocity depends (by equal other conditions) on the coefficient λ. The higher the ratio λ (shorter rod or longer stroke) is, the higher the value of the velocity is, and the maximum values move to the TDC.

The time derivative of the expression for velocity results in the equation for piston acceleration, which is also variable in magnitude and direction.

$$\boldsymbol{j = R.\omega^2.(\cos\varphi - \lambda.\cos 2\varphi)\ [m/s^2]}.$$

The maximum values for the acceleration are reached in TDC:

$$\boldsymbol{j_{max} = \omega^2.R.(1-\lambda)}.$$

The piston reaches an extreme negative value of acceleration at BDC. For $\lambda < 0{,}25$: $j_{min} = -\omega^2.R.(1-\lambda)$. Two local extremes appear for $\lambda > 0{,}25$.

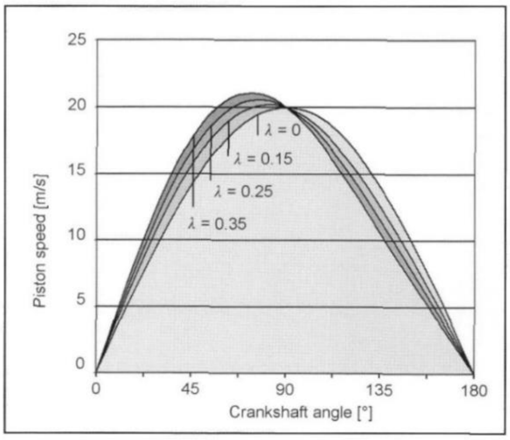


Fig. 81 Piston speed as a function of the crankshaft angle for different conrod ratios (90)

	Minimum	**Maximum**
Piston trajectory	$\alpha = 0°$, $\alpha = 360°$	$\alpha = 180°$
Piston velocity	$\alpha = 0°$, $\alpha = 180°$	$\alpha = 90°$, $\alpha = 270°$
Piston acceleration	$\alpha = 90°$, $\alpha = 270°$	$\alpha = 0°$, $\alpha = 180°$

Tab. 16 Parameters of piston (91)

Kinematics of Connecting Rod

After knowing the motion of point A and B and considering them together, it is possible to describe the motion of the connecting rod, which executes a complex planar movement. The axis of the small end bore (point B – gudgeon pin,) moves in the same manner as the piston – a straight reciprocating motion along the cylinder axis. The axis of the big end bore (point A – crankpin) rotates along a circular path with the crankshaft around axis O. All other points of the connecting rod perform movement along curves close to ellipses.

An important characteristic, which defines its position is the rod (pivot) angle β, measured from the vertical axis (also dependent on angle φ):

$$\beta = arcsin(\lambda.\,sin\,\varphi)$$

Maximum values are reached by $\varphi = 90°$ and $\varphi = 270°$. The values of the angle for common automotive engines are $\beta_{max} = 10 \div 18°$:

$$\pm\boldsymbol{\beta}_{\mathbf{max}} = \pm\mathbf{arcsin}\,\boldsymbol{\lambda}$$

Since the angle β has not negligible effect on the transfer of the forces inside the crank mechanism, it is important to understand its variation. The angle has not negligible effect on the transfer of the forces inside the crank mechanism. Moreover, it is involved in the equations for their magnitude.

	Maximum	**Minimum**
Angular displacement	$\alpha = 90°$, $\alpha = 270°$	$\alpha = 0°$, $\alpha = 180°$
Angular velocity	$\alpha = 0°$, $\alpha = 180°$	$\alpha = 90°$, $\alpha = 270°$
Angular acceleration	$\alpha = 90°$, $\alpha = 270°$	$\alpha = 0°$, $\alpha = 180°$

Tab. 17 Parameter of connecting rod (92)

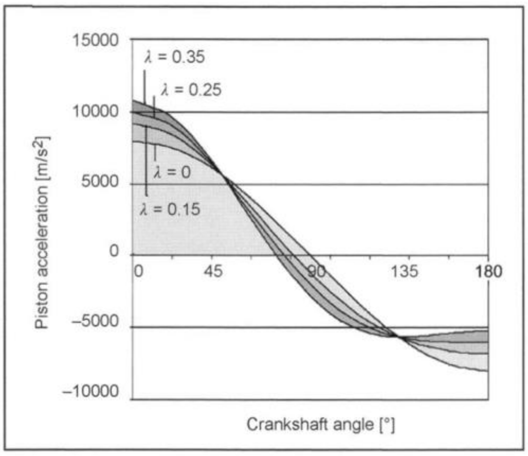


Fig. 82 Piston acceleration as a function of the crankshaft angle for different conrod ratios (93)

9.1.1.4. Calculation of Kinematic Parameters – Offset Cranktrain

Kinematics of Piston

The piston displacement (trajectory) is expressed as:

$$S_x = R.\left[\sqrt{\left(\frac{1}{\lambda}+1\right)^2 - k^2} - \cos\varphi - \frac{1}{\lambda}.\cos\beta\right]\ [m], k = e/R.$$

After some operations, the expression can be written with a small inaccuracy as a function of the angle of rotation φ:

$$S_x = R.\left[(1-\cos\varphi) + \frac{\lambda}{4}.(1-\cos 2\varphi) - \lambda.k.\sin\varphi\right]\ [m].$$

In this case, the initial position of the mechanism is also given by the angle $\varphi = 0$, when the crank is vertically on the top. However, both end positions of the piston are reached by angle φ_1 (TDC) and φ_2 (BDC), given by the following relations:

$$\varphi_1 = arcsin\left(\frac{\lambda.k}{\lambda+1}\right)\ [°CA]$$

and

$$\varphi_2 = 180 - arcsin\left(\frac{\lambda.k}{\lambda-1}\right)\ [°CA]$$

$$\sin\beta = \frac{R}{L}.\sin\varphi = \lambda.\sin\varphi.$$

According to the particular offset (-e or +e), the TDC can be reached before $\varphi = 0$, i.e. φ_1 can be with a negative value. An interesting fact is that both distances from TDC to BDC and are the same, i.e. the piston travels longer from TDC to BDC than from BDC to TDC.

The distance between TDC and BDC is longer than $2.R$, but for small values of e it can be neglected. In other cases, the relationship takes the form:

$$S_x = R.\left[\left(1+\frac{1}{\lambda}\right).\sqrt{1-\left(\frac{\lambda.k}{\lambda+1}\right)^2}+\left(1-\frac{1}{\lambda}\right).\sqrt{1-\left(\frac{\lambda.k}{\lambda-1}\right)^2}\right]\ [\mathrm{m}].$$

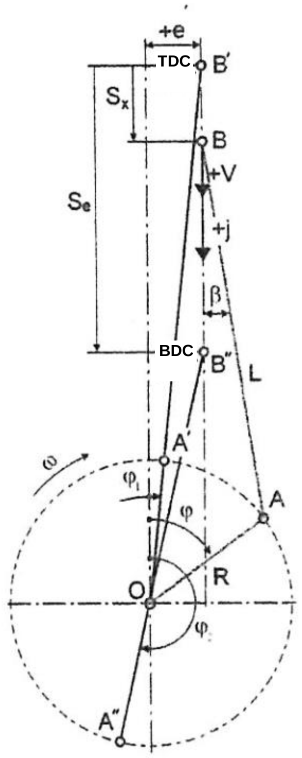


Fig. 83 Kinematics of Offset Cranktrain (94)

Description of the diagram: φ – angle of rotation of crankshaft [°, rad], TDC at point B', BDC at point B''; $R = \overline{OA} = 2.R$ – crank radius [m]; $L = \overline{AB}$ – connecting rod length; β – angle of displacement of connecting rod from the cylinder axis; S_x – displacement of piston [m]; point B – axis of piston pin; e – offset, point A – axis of crankpin; point O – axis of the crankshaft = the axis of rotation.

According to the particular offset (-e or +e), the TDC can be reached before $\varphi = 0$, i.e. φ_1 can have a negative value. An interesting fact is that both the distances from TDC to BDC and are the same, but i.e. the crank travels different distance from TDC to BDC and from BDC to TDC.

The distance between TDC and BDC is longer than $2.R$, but for small values of e it can be neglected. In other cases, the relationship takes the form:

$$S_x = R.\left[\left(1+\frac{1}{\lambda}\right).\sqrt{1-\left(\frac{\lambda.k}{\lambda+1}\right)^2}+\left(1-\frac{1}{\lambda}\right).\sqrt{1-\left(\frac{\lambda.k}{\lambda-1}\right)^2}\right]\ [\mathrm{m}].$$

The increase of the piston stroke for maximum values for automotive engines of λ (0,33) and k (0,10) is only about 0,05%, i.e. for a stroke of 100 mm, there would be an increase of 0,05 mm. So, the latest formula makes only sense for research of cranktrains with a very short connecting rod and a large eccentricity (offset).

The velocity of the piston is obtained as a time derivative of the expression for piston trajectory, also as a function of the angle of rotation φ:

$$V = R.\omega.\left[\sin\varphi + \frac{\lambda}{2}.\sin 2\varphi - \lambda.k.\cos\varphi\right]\ [m/s].$$

It is obvious that the velocity is not equal to zero by $\varphi = 0°$ and by $\varphi = 180°$, but by $\varphi = \varphi_1\ (TDC)$ and $\varphi = \varphi_2$ (BDC).

The time derivative of the relation for velocity gives the equation for piston acceleration:

$$\mathrm{j} = \mathrm{R}.\omega^2.[\cos\varphi + \lambda.\cos 2\varphi + \lambda.\mathrm{k}.\sin\varphi]\ [\mathrm{m/s^2}].$$

The position of the connecting rod is defined by the angle $\beta = arcsin(\lambda.(\sin\varphi - k))$. It is in a vertical position by $\varphi = \arcsin(k)$. Since the displacement in both directions differs, maximum values can be found by:

$\beta^{+}_{\max} = \arcsin(\lambda.(1-\mathrm{k})$ by $\varphi = 90°$ and $\beta^{-}_{\max} = -\arcsin(\lambda.(1+\mathrm{k})$ by $\varphi = 270°$.

By analysis of the kinematic relations, it can be found that the formulas for offset cranktrain are universal. So, they can be used in both cases, i.e. by $k = 0$ the expressions for inline cranktrain are obtained.

9.1.2. Dynamic Analysis of Engine Slider-crank Mechanism

9.1.2.1. Mass Reduction

To simplify the dynamic calculations, the real crank mechanism is replaced by a dynamically equivalent system (model) of concentrated masses and massless rigid links. These masses, called reduced masses, are concentrated at certain points and represent the mass of several components in the dynamic model, which is a two-mass (Fig. 84). One of the masses is concentrated at point A (crankpin axis) and the other at point B (piston pin axis).

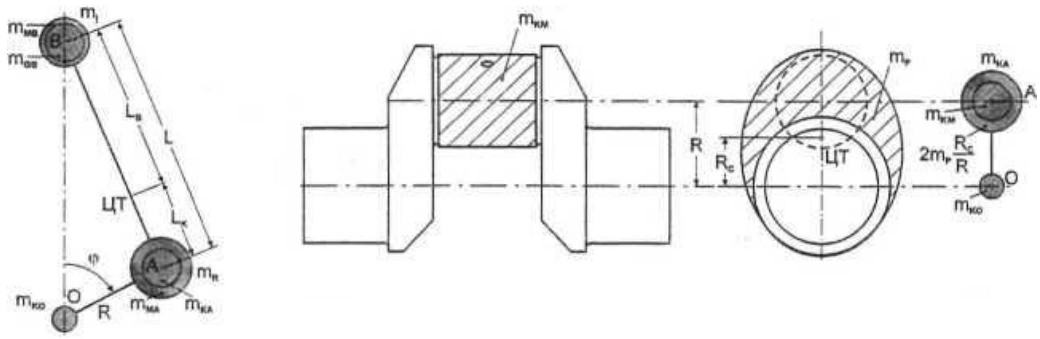


Fig. 84 Two-mass model of cranktrain (95)

The point of application of the inertial forces of the piston group components is the axis of the piston pin. That is why the masses of these components (piston, piston pin, piston rings, retaining rings) are summed up into the mass m_B and concentrated at point B.

To avoid the complex planar motion of the connecting rod, its mass is divided between these two points. If the position of the centre of gravity of the connecting rod is known, the following formulas are used to determine its two reduced masses:

$$m_{MB} = \frac{L_k}{L}.m_M\ [kg];\ m_{MA} = \frac{L_B}{L}.m_M\ [kg];$$

m_M – mass of the connecting rod. If it is impossible to determine theoretically the position of the centre of gravity of the connecting rod, approximate formulas can be used in the calculations:

$$m_{MB} = \ 0{,}25.m_M\ [kg];\ m_{\mathrm{MA}} = 0{,}75.m_M\ [kg].$$

The mass of the crankshaft crank throw is divided into two parts. The mass of the main journals and the part of the webs symmetrically placed around the axis of rotation is balanced. They do not cause centrifugal inertial forces and are concentrated at the centre of rotation in point O.

The unbalanced masses of the crank that are concentrated at point A are the mass of the connecting rod journal (crankpin) and the unbalanced parts of the crank webs. Because the centre of gravity of the unbalanced part of the crank webs does not lie at point A, these masses are reduced to the axis of the crankpin. Then for the mass of the crank reduced to point A:

$$m_{KA} = m_{KM} + 2.m_P.\frac{R_c}{R}\ [kg]$$

Considering the done reduction of the masses of the components of the crank mechanism, the resulting dynamic model consists of the following two masses:

– mass of reciprocating moving parts concentrated at a point B:

$$m_j = \ m_{GB} + m_{MB}\ [kg].$$

– mass of the unbalanced parts performing a rotational motion concentrated at point A:

$$m_R = \ m_{KA} + m_{MA}\ [kg].$$

The magnitude of the mass m_j determines the inertial forces acting in the cranktrain caused by reciprocating moving parts. The mass of the rotating unbalanced parts of the crank and connecting rod m_R determines the magnitude of the centrifugal inertial force. The aim is to completely balance this force by means of counterweights on the crankshaft web.

In fact, there is a third mass in the dynamic model of the cranktrain, concentrated at point O, the centre of rotation of the crankshaft. However, it does not create inertial forces, so it is not included in the dynamic analysis, but it is taken into account in the determination of the equivalent mass moment of inertia of the mechanism.

9.1.2.2. Analysis of Engine Slider-crank Mechanism

The dynamic analysis of the engine cranktrain studies the forces and moments acting in the mechanism. They arise from the gas pressure inside the cylinder and moving masses. Based on an accurate determination of total forces, the strength calculations of the individual engine components can be carried out in order to design, size up, and optimise them precisely and to ensure their durability and reliability, and the torque non-uniformity and the degree of engine non-uniformity are determined.

Forces Acting in the Slider-crank Mechanism

During engine operation, the following forces and relevant moments act on the components of the crank mechanism and engine frame:

– **gas pressure forces (primary forces)** – internally balanced and do not transmit to the engine support frame (do not cause vibrations);

– **inertial forces (secondary forces)** – caused from rotation and reciprocation (oscillation) of the moving masses, not internally balanced (cause vibrations);

– **frictional forces**;

– **the weight of the engine components**.

All unbalanced forces cause reaction forces and moments in the engine supports acting on the foundation. During every cycle (720 °CA for a four-stroke engine), the forces acting on the engine cranktrain vary continuously in magnitude and direction. Again, for the dynamic analysis, it is assumed that the angular velocity ω is constant during the operating cycle in order to simplify the expressions and represent them as a function of the crank position (angle of rotation φ) of the crankshaft.

Since the determination of the frictional forces is a complex task and due to their relatively small absolute magnitude (in comparison with other forces), they can be neglected during the initial dynamic analysis. The same also applies to the relatively small weight of the individual components. The calculations are therefore reduced to the determination of the gas pressure and inertia forces and their sum effect.

Gas Pressure Forces

The total gas pressure force is a sum of the gas pressure forces that act simultaneously on the top of the piston crown, cylinder wall and cylinder head from one side and the piston bottom from the other.

The pressure of the gases inside the engine cylinder changes throughout the operating cycle. The variation of the pressure as a function of crankshaft rotation angle φ, i.e. $p = f(\varphi)$, is obtained from the indicator diagram, [MPa]. For dynamic and strength analysis, this is very handy, since the pressure forces are proportional to the pressure and thus a function of φ, i.e.

$P_g = f(\varphi)$. In case, a new engine is being designed, the indicator diagram and dependency of the pressure to the crankshaft angle are usually estimated from an initial thermodynamic analysis.

The gas pressure inside the engine crankcase is roughly constant and usually is considered equal to the atmospheric pressure, $p_o = 0{,}1\ MPa$.

The positive gas pressure force in the engine cylinder directs down towards the crankshaft axis. The crankcase gas pressure acts in the opposite direction. The total gas force can be expressed by:

$$P_g = (p - p_o).A_p = (p - p_o).\frac{\pi.B^2}{4}\ [N], p(\varphi)\ [MPa], A_p\ [mm^2], B\ [mm].$$

This gas pressure force is transmitted through the gudgeon (piston) pin, connecting rod and crankshaft to the main bearings and the engine frame (block). During this force transmission torque T_{tq} rises on the crankshaft and a reaction torque T_r rises in the frame. The gas pressure acts not only on the piston but also on the other surfaces of the cylinder in the working space. The pressure forces which act on the cylinder walls perpendicularly to the cylinder axis are balanced. The pressure acting on the cylinder head results in a compressive load force P'_g, with the same magnitude, but in the opposite direction as the compressive load force P_g acting on the piston. Since the cylinder head is also part of the engine frame, these pressure forces are balanced inside the engine frame (they are internally balanced). This means that the gas pressure forces do not cause vibrations in the direction of the cylinder axis. The reaction moment causes angular (torsional) vibrations around the crankshaft axis.

Inertial Forces

The inertial forces are a consequence of the components' masses moving with acceleration. Two types of inertial forces are present in the slider-crank mechanism: inertial centrifugal forces due to rotating masses and inertial forces due to reciprocating masses.

The inertial forces caused by the reciprocating movement of the masses P_j are directed along the cylinder axis. They transmit to the engine frame (block) via the crank mechanism and cause vibrations along the axis and varying reactions in the engine support. The centrifugal inertial force P_R is caused by the rotation of masses and acts in the direction of the crank throw radius, causing vibrations in the engine body (frame) and directionally alternating reactions in the engine support (mounting).

An alternating torque T_{tq} acts on the crankshaft and is transmitted to the connected working machine. This torque produces a reaction torque T_r which acts on the engine frame and causes alternating reactions in the engine supports (frame).

For strength calculations that consider the varying loads characteristic of reciprocating internal combustion engines, it is necessary to determine both the maximum and minimum values of the forces and moments and their mean values. It is therefore necessary to determine these values in multiple crankshaft positions during a single duty cycle. In addition, for internal combustion engines for transport vehicles, it is necessary to obtain the forces for several calculation modes.

The centrifugal inertial forces are caused by the rotational motion of the masses of the mechanism concentrated at point A:

$$P_R = -m_R.j_A = -m_R.R.\omega^2\ [N], m_R\ [kg], R\ [m], \omega\ [1/s]$$

where m_R – *rotating masses;* j_A – *acceleration of point* A.

The effect of the centrifugal inertial forces is transmitted directly into the engine supports (mounts) and their balancing is mandatory. This is done easily by the mean of counterweights on the extension of the crankshaft webs (arms). These inertial forces are therefore not involved in the dynamic analysis, but they are taken into account in some strength analysis.

The **inertial force** caused by the reciprocating motion is expressed by the masses concentrated at point B and the acceleration of that point:

$$P_j = -m_j.j = -m_p.R.\omega^2.(\cos\varphi + \lambda.\cos 2\varphi)\ [N], m_k\ [kg], R\ [m], \omega\ [1/s].$$

This inertial force has the same nature of alternation as the acceleration but is mirror-reversed. It is positive when directs towards the crankshaft axis. The inertial force of the reciprocating moving masses is transmitted by the crank mechanism to the crankshaft bearings. It is not internally balanced, occurs in the engine supports and should be balanced.

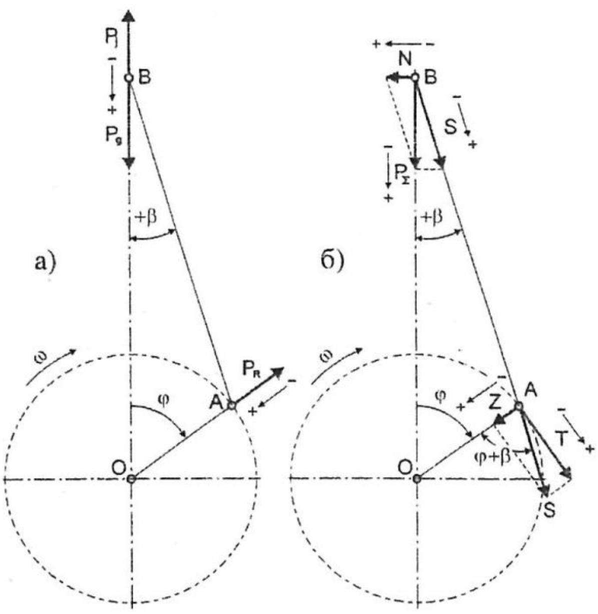


Fig. 85 Force distribution in cranktrain (96)

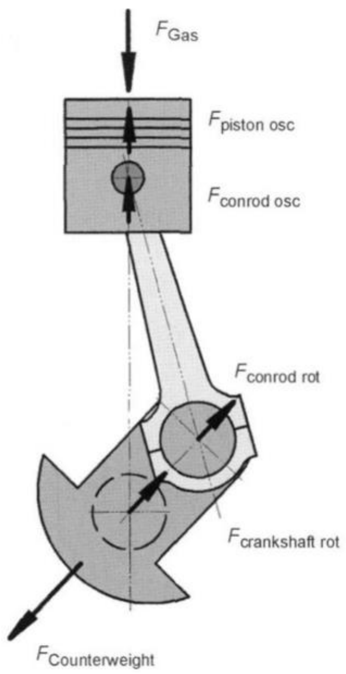


Fig. 86 Inertia forces in cranktrain (97)

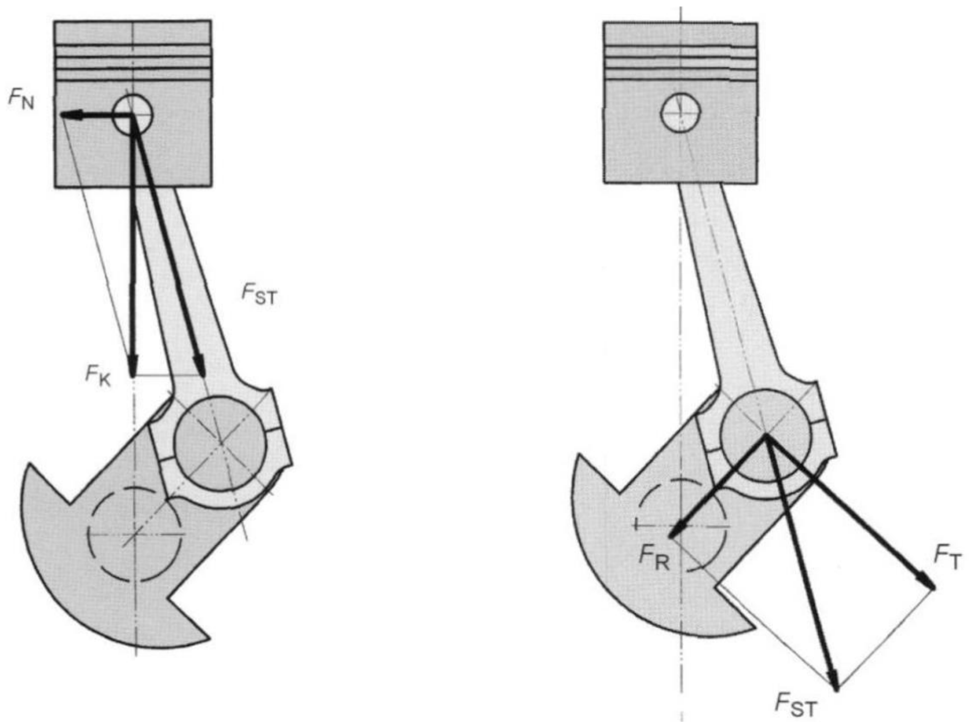


Fig. 87 Forces on piston and in connecting rod (98)

Total force in the cranktrain – The gas pressures force is transmitted from the piston via the gudgeon (piston) pin to the other components of the crank mechanism, i.e. its point of application is point B. The inertial force P_j has the same point of application, which make it possible to sum both forces:

$$P_\Sigma = P_g + P_j \ [N]$$

The resulting total force is the source of mechanical load engine components. It has a direction along the cylinder axis. The positive direction is towards the crankshaft axis.

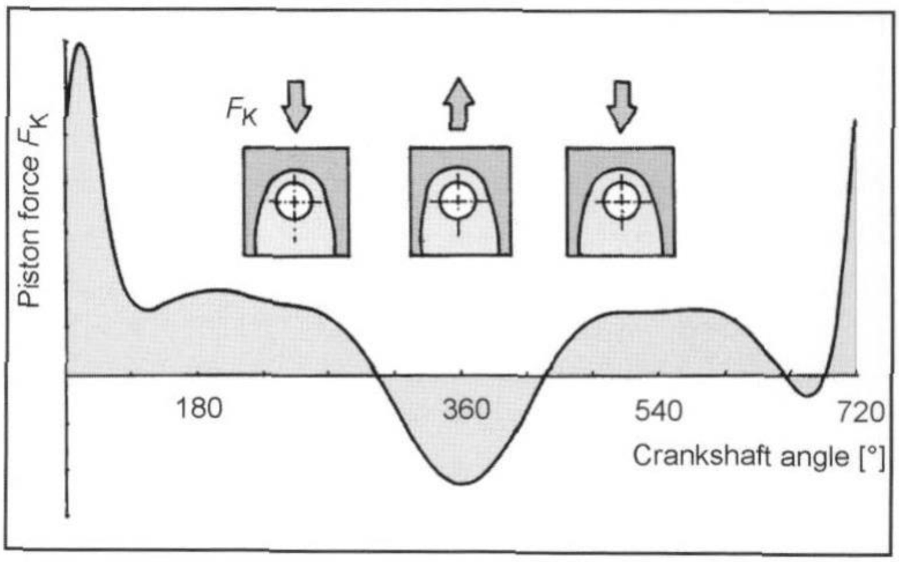


Fig. 88 The course of total force on piston (99)

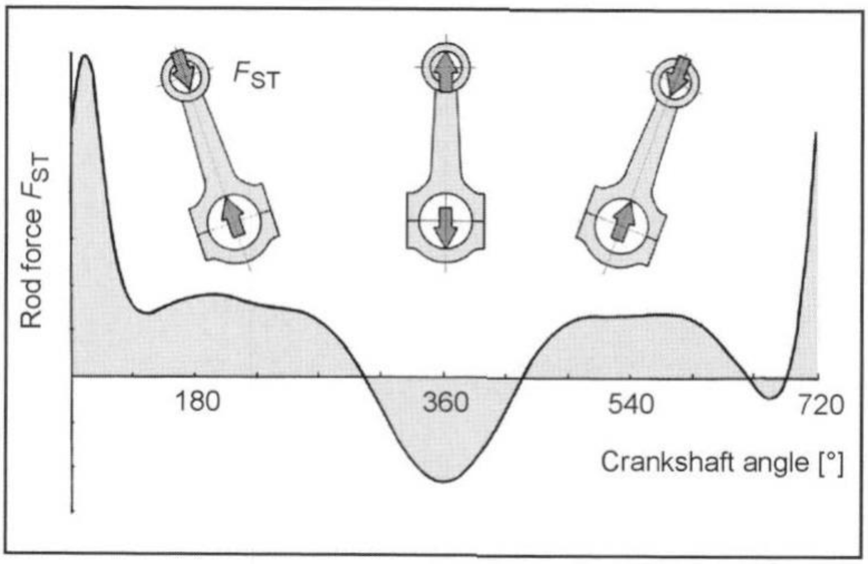


Fig. 89 The course of force in connecting rod (100)

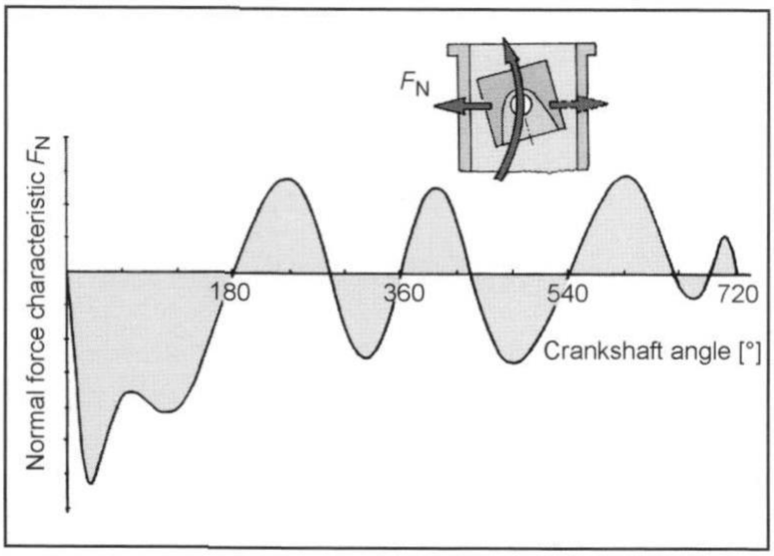


Fig. 90 Course of normal force on piston (101)

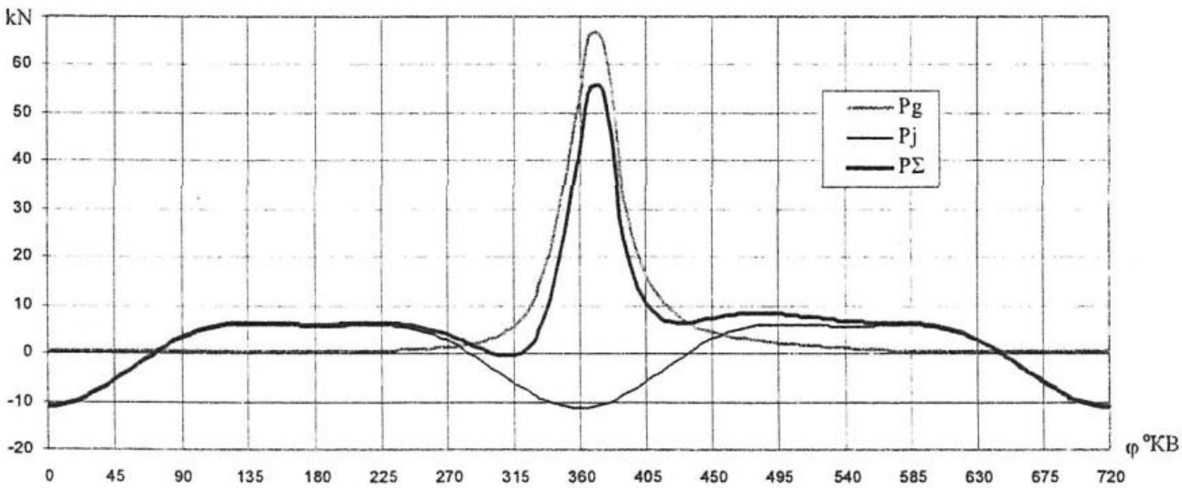


Fig. 91 Gas, inertial and summary force of diesel engine (102)

Decomposition of Forces in Engine Cranktrain

It is necessary to be familiar with the load forces acting inside the crank mechanism in order to perform a strength analysis of the individual component of the engine. The relations for these forces can be obtained by following the action of the total force and its decomposition in the crank mechanism.

The action of the total force P_{Σ} is transmitted into the cylinder wall perpendicularly to its axis, and into the connecting rod along its axis.

The force N acting in a direction perpendicular to the cylinder axis is called normal force and loads the cylinder wall. The positive direction of the normal force is shown in Fig. 85.

$$N = P_{\Sigma}.tg\beta\ [N]$$

The force S acts along the connecting rod axis and transmits onwards to the crankshaft. It is positive when pushes the connecting rod and negative when pulls (stretches) it.

$$S = \frac{P_{\Sigma}}{cos\,\beta}\ [N]$$

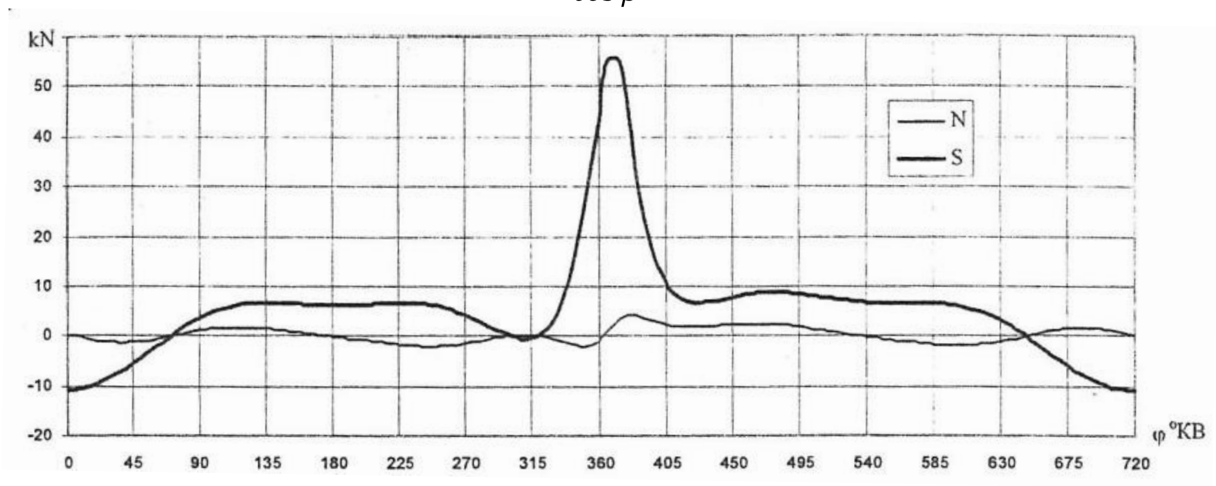


Fig. 92 Normal force and force on connecting rod of diesel engine (103)

The force S is transmitted to the connecting rod journal (crankpin) of the crankshaft. Its direction concludes an angle $\varphi + \beta$ with the axis of the crank throw, which changes with the position of the crankshaft φ. To simplify the analysis of the crankshaft loads, this force is decomposed into two components:

– force Z – radial, along the crank radius:

$$\mathbf{Z = S.cos(\varphi + \beta) = \frac{P_{\Sigma}.cos(\varphi + \beta)}{cos\,\beta}\ [N]}$$

– force T – tangential, perpendicular to the crank radius:

$$\mathbf{T = S.sin(\varphi + \beta) = \frac{P_{\Sigma}.sin(\varphi + \beta)}{cos\,\beta}\ [N]}$$

The force Z is positive when pushing the crank, whereas and tangential force T is positive in the direction of crankshaft rotation. The course of every force is a function of the crank angle φ.

Engine torque – Engine torque is a result of the tangential force T acting on the crankshaft and its reaction in the crankshaft support. It is variable and can be represented as a function of the rotation angle of the crankshaft φ.

Torque per cylinder – The tangential force T acting inside the crank mechanism in each cylinder creates a torque that is transmitted to the crank of the corresponding cylinder. This torque is calculated using the formula:

$$T_{tq} = T(\varphi).R\ [Nm]$$

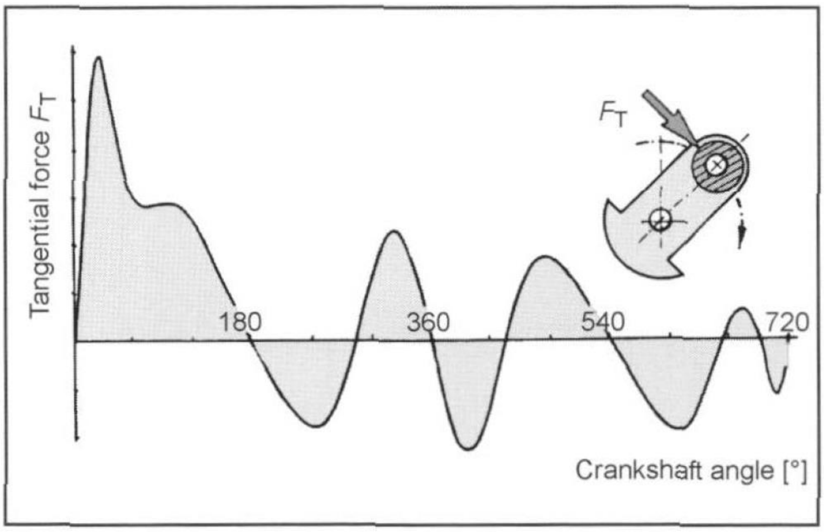


Fig. 93 Course of tangential force (104)

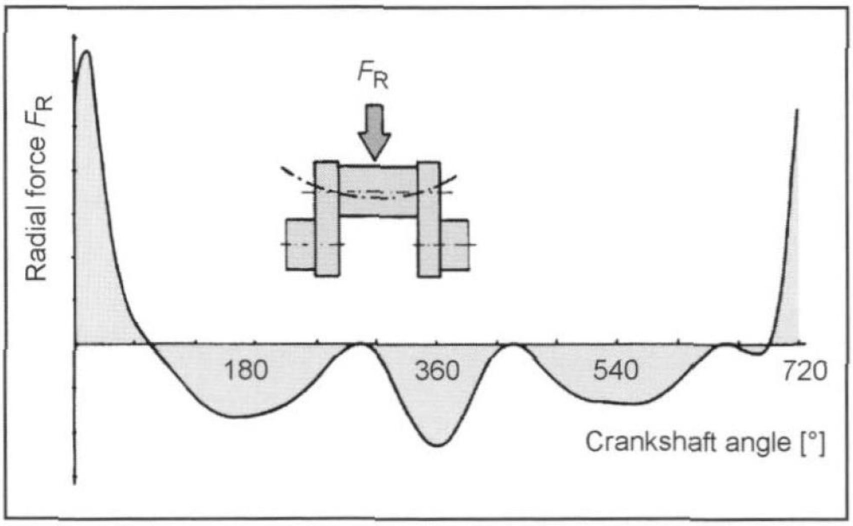


Fig. 94 Course of normal force on piston (105)

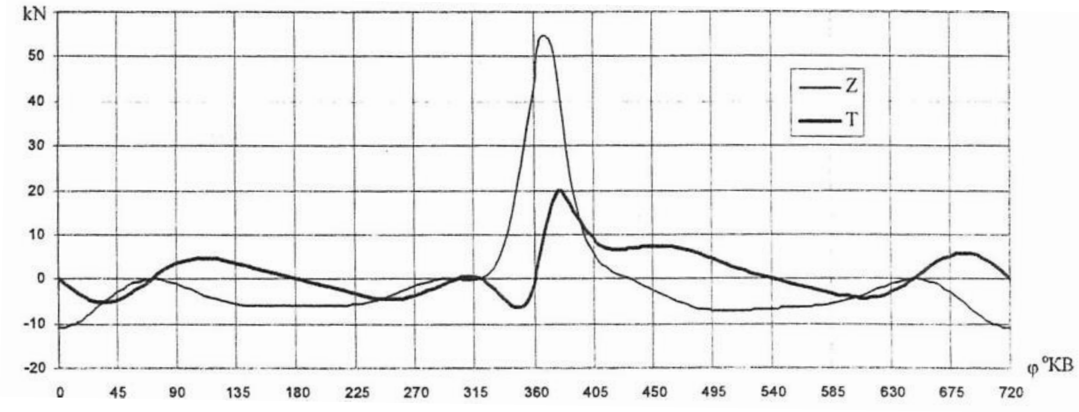


Fig. 95 Force on crank and tangential force of diesel engine (106)

The torque of a cylinder has the same character of variation as that of the tangential force, and its alternating has a starting point related to the rotation of the particular crank.

Total torque of a multi-cylinder engine. In a multi-cylinder engine, the torque from all cylinders is summed successively towards the power output end of the crankshaft, where the total (cumulative) torque is obtained. In all cylinders, a torque of relatively the same magnitude and shape is produced, but the starting moment differs. The torque, produced in each individual cylinder, is phase-shifted relative to that in the first cylinder according to the angular position of the crankshaft crank throws and the firing order of the engine. The total torque is obtained by summing the deflated torques from all cylinders. The variation curve is cyclic with a period equal to the angle between the operating strokes of two successively operating cylinders. Depending on the number of cylinders i the period is determined:

$$\vartheta = \frac{720°}{i}.$$

9.2. Estimation of Engine Friction

An important moment during the preliminary engine proposal is an initial estimation of the frictional losses in the IC engine and especially in the engine cranktrain because they are the most noticeable.

As already mentioned, the cranktrain transforms the reciprocating motion of the piston caused by the gas pressure force into the rotational motion of the crankshaft used afterwards to drive the vehicle. During operation, mechanical friction between the moving engine components and their contact surfaces appears, which decreases the overall efficiency of the engine. The total mechanical losses of an engine are divided among the main components as follows: 40%-50% to the piston group assembly, 20%-30% to the bearing and crankshaft, 7%-15% to the valvetrain, and 20%-25% to engine auxiliary units. While the friction is distributed 50%-68% to the piston assembly, 25%-35% to the bearing and crankshaft, and 10%-20% to the valvetrain.

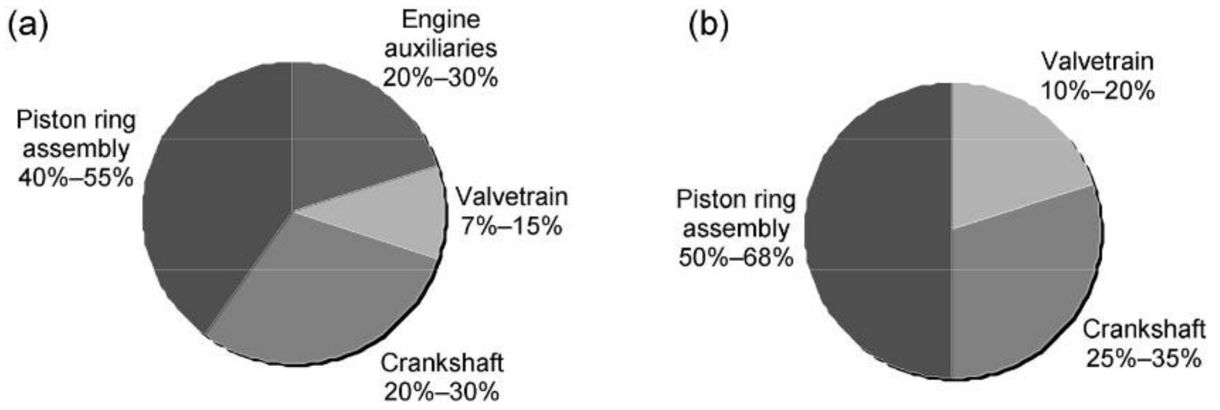


Fig. 96 Distribution of total mechanical losses (a) and friction (b) in a diesel engine (107)

It is obvious that the cranktrain has a significant impact on the engine's mechanical friction losses. The effort is focused on the reduction of mechanical losses and friction between the moving components of the engine mechanical subsystems in order to improve the overall efficiency of the engine. This, in turn, will result in lower fuel consumption and lower production of harmful and CO_2 emissions from the ICE.

During the design process, a variety of practical and mathematical approaches, methods, and models are used to estimate and predict the mechanical frictional losses on IC engines, which can be divided into separate groups. The experimental estimation methods include direct measurements of losses on the engine: of the entire cranktrain, crankshaft and main bearings, valvetrain, oil pump, auxiliary units, and the entire engine. In contrast, calculation models include analytical, empirical, semi-empirical, computational and FEA models.

The utilisation of experimental methods is available first after building a first functional prototype. It is a demanding process and often does not offer a detailed description of the friction in particular engine subsystems and components. In contrast, using a large amount of input data, modern computational techniques are capable of presenting a more precise description and estimation of the friction losses. However, a variety of the required input values are not easy to find and some of them may be retrieved from engine measurements. In the case of developing a new engine, where experimental methods cannot be used, computational methods are very practical for initial estimation.

Various models exist for both spark-ignition and compression-ignition engines (in terms of piston groups), as well as models for the main bearings and the connecting rod bearings. For quick initial estimation of the mechanical losses, simple analytical methods, approaches, and techniques are applied. Despite the fact that they are not able to provide the most accurate results, they benefit from less time and less input data needed, in comparison with more advanced and detailed computational models. They offer a relatively easy calculation and prediction of the frictional power of the crank train and its components.

The engine's mechanical losses can be expressed as the difference between the energy released during the combustion and expansion of the gases inside the cylinders (indicated power) and the energy obtained at the shaft output (brake power). Mechanical losses are often expressed by the parameter friction mean effective pressure (FMEP), which is the difference between the indicated mean effective pressure (IMEP) and the brake mean effective pressure (BMEP). [28, 32, 37, 70]

$$FMEP = IMEP - BMEP.$$

Empirical formulas are commonly used for an approximate estimation of the frictional losses in an IC engine. Many friction models depend only on engine operating speed and/or mean piston speed. A common friction model, the Chen-Flynn relation, depends on the maximal combustion (cylinder) pressure p_{max} and mean piston speed c_s. [42]

$$FMEP = 0{,}138 + 0{,}005.\, p_{max} + 0{,}06.\, c_s.$$

List of Used Quantities and Their Units

A_p	$[mm^2]$	piston area
B	$[mm]$	engine bore
$B/S, x$	$[-]$	bore to stroke ratio
c_s	$[\mathrm{m}/s]$	Mean piston speed
L	$[m]$	connecting rod length
M / T	$[Nm]$	engine torque
m_M	$[kg]$	connecting rod mass
m_{KA}	$[kg]$	total rotating mass
m_{MB}	$[kg]$	piston group mass
m_R	$[kg]$	rotating moving masses
m_j	$[kg]$	reciprocating moving masses
i	$[-]$	Number of cylinder
n	$[min^{-1}]$	engine speed per 1 minute
n_s	$[s^{-1}]$	engine speed per 1 second
p_e	$[MPa]$	brake mean effective pressure
p_g	$[MPa]$	cylinder gas pressure
p_o	$[MPa]$	pressure in crankcase
P_e	$[kW]$	engine brake power
$\mathrm{P_p}$	[N]	gas force
P_j	[N]	inertia force from reciprocation motion
P_R	[N]	inertia force from rotation
P_Σ	[N]	total force
r	$[mm]$	radius
R	$[mm]$	crank radius
t	$[s]$	time
j_A	$[m.s^{-2}]$	crank acceleration
j	$[m.s^{-2}]$	piston acceleration
S	$[m]$	piston stroke
S_A	$[m]$	crank trajectory

$\boldsymbol{S_x}$	$[m]$	piston trajectory
$\boldsymbol{V}$	$[\mathrm{m/s}]$	piston velocity
$\boldsymbol{V_A}$	$[\mathrm{m/s}]$	crank velocity
$\boldsymbol{V_z}$	$[\mathrm{dm}^3]$	engine displacement
$\boldsymbol{\alpha}$ / $\boldsymbol{\varphi}$	$[°]$	crank angle
$\boldsymbol{\beta}$	$[°]$	connecting rod angle
$\boldsymbol{\varepsilon}$	[-]	compression ratio
$\boldsymbol{\lambda}$	[-]	connecting rod ratio
$\boldsymbol{\eta}$	[-]	efficiency
$\boldsymbol{\omega}$	$[rad/s]$	angular velocity of crank
$\boldsymbol{\omega_o}$	$[rad/s]$	angular velocity of connecting rod

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