Preliminary proposal of an internal combustion engine as a range extender

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Summary

The current paper focuses on a research for a basic preliminary proposal of an internal combustion engine which will be used as a range extender for an electric vehicle. It presents a basic description of the power train concept and also important points, problems and challenges at the beginning of the engine design process. Some of them are the proper selection of the engine configuration (number of the cylinders, their configuration, type of the cycle, etc.) or determination of the main engine parameters (the diameter of the piston (bore) and its travel distance (stroke) and the connecting rod length) with respect of how they affect the engine cycle, engine operation and engine performance. For this aim different simulation models based on parametric modelling will be used. The engine output parameters will be presented and commented.

Keywords: hybrid vehicle, range extender, engine proposal, engine design, bore/stroke ratio, connecting rod length

1. Introduction

1.1. Range-extended electric vehicle

One of the possible power train configurations for hybrid vehicles (HV) represents an electric vehicle (EV) with extended range. This kind of power train has to eliminate one of the main disadvantages of the electric vehicles – the limited range. That means the pure electric vehicle is supplemented with additional (besides the main electrical) power source (auxiliary power unit) to extend its range in a more acceptable limit. Most frequently this source of power is a specially designed small internal combustion engine – so called range extender or range extender engine. It drives an electric generator to charge the batteries to supply the electric motors of the vehicle with electricity. There are some names (terms) for this kind of vehicle: extended-range electric vehicle (EREV) or range-extended electric vehicle (REEV) or range-extended battery-electric vehicle (BEVx).

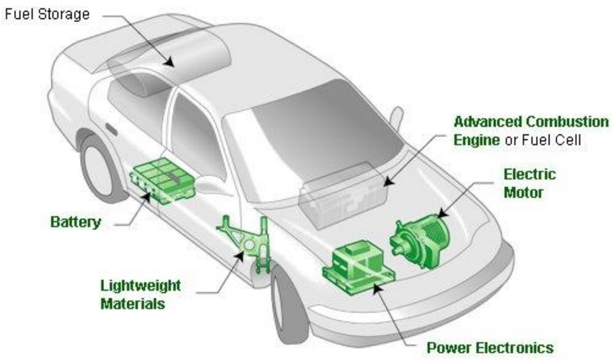

Fig. 1. *Range extended electric vehicle (1)*

The REEV operates in a pure electric mode in case there is enough power in the electric battery storage system. The auxiliary power system is connected when the main one (the batteries) is nearly exhausted. It is desirable to use the vehicle mainly in the electric mode, which requires a regular battery charging from a external source (from a charging station or a wall power plug) in the time the vehicle is not in use (etc. during the working day, at night). This means the battery system has to be designed to provide enough power for a daily use (range up to 200 km). So, the range extender engine will be used to ensure occasional longer journeys. (4)

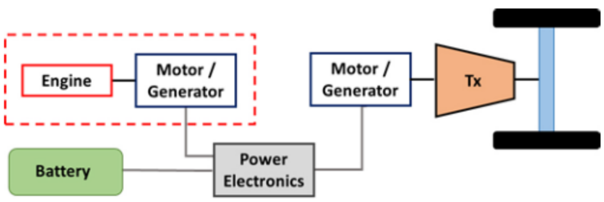

Fig. 2. *Diagram of a REEV power train (2)*

1.2. Internal combustion engine as a range extender

A competitive range extender engine for an electric vehicle should meet all requirements and demands. Some of them are low fuel consumption, excellent efficiency, low exhaust emissions, low costs, small overall assembly dimensions and weight, good NHV (Noise, Vibration, and Harshness) properties and etc.

The range extender engine has to be small, compact and light. Single, two or three cylinder configurations, inline or V-type ones are possible. This means a displacement of the engine up to around 1000 cm^3. Natural aspire petrol engine are most suitable, because they are uncomplicated and cheaper in comparison with turbocharged and diesel ones. Such an engine will be able to produce up to 40 kW brake power (naturally aspired

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versions), which is enough for a small family (compact) electric vehicle.

Fig. 3. *Range extender unit Getrag (3)*

Usually an engine with a rated power of 25 to 30 kW can provide an acceptable range and travel speed. The biggest range extender systems are able to extend the total vehicle range up to 1000 km. (4)

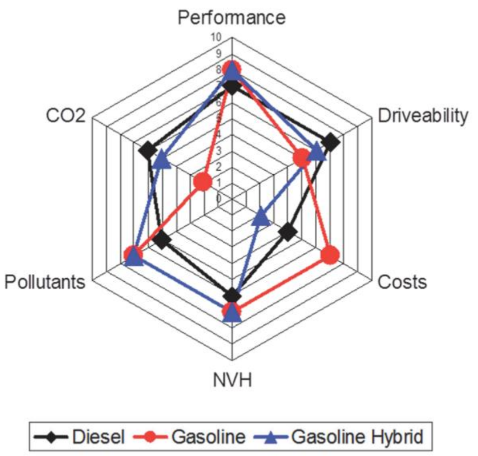

Fig. 4. *The comparison of RE development targets (4)*

2. Preliminary proposal of ICE main parameters

2.1. The main parameters of an engine

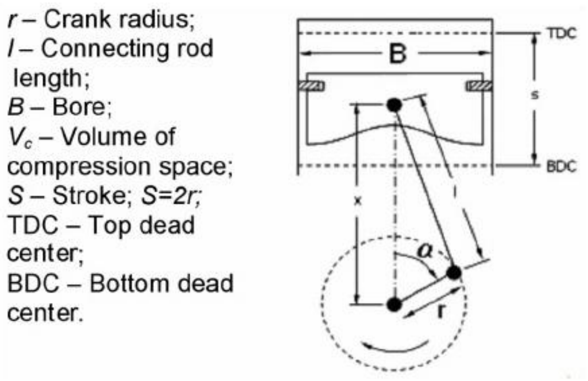

Fig. 5. *The main geometric properties of a reciprocation internal combustion engine (5)*

Generally, the main and most important parameters of every reciprocating internal combustion engine, which should be considered during the design process, are the engine cylinder bore B (the diameter of the cylinder), the piston stroke S (the distance between the upper and lower position of the piston) and the length of the connecting rod l. These parameters describe the engine geometrically. The relations (proportion) between them significantly affect the design, dimensions, output performance and behaviour of the engine. The parameters are shown in fig. 5.

2.2. Basic research goals

This paper will focus on a simple research which reveals the impact of the cylinder bore, the piston stroke and the length of the connecting rod on an internal combustion engine performance with consideration of the requirements and application of the engine as a range extender for an electric vehicle. It will also prepare a basic input data for further research. This will be done by using of computer simulations in specialized software. Also a simplified CAD design proposal will be presented.

2.3. Simulation model description

GT-SUITE is a leading software simulation system used in all fields of the automotive industry. It offers a lot of capabilities and functionalities for ICE design and simulation. For our investigation, a simple simulation model of a single-cylinder engine will be used. This model consists of two parts (sub models): a part for basic engine analysis of the engine cycle and a part for analysis of the mechanical properties of the engine crank train system. It contains all necessary input information (dimensions, physical properties, etc.) to describe a given engine. So, the effect of different engine parameter changes can be shown. The initial main parameters of the single-cylinder natural aspired petrol engine are shown in table 1.

Table 1. *The initial main properties of the model engine.*

Bore B [mm]	86
Stroke S [mm]	86
Crank radius R [mm]	43
Connecting rod length l [mm]	175
Compression ratio ε [-]	9,5

The value of the compression ratio is set to 9,5. The complete simulation model is shown in fig. 6. At this stage of the engine proposal a lot of unknown parameters are available, so some assumptions should be done. Some items of the engine that can also significantly affects the engine behaviour will be considered and maintained similar for all variants, e.g., compression ratio, ignition and valve timing, intake and exhaust system, etc., however their further design, tuning and optimization are needed.

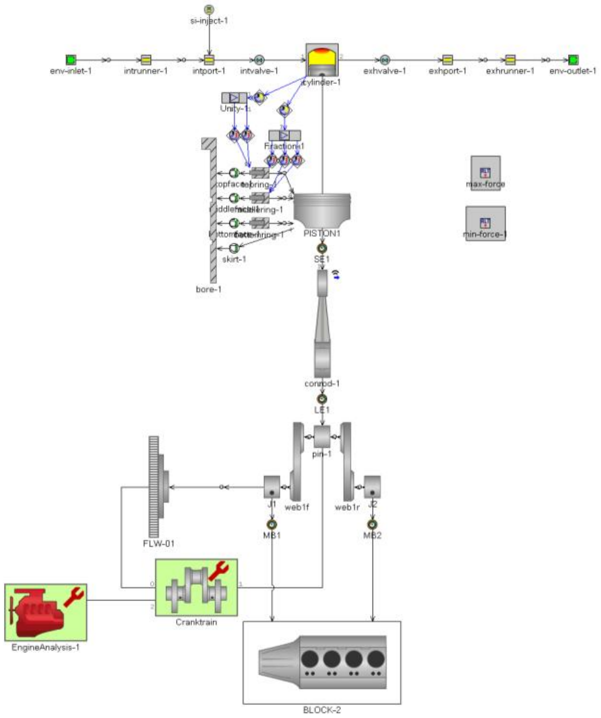

__Fig. 6.__ Single cylinder engine simulation model in GT-SUITE

3. Bore/Stroke (B/S) ratio

3.1. Basic description

One of the basic questions that should be solved firstly, at the beginning of the engine design process is the proper determination of the ratio between the cylinder bore (the diameter of the cylinder) and the piston stroke (distance between the TDC and the BDC) – so called Bore/Stroke ratio (B/S ratio). The initial model of the crank train considers an engine with equal value for both the bore and the stroke. This configuration is called a square (square engine). Of course, this ratio can be different from 1. In case the ratio value is higher than 1, the bore is bigger than the stoke and this engine configuration is called over square or short stroke engine. And vice versa: if the ratio value is lower than 1, the engine bore is smaller than the stroke and the configuration is called under square or long stroke engine.

The value of the bore/stroke ratio also affects the overall dimensions and weight of the engine, the design the crankshaft, but also the connecting rod length, the average speed of the piston and its height, the compression ratio, the size (area) of intake and exhaust valves, the engine vibrations, the cylinder surfaces area, etc.

An over square engine can be more compact, the velocity of the piston will be lower and 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.

Usually an engine with a small value of cylinder bore and large value of the piston stroke is used for better low speed performance (better low end torque for convenient city driving, trailer towing etc.). By contrast, an engine with higher value of cylinder bore and smaller value of the stroke is better for high speed performance.

Higher piston speed increases friction and wearing, but it also increases load forces in the connecting rod. In some cases it is possible to reduce the stresses in the connecting rod by using of a longer connecting rod in combination with shorter and lighter piston. [1, 17]

3.1. Definition of the assignment

For preliminary research different bore/stroke ratios, which means different combinations of the bore and stroke will be considered. It is suitable to maintain the engine displacement at a constant value, to find out the influence of the ratios on the engine behaviour. Next, the 9 different ratios for an engine with cylinder displacement of 500 cm^3 will be studied. An overview can be seen in table 2.

***Table 2.** Review of considered bore/stroke ratios*

B/S Ratio	Bore B [mm]	Stroke S [mm]	Displ. [cm3]
0,6	73	120	502,2
0,7	76	109	494,5
0,8	80	100	502,6
0,9	83	92	497,8
1,0	86	86	499,6
1,1	89	80	497,7
1,2	91	76	494,3
1,3	94	72	499,6
1,4	96	69	499,4

At this point the considered engine variants can be compared from the point of view of mean piston speed (average distance travelled by the piston per unit of time), which is defined by the following relation:

$$c_s = \frac{2.S.n}{60} = \frac{S.n}{30}\ m/s \qquad (1)$$

In fig. 7 the mean piston speed of all variants is shown. We can note that the engine with lower mean piston speed will produced low noise and vibrations.

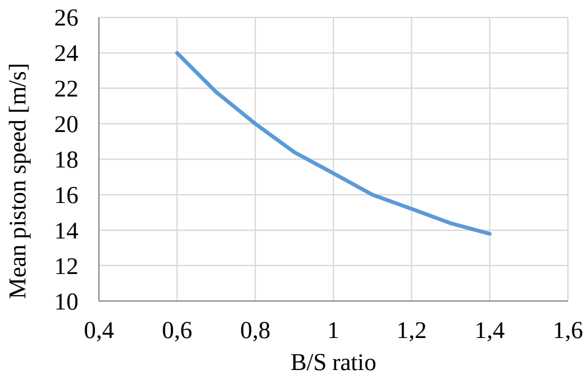

__Fig. 7.__ Mean piston speed at 6000 rpm

3.2. Simulation results overview

Some of the output parameters of the engines, as the maximal brake power produced at a specific speed and brake mean effective pressure and brake fuel consumption at that speed, are shown in table 3.

Table 3. *Results from engine simulation*

B/S ratio	Brake power [kW]	Speed n [rpm]	BMEP [bar]	BSFC [g/kWh]
0,6	22,3	5250	10,14	251,89
0,7	23,8	5500	10,50	248,29
0,8	25,6	5750	10,65	246,29
0,9	26,8	6000	10,77	245,83
1,0	28,0	6250	10,76	246,20
1,1	29,0	6250	11,19	242,00
1,2	29,5	6500	11,03	243,33
1,3	30,5	6500	11,26	241,19
1,4	31,0	6500	11,45	239,55

The output power, which the engines produce, is also presented graphically in fig 8. We can see that the first two engines with the lowest bore/stroke ratio (0,6 and 0,7) does not meet the requirement for the power. It produces maximal power of 22,3/23,8 kW at 5250/5500 rpm. In fig. 9 a comparison of the output power of all variants at the speed of 5250 rpm is shown. It is clear that the engine with higher B/S ratio performs better.

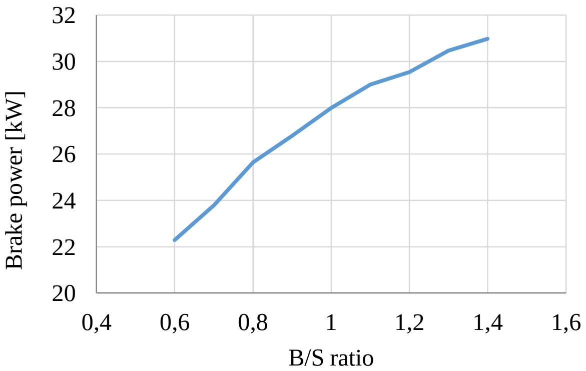

Fig. 8. *The output power of the different engine variants*

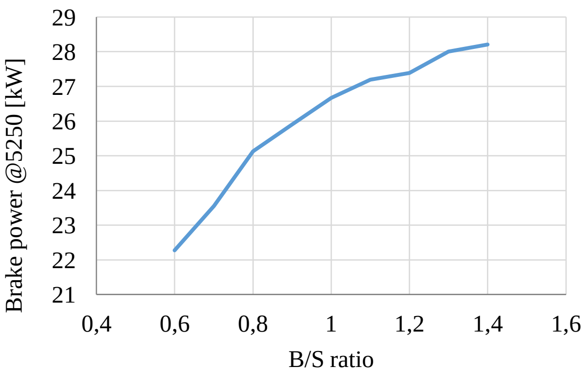

Fig. 9. *The output power of the engine variants at 5250 rpm*

In fig. 10 the specific brake fuel consumption at the speed where the engine produces its maximal power is shown. We can see that the engine variants with higher bore/stroke ratio have a better fuel economy.

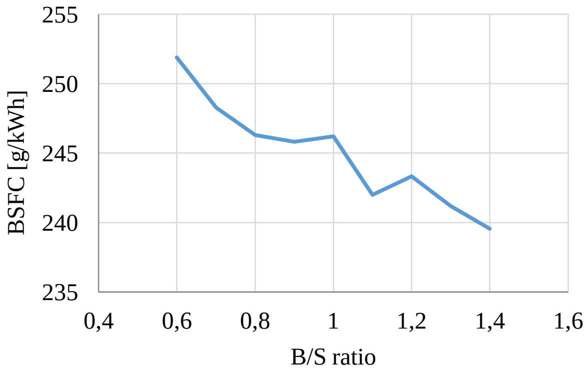

Fig. 10. *The brake specific fuel consumption (BSFC) of the engine variants*

In fig. 11 the brake mean effective pressure is presented. Mean effective pressure is a quantity (a fictive one, it is not a real pressure) which measures the ability of an engine to work independently to its displacement.

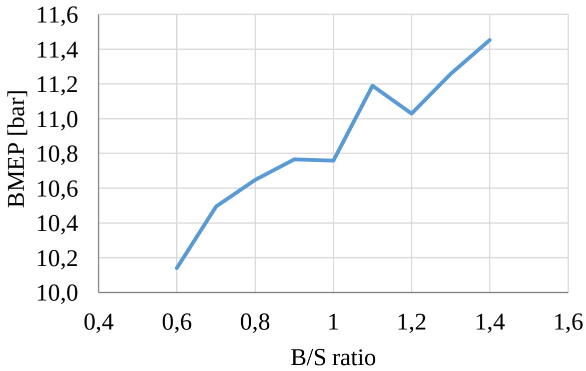

Fig. 11. *The brake mean effective pressure (BMEP)*

In next figure (fig. 12) the change in the maximal value of the peak pressure is shown. It is achieved at 4250 rpm for all variants. A higher peak pressure means a higher load of the engine components. This should be taken into account during the design and optimization of all separate engine components (connecting rod, crank shaft, bearings, etc.).

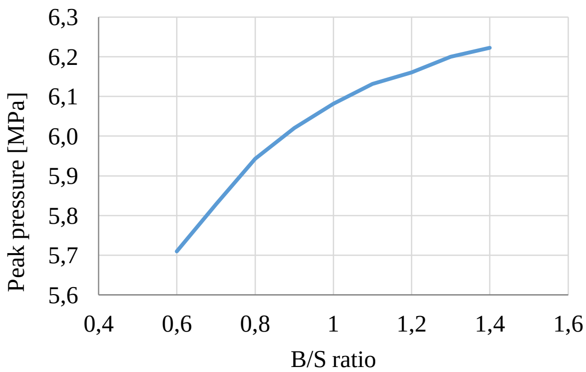

Fig. 12. *The change of the peak cylinder pressure*

Then the change in the brake torque of the engine variants is presented. It is clear that torque rises with rising B/S ratio, but there are also some drops, e.g., at ratio 0,9 and 1,2.

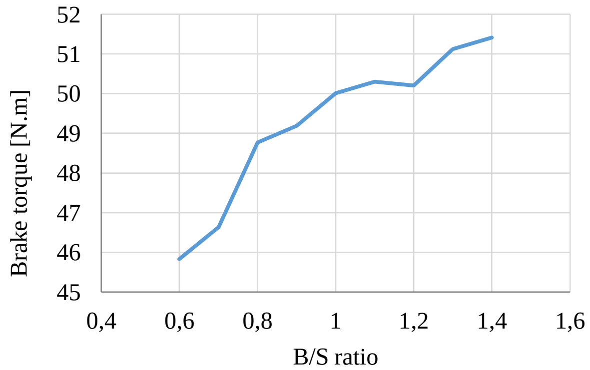

***Fig. 13.** The brake torque*

Fig. 14 presents the indicated efficiency of engine at the speed where the maximal brake power is achieved. Fig. 15 shows the maximal value of indicated efficiency of the engine variants.

The over square variants achieve the maximal power with better efficiency. An interesting variant for further analysis can be the one with B/S ratio 1.1.

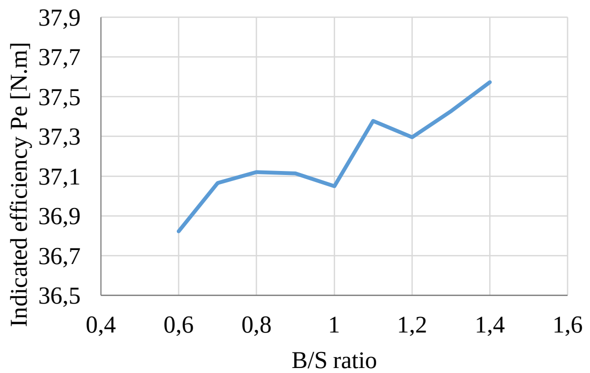

***Fig. 14.** The change of the peak cylinder pressure*

The square engine has the highest maximal efficiency. It is possible to notice the slight drop in over square variants.

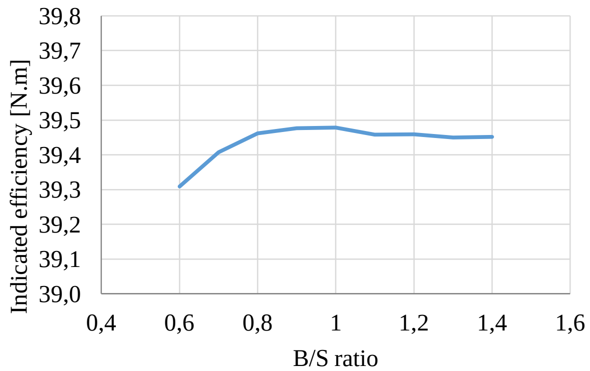

***Fig. 15.** The change of the peak cylinder pressure*

4. Connecting rod ratio

4.1. The connecting rod

One of the main components of the internal combustion engine (ICE) crank train is the connecting rod. It joins physically the piston to the crankshaft and of course it also interconnects their motions (reciprocation and rotation). The weight of the connecting rod affects the resulting forces in the crank train. It is very important to design the connecting rod properly. The lighter one will improve the response and acceleration of the engine. On the other hand good mechanical properties (strength) are also very essential. A well designed connecting rod should handle the load during the whole cycle in any engine operation mode – to withstand the compression and tensile load forces. At this point also a very important moment is an appropriate design of piston with optimization of its weight, and at last but not at least a selection of corresponding engine operation speed (rpm – revolutions per minute). So, the main effort is to design an enough light and strength connecting rod for a stated crank train system. In some cases, a modification of the length of the connecting rod (the engine stroke remains the same) will allow to obtain in some situations a slightly better engine performance or a slightly longer engine lifespan. So this article will research what a real benefit (what difference) of changing the connecting rod length is.

***Fig. 16.** Connecting rods with different design and length (6)*

4.3. Calculation of the connecting rod ratio

The length of the connecting rod is related to one of the basic and most important ICE parameters, which is so called connecting rod ratio (mostly labelled with λ). This parameter shows the geometrical configuration of the engine crank train and also could significantly affect the engine cycle, resulting behaviour and lifespan of the engine. It presents a mathematical relationship between the connecting rod length l and the crank radius R (which is a half from the engine stroke S). The ratio is defined by the follow relation:

$$\lambda = \frac{R}{l} = \frac{S}{2.l}. \qquad (2)$$

Sometimes the connection rod ratio is presented as a relationship between the connecting rod length l and the length of the engine stroke S. The definition is given with the relation below:

$$n = \frac{l}{S}. \qquad (3)$$

However, in this paper the first definition will be used. Also only the impact of the connecting rod length change will be studied.

In fig. 5 the main parameters of the engine crank train are shown. The length of the connecting rod l is defined as a distance between the axis (centre) of the crank pin and axis (centre) of the piston pin (gudgeon/wrist pin). The crank radius is defined as a distance between the axis (centre) of the crank pin and the axis (centre) of the main journal of the crankshaft.

Usually there is a limited range of values for the connecting rod ratio used in most conventional ICE. This comes from the requirements for the engine design – the connecting should be longer than engine stroke (it is not possible to be the same length as the stroke) and should not be too long (already a length of twice the stroke will make the engine very tall). Nowadays, the usual value of the ratio λ is between 0,23 (a longer rod) and 0,33 (a shorter rod), which means that the length of the connecting rod is approximately from 1,5 to 2,2 times bigger the engine stroke.

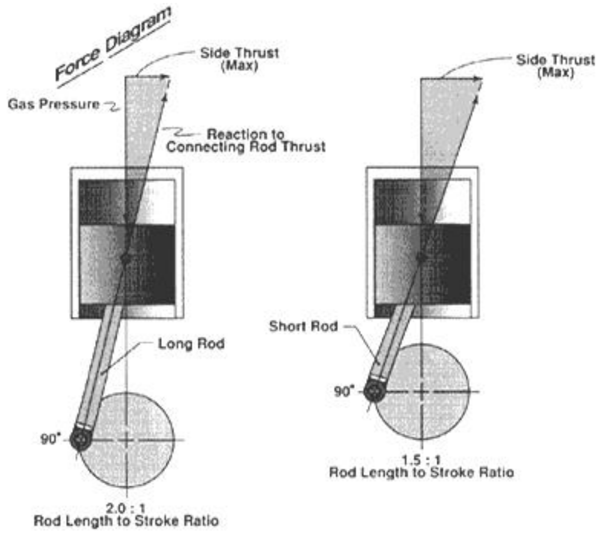

Fig. 17. *Comparison of force load in crank train (7)*

The length of the connecting rod l (at same value of engine stroke) affects the angle β between its axis and the axis of the cylinder and thus, changes the kinematic properties of crank train. This angle determines the force load of the piston skirt and the cylinder wall, and thus also influences the friction between them and force load of the connecting rod. E.g. a shorter connecting rod (a higher value of λ) will increase the angle and thus bigger side forces will occur, the friction and the wear of the piston skirt and the cylinder wall surface will increase, and last but not least, a little bit higher engine operating and oil temperatures will occur. On the other hand, a shorter connecting rod means a compact engine design – e.g., an overall lower and lighter engine block or shorter pistons can be used. The engine will have a better throttle response and low end (speed) torque. Also the ignition timing can be adjusted for an additional slight increase. This case is suitable for a comfortable everyday driving and vice versa.

The charge exchange process is also affected: during the intake stroke the cylinder vacuum after TDC increases faster (the piston moves faster). This allows using bigger intake manifold volumes and intake ports.

Ignition spark advance should be also modified: an earlier timing (more advance, a few degrees) is required, as the chamber volume is larger (piston is farther from TDC) at the same point of rotation; the engine may be also less knocking sensitive, as the chamber volume increases more rapidly ATDC, lowering combustion pressure. [3, 12, 14, 15]

Fig. 18. *An example for different length of the connecting rod and piston height (8)*

4.3. Long connecting rod – basic description

A longer connecting rod causes the piston to stay longer time at and around TDC. This keeps the combustion chamber volume smaller and extends the compression state, which improves the combustion (it is faster) and cylinder pressure and temperature. It is possible to obtain a slightly more power out of the air/fuel mixture. The engine will performs better (torque and power) at middle and higher speeds. On the other side, the engine friction will be reduced, caused by reduced connecting rod angle (smaller side force between the piston and cylinder wall). For this case, a lighter piston with a shorter compression height can be used which balances the slightly higher connecting rod weight.

The filling of the engine cylinder (volumetric efficiency) is worsened due to reduced air flow velocities into the cylinder mainly at low engine speeds. Immediately after the TDC the speed of the piston will be slightly lower and the maximum value will be gained later. The low end torque and throttle response will be worsened. For this engine configuration longer intake manifold runners with slightly smaller port diameters will ensure higher flow velocities. Also the camshaft profiles should be selected with caution. E.g., cams with longer duration will cause a decrease of the cylinder pressure during the close period of the intake stroke. [3, 12, 14, 15]

4.4. Short connecting rod – basic description

A shorter connecting rod will ensure better flow velocities during the intake and exhaust strokes, mainly at lower engine speeds. This will improve the low end torque, caused by improved (higher) cylinder vacuum mostly after the TDC, when the intake stroke begins. Higher flow velocities during the intake stroke allow obtaining a better mixed and more homogenous air/fuel mixture in the engine cylinder, which also positively affects the engine performance. The piston velocity is higher after the TDC and this increases the volume of the combustion chamber volume faster, which delays the position of the maximum cylinder pressure. The timing of the cams can be more radical (mainly intake valve closing).

On the other hand, a higher piston velocity after TDC negatively affects the combustion of the fuel mixture. Mainly at higher engine speeds it can lower the total cylinder pressure. The time at and around the TDC is shorter, so the piston will go down faster and the pressure and temperature in the cylinder will decrease also faster. [3, 12, 14, 15]

4.5. Result overview

Fig. 19 shows a simplified parametric design proposal of the connecting rod with basic parameters (dimension) built in CAD system PTC Creo Parametric. It can be modified to meet the connecting rod design configuration.

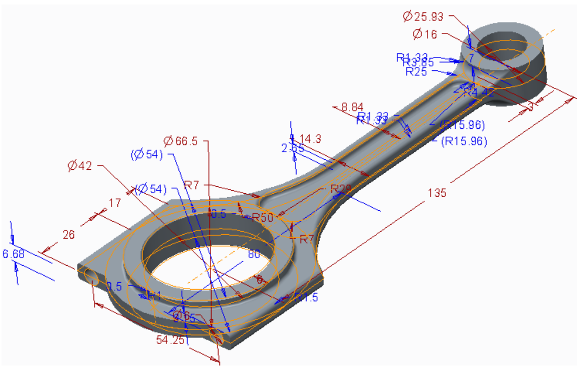

Fig. 19. *A parametric model of the connecting rod*

In table 4 the basic parameters are shown. Two additional variants are considered. The default length of the rod is 175 mm. Two others are defined – 100 mm and 250 mm. These lengths are moreover unrealistic and were selected to show the affect in a more distinguishable scale. At this stage, the current study does not engage the mechanical properties (strength, fatigue, safety factors etc.) of the design variants of the connecting rod. It aims only to show some of the parameters (etc. engine performance) which are influenced by changing of the connecting rod length. The square variant will be used.

Table 4. *Review of considered bore/stroke ratios.*

Conrod length [mm]	100	175	250
Conrod mass [g]	204	229	253
Conrod rot. mass [kg]	155	166	178

First of all, the impact of the connecting rod length to the crank train kinematic properties should be shown. Common definition is shown below.

$$x = R.\left[(1-\cos\alpha)+\frac{\lambda}{4}.(1-\cos 2.\alpha)\right] \tag{4}$$

$$v = R.\omega.\left(\sin\alpha+\frac{\lambda}{2}.\sin 2.\alpha\right) \tag{5}$$

$$a = R.\omega^2.(\cos\alpha-\lambda.\cos 2\alpha) \tag{6}$$

Fig. 20 presents the position of the piston. We can see a difference in the motion. In case of a shorter connecting rod, the piston travels faster after TDC, which causes a faster increase of the volume of the combustion chamber.

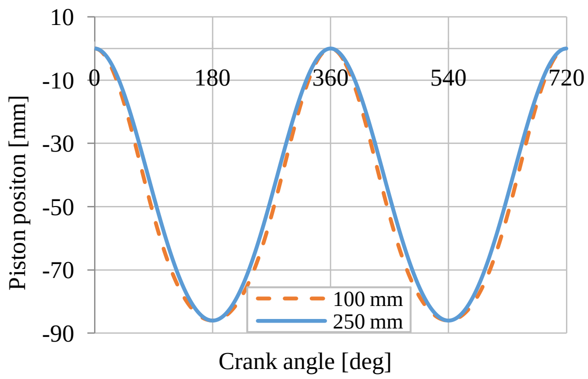

Fig. 20. *Piston position*

The velocity of the piston is presented similarly in fig. 21. Also there is evident the change of the velocity, where the shorter connecting rod increases the movement of the piston.

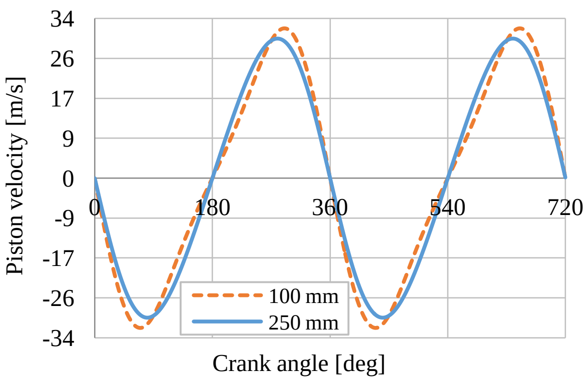

Fig. 21. *Piston velocity*

A significant change in piston acceleration can be also seen in fig. 22.

All three figures prove the affect trends, which were state in the theoretical description above.

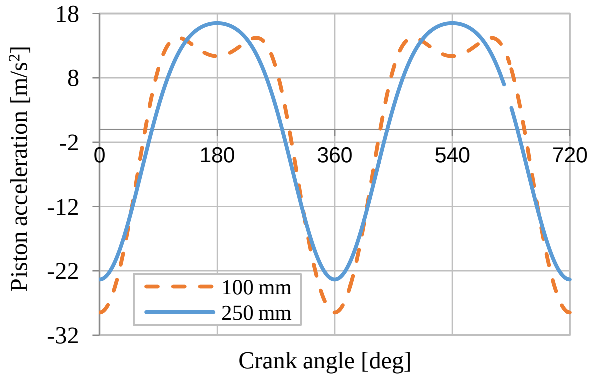

Fig. 22. *Piston acceleration*

As well it is very important and interesting to study the behavior of force load in relation to connecting rod length change.

We can see the change of maximal force loading axially the connecting rod in fig. 23. The shorter length of the connecting rod really decreases the force value nearly in all speed range. The absolute values are shown.

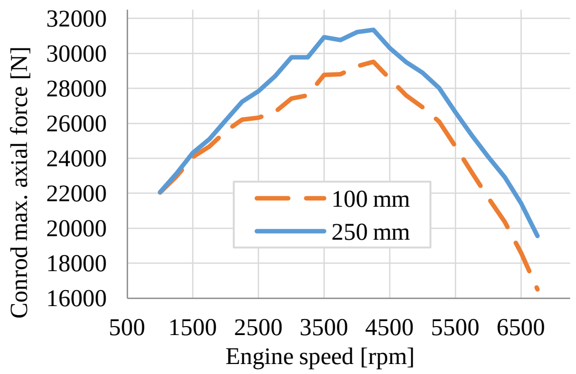

Fig. 23. *Maximal load force in the connecting rod*

The next figure 24 shows the progress of the force during the whole one cycle (2 strokes). Maximal values occur at engine speed of 4250 rpm.

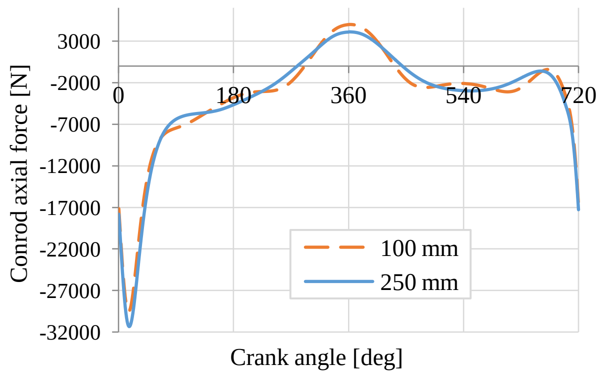

Fig. 24. *Connecting rod axial force at 4250 rpm*

On the other hand, we can notice how the shorter connecting rod increases the side force between the piston and cylinder wall. The progress is shown below. The peak difference is around 2500 N. This negatively will affect the friction and wear of the components.

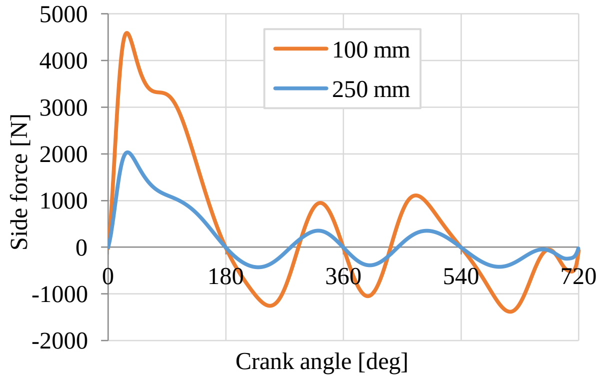

Fig. 25. *Side force at 4250 rpm*

In the next step, the impact on the engine output parameters will be presented. In fig. 26 and fig. 27 respectively, the brake torque and the brake power of the engines are shown. The engine with a shorter connecting rod produces a little more torque at low speeds (rpm). In the middle speeds there is a drop of torque in comparison with a longer rod. At upper (higher) engine speeds a shorter rod performs slightly better.

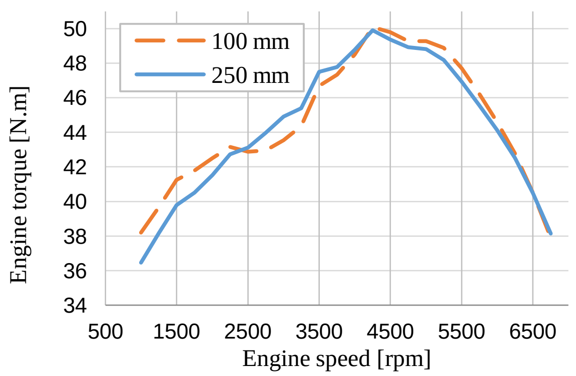

Fig. 26. *Differences in engine torque*

The comparison of the brake power of the engines with both connecting rod variants does not show any significant differences. At middle speeds the longer connecting rod performs slightly better, but at higher speeds there is noticeable worsening.

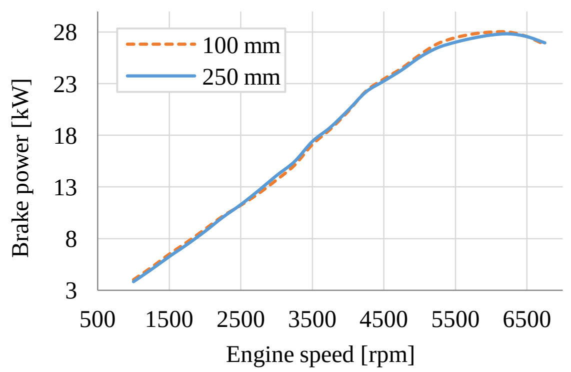

Fig. 27. *Differences in engine power*

5. Crank train design proposal

In this chapter a simplified design proposal of a single cylinder range extender will be described. Parametric CAD models of the crank train components will be used to show the differences in the design variants

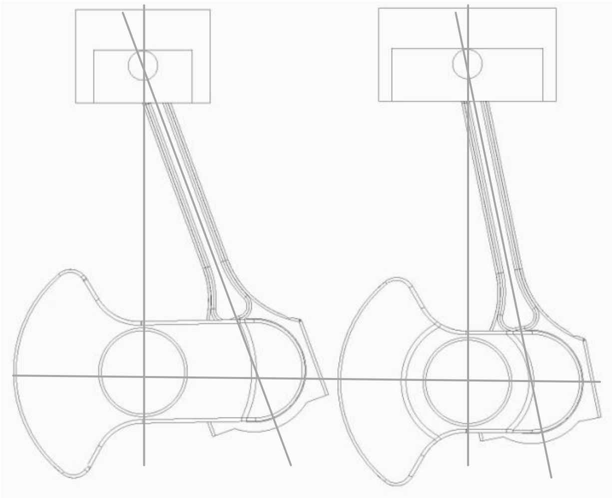

Fig. 28. *Side force at 4250 rpm*

In the figure above the comparison of variants with bore/stroke ratio of 0,6 and 1,4 is presented. Using the same connecting rod will make a difference in the crank train height of 51 millimetres and 25,5 millimetres in the weight for the long stroke variant. There is a space for optimization of the final proportion between bore, stroke and connecting rod length.

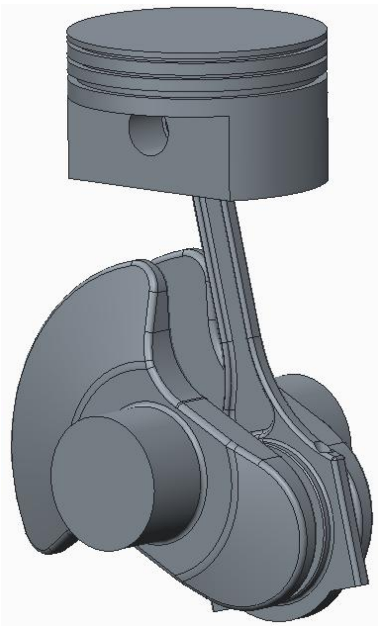

Fig. 29. *Single cylinder crank train*

6. Summary / Conclusion

This paper presents a short overview of the topic of the internal combustion engine used as a range extender for electric vehicles. It also deals with the main geometrical engine properties – the bore and stoke of the engine and the length of the connecting rod. The work also shows how the relations between these parameters can affect the performance and behaviour of the engine.

It presents a basic study of a possible configuration for a range extender – a single cylinder petrol engine with displacement of 500 cm^3. This study has to test the sensitivity of the output performance and behaviour of the engine. Different bore/stroke and connecting rod ratios are considered.

Without any further research and optimization of the parameters and engine systems it is not possible to give a final advice which configuration is the best and should be selected. However, at this stage there are some visible trends. Almost all engine properties (as power and torque, fuel consumption and efficiency) are improved with a higher value of bore/stroke ratio. Also it is clear how the change of the connecting rod length affects the engine properties. Nevertheless, a slightly over square configuration (B/S ratio 1,2-1,4) should be recommended. It will ensure a decent base for further research

A magic rule or relation for a right selection of the values of the ratios does not exist in the field of combustion engines. Both of them: the ratio between the bore and stroke of the engine, as well the ratio between the connecting rod length and length of crank radius depend on many different factors and on the engine purpose. They usually differ in some common ranges. There are a lot of different preconditions, requirements, rules and limitations that can determine which ratio could be the right one and also there is usually a space for varying with different possible combinations.

List of symbols

a acceleration of the piston (m/s^2)
B cylinder bore (mm)
c_s mean piston speed (m/s)
l connecting rod length (mm)
n engine speed, revolutions per minute (rpm)
R crank radius (mm)
S piston stroke (mm)
v velocity of the piston (m/s)
x travel distance of the piston (m)
α crank angle [°]
λ connecting rod ratio (-)
ω angular velocity (rad/s)

Literature and sources

[1] ALQAHTANI, Ali et MAZURKO, Paweł. Investigation into the effect of bore/stroke ratio on a single cylinder two stroke opposed piston engine. *Journal of KONES Powertrain and Transport*. February 2016. Vol. 23, No. 2. Pp 9-16. Available from: DOI: 10.5604/12314005.1213529

[2] CARLEY, Larry. *Understanding Rod Ratios* [online]. Engine Builders. [vid. 7.4.2017]. Dostupné z: http://www.enginebuildermag.com/2016/08/understanding-rod-ratios/

[3] *Connecting Rod vs. Stroke Analysis* [online]. Victory Library. [vid. 7.4.2017]. Dostupné z: http://victorylibrary.com/mopar/rod-tech-c.htm

[4] HAMMILL, Des. *How to Blueprint & Build a V8 Short Block for High Performance*. Dorchester: Veloce Pub, 1996. 112 p. ISBN 978-1-874105-70-1

[5] HAMMILL, Des. *The 4-Cylinder Engine Short Block High-Performance Manual*. Dorechester: Veloce Pub, 2011. 112 p. ISBN 978-1-845844-14-1

[6] HANSEN, Graham. *How to Build Big-Inch Chevy Small-Blocks*. North Branch: CarTech, 2011. 144 p. ISBN 978-1934709-66-5

[7] BAECHTEL, John et PARKHURST, Scott. *Competition Engine Building: Advanced Engine Design and Assembly Techniques*. North Branch: CarTech, 2012. 176 p. ISBN: 978-1-934709-62-7

[8] GT-Power User's Manual, GT-Suite version 7.4. Gamma Technologies Inc., September 2013.

[9] LINGENFELTER, John. *John Lingenfelter on Modifying Small-Block Chevy Engines: High Performance Engine Building and Tuning for Street and Racing*. New York: HP Books, 1996. 165 p. ISBN 1-55788-238-X

[10] MACEK Jan. *Spalovací motory*. Praha: Nakladatelství ČVUT, 2012. 262 s. ISBN 978-80-01-05015-6

[11] MARTIN, Isaac. *Ford Windsor Small-Block Performance: Parts and Modifications for High Performance Street and Racing*. New York: HPBooks, 1999. 201 p. ISBN: 978-1-55788-323-0

[12] MAVRIGIAN, Mike. *Modern Engine Blueprinting Techniques: A Practical Guide to Precision Engine Building*. Forest Lake: CarTech, 2013. ISBN: 978-1-61325-047-1

[13] MONROE, Tom. *Racing Engine Builder's Handbookhp1492: How to Build Winning Drag, Circle Track, Marine and Road Racingengines* [online]. New York: HP Books, 2014. ISBN: 978-1-101-50303-4 <http://www.myilibrary.com?id=707621>.

[14] *Rod Length Relationships* [online]. Stahl Headers. [vid. 7.4.2017]. Dostupné z: http://www.stahlheaders.com/lit_rod%20length.htm

[15] *Rod Ratio (Length/Stroke) Analysis* [online]. Mayfield Company [vid. 7.4.2017]. Dostupné z: http://www.mayfco.com/rods.htm

[16] SCRABA, Wayne. *1001 High Performance Tech Tips*. New York: HP Books, 1995. 169 p. ISBN 1-55788-199-5

[17] THORNHILL, D., DOUGLAS, R., KENNY, R. et FITZSIMONS, B. An Experimental Investigation into the Effect of Bore/Stroke Ratio on a Simple Two-Stroke Cycle Engine. *SAE Technical Paper* [online]. SAE International. September 1999. Available from doi:10.4271/1999-01-3342.

Images

(1) http://www.gedtv.us/images/webParts/extended_range_ev.jpg

(2) AGARWAL, A., LEWIS, A., AKEHURST, S., BRACE, C. et al., Development of a Low Cost Production Automotive Engine for Range Extender Application for Electric Vehicles, *SAE Technical Paper*, [online]. SAE International. April 2016. Dostupné z doi:10.4271/2016-01-1055

(3) http://electriccarsreport.com/wp-content/uploads/2010/02/Boosted-Range-Extender.jpg?x50765

(4) http://www.fev.com/fileadmin/user_upload/Media/Spectrum/en/Spectrum_49_E_WEB.pdf, page 4

(5) TARTAKOVSKY, L., BAIBIKOV, V., GUTMAN, M., VEINBLAT, M. et al., Simulation of Wankel Engine Performance Using Commercial Software for Piston Engines, *SAE Technical Paper* [online]. SAE International. October 2012. Dostupné z doi:10.4271/2012-32-0098. Page 5.

(6) http://www.howrah.org/images/articles/connecting-rods/connecting-rods-3.jpg

(7) http://forums.pelicanparts.com/uploads16/max_21260490831.gif

(8) http://img.photobucket.com/albums/v452/turbocharged400sbc/SI%20and%20SII%20Hybrid%203800%20V6/IMG_3832.jpg