

# I.C. Engine Having Performance Parameters Preferential for Automotive Operating Conditions

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## ABSTRACT

An idea of an I.C. engine with a disintegrated process is presented. As regards the process the engine consists of two specific systems: the cylinder-combustion chamber system and the intake-compression system. The working cycle takes place partially in the former and partially in the latter system. The disintegration of the process strongly influences the operating characteristics, fuel economy, ecological properties, and fuel tolerance of the engine. The torque characteristic of the engine could be highly elastic and preferential for automotive operating conditions. As regards the other criteria the proposed concept suggests certain superiorities in comparison with today's engines, but it points also to some problems which still have to be solved.

## INTRODUCTION

The performances of automotive vehicles are to a great deal dependent on matching the vehicle and engine. Decisive are the engine rating power and vehicle mass ratio and the engine torque full-load curve character. Two characteristic approximate torque curves common to today vehicle engines are presented in fig. 1. By the dashed line in the figure another torque curve is plotted, being rather uncommon for today engines. We find it as an interesting torque curve for vehicle engines, and a concept of an engine which could achieve this torque characteristic curve is proposed in the continuation. The plotted curves in the figure have quite occasionally the equal torque at maximum speed. In the figure are also plotted the power curves and an island inside the engine operation field in which today engines have their most economical working points. Finally the area of computed minimum specific fuel consumption of the here proposed engine concept is pointed out.

## WHAT DOES THE CONVENTIONAL ENGINE OFFER

Four stroke engines in all their varieties chiefly represent the power source for cars, lorries and off-road vehicles. Proper to all of them is the torque characteristic inside the area limited by full line curves in fig. 1. The dashed line is not their normal torque characteristic and it is questionable to reach it also by special

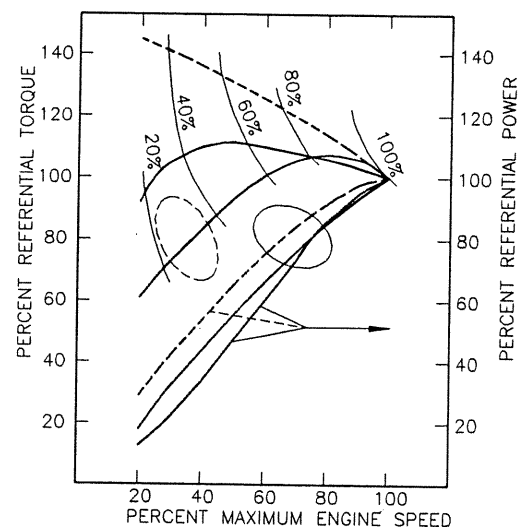


Fig.1 Character of full-load torque curves and location of the best fuel consumption domain inside the performance map common to conventional engines (full lines) and computed for the proposed (dashed lines) vehicle S.I. engine.

measures. The torque of engine depends (1) on the success of cylinder charging with fresh mixture expressed by volumetric efficiency, (2) on the quality of fuel energy transformation via heat in mechanical work, expressed by indicated efficiency, (3) on the mechanical losses of engine itself, expressed by mechanical efficiency, and (4) on the other limits like the thermal loads of cylinder assembly parts and the disturbances in combustion process like knock at S.I. engines. Influencing the volumetric efficiency by throttling is the most common case. Exploiting the dynamic phenomena in the intake and exhaust systems or applying a blower in the intake system, with or without aftercooling are usual measures for improving the volumetric efficiency. In the case of a mechanically driven blower by a C.V. transmission, the mass flow through the engine could be made to a great deal independent of engine speed, and the torque curve could be more in the direction plotted by the dashed line in fig. 1. The rest of the above mentioned factors, particularly the disturbances in the combustion

process are much more complex and also restrictive. The high mean indicated pressure at low engine speed, which regarding the charge could be realized, is questionable regarding the knock. Higher charge pressure and much more smaller piston speed and turbulence of charge dependent on it, reduce the speed of flame front, which means that there is a danger of knock. This also means that the high elastic torque characteristic is not attainable in conventional four stroke engine. The S.I. engine with such a torque characteristic must be resistant to knock, its inclination to knock is independent of engine speed and it should be resistant to destructive action of knock.

#### PROPOSED CONCEPT OF ENGINE

The proposed concept of engine is described more in details in [1]. It is here presented only briefly by a schematic diagram of the cylinder-combustion chamber system, from the point of view of design, fig. 2. In fig. 3 the complete engine is also schematically presented with one of possible versions of the intake-compression system in which the compressors of the compressor-cooler assembly are driven mechanically. One of the

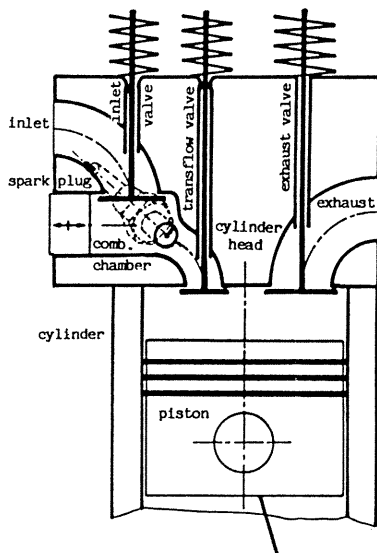


Fig.2 The proposed engine cylinder-combustion chamber system (from point of view of design) schematically presented.

compressors can also be a turbocompressor exploiting the energy of exhaust gas. For the engine it is characteristic that the working cycle goes on partially in the cylinder itself, partially outside it in the intake-compression system and in the combustion chamber, which in dependence on valve timing belongs to one or the other system. The compression phase goes on in the intake-compression system, then follows the inlet of the compressed medium in the combustion chamber, where the mixture is ignited and combustion goes on in it. After the opening of the transflow valve the combustion continues in both spaces: in the combustion chamber and in the cylinder. The expansion goes on in the cylinder-combustion chamber system, then follows the blowdown and the discharge of combustion products in the surrounding. Owing to separation of the particular phases of the process into separate systems we speak about the engine with a disintegrated process.

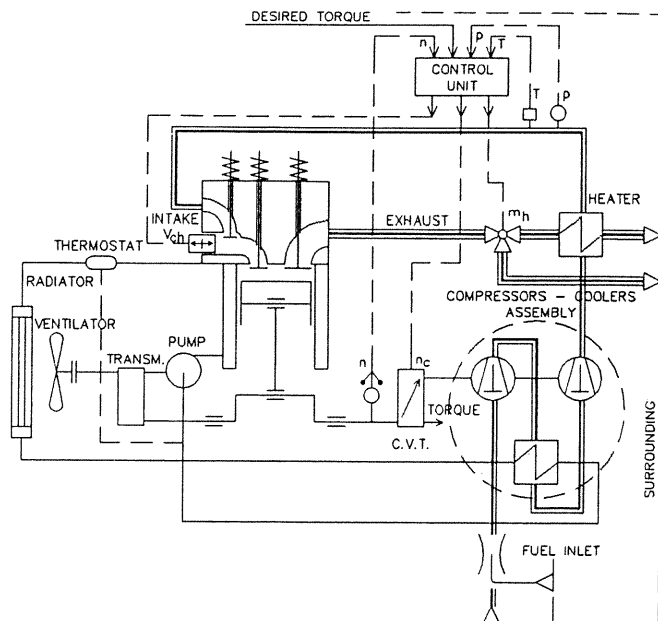


Fig.3 Version of engine intake-compression system with a mechanically driven compressors assembly, schematically presented.

#### Working Capability - Specific Power

The disintegration of the process enables a simultaneous course of particular cycle phases in separated systems. In each revolution of the engine shaft the process is completed. It means the engine is two-stroke acting, but liberated of the well known weaknesses accompanying the conventional twostrokers as regards the process and engine durability. The weaknesses like worse volumetric efficiency, fresh mixture escape and oil layer interruption owing to ports in the cylinder liner, are surpassed with the proposed concept of engine. The engine could be competitive to today four stroke engines. It also means that the working capability of the proposed engine compared to four-stroke engines could be doubled.

#### Inclination To Knock, Fuel Toleratation

The engine knock is the main limitation factor which does not enable to transform the normal torque characteristic of four stroke engines into the torque curve plotted by the dashed line in fig. 1. In the continuation we will discuss the promises of the proposed concept of engine from this aspect. The combustion chamber is separated from the cylinder, therefore its shape is independent of the geometry of the engine cylinder. The cross-sections of timing organs with reference to the combustion chamber can be relatively small. In such conditions the combustion chamber can be shaped to a great deal like a geometric cylinder; this means like a compact chamber, which in a relatively simple way enables a change of volume not inflicting its cylindrical geometry and compactness.

The mixture flow conditions in the chamber immediately before the combustion beginning defines the inlet stream which directly foreruns the combustion. The inlet mixture velocity is high, in the beginning even sonic, which guarantees a high turbulence of medium in the chamber also in the combustion phase. With inlet valve

location it is possible to attain also a settled whirling motion in the chamber. Besides, the flow conditions in the chamber are independent of engine speed, which means that the danger of knock at low engine speed does not increase. Consequently, the inclination to knock is independent of engine speed and owing to compactness of combustion chamber and high turbulence conditions in it, the danger of knock is reduced. There exists a possibility that the combustion starts and ends in a closed combustion chamber, or at later ignition the combustion goes on partially in the chamber partially in the cylinder. After opening of the transflow valve the combusted, burned and still unburned medium flows out of the chamber in the cylinder space, where the combustion continues and also ends. The circumstances in this case are approximately similar to the circumstances in the prechamber stratified charge engines.

On the other hand, the expected knock resistance of the proposed concept of engine also means lower sensitivity of engine to fuel as regards the octane number, and other physical and chemical properties of fuels. In short, we can speak about a multi-fuel capability of the engine.

Owing to separation of the combustion chamber it is to expect a reduced destructive action of knock limited to the combustion chamber and maybe even harmless. All the above mentioned proposals are presumptions for which in today engine practice we cannot get a confirmation.

#### Poisonous Emission

The poisonous substances ( $\text{CO}$ ,  $\text{CH}$ ,  $\text{NO}_x$ ) formation mechanisms remain unchanged also in conditions of the here proposed concept of engine. There is not to expect a difference in emission from primary combustion. Also in this case it will be necessary in order to reduce emission to exploit the direct and indirect measures known from conventional engines practice. Perhaps, larger specific power of two stroke acting engines and combustion in the separated combustion chamber could make possible to exploit the exhaust gas recirculation (EGR) to a greater extent without reducing the engine capability regarding the working properties. But, it is to expect an improvement of HC-emission from secondary sources like extinction of flame on the walls, crevices, absorption and adsorption of fuel in oil layer and deposits on the surfaces of combustion chamber. The cylinder itself should not be the generator of this emission, because the fresh air-fuel mixture should never be present in it. Hydro-carbons could originate from extinction and deposits on the combustion chamber walls, but owing to its compactness and high turbulence, the boundary layer is thin and therefore the total volume of the boundary layer is small. For both reasons the knock resistance of combustion chamber is high, permitting higher wall temperatures which reduce the thickness of the boundary layer even further. Higher wall temperatures also hinder the rise of deposits. From this follows that the secondary sources of HC compared to conventional engines could be highly reduced.

#### ENGINE PERFORMANCE MAP - COMPUTED

In previous chapters it was established that the factors which in S.I. conventional engines restrict the extreme of torque at lower engine speed are not restrictive in the proposed engine.

Owing to a disintegrated cycle separate computations were made (1) of the indicated processes in the cylinder-combustion chamber system, (2) of the indicated process in the intake-compression system, and (3) of the indicated parameters of the proposed concept of the complete engine. Finally, to get effective parameters, the engine mechanical losses had to be subtracted from the indicated ones and presented in engine performance map.

#### Indicated Cycle In Cylinder-Combustion Chamber System

The indicated performances of the working process for different characteristic parameters are computed by an appropriate model of the process in the cylinder-combustion chamber system. The characteristic parameters are also the control parameters controlling engine load within normal engine speeds. The cylinder-combustion chamber system total volume and the sum of the nominal combustion chamber plus the cylinder dead space volume ratio correspond to the compression ratio of four stroke engines. The same ratio of volumes, but with instantaneous volume of combustion chamber means the instantaneous expansion ratio ( $\xi$ ), a parameter of importance for the proposed concept of engine. Also the cylinder and crank mechanism geometries are identical to those in conventional engines practice. The control factors of the proposed concept of engine are (1) the pressure and (2) the temperature of medium in the intake-compression system immediately before entering in the combustion chamber and (3) the volume of the changeable combustion chamber. The analogous control factor at conventional four-stroke engines in general is the pressure of the cylinder charge, partially also of its temperature. Meanwhile the change of compression ratio, which is analogous to the change of combustion chamber volume of the here proposed engine, is not introduced in the conventional engines practice. That means that the proposed concept of engine introduces a new control factor which at partial load exploits the overexpansion as a thermodynamic measure which, together with intake medium temperature increased by regenerative preheating, contributes to improvement of the indicated process efficiency. Some computed results of the indicated process in the cylinder-combustion chamber system, for different values of control parameters ( $p/T/V_{c_h}$ ) at a given engine speed are presented by the points in the performance field in the upper part of fig.4. The particular points are connected together by the lines along which the indicated parameters are changed by change of control parameters, namely that one which, in the designation of the curve, is not quantitatively given. The diapason of quantities of control factors are evident from the figure.

The histograms of characteristic magnitudes of the indicated process like mass, pressures and temperatures in the cylinder and in the combustion chamber are presented for some examples in fig.5. For each example inside the diagrams the following data are printed: the geometrical data of cylinder-combustion chamber system, the engine operation data, the timing data, and the computed performances of process given by the mean indicated pressure and indicated efficiency. The diagrams 5A and 5B appertain to points marked by A and B on the upper envelope of the performance field in fig. 4. The diagram 5B<sub>1</sub> also belongs to point B only in comparison to the diagram 5B the

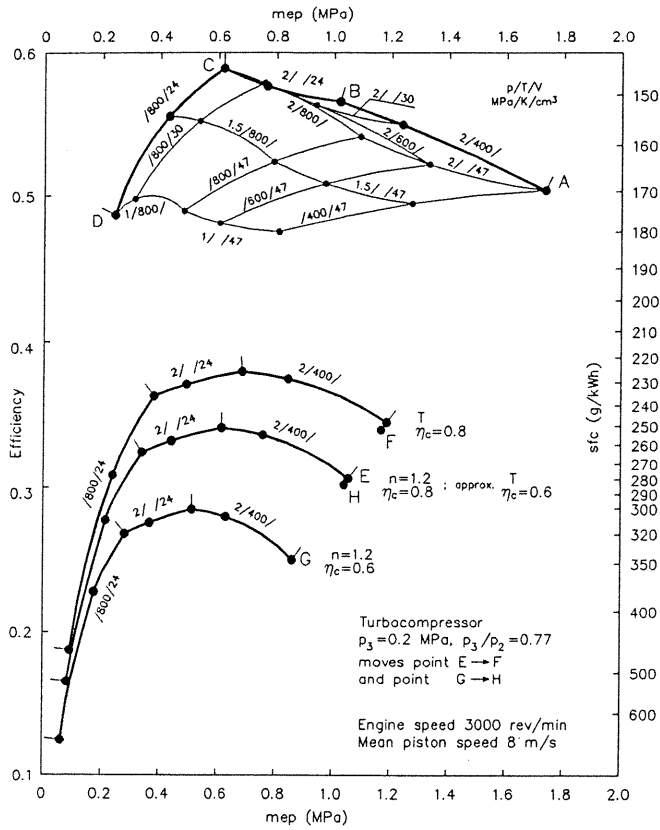


Fig.4 The field of engine mean effective pressure and efficiency.

time of ignition is moved towards much earlier beginning of combustion. Result: the imep and efficiency remain nearly unaffected. This speaks for small sensitiveness of the process as regards ignition timing.

Indicated Cycle In Intake-Compression System

The work necessary for medium transport through the engine is in the case of the proposed engine only partially involved in the indicated magnitudes of the process (in the cylinder-combustion chamber system (work of discharge stroke)). The rest of medium transport work and the compression work is consumed in the intake-com-

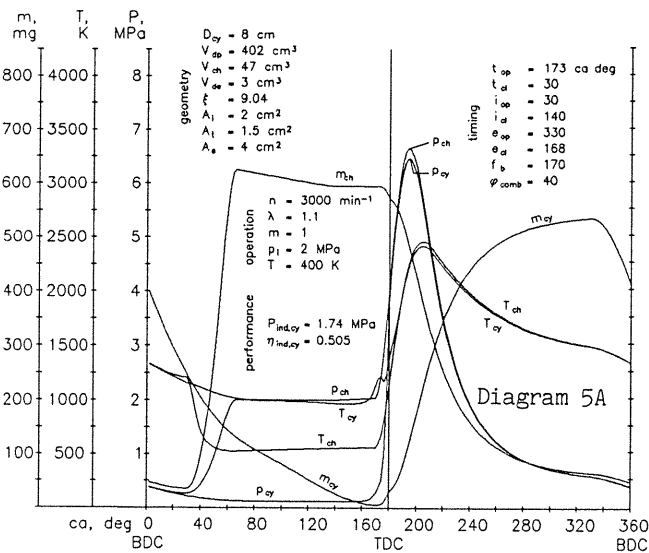


Diagram 5A

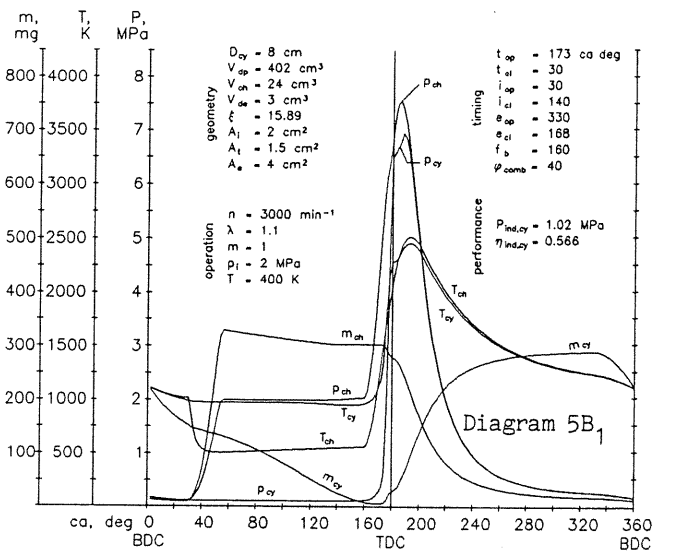
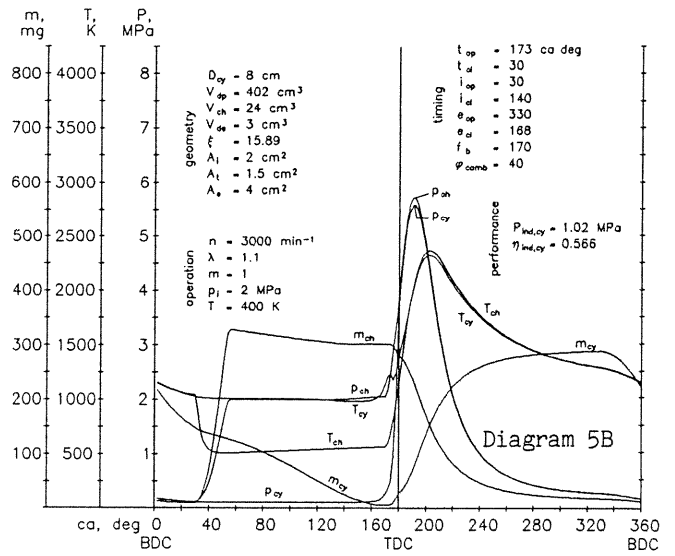


Fig.5 Histograms of pressure, temperature and mass in the combustion chamber and in the cylinder space versus crank angle. Designations:  $p_{cy}$  - pressure in cylinder,  $p_{ch}$  - pressure in chamber,  $T_{cy}$  - temperature in cylinder,  $T_{ch}$  - temperature in chamber,  $m_{cy}$  - mass of medium in cylinder,  $m_{ch}$  - mass of medium in chamber,  $D_{cy}$  - cylinder diameter,  $V_{sp}$  - displaced volume,  $V_{ch}$  - volume of chamber,  $V_{de}$  - dead volume of cylinder,  $\xi$  - expansion ratio,  $A_i, A_t, A_e$  - effective cross-section of intake, transflow-exhaust valve respectively,  $n$  - engine speed,  $\lambda$  - relative air fuel ration,  $m$  - characteristic of combustion (Vibe equa.)  $p_i$  - intake pressure,  $T_i$  - intake temperature,  $t_{op}$  - transflow valve open (crank angle degrees after BDC),  $t_{cl}$  - transflow valve close,  $i_{op}$  - intake valve open,  $i_{cl}$  - intake valve close,  $e_{op}$  - exhaust valve open,  $e_{cl}$  - exhaust valve close,  $f_b$  - combustion begins,  $\varphi_{comb}$  - duration of combustion,  $P_{ind,cy}$  - mean indicated pressure,  $\eta_{ind,cy}$  - indicated efficiency.

pression system and both carried away by the indicated work profited by expansion work in the engine cylinder. The determination of consumed work in the intake-compression system require the

knowledge of the working characteristic of the applied compressor(s). However, to get any approximate value about the indicated performances of the complete engine, the indicated work of compressor(s) should be computed like technical work for any chosen compression process (isothermal, isentropic, polytropic) increased for total losses of compressors of volumetric type, approximately expressed by compressor total efficiency ( $\eta_c$ ) besides polytropic exponent  $n$  as curves parameters in fig.4).

#### Indicated Cycle Of Engine

The consumed work in the intake-compression system must be subtracted from the indicated work in the cylinder-combustion chamber system. The rest of the work is to be considered as indicated work of the complete engine, which is comparable with the indicated work of conventional engines cycle.

#### Effective Engine Performances

To get effective (net) performances of the proposed concept of engine the mechanical losses must be subtracted from the indicated values. In the first approximation it could be accepted that the mechanical losses are dependent only on engine speed and independent of load [2]. In fig.4 three thickly drawn curves designated by /800/24, 2/ /24 and 2/400/ show the upper boundary—the upper envelope of performances field indicated in cylinder-combustion chamber system. For these curves the effective processes are computed and the curves of effective efficiencies versus mean effective pressure are plotted in the lower part of the diagram. Among the three curves the upper one belongs to the case when the compression process is isothermal at total compressor efficiency of 0.8, the middle one to polytropic compression with exponent of 1.2 at total efficiency of 0.8 and approximately also to isothermal

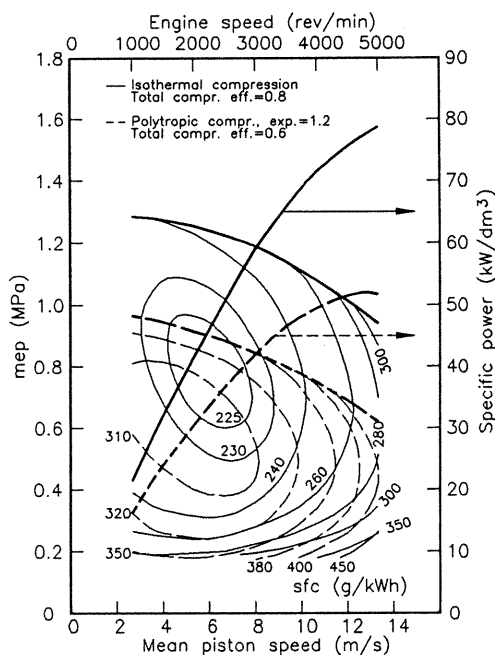


Fig.6 Engine performance map; bottom map for polytropic compression process with exponent 1.2 and total compressor efficiency 0.6, upper map for isothermal compression process and efficiency 0.8.

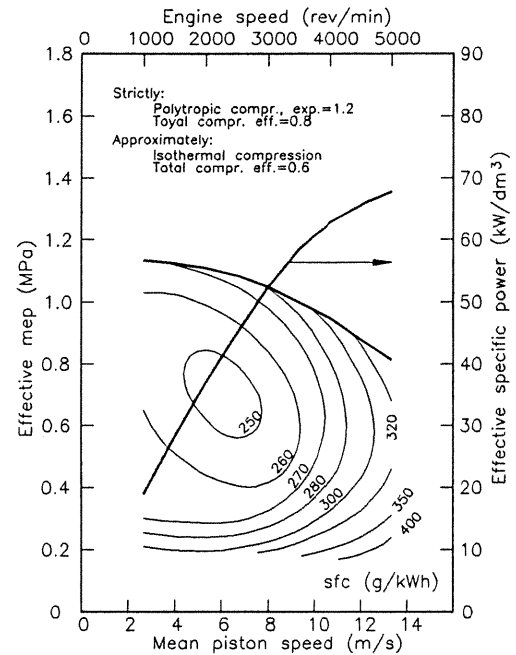


Fig.7 Engine performance map for isothermal compression process and total compressor efficiency 0.6, or approximately also for polytropic process with exponent 1.2 and efficiency 0.8 (expected possibilities of volumetric compressors technology).

compression process and total efficiency of 0.6, and the lower one to polytropic compression process with exponent 1.2 and total compressor efficiency of 0.6. This means the lower curve is easy to attain with today's compressors technology, the middle curve requires the maximum of today's compressors technology attainments, but the upper curve requires further development of volumetric compressors.

Turbocompressor as one step in the compressors assembly improves the effective parameters of engine at full load, at turbocompressor parameters given by  $p_3/p_4 = 2$  and  $p_3/p_2 = 0.77$  the full load point G moves to H and point E to F on the fig.4. This means the introduction of turbocompressor moves the lower curve towards the middle one and the middle curve towards the upper one, or in other words, with turbocompressor the middle curve is almost attainable with today's technology of volumetric compressors, and even the upper curve.

Similar points and curves of effective parameters are computed also for other engine speeds and the performance maps made are presented in fig.6 and 7. Two maps in fig.6 performed on the basis of the upper and lower curves (fig.4) show two boundary cases of possibilities, but the map in fig.7 shows the most expected possibility offered by today's technology of volumetric compressors.

The control of engine load according to the above envelope of the engine indicated performance field requires in the domain of higher load the control of combustion chamber volume, in the domain of middle load the control of temperature of medium entering in the combustion chamber, and in the domain of low load the control of its pressure. Applying the electronic processors such control is feasible today. However, while the

volumetric and pressure control are respondent, the temperature control is not. In engine operation transient conditions, so in the direction of torque increase as in the direction of torque decrease, a mixed control must be applied, first pressure or volume control and only later a stepwise temperature control should be attained to get the engine operation to optimal fuel consumption point. In transient conditions the operation of the engine would be outside the optimal domain of the indicated processes in the cylinder-combustion chamber system, i.e. below the envelope in the domain of worse economy.

#### CONCLUSIONS

1. Mechanical performances of the proposed concept of engine differ from the conventional four-stroke S.I. engines in the following:
  - increased effectivity of engine owing to its two-stroke action
  - highly elastic torque characteristic.
2. The economical characteristic of engine, which should be equivalent and at partial loads also superior to S.I. engines of today, requires a sophisticated intake-compression system composed of compressors assembly and heat exchangers. The location of minimum specific fuel consumption in engines performance maps is moved from upper engine speed and load domain in the case of conventional engines to lower engine speed and load domain in the case of the proposed engine.
3. The introduction of a turbocompressor as one step in compressors assembly is possible and it contributes to engine fuel economy, particularly at full load.
4. The separated combustion chamber makes possible:
  - to form the combustion chamber in an almost square cylinder shape at all its sizes;
  - relatively simple change of its volume without inflicting its compactness;
  - High turbulence of medium in the combustion phase caused by the high inlet velocity (also sonic) immediately before the beginning of combustion. The engine speed does not influence the medium turbulence level;
  - The compact shape of combustion chamber and turbulence which is independent of engine revolutions, increase the engine knock resistance also at lowest engine speeds;
  - Combustion chamber separated from the space in the cylinder by a valve probably reduces the harmful effect of knock upon the engine mechanism and its structure although the knock appears.
5. As regards the valves, its timing, and timing of ignition it is to conclude:
  - The design solution of the cylinder and cylinder head involving the combustion chamber permits the dimensions of valves with suitable flow cross-sections;
  - The timing and flow cross-section of the exhaust valve influence the process like in four-stroke engines;
  - The timing of the transflow valve partially accepts the function of ignition timing and therefore the timing of ignition does not influence the process so decisively as it does in conventional four-stroke engines.
6. This concept of engine offers a three-mode quantitative load control on the upper envelope of its performances field:
  - The control with variation of combustion chamber volume is applied in the domain of upper load, the control by temperature of the inlet medium (thermal control) in the domain of middle load, and the control by pressure of inlet medium in the domain of lower load;
  - The control inside the operation field, below the envelope, is applied in transient regimes owing to the endurance of thermal control mode.
7. The emission characteristic of the proposed concept of engine could be improved regarding HC-emission from secondary sources. Better emission from primary sources is not to expect. The treatment of poisonous substances outside the cylinder will still be necessary.
8. The separated combustion chamber which makes the engine more resistant to knock promises to solve the problem of fuel tolerance of engine regarding the octane number and regarding the fractions composition of fuel.
9. The feasibility of the proposed engine is critical in the next two points:
  - High heat loading of some cylinder components, particularly of the transflow valve;
  - The complicated compressors assembly composed even of more compressors of volumetric type or of a combination of a turbocompressor and a mechanical driven compressor intensively cooled or with coolers between some compression steps. The quality of compressors expressed by total efficiency should be high. Such compressor equipment for engine applications have not yet been devised and require further development in the future.

#### ACKNOWLEDGEMENTS

The support provided by the Research Council of Slovenia is gratefully acknowledged.

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APPENDIX

Model Of Cycle In Cylinder-Combustion Chamber System

The model of the engine cylinder-combustion chamber system is schematically presented in fig. A1. The process going on in the intake-compression system is not involved in this model, and the parameters of the medium before the inlet in the combustion chamber are chosen in advance. The model consists of the components marked by symbols which are explained in the figure caption. The model is actually composed of two open systems whose components are marked in the same figure by general indices alphabetically from a to e, which, once coupled to the chamber, secondly to the cylinder, get designations correspondent to the model indices.

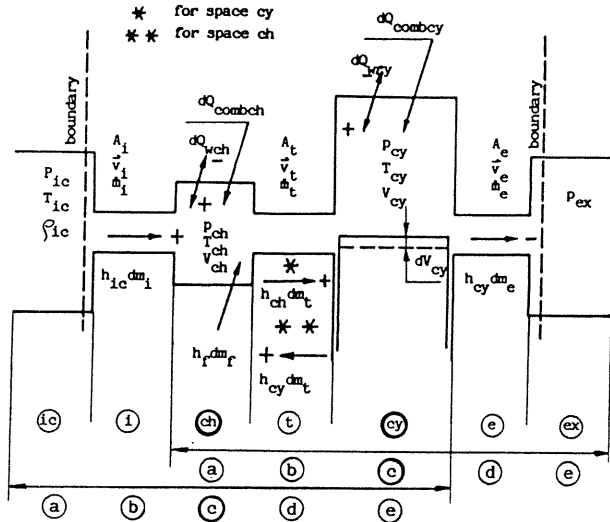


Fig.A1 Model of the engine cylinder-combustion chamber system. Indices: ic-intake compression system, i-intake valve cross-section, ch-combustion chamber, t-transflow valve cross-section, cy-cylinder, e-exhaust valve cross-section, ex-exhaust channel.

For each of the systems the laws of energy and mass conservation are applied. The mass-energy conservation law written in the difference form in accordance to general signs gets the form:

$$\Delta T_c = \frac{1}{c_v m_c} \left[ Q_{comb} \pm Q_w + c_p T_a \Delta m_b - c_v T_c \Delta m_b - m_c R T_c \Delta m_d - \frac{m_c R T_c}{V_c} \Delta V_c \right]$$

where: T-temperature,  $Q_{comb}$ -combustion heat,  $Q_w$ -wall heat,  $c_p$  and  $c_v$ -specific heat, m-mass, V-volume and R-gas constant. Both systems are connected together or separated, depending on the valve timing and likewise connected or separated to the intake-compression system or to exhaust channel. Regarding the mentioned connections and regarding the particular processes running in one or the other system, the different phases should

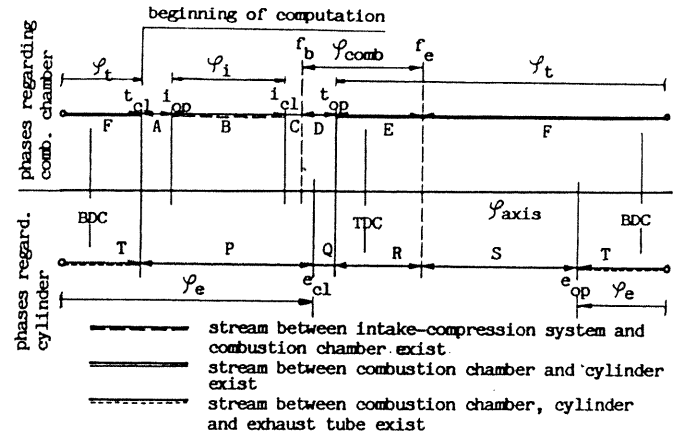


Fig.A2 Phases of working process in engine cylinder-combustion chamber system. The symbols are explained in caption of fig.5 in main body of article.

be defined. Some phases of one system will be independent of the other, some will be inter-related. Fig. A2 presents the different phases on the crank angle degrees axis during several different but characteristic events (opening or closing of valves, beginning and stop of combustion etc.). The symbols of some events are explained in the caption of fig. 5 in the main body of the paper. For these particular phases adequate expressions are derived from the equation above. Further, the geometric parameters of engine cylinder and crank mechanism are defined. The kinematic equations of the crank mechanism, Vibe formula /A1/ for heat release law and Woschni equation /A2/ for heat transfer coefficient are applied. The average temperature of surfaces in the cylinder and chamber for fixed surfaces and for the cylinder liner are computed in accordance with reference /A3/. The geometry of valves is given by the maximum effective cross-section and by a given discrete valve moving law. The mass flow through the valve cross-section is computed by the equations /A4/ for compressible medium flow.

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