MODELLING OF THE INLET SYSTEM AND EXAMINATION OF THE EFFECT OF INLET PIPE LENGTH ON ENGINE OPERATION PARAMETERS

Maciej Lisowski

West Pomeranian University of Technology
Szczecin, 70-310

tel.: 091 449 40 77, fax: 091 449 48 20
e-mail: mlisowski@ps.pl

Abstract

In the paper are presented assumptions for developing a computational model of engine inlet system. The aim of developing an inlet system model will be determination of the effect of different factors on the value of engine filling coefficient and, what is connected with that, on the parameters of its operation. Intention of the author is that a model developed will be of developmental character and would be capable of being expanded by further components enlarging the range of its applications. In the paper are also presented the results of test bed examinations for the SW-680 engine with the inlet system enabling modification of the length of inlet pipes. Findings are presented in the form of external characteristics where the effect of inlet pipe length on such parameters like power or torque, as well as on the rotational speed at which a maximum torque, is visible. The effect of inlet pipe length on torque rotational speed can be of significant importance in case of a possibility of shaping the engine characteristic and matching the area of engine operation with vehicle operating conditions, which is presented on universal characteristics in the form of resistance-to-motion lines. Test bed examinations have also served a purpose of verifying the model developed. The author is aware of the fact that the model presented requires further development. After considering certain corrections, it will be possible to develop a working computer programme being of use in simulation tests for the inlet systems of different engines.

Keywords: *inlet pipe, wave phenomena, modelling, power, torque.*

1. Introduction

As a basic assumption for designing the inlet system, its most favourable effect on maximum engine power as possible is usually adopted. At the same time, one must not forget about the value of maximum moment and that of rotational speed at which that value is being reached. The inlet system exerts a fundamental effect on the filling of engine cylinders and the speed at which a maximum filling occurs. Therefore, the inlet system shapes the course of torque line in the function of rotational speed, through which it also affects vehicle dynamic properties [3, 8-9].

During engine operation, wave phenomena occur in inlet pipes. These phenomena should be considered from the point of view of their use for improvement of cylinder filling. The filling of engine cylinders depends on the pressure present before the inlet valve, in particular in the final stage of its closing. An increase in the filling will be obtained when pressure is large enough at the time of inlet valve closing. Reaching the inlet valve by overpressure wave at an appropriate moment is being obtained by proper selection of the length of inlet pipe. By changing the length of inlet pipe, it is possible, depending on needs, to affect the rotational speed at which a maximum torque is being reached by engine. Making use of that property, short inlet pipes are being selected for racing car engine so that the maximum torque will occur near the rotational speed of maximum power, whereas it is required for utility cars that the maximum moment will be available just at small rotational speed, securing proper driving dynamics under moderate speed conditions being characteristic for utility vehicles. Engines of such vehicles should be equipped with the inlet system with relatively longer inlet pipes [3].

The length of inlet pipes can be selected basing on analyses and theoretical calculations or findings of experimental research. Enormous complexity of this problem and the necessity of using many simplifications causes that theoretical calculations only allow, with a lesser or greater approximation, determination of inlet system parameters. Final dimensions of inlet pipes are being only determined based on test bed examination results.

For utilisation of wave phenomena in the inlet pipe for the cylinder filling, their course should be learned in the first place. After opening the inlet valve, a piston moves from the upper dead position (UDP) to the lower one (LDP) making a suction stroke. Near the inlet valve, a negative pressure wave develops which moves at the speed of sound towards the open end of inlet pipe. There, a reflection takes place with a simultaneous change in the pressure sign and then, as an overpressure wave, returns to the valve. This is a so called first reflection. There can be many such reflections. Reflections fade after closing the valve since they do not change their sign when rebounding on a rigid obstacle (valve). Time when the pressure wave moves from the valve to the end of inlet pipe and back can be determined from the following dependence [3]:

$$t = \frac{2 \cdot L_d}{a} \quad [s], \tag{1}$$

where:
L_d - inlet pipe length [m],
a - speed of sound [m/s].

This dependence [relationship] can be also presented by means of the crank angle:

$$\tau = \frac{12 \cdot n \cdot L_d}{a} \quad [^\circ owk], \tag{2}$$

where: n - rotational speed [1/min].

2. Modelling of the filling process

During engine operation, the air flows through the following systems connected in series: environment, inlet pipe, cylinder, and environment. Parameters that describe environment are as follows: air temperature (T_0), air pressure (p_0) and air density in environment (ρ_0). Inlet pipe is described by air temperature in pipe $T_{(x,t)}$, air pressure in pipe $p_{(x,t)}$, air flow velocity in pipe $u_{(x,t)}$, air density $\rho_{(x,t)}$, length L_d and section surface A. Cylinder is described by combustion space surface $A_{c(t)}$, combustion space volume $V_{c(t)}$, pressure $p_{c(t)}$, temperature $T_{c(t)}$, air density in cylinder $\rho_{c(t)}$ and working medium mass in cylinder m_c. Furthermore, valve flow surface A_s as well as valve flow resistance coefficient μ_s and cylinder wall temperature T_{sc} should be taken into consideration [8].

Inlet system air parameters can be described by mass conservation, momentum and angular moment conservation and energy conservation equations, taking into consideration relations resulting from the equation of momentum and angular moment conservation, as well as by the equation of semi-ideal gas [8]:

$$\frac{\partial \rho}{\partial t} + u\frac{\partial \rho}{\partial x} + \rho\frac{\partial u}{\partial x} = 0, \tag{3}$$

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + \frac{1}{\rho}\frac{\partial p}{\partial x} = 0, \tag{4}$$

$$a_1\frac{\partial p}{\partial t} + p\left(\frac{a_2}{\partial t} + u\frac{\partial a_2}{\partial x} + a_2\frac{\partial u}{\partial x}\right) + a_1 u\frac{\partial p}{\partial x} = 0, \tag{5}$$

$$p = \rho \cdot R \cdot T, \tag{6}$$

where:

$a_1 = \frac{1}{\chi - 1}$, $a_2 = \frac{\chi}{\chi - 1}$,

χ - adiabatic exponent.

Processes that take place in cylinder can be described by mass balance and energy conservation equations as well as by the equation of cylinder air state [8]:

$$\frac{dm_c}{dt} = \frac{dm_s}{dt} + \frac{dm_w}{dt}, \tag{7}$$

$$h_c A_c \left(T_{sc} - T_c\right) + C_p T_{x=L} \frac{dm_s}{dt} + C_p T_c \frac{dm_w}{dt} = C_w T_c \frac{dm_c}{dt} + m_c c_v \frac{dT_c}{dt} + p_c \frac{dV_c}{dt}, \tag{8}$$

$$p_c V_c = m_c R_c T_c, \tag{9}$$

where:

x - distance of pipe -section from its entry,

m_s - mass flowing from pipe into cylinder,

m_w - mass flowing out of cylinder,

h_c - coefficient of heat exchange [transfer] with walls.

Solving of the aforesaid equations is very time-consuming and requires development of an appropriate computer programme. Intention of the author is to develop such a calculation programme that will allow simulation of phenomena taking place in engine inlet system.

3. Results of engine test bed examination

The object of test bed examinations was compression-ignition engine of the SW-680 type. This engine was used in the Jelcz motor trucks, buses and agriculture and construction machinery.

The engine had a dynamic supercharging system with single inlet pipe. When accomplishing the assumed research programme, a number of external characteristics was made for different inlet pipe lengths L_d (Fig. 1) [6], out of which the following lengths: 743, 843 and 1143 mm, were qualified for further examinations [7].

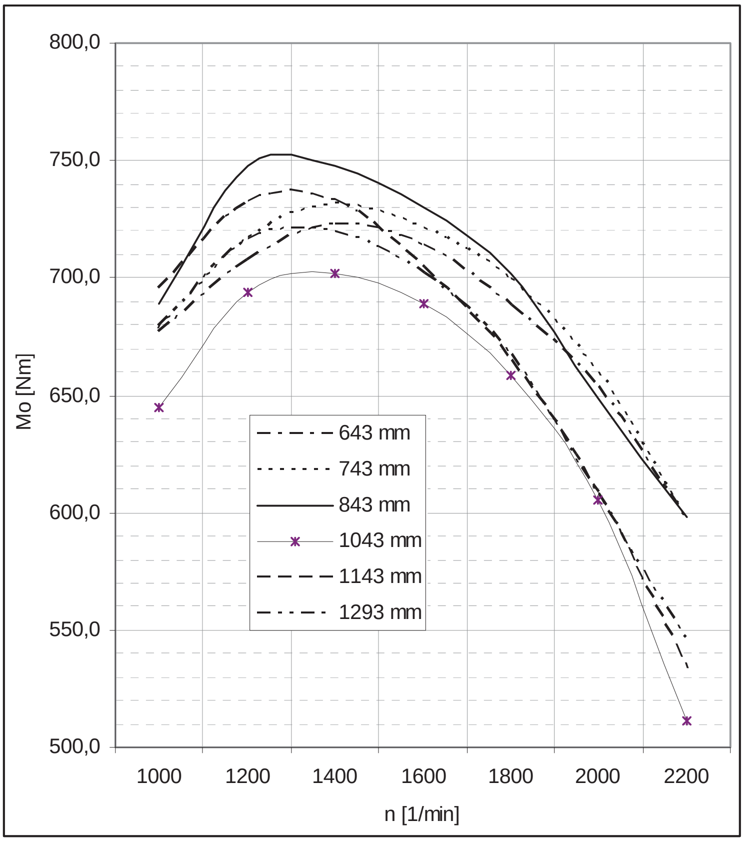

Fig. 1. Collective power characteristic for different inlet pipe lengths [6]

During further examinations, engine partial and full-load characteristics were made for the qualified lengths. Torque (M_o) and effective power (N_e) external and partial characteristics for partial loads (50 and 75 % of engine maximum load) are presented in Figs 2, 3, 4 and 5. The 50 and 75 % loads are most frequently met during normal operation of motor truck engine [1, 4-5].

It is seen in Fig. 2 and Fig. 3 that at 50% engine load the engine with 843 mm long inlet pipes reaches the maximum power (77 kW) and the maximum torque (400 Nm). The engine with 743 mm long inlet pipes proved to be slightly worse, by about 3%, but that with 1143 mm long inlet pipes turned out to be definitely worse in this respect, i.e. by about 20%.

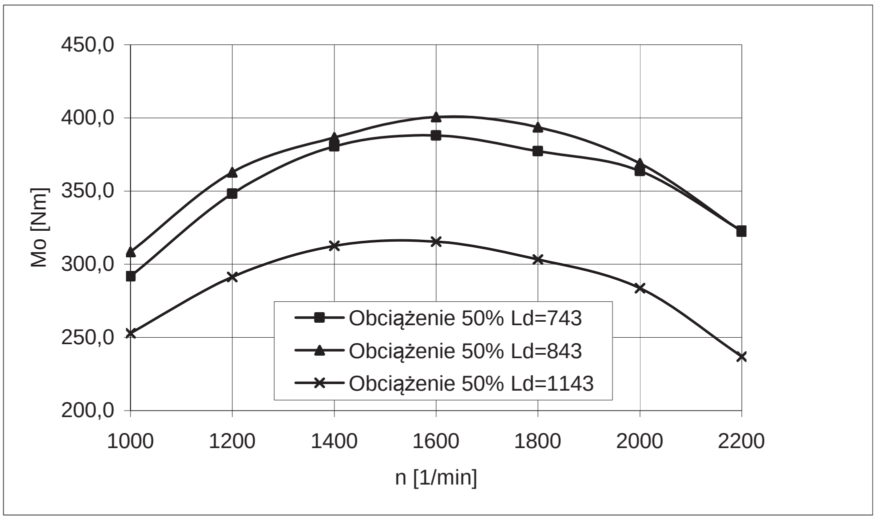

Fig. 2. Dependence of M_o on n (partial characteristic) for 50% load (L_D = 743, 843, 1143 mm) [7]

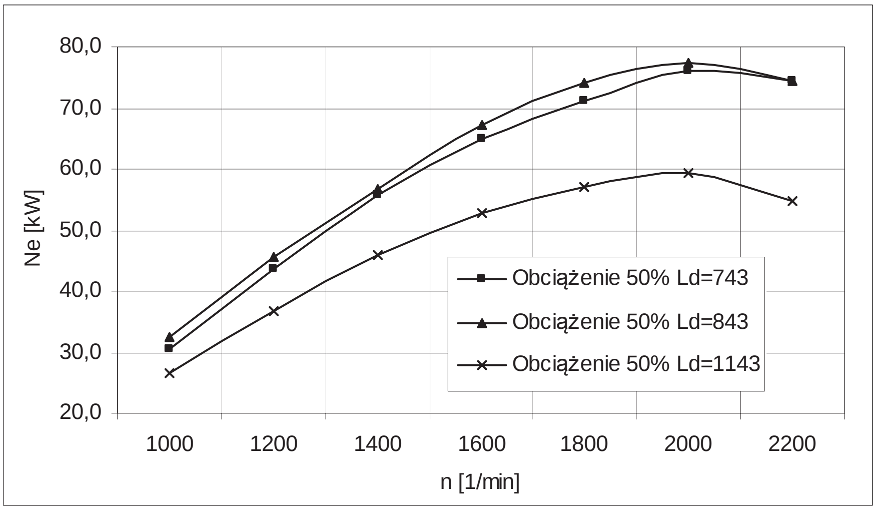

Fig. 3. Dependence of N_e on n (partial characteristic) for 50% load (L_D = 743, 843, 1143 mm) [7]

At the load amounting to 75% of engine maximum load (Figs 4 and 5), torque and power lines for the engine with 743 mm long inlet pipes were highest situated. The torque and power line for the engine with 843 mm long inlet pipes was situated about 3% below, within the range of medium rotational speeds. At the extremes of effective rotational speed range, a difference in power and moments was small. Differences between power and torque values for the engine with 843 mm long inlet pipes and that with 1143 mm long inlet pipes were however very clear, to the detriment of longer inlet pipes; these differences amounted to 10%, being within the range of speeds close to the rated ones. For low rotational speeds, these differences were very small.

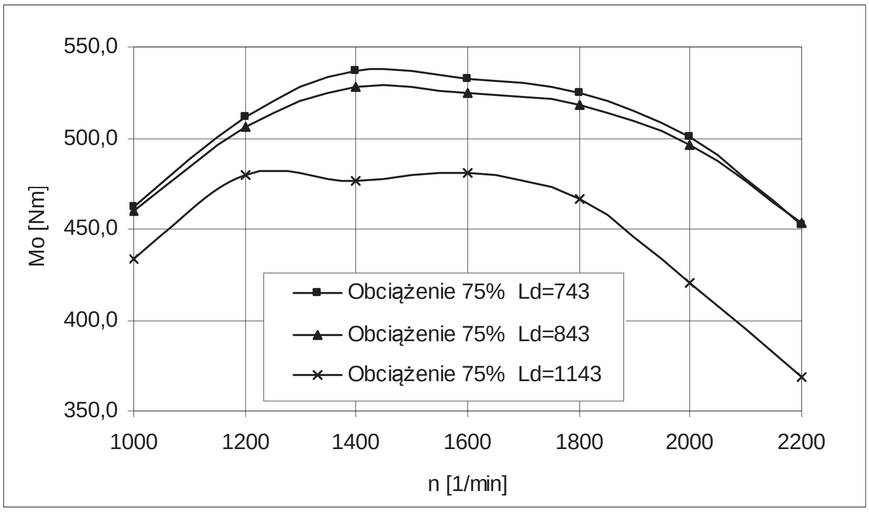

Fig. 4. Dependence of M_o on n (partial characteristic) for 75% load (L_D = 743, 843, 1143 mm) [7]

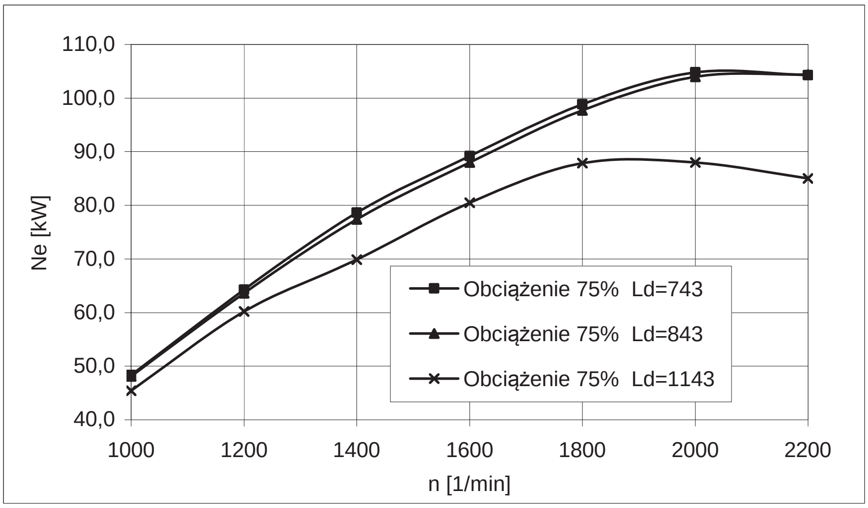

Fig. 5. Dependence of N_e on n (partial characteristic) for 75% load (L_D = 743, 843, 1143 mm) [7]

When analysing and comparing engine universal characteristics (Figs 6, 7, 8), it is possible to state that the engine with 843 mm long inlet pipes proved to be the best in respect of work efficiency (Fig. 6.). The field of relatively low specific fuel consumption, limited by isoline 220 g/kWh, takes up a considerable part of the whole area of characteristic. The fact that this field is within the range of low and medium rotational speeds, i.e. from 1000 1/min to about 1600 1/min, and medium loads, from about 400 to 650 Nm, is worthy of special attention.

Engines with 743 and 1143 mm long inlet pipes are characterised by slightly worse isoline 220 g/kWh distributions for specific fuel consumption. In both cases, these fields cover a considerably smaller area of the universal characteristic than for the engine with 843 mm long inlet pipes. Therefore, probability that engine operating point during normal operation will be situated precisely in that field is very small.

During motor-car motion, the road surface as well as other elements of environment affects a vehicle with certain resisting forces. Obviously, the forces acting upon a vehicle affect at the same time its power transmission system. Considering the values of transmission ratio and wheel dynamic radius, it is possible to determine engine load torque. Resistance torque lines were determined for the Jelcz 315 motor truck moving on the fifth and sixth gear [2].

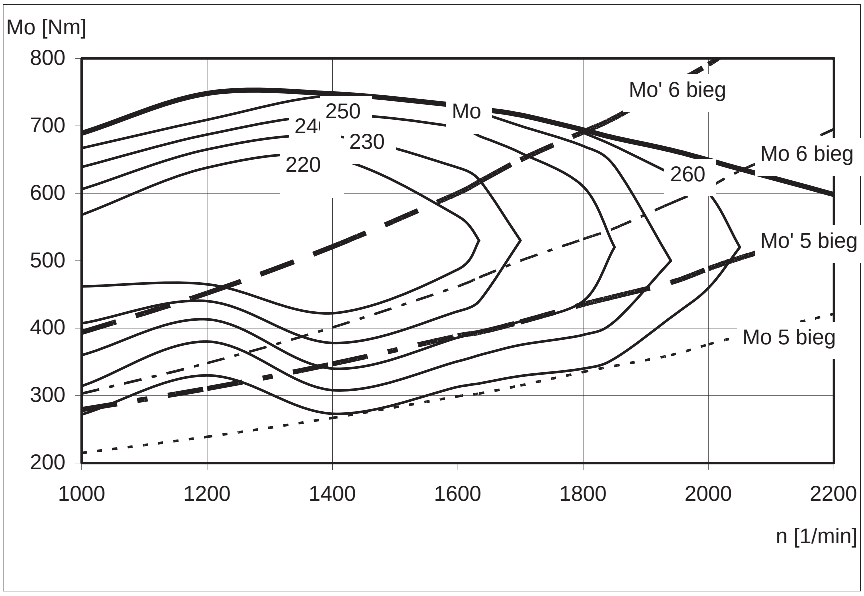

Fig. 6. Universal characteristic for inlet pipe length L_D = 843 mm

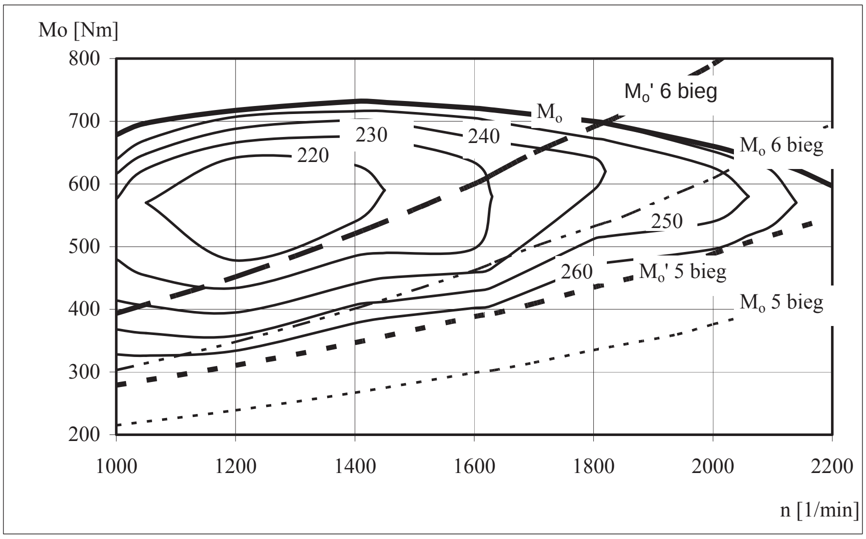

Fig. 7. Universal characteristic for inlet pipe length L_D = 743 mm

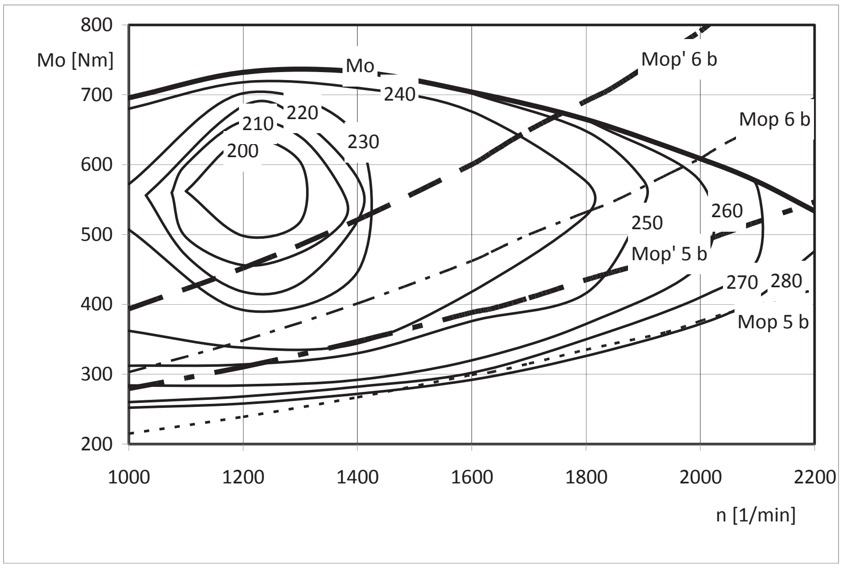

Fig. 8. Universal characteristic for inlet pipe length L_D = 1143 mm

As one can see, engine operating points will never be situated in the field limited by isoline 220 g/kWh for factory-made transmissions (lines M_o 5 b and M_o 6 b). In order to bring engine operating points into the area with low specific fuel consumption, the author reduced the value of axle ratio from 1.96 to 1.5 (lines M_{op}' 5 b and M_{op}' 6 b). After this modification, the line of resistance-to-motion torque for the sixth gear (M_{op}' 6 b) for 843 mm long inlet pipes is found, to a considerable extent, within the area limited by isoline 220 g/kWh. For the length of 743 mm, this line is not within the interesting area, while for that of 1143 mm only a small part of it is found in this area.

5. Conclusions

The inlet system of piston combustion engine is characterised by operation of the pulsatory character. Therefore, wave phenomena affecting the filling of cylinders have to develop in result of the cyclic process of charge induction into respective cylinders. Depending on inlet system geometry parameters, the cylinder filling can undergo improvement or deterioration. The problem of such a selection of the geometry of inlet system, in particular of the length of its inlet pipes, so to receive an improvement in the cylinder filling within the range of appropriate engine rotational speed range becomes significant. The length of inlet pipes can be determined based on test bed examinations but it is a time-consuming process that requires making a number of tests.

This process can be substantially reduced by replacement of the real system with an appropriately developed physical and mathematical model consisting in description of the inlet process by a number of equations. Depending on the representation degree of real processes, the equations describing them are more or less complicated. The statement that better representation will be accompanied by larger complexity of equations and larger labour consumption of their solving becomes obvious. However, the progress which has taken place in the development of information technology and the computing power of modern computers allows more and more common use of models that map processes occurring in technical equipment.

The model assumptions presented in this paper are solely an introduction to making a practical model that will allow a systemic approach to phenomena occurring in the inlet system and their effect of engine operation parameters.

The preliminary test bed examinations performed allowed determination of the effect of inlet pipe length on engine operation parameters. They will constitute a reference for results being obtained during modelling.

The most favourable, from the point of view of operation efficiency, is such a characteristic where fields limited by the isoline of specific fuel consumption as low as possible have the largest possible surface and are within the area of characteristic, in which engine operating points are found most frequently. It is an obvious fact that location of engine operating points, at which engine is being found most frequently and for the longest time, depends of its application. It is seen, based on the universal characteristics presented, that distribution of specific fuel consumption isolines for different inlet pipe lengths differs. Therefore, dynamic supercharging is one of the methods of engine characteristic shaping and its adaptation of the method of motor-car use.

In normal operation of motor truck, the working time of engine loaded in 50-75% is the longest. Due to the fact that engine characteristic can be controlled to a rather limited extent, there is a necessity of adapting engine working points to the existing universal characteristic. This can be done owing to application of an appropriate transmission in the power transmission system.

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