Scientific Papers. Series A. Agronomy, Vol. LXVIII, No. 2, 2025
ISSN 2285-5785; ISSN CD-ROM 2285-5793; ISSN Online 2285-5807; ISSN-L 2285-5785

DYNAMICS OF THE MOTOR MECHANISM OF INTERNAL COMBUSTION ENGINES

Anişoara DUMA COPCEA, Casiana MIHUȚ, Lucian Dumitru NIȚĂ, Nicoleta MATEOC, Teodor MATEOC, Ramona ȘTEF, Karel Iaroslav LAȚO, Carmen Claudia DURĂU, Daniel POPA, Dragoslav Vlad MIRCOV

University of Life Sciences "King Mihai I" from Timişoara,
119 Calea Aradului Street, Timişoara, Romania

Corresponding author email: vlad_mircov@usvt.ro

Abstract

Engines that equip self-propelled tractors and combines are diesel engines. These engines have a construction adapted to working over a wide range of speeds and powers. This ensures the operation of all agricultural machines. However, the areas of economic use are more restricted, which is why agricultural aggregates should be properly made up and exploited to fit into them. The tractor engine is used at maximum speed regime when it has maximum power requests or when operating machines is also necessary from the independent power source. The studies in this work were carried out on a D-110 Diesel engine, an engine that equips the U-650M tractor. Studies refer to the dynamics of the motor mechanism. The parts of the engine mechanism studied were: the cylinder, the piston, the piston bolt, the connecting rod, the crankshaft, and the steering wheel. In each of these parts, the size and weight were measured. For the calculation of the operating indices of the D-110 engine, measurements were made on a stand to try and roll internal combustion engines.

***Key words**: diesel engine, agricultural machines, tractor, cylinder, bolt, piston.*

INTRODUCTION

The cycle of an internal combustion engine can be divided into five phases:
-Intake or filling of the cylinder with motor fluid;
-Compression of the motor fluid intake in the engine cylinder;
-Ignition and burning of fuel;
-Gas relaxation;
-Evacuation of burnt gases from the engine cylinder (Copcea Anişoara Claudia Duma, et al, 2023; Anișoara Duma Copcea et al., 2024; Mihuț et al., 2024).

It follows from the above that the engine with compression fluid has a chemical energy that is partially or completely hot during its evolution.

In engines with compression ignition, mechanical energy is obtained by the engine fluid action on the pistons that have an alternative translation movement (Mateoc-Sîrb, et al., 2024). The translation movement of the pistons is transformed into a rotation movement of the engine shaft through connecting rod-crank mechanisms. In the four-stroke cycle, the succession of the processes is performed in four piston races, i.e., in two rotations of the crankshaft. (Relly et al., 2017). The periodic resumption of the motor cycle, which ensures the continuous operation of the engine, requires the emptying of the motor fluid cylinders that transmitted the mechanical work of the pistons (relaxed burn products), followed by the filling of the cylinders with a fresh motor fluid (Gunston, 1999). The diesel cycle has the advantage of high thermal efficiency due to the possibility of using high values of the compression ratio (Ɛ = 12-24) (Voleac, 2011).

The usual representation of the motor cycle is a diagram illustrating the variations of the gas pressure in the cylinder during the cycle. When gas pressure (p) is represented according to the volume (V) they occupy in the cylinder, the diagram is called the p-V diagram. The p-V diagram of the four-stroke actual engine cycle includes its phases (Mihuț et al., 2022; Mihut et al., 2023; Boca et al., 2019). The value and evolution of the gas pressure in the cylinder depends on the position of the piston, the moments of opening the intake holes (i.h.) and evacuation (e.h.), on their closing moments (b.e., b.a), of the injection (i), as well as of the nature of the process in which the cylinder gases are used: aspiration, compression, burning and relaxation, evacuation (Petrescu et al., 2009; Petrescu, 2011).

The main processes that take place in combustion engines in four stages, depending on the p-V diagram of the engine cycle are presented in Figure 1.

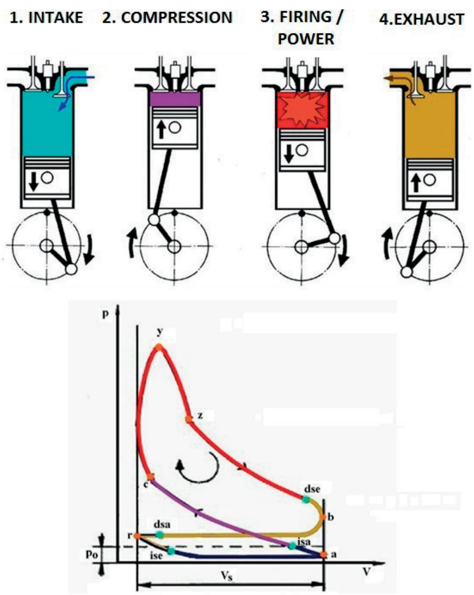

Figure 1. The main processes that take place in combustion engines in four stages, depending on the p-V diagram of the engine cycle

Time 1 – Air intake. After opening the intake hole with a certain advance compared to the IDP (inner dead point), the filling of the cylinder is possible under the action of the depression created by moving the piston to the ODP (outer dead point). The average speed of air entry through the intake hole is 70-90 m/s. During the movement of the piston to the ODP, in the so-called main phase of the intake, most of the air is aspired. The intake hole closes late compared to ODP for a complete air intake in the cylinder (Daicu Anatolie et al., 2015).

Time 2 – Compression. The piston moves from ODP to IDP. The intake and evacuation holes are closed. The air in the cylinder is compressed. The pressure and air temperature increase. The volume is shrinking. With a certain advance before the IDP, the fuel is injected into the cylinder. The very fine fuel drops sprayed in the cylinder by the injector are lit in contact with the compressed and heated air. The burning takes several thousands of seconds. The process of burning is carried out at constant volume and pressure.

Time 3 – Relaxation. The gases resulting from the burning of the fuel presses strongly on the piston forcing it to move from IDP to ODP. Relaxation is active time or engine time. The beginning of the relaxation is considered from the moment of reaching the maximum gas pressure; the end, in the opening of the exhaust hole (e.h.) (Levente et al., 2015).

Time 4 – Evacuation. Removal of the burnt gases from the engine cylinder begins with the opening of the exhaust hole and continues until its closing. The piston moves from ODP to IDP. The exhaust hole closes late compared to the IDP for a complete emptying of the burnt gases.

During a motor cycle, the crankshaft performs two rotations (720^0 RAC) and the camshaft, one rotation (Novorojdin, 2011).

For a complete intake and evacuation, it is necessary for the intake and evacuation holes to open in advance and close late from the dead points (Agape et al., 2018).

Each four-stroke internal combustion engine has a certain diagram with distribution phases, with certain values of advance and delay angles (Hăbăşescu et al., 2005).

MATERIALS AND METHODS

The motor mechanism achieves the transformation of thermal energy into mechanical energy, having the role of transforming the translation movement of the cylinder into a rotational movement of the crankshaft.

The piston of an internal combustion engine plays a special role due to the complex requests to which it is subjected. The power of the engine is limited by the resistance of the piston to thermal and mechanical demands.

The piston is a mechanical organ, in alternative translation motion, which, together with the parts that accompany it (segments and bolt), perform the following functions:

Achieving the volume variation inside the cylinder;

Ensuring the evolution of the motor fluid in the cylinder (gas intake and evacuation);

Guiding the rod movement by transmitting the gas pressure forces at the same time;
Contributing to the evacuation of the heat resulting during burning;
Ensuring the sealing of the cylinder, thus preventing gas leaks and excess oil penetration.
The pistons are made of metal materials: aluminium, steel, and, in some cases, cast iron. Because these materials have a thermal coefficient of expansion, the piston dimensions are not fixed but variable, depending on the temperature.
In order for the piston to move into the cylinder, between the piston and the cylinder there should be a clearance. This clearance is higher when the engine is cold and drops as the temperature increases. The thermal expansion of the piston is not uniform: it is higher in the area of the piston head due to the larger amount of material. The distribution of temperature in the body of the piston is not uniform, being larger in the head area and smaller in the lower part, in the area of the mantle. For this reason, the shape of the piston should be slightly conical, so that the expansion becomes cylindrical.
Due to the force of piston pressing on the bolts and thermal dilation, the piston mantle is deformed, taking the form of an ellipse, with the large axis arranged after the axis of the bolt. If the large axis of the ellipse is greater than the bore of the cylinder, the danger of fluctuating/locking the piston in the cylinder occurs. For this reason, the piston mantle is not continuous: it is cut in the areas below the axis of the bolt.
The diameter clearance of the piston induces a tilting effect around the bolt during the engine operation. This occurs with a shock: the cylinder starts vibrating and there are characteristic noises called "piston beat". In addition to the unpleasant acoustic effect, the piston beat intensifies the wear effect of the piston and segments. To diminish this tilting effect without changing the diameter clearance, the axis of the piston is shifted in relation to the cylinder axis.
A diesel engine is noted by the fact that the gas pressure in the cylinder has much higher values than in the case of a petrol engine. In petrol engines, the maximum pressure reaches around 60-90 bars while, in diesel engines, it reaches up to 130-160 bar. This requires diesel engines to use mechanical parts that have a much higher resistance.
It should be noted that a diesel engine piston is more robust compared to the petrol engine piston, has a taller mantle, and contains the combustion chamber in the piston head. The disadvantage is that higher robustness means greater mass. This is one of the reasons why the maximum speed of a diesel engine is lower, generally with 2,000-2,500 rpm than that of a petrol engine. All the moving parts of a diesel engine have a larger mass and, by default, larger inert. The maximum speeds at the level of a petrol engine are not possible in a diesel engine because it induces very large shocks and vibrations that can lead to partial or total engine damage.
Part of the heat resulting from the burning is evacuated through the piston. Most of the heat is evacuated through the segment-port region (70%); at the level of the mantle, around 25% are evacuated; and the rest is transmitted to the bolt, the crankcase gases, and the oil. The maximum temperature level depends on the operating regime of the engine.
Temperature can rise in two ways:
- By increasing the load, due to the greater amount of fuel inserted into the cylinder;
- By increasing speed, due to the greater number of cycles performed per time unit.
The connecting rod is the connection piece between the piston bolt and the crankshaft. It transmits the gas pressure force (during gas relaxation) from the piston to the crankshaft. The connecting rod includes the small head of the connecting rod, provided with a brass bushing in which the piston bolt, the connecting rod, the large head of the connecting rod and the connecting rod bearing, provided with semi-bearings, connecting the connecting rod of the crankshaft.
The connecting rod should withstand the intense mechanical requests produced by the gas pressure force and the inertia of the moving parts. The piston is fixed to the connecting rod through a bolt, which is fixed in the connecting rod and can rotate in the piston places. There is no direct contact between the foot of the connecting rod and the bolt: between them, there is a softer metal (bronze) that has the role of reducing friction.

Two screws are used to attach the connecting rod lid. Newer rods do not provide for bolts: they are fixed in the lid. On the lid and on the connecting rod, there are pins and holes that allow the lids to be installed only in one position. In an engine, the rod lids are not interchangeable: a lid will always be mounted at the same connecting rod.
Connecting rods are made of high resistance allied steels; in some cases, titanium-based connecting rods are used.
The piston bolt makes the articulated bond between the piston and the connecting rod. It has a cylindrical shape with an inner hole along the entire length. The mechanical resistance of the bolt should be high because it is subject to compression, shear, and bending requests. Because of the heavy operating conditions and of the need for wear resistance, the bolt is made of alloy steel.
To have an increased resistance to wear, the bolt is subjected to thermal hardening treatments (through high frequency currents). In thermal engines for cars, the bolt is floating, and there is clearance between the bolt, the shoulders of the piston, and the foot of the connecting rod. Due to this clearance, during the operation of the engine, an oil film is created between the moving parts that amortizes the shocks and reduces the friction.
To limit the axial displacement of the bolt in the piston places, the bolt is fixed with the help of metal rings located in special piston cans. Limiting the clearance is necessary because excessive clearance can lead to mechanical stress that could result in piston deformation.
The bearings are mounted in the head of the connecting rod, between the connecting rod and the crankshaft. On each connecting rod, there are two bearings, one in the lid and the other on the connecting rod. A bearing consists of a thin metal layer covered by an antifriction layer that comes in contact with the crankshaft bearing.
Between the bearing and the crankshaft, clearance is provided to allow the formation of a hydrodynamic oil layer to reduce the friction. To observe this clearance, the tightening of the connecting rod covers will always be done in the couple specified by the engine manufacturer.
Each bearing has a fixing spur that has the role of correctly positioning the bearing and, at the same time, of ensuring the mounting of the bearing only on the lid or only on the connecting rod. Like the rod lids, the bearings are not interchangeable: they are mounted all the time on the same piece.
The crankshaft is the most requested piece of the engine because, through the piston and the rod, it takes over the forces due to the pressure in the cylinder. The crankshaft is the piece that takes over the forces in the connecting rod, adds the mechanical things produced in the cylinders, and transmits the resulting energy to the wheels through the transmission. Also, the crankshaft involves some motor auxiliary systems (distribution, oil pump, water pump, compressor, alternator, etc.). The crankshaft is mounted in the engine block through the level bearings.
The items that make up a crankshaft for the engine are:
Level bearings (making the crankshaft rest on the motor block, in its places);
Handlers (on which the connectors are caught);
Arms (achieving the connection between the levels bearings and handlers;
Heads (coving areas).
Inside, the crankshaft has channels for the circulation of the lubrication oil that correspond to the feeding holes of the levels and handlers; most crankshafts have only one channel along them.
The crankshaft has a number of level bearings equal to the number of cylinders plus one. The level bearings are placed on the same geometric axis, and their width is different. The number of handlers is equal to that of the cylinders. The handler together with the two arms of the handler forms the crank (elbows). The diameter of the handler is smaller than that of the level bearing. The clearing of the bearings is done according to their number, thus ensuring a uniform engine operation and a balancing of the crankshaft.
The crankshaft is balanced with the help of the counterweights placed in the extension of the crankshaft arms (opposite them) and of the correct collapse of the crank. The balancing checking is done on special machines, and the share of the crankshaft, by partial releases (drilling or milling of counterweights).
Because it works under special disadvantageous conditions, the crankshaft has a complicated construction influenced also by the type of engine it equips. In general, the elements of the

crankshaft are oversized to achieve the rigidity necessary to limit the deformations.
Of all the engine organs, the crankshaft is subjected to the highest requests for the gas pressure force and the inertia forces of the masses with translation and rotation movement. Under the action of these forces in the elements of the crankshaft, important requests for stretching, compression, bending, and twisting occur, requests that have a shock character due to the mounting clearances, the high speed of pressure during the burning period, and the change of the forces application. The forces being variable produces dangerous demands of fatigue especially in the areas of voltage concentrators, such as the crossings from the arms and holes for lubrication oil. Apart from these, the crankshaft is stressed by additional stresses produced by the bending and twisting vibrations. The twisting vibrations of the crankshaft also produce disturbances in the functioning of the distribution mechanism, in the ignition distributor; in vehicle engines, they propagate in the mechanical transmission. All these requests cause the crankshaft deformation, which results in premature wear or even in breaking the crankshaft.
Given the difficult operating conditions, the construction and material of the crankshaft should meet the following important requirements: high mechanical stiffness and resistance, high wear resistance of the bearing surfaces, high fatigue resistance, manufacturing accuracy and dimensions, static and dynamic balancing, and avoiding resonance for both twisting and bending vibrations.
The construction and dimensions of the crankshaft depend on the type of engine, the number and disposition of the cylinders, the order of work, the balancing of the engine, etc.
Crankshafts are executed in two main variants: non-removable and removable, the first solution being the most used.
The front end of the crankshaft is constructively realized, taking into account that it serves mainly to operate the distribution, fan, cooling pump (except for air-cooled engines), and current generator (alternator or dynamo). It is also envisaged to place the oil sealing element, for example sealing ring and, in some engines, the location of the twisting shock absorber; the amplitudes of the twisting vibrations reach maximum values at the front end of the crankshaft, which justify the assembly of the shock absorber, if necessary, at that end.
Crankshafts are currently built of steel. The wide dimensioning demanded by the assurance of rigidity sometimes allows the use of quality carbon steel, being encountered in the manufacture of the connecting rod. For the most requested crankshafts, generally MAC, allied steels with Cr, Ni, Mo and possibly V, with breaking resistance up to 1,450 N/mm² are required.
The semi-finished product is elaborated by hot deformation – free moulding or forging. The first process is applied to small- and medium-sized crankshafts, whose final mass does not exceed 250 kg, using successively closed moulds (more heating). It presents the advantage of ensuring the continuity of the fibres of the material, as the semi-finished product has the shape of the crankshaft. This form is not made through free forging (large crankshafts), so that the elbows are obtained by removing the material between the arms by cutting, which causes the fibre to be interrupted and increases the cost of manufacture; newer technologies are based on the individual forging of each elbow, using a device that allows the eccentric movement of the respective handler bearing and then the rejection of the arms; continuous fibre is, thus, respected, and the homogeneity of the material and the resistance to fatigue are improved.
After elaboration, the semi-finished product is subjected to a normalization treatment to ease the processing. Large crankshafts forged with continuous fibre usually require axial recovery, and normalization is performed in a vertical position. Before finishing, the qualities of the crankshaft are improved by a chemical or thermochemical treatment. To increase wear resistance, the bearings are superficial. On the large scale, for this purpose, it applies to chill by induction or flame, on a depth of 3-5 mm, achieving a minimum hardness of 50 HRC.
Of great importance is the structure obtained by the treatment of return, which follows the hardening. For the hardening of the bearings, the cementation or fuelling is sometimes practiced, cheaper than hardening. In Cr - Mo or Cr - Mo - V steel crankshafts, bearing nitriding is very efficient, which increases not only hardness, but

also fatigue resistance; the treatment is more expensive.

Special results were obtained by constructing the crankshafts of some traction engines made of special micro-alloy steels. In the case of steel 49 Mn VS 3, the semi-finished product is uniformly cooled with air, without the need for normalization or correction of the geometric shape, and has a break resistance of 800-900 N/mm². From this material, for example, the crankshaft of a car engine with six cylinders in line is built; the crankshaft has non-removable counterweights at all arms and a mass of 35.5 kg. Optimizing its geometric form, the processing additions were reduced by lowering the semi-finished products by 10% and 35% of the volume of cutting operations.

Newer procedures for manufacturing the crankshaft are performed by casting.

The general characteristics of the crankshaft have some elements of superiority compared to those of the crankshafts performed from hot deformed semi-finished products: the mechanical characteristics can be located above the level of the forged piece, the resistance to creep is slightly higher, and hot behaviour is better. The technological difficulties, the less homogeneous structure, the more pronounced susceptibility of the dispersion of the physico-mechanical properties from one casting to another and the lower behaviour to fatigue have delayed the casting of crankshafts. Such obstacles were removed due to the progress made in the field of casting. As a result, the casting of crankshafts introduces important advantages, in relation to their forging. Thus, the consumption of metal is reduced by 30-70%, as the casting offers high accuracy of execution, greatly diminishing the processing additions (by 40-80%) because the amount of chips decreases 2.5-3 times and the number of operations with 20-25%. The mass of the finished crankshaft is 10-15% smaller, because the holes can be easily achieved along the bearings, which, in addition, ensures the uniform solidification and the increase of the rigidity.

The steering wheel is mounted on the engine crankshaft and has the role of accumulating a moment of inertia in the useful phase of the motor cycle (relaxation) that it gives in the energy consuming phases (evacuation, intake, and compression), thus realizing the uniform movement of rotation of the engine crankshaft. It is a massive piece of steel or cast iron on which the gear (the steering wheel) is mounted for training the crankshaft with the electromotor when starting the engine.

RESULTS AND DISCUSSIONS

The study of the dynamics of the motor mechanism aims to determine the forces and moments acting on the parts of the mechanism. This study is important for carrying out resistance calculations.

The forces that act in the motor mechanism are: inertia forces, pressure forces, gravitational forces, and reactions.

For the measuring of forces and moments, the dynamic model in Figure 2 is used.

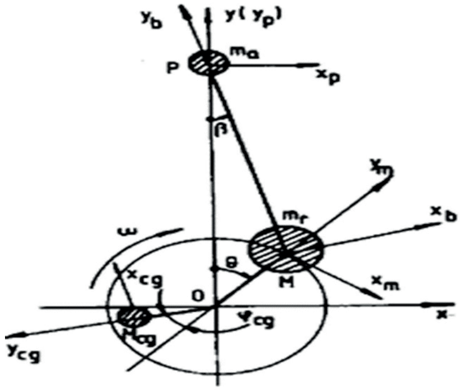

Figure 2. The dynamic model of the motor mechanism

m_a – masses in motion of translation; m_r – masses in rotational motion; m_{cg} – mass of counterweights

The values of the concentrated masses (of the subassemblies of the D-110 engine) are: $m_a = m_p + m_{ba}$,

where:

m_p – piston mass ($m_p = 0.86$ kg);

m_{ba} – connecting rod mass relative to piston in translational motion ($m_{ba} = 0.34$ kg)

Inertial forces are: $F_a = -m_a \cdot a_p$.

The pressure force of the gases is determined with the relationship: $F_p = \frac{\pi \cdot D^2}{4} \cdot p$

The force applied by the piston in the joint is: $F = F_a + F_p$.

The normal reaction is: $N = F \cdot tg\beta$.

The forces that act in the motor mechanism for a complete rotation of the crankshaft were calculated from 20 to 20 degrees. The results are centralized in Table 1.
Based on the data in Table 1, the forces diagram in Figure 3 was drawn.

Table 1. The forces acting on the motor mechanism

α (ºRAC)	F_p [daN]	F_a [daN]	F [daN]	N [daN]
0	295	- 79	216	0
20	389	- 72	317	27
40	192	- 51	141	23
60	96	-24	72	16
80	57	39	96	15
100	39	26	65	17
120	30	40	70	15
140	25	46	71	12
160	18	48	66	6
180	51	48	99	0
200	53	47	100	- 4
220	61	46	107	- 8
240	78	40	118	- 10
260	190	26	216	- 9
280	173	38	211	- 6
300	54	- 24	3	- 2
320	98	- 51	47	- 3
340	170	- 72	98	- 1
360	295	- 79	216	0

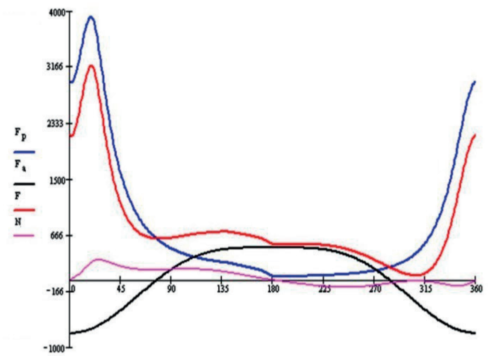

Figure 3. Diagram of forces acting on the motor mechanism

Analysing the diagram in Figure 3, it is found that the inertia forces have maximum values at the end of the race, i.e., when the piston is found in the dead points. This is due to the fact that the direction of travel of the piston changes.

CHECKING THE PISTON BOLT

The piston bolt is requested for shear in two ring cross sections. The sizes of the piston bolt are:
-Length: 88 mm;
-Outer diameter: 40 mm;
-Inner diameter: 23 mm.

Knowing the shape and dimensions of the transversal section, the permissible resistance of the material from which the bolt is made and the maximum size of the cutting force, the shear check is done with the relationship:

$$\tau_{ef} = \frac{T_{max}}{A_{ef}} \leq \tau_a$$

where:

τ_{ef} – the actual tangential voltage occurring in the cross-section;

T_{max} – maximum cutting force ($T_{max} = 400\ daN$);

A_{ef} – actual area required for shearing ($A_{ef} = 2 \cdot 0{,}785\left(D_1^2 - d_2^2\right) = 16{,}81\ cm^2$);

τ_a – permissible shear resistance of the material from which the piston bolt is made $\tau_a = 500\ bari$

By replacement, it is obtained:

$$\tau_{ef} = \frac{T_{max}}{A_{ef}} = \frac{400}{16{,}81} = 23{,}80\ bari < \tau_a.$$

In conclusion, the piston bolt withstands the shear request.

CHECKING THE CONNECTING ROD

The connecting rod mechanism is requested at combustion. It is considered that the main request is bending and that the task is distributed triangularly (Figure 4).

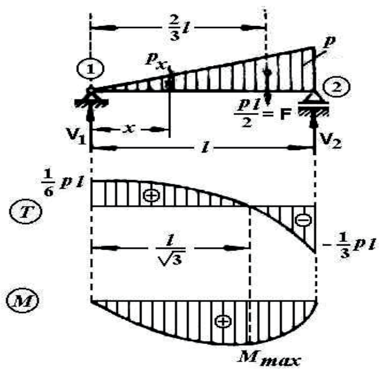

Figure 4. The connecting rod mechanism is requested at combustion

Because the load is distributed triangularly, its value differs from one cross-section to another of the connecting rod. For these reasons, the

connecting rod is made in the form of an equally resistant beam to another beam.
The total task that requires the connecting rod is: $F = p \cdot l / 2$.
The total load is considered to be applied in the centre of gravity of the triangle and the reactions in the supports (connecting rod bearings) are determined: $V_1 = \frac{1}{3}F = \frac{pl}{6}$ $V_2 = \frac{2}{3}F = \frac{pl}{3}$
In any section *x*, the distributed load has the intensity: $p_x = \frac{x}{l} \cdot p$
Taking into account that the derivative of the shear force with respect to the variable is equal to the distributed force taken with a changed sign ($\frac{dT}{dx} = -p$) and that the derivative of the bending moment with respect to the variable is equal to the shear force in the analysed section ($\frac{dM}{dx} = T$), cutting forces and bending moments can be determined by integrating these relationships.

$$T_x = -\int p_x dx = -\int \frac{x}{l} p dx = -\frac{px^2}{2l} + C_1$$

The constant C1 is determined by applying the relation in origin (*x* = 0), where $T = V_1 = \frac{pl}{6}$, hence: $T_x = \frac{pl}{6} - \frac{px^2}{2l}$.
It is observed that the cutting force varies parabolically from $\frac{pl}{6}$ (for $x = 0$) to $-\frac{pl}{3}$ (for $x = l$) and is cancelled at $\frac{pl}{6} - \frac{px^2}{2l} = 0$ for $x^2 = \frac{l^2}{3}$ and $x = \frac{l}{\sqrt{3}}$, respectively.
The expression of the bending moment in a section will be:

$$M_x = \int_0^x T_x dx = \int_0^x \left(\frac{pl}{6} - \frac{px^2}{2l} \right) dx = \frac{p\,l\,x}{6} - \frac{px^3}{6l} + C_2$$
.

The constant C_2 is determined for $x = 0$, where $M = 0$, hence $C_2 = 0$ and $M_x = \frac{plx}{6}\left(1 - \frac{x^2}{l^2}\right)$.

From the previous relation it follows that the bending moment varies according to a cubic parabola whose maximum is $x = \frac{l}{\sqrt{3}}$, hence:

$$M_{max} = \frac{pl^2}{9\sqrt{3}}.$$

From the relation $F = p \cdot l / 2$, it results $p = 2 \cdot F / l$.
Taking into account that the maximum force of inertia is $F_i = F = 400\ daN$ and that the length of the connecting rod, between the small head and the large head, is $l = 0{,}26\ m$, the distributed load will be:

$$p = 2 \cdot F / l = \frac{2 \cdot 400}{0{,}26} = 3077\ daN / m.$$

Therefore, the maximum moment will be:

$$M_{max} = \frac{pl^2}{9\sqrt{3}} = \frac{3077 \cdot \sqrt{3} \cdot 0{,}26^2}{27} = 13{,}346\ daNm = 1335\ daN \cdot cm$$

The dangerous net section of the connecting rod is at the small head, having the shape of an I_{26} profile. The cross-sectional strength modulus for profile I_{26} is: $W_z = 442\ cm^3$.
The actual voltage that occurs in the dangerous section, according to Navier's relation, has the value: $\sigma_{ef} = \frac{M_{max}}{W_z} = \frac{1335\ daN \cdot cm}{442\ cm^3} = 3.02\ bari$.
The connecting rod is made of cast iron alloyed with nodular graphite. The permissible strength for the connecting rod material is: $\sigma_a = 200 \div 400\ bari$, value much higher than the actual voltage in the dangerous net section. Therefore, the connecting rod is checked from the point of view of the resistance calculation.

CHECKING THE CRANKSHAFT

The crankshaft of the D-110 engine is made by alloy steel forging and treated in the outer layer in high frequency currents. It has five level bearings with a diameter of 85 mm and 4 bearings with a diameter of 75 mm. The length of the crankshaft between the first and last level bearing is 600 mm, and the handler range of 63 mm (r = S/2 – half of the piston race).
The main request of the crankshaft is the twist. The calculation is made for one of the handlers (smaller circular section).

The twisting checking relationship is: $\tau_{ef} = \dfrac{M_t}{W_p} \le \tau_a$, where:

τ_{ef} – effective tangential tension occurring in the cross-section of the spindle;

τ_a – permissible tensile strength of the material of which the crankshaft is made ($\tau_a = 300 \div 1000\ bari$);

M_t – maximum torque moment ($M_t = 29{,}5\ daN \cdot m$);

W_P – polar resistance modulus of the cross-section, given by the relationship: $W_P = \dfrac{\pi \cdot D^3}{16} = \dfrac{3{,}14 \cdot 7{,}5^3}{16} = 82\ cm^3$

The effective tension in the cross-section of the lever spindle will be: $\tau_{ef} = \dfrac{M_t}{W_p} = \dfrac{2950\ daN \cdot cm}{82\ cm^3} = 36\ bari < \tau_a$.

So, the crankshaft is checked at the twisting request.

CHECKING THE FLYWHEEL

The steering wheel is a heavy wheel, travelled on the crankshaft, having a high time of inertia in relation to the axis of rotation (*Figure 5*).

The steering wheel does not allow sudden variations in the angular speed, fixing the variation of the angular speed between the desired limits. The steering wheel works as a battery, storing energy when the movement accelerates, energy that then returns when the movement slows down. The disadvantage of the flywheel is that it makes it difficult to start or stop the car and reduce its maximum speed.

The D-10 engine steering wheel has the following features:

Outer diameter: D = 400 mm;

Generator (thickness): g = 80 mm;

Mass (without clutch): 36 kg.

Material of the flywheel: ash cast iron FC 25;

Material density: $\rho = 7{,}8\ grame / cm^3$.

The flywheel must ensure the quiet operation of the engine between two speed limits:

Minimum speed: $n_1 = 1000\ rot / min$ ($\omega_1 = 105\ rad / s$);

Maximum speed: $n_2 = 2600\ rot / min$ ($\omega_1 = 271\ rad / s$).

For this, the moment of inertia of the flywheel J_Δ should be as large as possible.

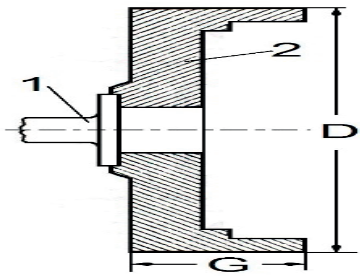

Figure 5. 1 - crankshaft; 2 - flywheel; D - flywheel diameter; G - Generator (thickness)

Moment of inertia relative to the axis of rotation J_Δ is determined by the relationship: $J_\Delta = \dfrac{1}{2} \cdot M \cdot R^2 \left[kg \cdot m^2\right]$, where:

M – flywheel mass [kg];

R – flywheel radius [m].

Angular velocity variation $\omega_2 - \omega_1$ during engine operation must not exceed n_r part of the average angular velocity: $\omega_m = \dfrac{\omega_1 + \omega_2}{2} = \dfrac{105 + 271}{2} = 188\ rad/\sec$

.Therefore, $\omega_2 - \omega_1 = \dfrac{\omega_m}{n_r} = \dfrac{\omega_1 + \omega_2}{2 \cdot n_r}$, hence: $n_r = \dfrac{\omega_1 + \omega_2}{2 \cdot (\omega_2 - \omega_1)} = \dfrac{105 + 271}{2 \cdot (271 - 105)} = 1{,}13$.

n_r It is called the regularity coefficient of the motor and the higher it is, the closer the motor has to the uniform one.

Applying the kinetic energy variation theorem, one can write: $E_{C_2} - E_{C_1} = \Delta L$, or $\dfrac{1}{2} J_\Delta \cdot \omega_2^2 - \dfrac{1}{2} J_\Delta \cdot \omega_1^2 = \Delta L$, hence $J_\Delta = 2 \cdot \dfrac{\Delta L}{\omega_2^2 - \omega_1^2} = \dfrac{n_r \cdot \Delta L}{\omega_m^2}$.

Taking into account that $\Delta L = P \cdot t$, where $P = 48000\ W$ and $t = 1\ s$, hence: $\Delta L = P \cdot t = 48000\ Jouli$, the axial moment of inertia will be:

$$J_{\Delta} = \frac{n_r \cdot \Delta L}{\omega_m^2} = \frac{1{,}13 \cdot 48000}{188^2} = 1{,}54\ kg \cdot m^2$$

Knowing the moment of inertia, the mass of the flywheel and its radius can determine the generator (thickness) of the flywheel.

The relationship $J_{\Delta} = \frac{1}{2} \cdot M \cdot R^2$ results in

$$M = \frac{2 \cdot J_{\Delta}}{R^2} = \frac{2 \cdot 1{,}54}{0{,}2^2} = 77\ kg$$

Knowing the density of the material, the volume of the flywheel (cylinder) *V* and its generator G are determined:

$$V = \frac{M}{\rho} = \frac{77000\ grame}{7{,}8} = 9872\ cm^3 = 0{,}785 \cdot D^2 \cdot G$$

$$G = \frac{V}{0{,}785 \cdot D^2} = \frac{9872}{0{,}785 \cdot 40^2} = 7{,}87\ cm,$$

which corresponds to the measurements made (G = 80 mm).

Therefore, the steering wheel is correctly sized and ensures the operation of the engine at constant speed.

CONCLUSIONS

The present paper presents a study on diesel engines that equip self-propelled tractors and machines. Using a vast bibliography, we analysed the evolution and performance of internal combustion engines and the dynamics of the motor mechanism.

Following the study conducted, the following conclusions resulted:

-Internal combustion engines that equip agricultural tractors are large mass engines that perform mechanical energy due to inertial mass properties;

-Diesel engines on tractors are "high engines" that perform the large cylinder race (they have a higher engine block than large bore engines) and therefore the crankshaft levers are larger, resulting in high torque;

-Using supercharged diesel engines with electronic injection in tractors reduces fuel consumption and polluting emissions, and improves their yield;

-Periodic technical checks and current internal combustion engines maintenance lead to reduction of costs and increase their reliability.

Resistance calculations on the subassemblies of the motor mechanism show that they resist the dynamic requests to which they are subjected during operation.

The steering wheel of the motor mechanism is correctly sized to keep the engine in constant speed for all operating regimes.

Regarding the energy base of the agricultural aggregates, the tendency to use tractors equipped with heavy power engines is noted. Under these conditions, their rational use in energy and technological aspect is ensured by the formation of combined aggregates, capable of performing several works, so that the duration of the work is reduced and savings regarding cost prices are made.

REFERENCES

Agape, I,, Gaiginschi, L., Ianuş, G. (2018). Studiu de caz asupra cauzelor defectării unui motor cu ardere internă destinat echipării unei combine Agricole. *Inginerie agrară și transport auto Vol.51, Simpozionul "Realizări și perspective în ingineria agrară și transport auto"* Chișinău, Moldova, 211-217.

Boca, C., Mura, A., Ilea, R., Duma Copcea, A. (2019). A comparative study regarding the use of internal combustion engines with different powers. *Research Journal of Agricultural Science*, 51 (3).

Duma Copcea, A., Mateoc Sîrb, N., Mihuț, C. (2024). Importanța eficienței economice în alegerea agregatelor de fertilizantă, *Scientific Papers Series Management, Economic Engineering in Agriculture & Rural Developm*ent, vol. 24, Numărul 1, 345-352.

Duma Copcea, A., Duma, C., Mihuţ, C.D., Sîrbu, C.C., Petanec, L.M., Mazăre, V., Scedei, D.N., Okros, A., Popa, D., Bungescu, S. (2023). Studii privind reducerea gradului de poluare la motoarele cu ardere internă. Știința în Nordul Republicii Moldova: realizări, probleme, perspective Ediția 7, 2023, Conferința"Ştiinţa în Nordul Republicii Moldova: realizări, probleme, perspective", 320-325.

Daicu, A., Novorojdin, D. (2015). Particularităţile utilizării carburanţilor mixşti în motoarele cu ardere internă pentru autovehicule. *Inginerie agrară și transport auto, vol 45*, Universitatea Agrară de Stat din Moldova, 297-301.

Gunston, B. (1999). Development of Piston Aero Engines. 2nd Edn., Patrick Stephens Ltd., Sparkford, UK., ISBN-10: 0-7509-4478-1, 21.

Hăbăşescu, I., Cerempei, V., Esir, M., Novorojdin, D., Banari, E., Lupaşcu, T., Dragalin, I. (2005). Indicii de performanţă a motorului cu aprindere prin comprimare alimentat cu biocombustibil. Energetica Moldovei. Aspecte regionale de dezvoltare, Ediția I, Conferința "Energetica Moldovei" Editia I, 2005, Chișinău, Moldova, 672-683.

Levente, B., Lelea, D., Birsan, N. (2015). Stand for Experimental Evaluation of Effects of Hydrogen Use in Internal Combustion Engines. Problemele Energeticii Regionale; Chişinău Vol. 2015, nr. 1.

Mateoc Sîrb, N., Otiman, P.D., Venig, A., Saida Feier, D., Mănescu, C., Duma Copcea, A., Mateoc, T. (2024). Analysis of the dynamics of organic agriculture in the european union with an emphasis on Romania. *Scientific Papers Series Management, Economic Engineering in Agriculture and Rural Development,* Vol. 24, Issue 2, 2024 PRINT ISSN 2284-7995, E-ISSN 2285-3952.

Mihuț, C., Mateoc-Sîrb, N., Duma Copcea, A., Niță, L., Ciolac, V., Okros, A., Popa, D. (2022). Assessment of soil quality limitative factors. a case study: Secaş, Timiş county, Romania. *Scientific Papers Series Management, Economic Engineering in Agriculture and Rural Development*, Vol. 22, Issue 1, 413-419.

Mihuț, C., Mateoc Sîrb, N, Duma Copcea, A., Niță, S., Lațo, I., Ciolac, V., Cozma, A., Okros, A. (2023). Assessing the state of fertility (quality) of the soils for sustainable agriculture in Remetea Mare, Timis county, Romania. *Scientific Papers Series Management, Economic Engineering in Agriculture and Rural Development,* Vol. 23, Issue 2, 487.

Mihuț, C., Niță, L., Duma Copcea, A., Rinovetz, A. (2024). Assessment of the productive capacity of agricultural lands for their sustainable use. *Scientific Papers Series Management, Economic Engineering in Agriculture and Rural Development*, Vol. 24, Issue 1, 2024 PRINT ISSN 2284-7995, E-ISSN 2285-3952.

Novorojdin, D. (2011). *Îmbunătăţirea performanţelor MAI cu utilizarea carburanţilor alternativi în regim instabil de funcţionare*, http://repository.utm.md/handle/5014/7831

Petrescu, F.I. (2015). Machine Motion Equations at the Internal Combustion Heat Engines. *American Journal of Engineering and Applied Sciences, 8*(1), 127-137.

Petrescu, F.I., Petrescu, R.V., (2015). Forces of Internal Combustion Heat Engines. *International Review on Modelling and Simulations* (I. R E .MO.S.), Vol. 7, n. 1.

Petrescu, F.I., Petrescu, R.V. (2011). Dinamica Mecanismelor de Distributie. *Create Space Publisher, USA., ISBN-10: 978-1-4680-5265-7,* 188.

Petrescu, F.I., Petrescu, R.V. (2009). Engine Design, Acta Technica Napocensis, Series: Applied Mathematics and Mechanics, CNCSIS 118 B, IS SN 1221-5872, Vol. Ib, ClujNapoca, 533-536.

Petrescu, R.V., Aversa, R., Akash, B., Bucinell, R.B., Calautit, J.K., Apicella, A., Petrescu, F.I. (2017). Forces at Internal Combustion Engines. *American Journal of Engineering and Applied Sciences 2017, 10 (2): 382.393 DOI:* 10.3844/ajeassp.2017.382.393.

Voleac, P. (2011). Aprecierea stării tehnice a motoarelor cu ardere internă aplicând metoda diagnosticării. Universitatea Agrară de Stat din Moldova,http://repository.utm.md/handle/5014/7834