Запропоновано модель робочих процесів двигуна автогрейдера при несталому навантаженні в технологічних процесах дорожньо-будівельних робіт. Визначені теоретичні залежності роботи двигуна автогрейдера. Встановлено закономірності впливу характеру навантаження на зміну частоти обертання колінчастого валу двигуна, циклової подачі палива і годинної витрати палива. Запропоновані залежності дозволяють оптимізувати роботу двигуна в несталих режимах

Ключові слова: експлуатація автогрейдера, режими роботи, несталe навантаження, динамічні характеристики, коливальний процес

Предложена модель рабочих процессов двигателя автогрейдера при неустановившейся нагрузке в технологических процессах дорожно-строительных работ. Определены теоретические зависимости работы двигателя автогрейдера. Установлены закономерности влияния характера нагрузки на изменение частоты вращения коленчатого вала двигателя, цикловой подачи топлива и часового расхода топлива. Предложенные зависимости позволяют оптимизировать работу двигателя в неустановившихся режимах

Ключевые слова: эксплуатация автогрейдера, режимы работы, неустановившаяся нагрузка, динамические характеристики, колебательный процесс

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MODELING OF OPERATION PROCESSES OF A MOTOR GRADER ENGINE DURING WORK UNDER UNSTEADY LOAD

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1. Introduction

A motor grader can be represented as a dynamic system with variable weight and velocity, which is influenced by load and position of the control element.

The indicated features affect velocity mode, thermal state of the engine, which in turn influences the course of processes of fuel and air supply, mixture formation, combustion, indicative and effective parameters of the engine.

When analyzing actual operating conditions, we will compile a schematic model of a motor grader engine. Let us examine operation of the engine while performing excavation works. In this case, the mass and gear, in which a motor grader operates, are constant, and the disturbance impact is only a change in the load.

The classic theory of internal-combustion engines has devised a whole range of evaluation indicators of the perfection of thermodynamic process and of the design of engine as a whole. However, when studying transient and unsteady processes, a number of indicators become difficult to determine. That is why it is necessary to select the basic indicators from a whole variety of them. The main indicators will make it possible to identify disruptions in the processes and changes in the output indicators of engine under transient modes with minimal consumption of resources and time.

2. Literature review and problem statement

An analysis of the processes that occur in internal combustion engines (ICE), taking into account their operation conditions, was performed by authors of papers [1, 2]. Contemporary software systems make it possible to simulate operation processes in the diesel engines, considering quantitative analysis of key parameters of an engine, which influence efficiency of thermal energy conversion.

When engine operates under unsteady load, there occurs a mismatch in the operation of all systems [3], as the engine was designed originally to work under a constant load, whereas it actually works under an unsteady load, which results in a decrease in the technical-economic parameters of an engine and a motor grader as a whole.

A method for the construction of a dynamic regulatory characteristic, taking into account random oscillations of resistance moment at the crankshaft of internal combustion engine, is proposed in article [4]. Due to continuous oscillations in the engine rotations, the following will change: filling ratio η_v, coefficient of air excess α, mechanical efficiency coefficient η_m and indicative efficiency coefficient.

Deterioration of effective indicators of engine under unsteady load depends on the amplitude and oscillations of rotation rate of the crankshaft of engine. The magnitude of load coefficient of a tractor engine is lowered in proportion to a decrease in torque safety factor. That is why the engines that operate under conditions of unsteady load should possess a higher torque safety than those under constant loads. In paper [5], in order to optimize fuel consumption of a motor grader, it was proposed to use a technique of shifting the curve of a grader engine capacity with the help of fuzzy adaptive control. There are known procedures for providing the engine with optimal temperature condition in the process of operation [6] through the use of comprehensive combined heating, as well as intelligent systems for control over cooling [7, 8].

Papers [9, 10] address the tasks of optimization of combustion processes in cylinders and thermal treatment processes. Based on the improved engine, a stationary bench test was conducted involving load testing at constant velocity and on roads. However, the character of the load of motor grader engines is transient, not stationary. At the same time, standards for engines testing regulate indicators under stationary modes. Under conditions of motion of a vehicle, operation and control parameters seriously deviate from the established values with large fluctuations, for example, ignition advance angle significantly slows down under conditions of sharp deceleration. That is why a simulation of operation processes of a motor grader engine under unsteady load is a promising scientific task.

3. Research goal and objectives

The goal of present study is to improve efficiency in the construction of technological processes of road-building works when laying road and railroad tracks by selecting the rational operating processes of a motor grader engine.

To accomplish the set goal, the following task was solved: we synthesized the model of working processes of a motor grader engine under unsteady load taking into account its operation conditions.

4. Materials and methods for examining working processes of a motor grader engine

The influence of control parameters of an engine on the indicators of its operation under unsteady load was explored using the method of simulation.

Dynamic processes in the internal combustion engines are described by linear differential equations within the pre-selected linear zones of change in the examined parameters. To obtain differential equations, we used the method of identification. Coefficients of differential equations were selected by the transitional curves obtained during load throw off and on.

5. Modeling of working processes of a motor grader engine

Dynamic qualities of a motor grader engine are defined by differential equations, which in turn are determined by experimental methods.

The linearity of dynamic properties can be verified experimentally. Guided by the dynamic studies presented in paper [2], we defined research zones by the linear sections of change in the analyzed indicators from a perturbing index by the stationary characteristic.

The studies that involved aperiodic impacts enabled us to receive dynamic characteristics, which represent the sum of free and forced motion of a section or a system, while frequency characteristics reflect only the forced motion of a section. Amplitude-frequency characteristic shows a degree of attenuation or amplification of harmonic signal that passes through a linear system [2].

Underlying the procedure for developing a mathematical model for the indicators of working process of an engine with the help of aperiodic impacts is the model proposed in paper [4].

The following magnitudes were chosen as the main indicators: rotation rate of the engine shaft (n), cyclic fuel supply (g_c), and air consumption (G_b). Employing the transition functions, we will define the kind and coefficients of differential equations whose solutions describe a change in the indicator during transition process.

When examining the regulatory branch, the engine's indicators are described by third order linear differential equations:

$$T_{3n}^3 \frac{\mathrm{d}^3 n}{\mathrm{d}t^3} + T_{2n} \frac{\mathrm{d}^2 n}{\mathrm{d}t^2} + T_{1n} \frac{\mathrm{d}n}{\mathrm{d}t} + n_0 = K_n \cdot \Delta M_c, \tag{1}$$

$$T_{3g}^3 \frac{\mathrm{d}^3 g_c}{\mathrm{d}t^3} + T_{2g} \frac{\mathrm{d}^2 g_c}{\mathrm{d}t^2} + T_{1g} \frac{\mathrm{d}g_c}{\mathrm{d}t} + g_{c_0} = K_g \cdot \Delta M_c, \tag{2}$$

$$T_{3b}^3 \frac{\mathrm{d}^3 G_b}{\mathrm{d}t^3} + T_{2b} \frac{\mathrm{d}^2 G_b}{\mathrm{d}t^2} + T_{1b} \frac{\mathrm{d}G_b}{\mathrm{d}t} + G_{b_0} = K_b \cdot \Delta M_c, \tag{3}$$

where T_i are the coefficients of differential equations; K_i are the factors of amplification; n_0 is the initial value of rotation rate of the engine crankshaft, min^{-1}; g_{c0} is the initial value of a cyclic fuel supply, g/cycle; G_{b0} is the initial value of the hourly air consumption by the engine, kg/h; ΔM_c is the law of change in the resistance moment acting on the crankshaft of the engine, N·m.

By applying the superposition principle, with the help of equations (1)–(3), it is possible to analyze behavior of the engine and its systems under all kinds of load. For this purpose, magnitude ΔM_c in the equations is replaced with the required law of a load change, and solutions to these equations determine the change in the examined indicators.

To determine effective indicators of engine performance, the basic equation of dynamics was used:

$$\Delta M_b(t) = \Delta M(t) + J_r \frac{\mathrm{d}\omega}{\mathrm{d}t}, \tag{4}$$

where $\Delta M_b(t)$ is the change in the engine torque, N·m; ω is the angular velocity of the engine crankshaft, s^{-1}; J_r is the inertia moment of a motor grader, reduced to the engine shaft, kg·m^2.

Reduced inertia moment can be represented mathematically in the form of the following dependence:

$$J = J_d + J_a, \quad (5)$$

where J_d is the reduced inertia moment of engine masses, kg·m²; J_a is the reduced inertia moment of motor grader masses, kg·m².

To determine a reduced inertia moment of the engine, a procedure presented in paper [2] is used.

As a result of reduction of the inertia moments of engine masses to the crankshaft axis, we receive the following dependence:

$$J_d = i_0\left(m_h + m_\omega\right)R_{cr}^2 + J_{cr}, \quad (6)$$

where i_0 is the number of cylinders; m_h is the mass of parts in the cylinder, making reciprocating motion, kg; m_ω is the weight of a part of a connecting rod, related to the axis of the crankpin, kg; R_{cr} is the radius of a crank, m; J_{cr} is the inertia moment of the engine crankshaft and of the masses rotating with it relative to the axis of the main journal, kg·m².

Mass m_h consists of the mass of a piston group m_n and a part of the mass of the connecting rod equal to 0.275 m_{cr} related to the axis of the upper head of the connecting rod, hence:

$$m_A = m_n + m_L\frac{L_B}{L}. \quad (7)$$

The remaining mass of connecting rod is related to the axis of the shaft crankpin:

$$m_{BL} = m_L\left(1 - \frac{L_B}{L}\right). \quad (8)$$

The reduced inertia moment J_{cr} of the engine crankshaft includes the reduced inertia moments of cranks J_c, and flywheel J_m:

$$J_{cr} = i_d J_c + J_m, \quad (9)$$

where i_d is the number of cranks of the crankshaft.

The reduced inertia moment of the flywheel will be determined from formula:

$$J_m = \frac{m_M D_M^2}{8}, \quad (10)$$

where m_M is the mass of the flywheel, kg; D_M is the diameter of the flywheel, m.

Similarly, we determine inertia moment of the coupling. Reduced inertia moment J_k is determined as the sum:

$$J_k = J_1 + J_2 + J_3, \quad (11)$$

where J_1 is the inertia moment of the mass of rotating cylindrical part of the shaft (main journal), kg·m²; J_2 is the inertia moment of the remaining rotating cylindrical part of the shaft (crankpin), kg·m²; J_3 is the inertia moment of the crank cheeks, kg·m².

$$J_1 = \frac{\pi l \rho}{32}\left(d_k^4 - d_b^4\right), \quad (12)$$

where ρ is the density of the shaft material, kg/m³; l is the length of the main journal, m; d_k is the outer diameter of the main journal, m; d_b is the drilling diameter, m.

$$J_2 = \frac{\pi l_{cp}\rho}{4}\left(d_{cr}^2 - d_{cp}^2\right)\cdot\left(R^2 + \frac{d_{cr}^2 - d_{cp}^2}{8}\right), \quad (13)$$

where d_{cr} is the diameter of the crankpin, m; d_{cp} is the drilling diameter of the crankpin, m; l_{cp} is the length of the crankpin, m.

If the shaft cheek is of shape close to a parallelepiped, the inertia moment of two cheeks can be determined from formula:

$$J_3 = 2\cdot a\cdot b\cdot h\cdot l_0\cdot\left(p^2 + \frac{a^2 + l_0^2}{12}\right), \quad (14)$$

where b is the width of the cheek, m; h is the thickness of the cheek, m; a is the cheek height, m; p is the distance of the centre of gravity of cheeks from the axis of rotation, m.

When the engine's velocity mode changes, the number of revolutions of the camshaft, rollers and rotors of auxiliary units and velocity of valves also change. Therefore, their inertia moment should also be reduced to the engine's crankshaft. However, the magnitude of this reduced inertia moment is small compared with moment J_d and therefore it is evaluated approximately by the introduction of factor X_d=1.1...1.2.

Then, the total reduced inertia moment of the engine:

$$J_d = X_d J_k. \quad (15)$$

When conducting theoretical and experimental studies, we assume that the inertia moment of the pendulum machine and the cardan joint, connecting the engine with the pendulum machine, represents a reduced inertia moment of the motor grader.

The most appropriate by inertia moment is the operation of a motor grader during excavation works.

Working speed of a motor grader in the fifth gear is 10 km/h. Depth of the soil cut is 8...10 cm. Soil type is medium loam with a relative humidity of 13...16 % and specific resistance ≈5 N/cm².

A reduced inertia moment is determined from the following dependence:

$$J_2 = J_d + J_T + J_c, \quad (16)$$

where J_T is the inertia moment of the rotor of a pendulum machine, J_T=1.7 kg·m²; J_c is the inertia moment of cardan joint with flanges, J_c=0.7025 kg·m².

The inertia moment of the rotor of a pendulum machine is determined according to technical characteristic; in this case, it takes into account inertia moments of intermediate connections.

The most unfavorable conditions of engine operation are the abrupt loading and abrupt load throw-off. That is why subsequent calculations will be carried out for this type of engine load. Consider a solution to the equations of load gain (load throw off). A change in the rotation rate of the engine crankshaft:

$$T_{3n}^3\frac{\mathrm{d}^3 n}{\mathrm{d}t^3} + T_{2n}\frac{\mathrm{d}^2 n}{\mathrm{d}t^2} + T_{1n}\frac{\mathrm{d}n}{\mathrm{d}t} + n_0 = K_n\cdot\Delta M_c. \quad (17)$$

A change in the cyclic fuel supply:

$$T_{3n}^3\frac{d^3n}{dt^3}+T_{2n}\frac{d^2n}{dt^2}+T_{1n}\frac{dn}{dt}+n_0=K_n\cdot\Delta M_c. \qquad (18)$$

A change in the air consumption:

$$T_{3b}^3\frac{d^3G_b}{dt^3}+T_{2b}\frac{d^2G_b}{dt^2}+T_{1b}\frac{dG_b}{dt}+G_{b0}=K_b\cdot\Delta M_c. \qquad (19)$$

General form of solution to third order differential equations is represented by the following expression:

$$E_{g.s.n.u}=E_{g.s.u}+E_{p.s.n.u}, \qquad (20)$$

where $E_{g.s.n.u}$ is the general solution to a non-uniform equation; $E_{g.s.u}$ is the general solution to a non-uniform equation; $E_{p.s.n.u}$ is the particular solution to a non-uniform equation.

Particular solution to a non-uniform equation takes the form:

$$E_{p.s.n.u}=f(K_i\cdot\Delta M_c), \qquad (21)$$

where $\Delta M_C=f(t)$ is the law of change in the resistance moment acting on the crankshaft of the engine.

When solving differential equations, we select a law of change in resistance moment that fully describes a change in the resistance moment reduced to the crankshaft of the engine during operation of a motor grader.

General solution to a third order differential equation in the field of vibrational convergent processes will take the form:

$$y=e^n\left(C_1\cos\omega t+C_2\sin\omega t\right)+C_3e^{K_d}+K\Delta M_c, \qquad (22)$$

where C_1, C_2, C_3 are the constants of integration, dependent on the initial conditions and complex roots of equation; K_d is the coefficient of a differential equation; K is the amplification factor.

To facilitate the record, we will replace in the equation:

Initial conditions of solution to the differential equation:

$$\frac{d^3y}{dt^3}\Rightarrow\dddot{y},\quad \frac{d^2y}{dt^2}\Rightarrow\ddot{y},\quad \frac{dy}{dt}\Rightarrow y.$$

Solution to the system of equations will take the form:

$$\begin{cases} y(0)=y(0), \\ \dot{y}(0)=0, \\ \ddot{y}(0)=0 \end{cases} \qquad (23)$$

or

$$\dot{y}=e^n\begin{bmatrix}(\gamma C_1+\omega C_2)\cdot\cos\omega t+ \\ +(\gamma C_2-\omega C_1)\cdot\sin\omega t\end{bmatrix}+K_3C_3e^{K_3t}, \qquad (24)$$

$$\ddot{y}=t_n\begin{bmatrix}(\gamma^2C_1+2\gamma\omega C_2)\cdot\cos\omega t+ \\ +(\gamma^2C_2+2\gamma\omega C_1)\sin\omega t\end{bmatrix}+K_3^2C_3t^{K_3t}. \qquad (25)$$

After transformation and solution of equations (24) and (25), we find the values of integration constants.

When solving a differential equation, we used the principle of numerical integration and determined optimal values of the law of load (ΔM_C), the most appropriate for the operation of a motor grader during excavation works.

Assume there is a differential equation:

$$T_3{}^3\frac{d^3y}{dt^3}+T^2{}_2\frac{d^2y}{dt^2}+T_1\frac{dy}{dt}+y=k\cdot x(t),$$

where $x(t)$ is the function that determines influence on the system; $y(t)$ is the function (response of the system); T_1, T_2, T_3, k are the constant coefficients.

Parameters $x(t)$, T_1, T_2, T_3, k are set.

Initial conditions are $y(0)$, $\dfrac{dy}{dt(0)}$, $\dfrac{d^2y}{dt^2}$. It is required to find $y(t)$.

Rewrite the differential equation in the following form:

$$\frac{d^3y}{dt^3}=\frac{k}{T_3^3}\cdot x(t)-\frac{T_2^2}{T_3^3}\cdot\frac{d^2y}{dt^2}-\frac{T_1}{T_3^3}\cdot\frac{dy}{dt}-\frac{1}{T_3^3}y. \qquad (26)$$

The idea of numerical integration, employed in the present study, can be explained by the structure, shown in Fig. 1. Let us explain the principle of operation.

To start with, we will note that index i will denote the function count at the i-th moment of time.

Let the first count of input impact x^1 enter the input. Multiplied by the constant, the signal enters the input of the adder. The values of counts at other inputs of the adder are determined by the initial conditions:

$$y^1=y(0), \qquad (27)$$

$$\frac{dy^1}{dt}=\frac{dy}{dt}(0), \qquad (28)$$

$$\frac{d^2y^1}{dt^2}=\frac{d^2y}{dt^2}. \qquad (29)$$

At the output of the adder, we will obtain the count $\dfrac{d^3y^1}{dt^3}$. Next, by numerically integrating $\dfrac{d^3y}{dt^3}$, assuming that all the previous counts were equal to zero, we obtain $\dfrac{d^2y^2}{dt^2}$. The two subsequent integrators yield values y^2, $\dfrac{dy^2}{dt}$.

Then the next count x^2 enters the input. It enters the input of the adder. The values, defined by counts y^2, $\dfrac{dy^2}{dt}$, $\dfrac{d^2y^2}{dt^2}$. enter other inputs.

At the output of the adder, we obtain the count $\dfrac{d^3y^2}{dt^3}$. After this, the operations are repeated. The resulting range of values $\{y^1\}$ is the desired numerical solution to the differential equation.

Changes in the engine indicators under transient modes are described by linear differential equations. Coefficients of differential equations characterize the quality and duration of the examined transient processes. That is why, in order to assess dynamic properties of the engine and its systems, we apply coefficients of differential equations and the time of delay in parameter's change caused by disturbance.

To optimize calculations, changes in parameters in the transition process during loading throw off and loading gain are taken in relative magnitudes. The maximum value is recognized as unity ($A_{max}=1$), the minimum value – as zero ($A_{min}=0$).

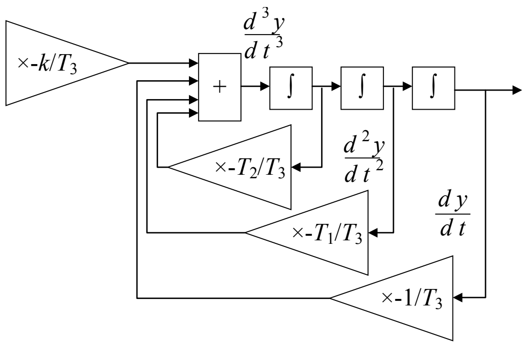

Fig. 1. Structural circuit explaining the principle of numerical solution to a differential equation

When modeling transient processes, the values of coefficients of differential equation were taken based on data obtained in the course of laboratory experiments. An increase in the range of coverage makes it possible to determine the effect of coefficients of differential equations on the course of a transition process.

In order to develop theoretical data, it is necessary, based on the results of obtained experimental data, to assign coefficients of differential equations and then solve it with the given coefficients. It is necessary to compare the obtained transient curves for the convergence with experimental ones in order to confirm adequacy of the mathematical model.

6. Consideration of results of examining working processes of a motor grader engine

Requirements to the engines of construction and road machinery are basically identical to the requirements for motor tractor engines. However, specific conditions in the operation of specified machines predetermine a number of special, additional requirements to the purification of oil, air and fuel, reliability of cooling systems, starting systems, control and standard equipment.

The study of character of unsteady load revealed that the most appropriate method to enhance operating efficiency of a motor grader engine is to improve the connection between operation of the engine systems and the character of unsteady load on the blade of a motor grader.

Analytical method for selecting the operating parameters makes it possible to determine the optimal magnitude of engine load, taking into account the features of the load mode. The optimal magnitude of engine load is selected depending on the load character and engine characteristics. This magnitude, during operation under unsteady load, is always lower than a nominal torque of the engine. The proposed mathematical model of operation processes of a motor grader engine under unsteady load allows us to improve effectiveness of the construction of technological processes of road-building works when laying roadways and railway tracks by taking into account working conditions of the elements of the system "driver – motor grader – road medium".

Optimization of operation processes of the engine enables us to improve significantly the character of engine operation under unsteady load. The main indicators include the following magnitudes: rotation rate of the engine motor shaft *(n)*, cyclic fuel supply (g_c), and air consumption (G_b). As a result, more qualitative work of the engine makes it possible to improve technical and economic characteristics of a motor grader.

It should be noted that in the present study we made an assumption about stationarity of engine indicators over time, which somewhat limits the application of the obtained results.

Economic effect when employing the optimization of operation processes of an engine consists of the following components: improvement of engine operation under unsteady load, characterized by an abrupt change in resistance moment, which will increase efficiency of the use of a motor grader under actual operating conditions.

7. Conclusions

The developed mathematical model of operation processes of a motor grader engine under unsteady mode makes it possible to apply known theoretical provisions to improve the system of regulation of air supply in commercially available motor grader engines. The model is presented in the form of theoretical dependences of operation processes of the diesel motor grader engine, described by third order differential equations. The specified dependences make it possible to establish the patterns of influence of coefficients of differential equations and load character on a change in the rotation rate of crankshaft of engine, cyclic fuel supply and hourly fuel consumption, which would ultimately optimize engine performance under unsteady modes.

Numerical modeling was carried out of load throw off and load gain of a motor grader engine using third-order differential equations in relative magnitudes. It was determined that at a decrease in the values of coefficients of a differential equation, the transition process proceeds more intensively. In this case, time of delay in the response to disturbance and the duration of damping the oscillation process decrease.

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DETERMINING THE MAGNITUDE OF TRACTION FORCE ON THE AXES OF DRIVE WHEELS OF SELF-PROPELLED MACHINES

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Наведено принципи утворення тягової сили тракторів та автомобілів. Точка прикладання тягової сила знаходиться на осі приводного колеса. Причому, величина тягової сили приблизно у два рази перевищує реактивну силу, що передається від дороги на приводне колесо. Визначення принципів утворення тягової сили дозволяє інтенсифікувати дослідження технічних та енергетичних засобів загалом та обґрунтувати принципи зменшення негативного впливу рушіїв на родючий шар ґрунту, зокрема

Ключові слова: тягова сила, приводне колесо, самохідні машини, енергетичний засіб, трансмісія

Приведены принципы образования тяговой силы тракторов и автомобилей. Точка приложения тяговой силы находится на оси ведущего колеса. Причем, величина тяговой силы примерно в два раза превышает реактивную силу, что передается от дороги на приводное колесо. Определение принципов образования тяговой силы позволяет интенсифицировать исследования технических и энергетических средств в общем и обосновать принципы уменьшения негативного влияния движителей на плодородный слой почвы, в частности

Ключевые слова: тяговая сила, приводное колесо, самоходные машины, энергетическое средство, трансмиссия

1. Introduction

A clear understanding of the application processes that are means for the purpose of their further improvement. It also concerns the issues of formalization of the process of creating traction force on the drive wheel of tractors and vehicles. This is especially important for the agricultural machine-tractor units that cause a number of problems related to the interaction between running systems and the surface of fertile layer of soil. Solving the problems aimed at bringing down negative impact of engines on the fertile layer of soil is a relevant task today.

2. Literature review and problem statement

The occurrence of traction force on the drive wheel is often explained by the fact that at the point of contact between the wheel and road surface the torque (Fig. 1–3) causes tangent force F_O. The counteraction of road to this tangent force