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DETERMINATION OF OPTIMUM CONSTRUCTIONAL DATA OF FEED DEVICE OF FRICTION MODIFIER WITH PNEUMATIC DRIVE

Julia Baranych, Jaroslav Mushkajev, Olexsander Klujev

Volodymyr Dal East-Ukrainian National University, Lugansk, Ukraine

Summary. The articles describes mathematical model of operation of pneumatic drive device, which increases traction coefficient of locomotive with rails and describes solution of this model in real conditions. Also the principle pneumatic scheme of feed of traction modifier is suggested for locomotive 2ТЭ116.

Key words: friction modifier, slip velocity, pneumatic drive**,** friction coefficient**.**

INTRODUCTION

Recently the friction modifiers Centrac **VHPF, HPF, LCF**, designed by a company Portec Railway Maintenance Products and company Kelsan Lubricants and applied in a number of countries (The USA, Canada, France) are “know-how” in area of high positive friction supplying in contact “wheel-rail”. The new generation of friction modifiers is usually used as solid lubricant or in the fluid form and it is feed as applicator, which is installed on locomotive equipment. The friction modifiers must have good holding capacities and good water resistance to maintain frequent contact effects. The value of friction coefficient depends on slip velocity of rolling-stock wheels and a coating thickness, which is produced by the friction modifiers [1,2].

The specialized friction modifiers **HPF** improve effectively the friction coefficient and remove squealing or other types of high noise levels on a track with rails, which is subjected scalloping wear and wheel spin. However, in practice these devices are ineffective without special control system or they cause the overrun of material. The article describes the algorithm of discrete method of solid lubricant feed, which consists of abrasive and silicate film-forming admixture; mathematical simulation of operation of feed device of friction modifier with pneumatic drive is developed, and principle pneumatic scheme of feed of friction modifier is suggested for locomotive 2ТЭ116 [3], it is shown on a fig. 1.

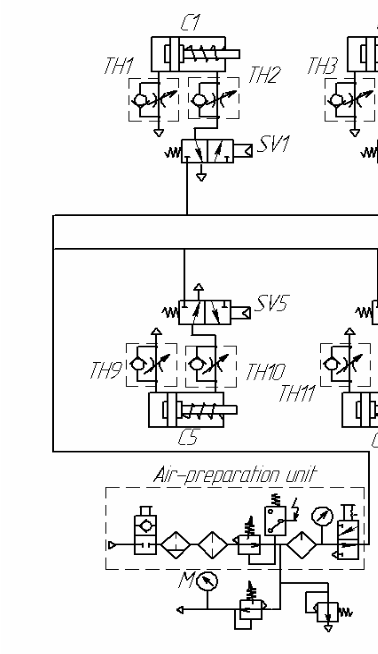

Fig. 1. Principle pneumatic scheme of feed device of friction modifier for locomotive 2ТЭ116

OBJECTS AND PROBLEMS

The device must satisfy dynamic characteristics (for example, velocity of solid lubricant in moment of contact with surface) and static characteristics (maximal press power); control simplicity.

Determination of dimensions of structural components is based on the followings set values:

1. velocity of solid lubricant in moment of contact with surface must not exceed a set value $V_{\max}$, in order to avoid impact loads which can cause the destruction (spalling) of solid lubricant;

2. press power of solid lubricant must not exceed a maximum load $F_{\max}$;

The motion of moving parts of compressed air cylinder with solid lubricant is considered on a figure 1. The moving direction of piston is chosen on the right – to the left. Dynamic equation for system piston-solid lubricant is determined as [4]:

$$(m_1+m_2)\frac{dV}{dt}=p_1F_1-p_2F_2+p_aF_p-R_0-cy-k_{mp}V-R_{cm}sign(V), \quad (1)$$

where :m_1 - reduced mass of moving parts of compressed air cylinder; m_2 - reduced mass of solid lubricant; V - rate of movement of piston-rod of compressed air cylinder; $p_1, p_2\, p_{атм}$ - absolute pressures in rod end, in blind side, atmospheric pressure accordingly; F_1, F_2, F_p - areas of rod end, blind side, and area of piston-rod accordingly; c - spring constant; k_f - friction coefficient, proportional to speed in the first degree; R_f - force of friction of moving parts of compressed air cylinder; R_o - effort of the initial spring pressing; $sign(V)$ - a sign of velocity value.

Taking into account observations, this equation can be presented as:

$$\frac{dV}{dt} = \frac{p_1F_1 - p_2F_2 + p_aF_p - R_0 - cy - k_fV - R_f sign(V)}{m_1 + m_2} . \tag{2}$$

Let's consider the pneumatic discharge line (from input throttle to rod end of compressed air cylinder), taking into account the dead zone of chamber of compressed air cylinder:

$$\frac{w_{01} + w_{m1} + F_1 y}{kRT_1} \cdot \frac{dp_1}{dt} = G_1 - \frac{p_1F_1}{RT_1}V , \tag{3}$$

where: w_{01} - dead zone of rod end of compressed air cylinder; w_{m1} - volume of discharge pipeline; $k = 1{,}4$ - adiabatic exponent; $R = 287 \frac{J}{\kappa g \cdot {}^0K}$ – gas constant for dry air; T_1 - absolute temperature in rod end of compressed air cylinder and in discharge pipeline; G_1 - air-mass flow, acting through input throttle in discharge pipeline; p_1 - absolute pressure of air in rod end of compressed air cylinder; F_1 - area of pressure of rod end.

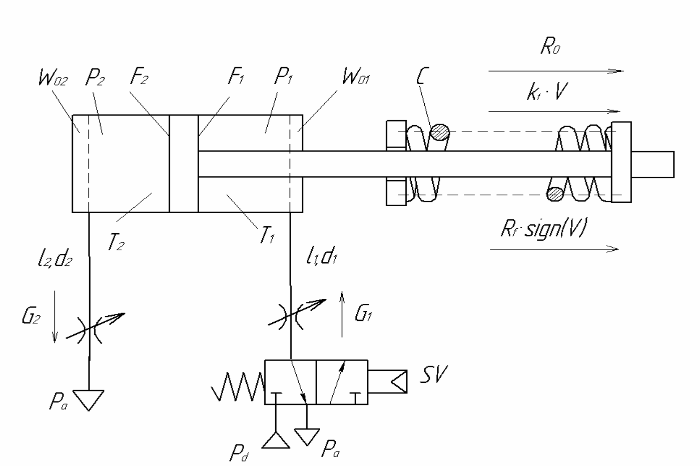

Fig. 2. Design model of operation of compressed air cylinder

Let's consider the pneumatic output line (from blind side of compressed air cylinder to output throttle), taking into account the dead zone of chamber of compressed air cylinder:

$$\frac{w_{02}+w_{m2}+F_2(H-y)}{kRT_2}\cdot\frac{dp_2}{dt}=\frac{p_2F_2}{RT_2}V-G_2, \qquad (4)$$

where: w_{02} - dead zone of blind side of compressed air cylinder; w_{m2} - volume of output pipeline; T_2 - absolute temperature in blind side of compressed air cylinder and in output pipeline; G_2 - air-mass flow, acting through output throttle in an atmosphere; p_2 - absolute pressure of air in blind side of compressed air cylinder; F_2 - area of blind side.

Air-mass flow through input throttle can take different values in accordance to subsonic or supersonic gas flows:

$$G_1=\begin{cases}\mu_1 f_1 p_d\sqrt{\dfrac{2k}{(k-1)RT_1}\left[\left(\dfrac{p_1}{p_d}\right)^{\frac{2}{k}}-\left(\dfrac{p_1}{p_d}\right)^{\frac{k+1}{k}}\right]}; & \dfrac{p_1}{p_d}>\sigma_c\\ \mu_1 f_1 p_d\left(\dfrac{2}{k+1}\right)^{\frac{1}{k-1}}\sqrt{\dfrac{2k}{(k+1)RT_1}}. & \dfrac{p_1}{p_d}\le\sigma_c\end{cases} \qquad (5)$$

where: μ_1 - discharge coefficients of input throttle; f_1 sectional area of discharge pipeline; p_d - discharge pressure [Pa]; $\sigma_c=\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}=0{,}528$ - critical relation of pressures for dry air.

Air-mass flow through output throttle can take different values in accordance to subsonic or supersonic gas flows:

$$G_2=\begin{cases}\mu_2 f_2 p_2\sqrt{\dfrac{2k}{(k-1)RT_2}\left[\left(\dfrac{p_a}{p_2}\right)^{\frac{2}{k}}-\left(\dfrac{p_a}{p_2}\right)^{\frac{k+1}{k}}\right]}; & \dfrac{p_a}{p_2}>\sigma_c\\ \mu_2 f_2 p_2\left(\dfrac{2}{k+1}\right)^{\frac{1}{k-1}}\sqrt{\dfrac{2k}{(k+1)RT_2}}. & \dfrac{p_a}{p_2}\le\sigma_c\end{cases}, \qquad (6)$$

where: μ_2 - discharge coefficients of output throttle; f_2 - sectional area of output pipeline; p_a - atmospheric pressure.

The obtained differential equations (1–6) can be presented in Cauchy’s form, that allows to apply for their solutions one of the known numeral methods of integration. A method of Runge-Kutta is most preferable, because it has high accuracy at rapid convergence. The assumptions of concerned mathematical model are such as:

1. Thermodynamics process is adiabatic in chamber of compressed air cylinder;

2. Temperatures T_1 and T_2 are average absolute temperatures, and are accepted equal value in calculations and does not change during the cycle of device operation;

3. Time, during which pressure of air before input throttle rises to deliver pressure, has so small value, that it can be considered equal to the zero;

4. The gravity work, operating compressed air cylinder, is neglected on account of small value of it;

5. The dead zone into pneumatic line is neglected as the walls of pneumatic lines are not deformed.

The values of basal characteristics of air are presented in a table 1.

The solution of equations is fulfilled in systems of MATLAB, data of calculations on the program are shown on figures 3, a –d.

Table 1. **Numerical values of basal characteristics of air**

Characteristic	Clause	Numerical values
1. Density $\rho, кг/м^3$	$p = 1.013 \cdot 10^5 Па$	1.207
2. Specific gravity $\gamma, H/м^3$	$p = 1.013 \cdot 10^5 Па$	11.82
specific volume $V, м^3$	$t = 20℃$	0.83
3. Gas constant $R, \frac{J}{кg \cdot {}^0K}$	Dry air	287
(RH 80%)	Moist air	289
4. Coefficient of dynamic viscosity $\mu, Pa \cdot c$	$t = 20℃$	17.88•10-6
5. Heat capacity $C_p \frac{J}{кg \cdot {}^0K}$	$t = 0 - 100^0 C$	18.4•106
$C_v \frac{J}{кg \cdot {}^0K}$	c = const	0.72•103

Air for the delivery of system of pneumatic drive is taken from delivery system of sanding apparatus and previously it is reduced to delivery pressure of pneumatic drive $p_{dp} = 0{,}2MPa$.

The curves of moving, rate of movement, discharge pressure and output pressure are shown on figure 3.

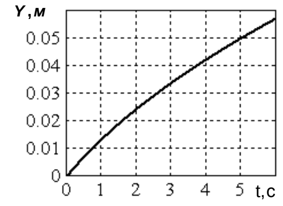

Fig. 3a. Motion curve

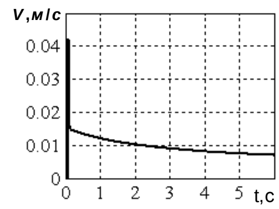

Fig. 3b. Rate of movement curve

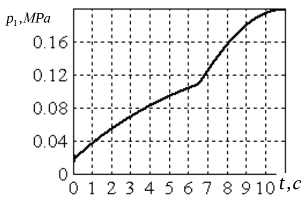

Fig. 3c. Discharge overpressure curve

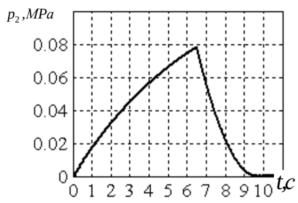

Fig. 3d. Output overpressure curve

CONCLUSIONS

Thus, calculations proved that this mathematical model allowed to choose optimum constructional data of feed device of traction modifier, for example, for pneumatic drive (area of work surface, length of piston-rod) from condition of absence of destruction (spalling) of solid lubricant, which is determined by rate of movement of solid lubricant in moment of contact and rate of increase of pressure power after moment of interaction.

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ВЫБОР ОПТИМАЛЬНЫХ КОНСТРУКТИВНЫХ ПАРАМЕТРОВ УСТРОЙСТВА ПОДАЧИ АКТИВИЗАТОРА СЦЕПЛЕНИЯ С ПНЕВМАТИЧЕСКИМ ПРИВОДОМ

Баранич Ю.В., Мушкаев Я.В., Клюев А.С.

Аннотация. В работе обосновывается и предлагается математическая модель пневматического привода устройства повышения коэффициента сцепления локомотива с рельсами. Приведено решение данной модели при условиях, приближенных к реальным. Также предложена принципиальная пневматическая схема устройства подачи активизатора сцепления для 2ТЭ116.

Ключевые слова: трение, модификаторы, присадки колесо-рельс, пневматический привод.