COMPUTER SIMULATION OF PNEUMATIC ENGINE OPERATION

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Abstract

A mathematical model for the three cylinder pneumatic engine is proposed, which allows calculating both the dynamic characteristics of piston motion and flowing gas parameters without using any fitting procedures. The corresponding computer code in MATLAB-SIMULINK software is developed and numerical simulation of the engine operation has been accomplished With the help of MATLAB-SIMULINK software and AUTOMOTIVE STUDIO software. The approach proposed allows calculation of a wide set of thermodynamic and operational parameters for various pneumatic cylinders and can be used for development of the highly efficient pneumatic engine intended for vehicle propulsion.

Keywords: *Pneumatic Engine, Simulation, Automotive studio, Fabrication*

1. INTRODUCTION

At the present time, a new direction in designing automobile using the compressed air technologies and pneumatic power plants is being developed[1-3].compressed air engine having the high efficiency as compared to gasoline engine and correspondingly low consumption of compressed air are necessary for development of non-polluting pneumatic automobile that run on compressed air. The piston expansion machine based on pneumatic cylinder most closely correspond to this criterion[4].recent developments in pneumatic servo system and innovative pneumatic components[5,6] show important advances, which are expected in vehicle applications also.

Design optimization for a pneumatic engine of a given set of characteristics is possible as a result of mathematical modeling of working cycle. Therefore, the development of a adequate mathematical model is a reasonable scientific and technical approach.

The purpose of making the new design of the pneumatic engine and mathematical model developed is to determine the basic dynamic parameter and the effectiveness of new design. The dynamic parameters such as air pressure in the cylinder, position and speed of the piston in time, cycle frequency and the operational characteristics such as power, efficiency, specific work, air consumption, etc. of the pneumatic engine are being considered in this analysis of three cylinder pneumatic engine.

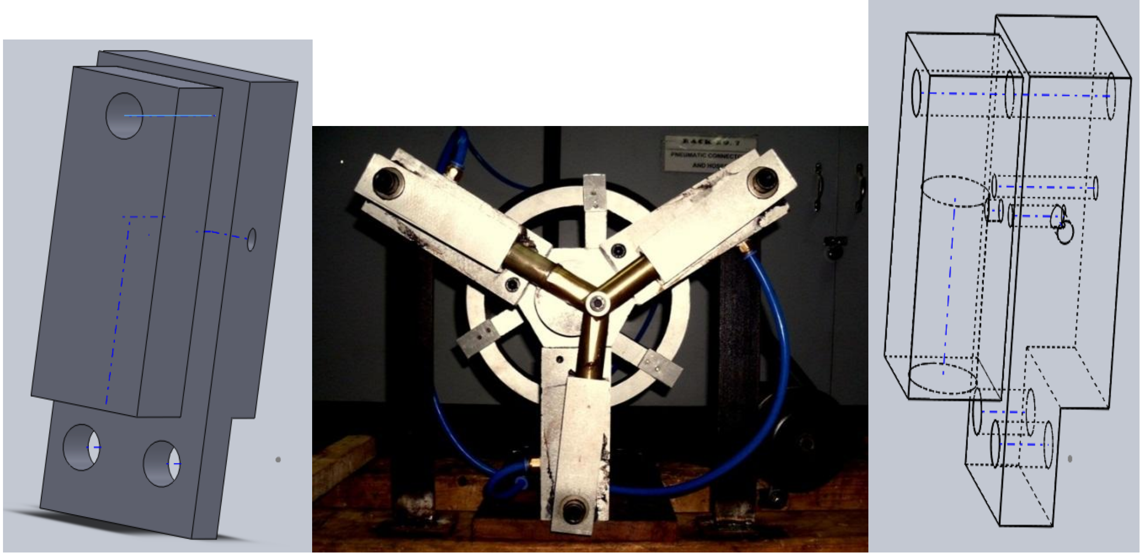

Fig1:-Design of engine parts in solidworks software and the pneumatic engine which is fabricated.

2. DESIGN CONSIDERATION :

Each component in engine frame and cylinder has design limit. To ensure that these are not exceeded in operation, each frame and each cylinder has a design rating above which it may not be used. The loads used to rate pneumatic engine are discussed below. All three cylinders have a maximum allowable inlet pressure. All engine components are subjected to alternating loads and the rated pressure of a cylinder will be based on fatigue considerations.

Every cylinder has a minimum clearance it can be built with. This controls the volumetric efficiency of the cylinder and hence the capacity for a given pressure ratio and air consumption. The clearance of a cylinder can usually be increased if the maximum capacity is not needed for a given application.

Cylinders has a fixed number of valves and valve size. A cylinder with a few or small valves for its size will have losses and will give poor efficiency.

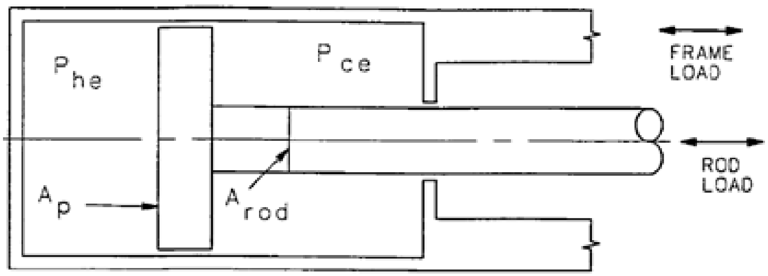

Fig:-2 Frame and rod loads

Each cylinder exerts a rod load on the running gear components, and a frame load on the stationary components. These can be evaluated by considering the forces acting on the various components.

Frame Load=$P_{HE} A_P - P_{CE} (A_P - A_{ROD})$

Where

P_{HE} and P_{CE} = Pressure in the head end and crank end of the cylinder

A_P and A_{ROD} =The area of the piston and piston rod

The frame load will vary through the cycle as the pressures in the head end and crank end of the cylinder vary. The maximum tensile and maximum compressive stresses are calculated. These are the loads the stationary components and bolting must be designed to resist. The rod load, the force exerted on the piston rod, connecting rod and crankshaft is different for each component. It is the frame load plus the inertia of all the parts outboard of the component that is of interest. The inertia is the mass times the piston acceleration, and varies through the cycle.Once the suction and discharge pressure, the suction air temperature, the required flow rate and the air composition are determined, a engine can be selected to design.

The engine speed and the stroke will then be determined by the horsepower requirement. Alow horsepower application will require a light, low stroke, high speed engine. A high horse power application will require a heavy, long stroke, low speed engine.

3. DEVELOPMENT OF MATHEMATICAL MODEL :

The schematic diagram of a pneumatic engine is represented on fig. In this we consider only one cylinder in spite of three, as the working of the all three cylinder are same and the mathematical model have no effect on, which cylinder we used under observation.

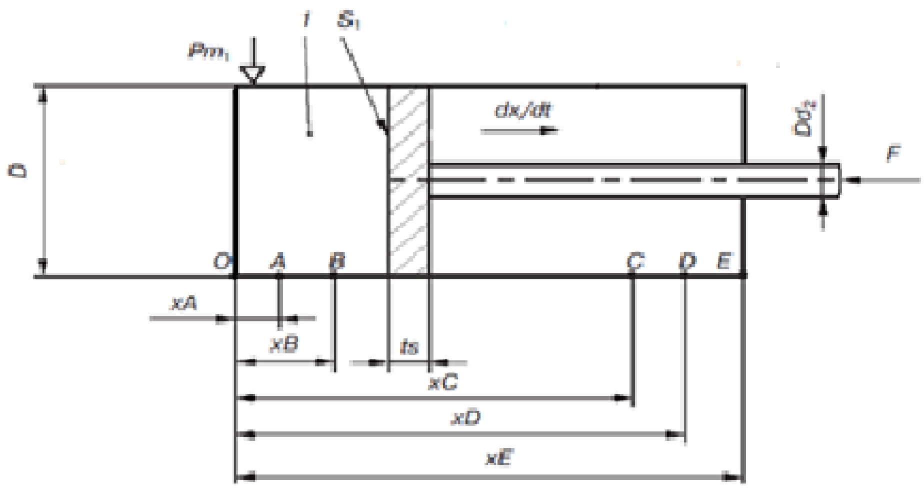

Fig.:-3 the schematic of the pneumatic cylinder

In the above cylinder, we will first consider a stroke of the piston from left to right. The movement of the piston from point A to B corresponds to the process of filling of the cylinder by compressed air from inlet manifold P_{m1} and exhaust of simple air from the exhaust manifold P_{m2}. At the point B the supply from the inlet manifold is cut off by closing of the inlet valve, as the axis of the cylinder and supporting assembly orifice are not co-inside due to cylinder harmonic motion. The movement of the piston between B and D corresponding to the process of expansion of the working air in the cylinder, as the compressed air is expand. At the point D the exhaust valve is open, as the axis of the cylinder and supporting assembly's exhaust orifice are co-inside, due to the cylinder harmonic motion. All three cylinder which are placed at 120^0 to each other, done the same work vice versa.

For a description of the dynamics of the piston movement between the point A and D, it is necessary to determine the parameters of working gas state. The equation of the piston motion generally can be written as:-

$$\frac{d^2x}{dt^2}M = p_1 s_1 - F \tag{1}$$

Where p_1, is the pressure in the working cylinder,$s_1$is the useful areas of the piston for cylinder is the resistance force; which consist of the force of friction F_{fr}, and loading force F_L; M is the mass of piston with all moving parts(roads, crank shaft mechanism, etc.)

In the above equation of the piston motion. The quantity p_1 is unknown. Now we have to derive the equation of pressure at every stage of motion of piston measured from the right edge of the piston.

Let we suppose that all the thermal energy dQ_{m1}, which is admitted with the air in cylinder is changed to internal energy dU_1, and the work of the gas expansion dW_1, and according to the first law of thermodynamics, the equation of the energy balance is written as

$$dQ_{m1} = dU_1 + dW_1 \tag{2}$$

Assuming, that pressure in the system of receiver manifold does not vary during filling of the working cylinder, we use the relation $dQ_{m1} = dH_{m1}$. In this case the quantity of thermal energy, which has arrived from the inlet(P_{m1}) to the cylinder, is equal to the product of the mass of gas dm_{m1} and the specific enthalpy($dQ_{m1} = h_{m1}dm_{m1}$) and the change of the gas internal energy dU_1 and work dW_1 made by it are equal accordingly $dU_1 = d(u_1 m_1)$ and $dW_1 = p_1 dV$. Therefore, eq.2 can be written in the following form

$$h_{m1}dm_1 = u_1 dm_1 + m_1 du_1 + p_1 dV_1, \tag{3}$$

Where u_1 is specific internal energy of gas in the cylinder, V_1 is a volume of cylinder, m_1 is amass of gas entering in the cylinder, quantities with index m relate to the manifold of pipeline.

We can express in equation 3 the values of enthalpy and internal energy through the product of temperature and heat capacity at constant pressure c_p and volume c_v, according to equation

$$c_p T_{m1} dm_1 = c_v T_1 dm_1 + c_v m_1 du_1 + p_1 dV_1 \tag{4}$$

The equation of state of the real gas in the working cavity is written as

$$p_1 V_1 = z m_1 R T_1 \tag{5}$$

Where R is the gas constant and compressibility factor z determines the extent of non-ideality of the working fluid. Numerical calculation of the compressibility factor z for air or nitrogen, accomplished in the approach proposed in ref.[7] for the real gases, showed that in the pressure range considered (P=0.1…..3.5MPa) and temperature range T ≥210k the assumption that z = 1 is highly accurate, and the gas can be treated as ideal.

Substituting in equation 4 the value $m_1 dT_1$,taken from equation 5 with the approximation z=1, and using the common notation $\frac{c_p}{c_v} = k$ and $c_p - c_v = R$ where k is the adiabatic exponent, after simple transformations one can obtain the following expression

$$kRT_{m1} dm_1 = V_1 dp_1 + k p_1 dV_1 \tag{6}$$

We can then replace the mass of air dm_{m1}, entering the volume V_1 during time dt in equation 6, by the corresponding value of the consumption function G_1 (define as $dm_{m1} = G_1 dt$) and then express the equation relative to pressure

$$dp_1 = \frac{k G_1 R T_{m1} dt}{V_1} - k p_1 \frac{dV_1}{V_1} \tag{7}$$

We can also determine the function G_1 that describes the gas consumption. For this purpose we can find the parameters from Bernoulli's equation. According to Bernoulli's equation;

$$dh + vdv = 0$$

For isentropic flow this can be written as

$$\frac{dp_1}{\rho_1} + vdv = 0 \tag{1a}$$

If the flow is incompressible than $\rho = \text{constant}$

Then equation (1a) on integration fields

$$\frac{1}{\rho_1} \int dp_1 + \frac{1}{2} \int dv^2 = \text{constant}$$

$$\frac{p_1}{\rho_1} + \frac{1}{2}v^2 = \text{constant} \tag{2a}$$

As we know when the flow is isentropically decelerated to zero velocity, the resultant pressure is the stagnation pressure. Therefore, when v=0,

$p_1 = p_{m1}$,

$\rho_1 = \rho_{m1}$,

Thus, constant in equation (2a) , $\frac{p_{m1}}{\rho_{m1}} = \text{constant}$

Therefore, $\frac{p_1}{\rho_1} + \frac{1}{2}v^2 = \frac{p_{m1}}{\rho_{m1}}$,

In incompressible flow since,

$\rho_{m1} \approx \rho_1 = \text{constant}$,

$p_1 + \frac{1}{2}\rho_1 v^2 = p_{m1}$,

But for adiabatic,

$$h_0 = c_p T_0 = \frac{k}{k-1} R T_0 , \tag{3a}$$

$RT_0 = \frac{p_{m1}}{\rho_{m1}}$, $h_0 = \frac{k}{k-1}\frac{p_{m1}}{\rho_{m1}}$, $h + \frac{1}{2}v^2 = h_0$, $h = c_p T = \frac{k}{k-1}RT$, $kRT = \alpha^2$ And $RT = \frac{p_1}{\rho_1}$,

$$h = \frac{\alpha^2}{k-1} = \frac{k}{k-1}\frac{p_1}{\rho_1} , \tag{4a}$$

From equation (4a) & (3a),

$$\frac{\alpha^2}{k-1} + \frac{1}{2}v^2 = \text{constant} , \tag{5a}$$

At $T = 0$, $h = 0$, $v = v_{\max}$, From equation (3a),

$$h_0 = \frac{1}{2}v_{\max}^2 , \tag{6a}$$

At $v = 0, \alpha = \alpha_0$, From equation (5a),

$$\text{constant} = h_0 = \frac{\alpha_0^2}{k-1} , \tag{7a}$$

By equation (5a),(6a)and (7a),

$\frac{\alpha^2}{k-1} + \frac{1}{2}v^2 = \frac{1}{2}v_{\max}^2 = \frac{\alpha_0^2}{k-1} = h_0$,

Therefore,

$$\frac{k}{k-1}\frac{p_1}{\rho_1} + \frac{1}{2}v_1^2 = \frac{k}{k-1}\frac{p_{m1}}{\rho_{m1}} , \tag{8a}$$

Then from (8a) we have,

$$\frac{k}{k-1}\frac{p}{\rho}+\frac{1}{2}v^2=\text{constant},$$

Of the adiabatic braked stream(parameters of braking).in this case we will put the velocity of a stream equal to zero over the area of inlet valve into Bernoulli's, then we have

$$\frac{k}{k-1}\frac{p_1}{\rho_1}+\frac{1}{2}v_1^2=\frac{k}{k-1}\frac{p_{m1}}{\rho_{m1}} \quad (8)$$

Now one can find the velocity of air entering the first cylinder from Eq. 8

$$v_1=\sqrt{\frac{2k}{k-1}\left(\frac{p_{m1}}{\rho_{m1}}-\frac{p_1}{\rho_1}\right)} \quad (9)$$

From the adiabatic equation

$$\frac{\rho_1}{\rho_{m1}}=\left(\frac{p_1}{p_{m1}}\right)^{\frac{1}{k}} \quad (10)$$

One can find the value of air density ρ_1 in the working cylinder,

$$\rho_1=\rho_{m1}\left(\frac{p_1}{p_{m1}}\right)^{\frac{1}{k}},$$

By substituting expression (10) into formula (9) we have,

$$v_1=\sqrt{\frac{2k}{k-1}RT_{m1}\left[1-\left(\frac{p_1}{p_{m1}}\right)^{\frac{k-1}{k}}\right]}, \quad (11)$$

The gas consumption function can be defined as $G_1=v_1\rho_1\alpha_1\mu_1$, where a_1 is the area of the cross section of the inlet valve; μ_1 is a coefficient of gas consumption through the inlet valve. Let us substitute the velocity of gas, determined by formula 11, into the expression for the gas consumption. Then we obtain

$$G_1=\rho_1\alpha_1\mu_1\sqrt{\frac{2k}{k-1}RT_{m1}\left[1-\left(\frac{p_1}{p_{m1}}\right)^{\frac{k-1}{k}}\right]} \quad (12)$$

If we substitute the gas density in the cylinder from equation (10) to (12) we can derive

$$G_1=\rho_{m1}\left(\frac{p_1}{p_{m1}}\right)^{\frac{1}{k}}\alpha_1\mu_1\sqrt{\frac{2k}{k-1}RT_{m1}\left[1-\left(\frac{p_1}{p_{m1}}\right)^{\frac{k-1}{k}}\right]}=$$

$$G_1=\alpha_1\mu_1\sqrt{\frac{2k}{k-1}\rho_{m1}^2RT_{m1}\left[\left(\frac{p_1}{p_{m1}}\right)^{\frac{2}{k}}-\left(\frac{p_1}{p_{m1}}\right)^{\frac{k+1}{k}}\right]}=$$

$$G_1=\alpha_1\mu_1p_{m1}\sqrt{\frac{2k}{k-1}\frac{1}{RT_{m1}}\left[\left(\frac{p_1}{p_{m1}}\right)^{\frac{2}{k}}-\left(\frac{p_1}{p_{m1}}\right)^{\frac{k+1}{k}}\right]}$$

As a result, the function G_1 of the gas consumption from the unlimited volume (manifold) can be determined by the formula [8–11]

$$G_1 = \alpha_1 \mu_1 p_{m1} \sqrt{\frac{2k}{k-1} \frac{1}{RT_{m1}} [(\frac{p_1}{p_{m1}})^{\frac{2}{k}} - (\frac{p_1}{p_{m1}})^{\frac{k+1}{k}}]} \quad (13)$$

Where p_{m1} and T_{m1} are the gas pressure and temperature in manifold p_{m1}..

Note that the losses of gas pressure in the pipeline and local resistances are taken into account by introducing the coefficient of consumption μ [9, 11], which besides takes into account the compression of flow stream during exhaust, the speed of the gas as it approach the aperture and other factors. More often, this coefficient of consumption is determined experimentally or with the help of approximations. When the flow of gas occurs within a short span of a pipe at high velocity, the exhaust process is considered to be adiabatic, and it is possible to use formula 13. The process of compression of the gas in a working cavity is described by Eq. 6, which we can solve simultaneously with Eq. 13. As a result it is possible to determine the pressure p_1 in cylinder as a function of time. We must note that this process cannot be described by means of elementary polytropic process with a constant parameter, n. In reality, it follows from formula 13, that the consumption G_1 is a function of the pressure ratio, in which the numerator is the pressure of the medium into which gas flows, and the denominator is the pressure of the medium from which this gas moves.

We will present formula 13 for gas flow from a pipe in a more convenient form [9]

$$G_1 = \frac{\alpha_1 \mu_1 k p_{m1} \Phi(\sigma_1)}{\sqrt{RT_{m1}}} \quad (14)$$

Where, $\Phi(\sigma) = \sqrt{(\sigma)^{\frac{2}{k}} - (\sigma)^{\frac{k+1}{k}}}$; $\sigma_1 = \frac{p_1}{p_{m1}}$; $K = \sqrt{\frac{2k}{k-1}}$

In order to find the maximum of the gas consumption factor $\Phi(\sigma)$,let us set its derivative to zero, which one can be written as

$$\frac{2}{k}\sigma^{\frac{2}{k-1}} - \frac{k+1}{k}\sigma^{\frac{1}{k}} = 0,$$

From which one can obtain the critical ratio of pressures

$$\sigma^* = [\frac{2}{k+1}]^{\frac{k}{k-1}} \quad (15)$$

It is necessary to distinguish between two regimes of the flow; subcritical, when the consumption function G_1 is determined by the formula 13, and supercritical, at which the maximum critical consumption of gas G^* is obtained after substitution of the critical ratio of pressures from Equation 15 into Equation 14[9]

$$G^* = \alpha_1 \mu_1 p_{m1} \Phi(\sigma^*) \sqrt{\frac{2k}{(k-1)RT_{m1}}} \quad (16)$$

Where $\sigma^* = 0.5282$ and for k=1.4

If we substitute the value of the critical gas consumption G^* into Equation 7 instead of G_1, we can obtain the equation that describes the process during the supercritical mode in a cavity of changing volume

$$dp_1 = \frac{kG^* RT_{m1}}{V_1} dt - kp_1 \frac{dV_1}{V_1} \qquad (17)$$

The analysis of Equation. 7, 13, 16 and 17 shows, that the process of change of gas state in a cylinder being filled, both for variable and for constant volume does not coincide with any one of the elementary thermodynamic processes, which occur with a constant polytropic exponent. The process can be described using the variable polytropic exponent n, which, in the beginning of the process, equals the adiabatic exponent, and then monotonically decreases,

$$n = 1 + [\frac{\sigma_0 (k-1)}{\sigma}] \qquad (18)$$

Where, $\sigma_0 = \frac{p_0}{p_{m1}}$; At σ = 1, i. e. at the end of process, the value of polytropic exponent asymptotically approaches the isothermal exponent n =1.

Change of the gas state by polytropic process (with a constant polytropic exponent) is possible in a working cylinder when there is a constant mass of gas. Some examples include the internal combustion engines after closure of the inlet valve, in flap-valve pneumatic motors, where the chamber is completely isolated by plates from the inlet and outlet ports, and in compression-driven devices and accumulators, when there is expansion of the compressed gas. In the case of variable gas quantity in a cylinder, it is necessary to investigate the process by the energy balance stated in equ (6).

Thus, we can obtain the differential equation for determination of the gas pressure during the filling of cavity 1 in a general form, by substituting the value of gas consumption G_1 from expression 14 into Equation (7)

$$\frac{dp_1}{dt} = \frac{K\alpha_1 \mu_1 k p_1 \Phi(\sigma_1)\sqrt{RT_{m1}}}{V_1} - \frac{kp_1}{V_1}\frac{dV_1}{dt} \qquad (19)$$

Let's consider the expansion stage of the pneumatic cylinder operation; the process of gas expansion during the movement of piston from point B to point D. We will describe this polytropic process with a parameter 1 ≤ n ≤1.4 , that allows us to take into account, to a first approximation, the possible process of heat exchange in the pneumatic cylinder.

Then the pressure in the cylinder can be determined by the following expression,

$$p_1 = p_{1B} (\frac{xB - ts}{x - ts})^n \qquad (20)$$

Where xB is the corresponding distance traveled by the piston to arrive at position B (see Fig. 1); x is the current piston position; ts is a thickness of the piston; p_{1B} is the gas pressure at the beginning of the expansion stage.

As a result it is necessary to solve the next equation for determining the dynamic characteristics of the piston motion in the pneumatic cylinder

$$\left.\begin{array}{l} 0, x = xA \\ \dfrac{d^2x}{dt^2} = \dfrac{p_1 s_1 - p_2 s_2 - F}{M} \\ 0, x = xD \end{array}\right| , xA < x < xD \qquad (21)$$

Thus the total set of the equation, describing the dynamics of the pneumatic cylinder can be presented as

$$\left|\begin{array}{l} \dfrac{dp_1}{dt} = \dfrac{\mathrm{K}\alpha_1\mu_1 k p_1 \Phi(\sigma_1)\sqrt{RT_{m1}}}{V_1} - \dfrac{kp_1}{V_1}\dfrac{dV_1}{dt} \\ if \;\; x < xB; \\ p_1 = p_{1B}(\dfrac{xB - ts}{x - ts})^n \text{ if } x \geq xB; \end{array}\right.$$

$$\left.\begin{array}{l} 0, x = xA \\ \dfrac{d^2x}{dt^2} = \dfrac{p_1 s_1 - p_2 s_2 - F}{M} \\ 0, x = xD \end{array}\right| , xA < x < xD$$

For the numerical solution of the derived differential parameters, describe the physical sizes and operating conditions of the pneumatic cylinder.

4. MODELING OF THE OPERATION :

A computer program in MATLAB-SIMULINK and modeling in AUTOMOTIVE STUDIO(PNEUMATIC SIMULATION) has been developed for calculation of the dynamic parameters of the pneumatic cylinder by using the derived equations.

The following parameters and initial conditions were chosen for the simulation.

working fluid-air/nitrogen, R=298.8 J/kg-k ; adiabatic k=1.4 ; polytropic n=1.25exponents; initial pressure in the cylinder=.942bar ; pressure in the manifold=7bar ; gas temperature in the manifold=300k ; loading force=2N ; coefficient of friction=.15 ; gas consumption coefficient in cylinder=.7 ; diameter of input and output valve=6mm ; diameter of the piston=23mm ; -initial piston speed=0 ; -mass of piston=300gm ; piston stroke=50mm ; distance xA=2mm,xB=4mm,xC=47mm,xD=48mm ;

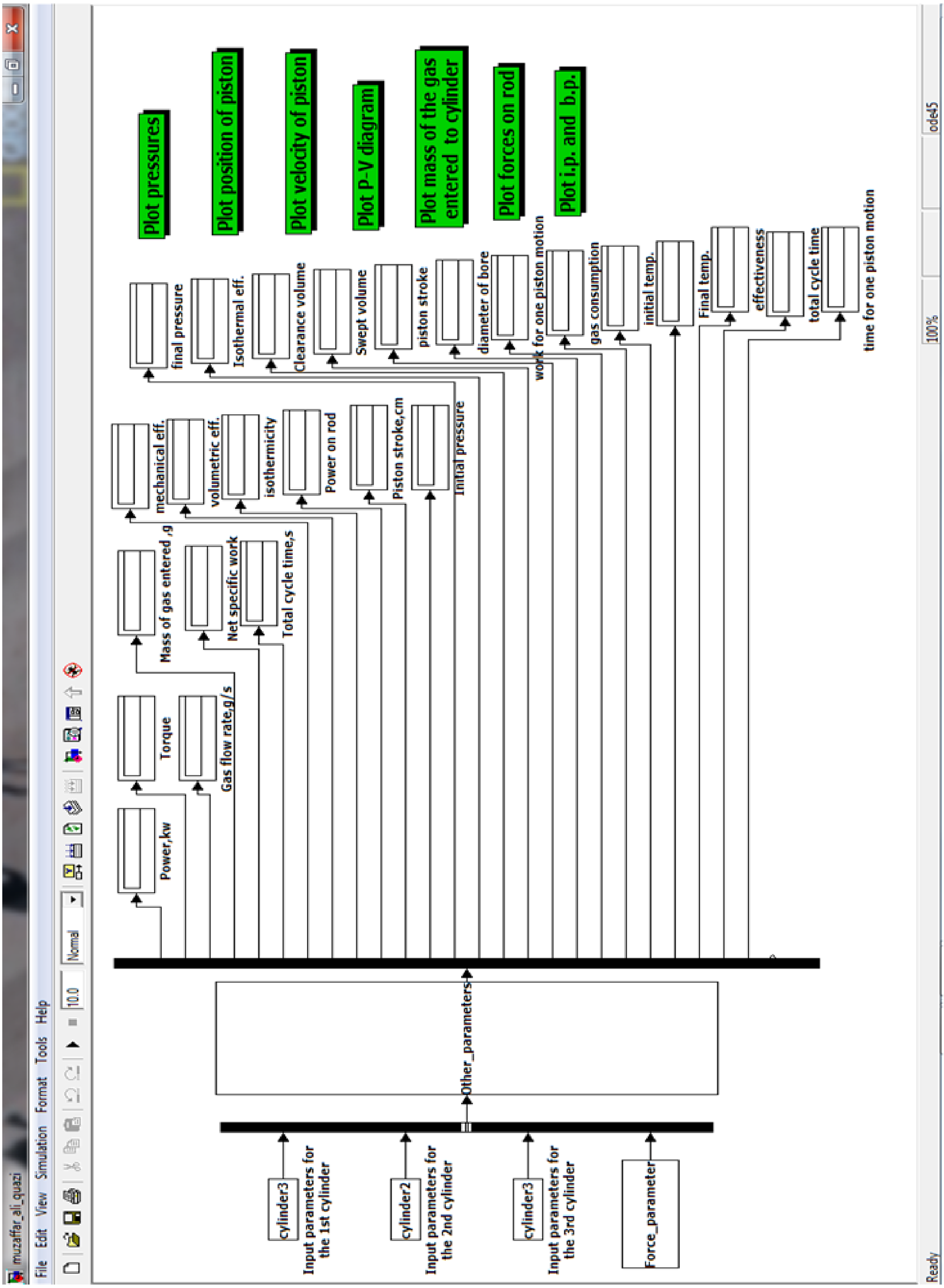

Fig:- 4 interface of the computer program for calculating the dynamical and operational characteristics of pneumatic cylinder

The result of numerical modeling of the gas pressure in the cylinder as the function of the piston position,pressure,speed and time are presented in fig.5 ,6 and 7.

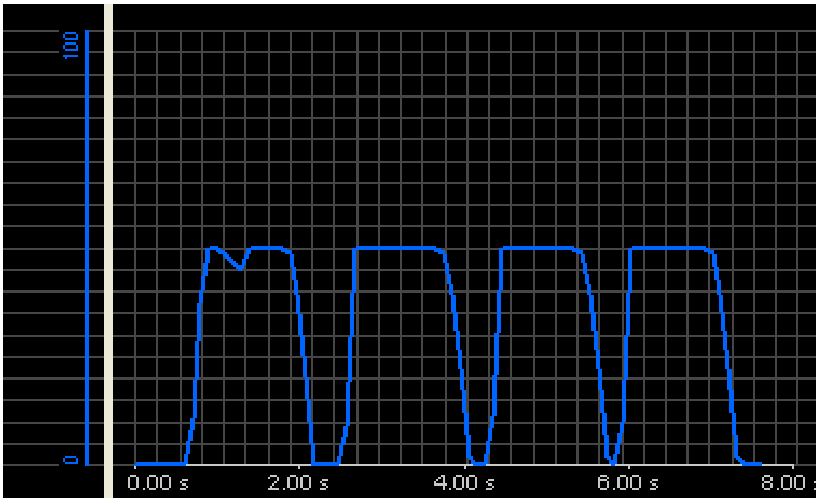

Fig 5.Displacement/time curve

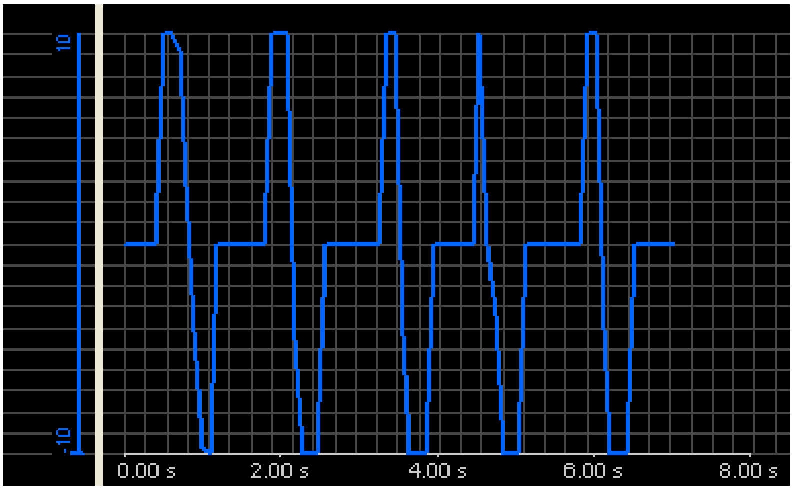

Fig 6. Speed/time curve

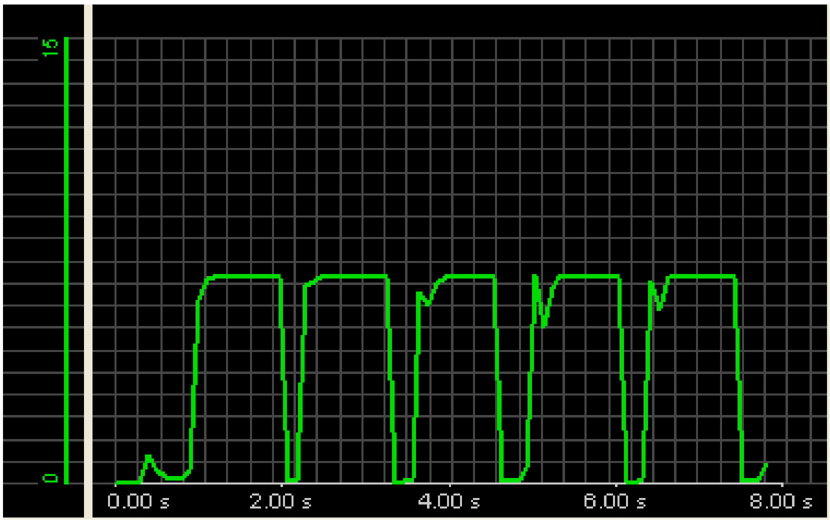

Fig 7.presure vs time curve

Now, let's analyze the physical process, which occur within the cylinder, in accordance with the numerical results obtained.

Until the force arising from the pressure in cylinder exceeds the force of friction and force of useful resistance. During this period there is filling of the constant volume of cylinder, and the piston remain motionless at point A.On the graph of pressure as a function of the piston motion this process is displayed by a practically vertical segment near the value of x=2mm.

The following stage of the pneumatic cylinder operation is the continuing process of filling of cylinder, for an already moving piston from point A up to B, the pressure in cylinder, which was establish after filling the cylinder while the piston was motionless, falls to 6bar.

After the piston passes point B, the outlet valve of exhaust is open, the inlet valve of the working cylinder is closed, and the piston continues to move to the right under the action of pressure force in cylinder, during this period pressure in cylinder decreases.Further movement of the piston there corresponds to the gas expansion with the closet inlet valve and open outlet valve. The final pressure in the cylinder (.924bar) accurately coincides with the pressure in the exhaust pipeline.

Note that the mathematical model developed here allows us to calculate the time dependence of the piston and velocity of the piston for pneumatic cylinder. The PV diagram is presented for the process under consideration, using the pressure dependences obtained for piston motion in the cylinder.

5. METHODOLOGY :

Data for each piston was entered into MATLAB Simulink computer simulation and AUTOMOTIVE STUDIO software, of pneumatic engine operation to generate flow velocities .These inputs include: the mass of the piston and all moving parts (piston, rod, rack, gear, etc.), useful areas of each piston, input and output pressures, gas temperature at inlet, and required piston velocity. The general equation of the piston motion can be written as follows:

$$M\frac{d^2x}{dt^2} = p_1 s_1 - p_2 s_2 - F$$

The MFR required for cylinder can be calculated with the following equation .

$$\dot{M} = \rho V A$$

Power produced by the engine:-

Power = $N = \frac{n}{n-1} Q_1 p_2 V_1 \eta_i (\frac{p_1}{p_2})^{\frac{n-1}{n}}$

Work per cycle = $\frac{n}{n-1} p_2 V_1 \eta_i [(\frac{p_1}{p_2})^{\frac{n-1}{n}} - 1]$

$$\eta_{vol} = 1 - \frac{V_c}{V_s}[(r_p)^{\frac{1}{n}} - 1]$$

6. RESULTS:

The main results obtained for the three cylinder pneumatic engine are as given below:-

Power(b.p.):-0.683hp

Power(I.p.):-0.853hp

Volumetric efficiency(ηvol):-0.72

Isothermal efficiency(ηiso):-0.89

What is cost per CFM(ft3per minute)?

A Good Approximation

Typical Compressor produces 4 CFM per 1 Hp

1 Hp = 0.746/0.9 = 0.829kW

Therefore, 1 CFM = 0.207kW

4rs/kw-hr, 1 cfm = .828rs/hr

It takes 0.69rs/km.

7. CONCLUSION:

The three cylinder engine is fabricated, and mathematical model developed in this paper allows one to accomplish the numerical simulation of the working process and to determine the main dynamic characteristics of the pneumatic cylinder. By using the results of the numerical calculations, the analysis of the particulars of changes of gas pressure in cylinder can be accomplished. The subsequent stages of pneumatic cylinder operation can also be studied to calculate the main operational characteristics.

NOMENCLATURE:	
A_{sp}	specific useful work, kJ/kg;
a_1, a_2	Cross sections of inlet and outlet valves, m^2;
B	Coeffient of friction;
C_v	Heat capacity at constant volume J / Kg. K ;
C_p	Heat capacity at constant pressure J / Kg. K;
D	Diameter of piston, m;
Dd_1,Dd_2	Rod diameters in cylinder, m;
d_1, d_2	Diameters of input and output valves, m;
F,F_L,F_{FR}	Resistance, loading and friction forces, N;
$\Phi(\sigma)$	Gas consumption factor;
G_1, G_2	Gas consumption functions for cylinder, kg/s;
G_{N2}	Specific gas consumption Kg / Kw-hr
H_{m1}	Enthalpy of gas entering the cylinder, J;
h_{m1}	Specific enthalpy, J/kg;
K	Adiabatic exponent;
M	Mass of piston, kg;

α_1, α_2	Coefficients for gas consumption in cylinder;
m1	Mass of gas entering the cylinder, kg;
N	Working power of cylinder, kW; n — polytropic exponent;
F	Frequency of operation, rot/min; p_1, p_2 — pressures in cylinder, MPa;
p_{m1}, p_{m2}	Pressures in 1st and 2nd manifolds, MPa;
p_{10}, p_{20}	Initial pressures in cylinder, MPa;
R	Gas constant J / Kg.K
Q_{m1}	Thermal energy entering the cylinder, J;
Q_{m2}	Thermal energy, which is removed from cylinder, J;
ρ_1, ρ_2	Gas densities in cylinder, kg/m^3;
ρ_{m1}, ρ_{m2}	Gas densities in 1st and 2nd pipelines, kg/m^3;
S_1, S_2	Useful areas of piston for cylinder, m^2;
σ_0, σ_1, σ_2, σ_*	Ratios of pressures;
T_{m1}, T_{m2}	Gas temperatures in 1st and 2nd manifolds, K;
T_1, T_2	Gas temperatures in cylinder, K;
U_1, U_2	Internal energies of gas in cylinder, J;
u_1	Specific internal energy of gas in the cylinder, J/kg;
V_1, V_2	Volumes of cylinder, m^3;
υ_1	Velocity of air entering the 1st cylinder ,m/s;
υ_{m2}	Velocity of air exhaust from thecylinder, m/s;
W_1, W_2	Works for gas expansion in cylinder, J;
X	Current piston position, m;
xA,xB,xC,xD	Distances (see Fig. 1), m; xE — piston stroke, m;
Z	Compressibility factor

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