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COMPUTER VISUALISATION AND SIMULATION OF FAST CYCLIC HYDRAULIC ACTUATOR DYNAMICS

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ABSTRACT

In the paper is presented complete mathematical model of generalised fast cyclic hydraulic actuator dynamics, based on Rieman's equations in addition with corresponding external and internal boundary conditions. Fast cyclic hydraulic actuator can be assumed as two serially connected compressible fluid flows controlled by supply and return variable flow restrictors enclosed in control servo-valve and separated by actuator piston. Exposed dynamic model includes several physical effects: fluid viscosity and compressibility, compression and expansion wave propagation in direct and reverse actuator modes, geometric and flow asymmetry, actuator equivalent inertia including remote mechanism of aircraft control surfaces, potential external load and arbitrary servo-valve control input. Solution of presented mathematical model is evaluated by using method of characteristics. Corresponding boundary conditions assumes actuator piston position as a movable boundary between direct and reverse part of actuator chamber. Main advantage of presented method is that corresponding computer package enables actuator simulation for its arbitrary states of function by using the same computational procedure.

INTRODUCTION

Computer simulation of fast cyclic hydraulic servo-actuator is actual in analysis of its performances. For very fast cyclic actuators its state is characterised of strongly expressed wave effects and high gradients of velocity and pressure changes along the fluid streamline. These effects are placed mostly in the source part of actuator. For slow actuators its dynamics and simulation procedure tends toward stationary state, which can be also evaluated by simulation of conventional mathematical models. In the paper are presented results of computer simulation and visualisation of generalised fast cyclic hydraulic actuator.

Here is presented complete mathematical model including transient state of fast cyclic hydraulic actuator, which can be described and determined by the following effects:
- ambiguity of the initial pressure conditions in actuator chambers;
- hydraulic static pressure drop and surge caused by propagation of damped hydraulic expansion and compression waves;
- pressure surge in the moment of activating reverse direction of piston motion.

For fast hydraulic actuators it is of interest to involve the effects of compression and expansion wave propagation along the inlet and outlet pipelines and actuator chamber. Reverse propagation of expansion wave produces corresponding pressure drop, depending on the pressure drop on the actuator servo valve. This effects increases with velocity of control servo-valve throttle and can be suppressed by its partial closing. In difference with the classic hydraulic actuator modelling in which the fluid is assumed as incompressible or quasi static compressible, wave effects introduces pressure disturbances during its propagation. These effects travel very fast and they are slightly damped by fluid viscosity. For digitally controlled actuators they can not be neglected. Each of the mentioned effects produces also corresponding actuator operational time delay, which is limiting factor of its cyclic velocity. Dominant influence on actuators time delay is caused by effect of fluid quasi-static compressibility. Pressure drop caused by propagation of expansion waves in the source pipeline of fast cyclic hydraulic actuator produces possible anomalies in its function.

ACTUATOR MATHEMATICAL MODELLING

Described effects can be evaluated by partial differential equations of continuity and momentum with additional fluid compressibility law for one dimensional flow. Effect of fluid viscosity can be entered as a friction between fluid streamline and pipeline wall. Local viscous effects at control servovalve and flow inlet and outlet of actuator cylinder represented by corresponding pressure lose coefficients are involved as boundary conditions. Method of characteristics is very applicative for solving this problem formulation. Corresponding model form is presented in the form of algebraic linear equations. Wave propagation is more visible in the solution form of characteristics and corresponds to the real physical effects. Equations of continuity and momentum for one dimensional fluid flow including the effects of wall friction are presented in the well known form of Rieman's partial differential equations of propagation small waves trough compressible medium. Rieman's partial differential equations including the wall friction effects and fluid compressibility law are given in the following form:

$$\begin{aligned}
&\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}=-c\frac{\partial \mu}{\partial x}-\frac{1}{2}u^{2}\frac{\partial \xi}{\partial x}\\
&\frac{\partial \mu}{\partial t}+u\frac{\partial \mu}{\partial x}=-c\frac{\partial u}{\partial x}\\
&\mu=\int_{\rho_0}^{\rho}\sqrt{\frac{dp}{d\rho}}\frac{d\rho}{\rho}
\end{aligned} \tag{1}$$

where are u fluid velocity trough the pipelines, p fluid static pressure, ρ fluid density, c velocity of sound, ξ coefficient of pressure loses for unit length, x co-ordinate along streamline and t time. Presented partial differential equations can be written in the form of characteristics as:

$$\begin{aligned}
&\frac{\delta_{+}P}{\delta t}=\frac{\delta_{+}}{\delta t}(\mu+u)=\left[\frac{\partial}{\partial t}+(u+c)\frac{\partial}{\partial x}\right](\mu+u)=-\frac{1}{2}u^{2}\frac{\partial \xi}{\partial x}\\
&\frac{\delta_{-}Q}{\delta t}=\frac{\delta_{-}}{\delta t}(\mu-u)=\left[\frac{\partial}{\partial t}+(u-c)\frac{\partial}{\partial x}\right](\mu-u)=\frac{1}{2}u^{2}\frac{\partial \xi}{\partial x}
\end{aligned} \tag{2}$$

where the characteristics P and Q are given in the following form respectively:

$$\frac{dx^{+}}{dt}=u+c \qquad \frac{dx^{-}}{dt}=u-c \tag{3}$$

For turbulent flow pressure losses can be accepted in the form:

$$\frac{\partial \xi}{\partial x}=\lambda\,\mathrm{sgn}(u)$$

where λ is coefficient of pressure loses per unit length of streamline. Corresponding equations of characteristics are given in the following form:

$$\begin{aligned}
&\frac{\delta_{+}P}{\delta t}=\frac{\delta_{+}}{\delta t}(\mu+u)=\left[\frac{\partial}{\partial t}+(u+c)\frac{\partial}{\partial x}\right](\mu+u)=-\frac{1}{2}\lambda u^{2}\mathrm{sgn}(u)\\
&\frac{\delta_{-}Q}{\delta t}=\frac{\delta_{-}}{\delta t}(\mu-u)=\left[\frac{\partial}{\partial t}+(u-c)\frac{\partial}{\partial x}\right](\mu-u)=\frac{1}{2}\lambda u^{2}\mathrm{sgn}(u)
\end{aligned} \tag{4}$$

For the case of small fluid compressibility, fluid density ρ and variable μ can be presented in linearised mathematical form as follows from the third of relations (1):

$$\rho=\rho_0\left(1+\frac{p}{\chi}\right) \qquad \mu=c\ln\left(1+\frac{p}{\chi}\right)\cong\frac{p}{c\rho_0} \tag{5}$$

where are ρ_0 fluid density for zero fluid static pressure and χ bulk module.

For the case that the fluid velocity u is small in ratio to the velocity of sound c and any of corresponding derivatives, partial differential equations (1) can be presented in linearised mathematical form, which is of interest for better understanding of given problem formulation:

$$\begin{aligned}
&\frac{\partial u}{\partial t}=-\frac{1}{\rho_0}\frac{\partial p}{\partial x}\left(-\frac{1}{2}u^{2}\frac{\partial \xi}{\partial x}\right)\\
&\frac{1}{c^{2}\rho_0}\frac{\partial p}{\partial t}=-\frac{\partial u}{\partial x}
\end{aligned} \tag{6}$$

Additional term in brackets is also small, but it can be assumed as the physical presentation of pressure loses caused by the wall friction effects. Neglecting the term in bracket, partial differential equations (6) describes the hydraulic impact effects. If exists significant velocity change during time in any point of the fluid streamline, it will cause the corresponding pressure drop along the streamline in the same point.

ACTUATOR SEPARATE FLOW MODELLING

Any direction change of actuator piston motion produces pressure discontinuity which is caused by inversion of fluid flow as result of connection change between supply and return pipelines and both actuator chambers. In the moment of change fluid flow direction each of actuator chambers change connections with system pump and return pipeline, producing corresponding discrete changes of pressure and velocity in actuator chambers. Possible pressure drop or surge is also caused by geometric asymmetry of servo valve.

Complete actuator system (points B through I), connected into hydraulic system into points O (supply) and K (return) to the power source hydraulic system, is presented on figure 1. Static pressure in supply and return pipelines are presented as p_s and p_0, respectively.

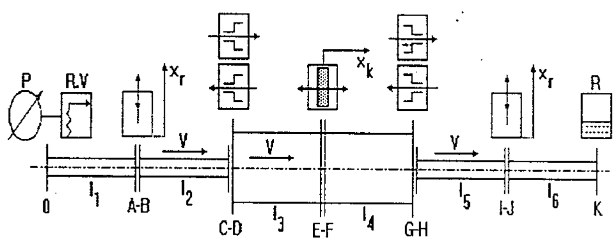

Fig.1

Full problem formulation includes equations of characteristics with corresponding boundary and initial conditions for each of assumed system subdomains. These domains correspond to the inlet and outlet pipelines, supply and return fluid flow sections between control servovalve and actuator piston. Wave effects of return flow are very small.

As a boundary condition at point O can be assumed nominal supply pressure of hydraulic system. In points A and B corresponding boundary conditions are defined as flow continuity between A and B, and pressure loss between A and B caused by control inlet servovalve throttle. In points C and D boundary conditions are the same type as for the points A and B with only difference in pressure loss caused by fluid viscous effects at cylinder flow inlet and outlet. At point E are defined boundary conditions of static pressure and piston velocity equal to the fluid velocity. In points I and J corresponding boundary conditions are defined as flow continuity between I and J, and pressure loss between I and J caused by outlet control servovalve throttle. In points G and H boundary conditions are the same type as for the points C and D with only difference in pressure loss caused by fluid viscous effects at cylinder flow outlet. At point F are defined boundary conditions of static pressure and piston velocity equal to the fluid velocity in addition of piston equilibrium ordinary differential equation. In point K boundary conditions are assumed as zero static pressure of return fluid flow.

Initial conditions are defined at each of domains. Initial conditions of fluid flow are assumed as zero fluid velocity. Initial conditions of fluid static pressure are assumed as supply static pressure (O-A) and cylinder static supply pressure (B-E and F-I). Initial and boundary conditions compatibility is completed for the exposed system model.

BOUNDARY CONDITIONS OF COMPLETE ACTUATOR SYSTEM

Boundary conditions are defined at seven points: point of separation fluid flow to actuator, input control valve, fluid inlet into actuator chamber, movable piston, fluid outlet from actuator chamber, output control servo-valve and point of connection to the return pipeline. Corresponding boundary conditions are defined in the form of continuity and Bernoulli equations or piston momentum equation in addition with pressure and velocity conditions. Left boundaries are determined with Q-characteristics, and right ones by P-characteristics. At the boundaries A&B through I&J corresponding values of μ and u must be determined by interpolation. At points E&F must be applied some of numeric integration methods for determination corresponding pressure value on the movable piston surface.

BOUNDARY CONDITIONS OF INLET AND OUTLET CONTROL SERVO-VALVE

Boundary conditions of fluid flow are given by the following algebraic non-linear equations on the changeably restrictor:

$$\begin{aligned} &\eta d x_r \sqrt{\frac{2}{\rho_0}(p_M - p_N)} = Q \\ &Q\rho_0 = \rho_M A_M u_M = \rho_N A_N u_N \end{aligned} \tag{7}$$

where index M and N denotes corresponding points on both sides of restrictor. It is of interest to establish the relation between derivatives of equation (7), because it defines corresponding pressure derivative along the streamline as the result of arising hydraulic impact. By derivation first of the equations (7) we gives:

$$\begin{aligned} &\frac{\partial Q}{\partial t} = \eta d \frac{\partial x_r}{\partial t}\sqrt{\frac{2}{\rho_0}(p_M - p_N)} + \\ &+ \frac{\eta d x_r}{\sqrt{2\rho_0(p_M - p_N)}}\left(\frac{\partial p_M}{\partial t} - \frac{\partial p_N}{\partial t}\right) = A\frac{\partial u}{\partial t} \end{aligned} \tag{8}$$

By substitution first of relations (6) into relation (8) and maximising its left side, we can give the following partial differential relation:

$$\eta d \frac{\partial x_r}{\partial t}\sqrt{\frac{2}{\rho_0}(p_M - p_N)}\bigg|_{M,N} = -\frac{A}{\rho_0}\frac{\partial p}{\partial x}\bigg|_{M,N} \tag{9}$$

Partial differential relation (9) shows the connection between input velocity of servo-valve throttle and pressure derivative along the fluid streamline. This relation proofs that existence of servo-valve throttle changes cause corresponding local changes of pressure, or produces the corresponding compression and expansion waves.

Instead of this formulation, we can use the following equivalent Bernoulli's equation with additional equation of continuity in the form:

$$\begin{aligned} &\frac{p_M}{\rho_M} - \left(\frac{A_M}{\eta_M d_M x_r}\right)^2 \frac{u_M^2}{2}\operatorname{sgn}(u_M) = \\ &= \frac{p_N}{\rho_N} + \left(\frac{A_N}{\eta_N d_N x_r}\right)^2 \frac{u_N^2}{2}\operatorname{sgn}(u_N) \\ &\rho_M A_M u_M = \rho_N A_N u_N \end{aligned} \tag{10}$$

BOUNDARY CONDITIONS ON THE FLUID INLET AND OUTLET OF ACTUATOR CHAMBER

Previous relations for actuator servo-valve can be applied for determination mentioned boundary conditions (10) in the following form of fixed restrictor:

$$\begin{aligned} &\frac{p_M}{\rho_M} - \frac{p_N}{\rho_N} \cong \eta_0\left(\frac{u_M^2}{2}\operatorname{sgn}(u_M) + \frac{u_N^2}{2}\operatorname{sgn}(u_N)\right) = \\ &= \eta_0(\operatorname{sgn}(u_N))\left[1 + \left(\frac{A_N}{A_M}\right)^2\right]\frac{u_N^2}{2}\operatorname{sgn}(u_N) = \\ &= \eta_0(\operatorname{sgn}(u_M))\left[1 + \left(\frac{A_M}{A_N}\right)^2\right]\frac{u_M^2}{2}\operatorname{sgn}(u_M) \\ &A_M u_M \cong A_N u_N \end{aligned} \tag{11}$$

where η_0 is corresponding coefficient of total pressure loses in this point of streamline.

BOUNDARY CONDITIONS OF ACTUATOR PISTON

Equations of piston equilibrium are given in the form:

$$p_M = p_N + \frac{k_a}{A_k} x_k + \frac{\delta_k}{A_k} u_k + \frac{m_k}{A_k} \dot{u}_k \qquad (12)$$
$$u_k = u_M = u_N$$

BOUNDARY CONDITIONS AT THE END POINTS

Boundary conditions in these points are caused by the performances and behaviour of hydraulic system power pump and its connection with relief valve. It means that corresponding boundary conditions can be defined in alternate form as follows:

$$p_0 = \begin{cases} p_{p\max}; 0 \le u < V_{max} \\ p(t); u = V_{max} \end{cases} \qquad (13)$$
$$u_0 = \begin{cases} u(t); p = p_{p\max} \\ V_{max}; p < p_{p\max} \end{cases}$$

The final form of boundary conditions in the point O are defined in the form:

$$p_0 = p_{p\max}; 0 \le u(t) \le V_{max} \qquad (14)$$
$$u_0 = V_{max}; p(t) \le p_{p\max}$$

In the point of connection to the return pipeline, boundary condition is:

$$p_k = 0 \qquad (15)$$

SIMULATION MODEL PERFORMANCES

It is more convenient to present given equations in nondimensional form by introducing the following nondimensional system co-ordinates and variables:

$$u = V_{max} \xi_u$$
$$x = L \xi_x$$
$$p = p_{max} \xi_p$$
$$T = \frac{L}{c}$$
$$t = T\tau$$

where V_{max} is maximal possible fluid velocity along the streamline caused by system pump, p_{max} is maximal possible fluid nominal static pressure and L is total length of the assumed streamline.

Computer package for fast cyclic hydraulic actuator simulation includes all important physical effects existing in transient state of its motion. As it is explained previously, for significant values of velocity and pressure gradients caused by fast servo-valve throttle wave effects expressed with mentioned travelling gradients and effects of wave reflection are of interest. It is well known that actual hydraulic cyclic actuators has the frequency range upon 100 Hz in correlation with the conventional ones, whose frequency is limited on 10 to 20 Hz. Presented results of computer simulation proofs that wave effects are of influence for the faster types of actuators. If we imagine some hypothetical types of compact actuators, whose frequency range are significantly grater than 100 Hz, we must include wave effects with its full influence. This produces completely different behaviour in compression with actual types of fast cyclic hydraulic actuators. Main difference is that wave reflection and corresponding velocity and pressure gradients are of same order as the frequency range of actuator input servo-valve control throttle. In that case quasi-static system behaviour practically degenerates and wave propagation effects takes a main role. As it is shown in the paper and attached references, inclusion of wave effects makes mathematical model more complicated because of inclusion partial differential equations instead ordinary once. Well known boundary conditions changes its relatively simplified formulation for the conventional models into very difficult procedure. It is commonly presented in iterative form including the interaction of two-connected boundaries with coupled parameters on its both sides which can be movable with arbitrary velocity and position.

In the paper [4] is presented more simplified one-sided actuator approximation whose outlet servo-valve part and reverse chamber were neglected by assuming quasi-static fluid flow with zero static pressure and constant fluid velocity along the stream-line. This approximation enables more simplified problem formulation because the effects of characteristics time delay between inlet and outlet actuator servo-valve parts are not present. Also we have only one point of control input. This approximation can be of interest, but given results are not of high quality. This fact is compensated with relatively simplified procedure of its feedback analysis and corresponding active control synthesis. In complete system model formulation, we must solve efficiently all of the mentioned problems, such as reverse of fluid flow in the return actuator subspace.

Density of nodal point distribution along the stream-line and along the simulation time depends of maximal velocity and pressure gradients. This fact can produce more computations, which usually increases cumulative computational errors. For usual actuator geometry, 200 points along the stream-line gives about 130.000 iterations for one second, or total 26 millions nodal points and corresponding calculations. In addition, each step of computation has the same number of medium nodal points. Assuming iterative procedure of nodal points calculation, we have to built very strictly procedure without any interpolations and extrapolations, except were they can not be overlaid. Method accuracy is very high because maximal errors of velocity and pressure distribution are very small. The corresponding differences between the results of one and two step iterative procedure for the case of 100 Hz actuator is less than $1x10^{-4}$ of maximal values. Corresponding results are shown on the figures 3.a and 3.b.

Time delay of characteristic is shown on the diagram on figures 5.d to 5.f. If the changes of tangent coefficients of characteristics are less than 2,5%, corresponding time delay is of same value or less. If the system discretisation assumes step of 0.5% unit time ratio, corresponding time delay compensation can be evaluated with no more than five iterations, which is very convenient result. But corresponding inversion of actuator connecting points produces more difficulties, which are explained on two following figures 2.a to 2.d.

Inversion of fluid flow between direct and reverse actuator modes is assumed as inversion of points of inlet and outlet servo-valve parts. Any other assumption of inverting points position is not correct, because it can produce non-existing high pressure and velocity gradients in reality. Compensation of characteristics time delay needs iterative procedure for calculation corresponding parameters in its domain between the points of inlet and outlet servo-valve zero throttle.

DISCUSSION OF GIVEN RESULTS

In accordance to assumed nondimensional units length ratio of total streamline length and corresponding nondimensional time ratio step of simulation loop, problem of numeric stability and convergence is of high interest. For equivalent time and co-ordinate step in assumed example whose co-ordinate discretisation is equal to 0.005 of unit, for simulation only one second of actuator activity is needed approximately 1.3×10^5 nondimensional time steps and loop execution iterations. For that very high number of iterations, numerical accuracy of calculation, spatially on the fixed and movable boundaries, is critical. For solving this problem, it is established corresponding interpolation or extrapolation procedure. In order to reduce computational oscillations around 'exact' solution, it is more effective to accept corresponding point of boundary calculations a little bit forward, or at least equal the greater time point from the both left and right boundaries. If we assume the middle point between the points on left and right boundaries, numerical convergence and corresponding time length of simulation process are limited on 4×10^3 iterations. This value is not acceptable for the practical purposes. If we assume maximal of these time values on the corresponding boundaries, accuracy is much better and it enables time simulation period of $1x\ 10^5$ iteration loops.

EXAMPLES AND GRAPHS

By using abilities of corresponding computer package actuator simulation results can be presented on several very impressive ways. Except 3-D diagrams, which enables presentation of each problem parameter as a 3-D surface depending on the streamline length and time axis, it is also possible to create the 'movies', which presents distribution of corresponding parameters along streamline axis for the assumed time sequences. In order to expose all of the relevant package performances, here are presented simulation results of some important cases of actuator and impute configuration. Shown diagrams correspond to the actuator input whose period is equal 40 nondimensional time units. This value corresponds to the actual actuator construction whose frequency input range is up to 100 Hz.

On figure 2 are presented 3-D results of actuator simulation. On figure 2.a is presented nondimensional velocity distribution along the unit nondimensional streamline and nondimensional time domain, shown on the figure 3.d (limited from 50.570 to 50.577 time units). For actual types of fast cyclic actuators 1 second is equal to 650 nondimensional time units. There are existing six domains of streamline, as it is previously explained. The first and the last ones correspond to the source and return pipelines of actuator. In these domains reverse fluid flow is very small and practically can be neglected. In the middle subdomains between inlet and outlet servo-valve parts, velocity distribution shows that reverse motion of the actuator piston and corresponding velocity distribution along streamline are of same order for the positive and negative control inputs in accordance to the figure 5.a Corresponding asymmetry of piston motion as results of direct and reverse actuator modes, which produces non-symetric piston position (different than zero) is presented for the both piston movable boundaries on figure 3.c. On figure 2.c are presented velocity distribution for the 12 characteristic actuator cross-sections including the movable boundaries of the both sides of the actuator piston. These results are calculated and shown for the each step of simulation loops, in difference with the diagram on figure 2.a which is calculated for the loops of 72 nondimensional time simulation steps. Exposed procedure is exclusively acceptable because of package and computer memory limits. Both of diagrams on figures 2.a and 2.c are calculated for two iteration steps. Corresponding difference between one and two iteration steps procedure of calculation nondimensional velocity at 12 characteristic cross-sections is presented on figure 2.e. Error of nondimensional velocity distribution is presented on 3-D diagram of figure 3.a. Corresponding maximal velocity distribution error is less than 1×10^{-4} of nondimensional unit velocity, or approximately 3×10^{-3} of existing velocity distribution. These results show that iteration procedure must be accepted.

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Everything shown on the figures 2.a, 2.c, 2.e and 3.a for nondimensional fluid velocity distribution is presented respectively on figures 2.b, 2.d, 2.f and 3.b for nondimensional static pressure distribution and corresponding pressure error calculations. Exposed conclusions can be applied on pressure distribution without any changes.

On figures 3.e and 3.f are presented respectively velocity and pressure nondimensional distribution along streamline at the final moment of simulation. Main difference with classical actuator modelling is nonexisting symmetric pressure drop at the inlet and outlet servo-valve parts. Velocity distribution practically degenerates to zero. On the figure 4.a is presented network of characteristic points, deflected in domain between inlet and outlet of actuator chamber as result of piston motion. Curving of presented mash is practically neglected in accordance with piston motion (figure 3.c), which is also shown for the all of streamline domains on the figure 4.b. On figures 4.c and 4.e is shown nondimensional velocity distribution in several sequences in the initial transient phase of actuator function. Corresponding nondimensional pressure distribution is shown respectively on the figures 4.d and 4.f. Significant difference with quasi-static actuator behaviour as result of classic modelling approach is visible on the mentioned diagrams.

On figure 5.a is presented diagram of assumed control servo-valve input throttle law for the actuator example whose results are shown on the all of exposed diagrams. On figure 5.b is presented fluid nondimensional velocity distribution during time for 12 characteristic cross-sections for each of time step calculation loops. On the other diagrams on figure 5 are presented nondimensional pressure distribution (figure 5.c) and nondimensional velocity distribution for different time simulation period (figures 5.d, 5.e and 5.f). Mentioned last three diagrams are short time sequences, because they explicitly show the wave travelling effects along streamline during time in transient actuator state. If the simulation period increases than any of the step velocity changes becomes smaller visible, because the velocity values also increases. Velocity step distribution is also visible on figure 5.b. These diagrams shows that the same computational procedure (and computer package) are used for the short and long time period simulation without any accuracy problems.

CONCLUSION

Computer simulation of fast cyclic hydraulic servo-actuator is very effective tool in procedure of its synthesis. Effect of fluid compressibility introduces additional wave propagation problems, which can be solved by some of numerical methods. For simulation purposes, it is recommended method of characteristics for system modelling. Presented results of actuator simulation proofs that exposed system model and corresponding computer package enables its simulation for arbitrary actuator configuration and states of function.

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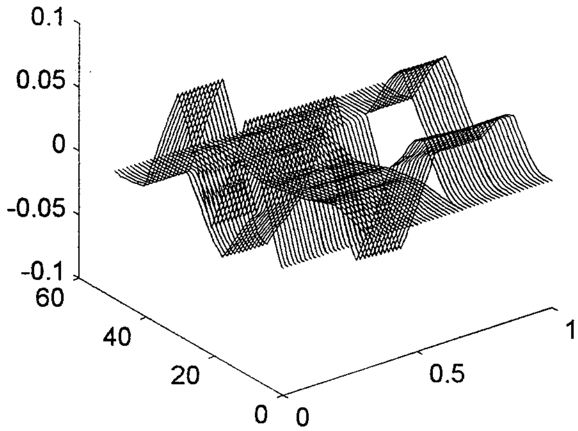
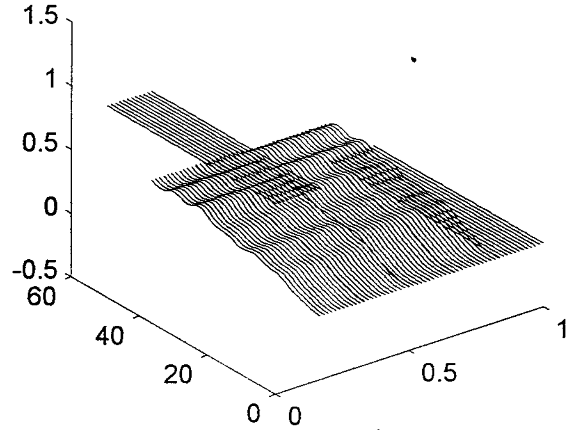


Fig.2.a.



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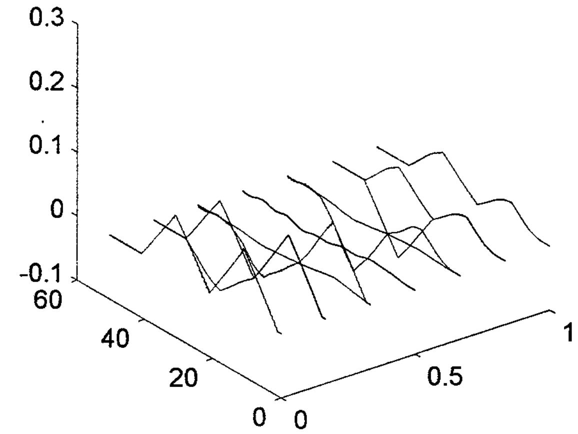

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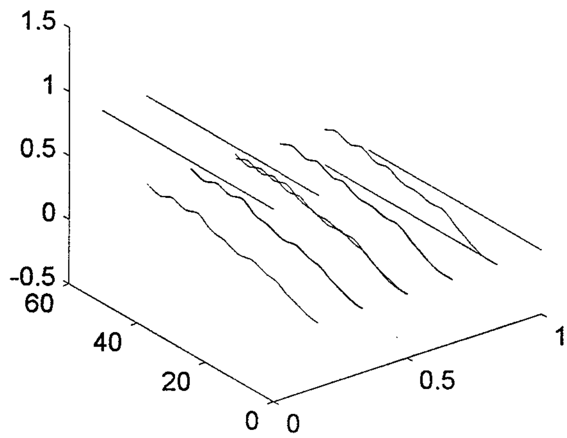

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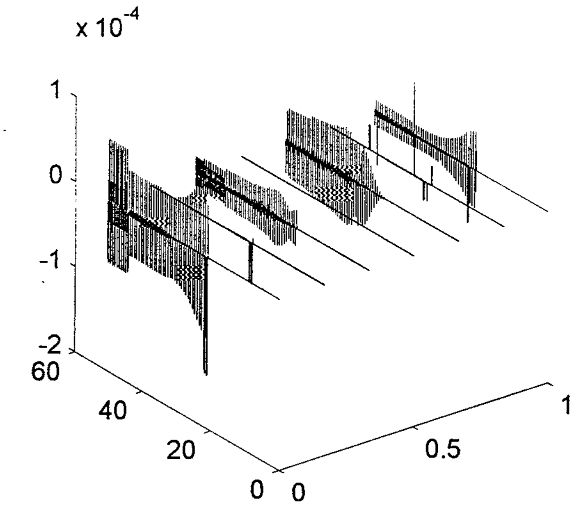

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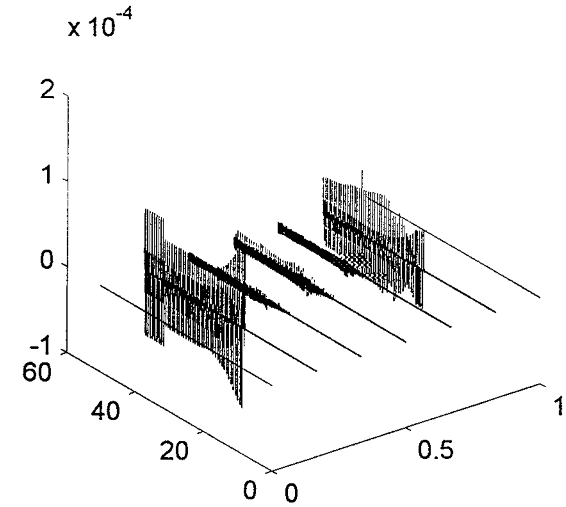

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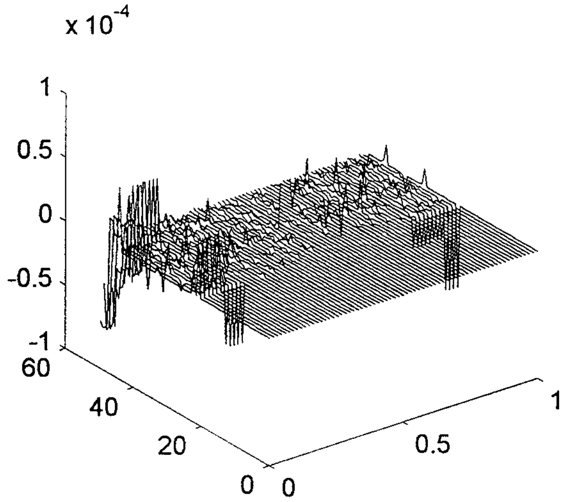
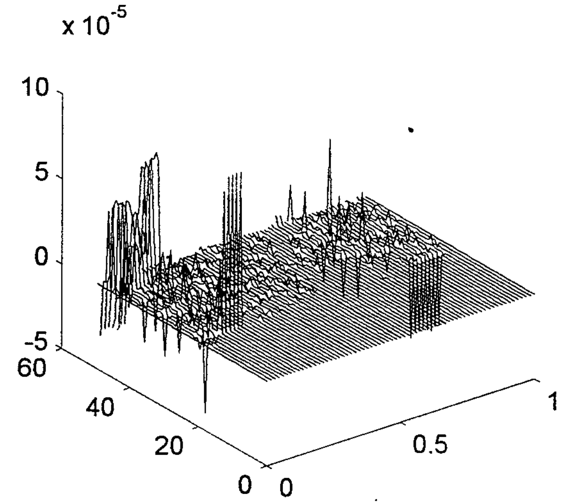


Fig.3.a.



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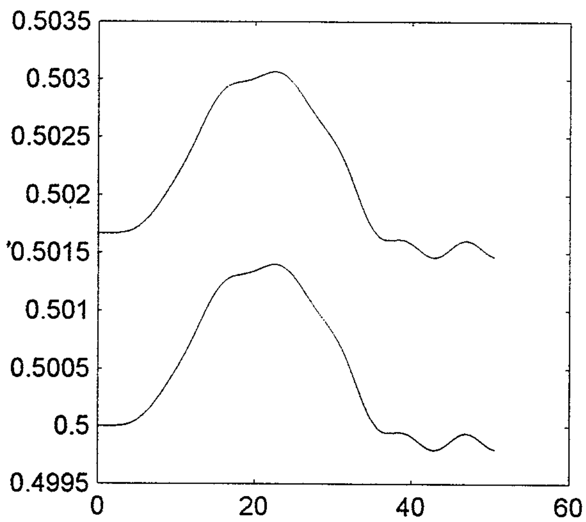
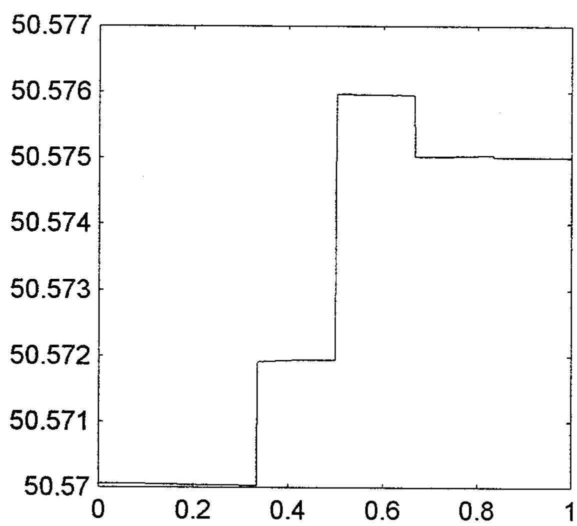


Fig.3.c.



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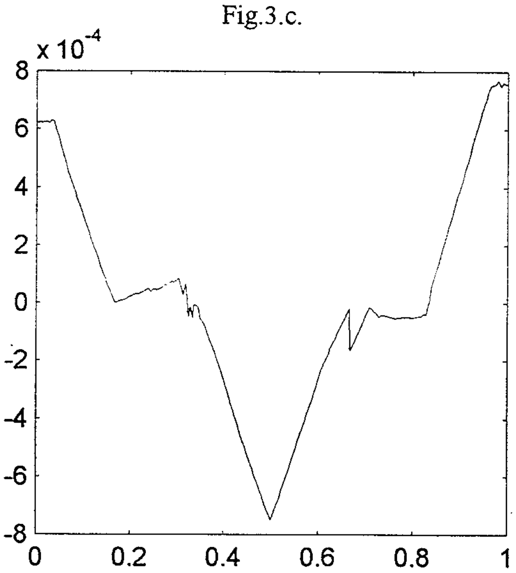

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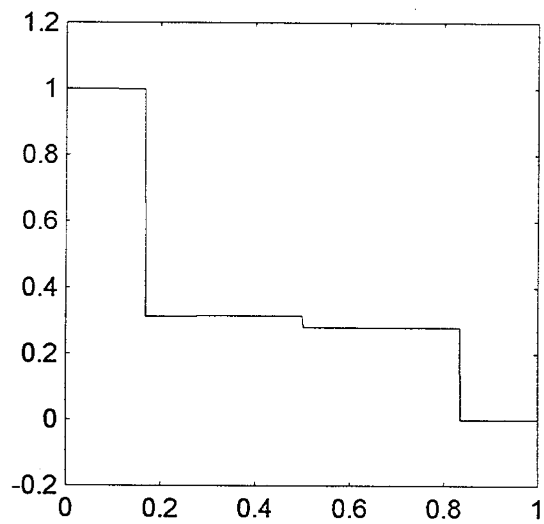

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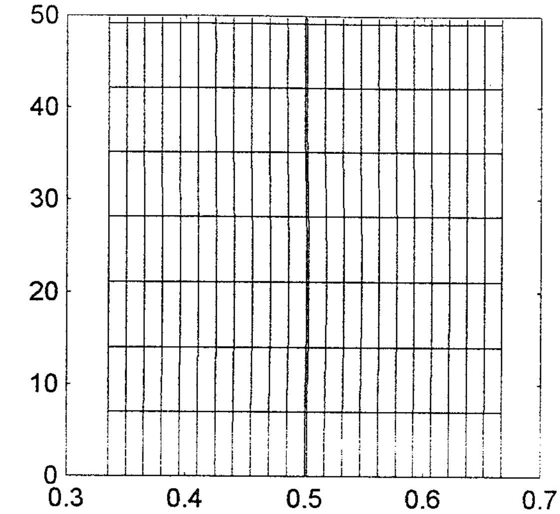

Fig.4.a.

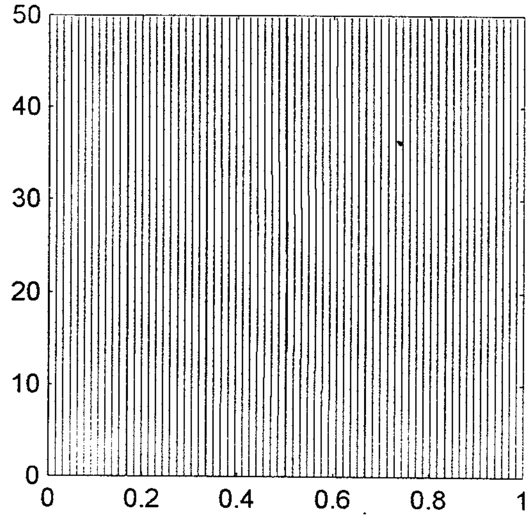

Fig.4.b.

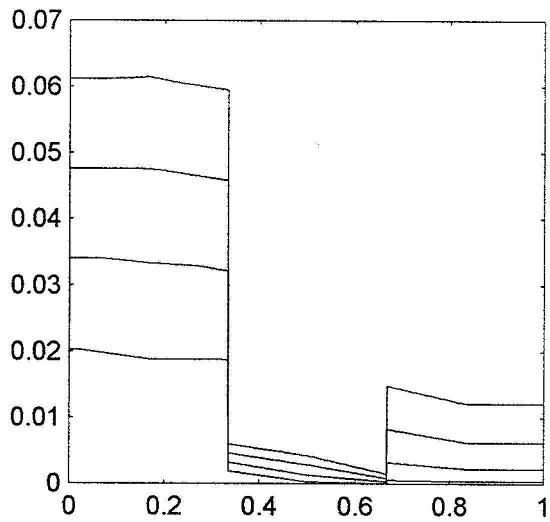

Fig.4.c.

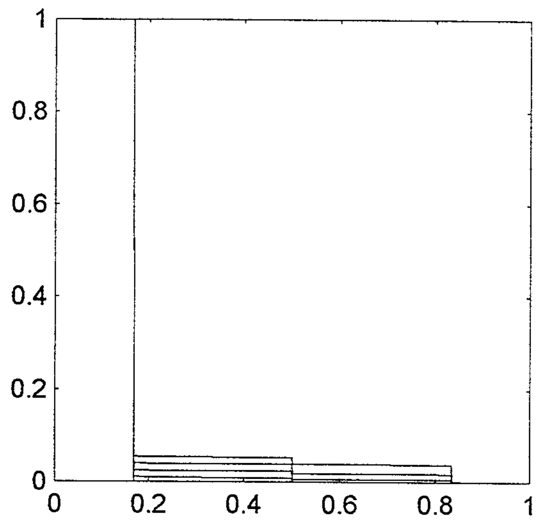

Fig.4.d.

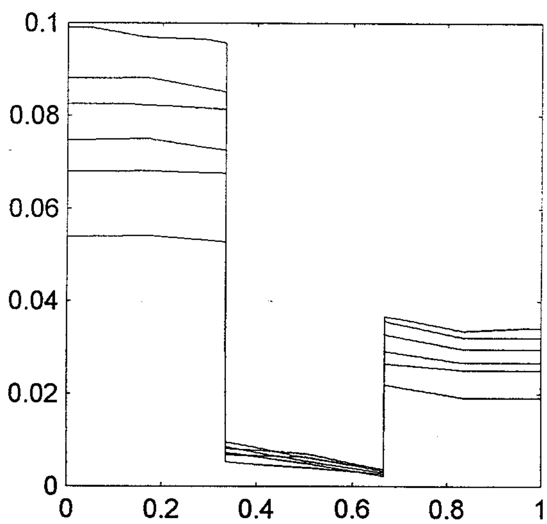

Fig.4.e.

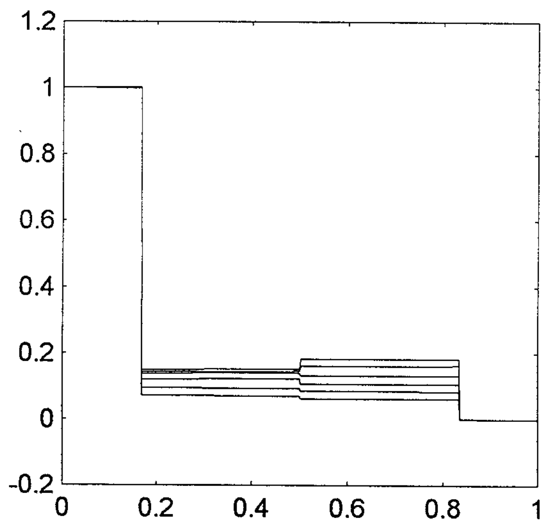

Fig.4.f.

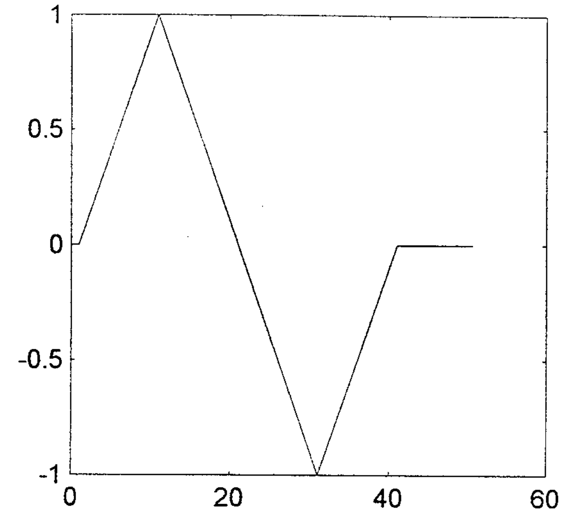

Fig.5.a.

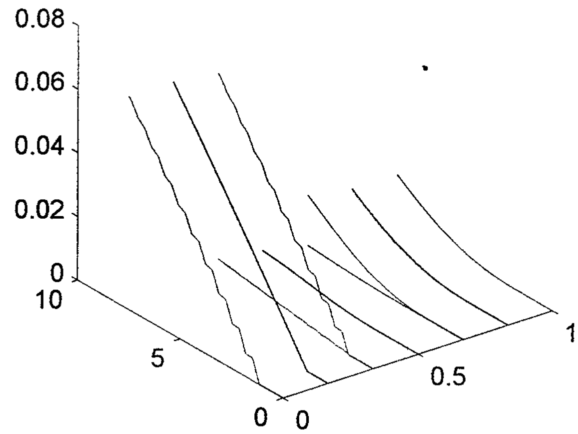

Fig.5.b.

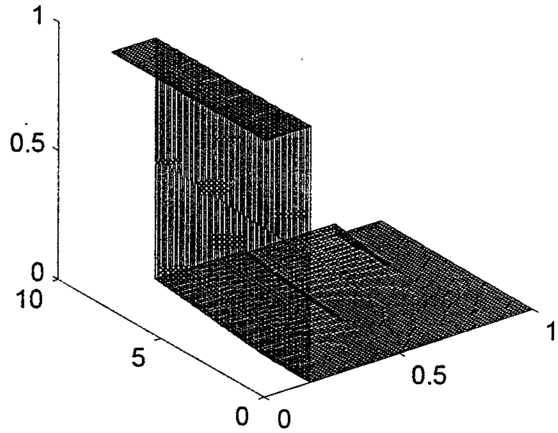

Fig.5.c.

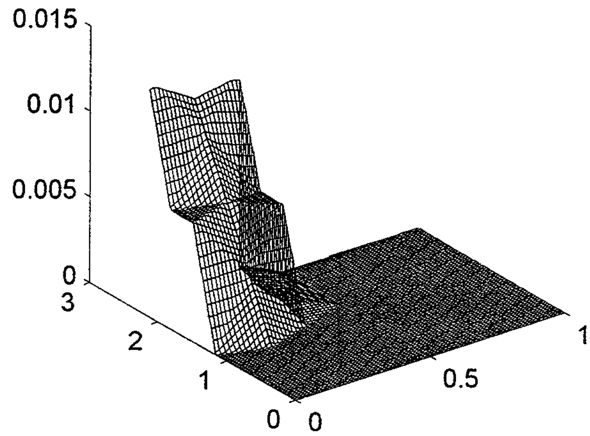

Fig.5.d.

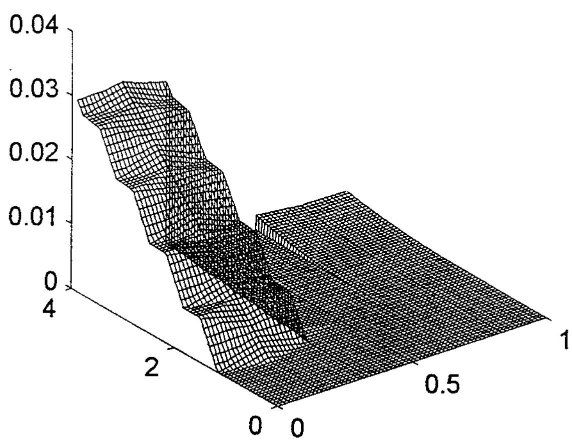

Fig.5.e.

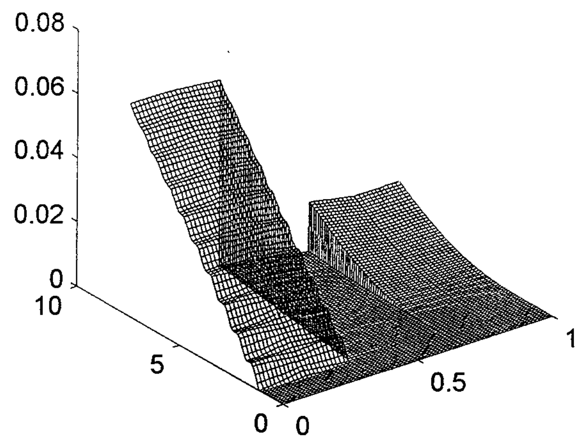

Fig.5.f.