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Reduction of Valve Seating Velocity in Electro Hydraulic Valve Actuation System

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ABSTRACT
The engine has two valves one in the inlet and other in the outlet. The valve timing is a significant thing of the engine. The valve actuation system is of two different types (1) Cam Based and (2) Camless Based actuation system. Initially valves were actuated by cam based system. Since these systems are mechanical, the performance diminishes overtime. To overcome this disadvantage and for accurate valve timing came the cam less actuation system. The electro hydraulic variable valve actuation system is one of the popular cams less actuation system in which the rotary spool valve is controlled by PI controller. The valve seating velocity is defined as the linear motion of the valve. The valve seating velocity is not optimized for good performance in the real engine. Hence we propose a system to optimize the valve seating velocity. The proposed system is modeled using bond graphs and the outputs are verified.
Keywords - bond graph, controller, hydraulic system, valve actuation, valve seating velocity

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I. INTRODUCTION

As we know, the engine valve motions, both timings and lifts, play a significant role in the resultant engine performances. They bring direct influences to engine pumping losses, in-cylinder turbulence and cylinder charge, that can lead to a considerable impact on engine output efficiency and emissions. Camless and cam-based valvetrain are the two types of the valve actuation system in engines. In the camless systems, there is no mechanical connection between the engine crankshaft and the valve train. A high level of flexibility in valve timing and valve lift is the main advantage of these systems over others. Electro-mechanical, electro-hydraulic and electro-pneumatic valve trains are all in this category. Major concerns including high cost, low reliability, high power consumption, high sitting velocity, and control complexity these are defects of the system.

The seating velocity is defined as the velocity when the valve lands on its seat. One of the main challenges in the variable engine valve actuation system is controlling the valve seating velocity. Engine valve seating velocity is essential to minimize valvetrain noise, vibration, and harshness (NVH) while maximizing durability. Many techniques have been implemented by several researchers to ensure varied controlled seating velocity in camless valve trains. An electromagnetic valvetrain (EMVT) with moving coil linear actuator achieves the low valve seating velocity. Seating performance of EMVT is evaluated by two indicators: seating velocity and holding force.PID current control to ensure the results shows that excellent valve seating performance has been achieved. Low valve seating velocity has been achieved by applying inverse system method to track a velocity-position profile in seating process [3]. In electro hydraulic fully flexible variable actuation system, PID control is used to vary the cam profile dwell portion according to the variation in speed, duration, lift, mode of the engine. In real time scenario, speed of the engine varies according to the valve profile. Dwell and periodic portion is controlled by the PID controller for good profile tracking performance [5]. dSpace system is used to achieve valve profile tracking errors within 140 microns for a 9.5 mm lift at engine speed up to 3000 rpm [6]. Cam-based valve train with the Hydraulic Variable Valve Actuation(HVVA) system replaces the single-cylinder gasoline engine. The potentials on improving the power, fuel economy and performance is obtained by GT-suite HVVA engine model. Genetic algorithm (optimization scheme) improved the engine output at full load. Fuel economy showed potential average improvement of 10.4% [7].

In electro hydraulic system, structural complexity is high since it includes inertia, damping and compliance. This structural complexity can be made simple by using bond graph. [8].

The 20 SIM software allows input of models in the form of equations, block diagrams, bond graphs and iconic diagrams. The output can be represented as x-t and x-y plots and as 3D-animations.Optimization and multiple run facilities of the simulator allow parameter estimation, controller tuning or sensitivity analysis. The above-mentioned features make 20 SIM a tool for research as well as education. Iconic diagrams can help a novice to get used to the bond graph notation. [9].When hydraulic valve system is built using bond graph, parameters such as length, inner diameter, wall thickness and material of the pipeline can be included [10].Bond graph model gives the same result as the ordinary differential equation state model. Hydraulic cylinder and spool valve is modelled by bond graph [11].The software facilities are explored by using both signal block diagram and the power bond graph elements within the same model. The friendly approach within the 20SIM software environment, also allows user defined sub models for facilitating the programming of the diverse needs for physical systems modeling, is an undisputable advantage [12].

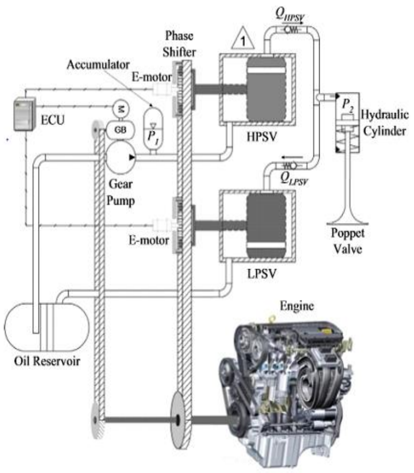

Fig 1 Schematic diagram of the variable valve actuation

Electro hydraulic valve actuation system is using the rotary spool valve rotated by servo motor controlled by the PI controller. Moreover it has motor pump connected to the engine shaft which rotates at half of the engine speed. The pump converts mechanical motion into hydraulic motion. The fluid flows through the high pressure spool valve which pushes the poppet valve against the spring force to supply fuel to the engine.

The retracting action to the cylinder is provided by the spring force which makes low pressure spool valve to be open thus, fluid flows to the reservoir. The rotary spool valve is operated by the servo motor which is controlled the PI controller. The system has

3.4 Bond Graph design parameters

The engine design parameters considered to

valve seating velocity of 150-200 mm/s which is not sufficient.

The main problem of the actuation system is to optimize the valve seating velocity in the engine. According to the literature, the existing system has velocity value of 150-200 mm/s [1]. The engine has satisfactory performance when the velocity is less than 100 mm/s. When the valve seating velocity is optimized, the engine gets high volumetric efficiency, low fuel consumption and low exhaust gas emission. Hence an attempt is made to optimize the velocity, by using bond graphs.

Objective

To reduce the valve seating velocity of the electro hydraulic actuation system to be less than 100 mm/s using bond graphs.

II. DESIGN WORK

1.1 Engine Parameters

Table 1 shows the engine parameters [1] values used to calculate the bond graph parameters. The input of the model is crank shaft rotation (rpm) and output is valve seating velocity.

Table 1 Engine parameters

PCyl,0 [bar]	5
r_{valve} [mm]	12.5
r_{cs} [mm]	39.3
l_{Rod} [mm]	169.2
P_O [bar]	1.01
A_{piston} [cm2]	64
$V_{Cyl,0}$ [cc]	56
C_p [kJ/kg.K]	1.0045
C_v [kJ/kg.K]	0.7175

1.2 Other Significant Parameters

Table 2 shows other significant system parameters used in the proposed system.

Table 2 Other parameters

ρ [kg/m3]	850
m [kg]	0.1
r_s[mm]	14.9
r_c[mm	15.01
l [mm]	8
K [N/m]	60000
β [bar]	17000
d [mm]	18

calculate the corresponding bond graph parameters are shown in Table 3

Table 3 Bond Graph Design parameters

MECHANICAL	Inertia (I)	Moment of Inertia of the crank shaft, Mass of the plunger, Mass of the motor shaft
	Compliance (C)	Spring stiffness of the plunger
	Resistance(R)	Frictional force of the motor
	Source of Effort (e)	Speed of the crank shaft
	Flow (f)	Velocity of the plunger
HYDRAULIC	Inertia (I)	Fluid inertence (I_f)
	Resistance(R)	Fluid Resistance(R_f)
	Compliance(C)	Accumulator Capacity(C_f)
	Source of Effort (e)	Oil Pressure
ELECTRICAL	Inertia (I)	Armature Inductance
	Resistance(R)	Armature Resistance

3.5 PID controller

PID controller continuously calculates an error value e(t) as the difference between a desired set point r(t) and a measured process variable y(t), and applies a correction based on proportional, integral, and derivative terms. As the opening of a control valve, the controller attempts to minimize the error over time by adjustment of a control variable u(t),to a new value determined by a weighted sum of the control terms. The position control of DC servo motors is very important since they are extensively deployed in various servomechanisms. Basically PID controllers are used to improve the transient response of DC servo motors. The suitable accurate servo controlled PID values [2] shown in the table 4.

Table 4 PID Controller Values

Constant gain (K)	0.8328
Proportional constant(Kp)	10.5
Integral constant(Ki)	0.0055
Derivative constant(Kd)	0.0755

3.6 Servo motor

Servomotor contains the closed-loop control system, which has the position sensor and speed control device which attain the required position of the shaft.The servomotors is used the optical rotary encoders to measure the speed of the output shaft and a variable-speed drive to control the motor speed. The PID control algorithm, allow the servomotor to be brought to its commanded position more quickly and more precisely, with less overshooting.As per [2] the servo motor parameters are used in this system.

Table 5 Servo Motor Properties

PARAMETER	VALUE	UNIT
Armature inductance(I)	0.035	Henry
Armature resistance(R)	2.45	Ohm
Moment of inertia(I)	0.022	$Kg.m^2$
Frictional Resistance(R)	0.005	Nm

III. PROPOSED SYSTEM

The source of effort is known as Se denotes the engine speed which is given as input. The Se1 denotes the reservoir pressure. Gyrators of the system insist conversion of one form of energy to other form of energy. Where the GY denotes the motor pump and GY1 denotes the poppet valve. The I denotes the mass where the rotational moment contain the inertia which is identified using (1)

Moment of inertia $I=mR^2/2$. m/N (1)

Resistance of the mechanical part is damping of the system which indicates the symbol R. We consider the system which not contains the damping system.

The Mechanical energy is converted into the hydraulic energy. The conversion part is the gyrator, which is the constant value to put in the system to consider the required output of the part.

The hydraulic part the flow of the liquid contains the inertance. Fluid inertance denoted by I which is calculated using (2)

Fluid Inertia $I = \rho L/A$. $Pa.s^2/m^3$ (2)

When surface of the pipe touches the fluid surface it creates some resistive force in the system. It is denoted by R. The compliance force is calculated by (3).

Fluid resistance $R= 8\mu L/2a^{4.\cdot}$ $Pa.s^2/m^3$ (3)

The accumulator and storage tank is one of the significant part of the hydraulic system which stores the hydraulic energy. It is denoted by C which is calculated by (4).

Fluid compliance $C= A/\rho g$ m^3/Pa (4)

The TF is the transformer function which is used to vary the effort or flow. As the diameter of the pipe changes, the hydraulic energy changes which is modelled by the transformer function. The Hydraulic resistance and Fluid Inertance is denoted using the R2 and I3.

Using the GY2 hydraulic energy is converted into the mechanical energy which is the constant value.

The plunger is actuated by the hydraulic power to get a linear motion. The Mass of the plunger is calculated by(1). The spring in the plunger opposes its movement. In this spring force is denoted by C which is calculated by (5).

Compliance force C=1/K. Kg (5).

The second part of the system is the controller part which contains the servo motor and PID controller as shown in fig 4. PID controller normally denotes as the PID symbol in the bond graph system. GY2, GY3 denotes gyrator which is used to convert electrical energy into mechanical energy. I6, I7 denotes the inductance of the electrical energy. R4, R6 denotes the electrical resistance of the system. I5, I8 denotes the mass moment of inertia in the rotary motion of the mechanical part. R5, R7 indicates the damping force of the system in the rotary motion. The output of the system is valve seating velocity denoted by I4. Because, the mechanical system flow parameter is velocity. The flow of the bond graph states the velocity and the effort is the force.

Table 6 Bond Graph values

Parameter	Value	Unit
Moment of inertia(I)	0.00109658	Kg.m^2
Moment of Inertia(I_1)	0.00054829	Kg.m^2
Fluid inertance(I_2)	0.035082	Pa.s^2/m^3
Fluid resistance(R)	19.7943	Pa.s^2/m^3
Fluid capacitance (C)	0.000954051	m^3/Pa.
Fluid inertance(I_3)	2.4050083	Pa.s^2/m^3
Fluid resistance(R_1)	20.413644	Pa.s^2/m^3
Mechanical compliance(C_1)	0.000016666	m/N
Mechanical inertence (I4)	100	Kg
Transformer constant	0.5	NIL
Transformer constant(TF_1)	0.9977336	NIL
Transformer constant (TF2)	0.997	NIL
Gyrator constant (GY)	2.8045	NIL
Gyrator constant (GY1)	1.25244	NIL

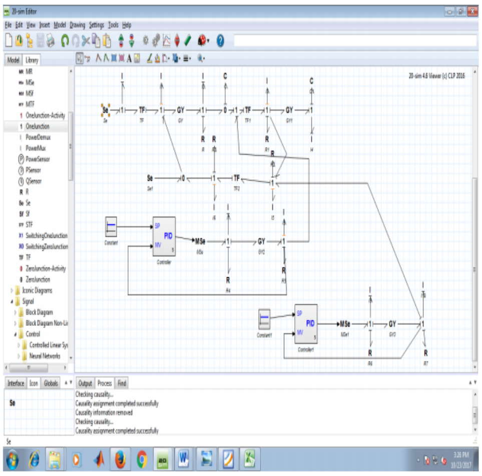

Fig 2 The proposed system modelled in Bond graph

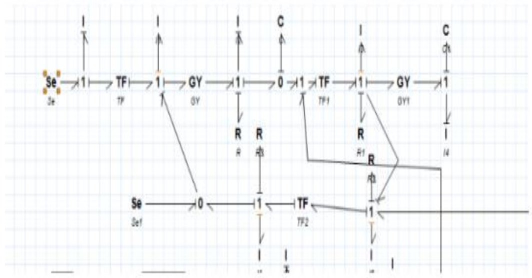

Fig3 Mechanical and Hydraulic part of the proposed system

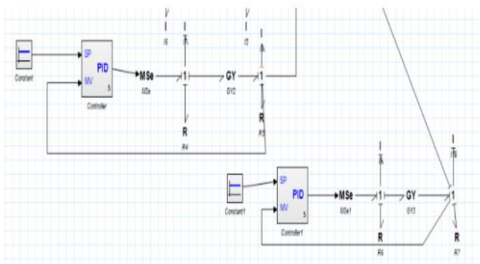

Fig 4 Controller part of the proposed system

IV. RESULTS AND DISCUSSION

The bond graph result are shown in below diagram

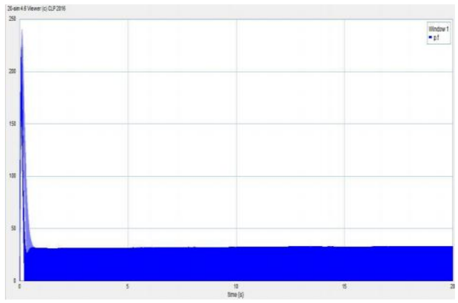

Fig 5 Valve seating velocity of the proposed system in 1000rpm.

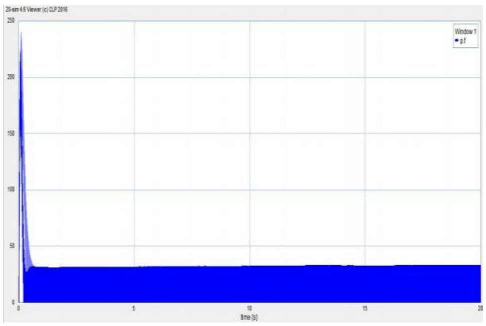

Fig 6 Valve seating velocity of the proposed system in 2000 rpm

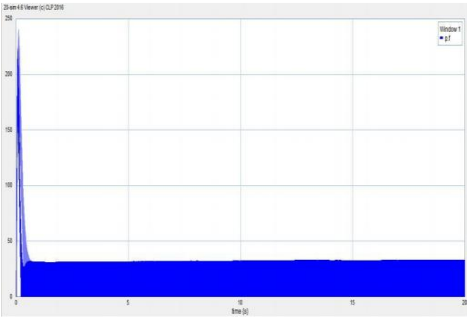

Fig 7 Valve seating velocity of the proposed system in 3000rpm.

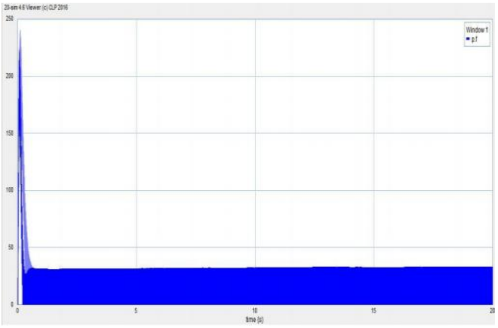

Fig 8 Valve seating velocity of the proposed system in 4000rpm.

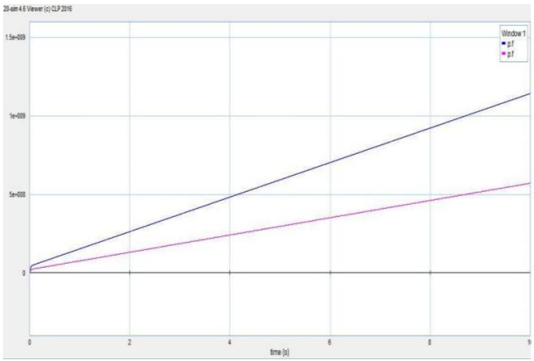

Fig 9 The motor speed and pump speed relationship in the proposed system

The Motor pump and Engine was connected and rotational speed of the pump is half of the engine rotational speed depicted the Fig 9.

Fig 5 shows the valve seating velocity graph of the proposed system. It increases exponentially and after 1 sec it gradually reduced to a constant value at an engine speed of 1000 RPM. The same velocity value is attained for increasing speed of the engine. The simulations are as shown in Fig 7, 8, 9.

As per the existing system [1] the valve seating velocity is initially 150-200 mm/s using the PI controller. It gives moderate performance for the system. This condition causes problems in proper fuel supply to the engine which diminishes the engine life and performance overtime. When the velocity is high, the fuel consumption is high, and so the engine performance decreases. Thus harmful gases are produced. As shown in fig 5 When PID controller is used, initially starting velocity value is 200-250 mm/s and then the velocity is maintained at a value less than 70mm/s (25.4 mm/s) which is optimized. This denotes that initially the fuel consumption is high due to start of the engine, and then the fuel consumption is optimized for an idle speed of 1000 rpm. Fig 6, 7, 8 depicts the valve seating velocity value using the PID controller at 2000rpm, 3000rpm and 4000rpm respectively. As the speed is increased, constant valve seating velocity value is observed which produces good performance in the system.

V. CONCLUSION

The existing electro hydraulic actuation system using PI controller has the seating velocity is 150 -200 mm/s. PID controller is used in the proposed system which is modelled by bond graph using 20 SIM software to reduce the valve seating velocity. The proposed system has valve seating velocity of 25-50 mm/s. The system simulations differ from the real time experimental setup. The model proposed should be implemented with more number of parameters under consideration and the results should be analyzed for various speeds in a real engine.

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VI. NOMENCLATURE

A_{piston}	Area of the piston [mm^2]
D	Diameter of the piston [mm]
HPSV	High pressure rotary spool valve
K	Engine valve return-spring stiffness [N/m]
L	Rotary spool valve port length [m]
Lf	Engine valve lift [m]
LPSV	Low pressure rotary pool valve
L_{tube}	Hydraulic tube length [m]
M	Engine valve moving mass [kg]
N_{engine}	Engine speed [rpm]
N_{pump}	Hydraulic pump speed [rpm]
Q_{HPSV}	Flow through HPSV [m3/s]
Q_{LPSV}	Flow through LPSV [m3/s]
$Q_{leakage}$	Leakage flow [m3/s]
Q_{pump}	Pump flow rate [m3/s]
r_c	Rotary spool valve casing radius [m]
r_{pe}	Pump to engine speed ratio
r_s	Rotary spool valve spool radius [m]
Rcs	Crank shaft radius
Re	Reynolds's number
t	Time

V_{disp} Pump displacement volume [m3/rev]
V1 Accumulator gas volume [m3]
V2 Hydraulic cylinder volume [m3]
x Engine valve displacement [m]
β Hydraulic fluid bulk modulus [pa]
θ Spool angular position [rad]
μ Oil dynamic viscosity [pa.s]
ρ Oil density [kg/m3]
ϕ Rotary spool valve port angle [rad]
ω Spool rotary velocity [rad/s]

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