# Active Lubrication For Hydrostatic Journal Bearing Using Adaptive Control

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**Abstract: Researches for improving both the load rejection performance and to maintain the air gap for minimum eccentricity in hydrostatic journal bearing, have been increasing in recent decades. Hydrostatic journal bearings are getting popularity from day to day due to their capability to support heavy loads, precise movement, and high stiffness. The performance factors of hydrostatic journal bearing depend on an external source of pressure that supply fluid at a certain pressure. Normally pump is responsible to supply fluid at a certain pressure, which has poor performance and low efficiency. To contribute a high-accuracy tracking control to maintain the clearance gap by dealing with nonlinearities and uncertainties, this paper presents servo valve with adaptive backstepping control strategy to provide active lubrication for positioning of a shaft in a hydrostatic journal bearing. To check the effectiveness of an adaptive backstepping technique, a number of experiments have been performed in Matlab/Simulink using different values of the speed of the shaft, external load and viscosity. It is found that the adaptive technique has better results as compared to PID strategy. The adaptive backstepping technique has fast response not only receiving uniform value of oil film thickness initially, but also has better load rejection performance. It is also found that proposed strategy is robust against different value of viscosity of SAE 30 grade oil as well as improvement in performance is better with increasing speed due to hydrodynamic effects as compared to PID strategy.**



**Keywords: Fluid Film Lubrication, Recess, Hydrostatic Journal Bearing, PID control, Adaptive Backstepping control etc.**


## I. INTRODUCTION

Hydrostatic journal bearing has some good characteristics such as; high accuracy, better stability and low friction that make it part of rotary machinery. One of the versatility of hydrostatic bearing is hybrid behavior at high speed to improve load rejection performance. Hydrostatic bearings are preferred in a lot of engineering applications due to their robust and favorable design characteristics. These characteristics include better ability to tolerate, absence of stick–slip characteristics, virtual independence of speed, negligible friction at small speed, zero wear of bearing surfaces, high fluid film stiffness and less vibrations, good damping, and better positional accuracy. Some good applications of hydrostatic bearings are found in the test rigs, Dynamometers as well as machine tool industry [1], in lock gates [2], and the circular saw guides which are used by the wooden product industry [3] and in the slippers of motors and axial piston pumps [4, 5].

Research has used different method to get better load rejection performance as well as improve shaft's eccentricity which is a common problem under high load conditions. Research shows that higher speed has a positive effect on stability of hydrodynamic and hydrostatic bearing. Also help to attain uniform oil film thickness quickly, while to work with high speed you need to design bearing with good material which will increase cost of bearing [6]. Surface roughness and thermal effects are also two important factors that are often considered to make better load rejection performance as well as to achieve uniform oil film gap around the shaft so that shaft eccentricity should be zero [7]. A new concept has been quickly introduced to obtain uniform oil film thickness by using magnetorheological fluids in hydrostatic bearing. In this new concept Magnetic field is used to obtain uniform oil gap around the shaft, advantages are in term of faster response to obtain uniform oil film thickness, but the problem is limitation of application for smaller loads [8]. Some research has been conduct to get better dynamics characteristics of hydrostatic journal bearing using different types of restrictors such as; circular, rectangular, elliptical and annular recesses. Performance comparison parameters were ratio of bearing to pocket area and value of restrictor design parameter [9]. Different methods of compensation have used such as an orifice, capillary and flow control valve. It is found that flow valve has better performance than orifice and capillary but problem comes in the form of no feedback and load rejection performance depends upon certain load limit [10]. The main focus of current research work is to find better control algorithms so that better active lubrications can be obtained [11], also to reduce cross coupling stiffness. There are some bearings which have little cross coupling stiffness [12, 13]. Some research has been conducted on active lubrication, gives superb contribution for control of hydrostatic bearing through active lubrication. They use the root locus and transfer function techniques to drive controller and to obtain better stability for the rotor bearing system. Unfortunately, there is not enough literature in the field of control hydrostatic bearing through active lubrication. Some literature on active lubrication for hydrostatic bearing can be found in [14-18].

Current research work presents the mathematical model as well as an advance Controller strategy (adaptive backstepping controller) is presented. This adaptive backstepping controller is better than PID control which is used by [19-21]. This paper doesn't take into account the turbulent flow, misalignment effects, temperature variations and inertia effects, bearing shell deformation. By controlling the pressure and flow into opposing bearing recesses with the help of servo control systems, significant modification of fluid film flow and forces can be obtained. A multirecesses hybrid journal bearing with active lubrication is termed active hybrid journal bearing (AHJB). The controlling shaft eccentricity by means of oil film monitoring (through servo valve) is main objective of this paper.

## II. WORKING PRINCIPLE

The active hydrostatic journal bearing has four recesses or pockets, which are aligned in pair along horizontal and vertical direction as shown in Figure 1. In conventional hydrostatic bearing, fluid is injected into four recesses at constant supply pressure through capillary restrictor or an orifice. Furthermore, to control bearing actively and dynamically, active flow is provided through horizontal and vertical recesses that are aligned in pairs. In order to control active lubrication, servo valve is an integral part that is controlled by electrical voltage signals. To explain geometrical characteristics, hydrostatic journal bearing with four recesses is shown in Figure 1.

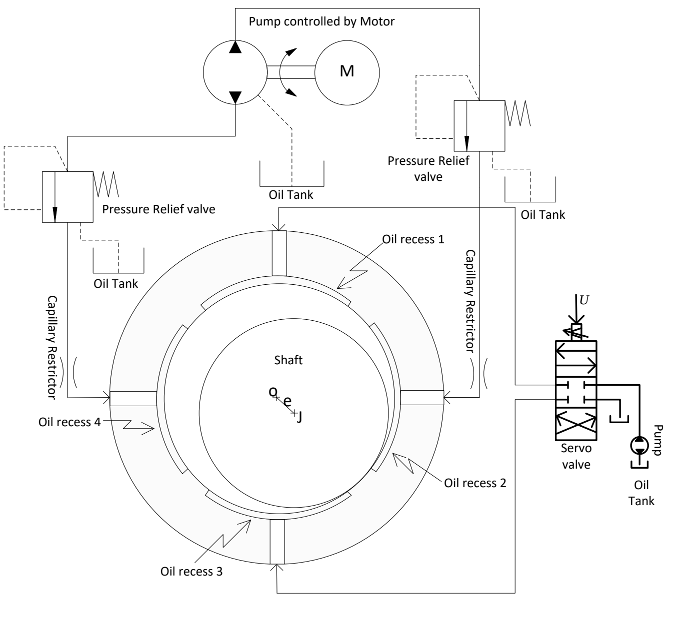


Figure 1 Structure of Hydrostatic bearing system

## III. PROBLEM FORMULATION AND COUPLED SYSTEM

The structure of the hydrostatic bearing system is shown in Figure 1. It consists of servo valve and hydrostatic bearing, hydraulic auxiliaries. The efficiency of hydrostatic journal bearing depends on the thickness of oil during operation. In the present study, the hydrostatic bearing with servo control is used by following assumptions that normally load is in vertical directions.

## IV. MATHEMATICAL MODELLING OF SERVO CONTROLLED HYDROSTATIC BEARING SYSTEM

Hydrostatic bearing is combination of some important parts that are displacement sensor, servo valve and hydraulic auxiliaries.

### A. Mathematical model of servo valve

To derive the mathematical model for servo controlled hydrostatic bearing, first mathematical model of servo valve is derived that can be described with first order transfer function when spool valve displacement is directly proportional to servo valve current [22].

$$\tau_v \dot{x}_v = K_i i_s - x_v \tag{1}$$

Where

$i_s$ is servo valve's current,

$x_v$ is spool valve's displacement

$K_i$ is servo valve opening/current gain,

$\tau_v$ is time constant.

The load flow is linear in a region where we have null opening and null load pressure [23, 24]

$$Q_{in} = K_q x_v - K_c P_L \tag{2}$$

Where $K_q$ is the flow/opening gain, $K_c$ is flow/pressure gain, $P_L$ is load pressure, $Q_{in}$ is flow entering into the bearing and $Q_L$ is load flow. Flow through the bearing is linear around the mid position of the shaft. Taking some assumption such as pipes from hydraulic cylinder to servo valve are thick and friction is negligible and modulus of elasticity is constant, help us to derive equation of continuity for system [25];

$$\begin{aligned} Q_L &= Q_{in} - Q_{out} = Q_{squeez} + Q_{compressible} + Q_{leakage} \\ Q_{in} &= A_e \frac{dh}{dt} + \frac{V_e}{\beta_e}\frac{dP_L}{dt} + \frac{lh_o^3}{12\eta b} P_L - \frac{U}{2} h + C_l P_L \end{aligned} \tag{3}$$

Where

$A_e$ is effective area for bearing

$V_e$ is effective volume for bearing

$U$ is surface velocity for fluid

$h_0$ is clearance for the bearing

$\beta_e$ is Bulk mdulous

$h$ is oil film thickness

$C_l$ is Leakage coefficient for bearing

$\eta$ is viscosity for fluid

$l$ is length for recess

$b$ is width for recess

According to newton's law, dynamic system will have a dynamic equilibrium if the sum of forces is equal to zero. So, the force balance equation of bearing is given by;

$$F_{film} - F_L = m\ddot{h} + B_d\dot{h} \quad (4)$$

Where

$F_{film}$ is force exerted by the oil film

$F_L$ is external load force

$m$ is mass for the shaft (journal)

$B_d$ is viscous damping of fluid

By combining equation 1 to 4 to get an overall mathematical model of the system which is shown in Figure 2.

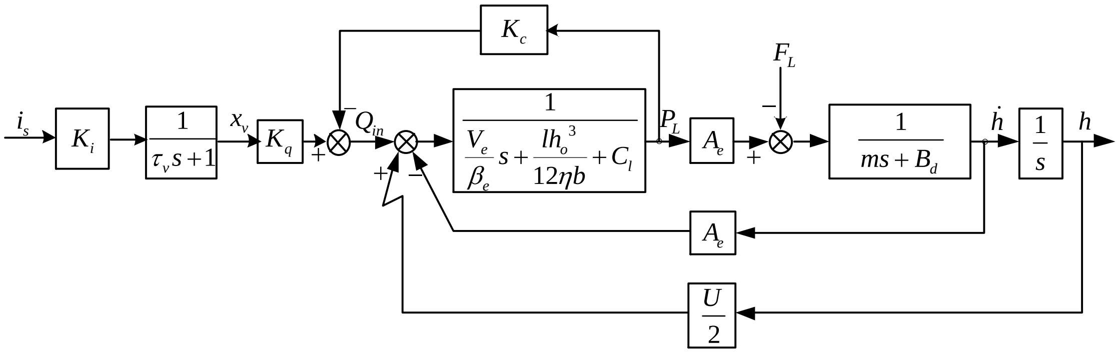

Figure 2 Schematic Diagram of Hydrostatic bearing controlled by servo-valve

## V. DESIGN STRATEGIES

### a) PID Design Strategy

Hydrostatic bearing is famous due to its ability to support load by thin pressurized layer. A Hydrostatic bearing is a bearing in which load is supported by a pressurized thin layer of fluid. Usually, pressure is created by using a pump and hydraulic accessories. Higher load produces eccentricity in the hydrostatic bearing. Eccentricity is distant between journal center and bearing. Eccentricity is zero or negligible whenever oil film is distributed uniformly around journal.. In this paper, PID control strategy is used to achieve uniform oil thickness. Experiments were performed with following value of $K_p = 90$, $K_i = 500$, $K_d = 0$ that gives best results. A schematic of PID control strategy is shown in Figure **3**.

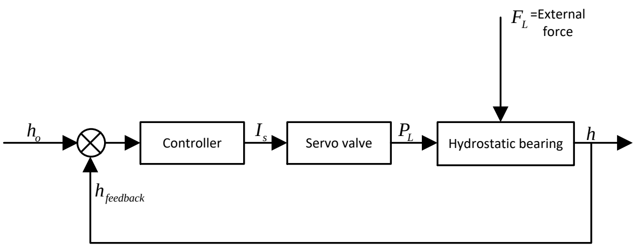


Figure 3 Block diagram of PID design strategy

### b) Adaptive Control Strategy

Lyapunov Theory

Let suppose there is a time variable differential equation of the type

$$\dot{x} = f(x,t) \quad (5)$$

The origin is an equilibrium point for equation (5) if $f(0,t) = 0 \quad \forall t \ge 0$. It is assumed that $f$ is such that solutions exist for all $t \ge t_0$.

***Definition 1****: Positive definite and semidefinite functions*

A continuously differentiable function $V : \Re^n \to \Re$ is called a positive definite in a region $U \subset \Re$ containing origin if

1. $V(0) = 0$
2. $V(x) > 0, \quad x \in U \quad \& \quad x \ne 0$

A function is called positive semidefinite if condition 2 is replaced by $V(x) \ge 0$.

***Definition 2***: Uniform Lyapunov's stability

The solution $x(t) = 0$ of an equation (5) is uniformly stable if for every $\varepsilon > 0$ there exist a number $\delta(\varepsilon) > 0$, independent of $t_0$, such that

$$\|x(t_0)\| < \delta \Rightarrow \|x(t)\| < \varepsilon \quad \forall t \ge t_0 \ge 0$$

The solution is uniformly asymptotically stable if it is uniformly stable and there is $c > 0$, independent of $t_0$, such that $x(t) \to 0$ as $t \to \infty$, uniformly in $t_0$, for all $\|x(t_0)\| < c$

***Definition 3****: Class K function*

A continuous function $\alpha:[0,a)\rightarrow[0,\infty)$ is said to belong to a class $K$ if it is strictly increasing and $\alpha(0)=0$ . It is said to belong to class $K_\infty$ if $a=\infty$ and $\alpha(\mathrm{r})\rightarrow\infty$ as $r\rightarrow\infty$.

***Theorem 1*** *Lyapunov's stability theorem: time varying systems*

Let $x=0$ be an equilibrium point for an equation (5) and $D=\left\{x\in\Re^n \,\|x\|<r\right\}$. Let $V$ be a continuously differentiable function such that

$$\alpha_1(\|x\|)\le V(x,t)\le\alpha_2(\|x\|)$$

$$\frac{dV}{dt}=\frac{\partial V}{\partial t}+\frac{\partial V}{\partial x}f(x,t)\le-\alpha_3(\|x\|)$$

For $\forall t\ge0$, where $\alpha_1$ $\alpha_2$ $\alpha_3$ are class $K$ functions. Then $x=0$ is uniformly asymptotically stable.

***Theorem 2*** *(Boundedness and convergence set)*

Let $D=\left\{x\in\square^n \,\|x\|<r\right\}$ and suppose that $f(x,t)$ is locally Lipchitz on $D\times[0,\infty)$ . Let V be a continuously differentiable function such that

$$\alpha_1(\|x\|)\le V(x,t)\le\alpha_2(\|x\|)$$

And

$$\frac{dV}{dt}=\frac{\partial V}{\partial t}+\frac{\partial V}{\partial x}f(x,t)\le-W(x)\le0$$

$\forall t\ge0,\forall x\in D$ Where $\alpha_1$ and $\alpha_2$ are class K functions defined on an interval $[0,r)$ and $W(x)$ is continuous on $D$ . Further, it is assumed that $\dot{V}$ is uniformly continuous in $t$ . Then all solutions to (5) with $\|x(\mathrm{t}_0)\|<\alpha_2^{-1}(\alpha_1(r))$ are bounded and satisfy

$$W(x(t))\rightarrow0 \quad as \quad t\rightarrow\infty$$

Moreover, if all the assumption holds globally and $\alpha_1$ belongs to a class $K_\infty$ , the statement is true for all $x(t_0)\in\square^n$ .

Let's suppose state variables are $x_1$ , $x_2$ , $x_3$ , $x_4$ for $h$ , $\dot{h}$ , $\ddot{h}$ , $\dddot{h}$ . The state equations for the system is given by;

$$\begin{aligned}\dot{x}_1&=x_2\\ \dot{x}_2&=x_3\\ \dot{x}_3&=x_4\\ \dot{x}_4&=Au-Bx_4-Cx_3-Dx_2+Ex_1-F\ddot{F}_L-G\dot{F}_L-HF_L\end{aligned} \tag{6}$$

Where

$$A=\frac{K_qK_iA_e\beta_e}{m\tau_VV_e}\ ,\ B=\left(\frac{B_d}{m}+\frac{\beta_eT}{V_e}+\frac{1}{\tau_V}\right)\ ,$$

$$C=\frac{B_dT\beta_e}{mV_e}+\frac{B_d}{\tau_Vm}+\frac{T\beta_e}{\tau_VV_e}+\frac{A_e^2\beta_e}{mV_e}\ ,$$

$$D=\frac{TB_d\beta_e}{\tau_VmV_e}+\frac{\beta_eA_e^2}{\tau_VmV_e}-\frac{A_e\beta_e}{mV_e}\frac{U}{2}\ ,$$

$$E=\frac{A_e\beta_e}{\tau_VmV_e}\frac{U}{2}\ ,\ F=\frac{1}{m}\ ,\ G=\frac{T\beta_e}{mV_e}+\frac{1}{\tau_Vm}\ ,\ H=\frac{T\beta_e}{\tau_VmV_e}$$

For ease of analysis fourth state equation can be written as;

$$\dot{x}_4=\int+gu \tag{7}$$

Where

$$\int=-Bx_4-Cx_3-Dx_2+Ex_1-F\ddot{F}_L-G\dot{F}_L-HF_L$$

$$g=A$$

An examination of the equation (6) shows that this system falls into the class of system known as a strict feedback form system. Therefore, a recursive Lyapunov design such as adaptive backstepping controller design will be appropriate.

In order to design IBSC, new state variables are defined as

$$z_1=x_1-x_d \tag{8}$$

$$z_2=x_2-\alpha_1(\mathrm{z}_1) \tag{9}$$

$$z_3=x_3-\alpha_2(\mathrm{z}_1,\mathrm{z}_2) \tag{10}$$

$$z_4=x_4-\alpha_3(\mathrm{z}_1,\mathrm{z}_2,\mathrm{z}_3) \tag{11}$$

Where $x_d$ is desired input, while $\alpha_1(\mathrm{z}_1)$ , $\alpha_2(\mathrm{z}_1,\mathrm{z}_2)$ and $\alpha_3(\mathrm{z}_1,\mathrm{z}_2,\mathrm{z}_3)$ are virtual control inputs that help system to attain convergence and stability. These virtual control inputs can be found with the help of Lyapunov function.

Step 1

Let's suppose positive semi-definite function for equation (8) is

$$V_1=\frac{1}{2}z_1^2 \tag{12}$$

The given Lyapunov function guarantees the stability of control system if

$$\alpha_1(\mathrm{z}_1)=-k_1z_1+\dot{x}_d \tag{13}$$

Then

$$\dot{V}_1=z_1z_2-k_1z_1^2 \tag{14}$$

Step 2

Positive semi-definite function for equation (9) is given by;

$$V_2 = V_1 + \frac{1}{2} z_2^2 \qquad (15)$$

The given Lyapunov function guarantees the stability of control system if

$$\alpha_2(z_1, z_2) = -k_2 z_2 - z_1 + \dot{\alpha}_1(z_1) \qquad (16)$$

Then

$$\dot{V}_2 = -k_1 z_1^2 + z_2 z_3 + z_2(z_1 + \alpha_2(z_1, z_2) - \dot{\alpha}_1(z_1)) \qquad (17)$$

Step 3

Positive semi-definite function for equation (10) is given by;

$$V_3 = V_2 + \frac{1}{2} z_3^2 \qquad (18)$$

The given Lyapunov function guarantees the stability of control system if

$$\alpha_3(z_1, z_2, z_3) = \dot{\alpha}_2(z_1, z_2) - k_3 z_3 - z_2 \qquad (19)$$

Then

$$\dot{V}_3 = z_3 z_4 - k_1 z_1^2 - k_2 z_2^2 - k_3 z_3^2 \qquad (20)$$

Step 4

Positive semi-definite function for equation (11) is given by;

$$V_4 = V_3 + \frac{1}{2} z_4^2 \qquad (21)$$

The given Lyapunov function guarantees the stability of control system if

$$u = \frac{1}{A}\left(Bx_4 + Cx_3 + Dx_2 - Ex_1 + F\ddot{F}_L + G\dot{F}_L + HF_L + \dot{\alpha}_3(z_1, z_2, z_3) - z_3 - k_4 z_4\right) \qquad (22)$$

Then

$$\dot{V}_4 = -k_1 z_1^2 - k_2 z_2^2 - k_3 z_3^2 - k_4 z_4^2 \qquad (23)$$

From equation (23), It is found that control system using IBSC law of equation (22), can guarantee exponential stability.

## VI. Results and Discussion

In order to check the effectiveness of ABSC over PID, different experiments have been done in Matlab/Simulink by using parameters given in Table 1.

a) Effectiveness of Strategy under different External load

In order to verify the effectiveness of strategy, first of all jerk load (square wave of amplitude 1200N, duration of 0.5 second) is used which acts on the shaft at a time of 0.6 second. It is found that IBSC strategy has better results than PID strategy as shown in Figure **4**(a). Figure **4**(a), shows that when the external load acts on the shaft, then IBSC has 17.8 μm changes in bearing clearance and PID has 20.27 μm changes in bearing clearance. So, ABSC has better results than PID. Further analysis is carried out using similar conditions with a different type of external load and the same results were obtained, shown in Figure **4**(b).

To check the linearity of IBSC against external load, a square wave of amplitude 600N, 1200N, 1800N, period 150 millisecond and pulse width is 50% of the period is used. It is found that changes in oil film thickness is 9 μm, 18 μm and 27 μm for 600N, 1200N and 1800N respectively as shown in Figure **4**(c). Which is basically a linear relationship of change in oil film thickness against external load. Similar results were obtained using step load and shown in Figure **4**(d).

**Table 1 Simulation Parameters**

| **Parameter** | **Symbol** | **Values** | **Unit** |
|---|---|---|---|
| **Recess Parameters** | | | |
| Initial thickness of oil film | $h_0$ | 3e-5 | m |
| Length of recess | $l$ | 0.08 | m |
| Length of land | $l_1$ | 0.01 | m |
| Width of recess | $b$ | 0.08 | m |
| Width of land | $b_1$ | 0.01 | m |
| **Spindle Parameters** | | | |
| Mass of Spindle | $m$ | 23 | Kg |
| Spindle diameter | $d$ | 0.06 | m |
| Shaft Speed | $N$ | 1500 | rpm |
| **Oil Parameters** | | | |
| Oil viscosity | $\eta$ | 0.025 | Pas |
| Oil bulk modulus | $\beta_e$ | $7\times10^8$ | Pa |
| Oil Density | $\rho$ | 900 | Kg/m$^3$ |
| Viscous Damping Coefficient | $B_d$ | 5e5 | $Ns/m$ |
| **Bearing Parameters** | | | |
| Bearing Diameter | $D$ | 0.06006 | $m$ |
| Bearing Length | $L$ | 0.06 | $m$ |
| Length of land | $L_1$ | 0.012 | $m$ |
| Shaft Diameter | $D$ | 0.06 | $m$ |
| Width of land | $B_1$ | $\pi D/16$ | $m$ |
| Clearance | $C$ | 3e-5 | $m$ |
| Effective area | $A_e$ | $1.88\times10^{-2}$ | $m^2$ |
| Effective Volume | $V_e$ | $2.8\times10^{-4}$ | $m^3$ |
| External Force | $F$ | 1200 | $N$ |
| **Servo valve Parameters** | | | |
| Opening/Current Gain | $K_i$ | $3.04\times10^{-5}$ | $m/A$ |
| Flow/opening gain at null pressure | $K_q$ | 2.7 | $m^2/s$ |
| Flow/pressure gain | $K_c$ | $1.75\times10^{-11}$ | $m^3 s^{-1} pa$ |
| Time Constant | $\tau_v$ | 0.001 | $s$ |

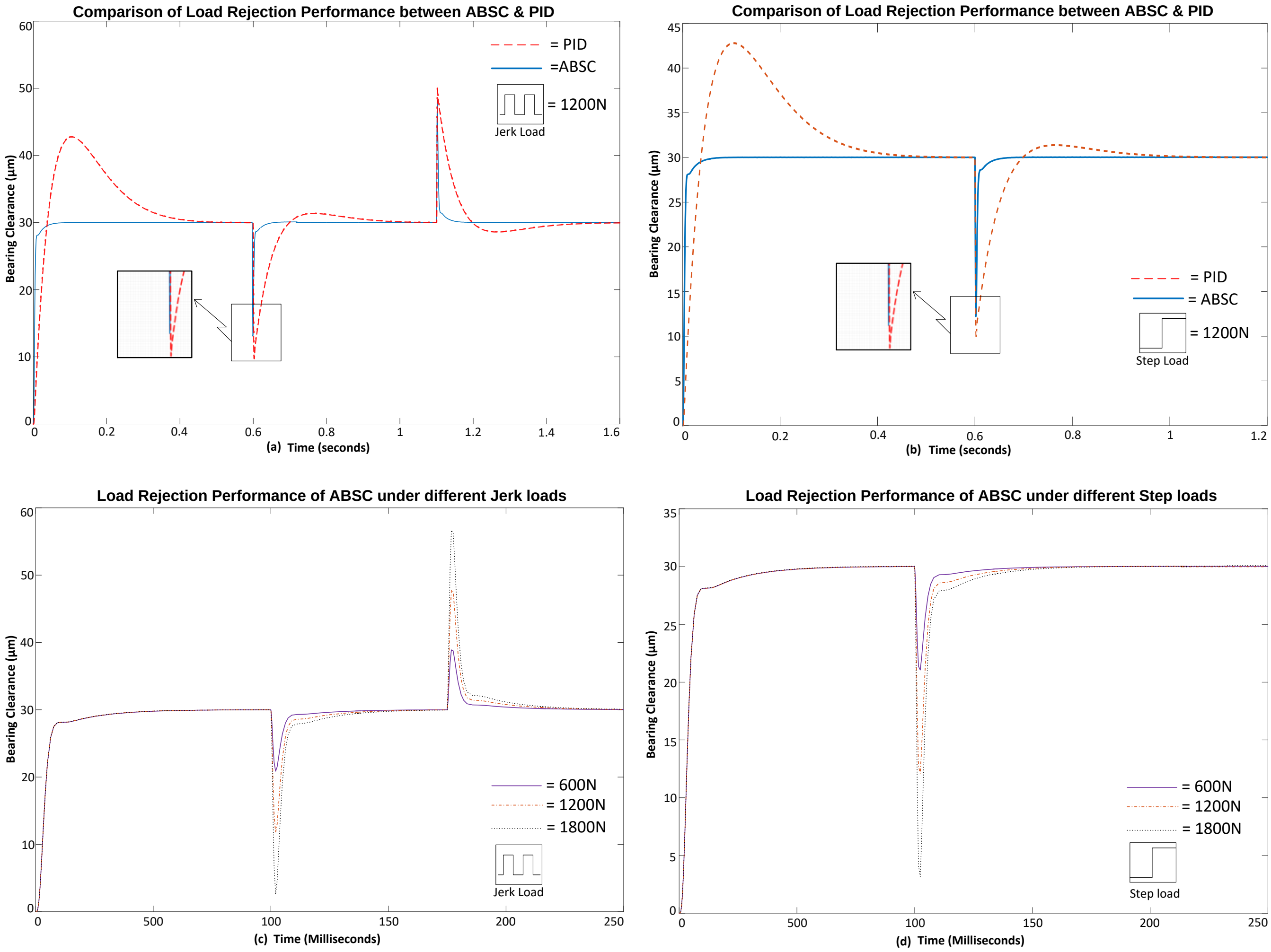


Figure 4 Effectiveness of strategy under different external load

b) Effectiveness of Strategy under different viscosity Values

In order to further verify the effectiveness of proposed strategy, SAE 30 oil is used with viscosity values (measured in mPa) 1124.11, 239.39, 74.55, 30.58, 15.28, 8.80 at temperature (Celsius) 0, 20, 40, 60, 80, and 100 respectively. It is found that strategy is not only robust against different value of viscosity but also will work for a long range of temperature for SAE 30 grade oil as shown in Figure 5(a). Robustness is not only checked under different value of viscosity for SAE 30 grade oil, but also checked different forms of external load such as constant load and jerk load. In Figure 5(a) and Figure 5(b), Strategy is checked against jerk load (square wave of magnitude 1800N with period 10 second and pulse width 5 second) and step load of 1800N with different value of viscosity for SAE 30 grade oil. It is found that there is a negligible change in response of oil film thickness that can be ignored.

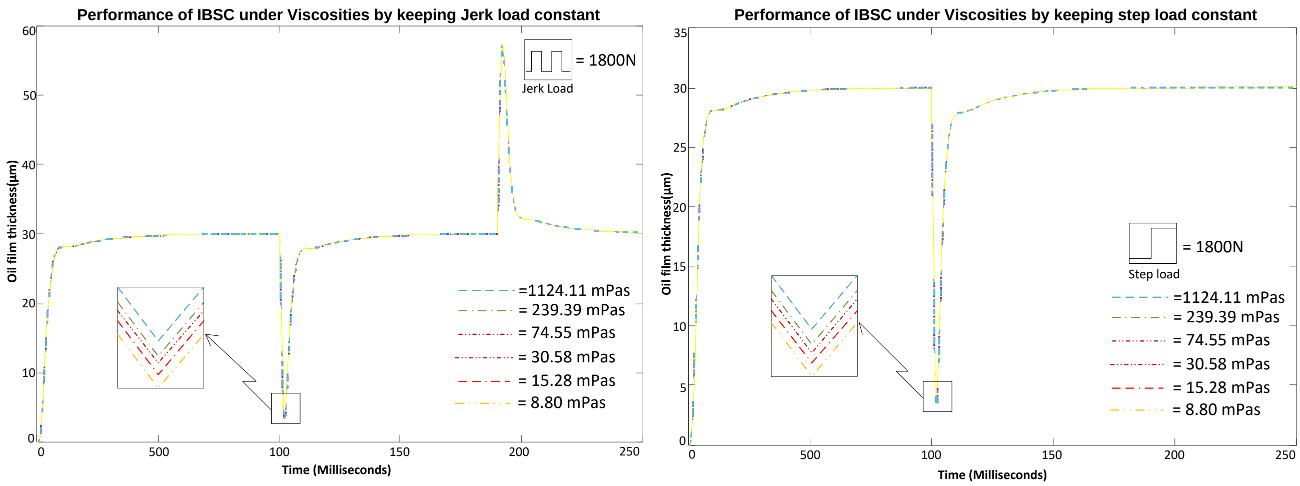


Figure 5 Effectiveness of strategy under different viscosity

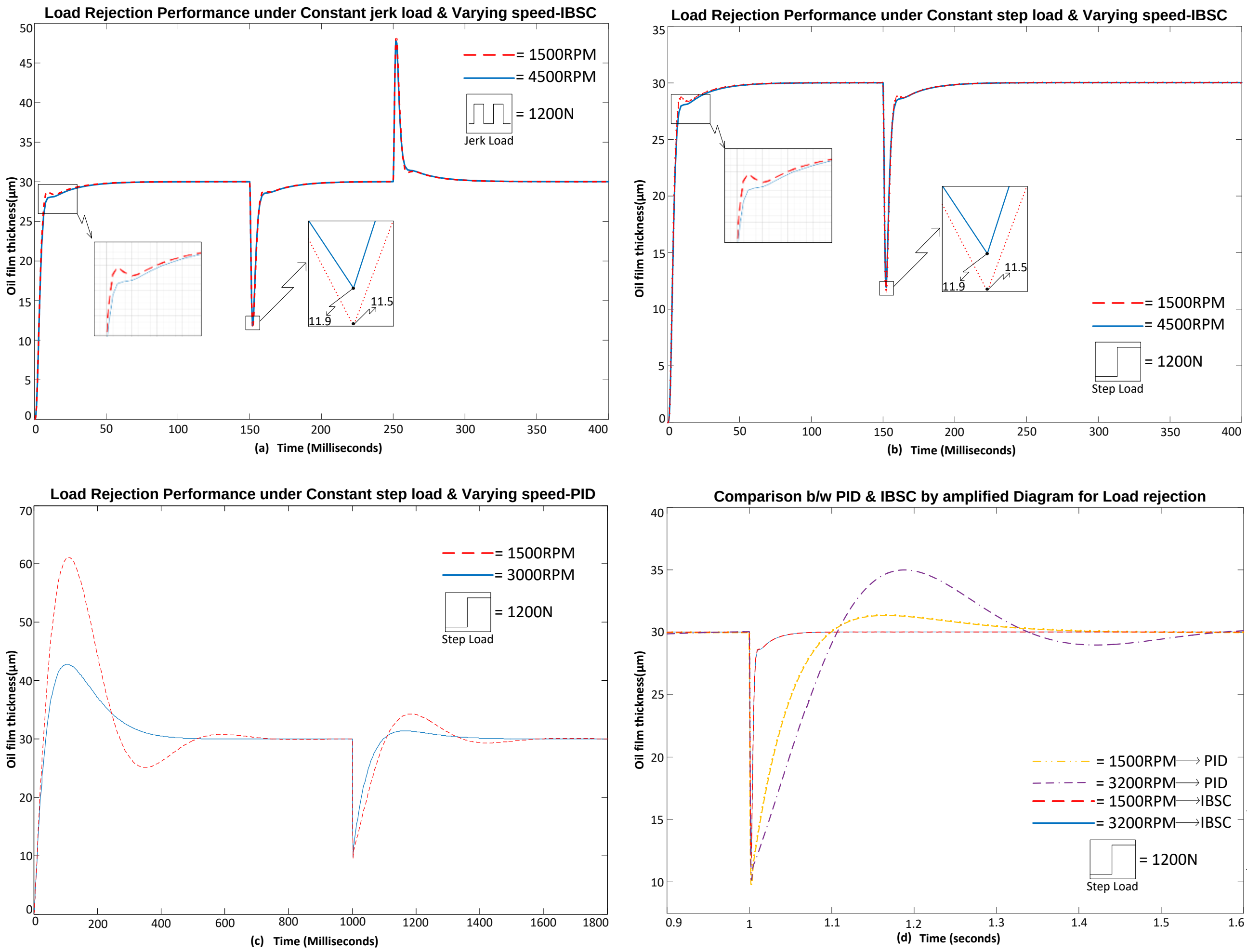


Figure 6 Effectiveness of strategy under different speed

c) Effectiveness of strategy under different speed

Hydrostatic bearings are like hybrid bearing which have dual properties of hydrodynamic bearings and hydrostatic bearings. This phenomenon is observed and prominent at higher speed of the shaft. In order to verify the hybrid nature of hydrostatic bearing, Different experiments have been done in Matlab/Simulink by using ABSC (Adaptive Controller).

First of all jerk load (square wave of amplitude 1200N, period 400 millisecond and pulse width is 25% of the period) is used which acts upon shaft at a time of 150 millisecond. To verify Hydrodynamic behavior of the hydrostatic bearing speed of the shaft is increased from 1500RPM to 4500RPM. It is found that increasing speed of shaft improves load rejection performance of hydrostatic bearing, and also it helps system to improve initial performance to achieve uniform oil film thickness when there is no external load on shaft. Results show hybrid behavior by ABSC are shown in Figure 6(a) and Figure 6(b) under different forms of external load. Similar observations have been found with a step load of 1200N applied (at a time of 150 millisecond) for a duration 400 millisecond through ABSC as shown in Figure 6(b). On the other hand PID strategy has poor results not only in case of initial performance, but also in improving dynamic performance as shown in Figure 6(c). Figure 6(d) shows the comparison of ABSC and PID. It shows that PID is unable to control oscillation at high speed while ABSC can easily control these oscillations in a hydrostatic bearing.

## VII. CONCLUSIONS

The research work presented in the present paper involved hydrostatic bearing controlled by servo-valve with the assumption that load acts in vertical direction. Results show that Adaptive backstepping control (ABSC) has better performance than PID control to achieve not only constant clearance gap around journal, but also help hydrostatic bearing to improve load rejection performance under different forms of external load. Results also show that IBSC has a faster initial response to achieve constant clearance gap as compared to PID control. On the other hand, results show that at high speed hydrostatic bearing acts like a hybrid bearing taking advantage of hydrostatic bearing properties and hydrodynamic bearing properties. Hybrid bearing properties are more prominent with ABSC strategy as compared to PID strategy. It is also found that ABSC strategy provides more direct stiffness against external load as compared to PID control strategy. ABSC is robust strategy against different values of temperature. Robustness is checked indirectly against different values of viscosities for SAE 30 grade oil and found negligible change in performance of bearing by ABSC as compared to PID which is unable to control the dynamics of hydrostatic bearing under high temperature. Present work involves controlling bearing clearance with an assumption that load acts in vertical direction future work will be to control bearing clearance against any type of load whether it is horizontal or vertical.

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