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Performances Of 6-Pocket Compensated Conical Hydrostatic Journal Bearing Under Micropolar Lubrication

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Abstract— **A theoretical analysis on the effect of number of pockets and semi cone angle on the performances of a conical hydrostatic journal bearing for a multirecess constant flow valve compensated under micropolar lubrication has been carried out. The numerical solution of the modified Reynolds equation for the conical bearing has been done using finite element method using necessary boundary conditions. The various dynamic characteristics have been presented to analyze the performance of bearing at zero speed against radial load and constant flow valve parameter.**

Index Terms— constant flow valve restrictor, conical bearing, micro polar lubrication, finite element method

I. INTRODUCTION

To improve the bearing characteristics and its stability various researchers employed various geometrical considerations as well as non-Newtonian lubricants. Conical bearing is among one of the classified. This bearing has significant advantages which supports both radial and axial loads. The performance can further be improved when conical geometry is implemented over hydrostatic / hybrid journal bearings with the use of various restrictors. The conical bearings are generally used in high speed turbine and precision machine tools etc.

The characteristics of multirecess conical hydrostatic thrust bearings were studied theoretically taking into account the effect of rotational lubricant inertia by Prabhu and Ganesan [1]. The characteristics of externally pressurized central recess conical bearings with non-constant film thickness under the assumption of isothermal laminar flow of a viscous incompressible fluid has been analyzed by Kalita et al. [2] & [3]. Khalil et al. [4] presented theoretical investigation the effect of laminar and turbulent in conical thrust bearing.

Sinha et al. [5] reported an externally pressurized non-constant gap conical bearing rotating with a uniform angular velocity considering incompressible lubricant and rotational inertia. Hong et al. [6] have presented a theoretical and experimental method to recognize the dynamic performance of an externally pressurized deep/shallow pockets hybrid conical bearings compensated by flat capillary restrictors. Verma et al. [7] have analyzed the multirecess hydrostatic journal bearing operating with micropolar lubricant. Sharma et al. [8] have studied theoretically the performance of a 4 pocket hydrostatic conical journal bearing system for various cone angles and also studied the effect of wear on its performance.

II. GOVERNING EQUATIONS

The six pocket hydrostatic conical journal bearing is shown in Fig.1. The journal is assumed to rotate with uniform angular velocity about its equilibrium position. For hydrostatic bearing design it must be possible to support the full operating load at zero speed as well as at high speed.

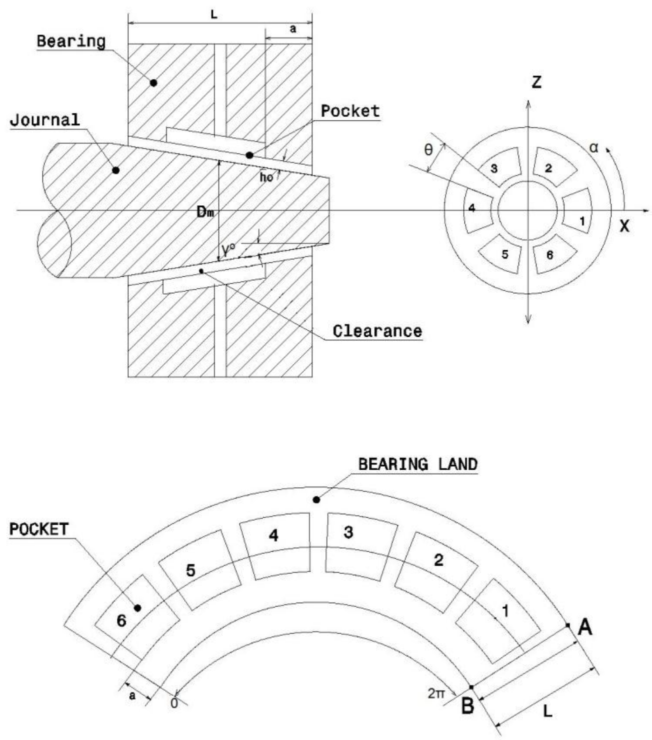

Fig 1: Six pocket hydrostatic conical journal bearing

The Reynolds equation for conical bearing is as follows [8]:

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$$\frac{1}{\beta^2}\frac{\partial}{\partial\alpha}\left[\frac{\bar{\emptyset}(N,l_m,\bar{h})}{12}\frac{\partial\bar{p}}{\partial\alpha}\right]+\frac{sin^2\gamma}{\beta}\frac{\partial}{\partial\beta}\left[\frac{\beta\bar{\emptyset}(N,l_m,\bar{h})}{12}\frac{\partial\bar{p}}{\partial\beta}\right]=\frac{\Omega}{2}\frac{\partial\bar{h}}{\partial\alpha}+\frac{\partial\bar{h}}{\partial\bar{t}}$$

where,

$$\beta=\frac{rsin\gamma}{R_j}\ ,\bar{p}=\frac{p}{p_s}\ ,\bar{h}=\frac{h}{C}\ ,\qquad \bar{t}=t\left(\frac{C^2p_s}{R_j^2\mu}\right)$$

$$\phi(N,l_m,\bar{h})=1+\frac{12}{\bar{h}^2l_m^2}-\frac{6N}{\bar{h}l_m}coth\left(\frac{N\bar{h}l_m}{2}\right)$$

$$N=\left(\frac{k}{2\mu+k}\right)^{1/2},\quad l=\left(\frac{\partial}{4\mu}\right)^{1/2}$$

III. STATIC AND DYNAMIC CHARACTERISTICS

Equations for load capacity

$$\bar{F}_r=\bar{F}cos\gamma=-\int_{1-tan\gamma}^{1+tan\gamma}\int_0^{2\pi}\bar{p}cos\gamma d\alpha d\beta$$

$$\bar{F}_x=-\int_{1-tan\gamma}^{1+tan\gamma}\int_0^{2\pi}\bar{p}cos\alpha cos\gamma d\alpha d\beta$$

$$\bar{F}_z=-\int_{1-tan\gamma}^{1+tan\gamma}\int_0^{2\pi}\bar{p}sin\alpha cos\gamma d\alpha d\beta$$

Equations for fluid film stiffness coefficients

$$\begin{bmatrix}\bar{S}_{xx} & \bar{S}_{xz}\\ \bar{S}_{zx} & \bar{S}_{zz}\end{bmatrix}=-\begin{bmatrix}\partial\bar{F}_x/\partial\bar{X}_j & \partial\bar{F}_x/\partial\bar{Z}_j\\ \partial\bar{F}_z/\partial\bar{X}_j & \partial\bar{F}_z/\partial\bar{Z}_j\end{bmatrix}$$

Equations for fluid film damping coefficients

$$\begin{bmatrix}\bar{C}_{xx} & \bar{C}_{xz}\\ \bar{C}_{zx} & \bar{C}_{zz}\end{bmatrix}=-\begin{bmatrix}\partial\bar{F}_x/\partial\dot{\bar{X}}_j & \partial\bar{F}_x/\partial\dot{\bar{Z}}_j\\ \partial\bar{F}_z/\partial\dot{\bar{X}}_j & \partial\bar{F}_z/\partial\dot{\bar{Z}}_j\end{bmatrix}$$

IV. FLUID FILM THICKNESS

The film thickness expression in non-dimensional form for conical journal bearing,

$$\bar{h}=\left(1-\bar{X}_jcos\alpha-\bar{Z}_jsin\alpha\right)cos\gamma$$

V. RESTRICTOR FLOW EQUATION

The constant flow valve restrictor should be able to supply a fixed quantity of lubricant flow through it, hence the flow $\bar{Q}_R$ of lubricant flow through it is expressed as

$$\bar{Q}_R=constant=\ \bar{Q}_c$$

Here, $\bar{Q}_R$ and $\bar{Q}_c$ represent the restrictor flow and pocket flow, respectively.

VI. BOUNDARY CONDITIONS

1. Flow of lubricant through the constant flow valve is equal to the bearing input flow.
2. All the nodes situated on the recess have equal pressure.
3. Nodes situated on the external boundary of the bearing have zero pressure.
4. At the trailing edge of positive region, The Reynolds condition is applied i.e.

$$\bar{p}=\frac{\partial\bar{p}}{\partial\alpha}=0.0$$

VII. SOLUTION PROCEDURE

The 2-D meshing of the six pocket hydrostatic conical journal bearing, with various nodes and elements is shown in Fig.2.

The solution of a hydrostatic conical journal bearing system requires the solution of the micropolar fluid flow equation with the restrictor flow equation as constraint and appropriate boundary conditions. The modified Reynolds equation governing the flow of the micropolar lubricant is solved along with restrictor flow equation by finite element method so as to obtain fluid film pressures.

The iterative procedure is repeated until the converged solution for the fluid film pressure field is obtained. The input data file is prepared for the computation of the static performance characterizes of the hydrostatic conical journal bearing. The module INPDAT reads input data for 2D mesh. The subroutine FLDTHK calculates fluid film thickness at nodal points for the tentative values of the journal center position. In subroutine FEQN, fluidity matrix is generated and globalized as per the connectivity of the elements. The subroutine CALL_BOUNDARY modifies the system equation for specified boundary conditions. The modified Reynolds equation is solved for the fluid film nodal pressure in subroutine CALL_SOLVER using the Gaussian elimination technique. The fluid film reactions $\bar{F}_x$ and $\bar{F}_z$ are calculated. Using the values of $\bar{F}_x$ and $\bar{F}_z$, the correction in journal center is established and also the journal center equilibrium position is established by an iterative method and the subroutine SPC calculates the bearing static performance characteristics of the journal bearing system.

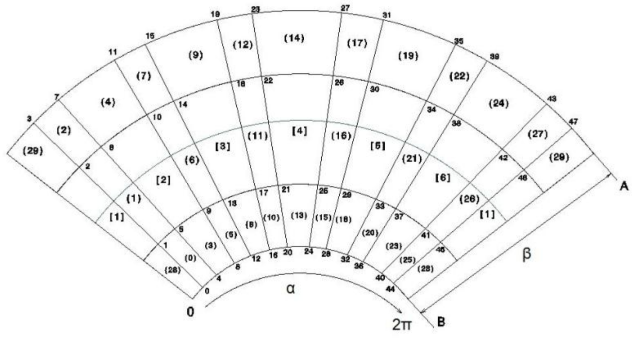

Fig.2: Discretized 6 pocket hydrostatic conical journal Bearing

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VIII. DISCUSSIONS OF RESULTS

The mathematical model developed is used to compute the performance characteristics of a constant flow valve compensated 6 pocket conical hydrostatic journal bearing. Since, for hydrostatic bearing design it must be possible to support the full operating load at zero speed, the results of zero speed are presented here. The results are computed for the following dimensionless values of bearing operating and geometric parameters:

$$\Omega = 0.0, \lambda = 1.0, \theta = 18^0, A_p/A_b = 0.333,$$
$$N^2 = 0.5, l_m = 20, \gamma = 10^0 - 40^0$$

Table 1: Validation of capillary compensated micropolar conical journal bearing ($l_m = 20$)

N^2	Comparisons	$\bar{p}_{max}$	$\bar{h}_{min}$	$\bar{Q}$	φ
0.4	Present	0.643	0.889	0.909	60.4
0.4	Ref [9]	0.650	0.880	0.890	59.0
0.5	Present	0.650	0.891	0.896	60.5
0.5	Ref [9]	0.650	0.890	0.880	59.0

The validation of the code for 4-pocket is shown in Table 1. The results are in good agreement with the literature. The variations of maximum fluid pressure($\bar{p}_{max}$), minimum fluid film thickness ($\bar{h}_{min}$), direct stiffness and damping coefficients with radial load ($\bar{W}_r$) have been plotted in Figs. 3 - 8. The $\bar{p}_{max}$ is seen to be increased for 6-pocket "2" than 4-pocket "1" with increase in $\bar{W}_r$. Whereas, the fluid film thickness $\bar{h}_{min}$ decreases much for 6-pocket "2" than 4-pocket "1" with increase in the values of $\bar{W}_r$.

The variations of stiffness coefficients ($\bar{S}_{xx}, \bar{S}_{zz}$) and damping coefficients ($\bar{C}_{xx}, \bar{C}_{zz}$) with $\bar{W}_r$ for "1" and "2" are shown in Figs. 5 - 8 for various semi cone angle (γ). The stiffness coefficient $\bar{S}_{xx}$ are found to be higher for "1" than "2" whereas $\bar{S}_{zz}$ are found to be higher for "2" than "1" for various semi cone angle (γ). In the similar way the damping coefficient $\bar{C}_{xx}$ is found to be higher for "1" than "2" and $\bar{C}_{zz}$ is found to be higher for "2" than "1" for all the semi cone angles throughout the radial load considered as can be seen in Figs. 7 & 8.

The variations of maximum fluid pressure($\bar{p}_{max}$), minimum fluid film thickness ($\bar{h}_{min}$), direct stiffness and damping coefficients with constant flow valve parameter ($\bar{Q}_c$) have been plotted in Figs. 9 - 14. The $\bar{p}_{max}$ is seen to be increasing for 6-pocket "2" than 4-pocket "1" with increase in ($\bar{Q}_c$). Whereas, the fluid film thickness $\bar{h}_{min}$ decreases much for 6-pocket "2" than 4-pocket "1" with increase in the values ($\bar{Q}_c$).

The variations of stiffness coefficients ($\bar{S}_{xx}, \bar{S}_{zz}$) and damping coefficients ($\bar{C}_{xx}, \bar{C}_{zz}$) with $\bar{W}_r$ for "1" and "2" are shown in Figs. 11 - 14 for various semi cone angle (γ). The stiffness coefficient $\bar{S}_{xx}$ are found to be higher for "1" than "2" whereas $\bar{S}_{zz}$ are found to be higher for "2" than "1" for various semi cone angle (γ). In the similar way the damping coefficient $\bar{C}_{xx}$ is found to be higher for "1" than "2" and $\bar{C}_{zz}$ is found to be higher for "2" than "1" for all the semi cone angles throughout the $\bar{Q}_c$ considered as can be seen in Figs. 13 & 14.

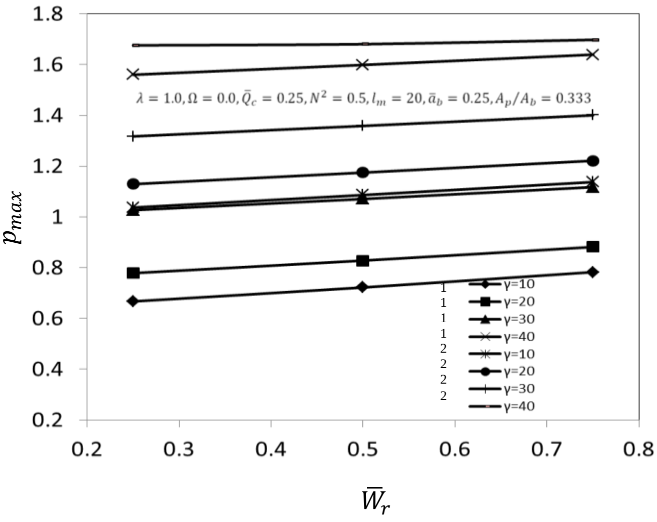

Fig. 3: Variation of maximum pressure vs. load for different cone angle

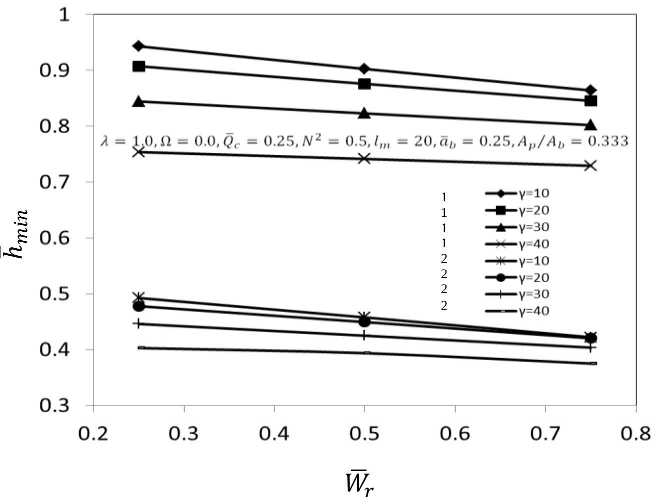

Fig. 4: Variation of minimum film thickness vs. load for different cone angle

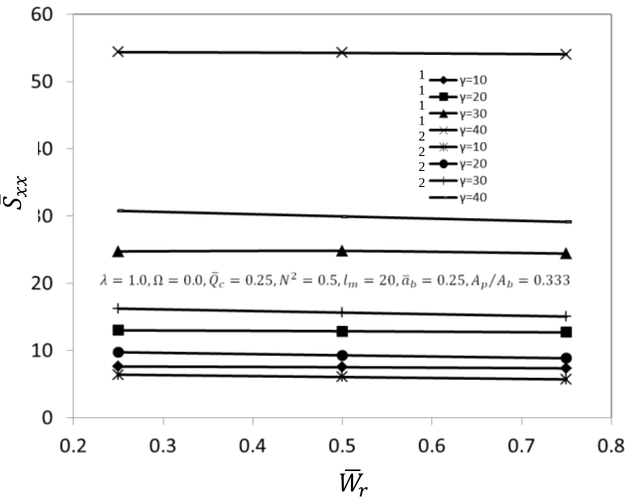

Fig. 5: Variation of direct stiffness vs. load for different cone angle

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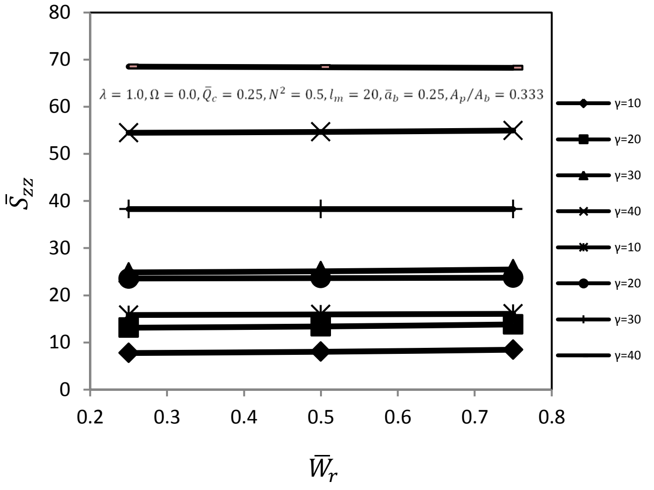

Fig.6: Variation of direct stiffness vs. load for different cone angle

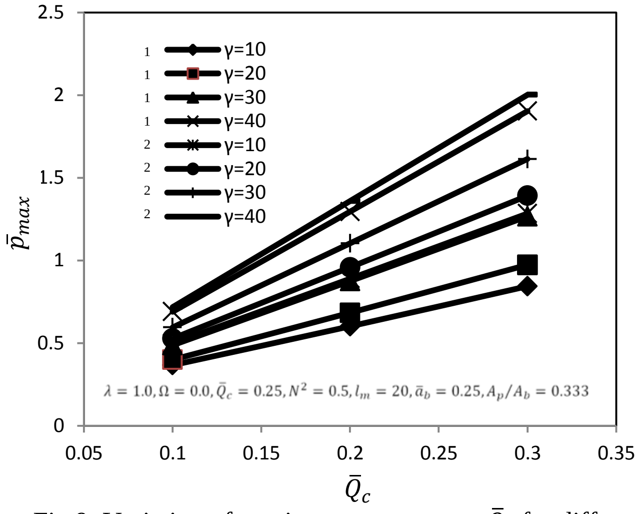

Fig.9: Variation of maximum pressure vs. $\bar{Q}_c$ for different cone angle

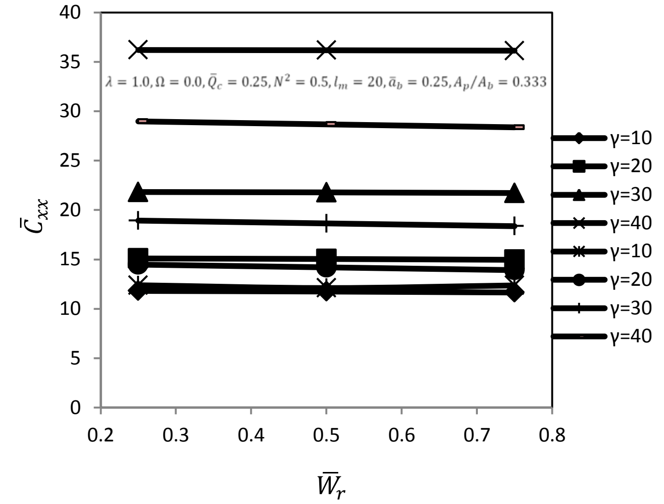

Fig.7: Variation of direct damping vs. load for different cone angle

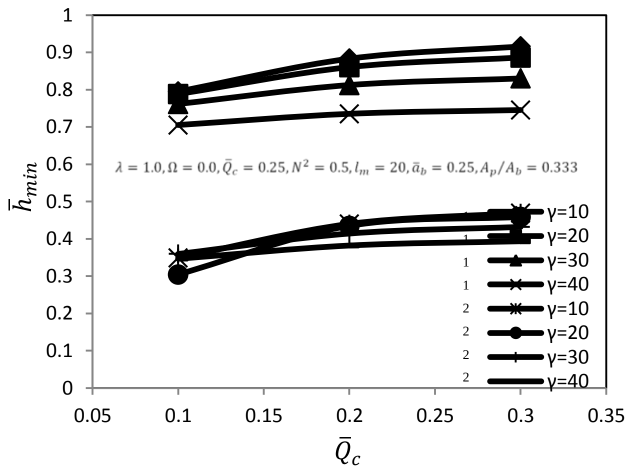

Fig.10: Variation of minimum film thickness vs. $\bar{Q}_c$ for different cone angle

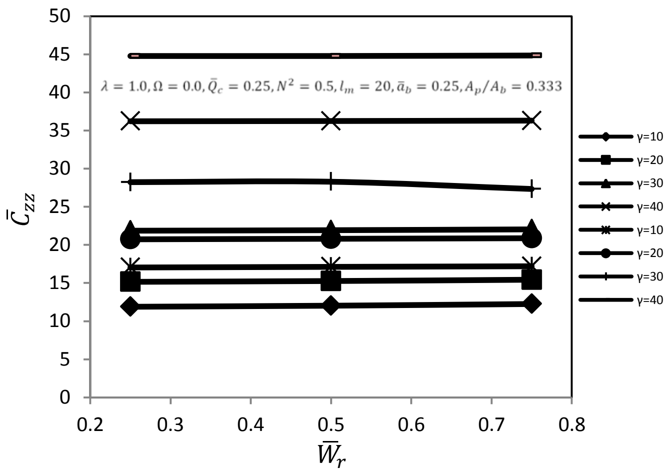

Fig.8: Variation of direct damping vs. load for different cone angle

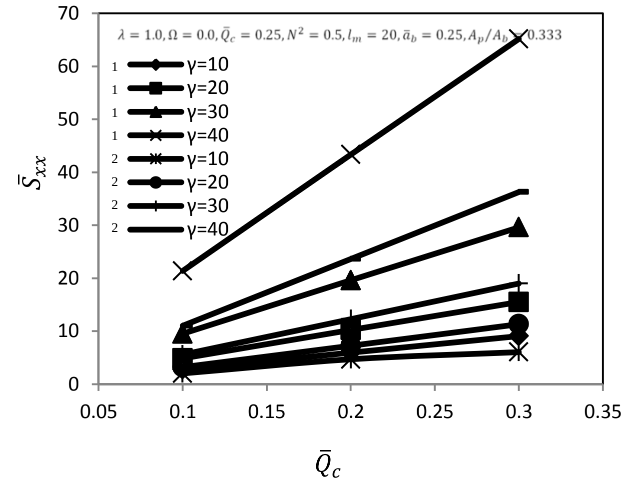

Fig.11: Variation of direct stiffness vs. $\bar{Q}_c$ for different cone angle

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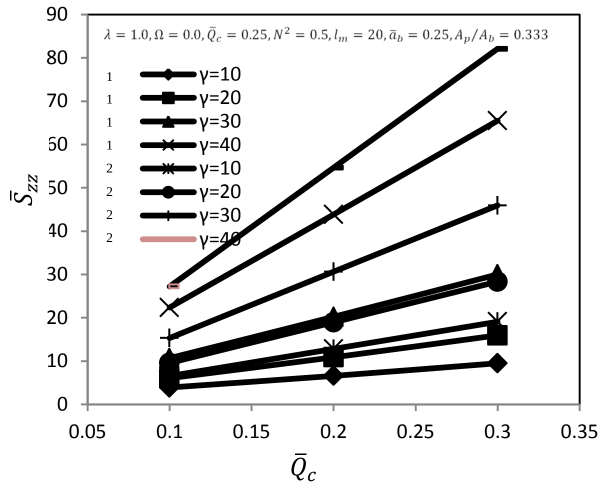

Fig.12: Variation of direct stiffness vs. $\bar{Q}_c$ for different cone angle

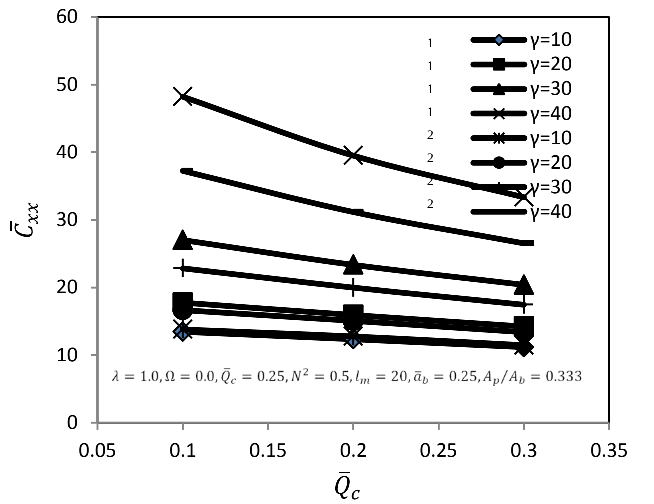

Fig.13: Variation of direct damping vs. $\bar{Q}_c$ for different cone Angle

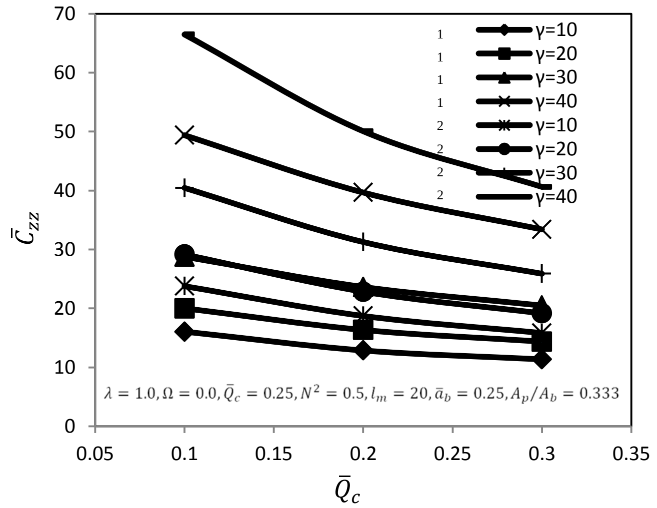

Fig.14: Variation of direct damping vs. $\bar{Q}_c$ for different cone angle

IX. CONCLUSIONS

A 6 pocket hydrostatic conical journal bearing operating under micropolar lubrication compensated with constant flow valve has been analyzed theoretically to determine the effect of pockets variations and semi cone angle on the performance of characteristics of the bearing. The results are presented here for bearing under zero speed condition. It is observed that the bearing develops large pressure with increase in number of pockets and semi-cone angles. Whereas, the fluid film thickness is found to be lower for higher pockets number and semi- cone angle. The stiffness and damping coefficients are also found to be improved.

X. NOMENCLATURE

a_b axial bearing land width, mm

c radial clearance, mm
D_m mean journal diameter of conical shaft, mm
F fluid film reaction , N
h_o outlet film thickness, mm
$l,,l_m$ bearing characteristic
L bearing length, mm
p_s supply pressure, Pa
R_j journal radius, mm
r radial coordinate
X_j , X_z journal center coordinate

Greek symbol

γ semi cone angle
α circumferential coordinate
β axial co-ordinate
μ dynamic viscosity (PaS)
λ aspect ratio, L/D_m
θ inter-recess angle
φ attitude angle, degree
Ω speed parameter
ε eccentricity ratio $e/$c
ϕ micro polar function

Non dimensional number

$\bar{F}_r$ (F/p_s)
$\bar{h}_{mim}$ minimum fluid film thickness
$\bar{p}_{max}$ maximum fluid film pressure
N coupling number
N_i, N_j shape function
$\bar{P}_j$ (p/p_s)
$\bar{Q}$ Q (μ/$c^3 p_s$)
$\dot{\bar{X}}_j, \dot{\bar{Z}}_j$ ($X_j.Y_j$) $/R_j$
$\bar{Q}_c$ constant flow valve restrictor design parameter

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