Hydrodynamic lubrication of a porous slider: shuns simplifying assumption of a small porous facing thickness

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Abstract. This investigation analyzes the lubricating characteristics of a ferrofluid based plane porous slider without the use of the simplifying assumption of a small porous facing thickness. The magnetic fluid flow model of Neuringer and Rosensweig has been adopted here. It is observed that the performance characteristics getting improved owing to magnetization of the lubricant. Further, significant deviations from the past results are observed regarding the ranges of some parameters for which the simplifying assumption yield that satisfactory results.

Keywords: slider bearing, porosity, magnetic fluid, infinite series solution, load carrying capacity.

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1. Introduction

Many authors have conducted investigations on the fluid film lubrication between porous plates (Prakash and Vij [1], Wu [2], Prakash and Vij [3], Bhat and Patel [4]). Sanni and Ayomidele [5] consider hydrodynamic lubrication and porous slider bearing without the use of the signifying assumption of a small porous facing thickness. Solutions were obtained in the form of infinite series. Das [6] studied the optimum load bearing capacity for slider bearings lubricated with couple stress fluids in magnetic field. It was observed that the couple stress effect improved the bearing performance with suitable inlet to outlet film thickness ratio.

Deheri et. al. [7] discussed the performance of longitudinally rough slider bearing with squeeze film formed by a magnetic field. It was noted that the bearing performance was significantly affected by all the parameters characterizing the surface roughness. Ochonski [8] investigated some new design of slider bearing lubricated with ferrofluid exploring the possibility of using them in computer and audio-visual equipment. Agrawal [9] dealt with a porous inclined slider bearing lubricated with a ferrofluid in the presence of an externally applied magnetic field. The load carrying capacity was found to be more as compared to that of a viscous porous inclined slider bearing.

Patel and Deheri [10] studied the effect of surface roughness on the performance of a magnetic fluid based parallel plate porous slider bearing with slip velocity. Here, in spite of the positive effect of a magnetization the slip velocity caused significant load reduction. Further, the friction was unaltered. Lin [11] observed that the performance characteristics of a finite porous slider bearing by using the Brinkmans equation to account for the effect of viscous shearing stress.

Singh and Gupta [12] evaluated the effect of ferrofluid on the dynamic characteristics of a curved slider bearing using Shliomis model based ferrofluid lubrication. It was observed that the effect of rotation of magnetic particle improved the stiffness and damping capacity of the bearing. Bhat and Deheri [13] extended the analysis of Agrawal [9] by considering the magnetic fluid based porous composite slider bearing with its slider consisting of an inclined pad and a flat pad. Here, it observed that magnetic fluid increased the load carrying capacity, unaltered the friction, decreased the coefficient of friction and shifted the centre of pressure toward the inlet. Deheri et. al. [14] discussed transversely rough slider bearings with squeeze film formed by a magnetic fluid. Here, it is concluded that the use of magnetic fluid as a lubricant increases the load carrying capacity, decreases the coefficient of friction and affects the centre of pressure marginally. Further, it is seen that the effect of magnetization[15] on the plane and secant shaped bearings is nominal while the effect of exponential and hyperbolic slider bearings is significant.

Here, it has been sought to study the effect of magnetic fluid lubrication on the configuration of Sanni and Ayomidele [5].

2. Analysis

Geometry and coordinates system of the problem is shown below. With usual

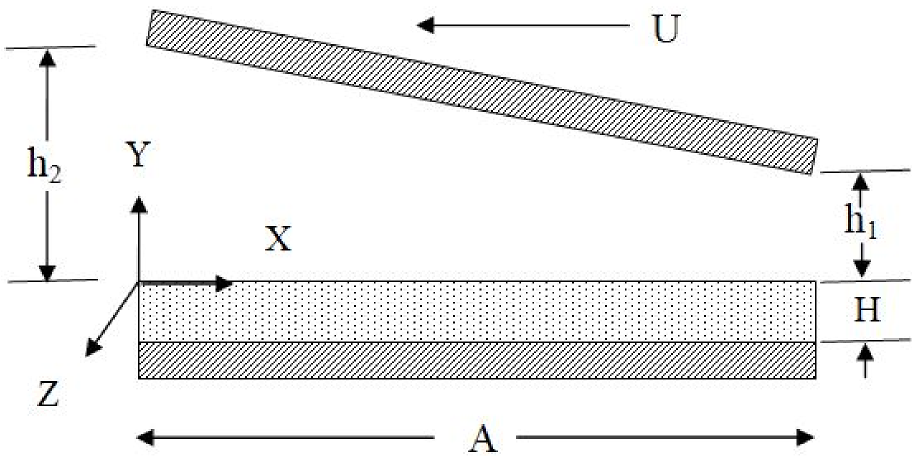

Figure 1: Geometry and coordinates of the bearing system

assumption of conventional lubrication theory, after neglecting the side leakage the modified Reynolds equation is found in the form of equation

$$\frac{\mathrm{d}}{\mathrm{d}x}\left(h^3\,\frac{\mathrm{d}p^*}{\mathrm{d}x}\right) = -6\mu U\frac{\mathrm{d}h}{\mathrm{d}x} + 12\phi\left[\frac{\partial P}{\partial y}\right]_{y=0}. \tag{1}$$

where P the pressure in the porous region, p^* is the pressure in the lubricant region , μ is the absolute viscosity of lubricant , $h = h_2 - (h_2 - h1)\frac{x}{A}$, ϕ is the permeability of the porous material, U is the tangential velocity of the slider.

In the porous region P is govern by the Laplacian equation

$$\frac{\partial^2 P}{\partial x^2} + \frac{\partial^2 P}{\partial y^2} = 0. \tag{2}$$

The boundary condition on p^* and P are

$$p^*(0) = p^*(A) = 0, \tag{3}$$

$$P(0, y) = P(A, y) = 0, \tag{4}$$

$$\left[\frac{\partial P}{\partial y}\right]_{y=-H} = 0. \tag{5}$$

Further, the matching condition for the continuity for the flow at the porous plate film interface is given by

$$P(x, 0) = p^*(x). \tag{6}$$

The pressure distribution in the porous matrix comes out to be

$$P=\frac{2}{\pi}\sum_{n=1}^{\infty}\frac{\overline{p}(n)(e^{n\pi H^*}\sinh n\pi\overline{y}+e^{-n\pi\overline{y}}\cosh n\pi H^*)\sin n\pi\overline{x}}{\cosh n\pi H^*}. \tag{7}$$

The matching condition yields the pressure in the film region to be equation 3. Now substituting this in equation 2 in equation 1 gets

$$p^*(x)=\frac{2}{\pi}\sum_{n=1}^{\infty}\overline{p}(n)\sin n\pi\overline{x} \tag{8}$$

In the view of the magnetic fluid lubrication equation 1 is converted to

$$\frac{\mathrm{d}}{\mathrm{d}x}\big(h^3\ \frac{\mathrm{d}}{\mathrm{d}x}\{p^*-0.5\mu_0\overline{\mu}H^2\}\big)=-6\mu U\frac{\mathrm{d}h}{\mathrm{d}x}+12\phi\big[\frac{\partial P}{\partial y}\big]_{y=0}. \tag{9}$$

where

$$H^2=kx(A-x)$$

k has been suitably chosen constant to manufacture a required magnetic strength. Substituting Equations 7 and 8 into Equation 9, one obtains

$$\begin{aligned}&6nA\frac{\mathrm{d}h}{\mathrm{d}x}\overline{p}(n)h^2\cos n\pi\overline{x}\\&-2\pi n^2\overline{p}(n)h^3\sin n\pi\overline{x}-0.5A^2\mu_0\overline{\mu}k\big[3h^2A\frac{\mathrm{d}h}{\mathrm{d}x}-2kn^3-6kxh^2\frac{\mathrm{d}h}{\mathrm{d}x}\big]\\&=-6A^2\mu U\frac{\mathrm{d}h}{\mathrm{d}x}+24A\phi n\overline{p}(n)\tanh n\pi H^*\sin n\pi\overline{x}.\end{aligned} \tag{10}$$

The constants $\overline{p}(n)$ are calculated using orthogonality of eigen function $\sin n\pi\overline{x}$ as

$$\overline{p}(n)=\frac{24\mu UAH^*\{1-(-1)^n\}(1-r)+4\pi\mu_0\overline{\mu}kA^2H^*nh_1^2D_n}{nh_1^2\{C_nH^*+48\Psi n\pi\tanh n\pi H^*\}}. \tag{11}$$

Where

$$\begin{aligned}r&=\frac{h_2}{h_1},\\D_n&=\big[\frac{r^3-(-1)^n}{n\pi}+\frac{6(1-r)^2[(-1)^n-r]}{n^3\pi^3}\big]\\&\quad-1.5(1-r)\big[\frac{r^2+(-1)^n}{n\pi}-\frac{2(1-r)^2[(-1)^n+1]-8(1-r)[(-1)^n+r]}{n^3\pi^3}\big]\end{aligned}$$

and

$$C_n=n^2\pi^2(1+r+r^2+r^3)+(1-r-r^2+r^3). \tag{12}$$

Substituting the value of $\overline{p}(n)$ in Equation 8 one obtains

$$\begin{aligned} & p^*(x) \\ & = \frac{2}{\pi}\sum_{n=1}^{\infty} \frac{24\mu U A H^*\{1-(-1)^n\}(1-r) + 4\pi\mu_0\overline{\mu}kA^2H^*nh_1^2D_n}{nh_1^2\{C_nH^* + 48\Psi n\pi\tanh n\pi H^*\}}\sin n\pi\overline{x}. \end{aligned} \tag{13}$$

The pressure distribution in non-dimensional form is found as

$$\overline{p} = \sum_{n=1}^{\infty} \frac{[48H^*[1-(-1)^n]](r-1) + 8\pi\mu^* n\frac{A}{h_1}D_nH^*}{n[C_nH^* + 48\Psi n\pi\tanh n\pi H^*]}\sin n\pi\overline{x}. \tag{14}$$

Hence, the load bearing capacity in dimensionless form can be calculated as

$$\begin{aligned} \overline{W} &= \int_0^1 \overline{p}d\overline{x} \\ &= \sum_{n=1}^{\infty} \frac{[48H^*[1-(-1)^n]](r-1) + 8\pi\mu^* n\frac{A}{h_1}D_nH^*}{n[C_nH^* + 48\Psi n\pi\tanh n\pi H^*]} \times \frac{1-(-1)^n}{n\pi}. \end{aligned} \tag{15}$$

3. Result and discussions

It is seen from equation 15 that the load carrying capacity enhances by

$$\sum_{n=1}^{\infty} \frac{8\pi\mu^* n\frac{A}{h_1}D_nH^*}{n[C_nH^* + 48\Psi n\pi\tanh n\pi H^*]}\,\frac{1-(-1)^n}{n\pi} \tag{16}$$

as compare to the conventional fluid based bearing system. This may be due to the effect that magnetization increases the viscosity of lubricant leading to increased pressure and hence the load. A closed observation of equation 15 suggest that the expression is linear with respect to μ^*, accordingly an increased in μ^* will lead to increased load carrying capacity. In the absence of magnetization this investigation reduces to the study of Sanni and Ayomidele [5].

The effect that load bearing capacity increases sharply with respect to magnetization is presented in Figures 2 4. The effect of H^* on the load carrying capacity with respect to μ^* remains negligible (C.f. Figure 2). Besides, the porosity effect up to the porosity value 0.001. Further, the porosity effect up to 0.001 remains either almost nominal or negligible as can be seen from Figures 6 and 7.

Indeed, some amount of fluid enters into the holes and when the surface dries up the fluids comes up to the surface. Therefore, the flow is maintained. But, the pressure decreases (and hence load carrying capacity decreases) because of the obstruction due to the presence of holes. When H^* increases more space is occupied by the fluid which results in increased pressure.

A composition of the result presented here and results of Sanni and Ayomidele [5] indicates that all the cases there appears elevated load.

4. Conclusions

It is established that the results presented here compare well with some of earlier published works. The Neuringer-Rosensweig model based magnetic fluid lubrication results.in significant enhancement of the performance of bearing characteristic. It is revealed that the porosity effect can be neutralized by the positive effect of magnetic fluid lubrication at least for smaller to nominal values of porosity. In addition, this type of bearing systems become some amount load under the absence of flow as well, a properly rarely seen in traditional lubrication.

5. Acknowledgement

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6. Nomenclature

A= length of bearing
h=film thickness = $h = h_2 - (h_2 - h_1)\frac{x}{A}$
h_1=minimum film thickness
h_2=maximum film thickness
r= film thickness ration =$\frac{h_2}{h_1}$
μ_0=permeability of the free space
$\overline{\mu}$=magnetic permeability
μ=absolute viscosity of the lubricant
U= tangential velocity of the slider
$H^* = \frac{H}{A}$
Ψ=permeability parameter= $\phi\frac{H}{h_1^3}$
p^*=Pressure in the lubricant region
H = Bearing wall thickness
ϕ= Permeability of porous material
P = Pressure in porous region
μ^*= Magnetization parameter
$\overline{P} = -\frac{h_1^2}{\mu UA}p^*$, Dimensionless pressure
$\overline{W} = -\frac{h_1^2}{\mu UBA^2}W$ Dimensionless load carrying capacity

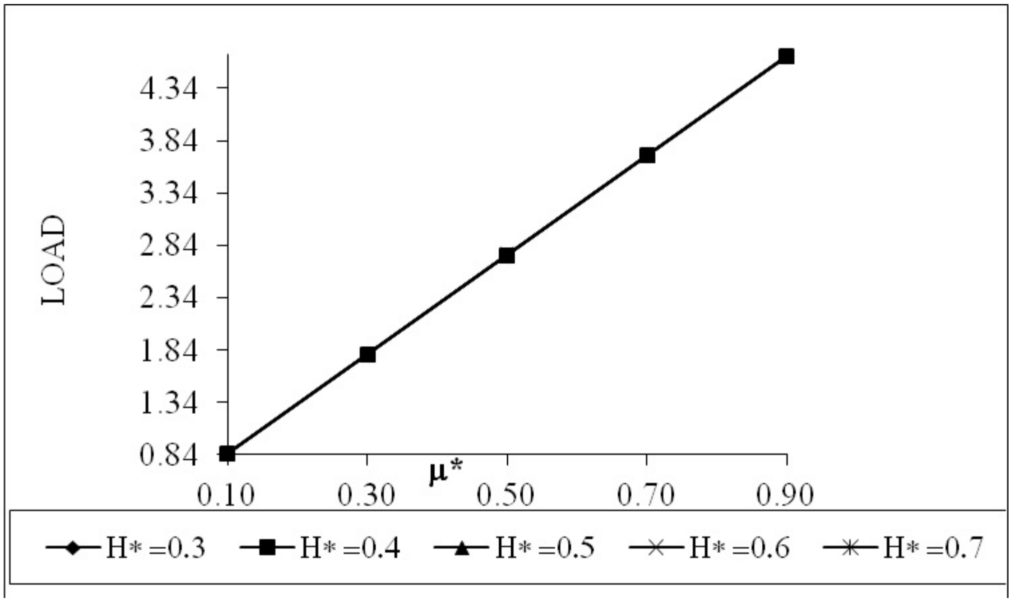

Figure 2: Variation of load caring capacity with respect to $\mu^* and H^*$

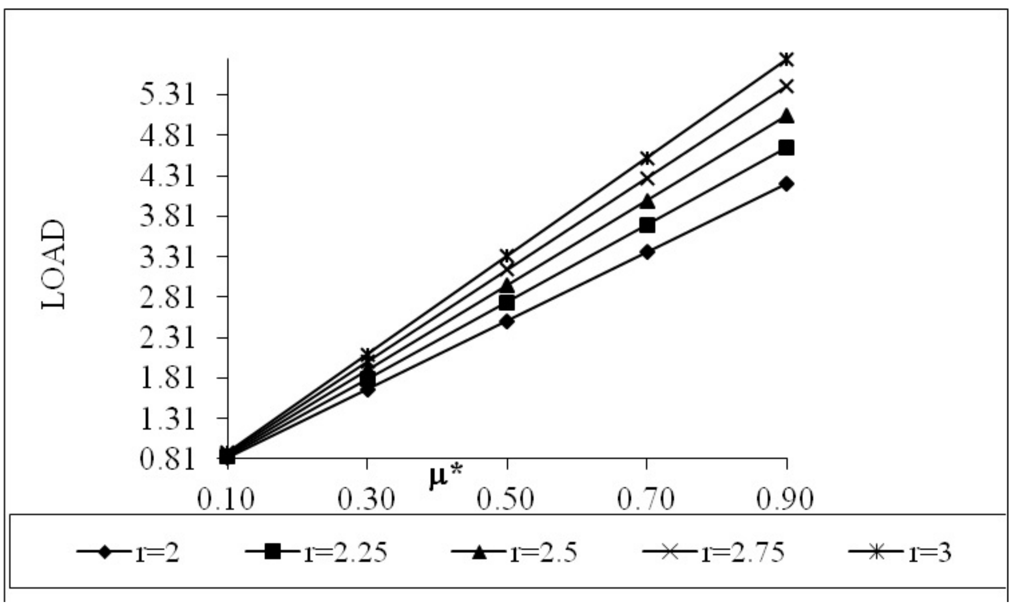

Figure 3: Variation of load caring capacity with respect to μ^* and r

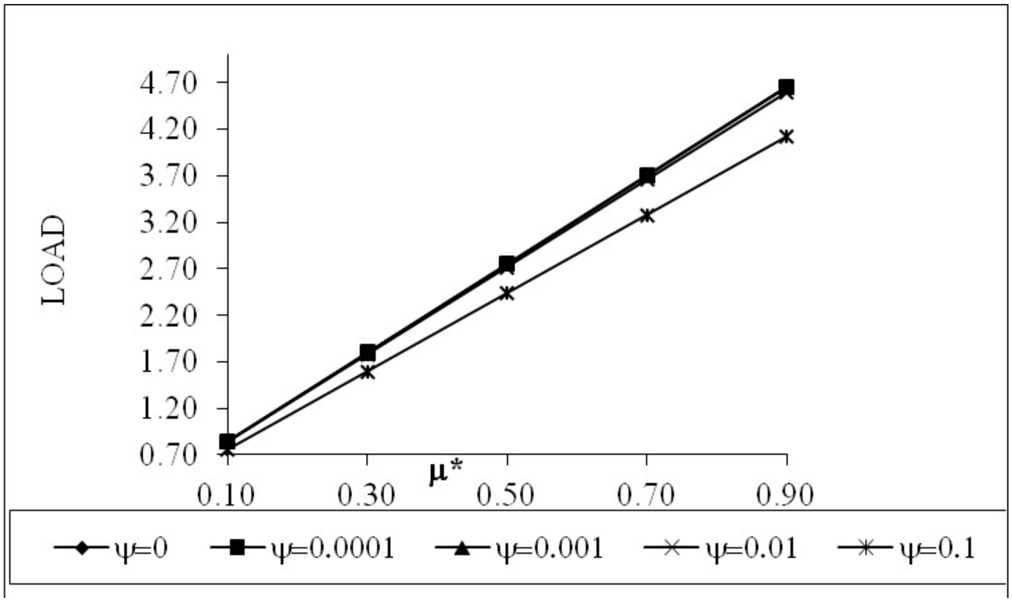

Figure 4: Variation of load caring capacity with respect to $\mu^{*} and \Psi$

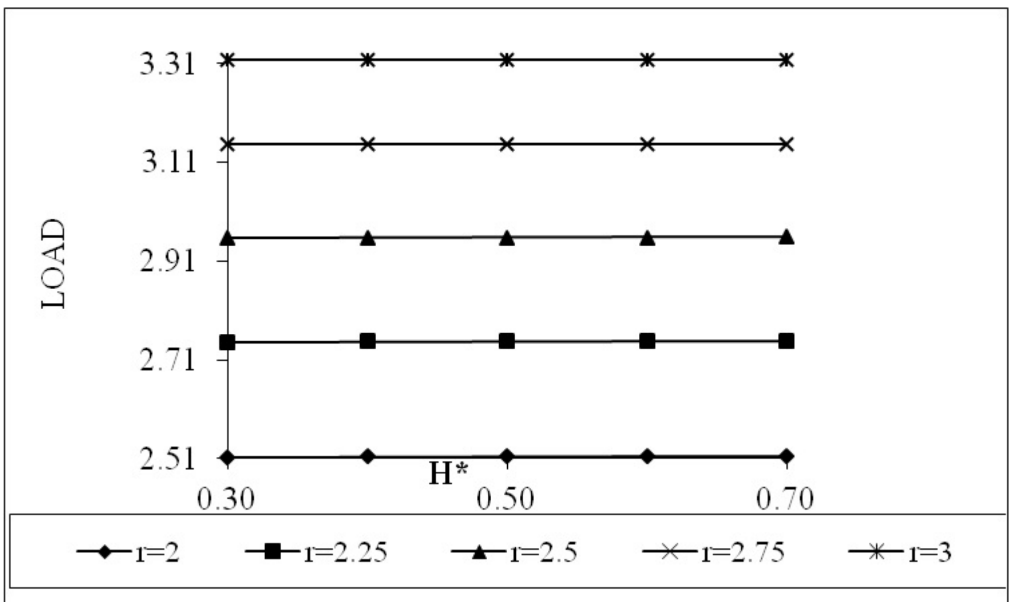

Figure 5: Variation of load caring capacity with respect to H^{*} and r

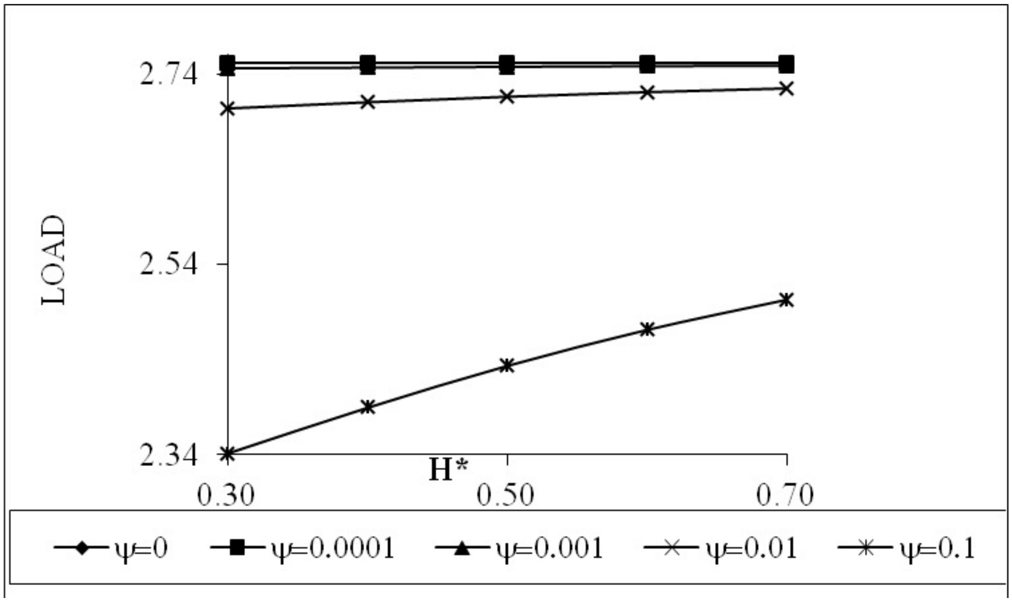

Figure 6: Variation of load caring capacity with respect to $H^* and \Psi$

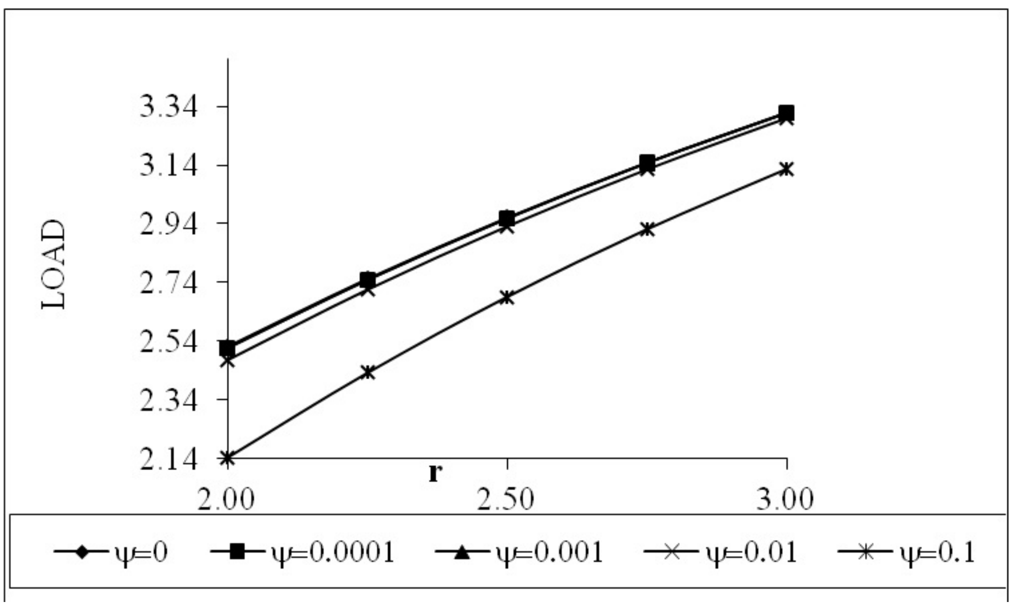

Figure 7: Variation of load caring capacity with respect to $r and \Psi$

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