CAPACITY FORCES IN SLIDE JOURNAL BEARING FOR LAMINAR, UNSTEADY LUBRICATION

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Abstract

The paper presents the results of numerical solutions modified by the Reynolds equation of laminar unsteady lubrication of a cylindrical slide journal bearing. The particular solutions are limited to isothermal models of bearing with infinite length, lubricated by Newtonian oil with the dynamic viscosity dependent on pressure. The disturbances are related with unsteady velocity of oil flow on the sleeve and on the journal. The results are shown in the diagram of hydrodynamic pressure and capacity forces in the dimensionless form in time intervals of displacement duration. In particular modified Reynolds equation and hydrodynamic pressure, capacity forces and numerical results are presented in the paper. The results confirm that the perturbation of velocity influence is stronger when the oil viscosity depends stronger on pressure.

Summary pressure and perturbation pressure change are periodical equal to perturbation of velocity period and the size of change depends on perturbation of velocity. When the perturbation of velocity on the pin has the same direction as peripheral velocity of the pin, the perturbation pressure is positive. When the peripheral velocity is in the opposite direction, the perturbation pressure is negative and it decreases summary pressure. Pressure increase and decrease are not symmetrical during the perturbation time. Presented results will going to be used as a comparison quantities in case of laminar unsteady flow of Non-Newtonian fluids in cylindrical bearing gap.

Keywords: *bearing, lubrication capacity forces, unsteady laminar flow, slide bearing*

1. Introduction

This article refer to the unsteady, laminar flows issue, in which modified Reynolds number $Re^{*}=Re\psi$ is smaller or equal to 2. Laminar, unsteady oil flow is performed during periodic and unperiodic perturbations [4],[5] of bearing load or is caused by the changes of gap height in the time. Above perturbations occur mostly during the starting and stopping of machine. Lubricated oil disturbance velocity the pin and on the bearing shell was also consider in the article. Reynolds equation system describing newtonian oil flow in the gap of transversal slide bearing was discussed in the articles [3],[4]. Mentioned equations were used in this article. Velocity perturbations of oil lubricated flow on the pin can be caused by torsion pin vibrations during the rotary movement of the shaft. Perturbations are proportional to torsion vibration amplitude, frequent constraint and to pin radius of the shaft. Oil velocity perturbations on the shell surface can be caused by rotary vibration of the shell together with bearing casing. This movement can be consider as kinematics constraint for whole bearing friction node. Isothermal bearing model can be approximate to bearing operation in friction node under steady-state thermal load conditions for example bearing in generating set on ship. In bearing calculation operating in pressure of the order of 10 MPa, dynamic viscosity change from pressure was taken into consideration.

2. Modified Reynolds Equation and Hydrodynamic Pressure

The unsteady, laminar and isotherm flow Newtonian oil in journal bearing gap is described for modified Reynolds equation [1],[2] from newtonian oil with constant and variable dynamic viscosity depended for pressure. In considered model we assume small unsteady disturbances and in order to maintain the laminar flow, oil velocity V_i^* and pressure p_1^* are total of dependent quantities $\tilde{V}_i$; $\tilde{p}_1$ and independent quantities V_i; p_1 from time [3], [5] according to equation (1)

$$V_i^* = V_i + \tilde{V}_i \qquad i = 1,2,3$$
$$p_1^* = p_1 + \tilde{p} \tag{1}$$

Unsteady components of dimensionless oil velocity and pressure we [4] in following form of infinite series :

$$\tilde{V}_i(\varphi; r_1; z_1; t_1) = \sum_{k=1}^{\infty} V_i^{(k)}(\varphi; r_1; z_1)\exp(jk\omega_0 t_0 t_1) \tag{2}$$

$$\tilde{p}_1(\varphi; z_1; t_1) = \sum_{k=1}^{\infty} p_1^{(k)}(\varphi; z_1)\exp(jk\omega_0 t_0 t_1)$$

where: i=1,2,3 ;ω_0 – angular velocity perturbations in unsteady flow; j=$\sqrt{-1}$ - imaginary unit.

Reynolds equation describing total dimensionless pressure p_1^* (sum steady and unsteady components) in oil journal bearing gap [1] by unsteady, laminar, isotherm Newtonian flow along with disturbances of peripheral velocity V_{10} on the journal and V_{1h} on the sleeve and disturbances of velocity on journal length V_{30} on the journal and V_{3h} on the sleeve has following form:

$$\frac{\partial}{\partial\varphi}\left\{\frac{(h_1)^3}{e^{Kp_1}}\left[\frac{\partial p_1^*}{\partial\varphi} - K\left(p_1^* - p_1\right)\frac{\partial p_1}{\partial\varphi}\right]\right\} + \frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left\{\frac{(h_1)^3}{e^{Kp_1}}\left[\frac{\partial p_1^*}{\partial z_1} - K\left(p_1^* - p_1\right)\frac{\partial p_1}{\partial z_1}\right]\right\} =$$
$$= 6\frac{\partial h_1}{\partial\varphi} + \frac{1}{2}\frac{\rho_1}{\eta_1}\mathrm{Re}\,Str\,\omega_0 t_0\left\{\frac{\partial}{\partial\varphi}\left[h_1^3\left(V_{10} + V_{1h}\right)\right] + \frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left[h_1^3\left(V_{30} + V_{3h}\right)\right]\right\}\sum_{k=1}^{\infty}A_k + \tag{3}$$
$$-6\left\{\frac{\partial}{\partial\varphi}\left[h_1\left(V_{10} + V_{1h}\right)\right] + \frac{1}{L_1^2}\frac{\partial}{\partial z_1}\left[h_1\left(V_{30} + V_{3h}\right)\right] - 2\left(V_{1h}\frac{\partial h_1}{\partial\varphi} + \frac{1}{L_1^2}V_{3h}\frac{\partial h_1}{\partial z_1}\right)\right\}\sum_{k=1}^{\infty}B_k$$

where $0 \le \varphi \le \varphi_e$; $0 \le r_1 \le h_{p1}$; $-1 \le z_1 \le 1$; $0 \le t_1 \le t_k$; $p_1^* = p_1^*(\varphi; z_1; t_1)$

Dynamic oil viscosity η is depended on pressure by Barrus formula [5] and has following form:

$$\eta = \eta_0 e^{\alpha\left(p - p_a\right)} \approx \eta_0 e^{\alpha p} = \eta_0\eta_1 \tag{4}$$

where: η_o- the dynamic oil viscosity for atmospheric pressure $p = p_a \approx 0$, η – the dynamic oil viscosity function, α – the pressure influence piesocoefficient of the oil viscosity , η_1 – dimensionless dynamic viscosity depending on pressure $\eta_1 = \exp(\alpha p)$.

Components of oil velocity V_φ ,V_r ,V_z in cylindrical co-ordinates r,φ,z have presented as V_1 , V_2 ,V_3 in dimensionless form:

$$V_\varphi = UV_1^* \qquad V_r = \psi UV_2^* \qquad V_z = \frac{U}{L_1}V_3^* \tag{5}$$

where: U – peripheral journal velocity U=ωR; ω – angular journal velocity; R – radius of the journal; ψ– dimensionless radial clearance $(10^{-4} \le \psi \le 10^{-3})$; L_1 – dimensionless bearing length:

$$\psi = \frac{\varepsilon}{R}; \qquad L_1 = \frac{b}{R} \tag{6}$$

where: b – length of the journal; ε – radial clearance.

Putting following quantities: dimensionless values density ρ_1, hydrodynamic pressure p_1^*, dynamic oil viscosity η_1, time t_1, longitudinal gap height h_1, radial co-ordinate r_1

$$\eta = \eta_0\,\eta_1, \quad \rho = \rho_0\,\rho_1, \quad z = b\,z_1, \quad h = \varepsilon\,h_1,$$
$$t = t_0\,t_1, \quad r = R\,(1+\psi\,r_1), \quad p = p_0\,p_1^*, \; K = \alpha p_0, \; p_0 = \frac{U\eta_0}{\psi^2 R^2} \tag{7}$$

Rule of putting dimensionless velocity and pressure quantities in unsteady and steady part of the flow stays similar. Following symbols with bottom zero index signify density, dynamic viscosity, pressure and time describe characteristic dimension values assigned to adequate quantities. Parameter K characterize dimensionless oil dynamic viscosity change from pressure. Dynamic viscosity oil is depended on pressure and presented as sum from steady part and unsteady in dimensionless form [1]:

$$\eta_1^* = \eta_1 + \tilde{\eta}_1; \qquad \eta_1 = e^{Kp_1}; \qquad \tilde{\eta}_1 = K\tilde{p}_1\eta_1 \tag{8}$$

Sum for series $\sum_{k=1}^{\infty} A_k$ and $\sum_{k=1}^{\infty} B_k$ in right side of Reynolds equation (3) are results from conservation of the momentum solutions and were define in work [1],[2]. The equation solution (3) for bearing with infinity length determine total dimensionless hydrodynamic pressure function in following [2]form:

$$p_1^*(\varphi) = \frac{1}{1-Kp_{10}}\left[-\frac{1-Kp_{10}}{K}\ln\left|1-Kp_{10}\right| - p_{10}\left(V_{10}-V_{1h}\right)\sum_{k=1}^{\infty}B_k + \right.$$
$$\left. + \frac{1}{2}\rho_1 \operatorname{Re}^* n\left(V_{10}+V_{1h}\right)\left(\varphi - h_{1e}^3\int_0^{\varphi}\frac{d\varphi}{h_1^3} - K\int_0^{\varphi}p_{10}\,d\varphi\right)\sum_{k=1}^{\infty}A_k\right] \tag{9}$$

$$\text{for} \quad 0 \le \varphi \le \varphi_e; \quad 0 \le t_1 \le t_k; \; p_1^* = p_1^*(\varphi; t_1)$$

Pressure p_{10} is located in the oil gap by steady flow and by constant oil dynamic viscosity. Disturbance pressure in unsteady flow part can be presented with common formula for constant and variable dynamic viscosity $(K \ge 0)$:

$$\tilde{p}_1 = \frac{1}{1-Kp_{10}}\left[\frac{1}{2}\rho_1 \operatorname{Re}^* n\left(V_{10}+V_{1h}\right)\left(\varphi - h_{1e}^3\int_0^{\varphi}\frac{d\varphi}{h_1^3} - K\int_0^{\varphi}p_{10}\,d\varphi\right)\sum_{k=1}^{\infty}A_k - p_{10}\left(V_{10}-V_{1h}\right)\sum_{k=1}^{\infty}B_k\right] \tag{10}$$

where: Re^* - modified Reynolds Number $\operatorname{Re}^* \equiv \psi \operatorname{Re}$, $n=\omega_0/\omega$ multiplication factor determined frequency periodical perturbations and frequency of journal rotation ω.

In disturbances of peripheral velocity case caused by journal bearing torsion vibrations of main engine, n value is equal to number of cylinder c in two-stroke engine or in four-stroke engine to number of cylinders c/2.

3. Capacity Forces

Capacity force W for cylindrical slide journal bearing has following components W_x and W_y to be determined [1] in the local co-ordinate systems in Fig. 1.

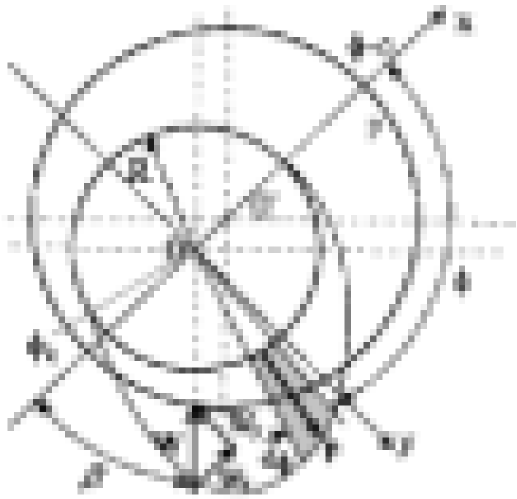

Fig 1: Capacity force W and components W_x and W_y in the local co-ordinate system.

Thus dimensionless components W_{1x} and W_{1y} of capacity forces W_1 are as follows [1]:

$$W_{1x} = \frac{W_x}{W_0} = -\int_0^{\varphi_k} p_1 \cos\varphi d\varphi; \qquad W_{1y} = \frac{W_y}{W_0} = -\int_0^{\varphi_k} p_1 \sin\varphi d\varphi \tag{11}$$

$$W_1 \equiv S_o = \sqrt{W_{1x}^2 + W_{1y}^2} = \frac{W}{W_0}$$

where: W_0 - characteristic value of capacity force $W_0 \equiv 2Rbp_0$
S_o – Sommerfeld Number for slide journal bearing

Hydrodynamic capacity force change caused by the pressure perturbation is calculate from:

$$\tilde{W}_1 = W_1^* - W_1 \tag{12}$$

Capacity force W is situated in the co-ordinate angle φ_w from a angle φ=0 calculated for film origin (Fig. 1):

$$\varphi_w = \pi - \beta = \pi - \text{arctg}\left|\frac{W_{1y}^*}{W_{1x}^*}\right| \tag{13}$$

4. Numerical Results

In numerical calculation example oil with constant density was assume, what is equivalent to quantity ρ_1. In presented calculation way an expression value is assumed $n\rho_1 Re^* = 12$, what is approximately equivalent to force over first frequency torsion vibrations force of six cylinder engine shaft. This take place by laminar unsteady flow. Time of reference t_o is a period of velocity disturbances dispersion. Dimensionless oil gap height for bearing dependent eccentricity λ is described as follows:

$$h_1 = 1 + \lambda \cos\varphi \tag{14}$$

In numerical calculations influence of velocity disturbances on the journal and on the sleeve were analyze. Examples apply to bearing with constant dependent eccentricity λ=0,6. Pressure distribution by wrapping angle and pressure distribution in time function at selected

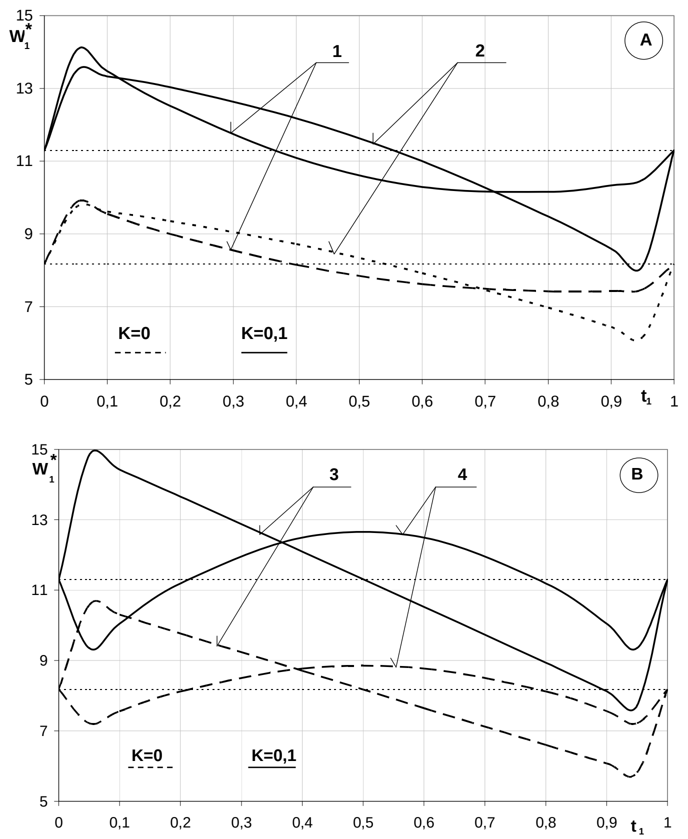

Fig.2 The dimensionless capacity forces W_1^* of slide journal bearing in the time t_1 by velocity perturbations:1) V_{10}=0,05; V_{1h}=0; 2) V_{10}=0,05; V_{1h}=0,025; 3) V_{10}=0,05; V_{1h}=0,05; 4) V_{10}=0,05; V_{1h}=-0,05

point at the journal surface. Numerical calculation results are presented by following tangential velocity perturbations:

1. velocity perturbations on the journal V_{10}=0,05,
2. velocity perturbations on the journal V_{10}=0,05 and on the sleeve V_{1h}=0,025,
3. velocity perturbations on the journal V_{10}=0,05 and on the sleeve V_{1h}=0,05,
4. velocity perturbations on the journal V_{10}=0,05 and on the sleeve V_{1h}=-0,05

Pressure distribution for all four mentioned alternative velocity disturbance are presented in work [2]

Pressure in the bearing during the perturbation is a total of stationary flow pressure and perturbation pressure according to (2) According to mentioned equation (2) if we provide stationary flow pressure p_1 we will obtain capacity force W_1. On the other hand if we provide summarise pressure p_1^* we will receive capacity force W_1^* as a result from this distribution Diagram 2A and 2B presents hydrodynamic capacity W_1^* in the time function t_1 for perturbation velocities cases marked with figure 1 and 2 (2A) and with figure 3 and 4 (2B). Diagram 2 also presents capacity calculation results for oil with constant viscosity (K=0). Capacity force in stationary flow is marked by horizontal lines with dots. Hydrodynamic capacity force W_1^* changes periodically with a period equal to perturbation velocity. In case of velocity perturbation in the bearing pin, increase of capacity force above the stationary condition value last no longer than half of the perturbation period and the increase of capacity force is bigger than the decrease in the remaining time. When perturbation of velocity on the bearing pin is in the same direction as a peripheral velocity of the pin it causes then bigger increase of capacity force than decrease. It is opposite in case of oil peripheral velocity perturbation on the shell surface, but his diagrams

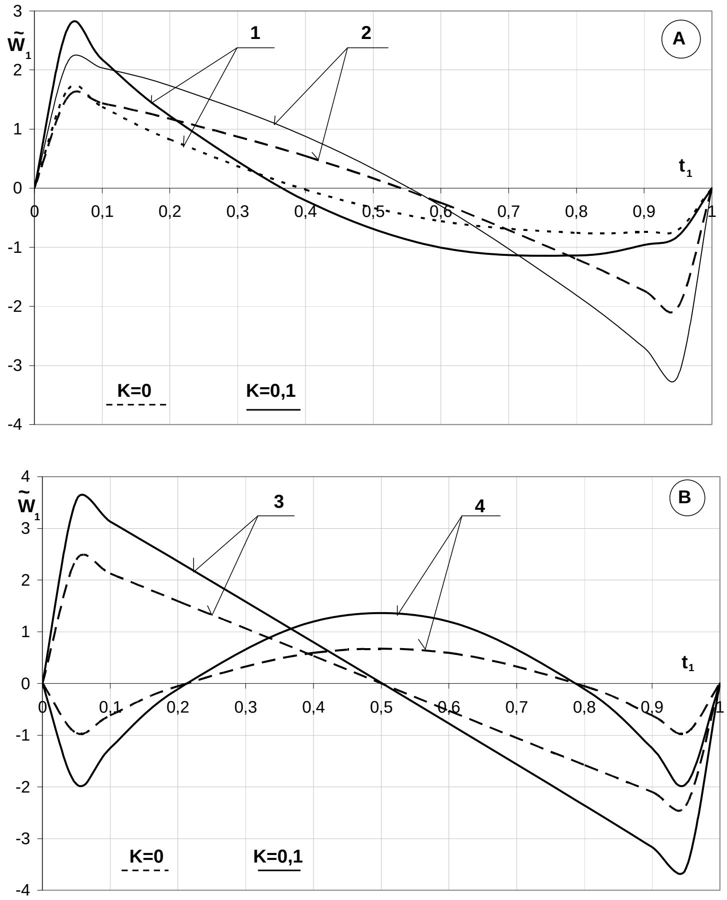

Fig.3 The dimensionless capacity forces $\tilde{W}_1$ of slide journal bearing in the time t_1 by velocity perturbations:: 1) V_{10}=0,05; V_{1h}=0; 2) V_{10}=0,05; V_{1h}=0,025. 3) V_{10}=0,05; V_{1h}=0,05; 4) V_{10}=0,05; V_{1h}=-0,05

are not presented in his article. The case 2 effects are shown on the diagram 2. Capacity force course in time is not symmetrical for different perturbation of velocity quantities on the pin and on the shell. . Presented modification of dimensionless capacity force W_1^* illustrate also a change of Sommerfeld Number So in the bearing node model. Diagram 2 presents hydrodynamic capacity force change in the four mentioned alternatives of velocity disturbance. In the case where disturbance have the same values but the signs are different, the force change has different character (Diagram 4). In 3 and 4 alternatives both diagram point out the symmetry with reference to disturbance time. Diagram 4 presents contact angle of a bearing hydrodynamic capacity force in dimensionless time function t_1 in two considered perturbation of velocity cases marked same as before. By stationary flow angle is marked with dot line. φ_w angle defines capacity force position changes

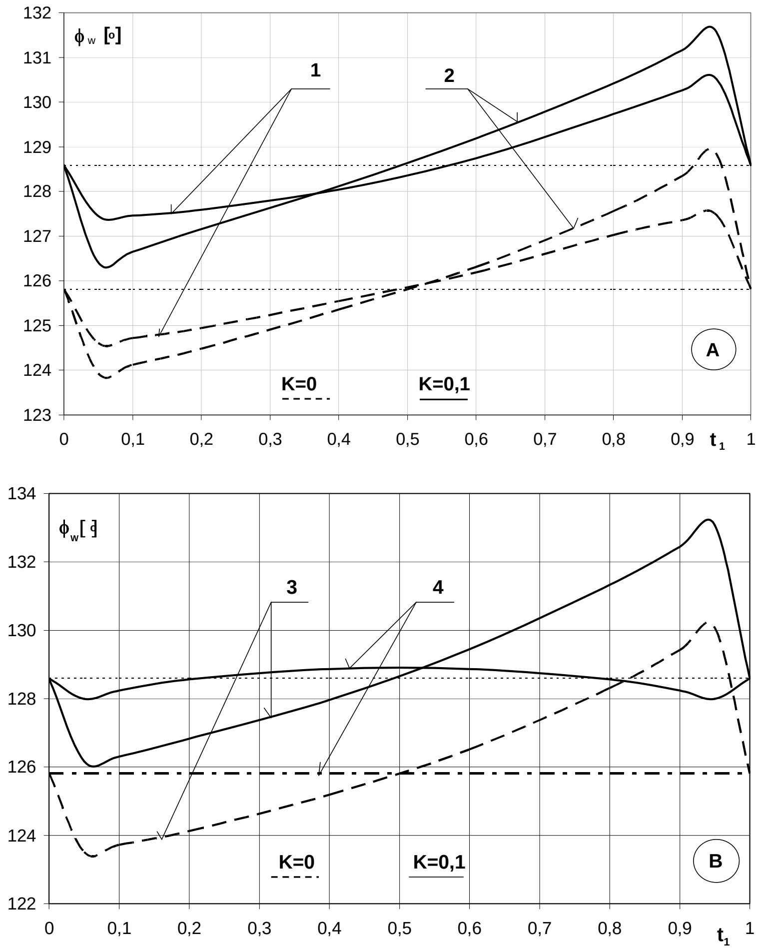

Fig.4 Angle φ_w situated capacity force W_1^* in slide journal bearing in the time t_1 by velocity perturbations: 1) V_{10}=0,05; V_{1h}=0 2) V_{10}=0,05; V_{1h}=0,025. 3) V_{10}=0,05; V_{1h}=0,05; 4) V_{10}=0,05; V_{1h}=-0,05

periodically and the period is equal to perturbation of velocity period. In all considered cases of perturbation velocity the φ_w angle change in time is not bigger than 4 degrees. Only in case 4 (Diagram 4 Fif.4B) for oil with constant viscosity the position angle is uniform (constant). Capacity force angle change for viscosity in dependence on pressure is insignificant comparing to change with constant viscosity and equals below one degree. Viscosity dependence on pressure causes angle φ_w increase for stationary flow.

5. Conclusions

Unsteady perturbation of velocity on the pin have influence on the hydrodynamic pressure distribution in the lubricated gap. The influence is stronger when the oil viscosity depends stronger on pressure. Summary pressure and perturbation pressure change are periodical equal to perturbation of velocity period and the size of change depends on perturbation of velocity. When the perturbation of velocity on the pin Has the same direction as peripheral velocity of the pin then the perturbation pressure is positive. When the peripheral velocity is in the opposite direction than the perturbation pressure is negative and it decrease summary pressure. Pressure increase and decrease is not symmetrical during the perturbation time. Although presented case consider isothermal bearing model with infinity width the results can be useful in preliminary pressure distribution estimation by laminar unsteady lubrication of cylindrical journal bearings with infinity width. Presented results will going to be used as a comparison quantities in case of laminar unsteady flow of Non-Newtonian fluids in cylindrical bearing gap.

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