



Parametric Study of Aerostatic Bearing Performance Analysis with CFD Validation

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Abstract: Aerostatic bearing is used in the recent times for the high precision applications like moment of inertia (MOI) measurement system, coordinate position measuring machine (CMM), dentist pneumatic drill, high speed train etc. The advantages of the aerostatic bearing application is it provide friction free movement between the static and rotating part and is also cleaner in the operation system compare to the conventional ball and oil bearing. The present study of this aerostatic bearing design has been considered with a centrally feeded air hole which is connected through an external compressor. The total supported load acts coaxially on the bearing rotor. The inlet mass flow rate goes vertical air-hole then radial gap between the bearing stator and rotor towards the atmosphere. The typical stator design has two circular pockets, one is at the centre of the stator and another just before the outlet. The objective of the present investigation is the parametric study of a stable bearing design which supports a heavy payload. The air bearing design modelling is achieved by the numerical analysis in the Matlab -Simulink platform. The parametric study involved the different pressure supply with force capability, bearing stiffness, pressure profile at the gap etc. The Matlab developed numerical results is validated through the CFD analysis in Ansys Fluent-16 which matches well with the analytical obtain results.

Keywords: air bearing capacity, stiffness, analytical analysis, CFD validation.

I. INTRODUCTION

Aerostatic bearing are mainly used in the high precession applications coordinate position measuring machine (CMM), dentist grinding tool, Semiconductor wafer processing ,computer and laptop hard disk drive for the friction less and clean in operating condition[1-2].The very recently design aerostatic bearing for moment of inertia measurement system for heavy payload support during measurement air-bearing frictional free torsional oscillation.[3] The oil bearing would provide better stiffness but operation system not provide clean so new research dimension of air bearing [4]. The inherent orifice resistance flow of the bearing gap play the role of the bearing stability [5].The aerostatic bearing with multi-pocket orifice restrictor performance investigated by Yuntang et al[6]. The various solution techniques for the finding the pressure variation thin-film lubration stady and transient investigated by Al-bender [7].The rectangular aerostatic bearing with centrally air hole feeding type of a computer slider has been analysed by Karkoub and Klkamel [8]. The experimentally observed in an external pressurised air bearing the pneumatic hammer

can easily avoided by using small diameter orifices [9]. The CFD method aerostatic bearing performance investigated by Mohamed E and Eleshaky [10] .They observed that pressure and load capacity decreases in the present of the bearing recesses as well as bearing stability performance. The counterbalance type aerostatic bearing analytical steady and transient model has been developed using the bearing passages resistance analytical steady and model validated through CFD analysis [11].The recently simple perturbation flow of high thrust air-bearing model has been developed and validated through CFD moving mesh dynamic method [12].The objectives of the parametric study of the air-bearing analytical model and validated through the CFD analysis.

II. AIR BEARING GEOMETRY DESCRIPTION AND MODEL DEVELOPMENT ASSUMPTION

The figure 1 shows the schematic view of the air bearing design model. The total load (W) means payload weight plus the external weight which are acting coaxially of the bearing axis. The centrally feed

air hole whose inlet is connected through external supply pressure p_c and the mass flow rate is \dot{m} . The steeped feeder air hole design is with radius r_1 for the length L_1 and the rest of the length has radius is r_2 . The air gap between the stator and payload support are two circular pockets. One is the inner circular pocket which radius is r_3 and air gap is (h_1+h_2) another pocket which located just before the outlet which radius is r_5 air gap is (h_1+h_2) rest of the radial direction air gap is h_1 . The mass flow rate goes through the air hole to the radial

direction gap between stator and payload to the atmosphere. During mathematical modelling the top gap different radial location r_2, r_3, r_4, r_5 and r_6 the pressure are p_{tin} , p_{tm} , p_{tm1} and p_{tout} respectively is considered. During the mathematical modelling the following assumption are considered the flow is compressible, isothermal, flow between gap is laminar flow, the viscosity is constant and flow of air obeys the ideal gas property.

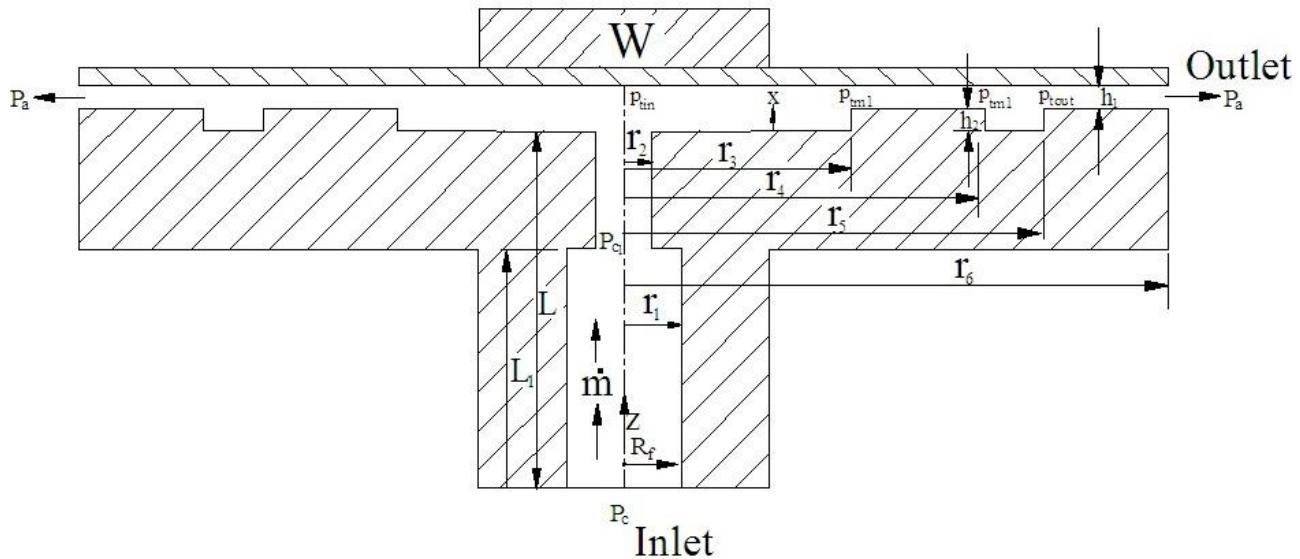


Figure 1: Schematic view of the air bearing model

III. AIR BEARING MATHEMATICAL MODELLING

The velocity of the top air gap in constant temperature condition written as

$$u_r = \frac{k_f x (h - x)}{r h} \quad (1)$$

Where x is the variation of the height from the stator top wall, at wall $x=0$ then $u_r=0$, and $x=h/2$ velocity will be maximum. The unknown parameter k_f is found out from continuity equation (2).

$$\dot{m} = \rho A u_r = \rho \int_0^h 2\pi r dx u_r \quad (2)$$

$$k_f = \frac{3 \dot{m}}{\rho \pi h^2} \quad (3)$$

The substituting the value k_f from equation (1) and find out the corrected velocity profile written as

$$u_r = \frac{3 \dot{m} x (h - x)}{\pi \rho h^3 r} \quad (4)$$

After that apply ideal gas law and then rearranging the equation (4) can written as

$$u_r = \frac{3 RT \dot{m} x (h - x)}{\pi p h^3 r} \quad (5)$$

Then Apply the Navier-Stokes equation with the following assumptions a) Steady State b) Axisymmetric problem and the velocity is only $u_r=f(x)$ Therefore N-S equations simplify to

$$\frac{1}{\mu} \frac{dp}{dr} = \frac{d^2 u_r}{dx^2} \quad (6)$$

$$\frac{1}{\mu} \frac{dp}{dr} = \frac{d^2}{dx^2} \left(\frac{3 RT \dot{m} x (h - x)}{\pi p h^3 r} \right) \quad (7)$$

After rearranging the equation (7) can written as

$$\frac{dp}{dr} = \frac{-6RT \dot{m} \mu}{\pi p h^3 r} \quad (8)$$

Then integrating equation (8) find out the expression for the radial pressure drops in the outer, outer pocket, middle and inner pocket gap segments. The expression for the different gap segments respectively can be written as.

The outer gap segment pressure drops expression

$$\int_{p_{tout}}^{p_a} d(p^2) = \frac{-12RT \dot{m} \mu}{\pi p h^3} \int_r^{r_6} \frac{dr}{r} \quad (9)$$

After performing the integration in the equation (9) and rearranging can be written as

$$p_{tout}^2 = p_a^2 + \frac{12RT \dot{m} \mu}{\pi h_1^3} \ln\left(\frac{r_6}{r}\right)$$

for $r_5 \leq r \leq r_6$ (10)

The outer pocket pressure drop expression can be written as

$$p_{tm}^2 = p_a^2 + \partial p_{tout}^2 + \frac{12RT \dot{m} \mu}{\pi (h_1 + h_2)^3} \ln\left(\frac{r_5}{r}\right)$$

for $r_4 \leq r \leq r_5$ (11)

$$\partial p_{tout}^2 = \frac{12RT \dot{m} \mu}{\pi h_1^3} \ln\left(\frac{r_6}{r_5}\right)$$

and

$$p_{tm1}^2 = p_a^2 + \partial p_{tout}^2 + \partial p_{tm}^2 + \frac{12RT \dot{m} \mu}{\pi h_1^3} \ln\left(\frac{r_4}{r}\right)$$

for $r_3 \leq r \leq r_4$ (12)

and $\partial p_{tm}^2 = \frac{12RT \dot{m} \mu}{\pi (h_1 + h_2)^3} \ln\left(\frac{r_5}{r_4}\right)$

The inner pocket pressure drop can be written as

$$p_{tin}^2 = p_a^2 + \partial p_{tout}^2 + \partial p_{tm}^2 + \partial p_{tm1}^2 + \frac{12RT \dot{m} \mu}{\pi (h_1 + h_2)^3} \ln\left(\frac{r_3}{r}\right)$$

$\partial p_{tm1}^2 = \frac{12RT \dot{m} \mu}{\pi h_1^3} \ln\left(\frac{r_4}{r_3}\right)$

for $r_2 \leq r \leq r_3$... (13) and

For the vertical branch of the hole, the flow is assumed to be a component in the z direction of the hole axis and the hole radius is R_f . The velocity profile through this hole in isothermal condition satisfied the boundary condition and similarly like the previous process find out the pressure difference for the hole the equation can be written as.

$$p_c^2 = p_{c1}^2 + \frac{8RT \mu m L_1}{\pi r_1^4} \quad \text{for } 0 \leq L \leq L_1$$
(14)

$$p_{c1}^2 = p_{ti}^2 + \frac{8RT \mu m (L - L_1)}{\pi r_2^4} \quad \text{for } 0 \leq L \leq L - L_1$$
(15)

The inlet supply pressure is P_c and the outlet is the atmospheric pressure the total mass flow rate from the inlet through the bearing passage can be written as

$$\dot{m} = \left[\frac{p_c^2 - p_a^2}{R_{gh}} \right]$$

Where

$$R_{gh} = \left[\frac{12RT \dot{m} \mu}{\pi (h_1 + h_2)^3} \ln\left(\frac{r_3}{r_2}\right) + \frac{12RT \dot{m} \mu}{\pi h_1^3} \ln\left(\frac{r_4}{r_3}\right) + \frac{12RT \dot{m} \mu}{\pi (h_1 + h_2)^3} \ln\left(\frac{r_5}{r_4}\right) + \frac{12RT \dot{m} \mu}{\pi h_1^3} \ln\left(\frac{r_6}{r}\right) + \frac{8RT \mu m L_1}{\pi r_1^4} + \frac{8RT \mu m (L - L_1)}{\pi r_2^4} \right] \quad (16)$$

IV. THE STATIC PERFORMANCE OF THE AIR BEARING

$$F = \left[p_{tin} \times \pi r_2^2 + 2\pi \left\{ \int_{r_5}^{r_6} p_{tout} r dr + \int_{r_4}^{r_5} p_{tm} r dr + \int_{r_3}^{r_4} p_{tm1} r dr + \int_{r_2}^{r_3} p_{tin} r dr \right\} \right]$$
(17)

The static top gap dynamics of the combined load (W) written as

$$\ddot{h}_1 = g [F - W] / W \quad (18)$$

The air bearing axial stiffness can be defined as the change of force capability with respect to change of air gap, the small change of air gap corresponding to the large variation of the force.

The total net static force can be found by integrating the radial pressure. Since the steady state axial force balance condition can be written as,

$$K = \frac{\partial F}{\partial h_1} \quad (19)$$

V. RESULT AND DISCUSSION

The numerically obtained all the results are presented here, the details of the air bearing configuration and bearing

design parameters are shown in fig.1 and Table 1 respectively.

Table 1: Air bearing design parameters

Symbol	Parameters description	Parameter value	SI unit
h_1	Total air gap	30×10^{-6}	m
h_2	Inner and outer pockets height	0.0014	m
L	Feeder air hole total length	0.035	m
L_1	Air hole length	0.03	m
P_c	Supply pressure	$6 \times 10^5, 8 \times 10^5, 10 \times 10^5$	Pa
P_a	Ambient pressure	1.1×10^5	Pa
R	Gas constant	287	KJ/Kg K
r_1	Feeder hole radius	0.005	m
r_2	Feeder hole radius	0.0001	m
r_3	1 st pocket outer radius	0.015	m
r_4	2 nd pocket inner radius	0.025	m
r_5	2 nd pocket outer radius	0.26	m
r_6	Outer radius	0.35	m
T	Ambient temperature	293	K
μ	Viscosity of air	0.00001820755	Ns/m ²

The total flow path resistance of the air-bearing is shows in fig.2 from the figure shows that resistance decrease with increase of the air gap, from the equation

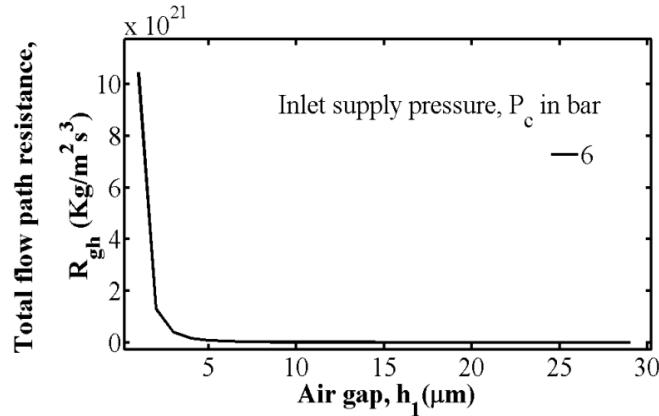


Figure 2: Total flow path resistance variation with air gap for fixed pressure P_c

(16) also it has been observed that the flow path resistance is independent of the inlet supply pressure.

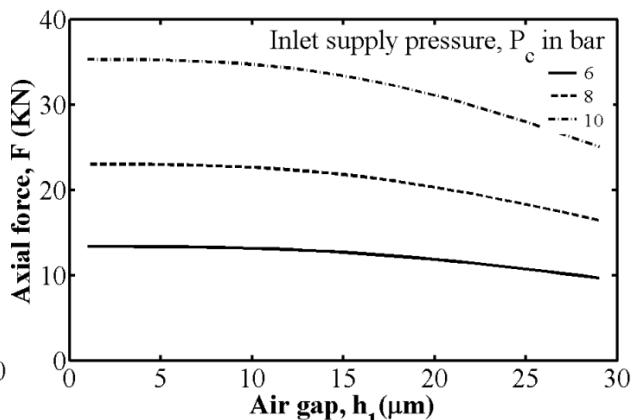


Figure 3: Axial force variation with air gap for different pressure p_c

The effect of the axial force capability in changing the inlet supply pressure for 30 micron air gap shows in fig.3. from this result it has been observed that force capability is increasing with increasing the inlet supply pressure.

The variation of the air bearing stiffness eqn.(19) with changing inlet supply pressure presented in fig.4. from this result it is observed that air-bearing stiffness increase with increase in the supply pressure and for different supply pressure stiffness monotonically decrease with increase in the air gap.

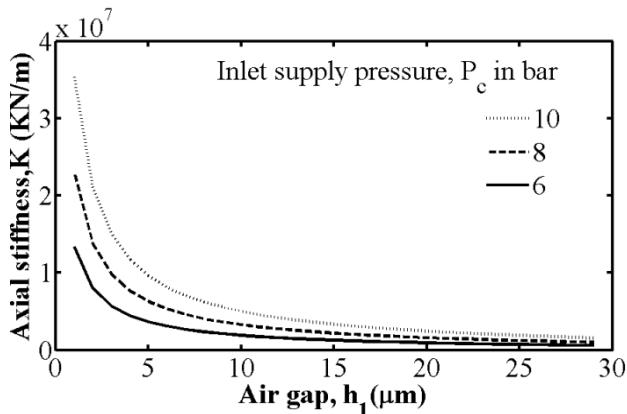


Figure 4: Axial stiffness variation with air gap for different pressure P_c

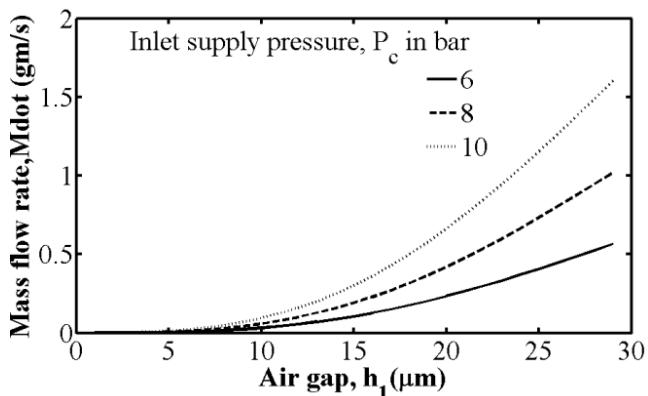


Figure 5: Total mass flow rate variation with air gap for different pressure P_c

The mass flow rate variation for the different supply pressure is shown in fig.5 from this figure it has been shows that increase of the supply pressure mass flow rate increase with the increase in the air gap.

The air gap pressure profile for the different supply pressure shows in fig.6 for 30 micron air gap, from this figure it's clear that radius r_2 to r_3 pressure is constant and radius r_3 to r_4 pressure decreases and radius r_4 to r_5 pressure remain constant again pressure decreasing to reach atmospheric pressure.

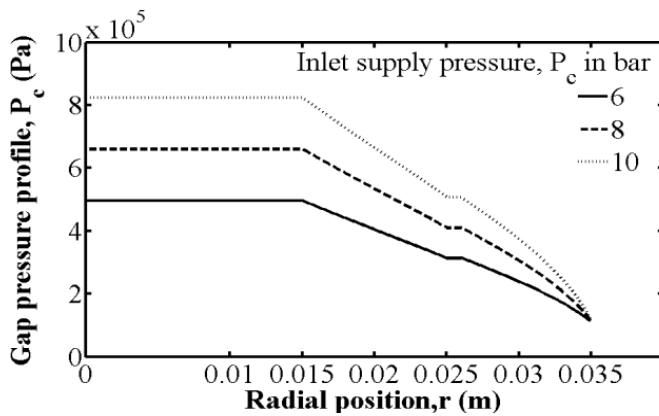


Figure 6: Gap pressure variation with radial location for different pressure P_c

The effect of the increasing outer radius r_6 from 0.03 m to 0.04 m axial force variation shows in fig.7 from figure it has been observed that force capability increasing with the increasing of outer radius.

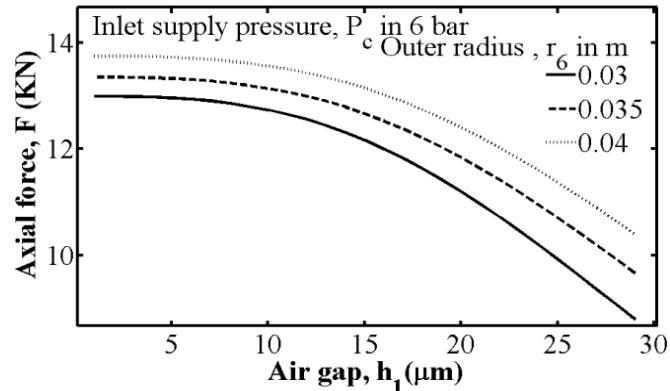


Figure 7: Axial force variation with air gap for changing outer in fixed pressure P_c

The effect of the outer pocket width changing for fixed supply pressure is shows in fig.8.

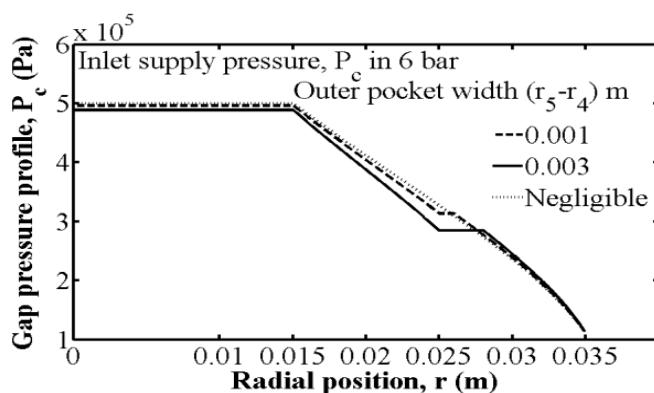


Figure 8: Gap pressure variation with radial location for changing outer pocket width

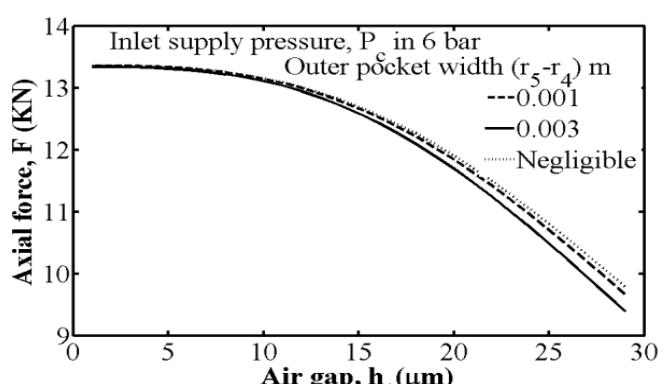


Figure 9: Axial force variation with air gap for changing outer pocket width

The similar effect has been made for the axial force capability is shows in fig.9 it has been seen that the force capability increase for negligible pocket width respect to others.

VI. THE VALIDATION OF THE ANALYTICAL METHOD WITH CFD RESULT

The comparison between the analytical solutions obtained from the Matlab-Simulink platform and the CFD solutions obtained from the Ansys -Fluent 16. In the CFD solution Model it is considered Laminar flow, axisymmetric compressible flow isothermal and no-slip boundary condition the Navier Stokes equation flow air is consider for ideal gas. The second order discretization scheme has been considered. The validation of air bearing model the vertical air hole length L is 0.005 m and radius of inlet r_2 taken 0.0001m. and all other parameter remain same as table 1. The steady flow model is solved considering the inlet supply pressure 7 and 5 bar Matlab and CFD model the atmospheric pressure is the outlet boundary conditions. The total air gap between rotor and stator 30 μ m has been taken in Matlab as well as in CFD model. The validated CFD and analytical solution is very good close matching and acceptable for the analytical model shown in fig.10.

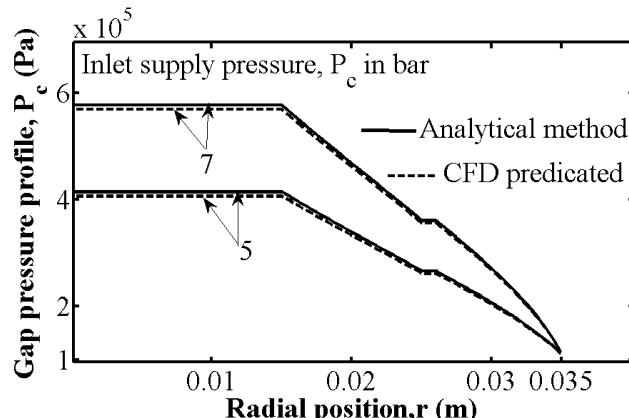


Figure 10: Validation of gap pressure analytical and CFD method

VII. CONCLUSIONS

This paper is mainly focused on the steady analysis of the air bearing performance for varying the different input parameters likes inlet pressure, outer radius, outer pocket width etc. The flowing conclusion has been given below:

1)The air bearing force capability increasing with increasing the outer radius, because increasing the outer radius means increase the net effective area and force is

directly proportional to the effective area so the force capability increase.

2)It has been also observed that when the outer pocket width reduces or eliminate the force capability slightly increases, because reduce or eliminate the outer pocket width pressure loss is reduce or eliminate respectively so the force capability increase.

3)The air bearing axial stiffness increasing with increasing the inlet pressure, because air bearing stiffness is directly proportional to the supply pressure so the increase the supply pressure air bearing stiffness increase.

4)The validation of the CFD and numerical model is accepted for the analytical solution method.

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