Design of gas foil thrust bearing for vertically operated turboexpander used in cryogenic application

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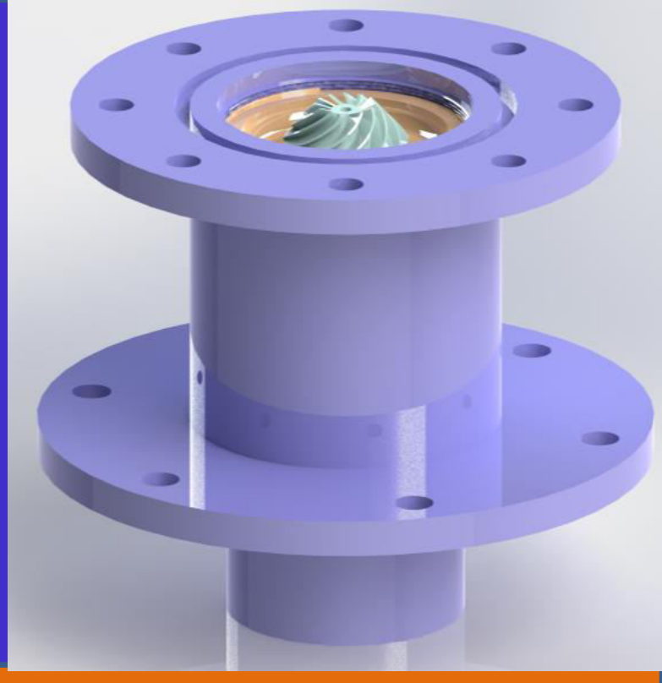
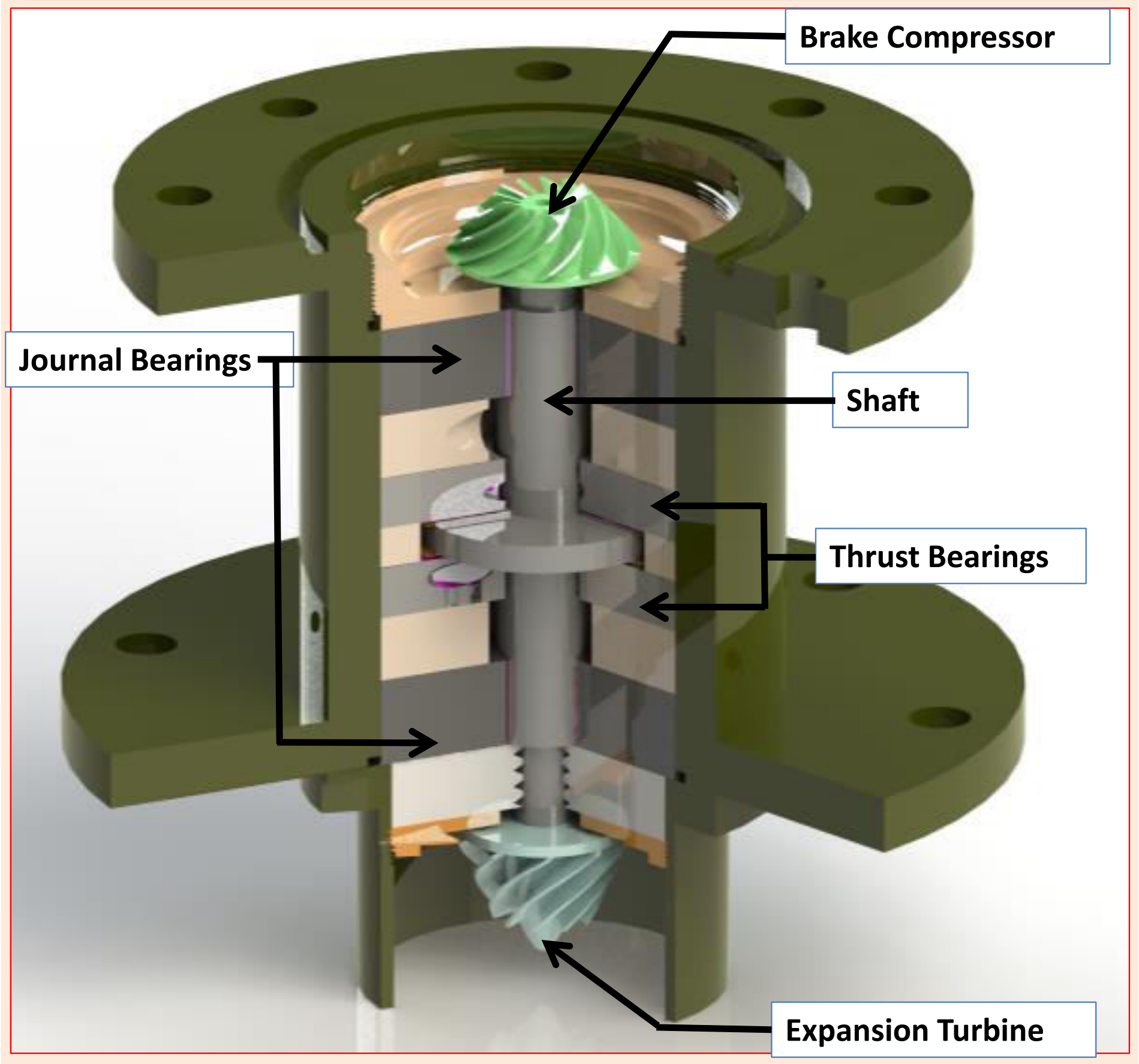
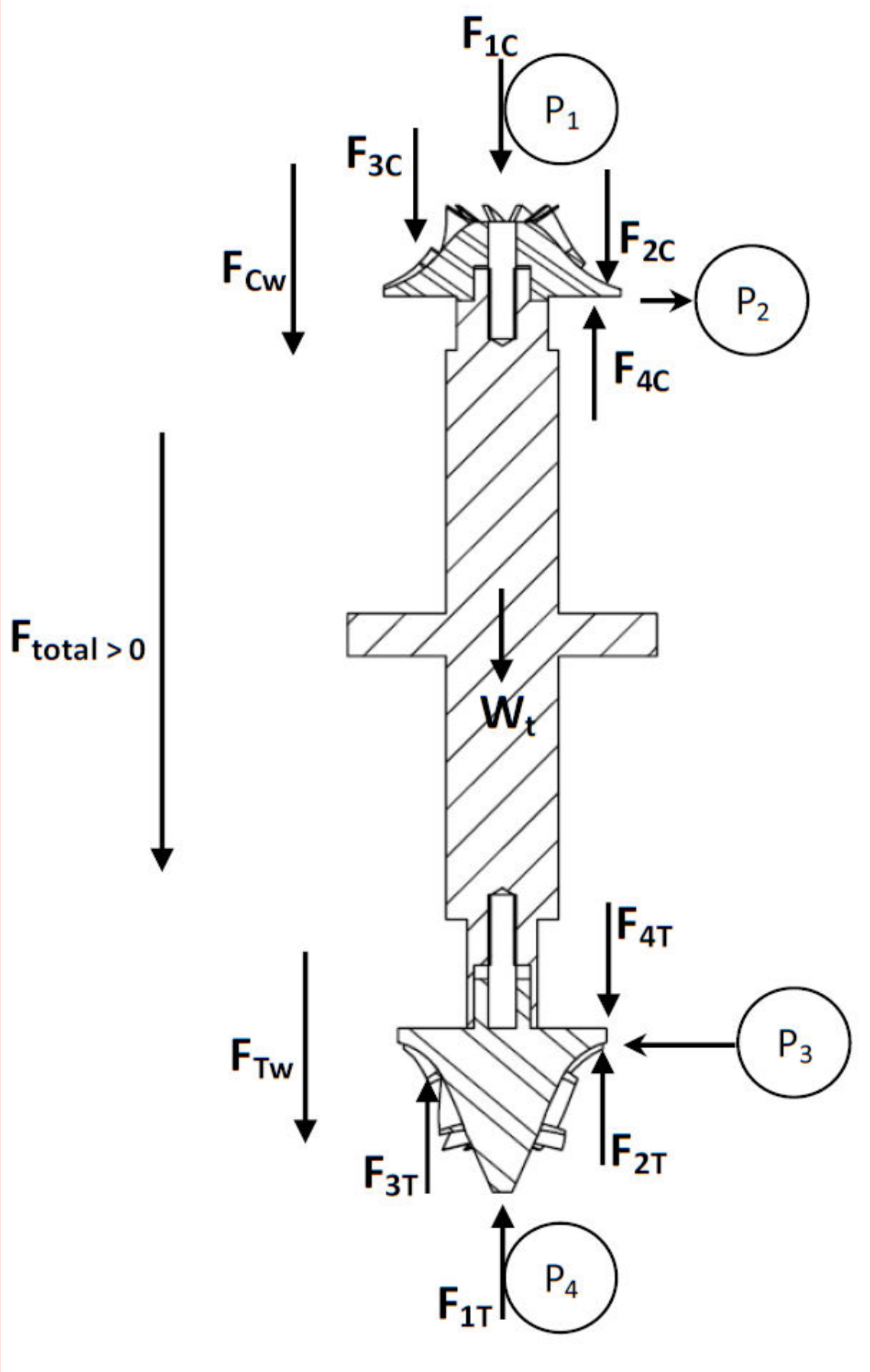


Introduction

Turboexpander is one of the most critical parts of large no of the cryogenic process plant. These turboexpanders in a typical cryogenic refrigerator run at the speed greater than 1, 00,000 RPM. Such operating conditions along with vertical orientation of the rotor impose rigorous constraints on axial or thrust bearing of turboexpander.

Anatomy of Turboexpander





$F_{1C} = A_{IC} p_{IC}$

$F_{2C} = A_{S,C}\left(\frac{p_{IC} + p_{OC}}{2}\right)$

$F_{3C} = \dot{m}_c \left(\frac{\dot{m}_c}{\rho_{IC} A_{IC}}\right)$

$F_{4,C} = A_{OC} p_{OC}$

$F_{1T} = A_{OT} p_{OT}$

$F_{2T} = A_{S,T}\left(\frac{p_{IT} + p_{OT}}{2}\right)$

$F_{3T} = \dot{m}_T \left(\frac{\dot{m}_T}{\rho_{IT} A_{IT}}\right)$

$F_{4,C} = A_{IT} p_{IT}$

Compressor Wheel		Turbine Wheel	
Inlet mean diameter (D_{IC})	12.00 mm	Inlet diameter (D_{IT})	29.60 mm
Outlet diameter (D_{OC})	33.70 mm	Outlet mean diameter (D_{OT})	13.32 mm
Inlet pressure (P_{IC})	3.868 bar	Inlet pressure (P_{IT})	4.319 bar
Outlet pressure (P_{OC})	4.451 bar	Outlet pressure (P_{OT})	1.020 bar
Density of the gas at inlet (ρ_{IC})	4.4235 kg/m^3	Density of gas at the inlet (ρ_{IT})	15.0194 kg/m^3
Mass flow rate ($\dot{m}_c$)	0.05179 kg/s	Mass flow rate ($\dot{m}_T$)	0.07646 kg/s
Mass of the rotor	101.80 ×10^{-3} kg		

Compressor Wheel		Turbine Wheel	
Pressure force at the inlet (F_{1C})	43.75 N	Pressure force at the outlet (F_{1T})	14.21 N
Pressure force over the shroud (F_{2C})	265.66 N	Pressure force over the shroud (F_{2T})	118.66 N
Impulse force (F_{3C})	5.96 N	Impulse force (F_{3T})	0.63 N
Pressure force at the outlet (F_{4C})	397.02 N	Pressure force at the inlet (F_{4T})	297.21 N
Total thrust load at compressor side (F_{CW})	- 81.66 N	Total thrust load at turbine side (F_{TW})	135.93 N

Gas Foil Thrust Bearings(GFTB)

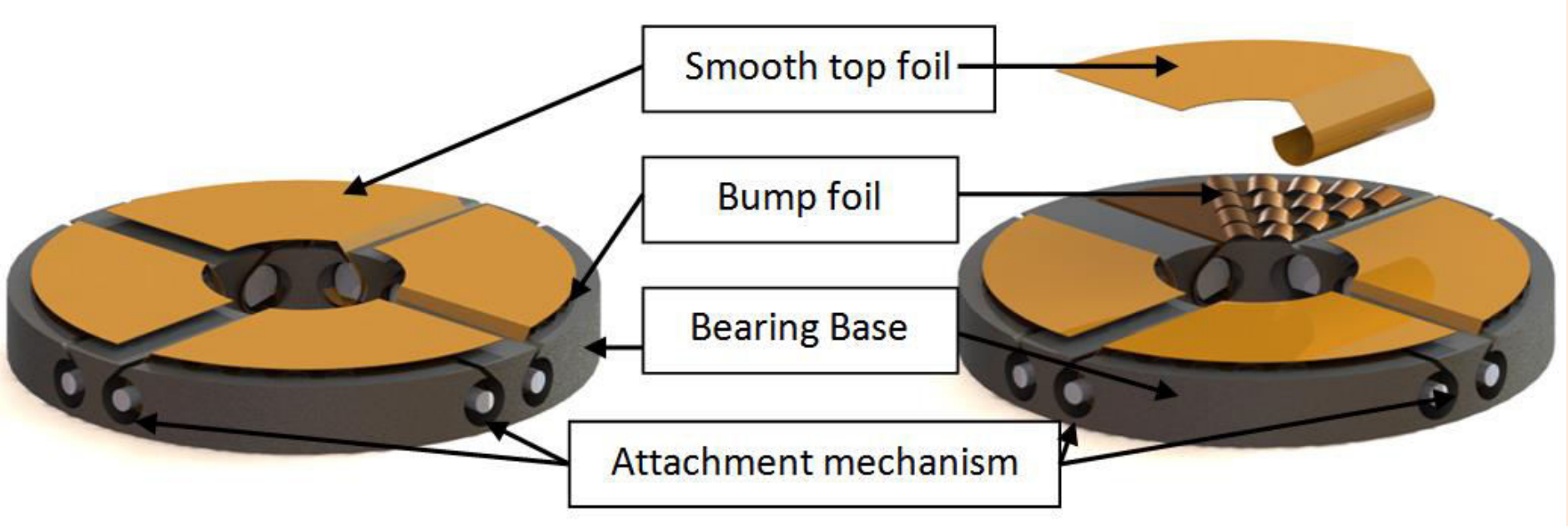
Foil bearings are:

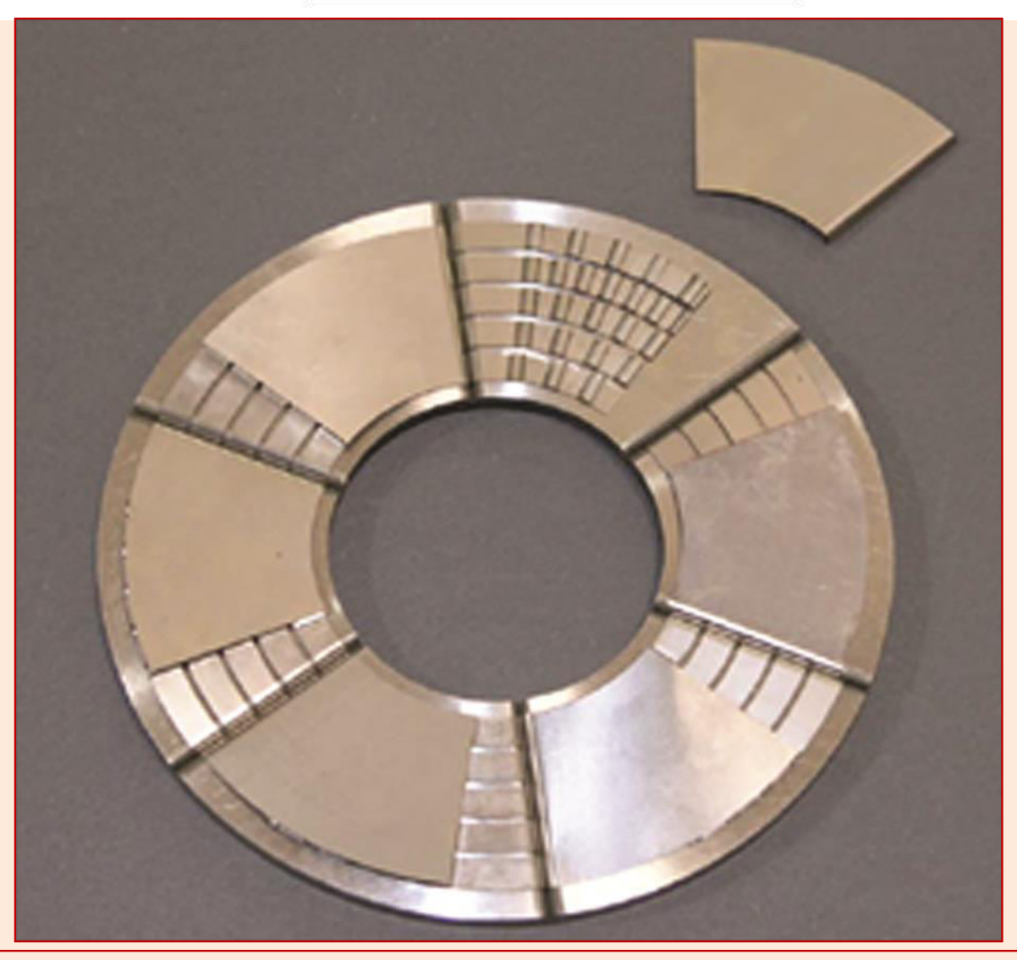
-Self acting bearings(Aerodynamic)

-Compliant bearings.

-Accommodated centrifugal and thermal growth of rotor.

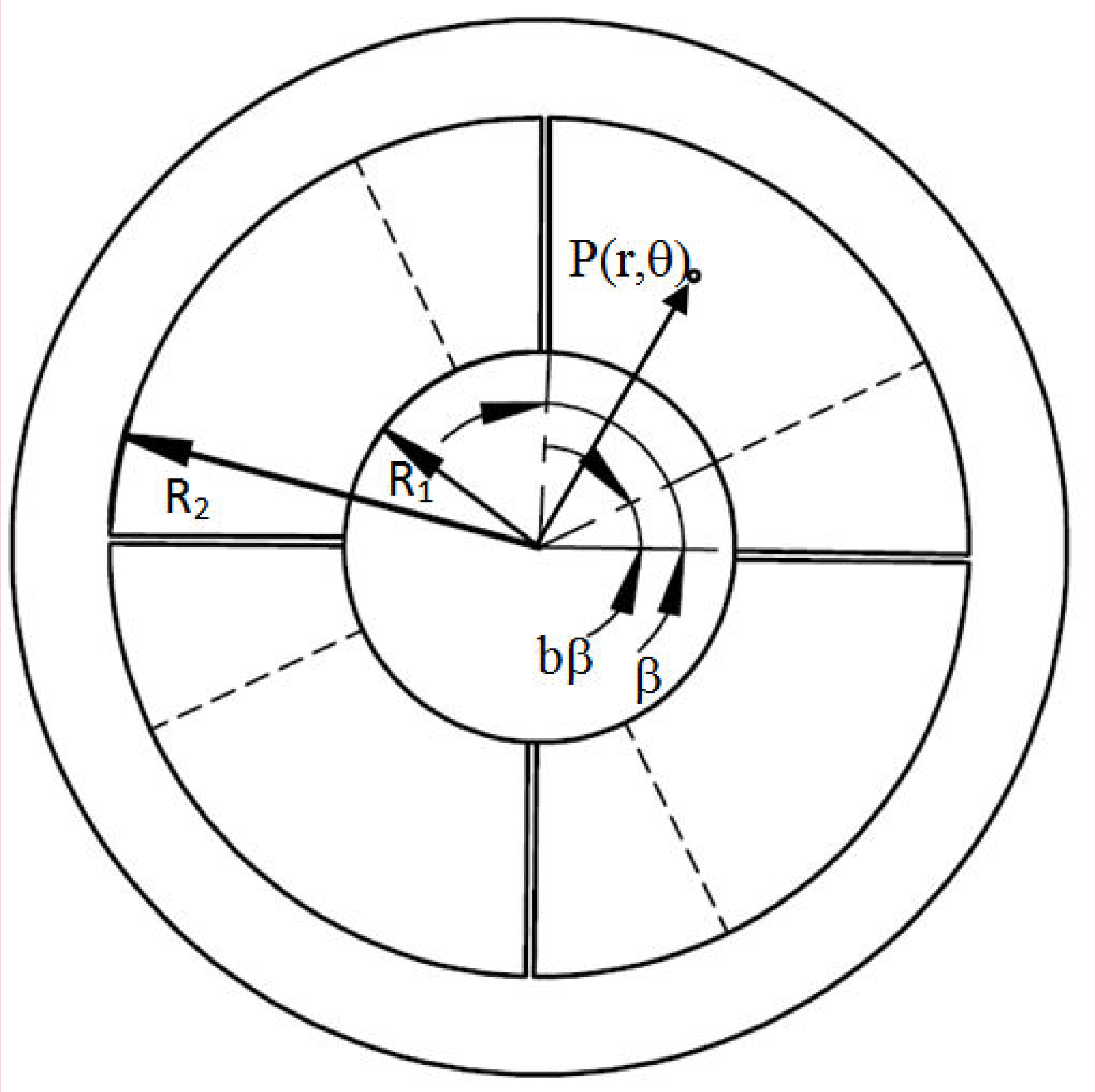
-Soft Failure.





Developed at NASA Glenn Research Center (**2009**)

Configuration of GFTB



R_1 : Inner radius of top/bump foil

R_2 : Outer radius of top/bump foil

β : Angular extent of a bearing pad

b : Extent of the ramp as a function of β

$P_{(r,\theta)}$: Pressure at coordinate (r, θ)

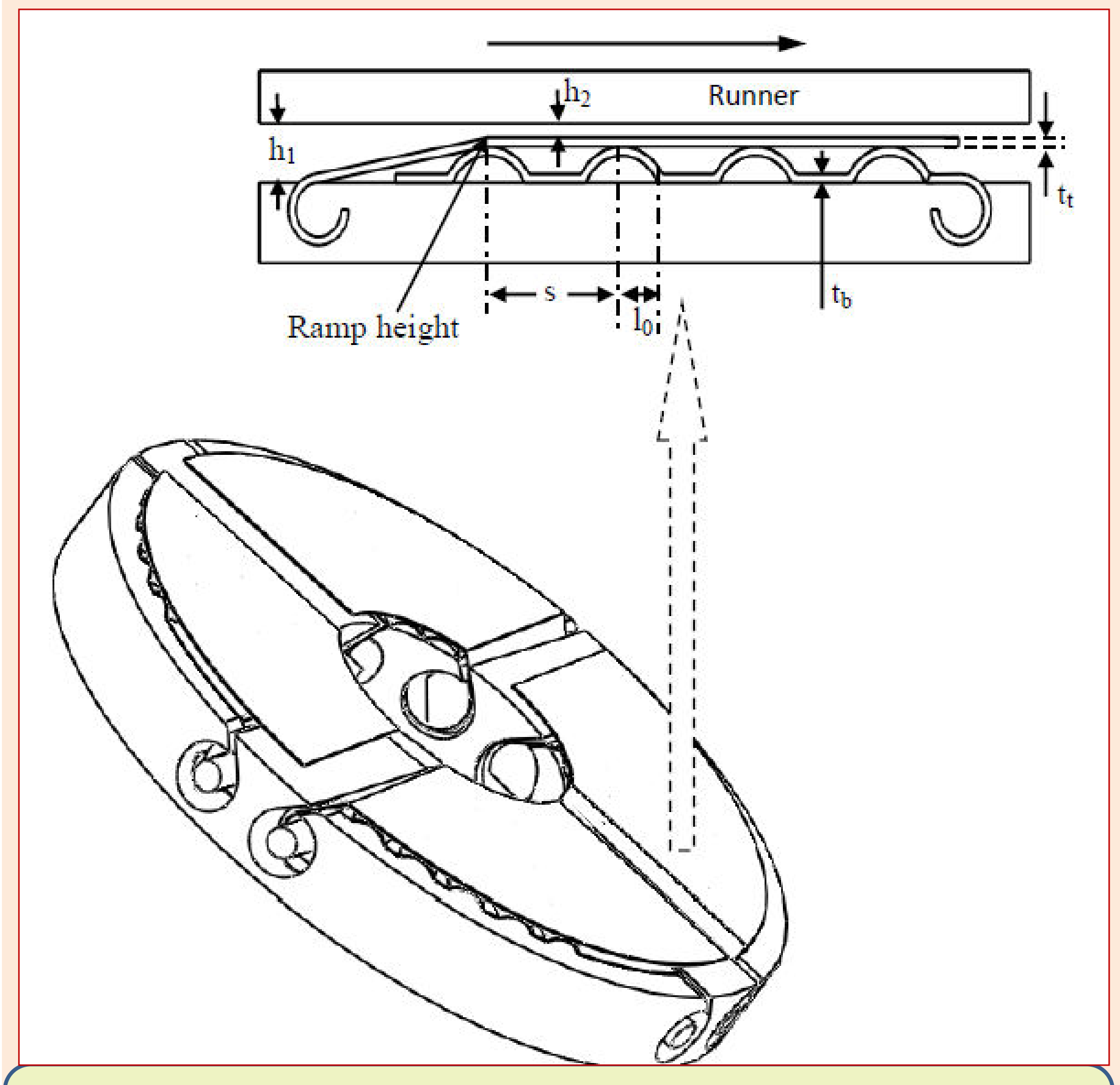
h_1 : Inlet film thickness

h_2 : Minimum film thickness

s : Bump pitch

l_0 : Half bump length

t : Thickness of foil



Aerodynamic Analysis

Assumptions :

-The fluid in the film is isothermal and behaves as a perfect gas.

-The stiffness of the bump foil is taken to be uniformly distributed and constant throughout the bearing surface.

-The top foil is assumed not to deflect about the bumps, but rather to follow the deflection of the bumps themselves.

Governing Equation

$$\frac{1}{\bar{r}}\frac{\partial}{\partial \bar{r}}\left(\bar{r}\bar{h}^3\bar{p}\frac{\partial \bar{p}}{\partial \bar{r}}\right) + \frac{1}{\bar{r}^2}\frac{\partial}{\partial \theta}\left(\bar{h}^3\bar{p}\frac{\partial \bar{p}}{\partial \theta}\right) = \Lambda\frac{\partial(\bar{p}\bar{h})}{\partial \theta}$$

$\bar{r} = \frac{r}{R_2}$, $\bar{p} = \frac{p}{p_a}$, $\bar{h} = \frac{h}{h_2}$ and $\Lambda = \frac{6\omega\mu_0}{p_a}\left(\frac{R_2}{C}\right)^2$

Normalised gas film thickness:

$$\bar{h} = 1 + \bar{g}(\bar{r},\theta) + \alpha(\bar{p} - 1)$$

$$\bar{g} = (\bar{h}_1 - 1)\left(1 - \frac{\theta}{b\beta}\right), 0 \le \theta \le b\beta$$

$$= 0 \qquad , b\beta \le \theta \le \beta$$

where, $\bar{h}_1 = \frac{h_1}{h_2}$, $\bar{g} = \frac{g}{h_2}$ and

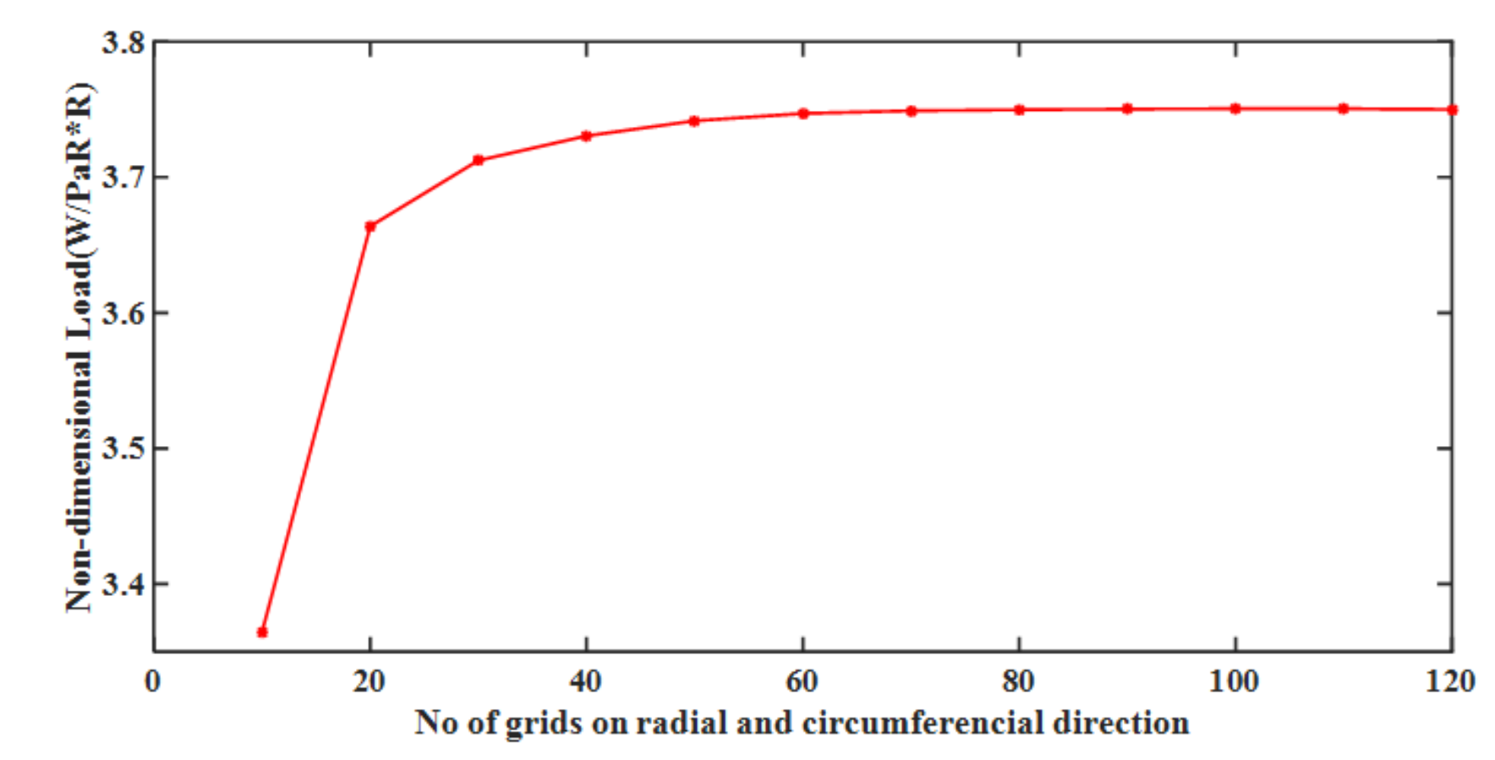
$$\alpha = \frac{2p_a s}{cE}\left(\frac{l}{t_b}\right)^3 (1 - \nu^2)$$

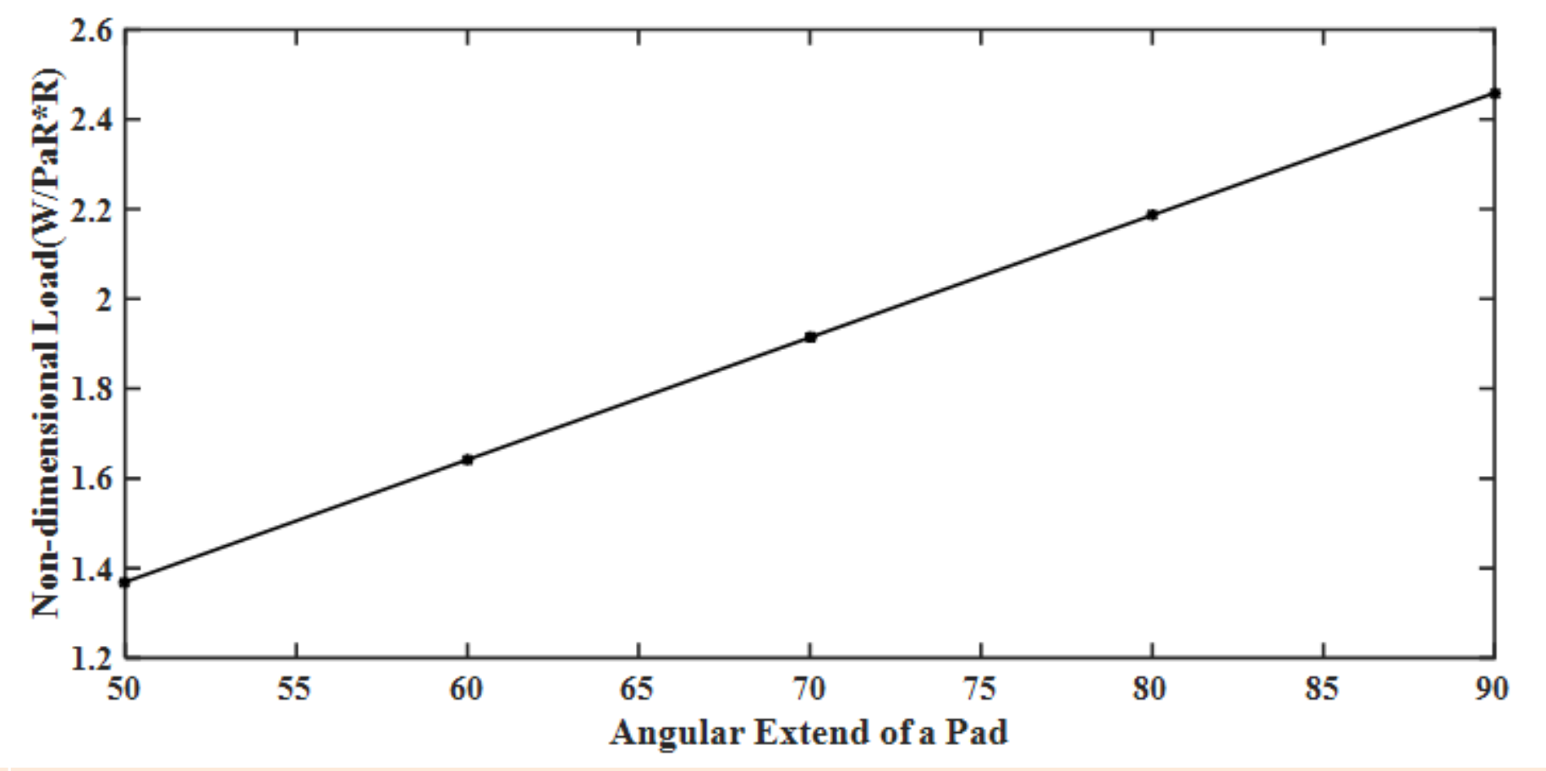
Boundary Conditions:

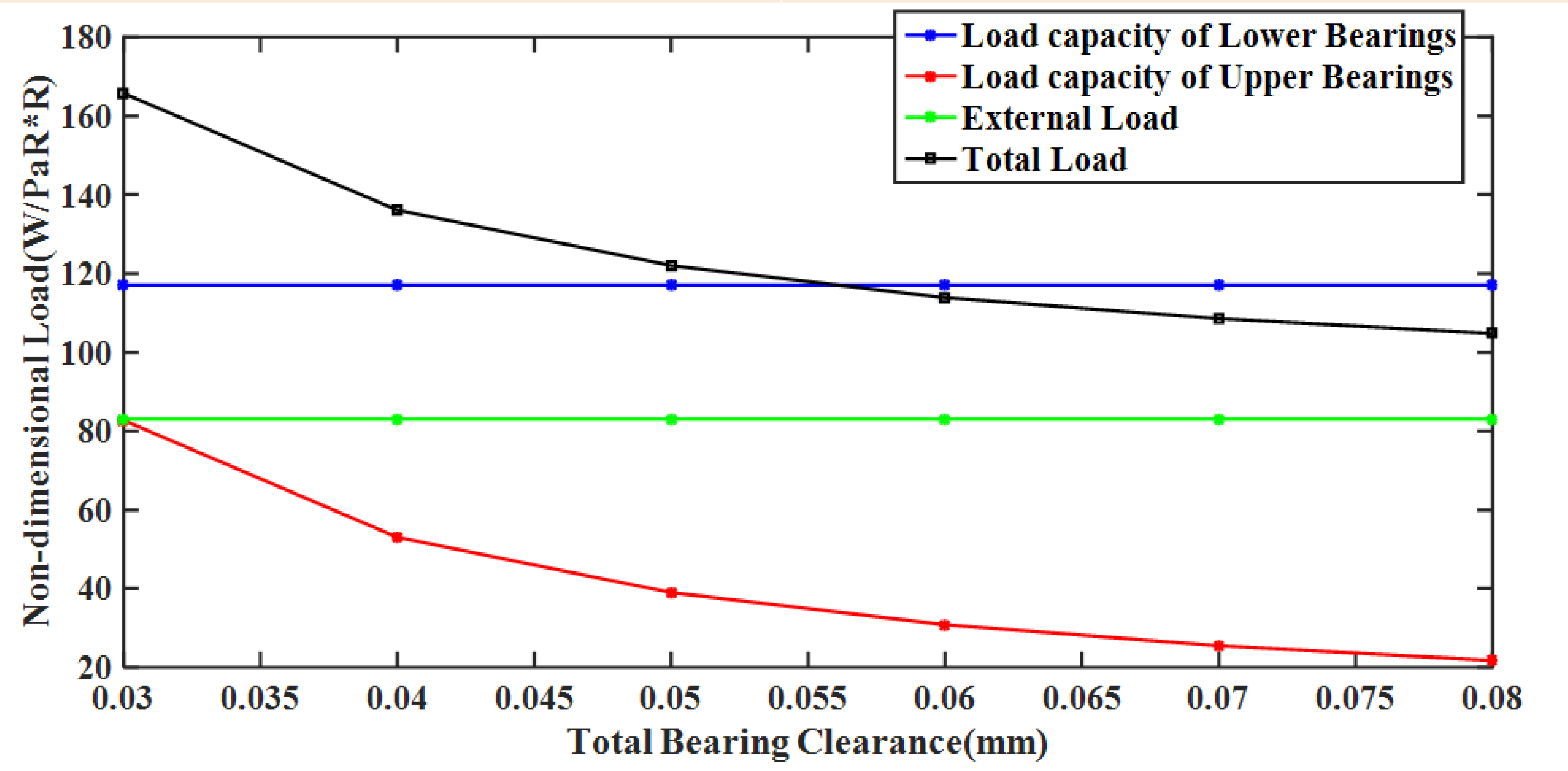
$$\left.\begin{array}{c} \bar{p} = 1 \ at\ \bar{r} = 1 \ and \ \left(1 - \frac{R_1}{R_2}\right) \\ \bar{p} = 1 \ at\ \theta = 0 \ and \ \beta \\ \bar{p} \ge 1 \ over\ bearing\ surface \end{array}\right\}$$

$$\bar{W} = \frac{W}{p_a R_2^{\,2}} = \int_0^{\beta}\int_{\frac{R_1}{R_2}}^{1} \bar{p}\,\bar{r}\,d\bar{r}\,d\theta$$

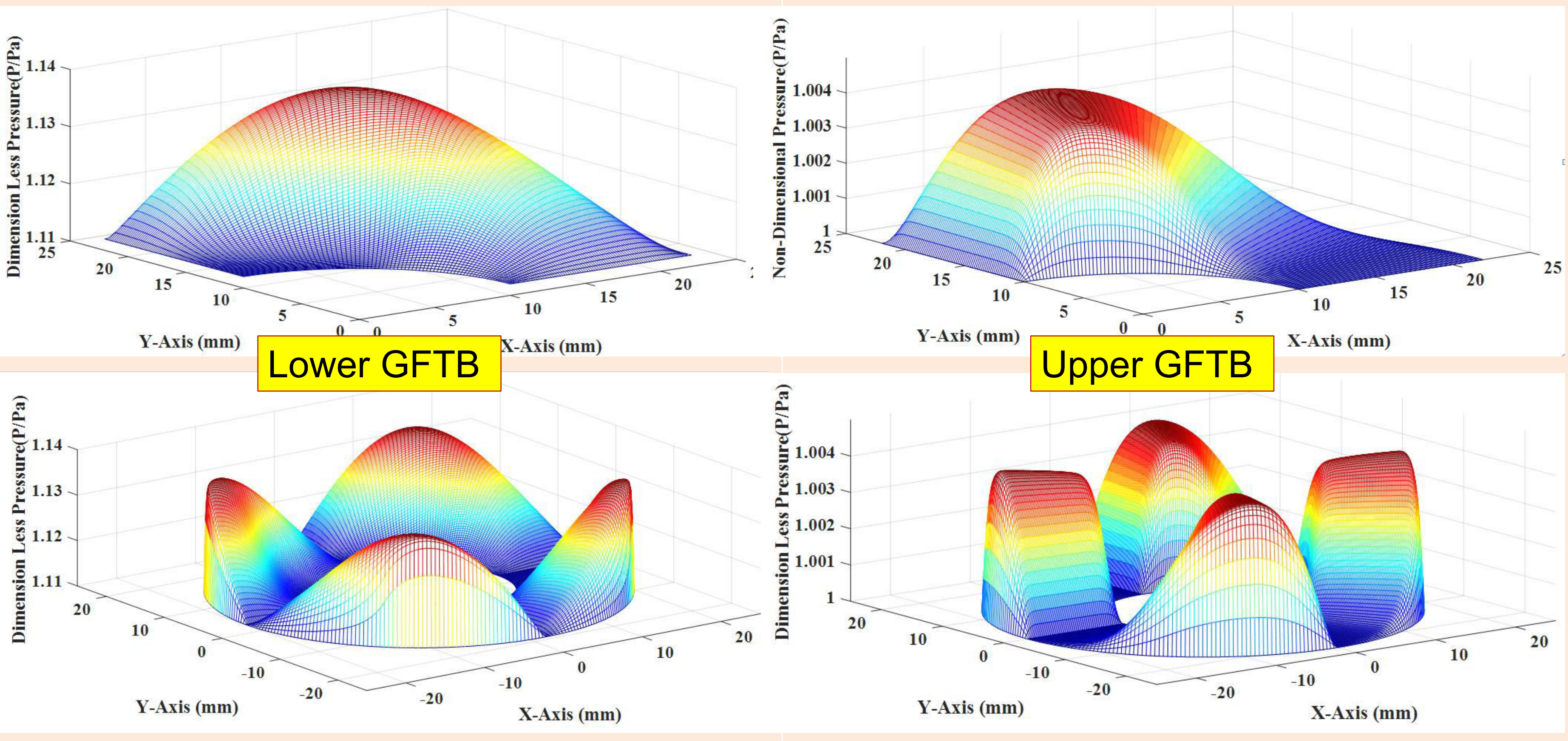
Numerical Results



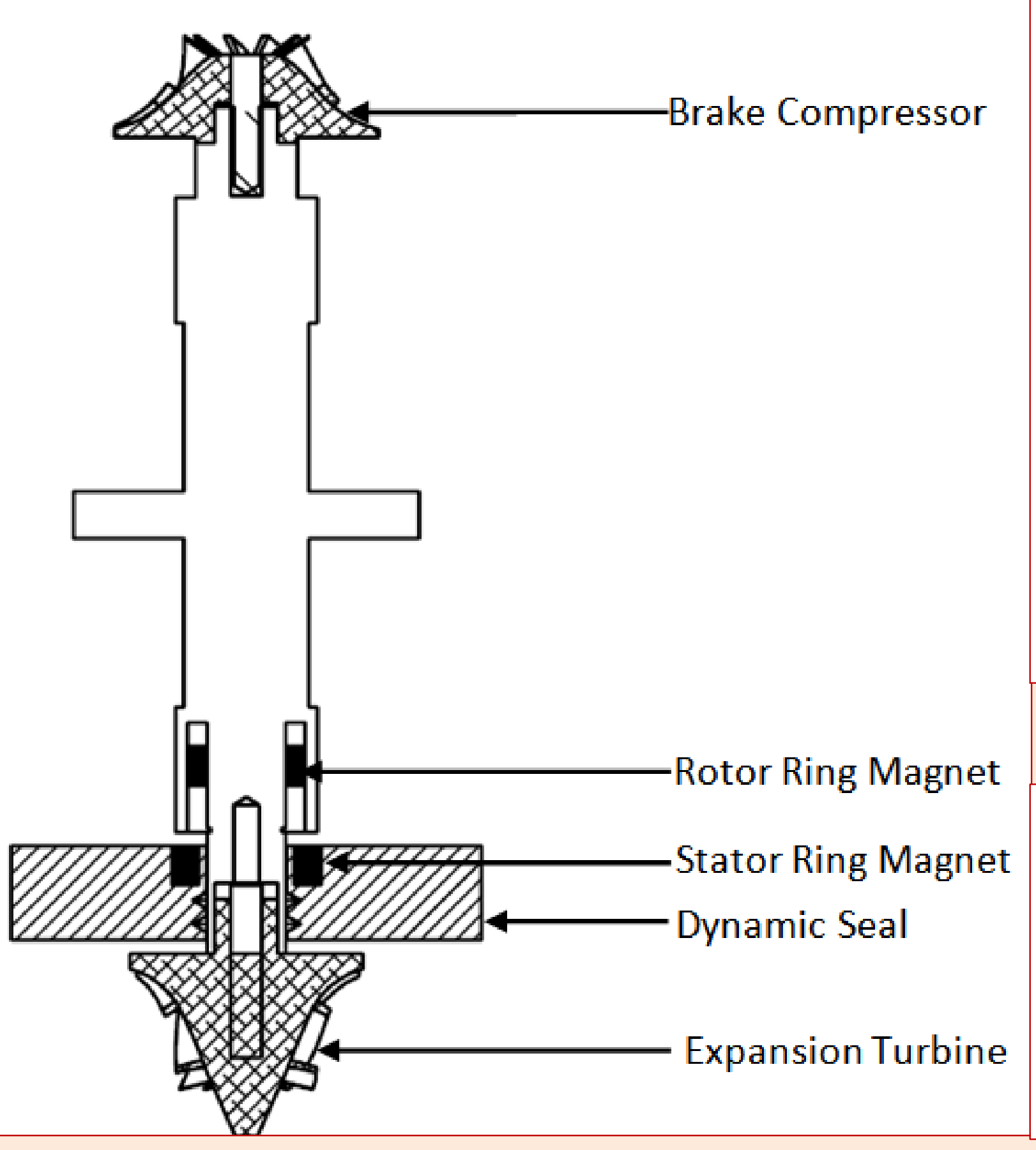


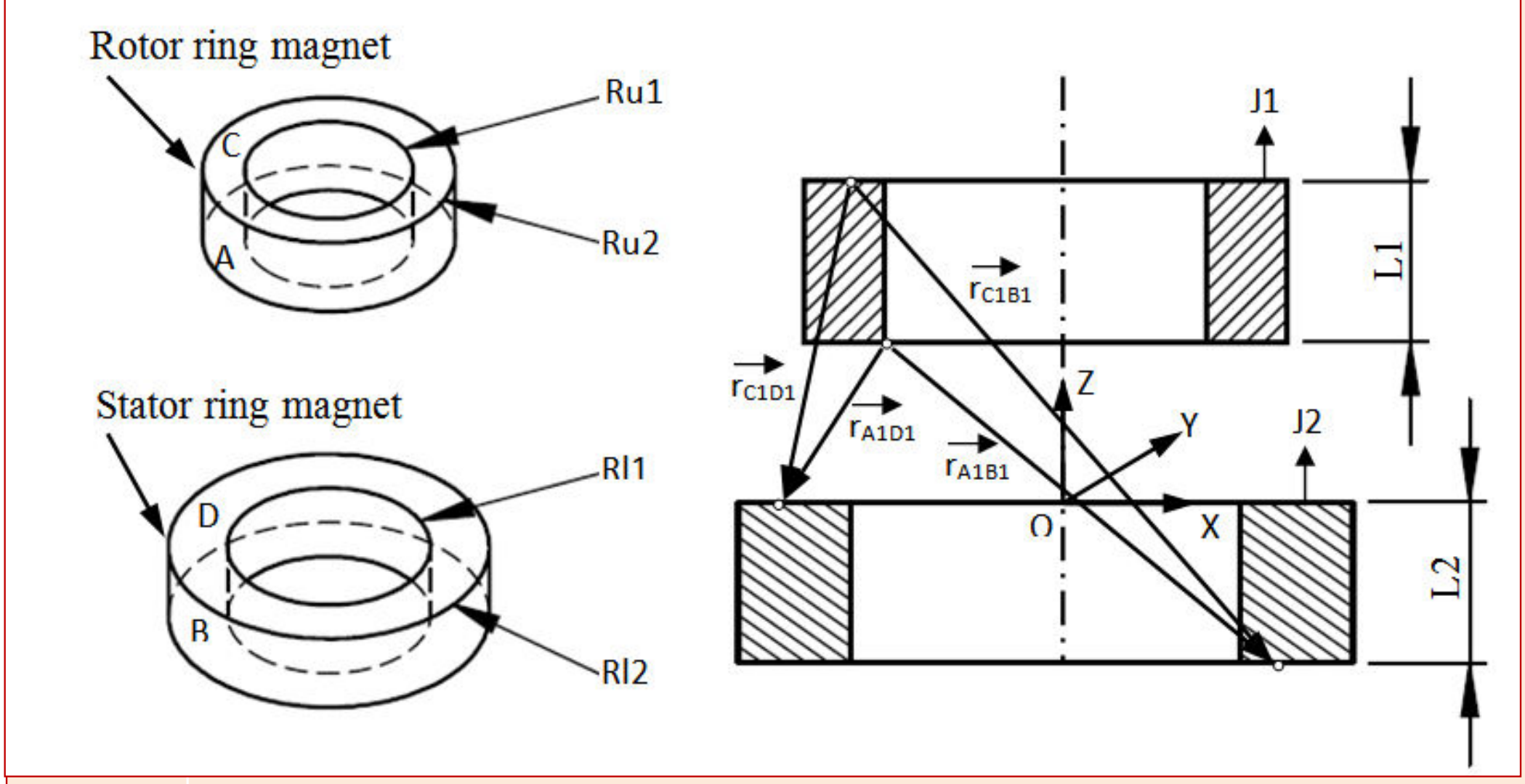


Pressure Profile



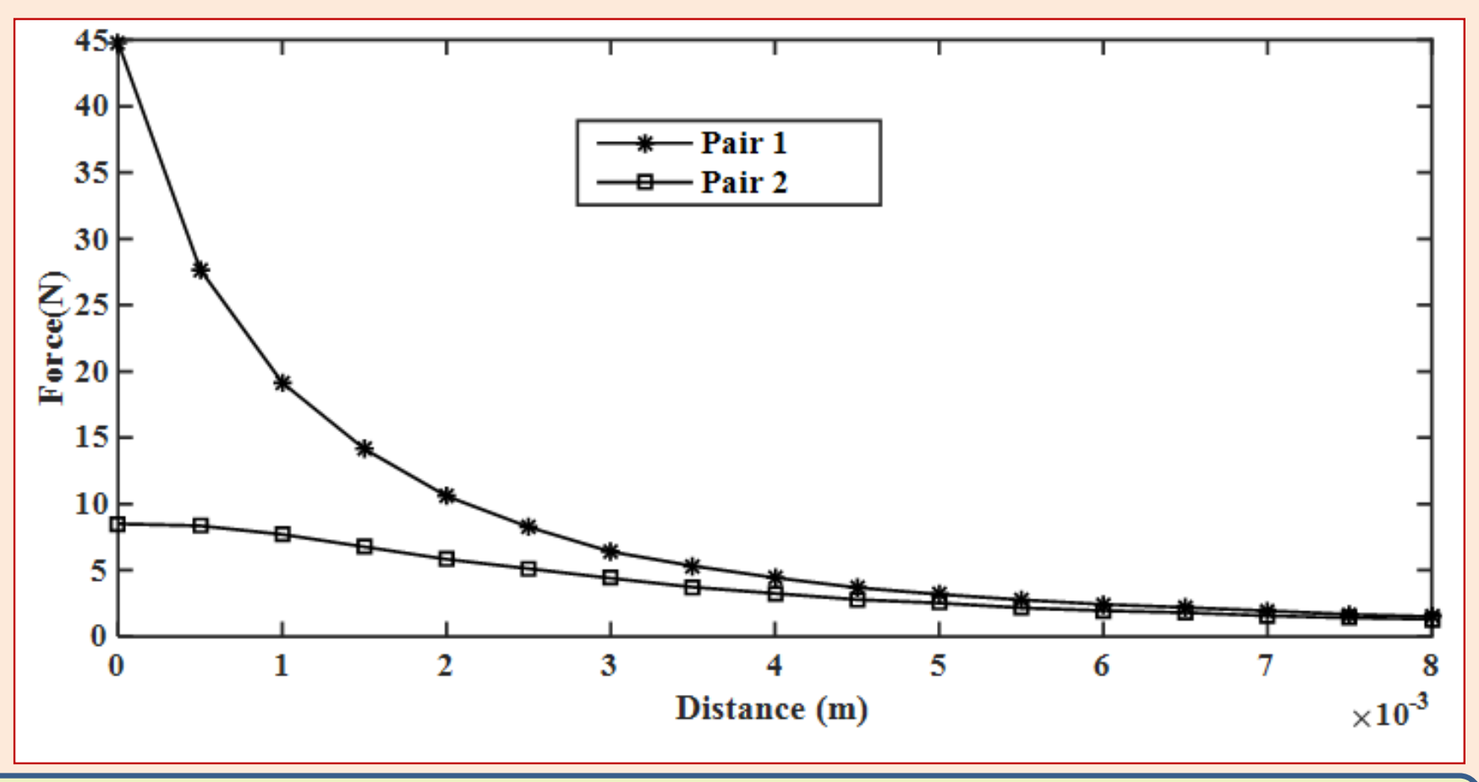
Axial Passive Magnetic Bearings





$$\vec{F}_{A1B1} = \frac{J_1 J_2 S_{A1} S_{B1}}{4\pi\mu_0 r_{A1B1}^{3}}\vec{r}_{A1B1}$$

Magnetic material	NdFeB, Grade N42
J1	1.4 Tesla
J2	1.4 Tesla
μ_0	10^{-6} N/A^2
Pair 1 configuration	Ru1=5 mm, Ru2=7.5 mm , L1=5 mm Rl1=5 mm, Rl2= 7.5 mm , L2=5 mm
Pair 2 configuration	Ru1=5 mm, Ru2=7.5 mm , L1=5 mm Rl1=6.35 mm, Rl2=9.525 mm , L2=3.175 mm



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