EFFECT OF ROTATION ON INTERNAL COOLING OF GAS TURBINE BLADES

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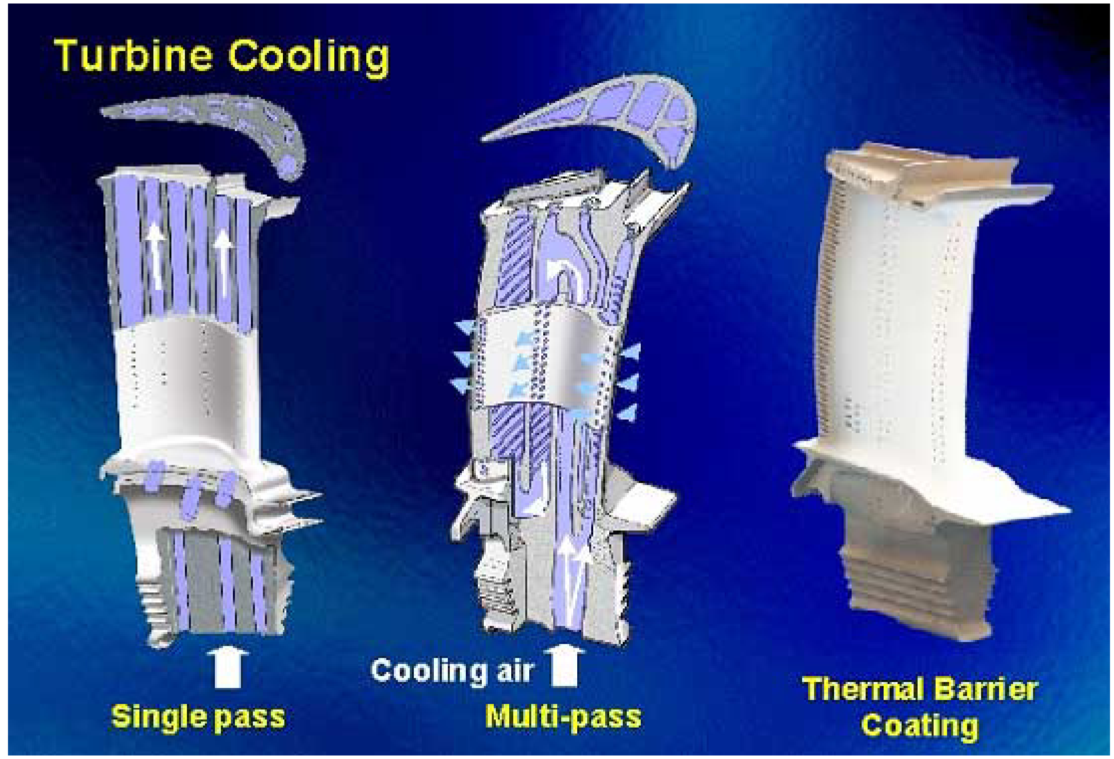
INTRODUCTION

Types of blade cooling techniques :

1. Internal convection cooling
2. Film cooling
3. Transpiration cooling



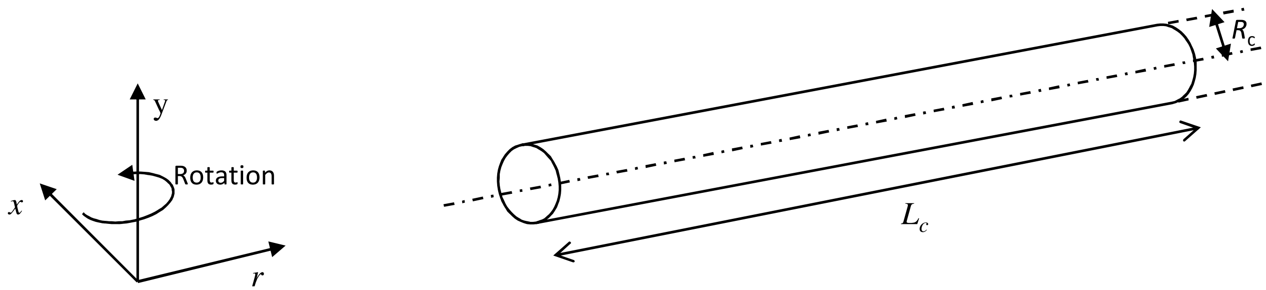
Different types of blade cooling techniques[8]

Conventional analysis :

1. Static blade
2. Thermal conductivity of the blade material is not considered

EFFECT OF ROTATION ON INTERNAL COOLING

- Case of orthogonally rotating channel has been considered
- Case of radially outward flowing coolant and radially inward flowing coolant considered separately



Momentum equation in rotating reference frame,

$$\rho \frac{Dv}{Dt} + 2\underbrace{\rho\left(\omega \times v\right)}_{\text{Coriolis term}} = -\nabla p - \rho\left[\underbrace{\omega \times \left(\omega \times r\right)}_{\text{Centrifugal term}}\right] + \mu \nabla^2 v$$

How does the coolant relative total temperature vary along the length of the channel?

1. Effect of external heat transfer (from hot gas to blade) on coolant temperature with conjugate heat transfer analysis

2. Effect of rotation on coolant relative total temperature for both radially outward flowing coolant and radially inward flowing coolant

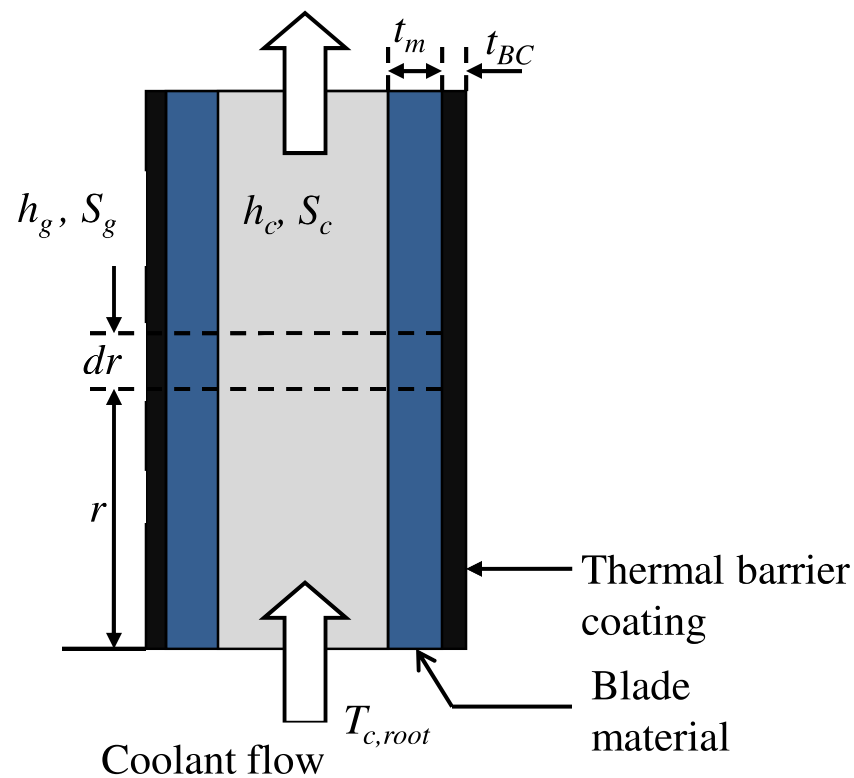
3. Effect of rotation with external heat transfer on coolant temperature for both radially outward flowing coolant and radially inward flowing coolant

EFFECT OF EXTERNAL HEAT TRANSFER ON COOLANT TEMPERATURE

Analytical formulation

$$h_g\, S_g\, (T_g - T_b) = h_c\, S_c\, (T_b - T_c)$$

$$m_c\, c_{pc}\, \frac{dT_c}{dr} = h_c\, S_c\, (T_b - T_c)$$



$$\boxed{T_c = T_g - (T_g - T_{c,root})\, e^{-\beta r}}$$

$$\text{where, } \beta = \frac{h_g\, S_g\, h_c\, S_C}{m_c\, c_{pc}\, (h_g\, S_g + h_c\, S_c)}$$

CONSIDERATION OF FINITE METAL THERMAL CONDUCTIVITY

$$U_g = \frac{1}{\left(\frac{1}{h_g} + \frac{t_{BC}}{k_{BC}} + \frac{t_m}{k_m}\right)}$$

$$\boxed{T_c = T_g - (T_g - T_{c,root})e^{-\beta r}}$$

$$\text{where } \beta = \frac{U_g S_g h_c S_C}{m_c c_{pc}(U_g S_g + h_c S_c)}$$

1. Gas side heat transfer coefficient, $h_g = 7000\,\mathrm{W/m^2K}$
2. Coolant side heat transfer coefficient calculated using Dittus-Boelter correlation,

$$Nu_{nr} = 0.023\mathrm{Re}_{Dh}^{0.8}\mathrm{Pr}^n$$

n=0.4 for heating of the fluid

n=0.3 for cooling of the fluid

$$Nu_{nr} = 0.023\left[1 + \frac{C}{(x/D_h)^m}\right]\mathrm{Re}_{Dh}^{0.8}\,\mathrm{Pr}^n$$

Geometric details

Channel cross-section	**Circular**
Surface condition	**Smooth**
Reynolds number, Re	**25,000**
Rotation number, Ro	**0.25**
Hydraulic diameter, D_h	**6.3 mm**
L/D_h	**12**
Blade root radial distance from engine centerline	**300 mm**
Non dimensional mid-span radius, e/D_h	**59.62**

Reynolds number,

$$\mathrm{Re}_{Dh} = \frac{\rho v D_h}{\mu}$$

Rotation number,

$$\mathrm{Ro} = \frac{\omega D_h}{v}$$

Blade material properties

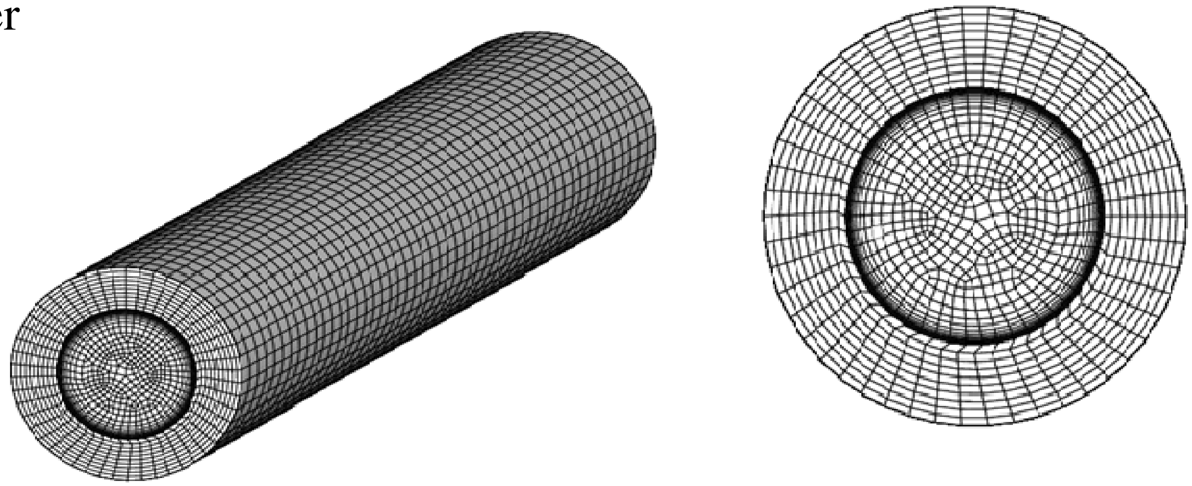
Material	Thickness(mm)	Thermal conductivity (W/m-K)
Blade material	**2.0**	**26**
Thermal barrier coating	**0.15**	**1.8**
Geometric model	**2.0**	**12.5**

COMPUTATIONAL FLUID DYNAMICS (CFD) SIMULATION SETUP

•Commercial software package- ANSYS FLUENT, GAMBIT

• **Boundary conditions:**

1. MASS FLOW INLET at inlet
2. PRESSURE OUTLET at the outlet
3. Convective boundary condition at the outside wall
4. Conjugate heat transfer

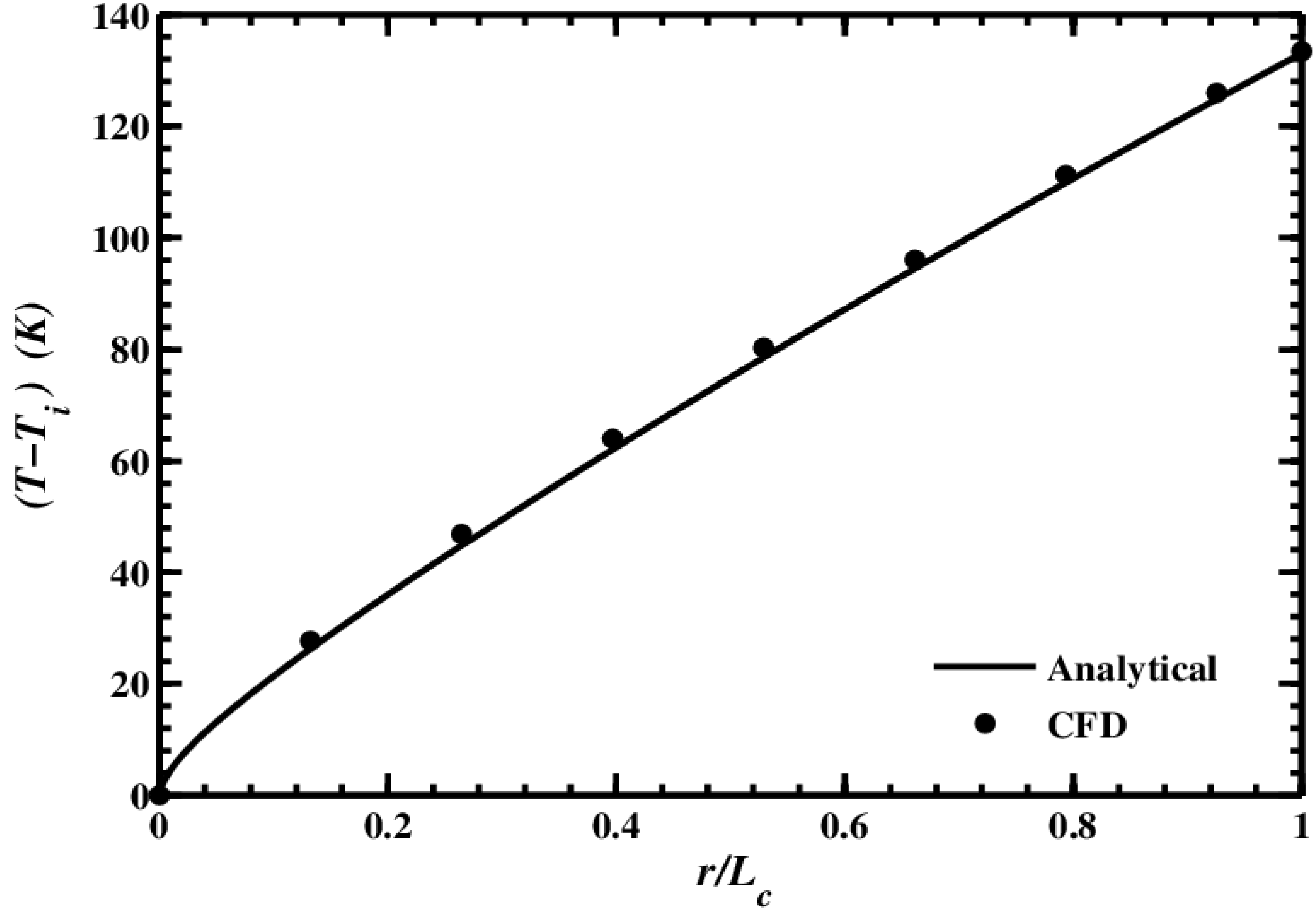


- Air with constant density
- Standard k-ε model turbulence model with enhanced wall treatment(with wall $y^+ < 1$)
- Convergence criteria: 1e-6

GRID INDEPENDENCE STUDY

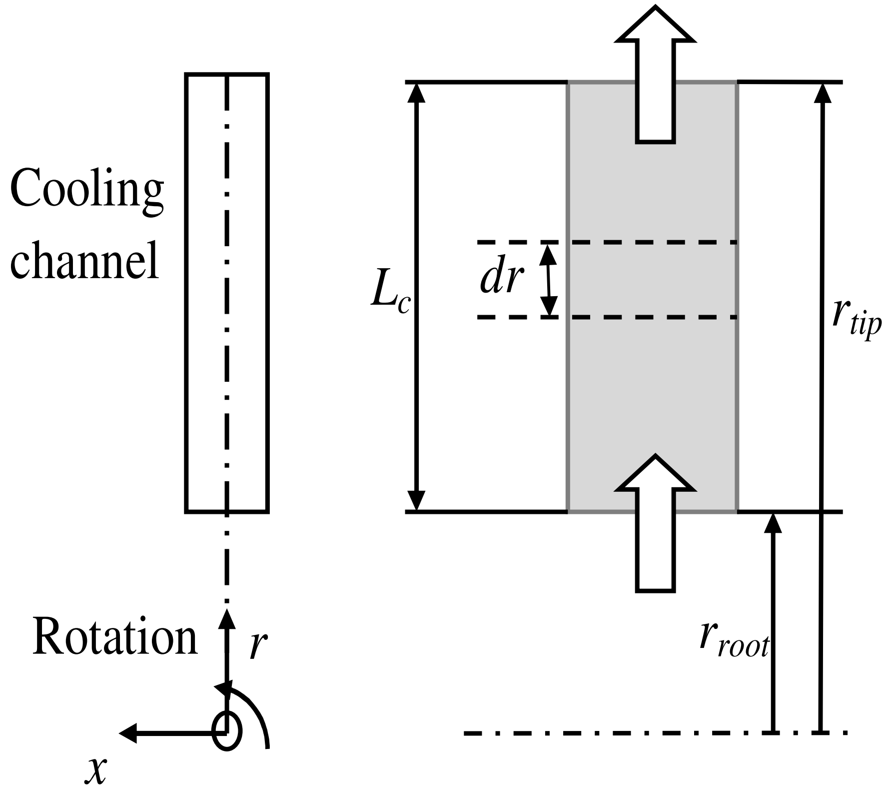
Grid independence test 1: Conventional internal cooling analysis with finite thermal conductivity of blade material without the effect of rotation (conjugate heat transfer case)

Grid refinement	Coarse	Standard	Fine
Number of grid points in θ-direction	34	50	76
Number of grid points in r-direction	33	50	75
Distance of the first grid point from the wall (fluid region)	2e-5 m	1e-6 m	0.75e-6 m
Boundary layer mesh successive ratio (fluid region)	1.4	1.35	1.3
Number of rows in boundary layer mesh (fluid region)	10	20	20
Total number of grids	25938	86400	209400
Pressure drop (Pa)	582.931	576.807	576.49



Comparison of increase in temperature obtained from analytical formulations and CFD for the non-rotating case (Re=25,000; Ro=0)

EFFECT OF ROTATION ON COOLANT RELATIVE TOTAL TEMPERATURE



$$\frac{dq}{dr}=\frac{d\hbar}{dr}+\frac{d(KE)}{dr}+\frac{d(PE)}{dr}+\frac{dW}{dr}$$

$$\frac{dq}{dr}=0 \qquad \frac{dW}{dr}=0$$

$$\frac{d(KE)}{dr}=v_{rel}\,\frac{dv_{rel}}{dr}$$

$$\frac{d(PE)}{dr}=g-\omega^2 r$$

$$\frac{d\hbar_{0c,rel}}{dr}-\omega^2\,r=0$$

$T_{0c,rel}$ =relative total temperature

$$T_{0c,rel}=\frac{\omega^2 r^2}{2c_p}+C$$

For radially outward flowing coolant,

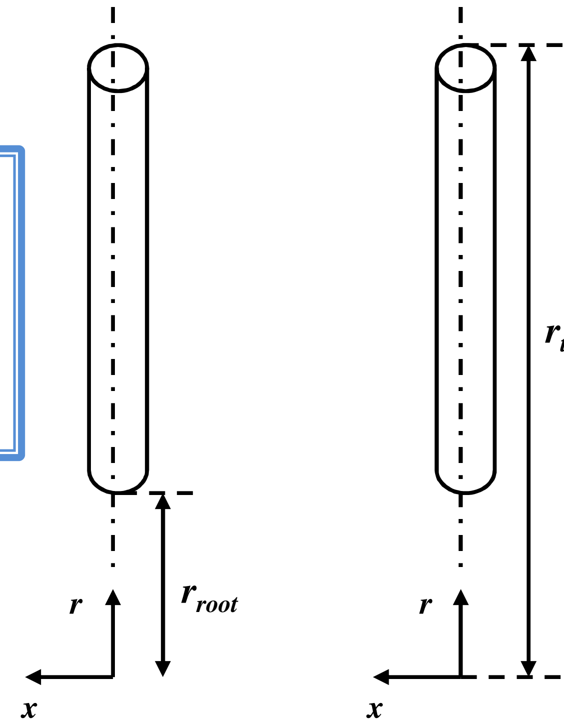
at $r = r_{root}, T_{0c,rel} = T_{0c,root}$

$$\boxed{T_{0c,rel} = T_{0c,root} - \frac{\omega^2 \, {r_{root}}^2}{2c_p}\left[1 - \left(\frac{r}{r_{root}}\right)^2\right]}$$

For radially inward flowing coolant,

at $r = r_{tip}, T_{0c,rel} = T_{0c,tip}$

$$\boxed{T_{0c,rel} = T_{0c,tip} - \frac{\omega^2 \, {r_{tip}}^2}{2c_p}\left[1 - \left(\frac{r}{r_{tip}}\right)^2\right]}$$



Radially outward flowing case

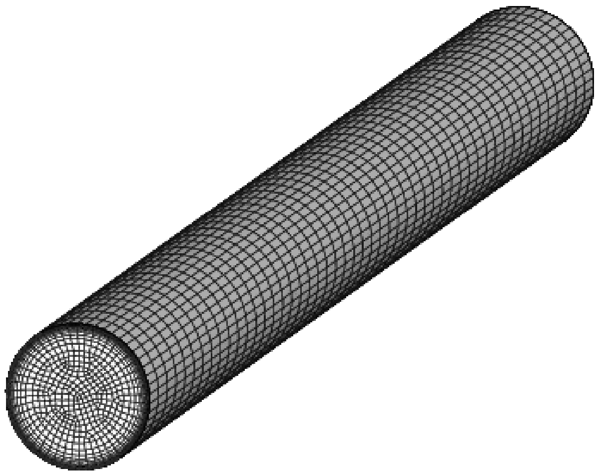
Radially inward flowing case

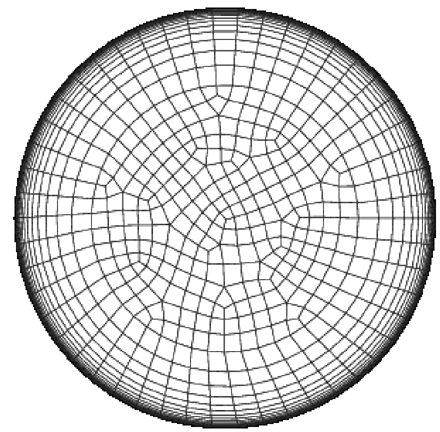
COMPUTATIONAL FLUID DYNAMICS (CFD) SIMULATION SETUP

•Commercial software package- ANSYS FLUENT, GAMBIT

• **Boundary conditions:**

1. MASS FLOW INLET at inlet
2. PRESSURE OUTLET at the outlet
3. Adiabatic wall





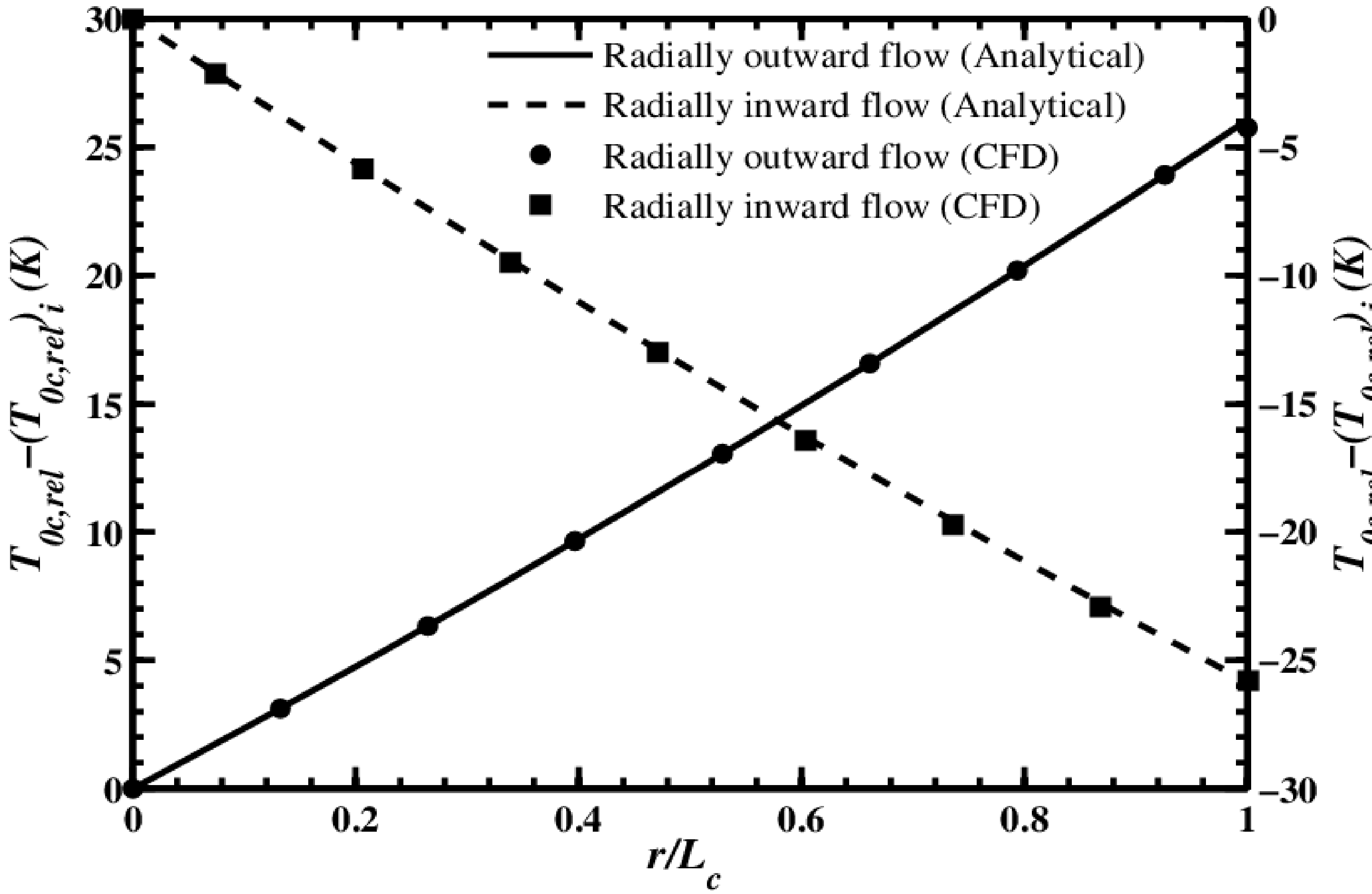
- Air as ideal gas
- Standard k-ε model turbulence model with enhanced wall treatment
- Rotating reference frame specification to include the effects of rotation
- Convergence criteria: 1e-6

GRID INDEPENDENCE STUDY

Grid independence test 2: Effect of rotation without external heat transfer

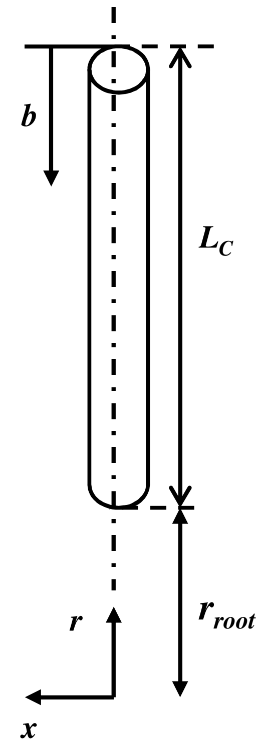
Grid refinement	Coarse	Standard	Fine
Number of grid points in -θ direction	34	50	76
Number of grid points in r -direction	33	50	75
Distance of the first grid point from the wall	2e-5 m	1e-6 m	0.75e-6 m
Boundary layer mesh successive ratio	1.4	1.35	1.3
Number of rows in boundary layer mesh	10	20	20
Total number of grids	14751	62150	155925
Pressure drop (Pa)	155199	155166	155161

COMPARISON OF ANALYTICAL AND CFD SOLUTIONS



Comparison of change in relative total temperature obtained from analytical formulation and CFD simulation for radially outward flowing case and radially inward flowing case (Re=25,000; Ro=0.25)

IMPROVED ANALYSIS WITH EFFECT OF ROTATION AND EXTERNAL HEAT TRANSFER



$$\frac{dq}{dr} = \frac{d\hbar_{0c,rel}}{dr} - \omega^2 r$$

$$\frac{dq}{dr} = \frac{U_g S_g h_c S_C (T_g - T_{0c,rel})}{m_c (U_g S_g + h_c S_c)}$$

For radially outward flowing coolant,

$$\frac{dT_{0c,rel}}{dr} - \frac{\omega^2 r}{c_{pc}} - \frac{U_g S_g h_c S_C (T_g - T_{0c,rel})}{m_c (U_g S_g + h_c S_c) c_{pc}} = 0$$

For radially inward flowing coolant,

$$\frac{dT_{0c,rel}}{db} + \frac{\omega^2 [r_{root} + (L - b)]}{c_{pc}} - \frac{U_g S_g h_c S_C (T_g - T_{0c,rel})}{m_c (U_g S_g + h_c S_c) c_{pc}} = 0$$

For radially outward flowing coolant,

$$\text{at } r = r_{root}, T_{0c,rel} = T_{0c,root}$$

$$\boxed{T_{0c,rel}(r) = T_g + G(r) + [T_{0c,root} - T_g - G(r_{root})]e^{-\beta(r - r_{root})}}$$

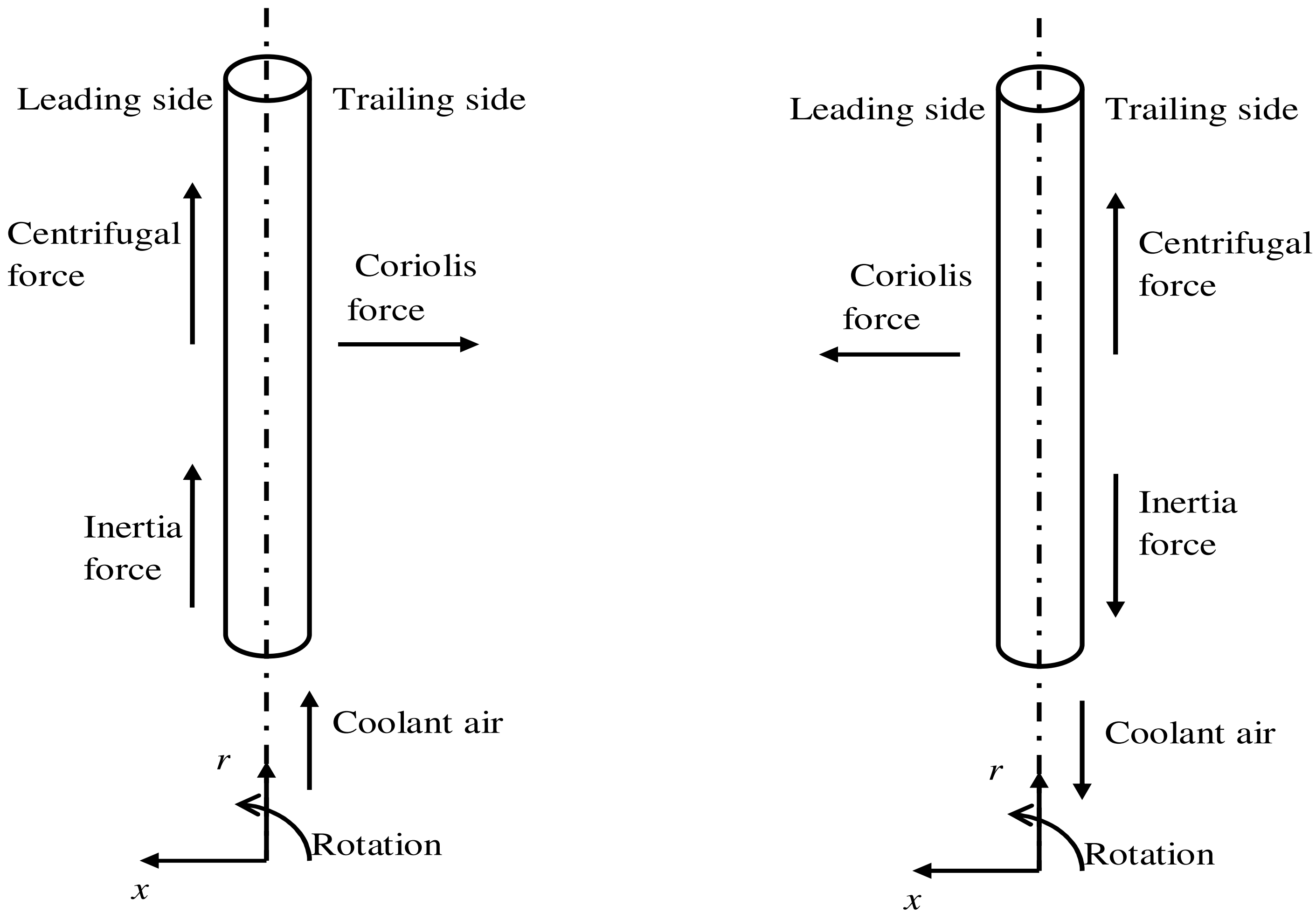
$$\text{where, } \beta = \frac{U_g S_g h_c S_C}{m_c c_{pc}(U_g S_g + h_c S_c)}, G(r) = \frac{\omega^2}{c_{pc}}\left[\frac{r}{\beta} - \frac{1}{\beta^2}\right]$$

For radially inward flowing coolant,

$$\text{at } b = 0, T_{0c,rel} = T_{0c,tip}$$

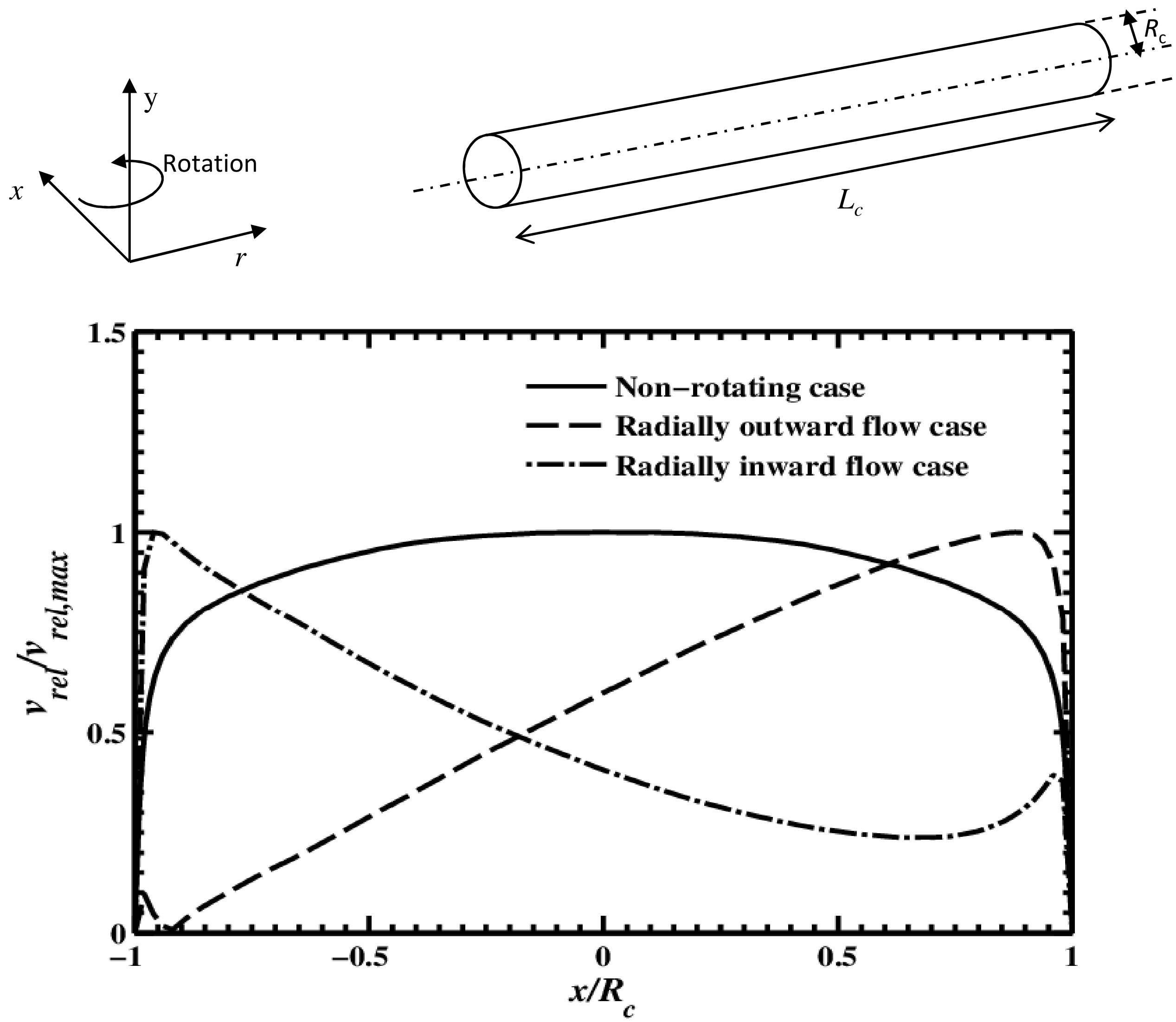
$$\boxed{T_{0c,rel}(b) = T_g - G(b) + [T_{0c,tip} - T_g + G(0)]e^{-\beta b}}$$

$$\text{where, } G(b) = \frac{\omega^2}{c_{pc}}\left[\frac{\{r_{root} + (L - b)\}}{\beta} - \frac{1}{\beta^2}\right]$$



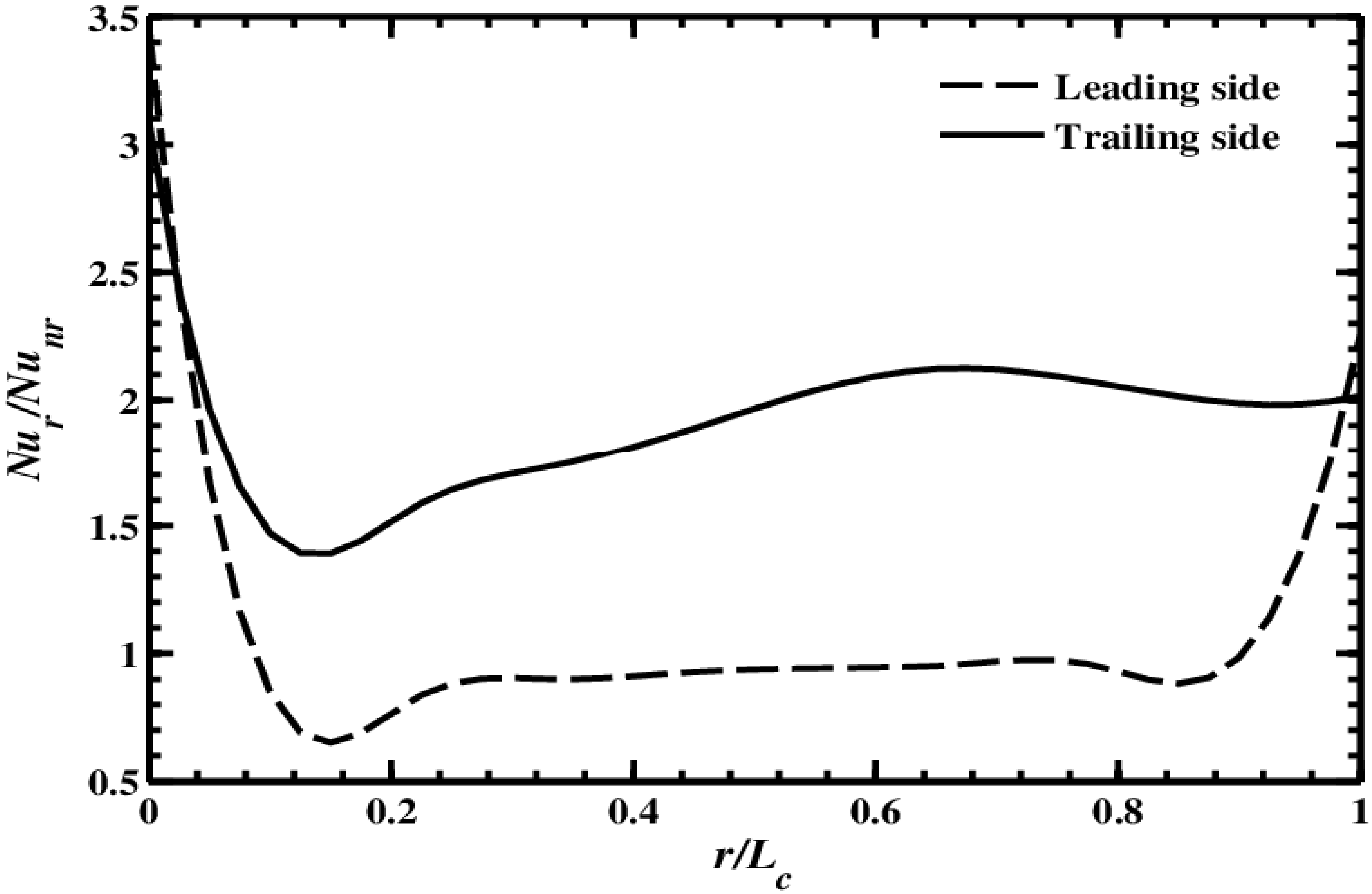
Radially outward flowing coolant **Radially inward flowing coolant**

Schematic of rotating coolant channel configuration and direction of different forces

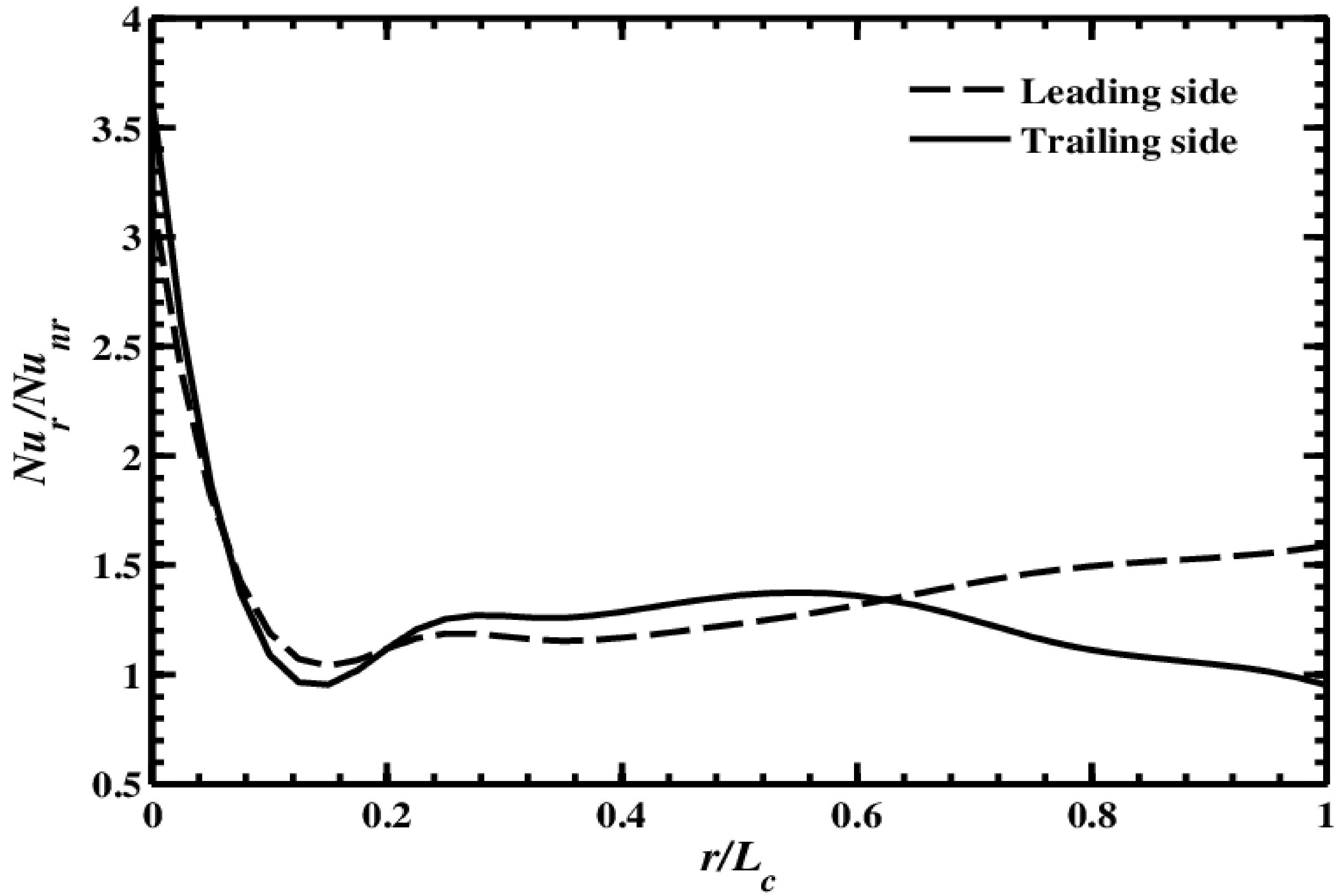


Comparison of relative velocity profile for non-rotating case, rotating radially outward flow and radially inward flow: prediction of present CFD computation

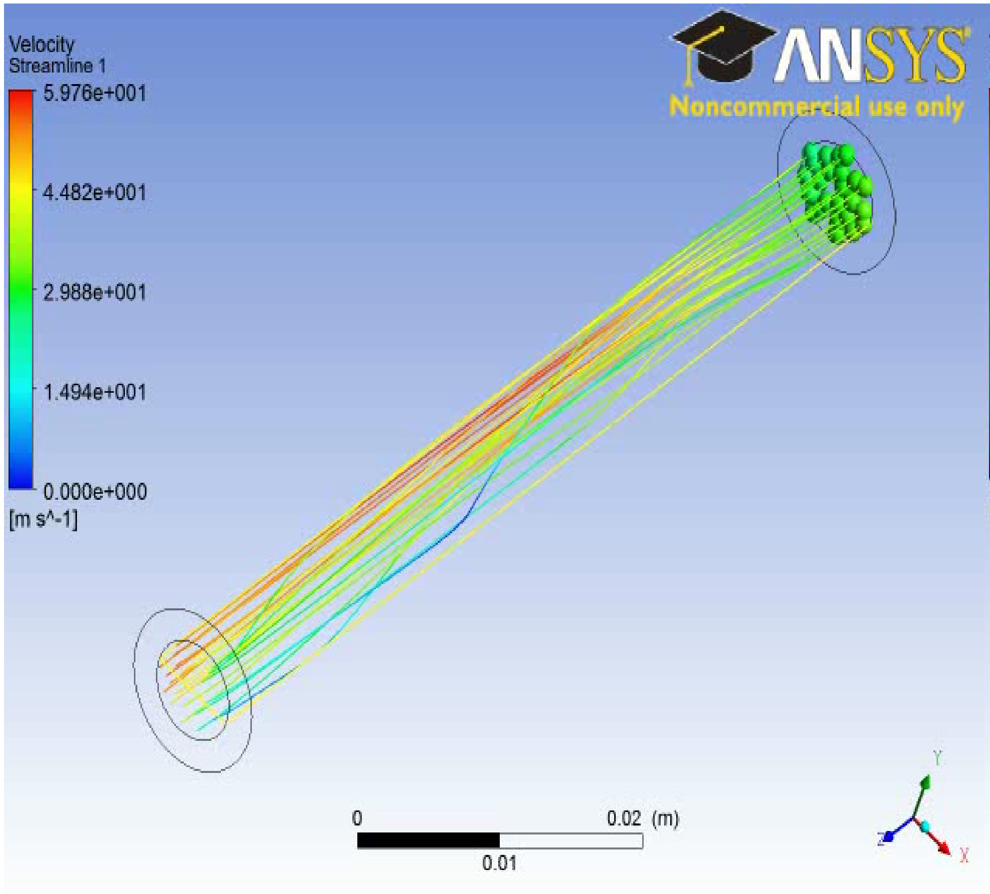
EFFECT OF ROTATION ON HEAT TRANSFER



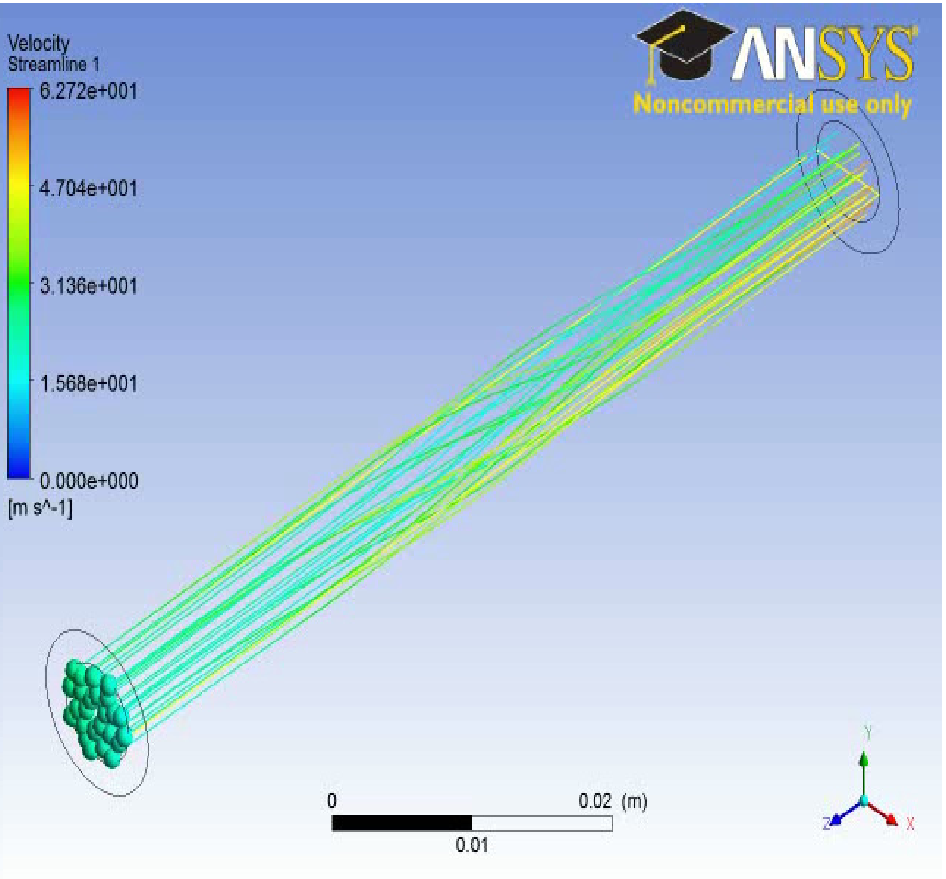
Nusselt number ratio at the leading and trailing side for radially outward flow: prediction of present CFD computation (Re=25,000; Ro=0.25)



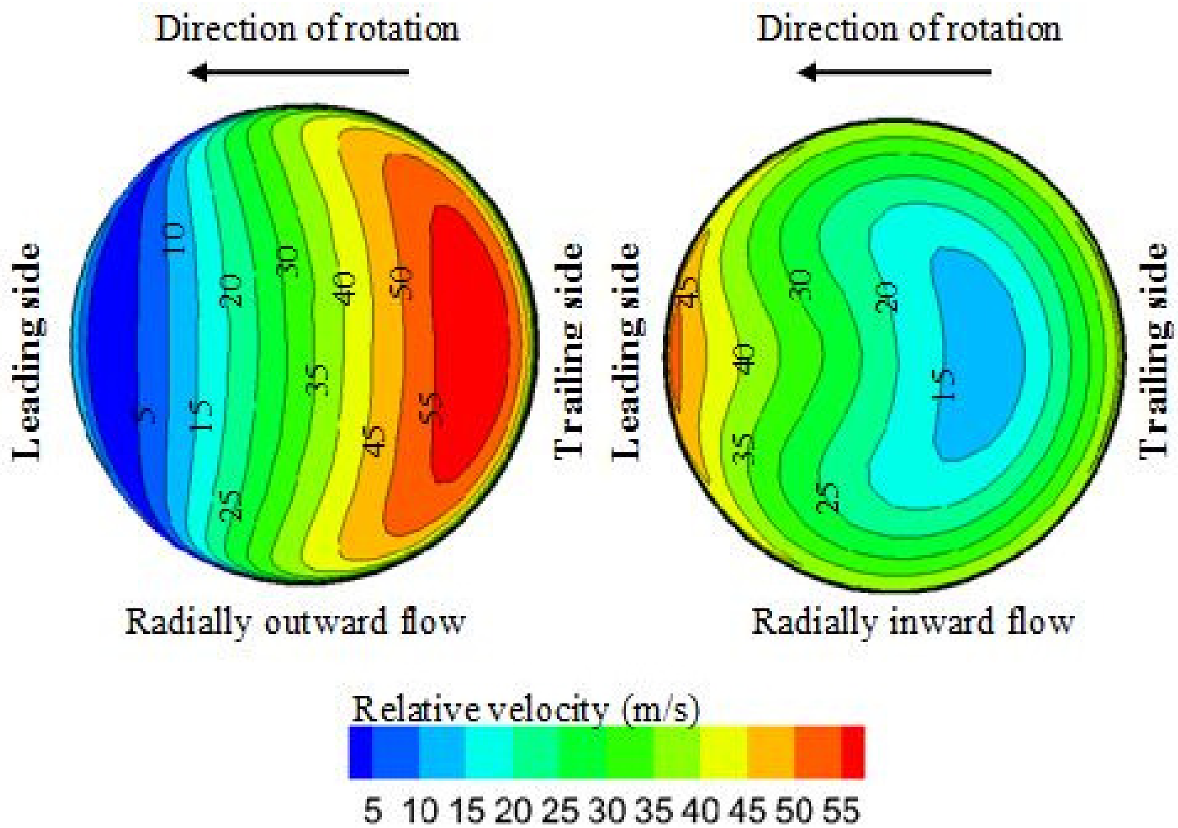
Nusselt number ratio at the leading side and trailing side for radially inward flow: prediction of present CFD computation (Re=25,000; Ro=0.25)



Radially outward flow case

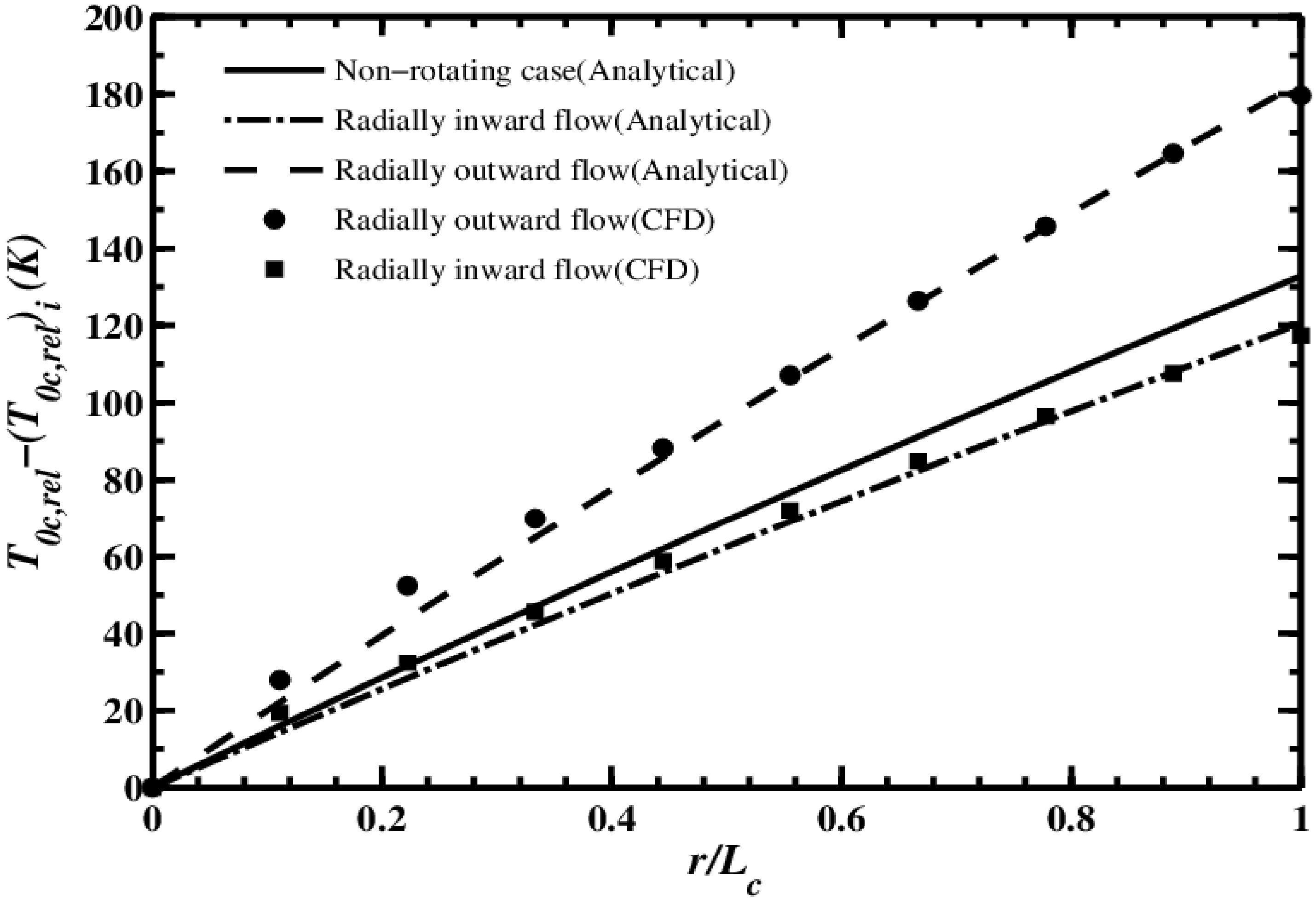


Radially inward flow case



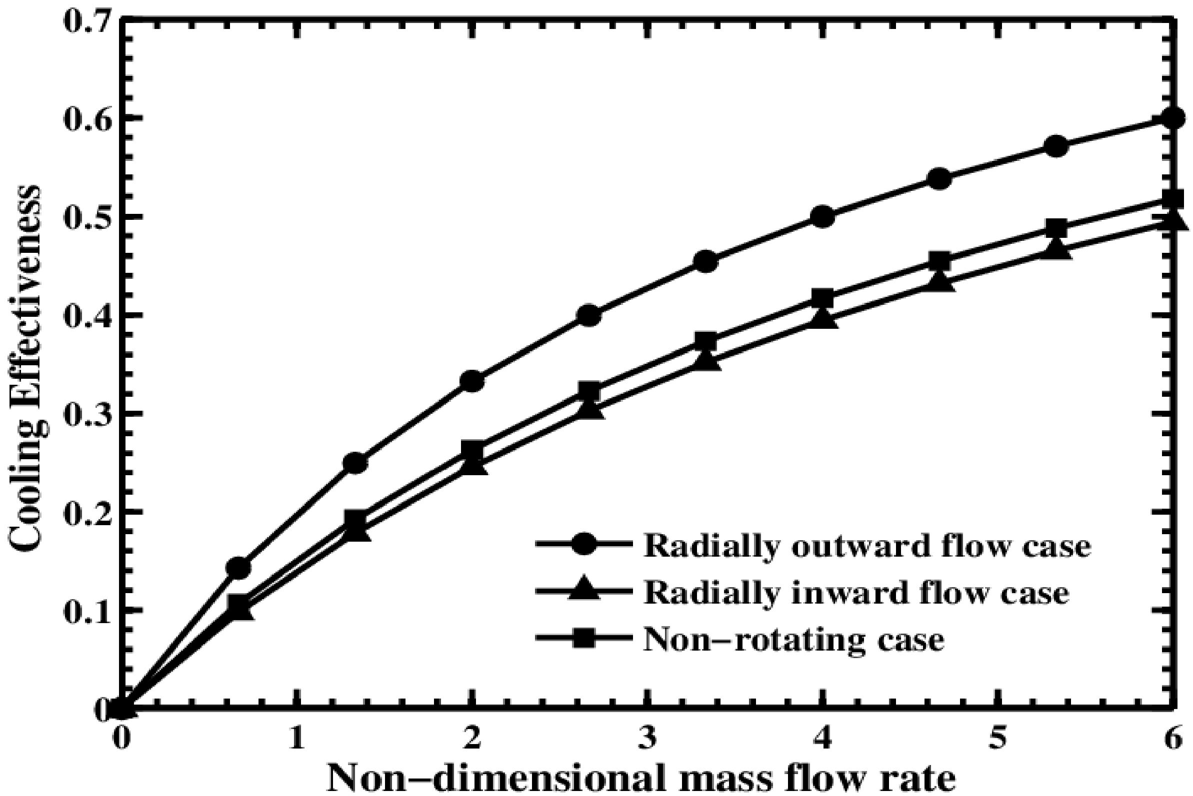
Mid-span relative velocity magnitude contour for radially outward and radially inward flow case respectively: prediction of present CFD computation (Re=25,000; Ro=0.25)

COMPARISON OF ANALYTICAL AND CFD SOLUTIONS



Comparison of change in relative total temperature obtained from analytical formulations and CFD simulation for non-rotating case, radially outward flow and radially inward flow case: prediction of present CFD computation

EFFECT OF COOLANT MASS FLOW RATE



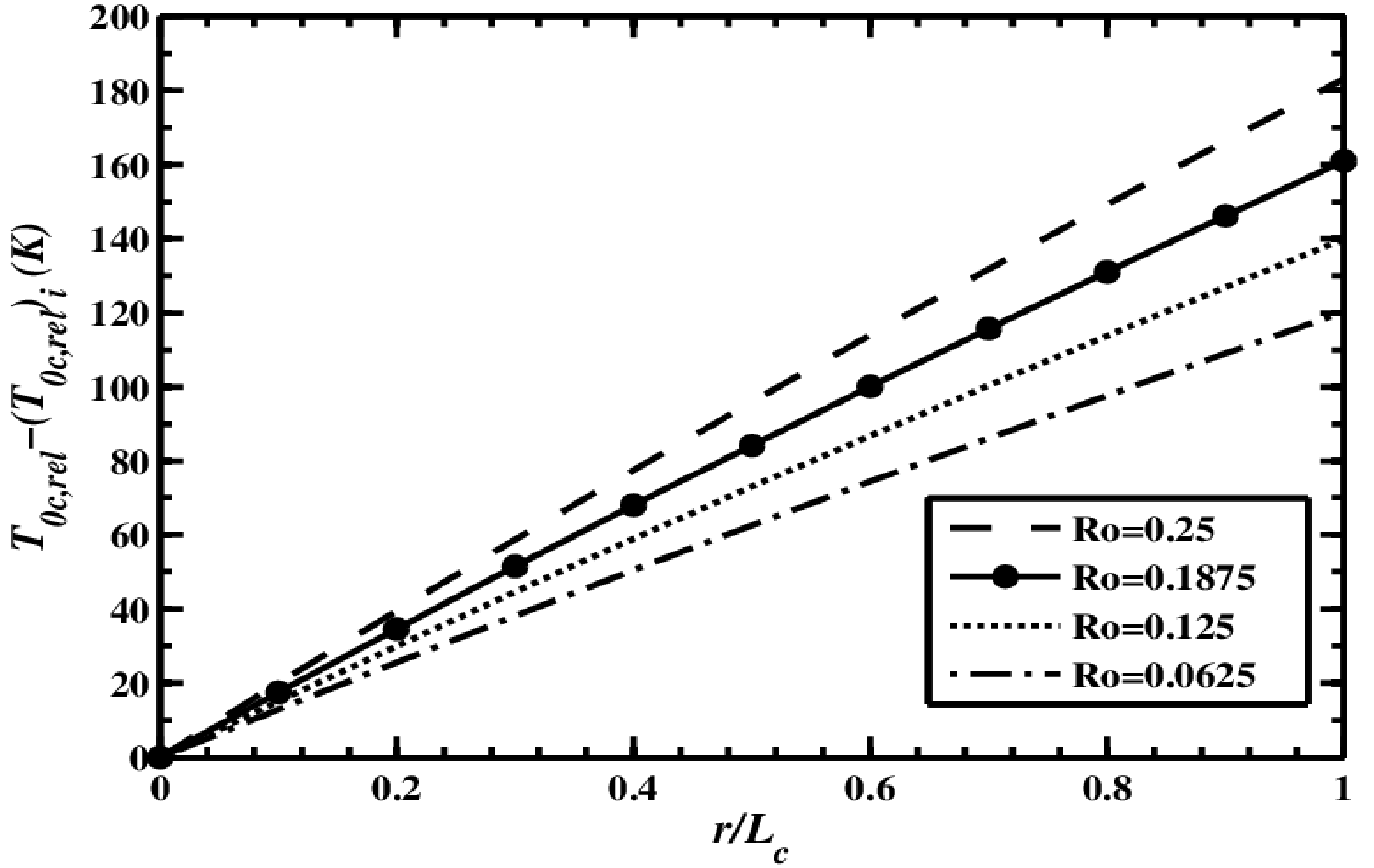
$$\eta = \frac{\left(T_{0c,rel}\right)_o - \left(T_{0c,rel}\right)_i}{T_{b,m} - \left(T_{0c,rel}\right)_i}$$

$$\varepsilon = \frac{T_g - T_{b,m}}{T_g - \left(T_{0c,rel}\right)_i}$$

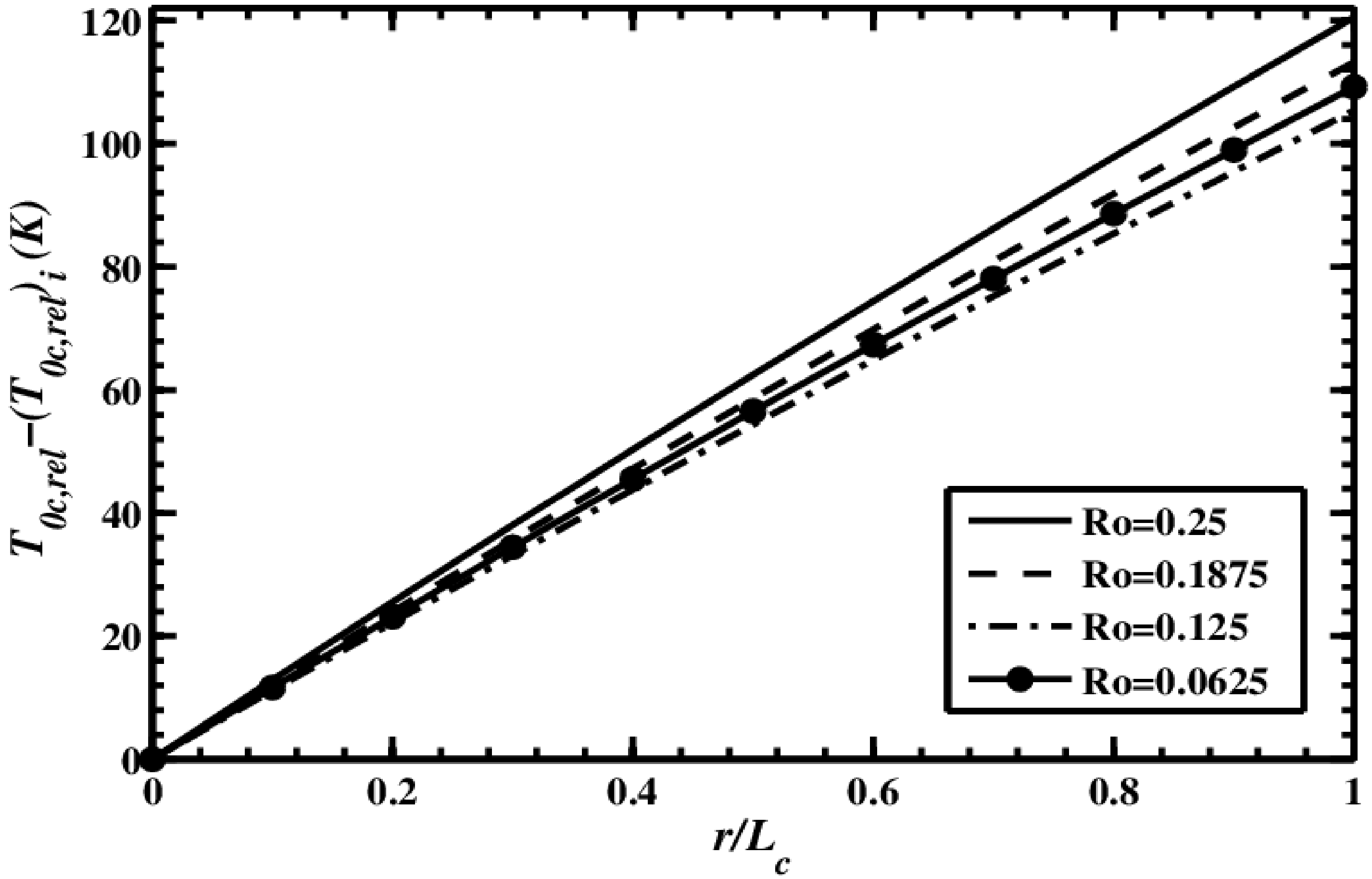
$$m^* = \frac{\varepsilon}{\eta(1-\varepsilon)}$$

Comparison of variation of convective effectiveness as a function of non-dimensional coolant flow, for non-rotating case, radially outward flow and radially inward flow case: prediction of present CFD computation

EFFECT OF ROTATION NUMBER



Change in relative total temperature for the radially outward flowing coolant case at different rotation numbers



Change in relative total temperature for the radially inward flowing coolant case at different rotation numbers

CONCLUSIONS

1. If only rotation is considered for the orthogonally rotating coolant channel then, the relative total temperature for radially outward flow case increases from inlet to outlet, while it decreases for the radially inward flow case.

2. Conventional analysis for static internal blade cooling has been extended to include the effects of finite thermal conductivity of the blade material and corresponding CFD simulations have been performed to validate the theory.

3. Finally, the effect of rotation with external heat transfer for orthogonally rotating coolant channel has been considered and analytical expressions have been derived which improve the static blade analysis by incorporating effects of rotation on the coolant relative total temperature.

4. Due to the combined effect of Coriolis force, centrifugal force and inertia force, heat transfer coefficients vary in leading and trailing side depending on the direction of Coriolis force for both the cases of radially outward flowing and radially inward flowing case.

5. Using the static blade analysis for predicting the coolant flow requirement leads to an over prediction of the coolant requirement for radially outward flowing case and under prediction in case of radially inward flowing case for same cooling effectiveness.

MAJOR REFERENCES

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