Large-eddy Simulation of Impinging Jet Heat Transfer

Bin Zhao

Department of Mechanical & Industrial Engineering
University of Illinois at Urbana-Champaign

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Acknowledgements

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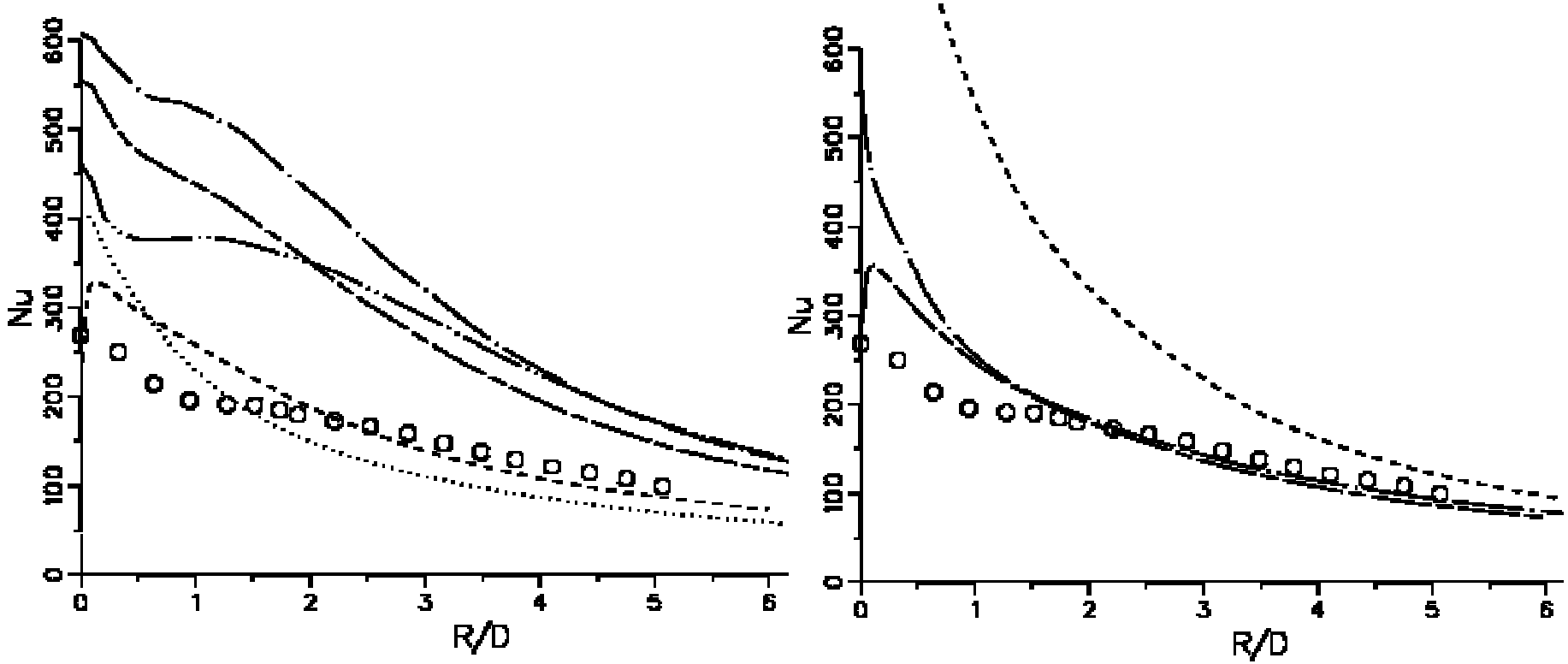
Background

- Heat transfer from impinging jets is difficult to calculate with traditional methods (k-ε wall laws).
- Particle motion is likely to be strongly affected by details of turbulent impinging of jet on NF in caster.

K-ε simulation of impinging jet



Details can be found in "Validation Exercise CFD 95, **Impinging Turbulent Round Jet with Heat Transfer**", A. Pollard (Coordinator, Chair), S. McIlwain (Coordinator), et.

http://www.cfdsc.ca/english/benchmarks/cfd95/index.html

Objective

- Study the transient behavior of fluid flow in impinging jet.
- Study the turbulent heat transfer characteristic of the impinging jet.

Previous Work

- LES flow simulations have insufficient grid refinement near walls for heat transfer.
- High grid density needed near the impingement plate in order to get the correct prediction of heat transfer rate without wall models.

Grid Spacing Restriction (I)

The distance between the first grid point and the plate must satisfy the following restriction in order to predict the heat transfer rate correctly

$$\Delta x^* \le \frac{1}{Nu_{\max}} \qquad \text{while} \qquad Nu_{\max} \approx \frac{\Delta\theta_{\max}}{\Delta x^*} \le \frac{1}{\Delta x^*}$$

Grid Spacing Restriction (II)

Definitions:

The dimensionless temperature is defined as:

$$\theta = \frac{T_{inlet} - T}{T_{inlet} - T_{surface}} \leq 1$$

The dimensionless length is defined as:

$$x^* = \frac{x}{D}$$

The Nusselt number is the dimensionless temperature gradient at the impingement plate. Using a simple first-order differencing scheme, the Nusselt number can be calculated using the following equation:

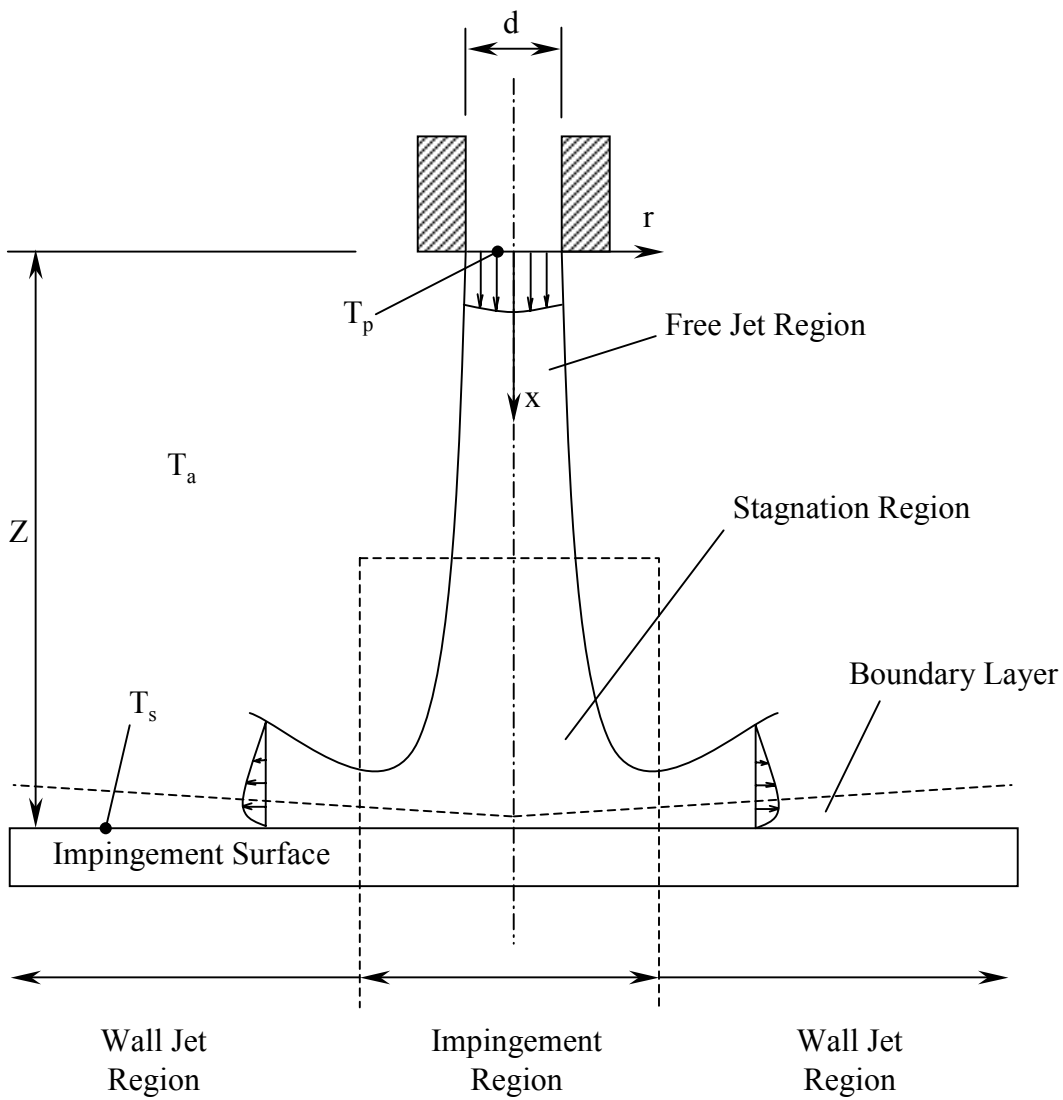
$$Nu = \left.\frac{\partial \theta}{\partial x^*}\right|_{wall} \approx \left.\frac{\Delta \theta}{\Delta x^*}\right|_{wall}$$

Test problem Setup

The simulation results are compared with the experiment of Hollworth and Gero (1985).

- Unconfined air jet impinging normal to an isothermal flat surface.
- The temperature of the jet is equal to that of the ambient air.
- The target plate is cooled below the ambient temperature.

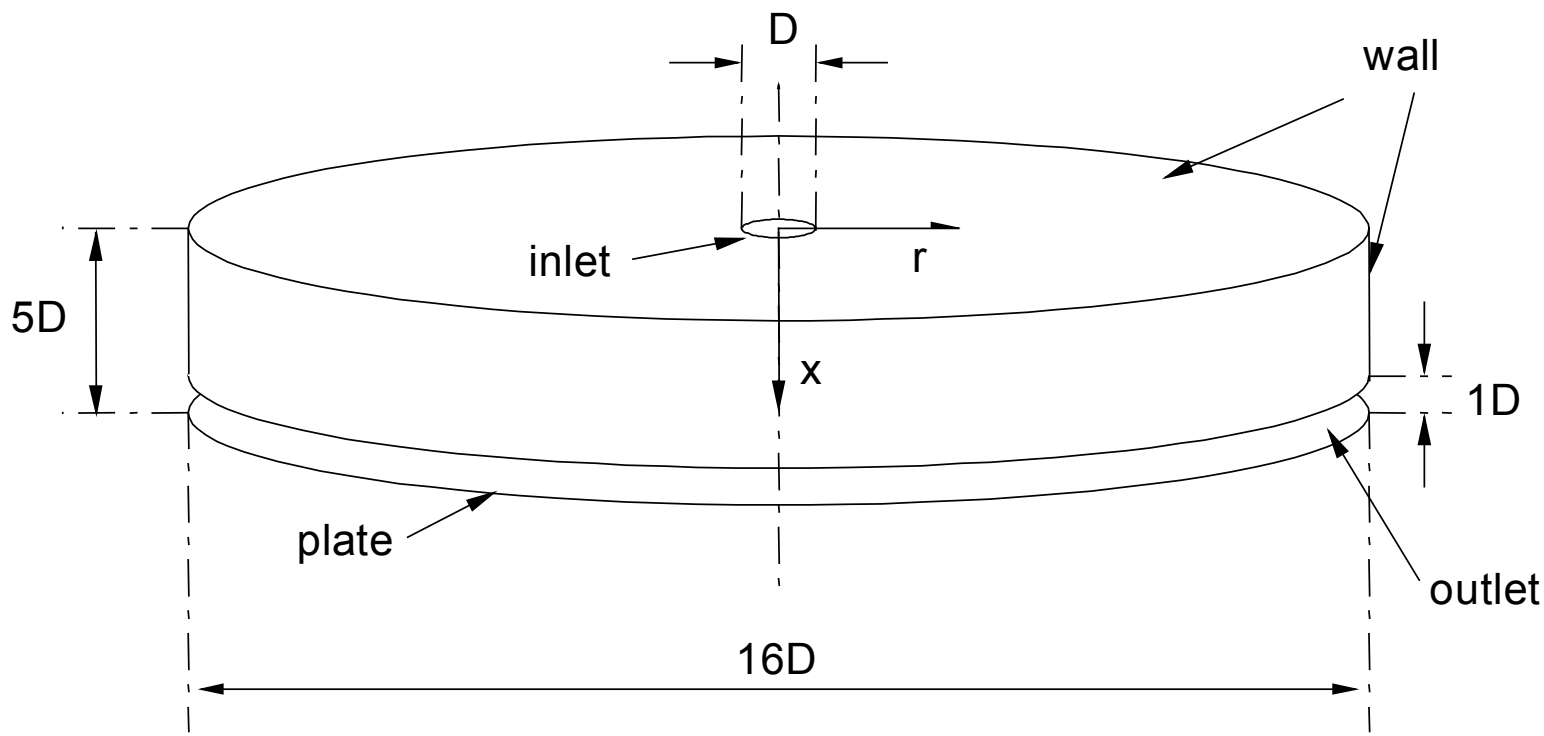
Schematic of an Impinging Jet



Test Problem Details

d	Inlet diameter	$10\,mm$
Z	Nozzle to plate distance	$50\,mm$
T_p	Inlet temperature	$24 \sim 25\,°C$
T_a	Ambient temperature	T_p
T_s	Impingement surface temperature	$7 \sim 8\,°C$
ρ	Density of air	$1.2\,Kg/m^3$
μ	Molecule viscosity of air	$17.85\times10^{-6}\ Ns/m^2$
k	Thermal conductivity	$0.25W/mK$
Pr	Prandtl number $\mu C_p/k$	0.71
Re	Renolds number	5000
V_b	Inlet bulk velocity $\dfrac{\mathrm{Re}\,\mu}{\rho d}$	$7.4375\,m/s$

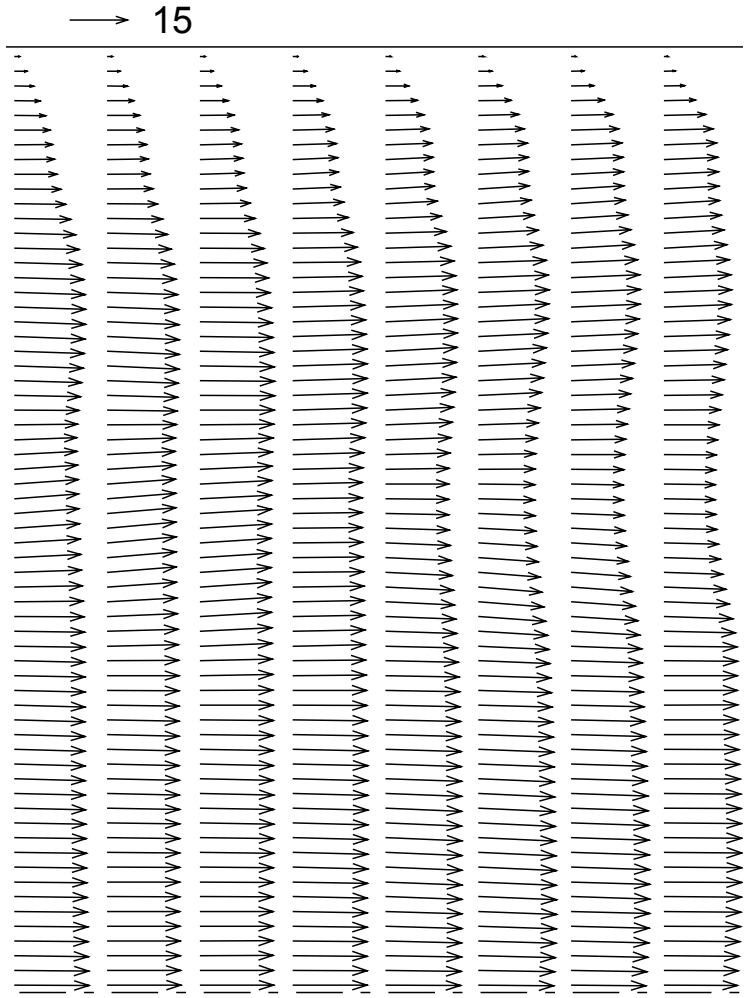
Simulation Domain

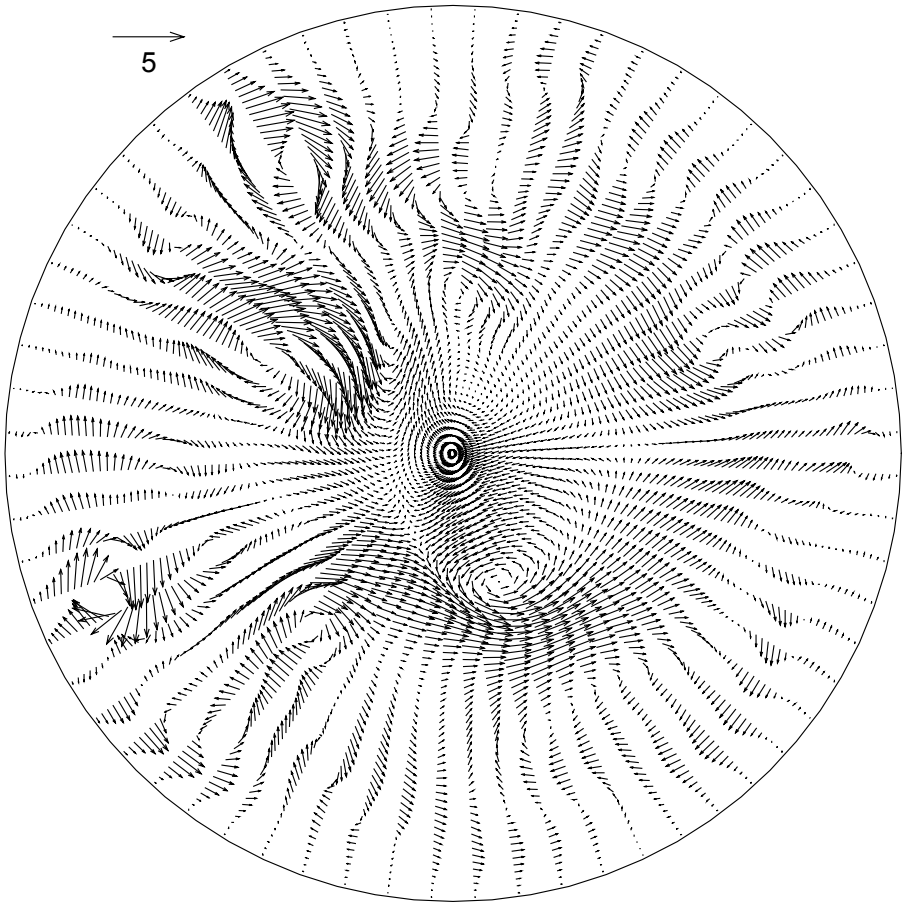


Simulation Overview

- Use LES3D code (2nd order accurate in space and time).
- Use Smagorinsky turbulence model.
- A cylindrical mesh with staggered discretization of velocities and pressures.
- Grid spacing increases from 0.0218 to 0.374 in the radial direction.
- Instantaneous flow fields from a fully developed turbulent pipe flow simulation are prescribed as the inlet to the computational domain in a time-varying manner.
- Same as last meeting but with finer grid in the circumferential direction.

Jet Inlet – Fully Developed Pipe flow





Simulation Details

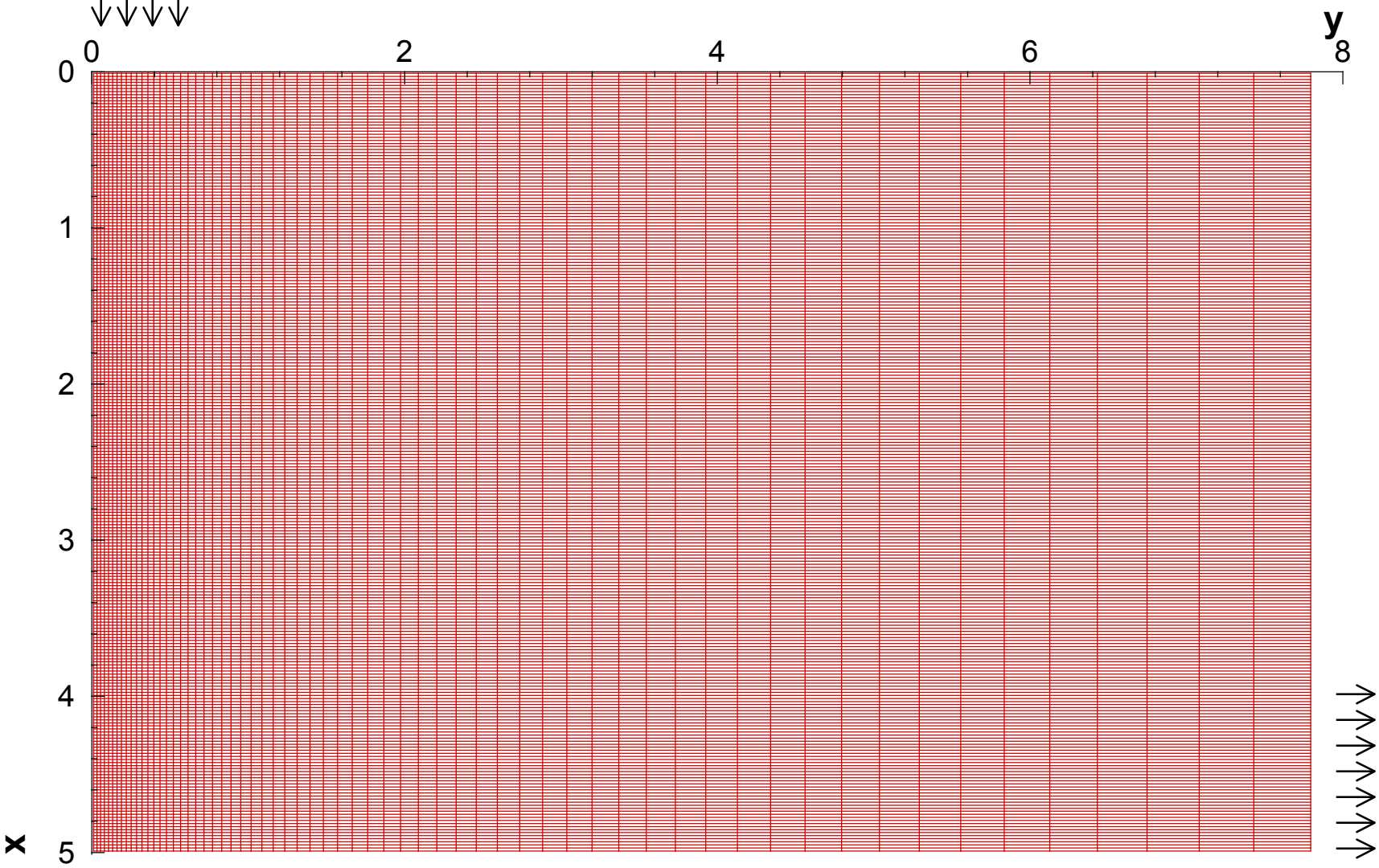
Grid	256×64×64	
d	Inlet diameter	1
$\bar{z}$	Nozzle to plate distance Z/d	5
θ_p	Inlet temperature	0
θ_s	Impingement surface temperature	1
ρ	Density	1
μ	Molecular viscosity	0.002925058
Pr	Prandtl number	0.71
Re	Renolds number $V_b d/\nu$	5000
$V_b = \dfrac{4\dot{m}}{\pi d^2}$	Inlet bulk velocity	14.6253

$\theta = \dfrac{T - T_p}{T_s - T_p}$ is the dimensionless temperature.

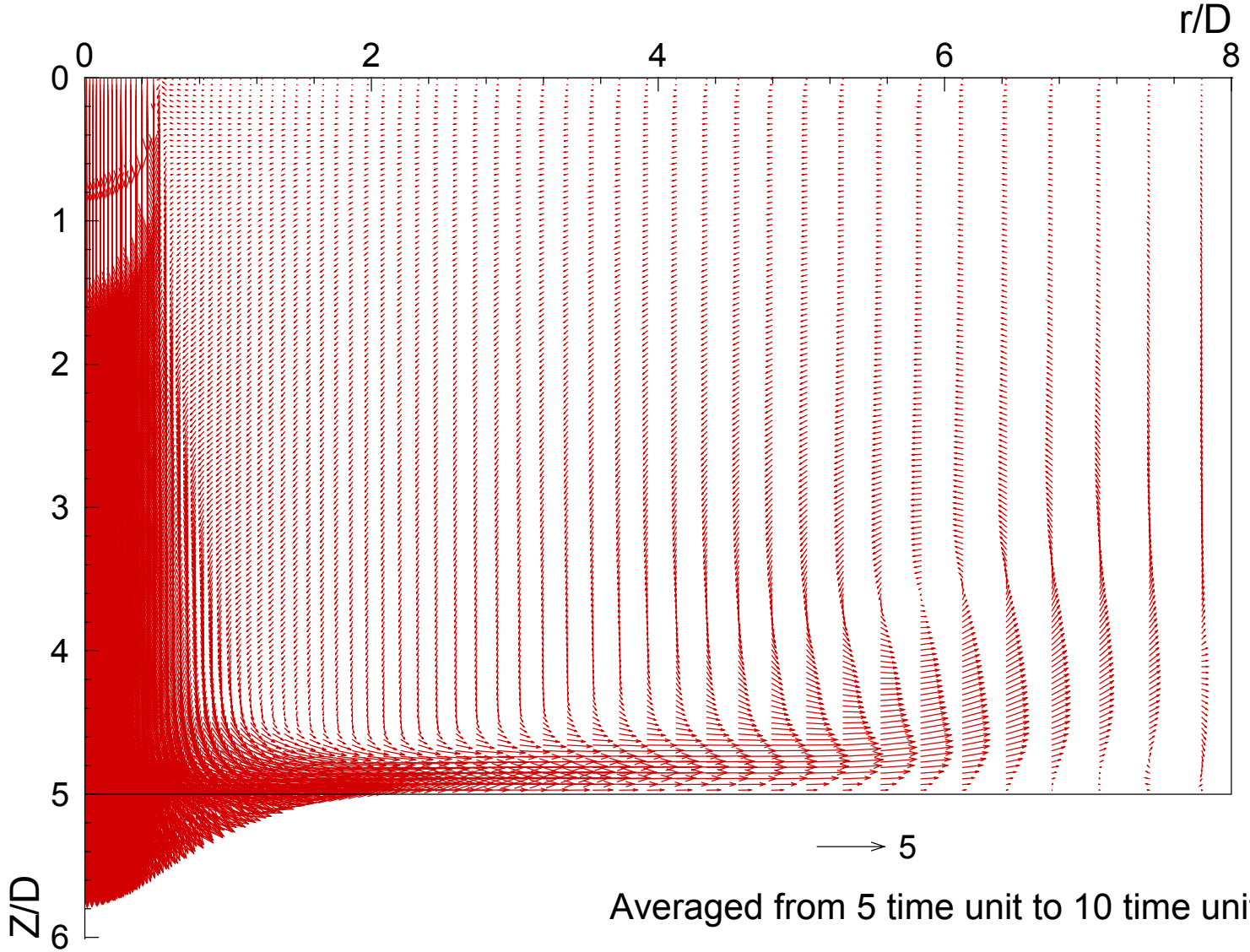
$t_c = d/V_b = 0.068375$ is the characteristic time, corresponds to 0.0013445s in the Test problem. So 1 time unit correspond to 0.01966s in the test problem.

The time step used in the simulation is 0.0001 time units.

Mesh Used in The Simulation

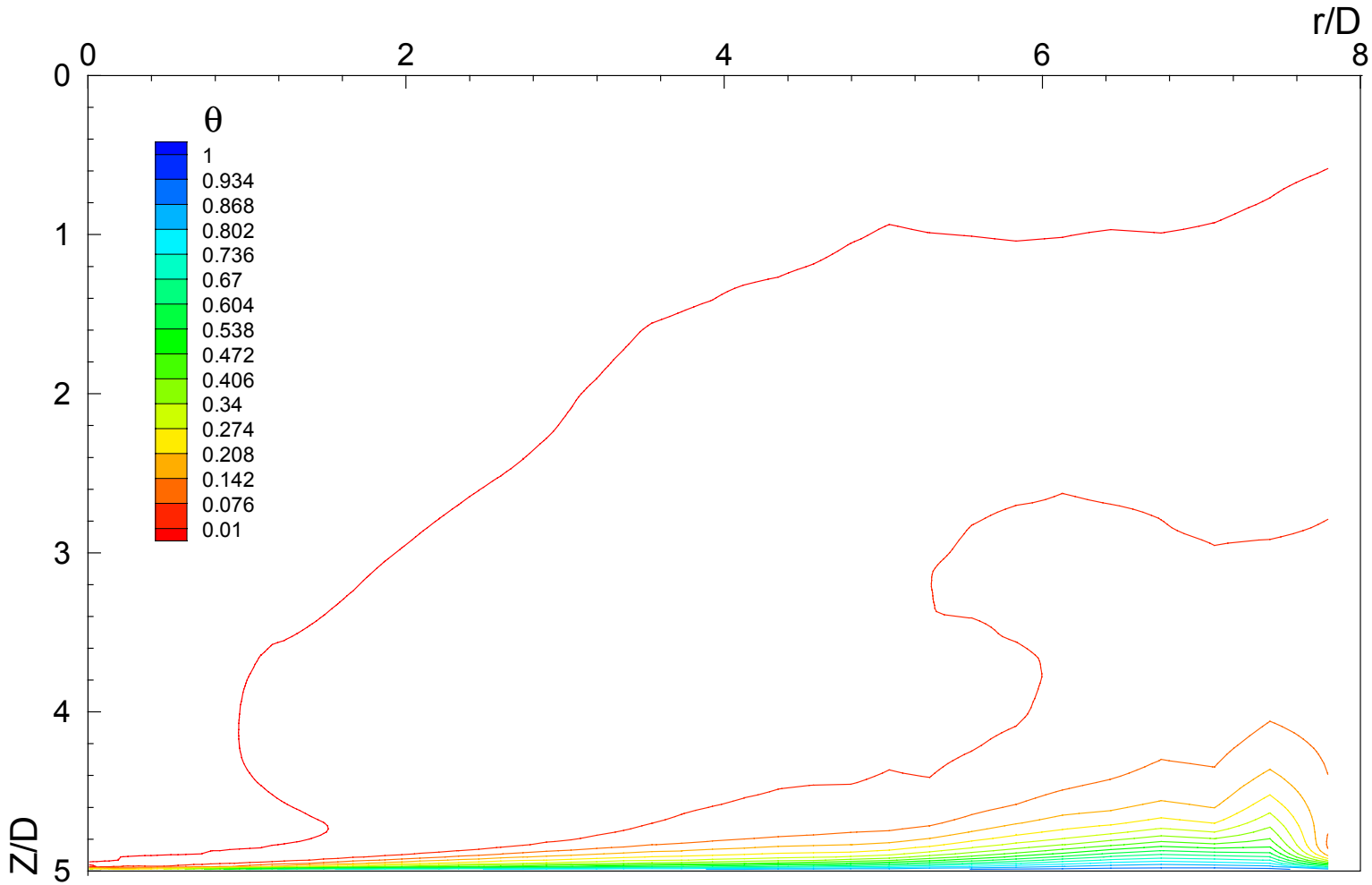


Mean Velocity Field



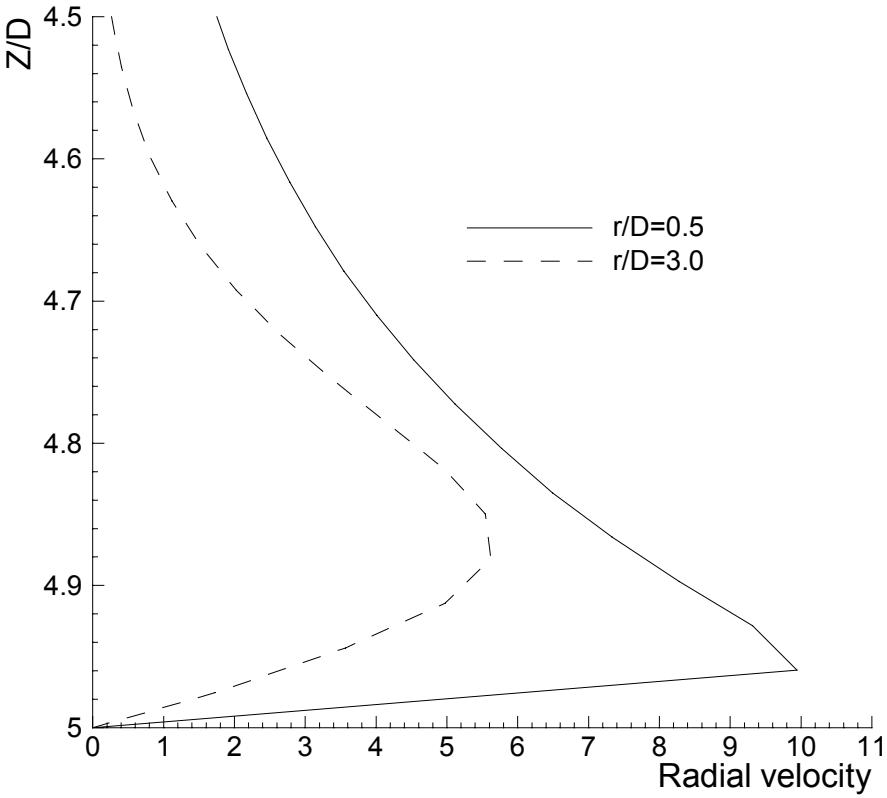
Averaged from 5 time unit to 10 time unit

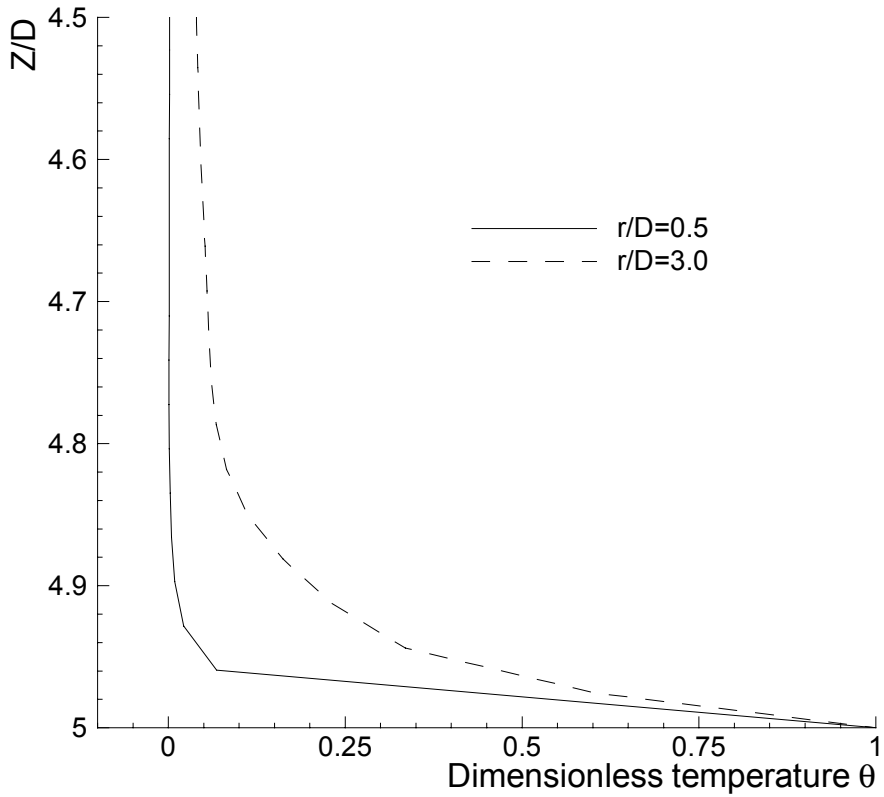
Mean Temperature Field



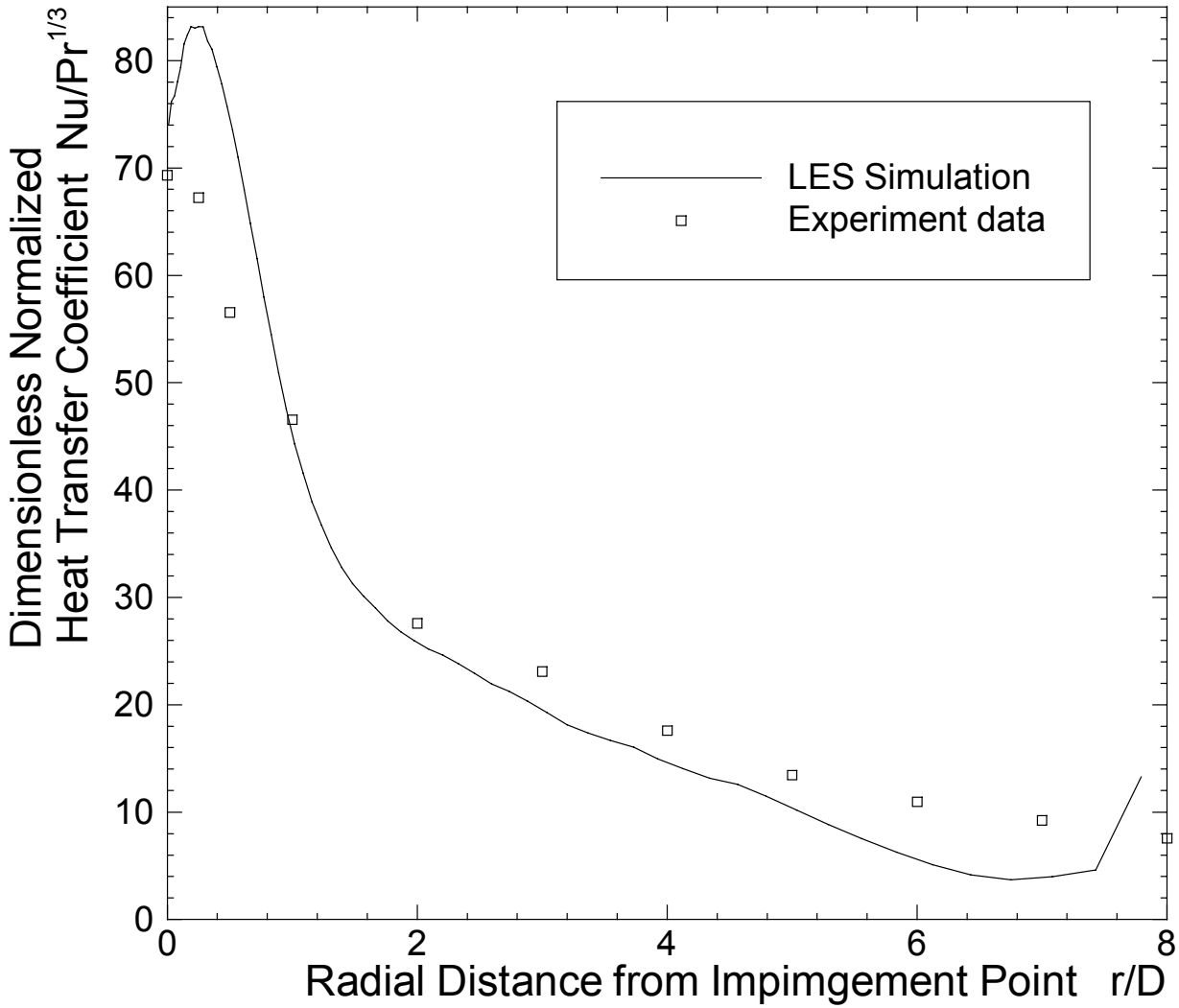
Averaged from 5 time unit to 10 time unit

Velocity and Temperature Profiles



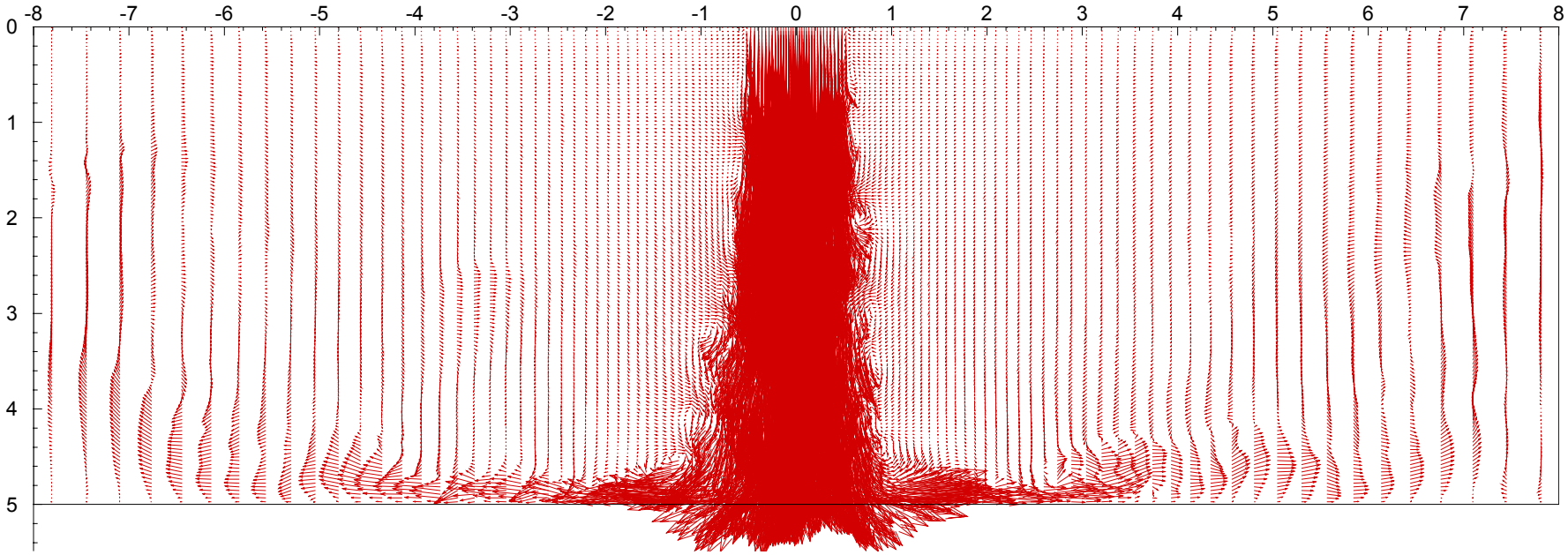


Mean Nusselt Number Distribution



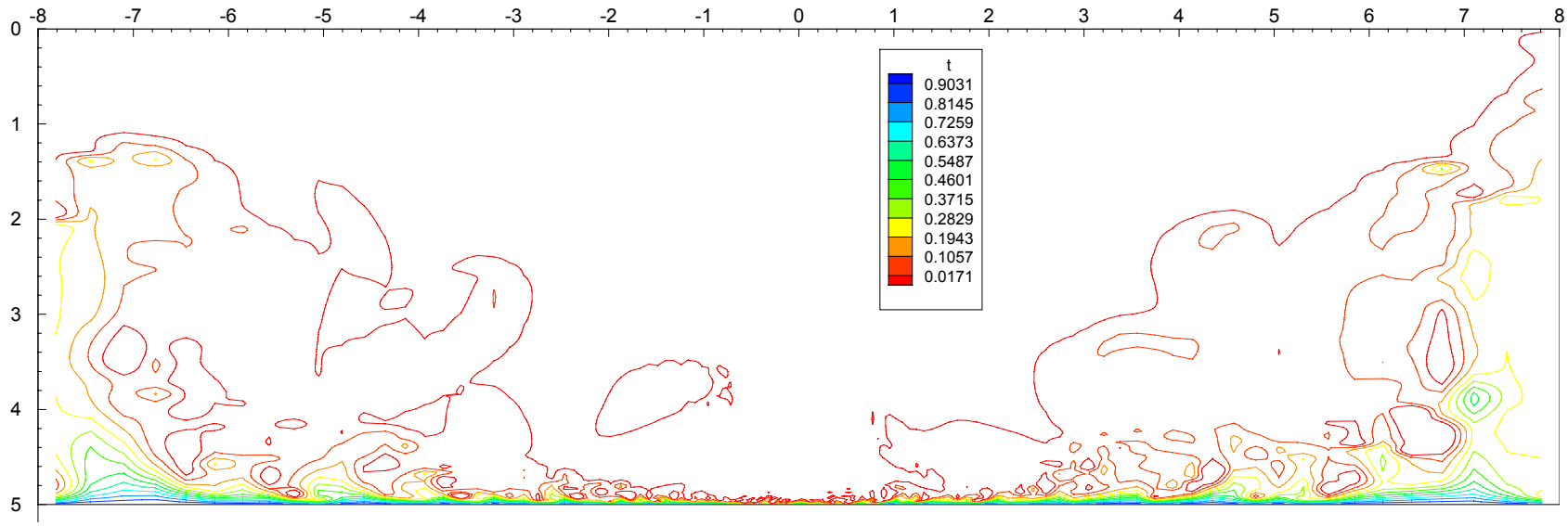
Averaged from 5 time unit to 10 time unit

Instantaneous Velocity Field



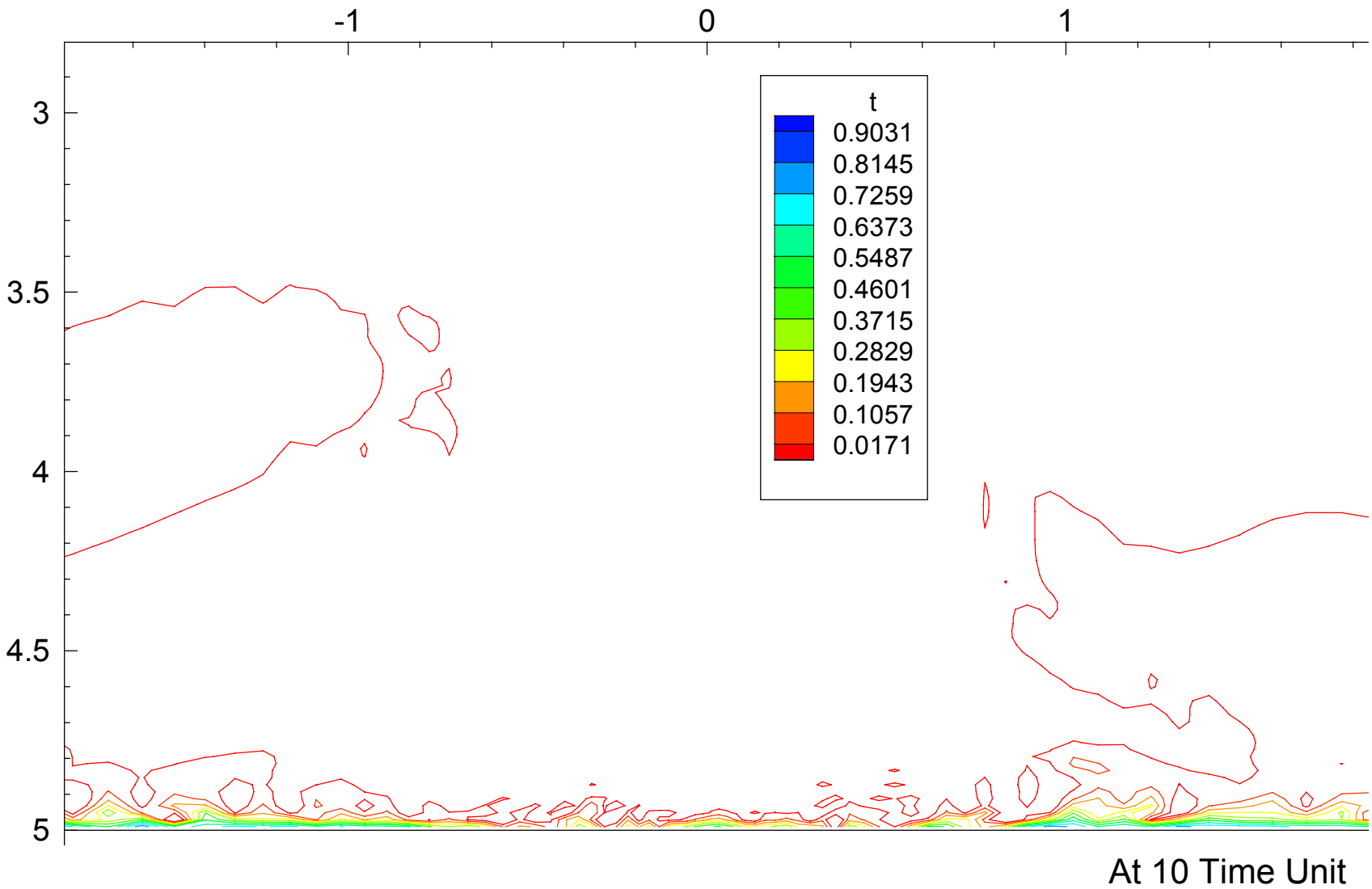
At 10 time unit

Instantaneous Temperature Field



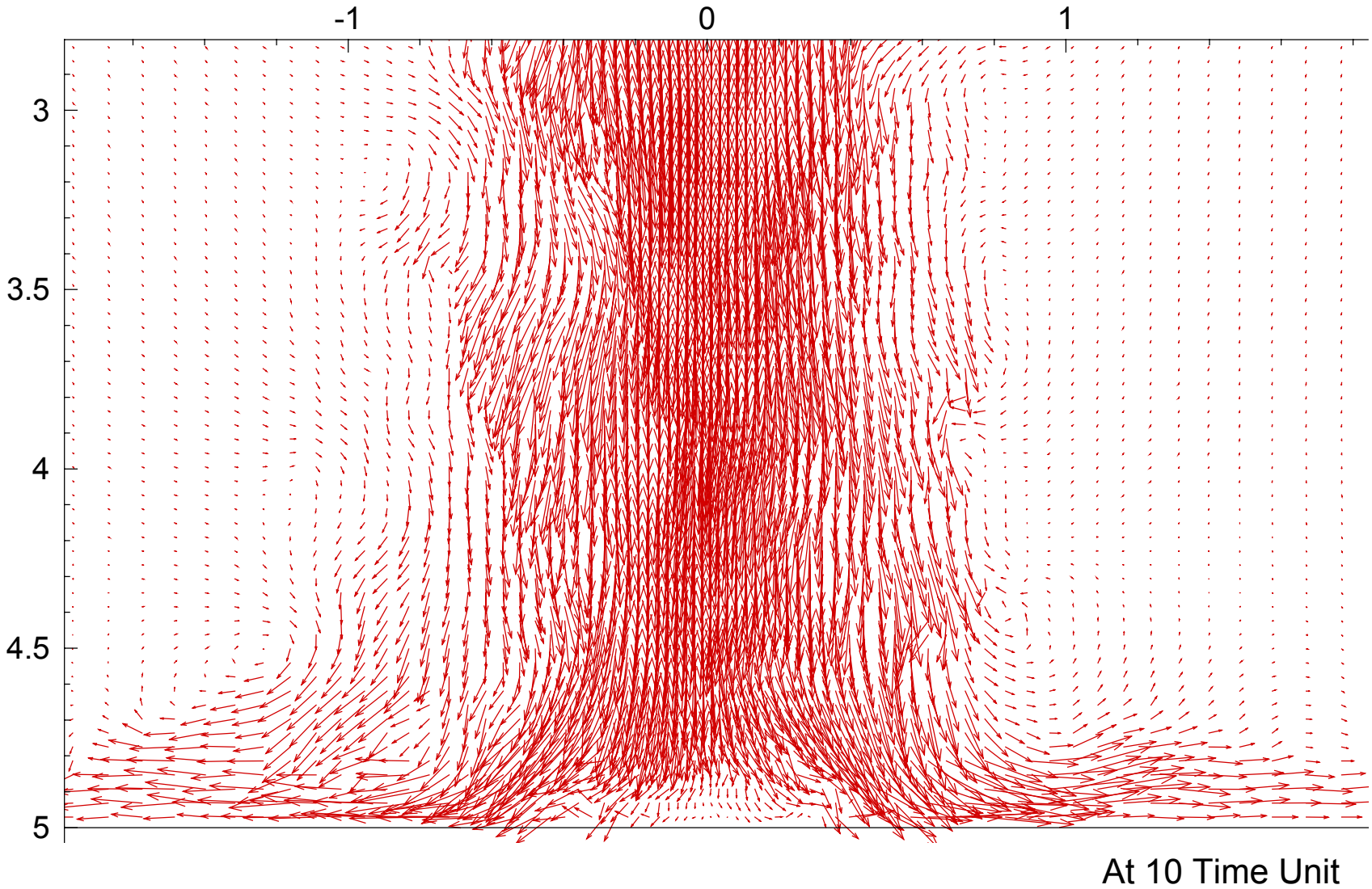
At 10 time unit

Instantaneous Temperature Field at Impingement Region



At 10 Time Unit

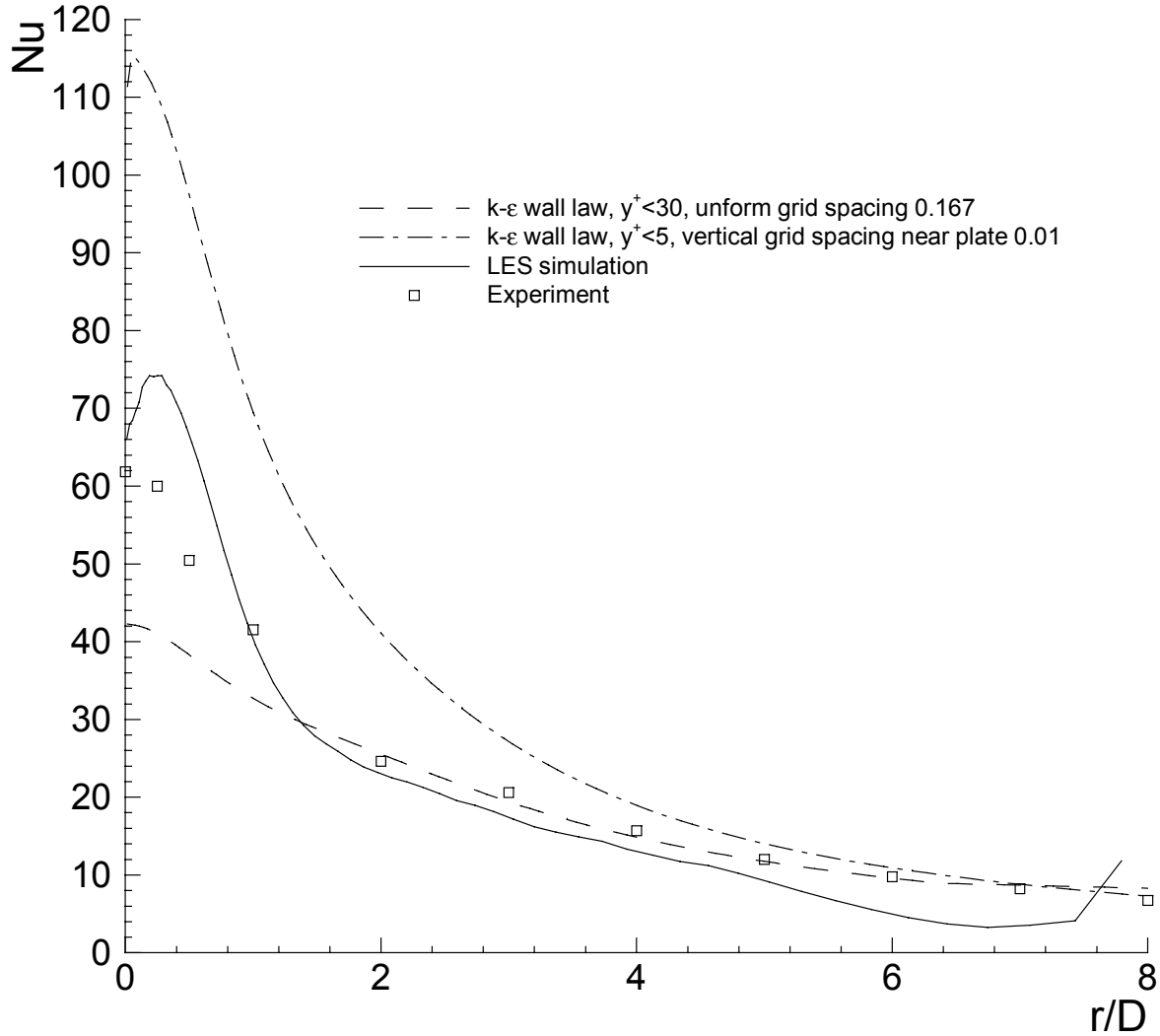
Instantaneous Velocity Field at Impingement Region



Observations

- Very steep gradients at impingement point, which require very fine mesh to capture.
- The shear between jet and ambient fluid generates vortexes which entrain fluid and cause jet to expend.
- The impact of the vortexes on the wall gives a local maximum heat transfer coefficient.
- Very small tight vortexes are generated at impingement point.
- The vortexes move along the bottom plate, while growing in size and slowing down.
- Unphysical numerical noise originates from impingement point and spreads out.

k-ε simulation of impinging jet



Mesh Calculation of a Real Caster

ARMCO Thin Slab Mold:

Casting speed	0.0254 m/s
Mold thickness	0.132 m
Mold width	0.984 m
Inlet height	0.077 m
Inlet width	0.032 m
Molecular viscosity	0.0056 kg/ms
Density	7020 kg/m^3
Prandtl number	0.1

Empirical Equations for Nu Estimation

$$Nu = K\,\mathrm{Re}^{a} \qquad \frac{Nu_{steel}}{\mathrm{Pr}_{steel}^{1/3}} = \frac{Nu_{air}}{\mathrm{Pr}_{air}^{1/3}}$$

Where:

$$a = 0.82 - \frac{0.32}{\left(1 - 1.95\left(\frac{r}{D}\right)^{1.8} + 2.23\left(\frac{r}{D}\right)^{2}\right)\left(1 - 0.21\left(\frac{x}{D}\right)^{1.25} + 0.21\left(\frac{x}{D}\right)^{1.5}\right)}$$

K is a constant for a given r/D.

Details can be found in:

K. Jambunathan, E. Lai, et., “A review of heat transfer data for single circular jet impingement”, Int. J. Heat and Fluid Flow, Vol. 13, No. 2, June 1992

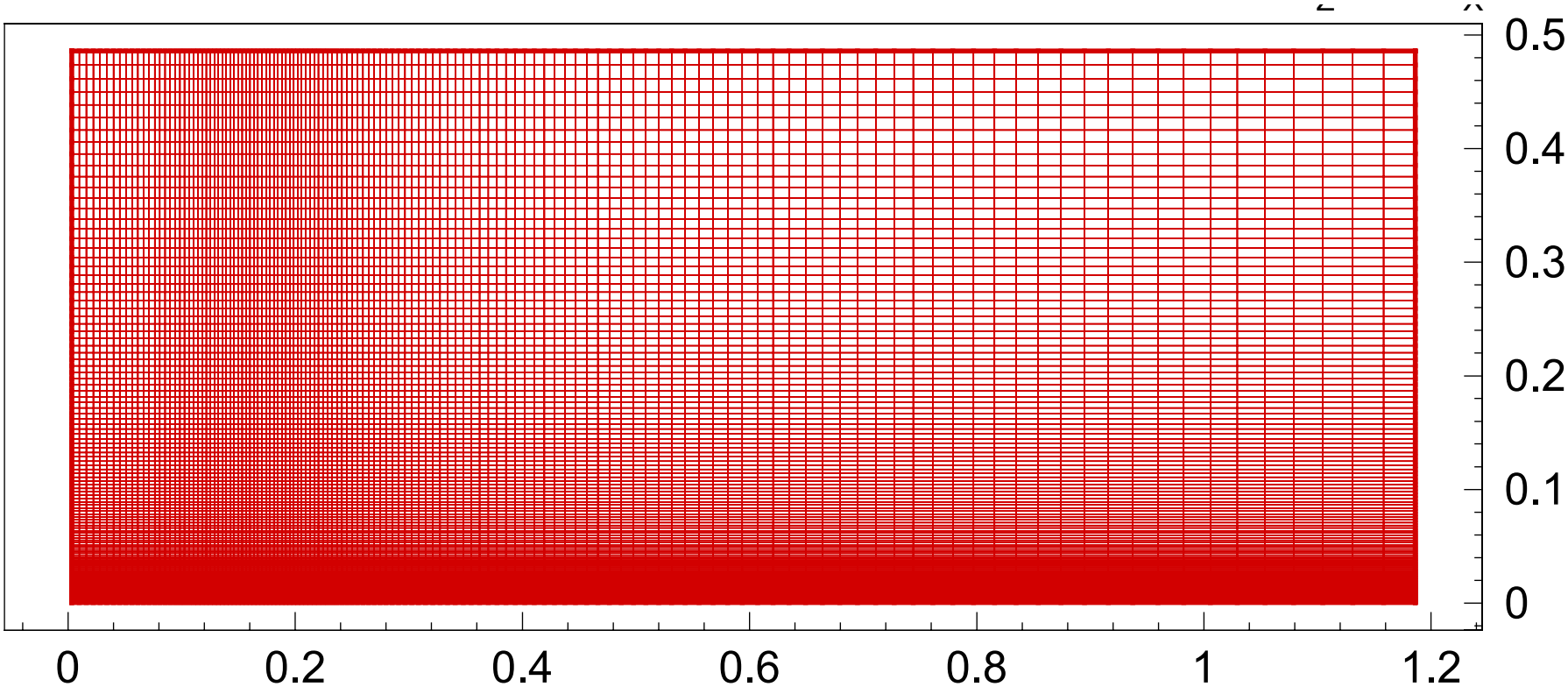
Grid Spacing Needed for Heat Transfer Prediction

Jet Reynolds number	47,000
Reference length	0.056 m/s
Nu_{max} at narrow face	72.4
Nu_{max} at wide face	24.1
Grid spacing needed at the narrow face wall	0.00774 m
Grid spacing needed at the wide face wall	0.00232 m

Using non-uniform grid in all three directions, we can get a mesh of 120×50×132, total of 792,000 grids, which satisfy the grid spacing restriction.

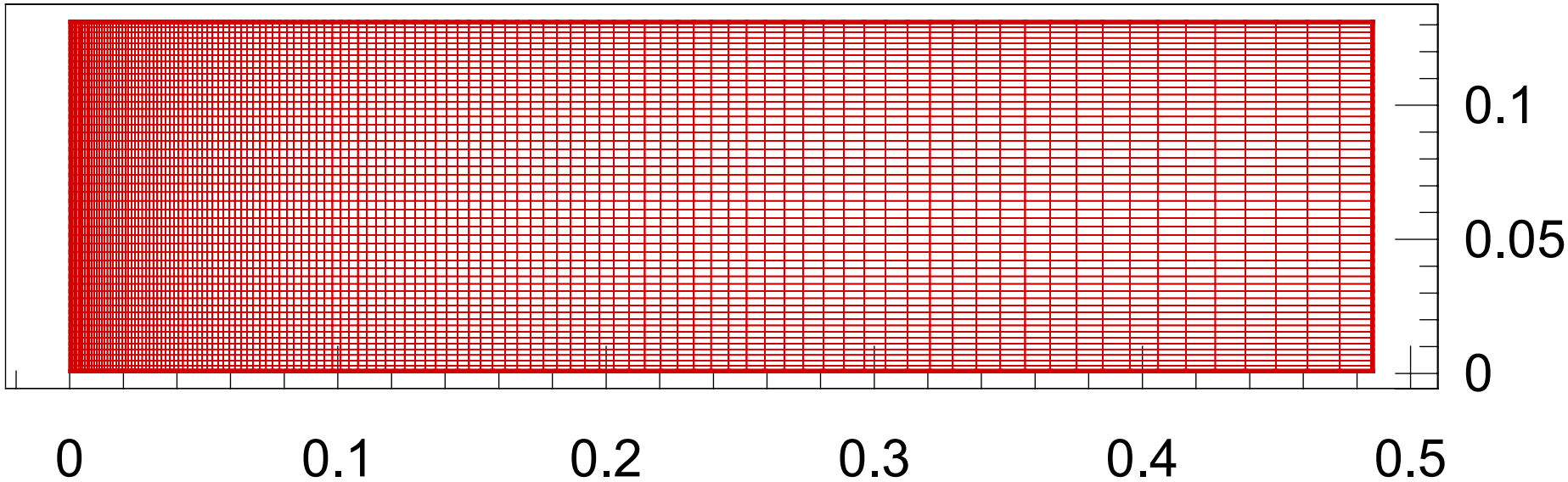
Mesh for a Real Caster

Front view:

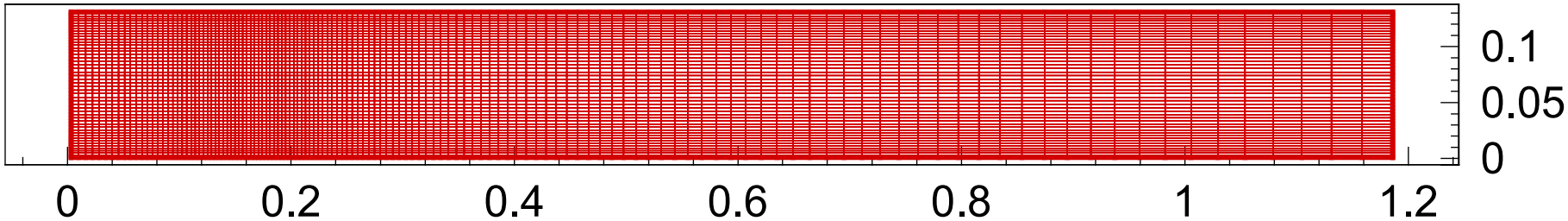


Mesh for a Real Caster

Side view:



Top view:



Problems Needed to Be Addressed in Future Simulations

- Even the 5,000 Re impinging jet simulation with a fine grid had unphysical temperature undershoots due to sharp gradients in the impingement region.
- Non-uniform grid in 2 or 3 directions is needed to give finer grid at the heat transfer wall.
- Pressure boundary condition is needed to describe the boundary in a more realistic way.

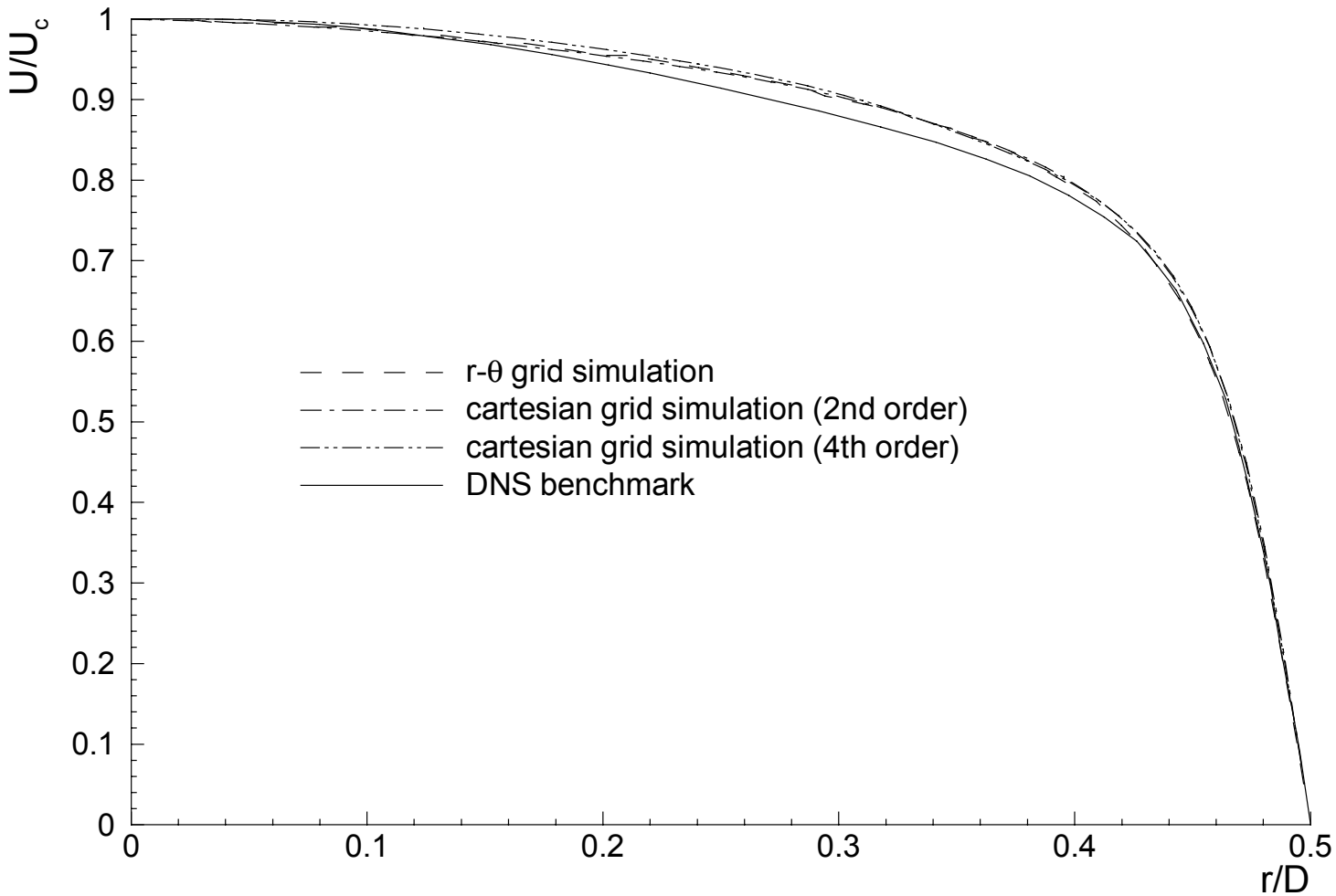
Features of New CART3D Code

- Flexible grid arrangement which can deal with more complex geometries.
- Grid can have non-uniform spacing in all three directions.
- Pressure Poisson equation is solved using advanced algebraic multi-grid solver.
- Flux limiter which can suppress the unphysical oscillations of the solution.
- Fourth order accurate convection terms which give more accurate solutions.

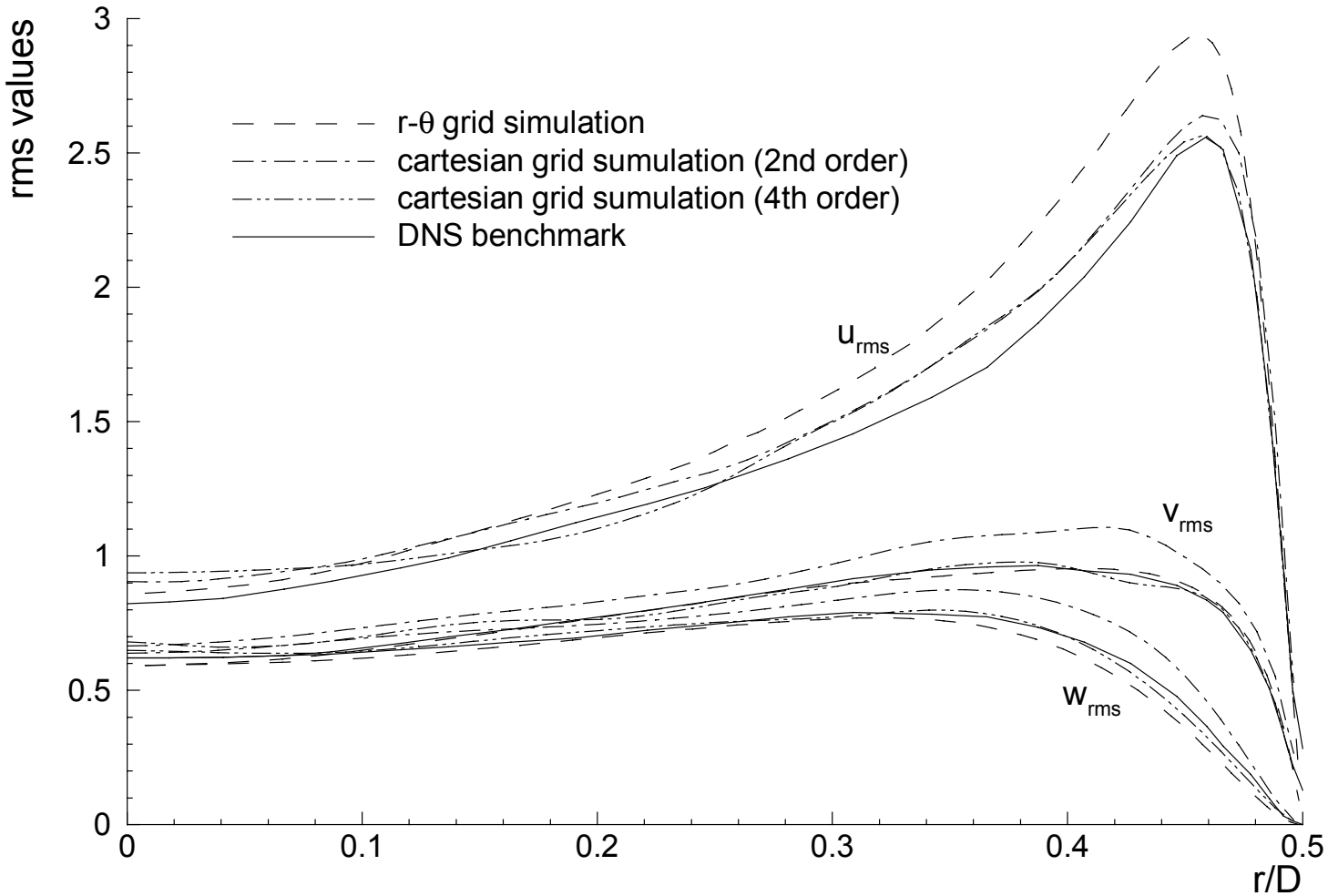
Validation of New Code

- Three simulations of fully developed pipe were carried out.
- Simulations are done in cylindrical grids as well as in Cartesian grids with stair-step approximation of the pipe boundary.
- Simulation were done with 2nd order convection terms and 4th order convection terms respectively.
- Comparison of mean and rms statistics with DNS benchmark results.

Mean Velocity Profile



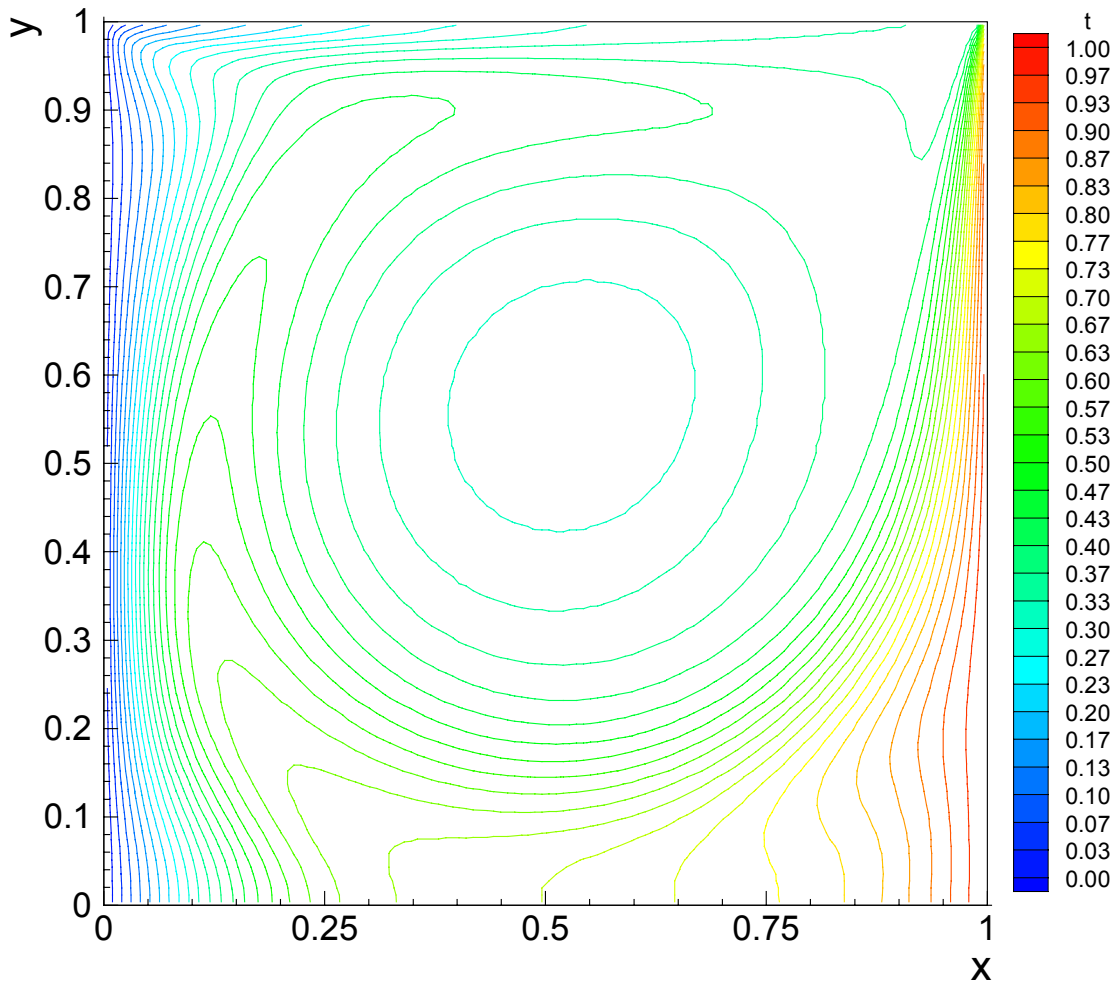
RMS Statistics



Validation of Flux Limiter

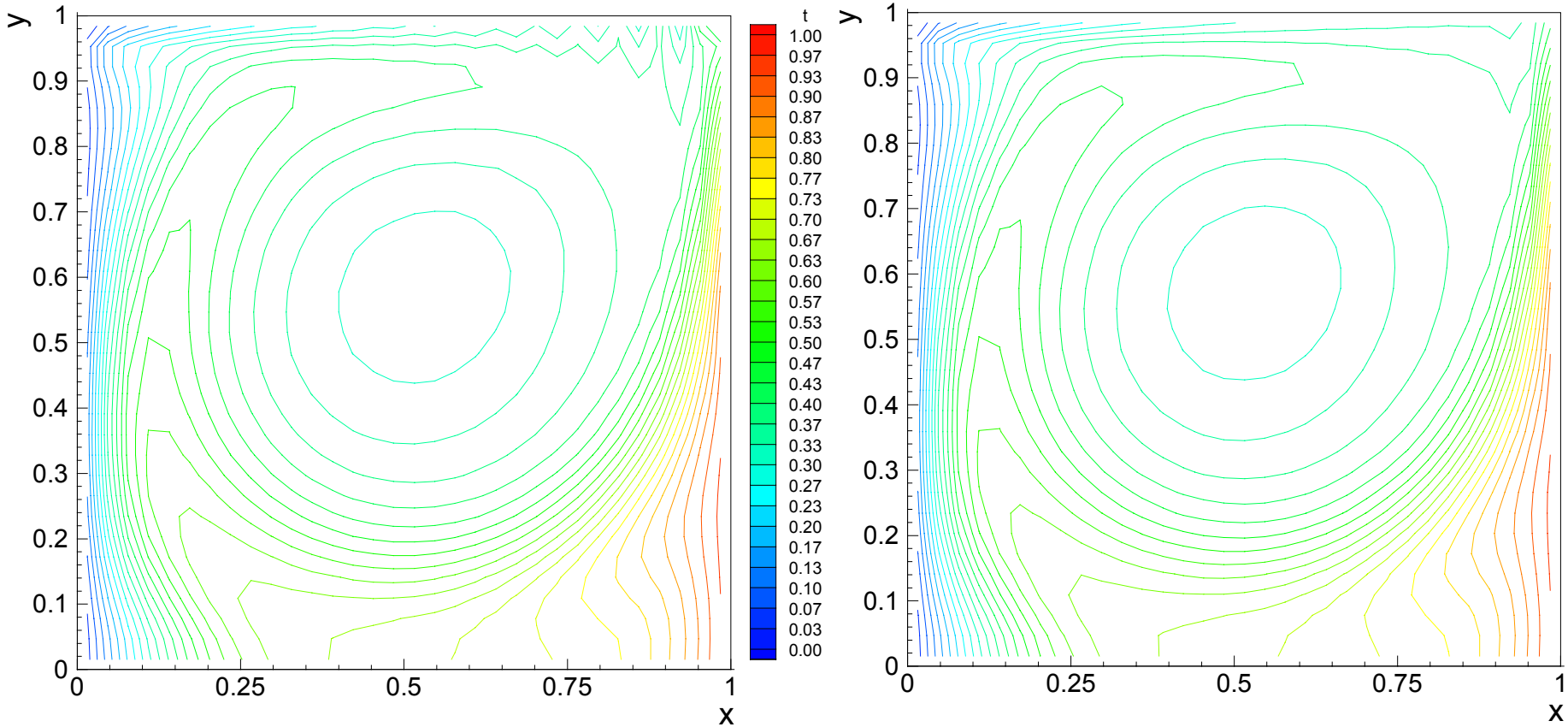
- 2 dimensional driven cavity with heat transfer were simulated to test the flux limiter.
- Flux limiter was applied only to the energy equation.
- Solutions with and without the limiter were compared with each other as well as DNS benchmark solution.
- The test simulations use a grid of 32×32, while the benchmark solution uses a grid of 128×128.
- Results show that the flux limiter suppresses unphysical oscillation while preserving good accuracy.

Temperature Contours



Benchmark Solution (Grid 128×128, no flux limiter)

Temperature Contours (Grid 32×32)



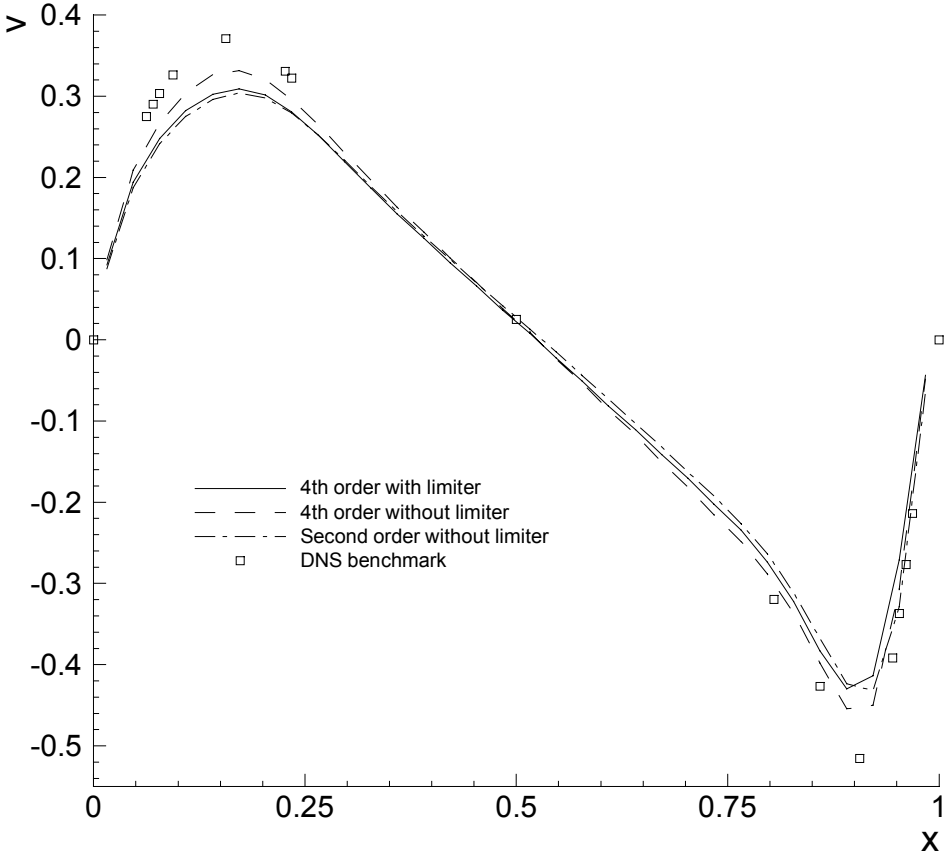
a) Solution without flux limiter

b) Solution with flux limiter

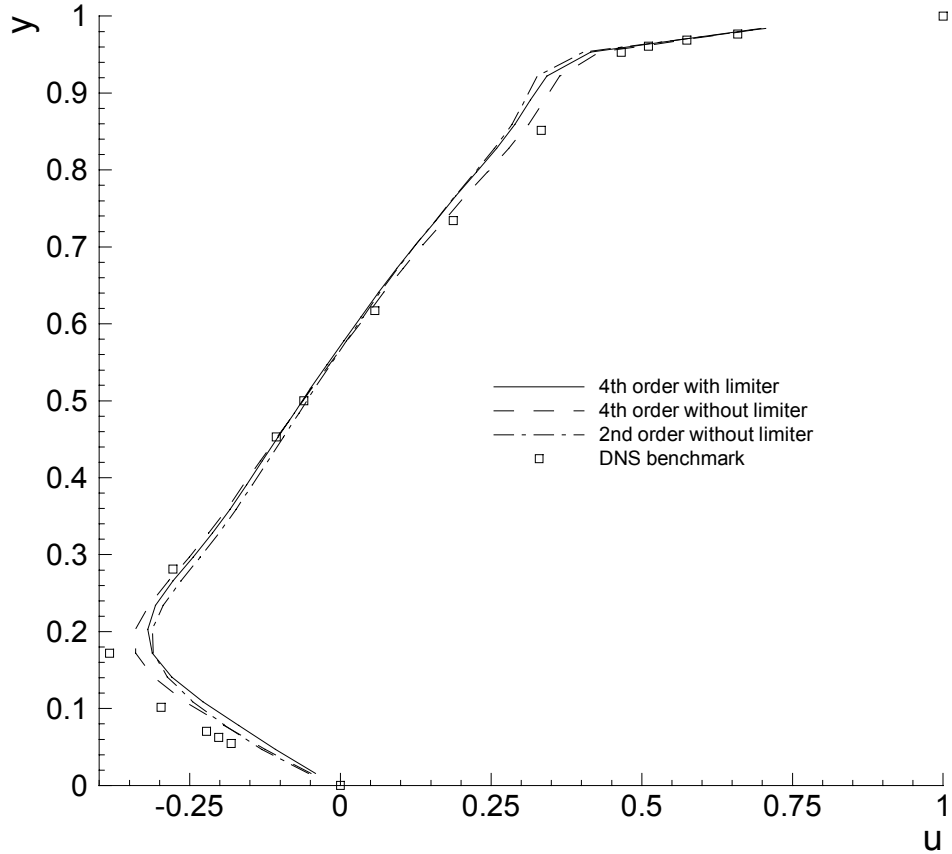
Validation of 4th Order Convection Scheme

- 2 dimensional driven cavity were simulated to test the fourth order convection scheme.
- Three cases were tested: 2nd order without limiter, 4th order without limiter and 4th order with limiter.
- The test simulations use a grid of 32×32, while the benchmark solution uses a grid of 128×128.
- The results show that fourth order convection scheme improves accuracy.

Centerline velocity comparison



a) Velocity profile along horizontal centerline



b) Velocity profile along vertical centerline

Direct Numerical Simulation of Fluid Flow and Heat Transfer in Liquid Mold Flux Layer

Objective

- Study the natural convection flow pattern in the slag layer driven by temperature difference.
- Quantify the heat transfer rate in the slag layer.
- Study the influence of bottom shear on the flow pattern and heat transfer rate.
- Study the influence of changing viscosity on the flow pattern and heat transfer rate.

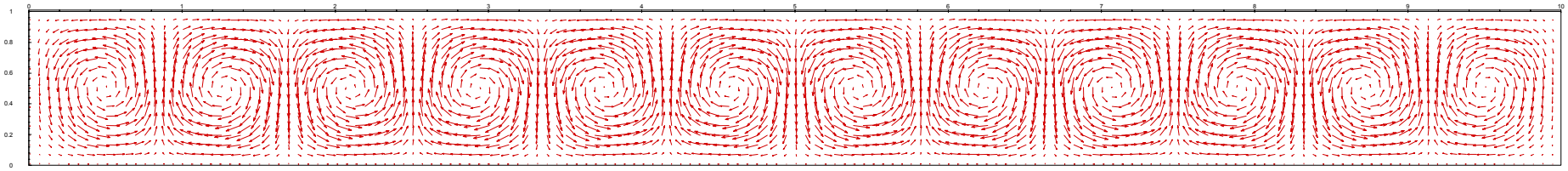
Previous work

- Sivaraj had done some 2 dimensional simulations of the slag layer assuming constant fluid properties (use commercial code “FIDAP” from FLUENT).
- The simulations were done on a coarse 160×16 grid.
- The breakdown of natural convection cell arrays under bottom shear was studied.

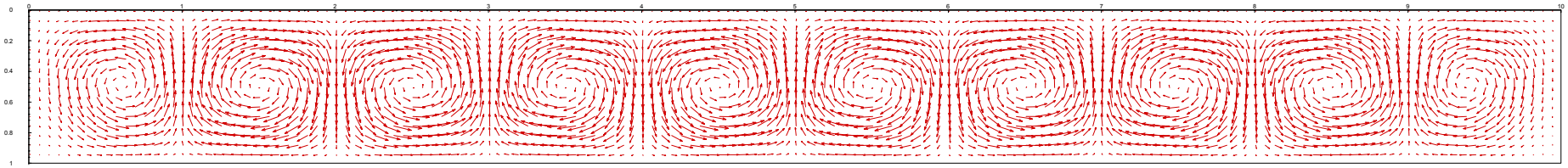
Preliminary results

- Two 2 dimensional simulations were done to get the natural convection pattern in a thin layer with aspect ratio of 10:1.
- A fine 640×64 grid was used.
- The results are compared with differential interferogram measurements of three-dimensional thermal cellular convection in rectangular box (K. R. Kirchartz and H. Oertel).

Velocity Field

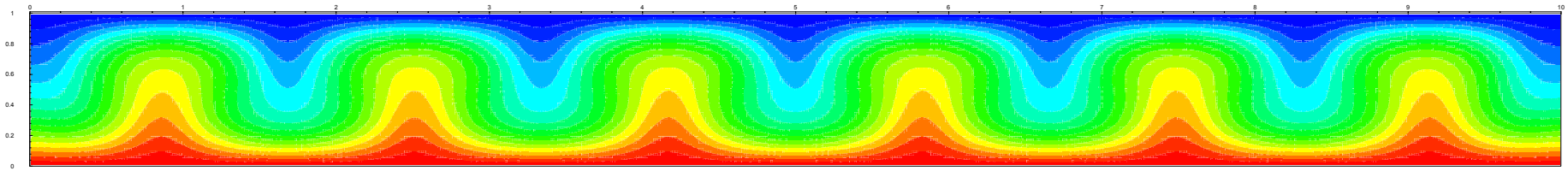


a) Adiabatic side walls



b) Perfect conducting side walls with linear temperature distribution

Temperature field



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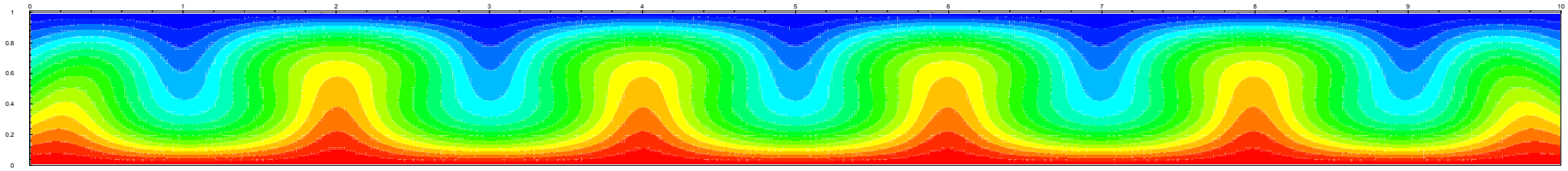
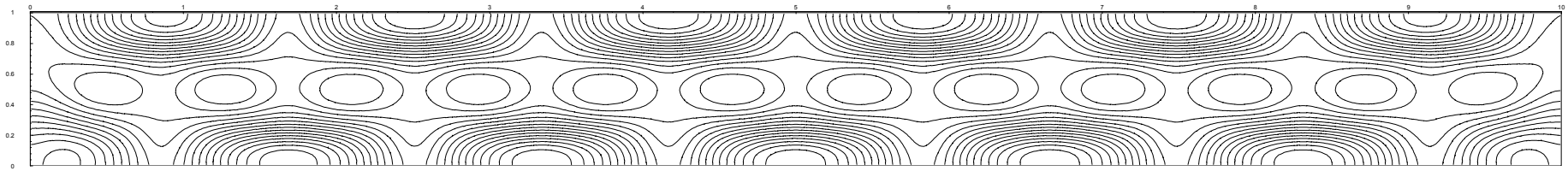


b) Perfect conducting side walls with linear temperature distribution

Contour of vertical density gradients



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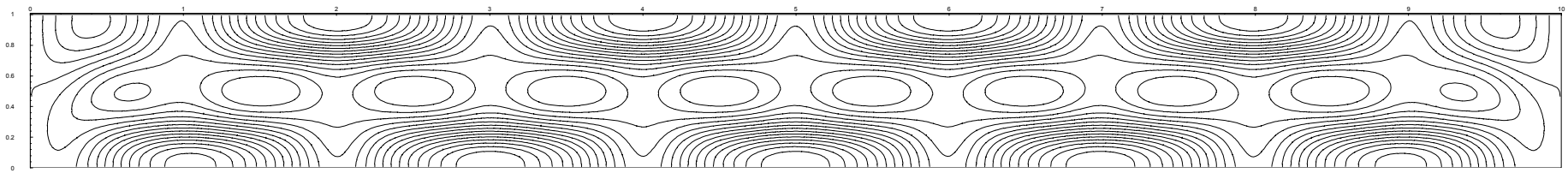
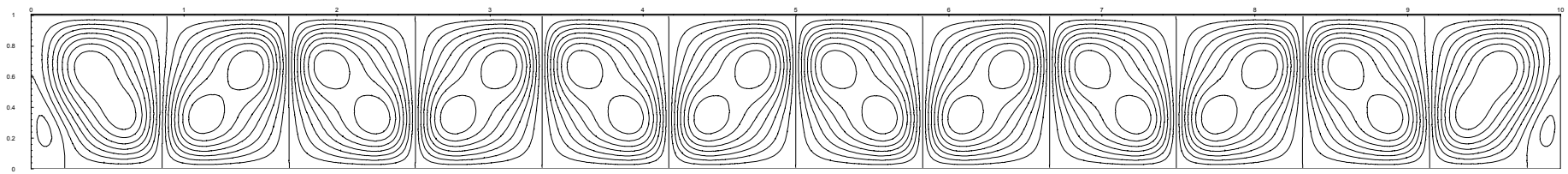


b) Perfect conducting side walls with linear temperature distribution

c) Differential interferogram experiment

Contour of horizontal density gradients



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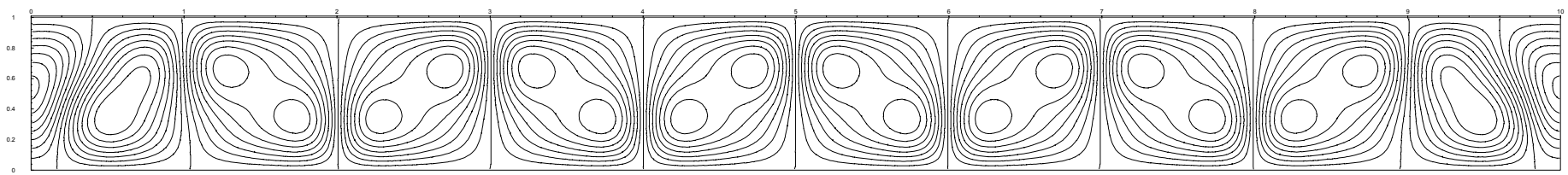
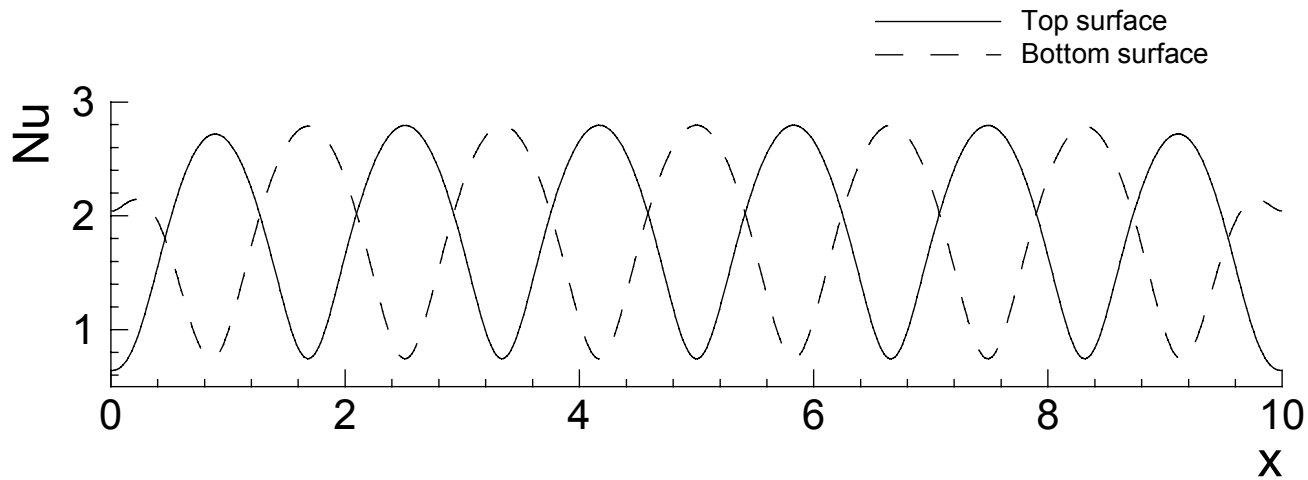


b) Perfect conducting side walls with linear temperature distribution

c) Differential interferogram experiment

Nusselt Number Distribution



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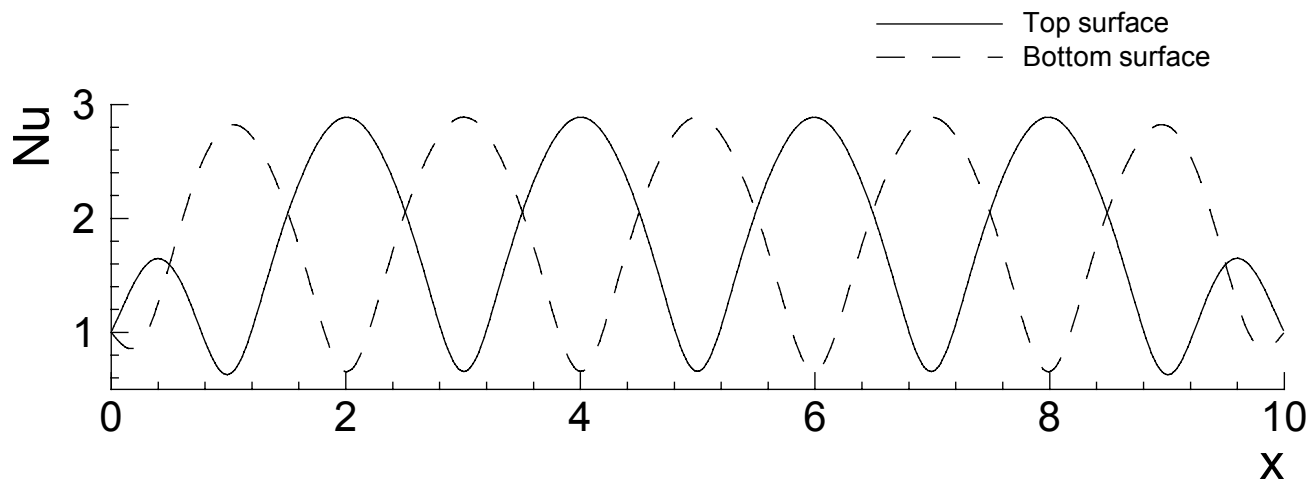

b) Perfect conducting side walls with linear temperature distribution

Future work

Impinging Jet Heat Transfer:

- Finish code development: add pressure boundary condition feature.
- Simulation of real caster.
- Simulation of confined or oblique impinging jet with heat transfer in simpler geometry.
- Choose optimal fast computation methods (e.g. k-ε wall laws) for heat transfer in steel caster.

Future Work

Simulation of Liquid Mold Flux Layer:

- ***Natural convection pattern and heat transfer in slag layer with shear at the bottom surface.***
- ***Account for the influence of changing viscosity with temperature.***
- ***Carry out 3-dimensional simulation.***