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Joint ICTP-IAEA Course on Science and Technology of Supercritical Water Cooled Reactors

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HEAT TRANSFER TO SUPERCRITICAL FLUIDS

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(SC11)
Heat Transfer to Supercritical fluids

Joint ICTP-IAEA Course on Science and Technology of SCWRs, Trieste, Italy, 27 June - 1 July 2011

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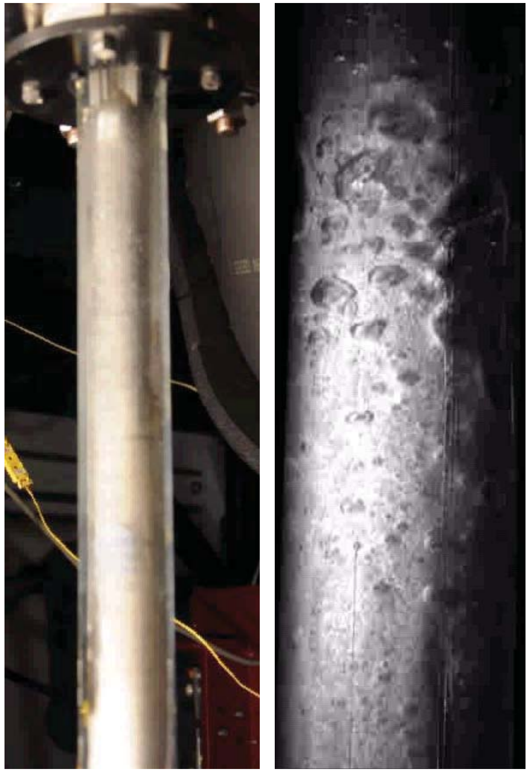
Objectives

- Outline differences in heat transfer under supercritical conditions
- Identify best methods to estimate heat transfer in supercritical fluids
- Understand the mechanisms for Heat transfer improvement/deterioration
- Understand the differences between tubes, bundles and other orientation heat transfer correlations
- Learn where to find Data/Resources for further investigation of SCWR heat transfer

Basic review of heat transfer

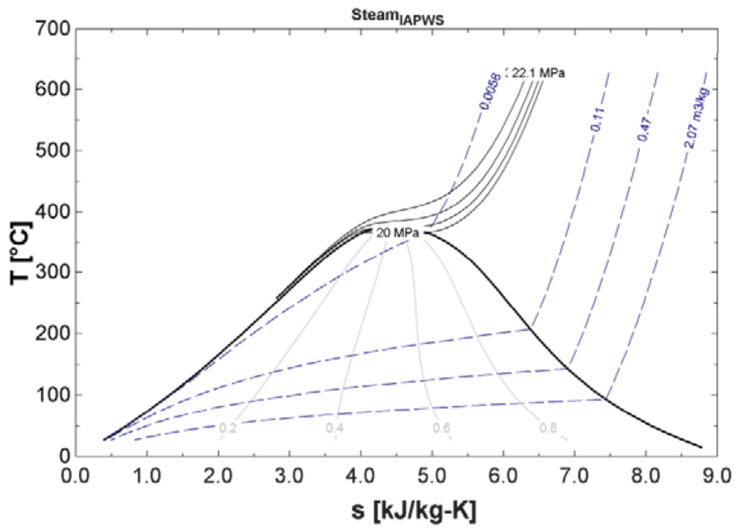
- Introduction to heat transfer for supercritical fluids
  - How does this differ from low pressure water with respect to heat transfer. One major area is CHF- (at pressures above the critical pressure we do not have to worry about boiling phase heat transfer)

At pressures below the critical point normal heat transfer phenomena occurs until a certain critical heat flux is achieved. Above this point rapid boiling occurs and you can get dry-out (CHF).



Movies of Simulated TRIGA fuel pin at 3000W atmospheric pressure, inlet temperature 93 C Flow rate ~0.08 kg/s – boiling phenomena

Movie of pool boiling CHF on wire



Introduction to heat transfer (Review)

- Review of heat transfer for “normal fluids”
- Heat transfer is the exchange of thermal energy from one thermodynamic system to another

  e.g., Nuclear systems – typically concerned with transferring heat from fuel pins to water

- Types of heat transfer:
  - Conduction, convection and thermal radiation

$$q''_x = -k\frac{dT}{dx} \qquad q''_x = h(T_s - T_\infty) \qquad q''_x = \epsilon\sigma(T_s^4 - T_\infty^4) \qquad \sigma = 5.67x10^{-8}[W/m^2K^4]$$

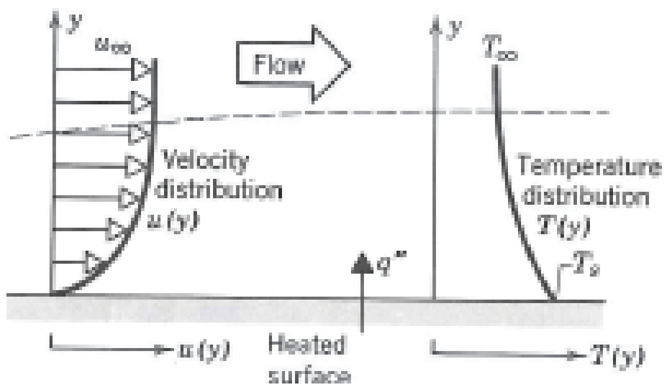
- In order to calculate the fuel pin temperature and the amount of energy we will get from a SCWR need to evaluate the heat transfer.
  - Conduction is straight forward
  - Unless at very high temperatures radiation is component is small
  - Convection coefficient (*h*) is most important.

Estimation of Convection

- Convection is comprised of two mechanisms
  - Random molecular motion (diffusion)
  - Bulk or macroscopic motion of the fluid

Convection between fluid motion and bounding surface when the two are at different temperatures.

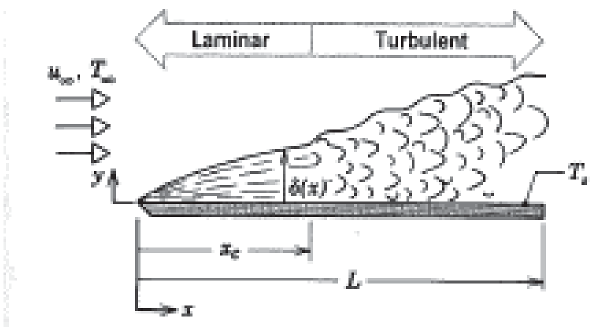
Recall fluid motion over a heated surface: (both hydrodynamic and thermal boundary layers will develop) we need to understand what is happening in the boundary layer.

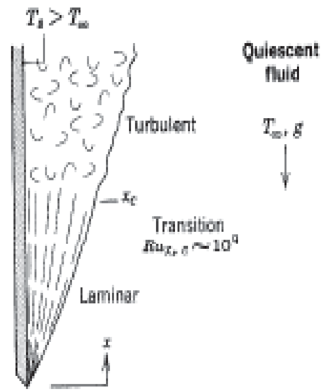


Random molecular motion dominates at y close to zero. Bulk motion is governed by the boundary layer that develops in the x-direction. The heat conducted into this layer is swept down stream and eventually transferred to the bulk.

Laminar –vs- Turbulent

- The range of heat transfer is dependent on the flow regime and the size/frequency of the turbulent eddies
- Laminar flow typically occurs during development or with low fluid velocities and has heat transfer coefficients around 10-1000 [W/m^{2}K]
- Turbulent flows are associated with high velocity and fully developed conditions and can have heat transfer coefficients in the 20-20000 [W/m^{2}K]





The Reynolds number (ratio of inertial forces to viscous forces) can be used to characterize laminar or turbulent flow

$$Re = \frac{\rho V D}{\mu}$$

Laminar convection coefficients

- For constant surface heat flux it is possible to develop an approximate analytical solution for the heat transfer coefficient if we assume constant properties:

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial r} = \frac{\alpha}{r}\frac{\partial}{\partial r}\left(r\frac{\partial T}{\partial r}\right)$$

Energy

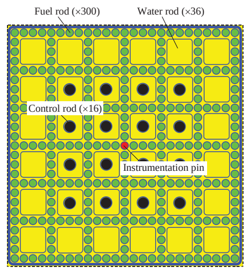
Boundary layer approximations

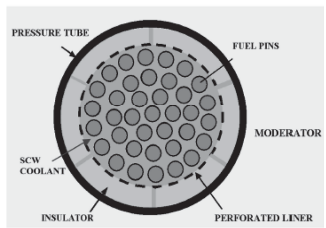
$$v = 0, (\partial u/\partial x) = 0$$
$$(\partial^2 T/\partial x^2) = 0$$

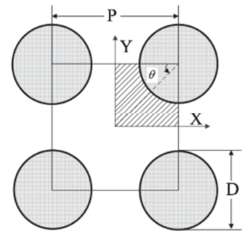
Results in $h = \frac{48}{11}\left(\frac{k}{D}\right)$ or $Nu_D = \frac{hD}{k} = 4.36$

Turbulent convection coefficients

- Under most reactor conditions the Reynolds number is sufficiently high such that turbulent flow occurs (Re >> 5000)
- It is difficult to develop analytical theory for turbulent flows – therefore we require either CFD solutions to the governing equations or can get the general trends by semi - empirical formulations which include the relevant non-dimensional parameters
- A fuel pin bundle flow channel to a first approximation can be considered internal flow.
- Under these conditions the Dittus Boelter “type equations” are widely used for engineering approximations and are in the form:.







$$Nu_D = C Re_D^x Pr^y \qquad Nu = \frac{hD}{k}$$

$$C = 0.023,\ x = \tfrac{4}{5},\ y = 0.4 \qquad Pr = \frac{c_p \mu}{k}$$

$$0.7 \le Pr \le 160$$

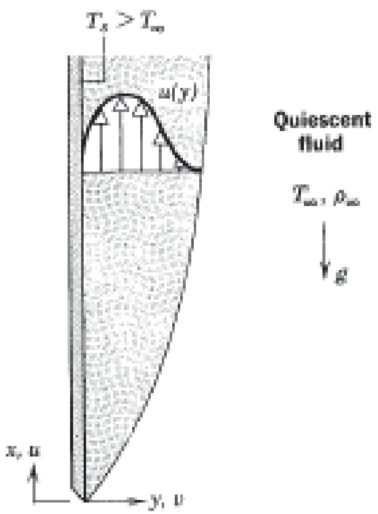
$$Re_D \ge 8{,}000$$

$$\tfrac{L}{D} \ge 10$$

These equations were developed from air water data at room temperature but are typically extended out side this range by using different C,x,y values adding some corrections for property variations. Most reactor systems have specific correlations for their geometries typically of this form.

Forced –vs- Free convection

- Forced convection – caused by external means (e.g., pump, fan)
- Free (natural) convection- flow is induced by buoyancy forces which arise from density differences caused by temperature variations



We can have:

Pure forced convection
- Gases – h=25-250 W/m^{2}K
- Liquids – h= 50- 20,000 W/m^{2}K

Pure Free convection
- Gases – h = 2-25 W/m^{2}K
- Liquids – h = 50- 1000 W/m^{2}K

Mixed convection
- Typically between forced and free

Boiling/condensation
- h = 2500- 100,000 W/m^{2}K

Supercritical fluids forced convection
- h= 1000-50000 W/m^{2}K

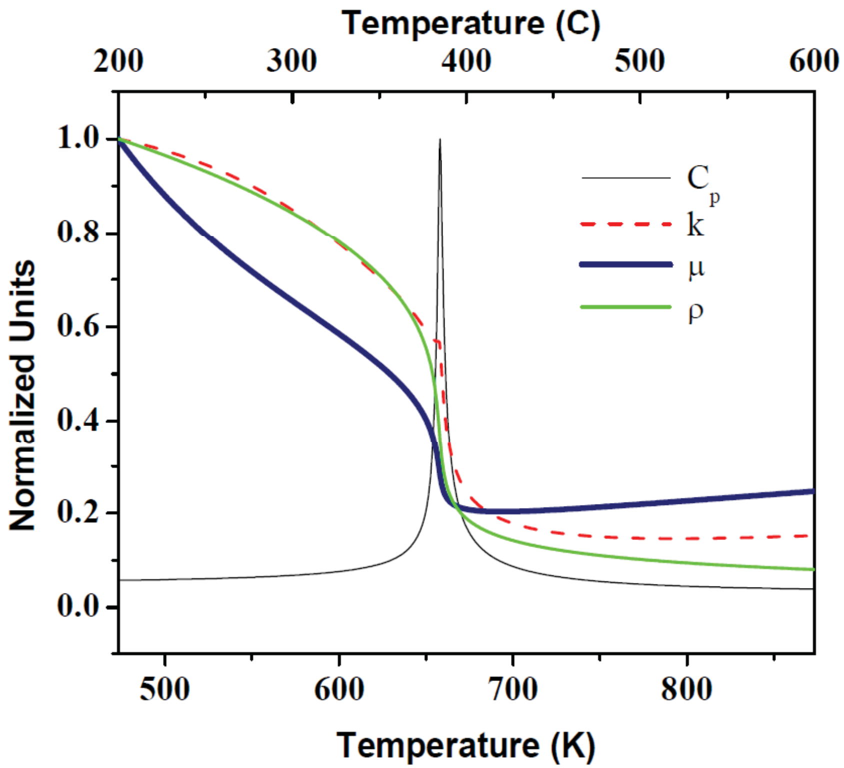
$$Gr = \frac{g\beta(T_w - T_b)D_h^3}{\nu^2}$$

$$\beta = -\frac{1}{\rho}\left(\frac{\partial\rho}{\partial T}\right)_p$$

Ratio of Buoyancy to viscous forces

$$Nu = C\left(Gr^a Pr^b\right)$$

What are the issues with SCF's

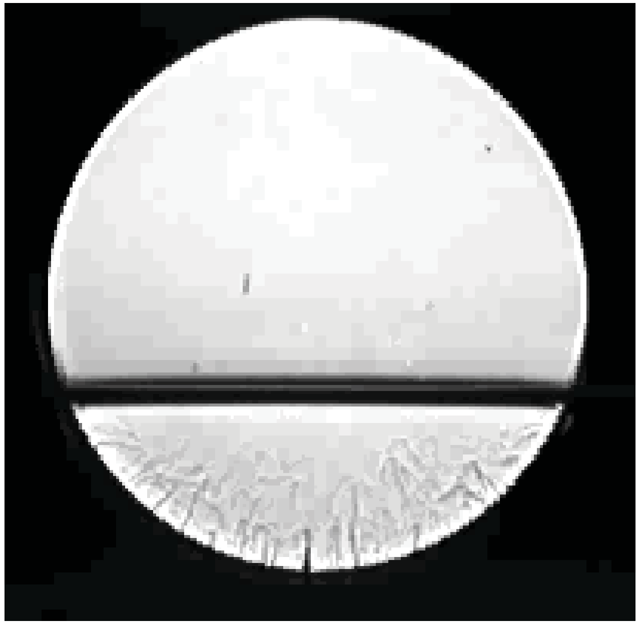


Property variations!

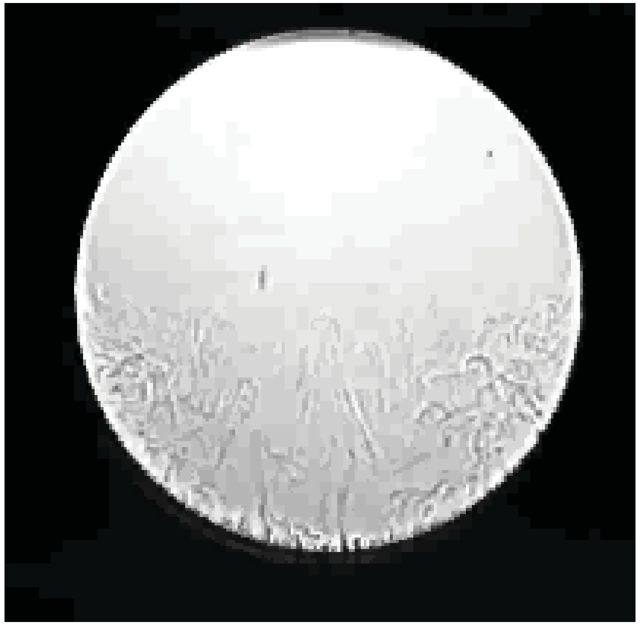
- Specific heat is theoretically infinite at the critical point and the location of the peak defines the pseudo critical point.
- Density changes by a factor of 10x
- Thermal conductivity changes by a factor of 6x
- Viscosity changes by a factor of 4x
- The fact that the properties change so drastically over a small temperature causes unique phenomena in the flow and in correlating heat transfer data.
- Requires modifications to the existing correlations

Thermophysical property variation of water as a function of temperature at 25 Mpa calculated with Steam_IAPWS formulae (Kestin et al., 1984, Saul et al., 1987)
NIST REFPROP - program
Lemmon, E.W., Huber, M.L., McLinden, M.O. NIST Standard Reference Database 23: Reference Fluid Thermodynamic and Transport Properties-REFPROP, Version 9.0, National Institute of Standards and Technology, Standard Reference Data Program, Gaithersburg, 2010.
EES (Engineering Equation solver) – This is convenient since it also allows you to solve non-linear sets of equations

A look at what happens going through the critical point P=220.6 Bar



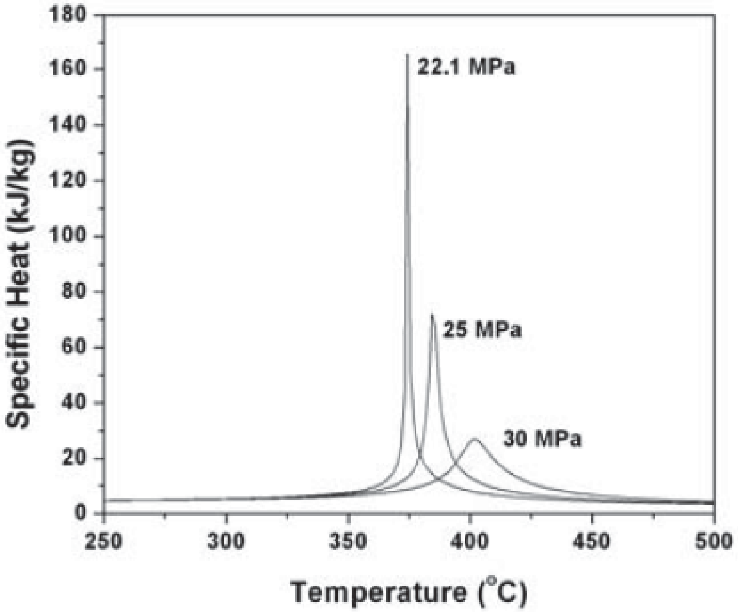
Heating up through critical point
TC- 373.9C

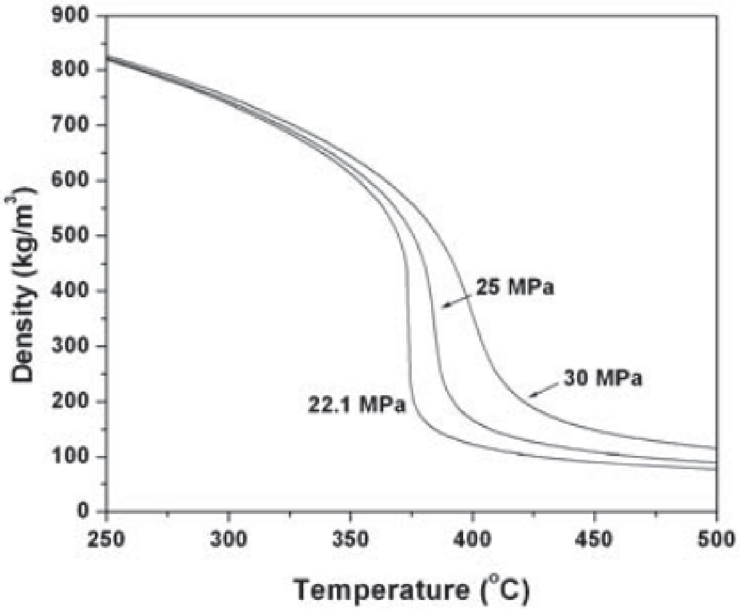


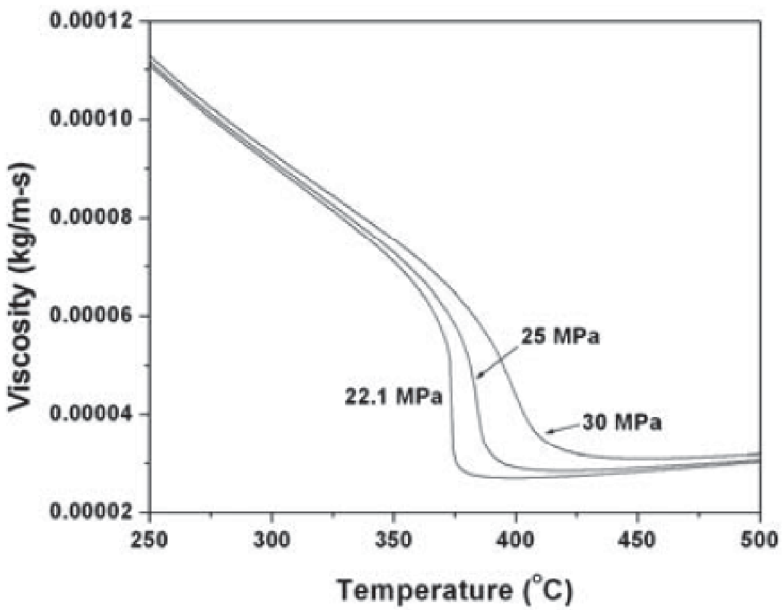
Cooling down from above critical point
TC=373.9C

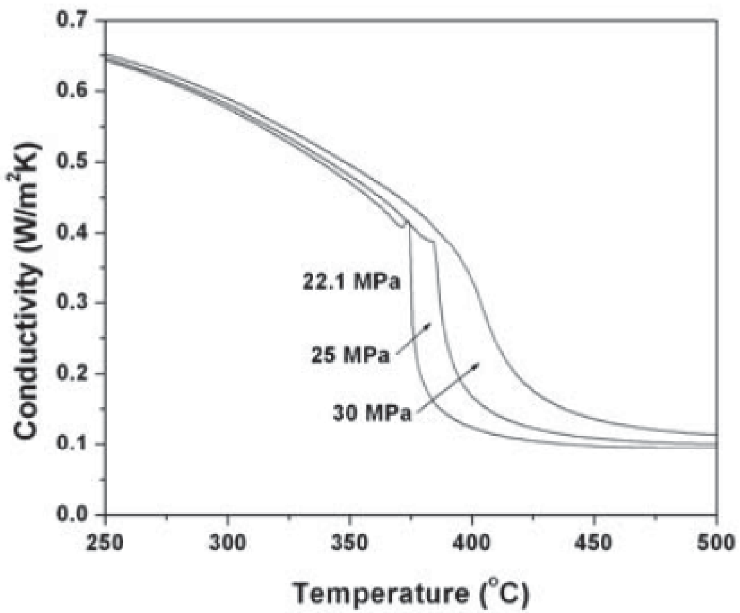
http://www.science.uva.nl/research/mgrd/video-cp.htm
(Universiteit van Amsterdam

Properties of water as a function of pressure



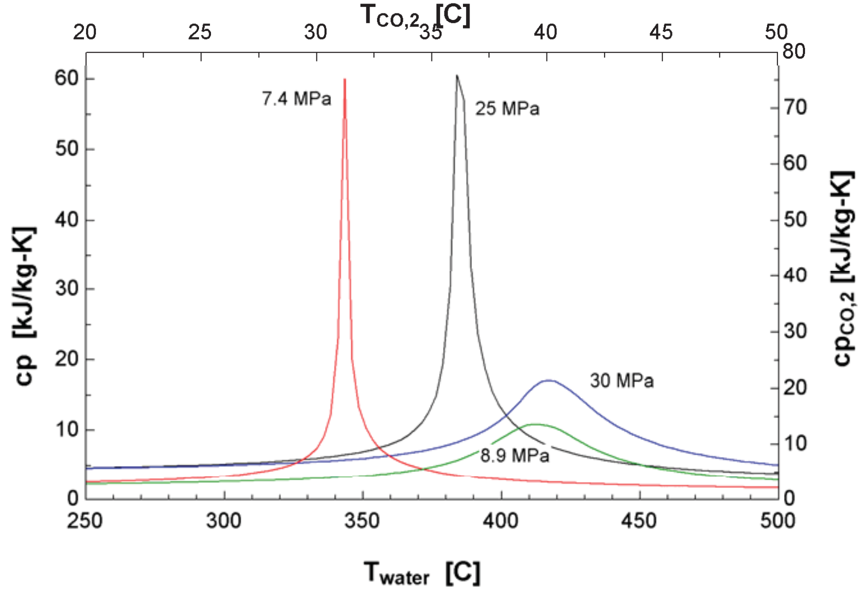




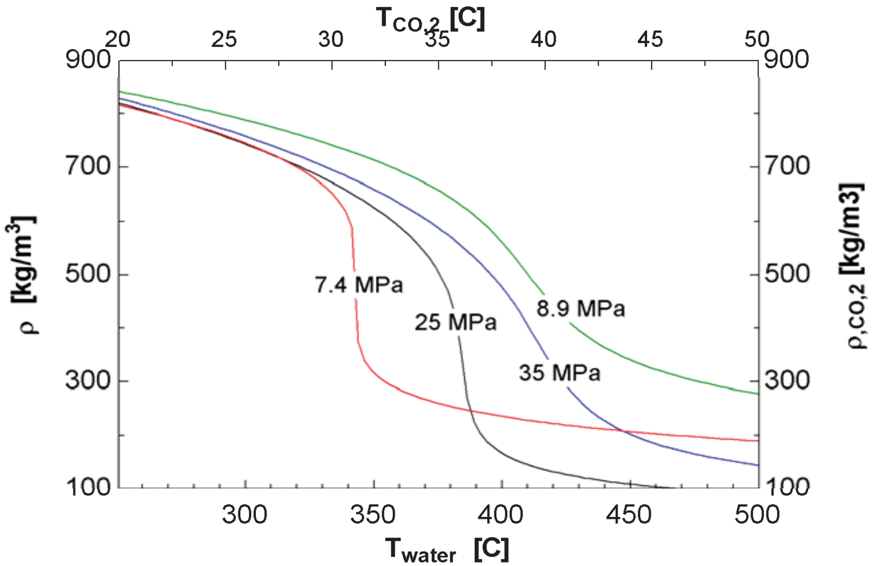


Quick way to calculate properties

```
"Calculation of properties using EES code"

P=35
PC=P_crit(Steam_IAPWS)
TC=T_crit(Steam_IAPWS)

cp=Cp(Steam_IAPWS,T=T,P=P)
rho=Density(Steam_IAPWS,T=T,P=P)
h=Enthalpy(Steam_IAPWS,T=T,P=P)
mu=Viscosity(Steam_IAPWS,T=T,P=P)
k=Conductivity(Steam_IAPWS,T=T,P=P)

cp_J=convert(kj,j)*cp
Prandlt=cp_J*mu/k
```



We can get the properties of any fluid in this manner and could compare the changes in properties of water with surrogate fluids CO2, helium, air, Refrigerants, etc.



We will focus on water for this lecture but data from other fluids with similar fluid property variations are relevant.

What does the data tell us

- Pioro and Duffy, Heat transfer and Hydraulic Resistance, 2007 has an excellent summary of data for a wide range of Supercritical conditions.
- There have been 100's of experiments measuring heat transfer with water and other fluids above the critical pressure.
- Most of the data have been in circular tubes, with a limited set in annuli and even fewer in bundle geometries.
- In general what was found is that the variation in properties affect the convection heat transfer - three different modes of heat transfer were observed
  - normal heat transfer – occurs at high mass flux low heat flux
  - Improvement in the HTC near the pseudo critical point.
  - Deteriorated heat transfer with low mass flux and high heat flux under some orientations
- The following tables have the conditions of several of the tests that have been conducted in the past – we will focus on a few for discussion purposes.

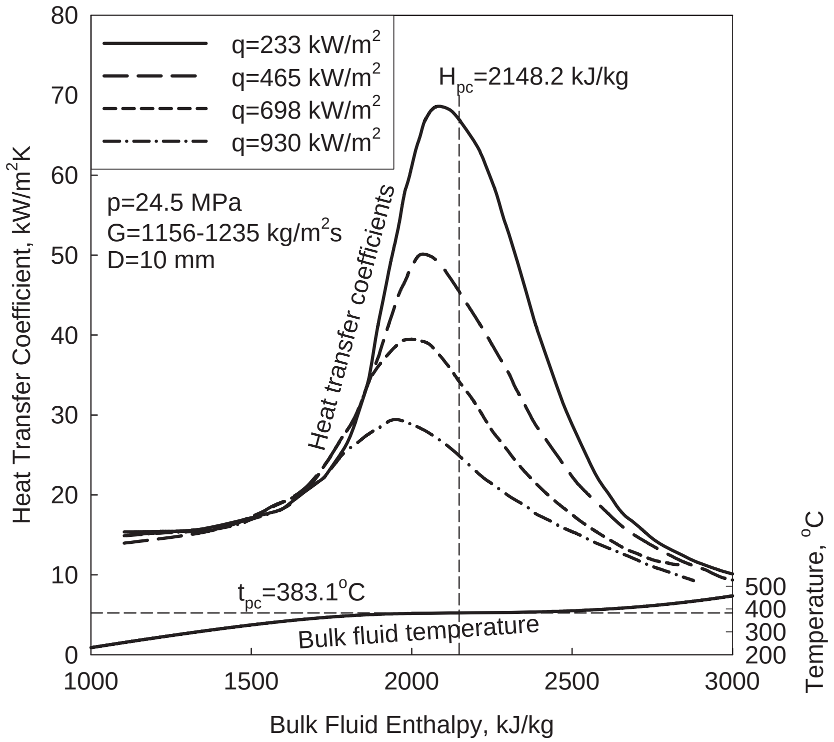
Reference	P,MPa	T, ºC	q [MW/m²]	G[kg/m²s]	Description
Randall 1956	27.6–55.2	t_b=204–538; t_w=204–760	0.31–9.44	2034–5425	Hastelloy C vertical tube (*D*=1.27; 1.57; 1.9 mm, *L*=203.2 mm)
Armand et al. 1959	23–26.3	t_b=300–380	0.17–0.35	450–650	SS and nickel tubes (*D*=6; 8 mm, *L*=250; 350 mm), upward flow
Doroshchuk et al. 1959	24.3	t_b=100–250	3.06–3.9	3535–8760	Silver tube (*D*=3 mm, *L*=246 mm), downward flow
Swenson et al. 1965	23–41	t_b=75–576; t_w=93–649	0.2–1.8	542–2150	SS tube (*D*=9.42 mm, *L*=1.83 m), upward flow (selected data are shown in Figures 5.2 and 5.3)
Smolin and Polyakov 1965	25.4; 27.4; 30.4	t_b=250–440	0.7–1.75	1500–3000	SS tube (*D*=10; 8 mm, *L*=2.6 m), upward flow
Vikhrev et al. 1967	24.5; 26.5	H_b=230–2750	0.23–1.25	485–1900	SS tube (*D*=7.85; 20.4 mm, *L*=1.515; 6 m) (selected data are shown in Figure 5.4)
Bourke and 1967	23.0–25.4	t_b=310–380	1.2–2.2	1207; 2712	Tube (*D*=4.06 mm, *L*=1.2 m)
Styrikovich et al. 1967	24	H_b=1260–2500	0.35–0.87	700	Tube (*D*=22 mm, *L* was not provided in the original paper) (selected data are shown in Figure 5.6)
Krasyakova et al. 1967	23	H_{in}=837–2721	0.23–0.7	300–1500	Vertical and horizontal tubes (*D*=20 mm, *L*=2.8 m), upward and horizontal flows
Shitsman 1968	10–35	t_b=100–250	0.27–0.7	400	Vertical and horizontal SS tubes (*D*/*L*=3/0.7; 8/0.8; 8/3.2; 16/1.6 mm/m), upward, downward and horizontal flows
Krasyakova et al. 1968	15; 18.8; 23	H_{in}=840–1890	0.23–0.7	300–2000	Vertical and horizontal SS tube (*D*=20 mm, *L*=2.2 m), upward, downward and horizontal flows
Alferov et al. 1969	14.7–29.4	t_b=160–365	0.17–0.6	250–1000	SS tubes (*D*/*L*=14/1.4; 20/3.7 mm/m)
Kamenetsky and Shitsman 1970	24.5	H_b=80–2300	0.19–1.33	50–1750	Vertical and horizontal SS tube (*D*=22 mm, *L*=3 m), non-uniform circumferential heat flux, upward and horizontal flows
Ackerman 1970	22.8–41.3	t_b=77–482	0.126–1.73	136–2170	Smooth (*D*=9.4; 11.9 and 24.4 mm, *L*=1.83 m; *D*=18.5 mm, *L*=2.74 m) and ribbed (*D*=18 mm (from rib valley to rib valley), *L*=1.83 m, six helical ribs, pitch 21.8 mm) tubes
Ornatsky et al. 1970	22.6; 25.5; 29.4	H_{in}=420–1400	0.28–1.2	450–3000	Five SS parallel tubes (*D*=3 mm, *L*=0.75 m), upward stable and pulsating flows
Barulin et al. 1971	22.5–26.5	t_b=50–500; t_w=60–750	0.2–6.5	480–5000	Vertical and horizontal tubes (*D*=3; 8; 20 mm, *L*/*D*<300), upward, downward and horizontal flows
Belyakov et al. 1971	24.5	H_b=420–3140	0.23–1.4	300–3000	Vertical and horizontal SS tube (*D*=20 mm, *L*=4–7.5 m), upward and horizontal flows
Ornatskii et al. 1971	22.6, 25.5, 29.7	H_b=100–3000	0.4–1.8	500–3000	SS tube (*D*=3 mm, *L*=0.75 m), upward and downward flows
Yamagata et al. 1972	22.6–29.4	t_b=230–540	0.12–0.93	310–1830	Vertical and horizontal SS tubes (*D*/*L*=7.5/1.5; 10/2 mm/m), upward, downward and horizontal flows (selected data are shown in Figure 5.7)

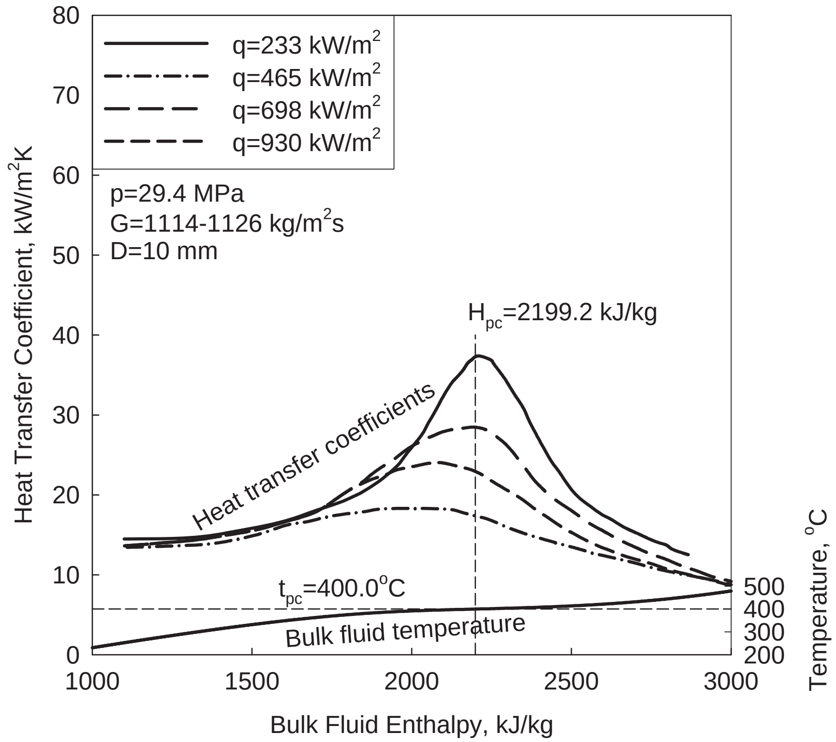
Edited from Pioro and Duffy 2007

Reference	P,MPa	T, °C	q [MW/m²]	G[kg/m²s]	Description
Glushchenko et al. 1972	22.6; 25.5; 29.5	H_b=85–2400	1.15–3	500–3000	Tubes (*D*=3; 4; 6; 8 mm, *L*=0.75–1 m), upward flow; *D*=3 mm, downward flow
Malkina et al. 1972	24.5–31.4	t_b=20–80	0.47–2.3	*u*=7–10 m/s	SS tubes (*D*=2; 3 mm, *L*=0.15 m)
Chakrygin et al. 1974	26.5	t_{in}=220	*q* was not provided	445–1270	SS tube (*D*=10 mm, *L*=0.6 m), upward and downward flows
Lee and Haller 1974	24.1	t_b=260–383	0.25–1.57	542–2441	SS tubes (*D*=38.1; 37.7 mm, *L*=4.57 m), tube with ribs
Alferov et al. 1975	26.5	t_b=80–250	0.48	447	Tube (*D*=20 mm, *L*=3.7 m), upward and downward flows
Kamenetskii 1975	23.5; 24.5	H_{in}=100–2300	1.2	50–1700	Steel tubes (*D*=21; 22 mm, *L*=3 m), non-uniform circumferential heat flux
Alekseev et al. 1976	24.5	t_{in}=100–350	0.1–0.9	380, 490, 650, 820	SS tube (*D*=10.4 mm, *L*=0.5; 0.7 m), upward flow
Ishigai et al. 1976	24.5; 29.5; 39.2	H_b=220–800	0.14–1.4	500; 1000; 1500	Vertical and horizontal SS polished tubes (*D*=3.92 mm, *L*=0.63 m – vertical; *D*=4.44 mm, *L*=0.87 m – horizontal)
Harrison and Watson 1976a,b	24.5	t_b=50–350	1.3, 2.3	940, 1560	Vertical and horizontal SS tubes (*D*=1.64; 3.1 mm, *L*=0.4, 0.12 m)
Treshchev and Sukhov 1977	23; 25	H_{in}=1331	0.69–1.16	740–770	Tubes (*L*=0.5–1 m), stable and pulsating upward flows
Krasyakova et al. 1977	24.5	t_b=90–340	0.11–1.4	90–2000	Tube (*D*=20 mm, D_{ext}=28 mm, *L*=3.5 m), downward flow (selected data are shown in Figure 5.5)
Smirnov and Krasnov 1978–1980	25; 28; 30	t_w=250–700	0.25–1	500–1200	SS tube (*D*=4.08 mm, *L*=1.09 m), upward and downward flow
Kamenetskii 1980	24.5	H_b=100–2200	0.37–1.3	300–1700	Vertical and horizontal SS tubes with and without flow spoiler (*D*=22 mm, *L*=3 m)
Selivanov and Smirnov 1984	26	t_{in}=50–450	0.13–0.65	200–10 000	SS tube (*D*=10 mm, D_{ext}=14 mm, *L*=1 m)
Kirillov et al. 1986	25	t_{in}=385	0.4; 0.6	1000	SS tube (*D*=10 mm, D_{ext}=14 mm, *L*=1 m)
Razumovskiy et al. 1990	23.5	H_{in}=1400; 1600; 1800	0.657–3.385	2190	Tube (*D*=6.28 mm, *L*=1440 mm), downward flow
Chen 2004	24	H_{in}=1350; 1600	300	400	SS vertical and inclined tubes (smooth with uniform and non-uniform radial heating and ribbed)
Pis'mennyy et al. 2005	23.5	t_{in}=20–380	Up to 0.515	250; 500	Vertical SS tubes (*D*=6.28 mm, L_h=600; 360 mm; *D*=9.50 mm, L_h=600; 400)
Kirillov et al. 2005	24–25	t_{in}=300–380	0.09–1.050	200–1500	SS tube (*D*=10 mm, *L*=1; 4 m)
Licht, et al 2008	25	t_{in}=200-500	0.05-1	200-1800	Inconel tube 4 cm diameter, circular annular and square annular L=1m

Edited from Pioro and Duffy 2007

Normal and improved heat transfer - Yamagata data

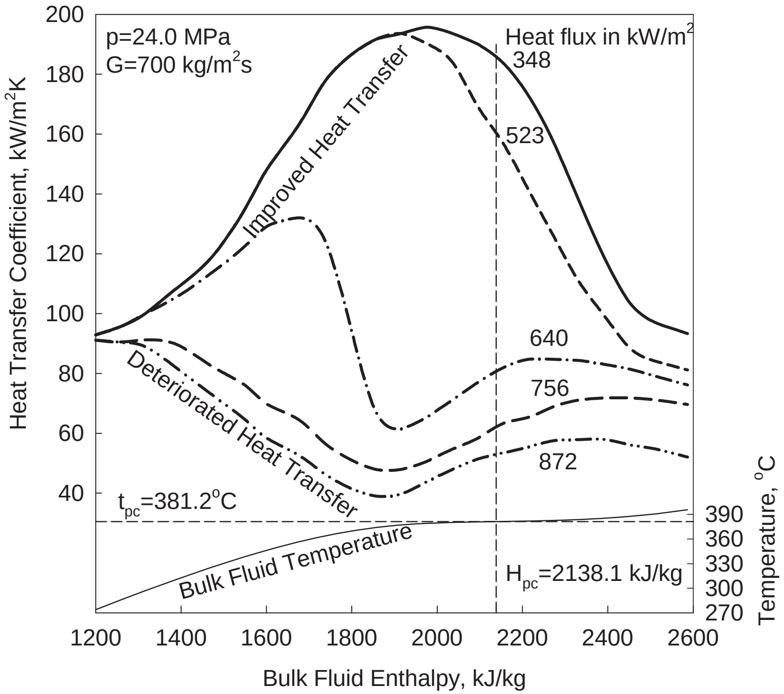




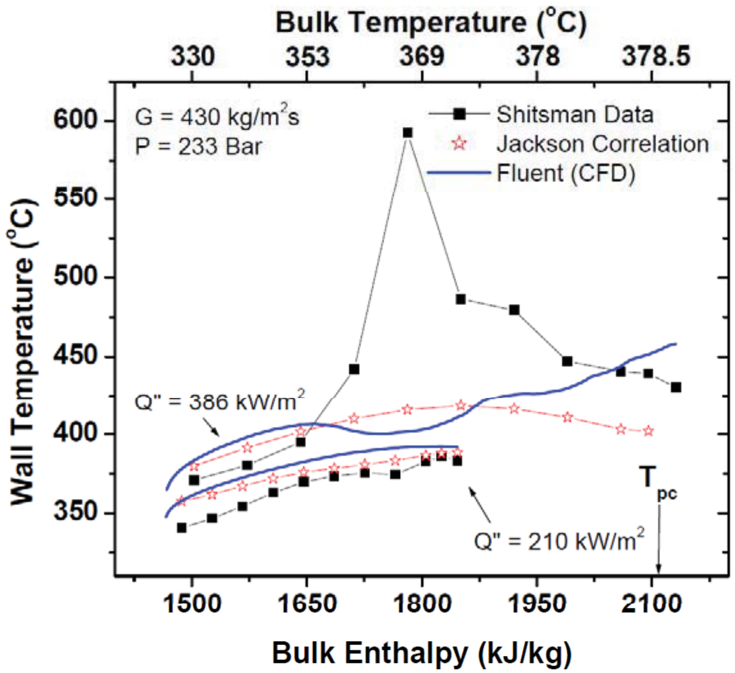
Yamagata et al 1972 a)24.5MPa, b)29.4 MPa

- Note drastic improvement in HTC near the critical point
  - magnitude of peak increases with lower heat flux
  - magnitude of peak decrease with increased pressure

Observation of deterioration



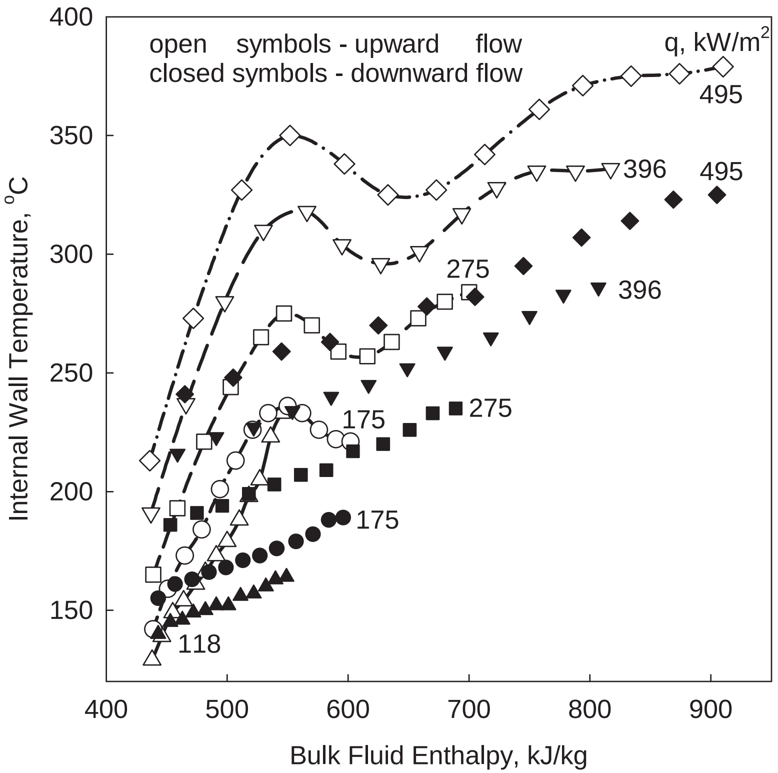
Styrikovich et al 1965



Shitsman et al. 1968

- Note as the heat flux is increased we start to see deterioration in the HTC
  - Deterioration starts to occur when wall temperature increases above pseudo critical temperature
  - It was observed as an increase in wall temperature along the tube
  - Mainly seen to occur in upward vertical flow and to a much lower extent in horizontal

Effect of orientation



Pis'mennyy et al. 2005

P=23.5MPa, G=248 kg/m^{2}s

The effect of orientation is seen in this data. Low mass flux and increasing heat flux in vertical up flow and vertical down flow

- Note the increase in the wall temperature in upward flow
- Also note the somewhat periodic change in wall temperature as you move along the tube
- The increase in wall temperature is not observed in downward flow

These modes of heat transfer

Normal

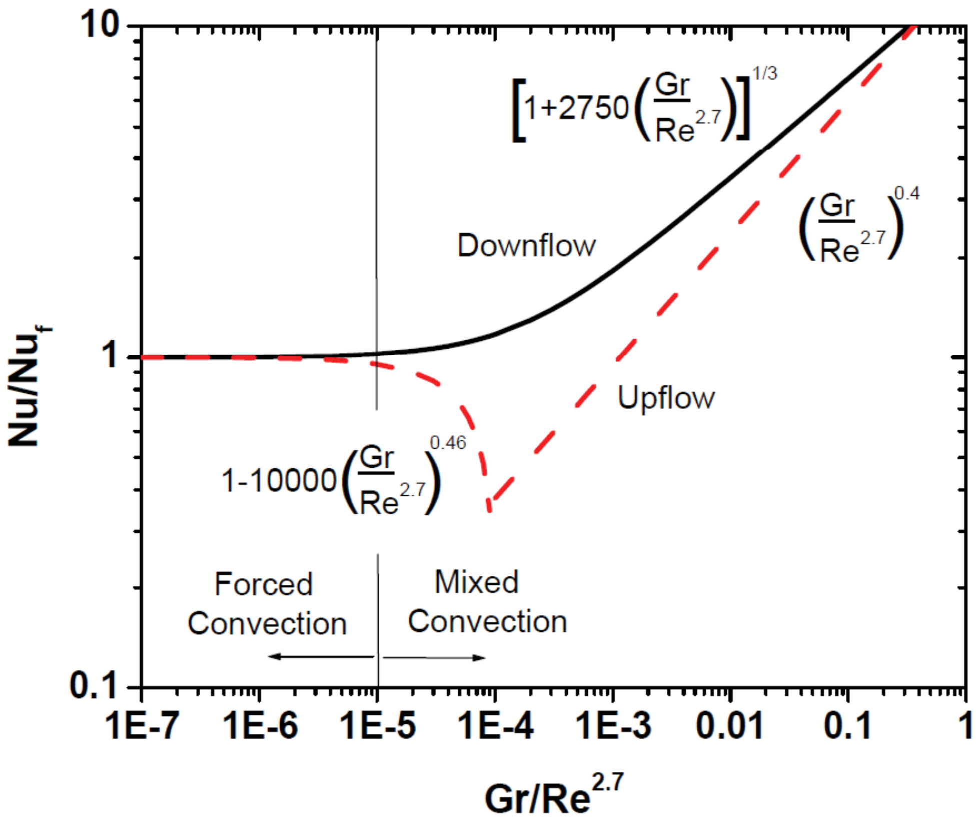
Improvement

Deterioration

have been seen actually been seen in many fluids where the properties change continuously but drastically. The larger the change the larger the effect

To correlate the data we have to categorize it based on the flow conditions (forced, mixed convection – i.e. buoyancy induced, acceleration affected)

As noted several studies showed that the deterioration typically occurs when the mass flux is low and the heat flux is high q"/G>0.4



This effect was attributed to a change in the turbulent shear stress due to the influence of buoyancy forces

$$\overline{Gr}_b / \mathrm{Re}_b^{2.7} < 10^{-5}$$

$$\frac{Nu_b}{Nu_{bo}} = \left[1 \pm C_B Bo_b^* \left[\frac{Nu_b}{Nu_{bo}}\right]^{-2}\right]^{0.46}$$

$$Bo_b^* = Gr_b^* / (\mathrm{Re}_b^m \, \mathrm{Pr}^n)$$

$$C_b \sim 10^5$$

Forced convection

Under forced convection conditions we have normal heat transfer with some improvement due to enhanced properties

$$Nu_D = CRe_D^x Pr^y$$

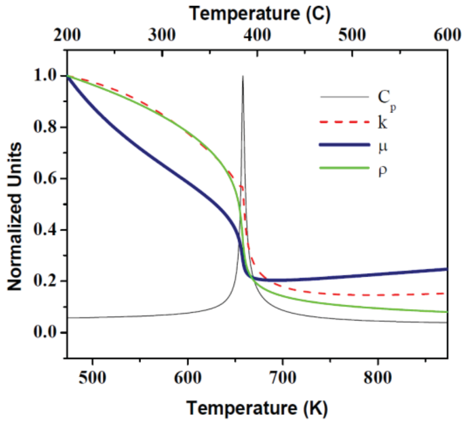
$C = 0.023, x = \frac{4}{5}, y = 0.4$
$0.7 \leq Pr \leq 160$
$Re_D \geq 8,000$
$\frac{L}{D} \geq 10$

$$Nu = \frac{hD}{k}$$

$$Pr = \frac{c_p \mu}{k}$$

$$Re = \frac{\rho V D}{\mu}$$

$$h \propto (\rho v)^{0.8} D^{-0.2} \mu^{-0.4} c_p^{0.4} k^{0.6}$$



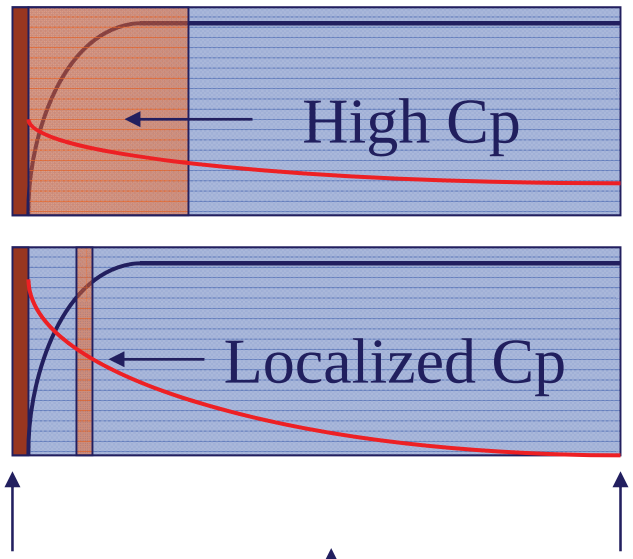
This means that we can correlate the data similar to what we did for normal fluids if we include factors that account for the property changes

$$\left(\frac{\rho_a}{\rho_b}\right)^c \quad \left(\frac{c_{p,a}}{c_{p,b}}\right)^d \quad \left(\frac{k_a}{k_b}\right)^e \quad \left(\frac{\mu_a}{\mu_b}\right)^f$$

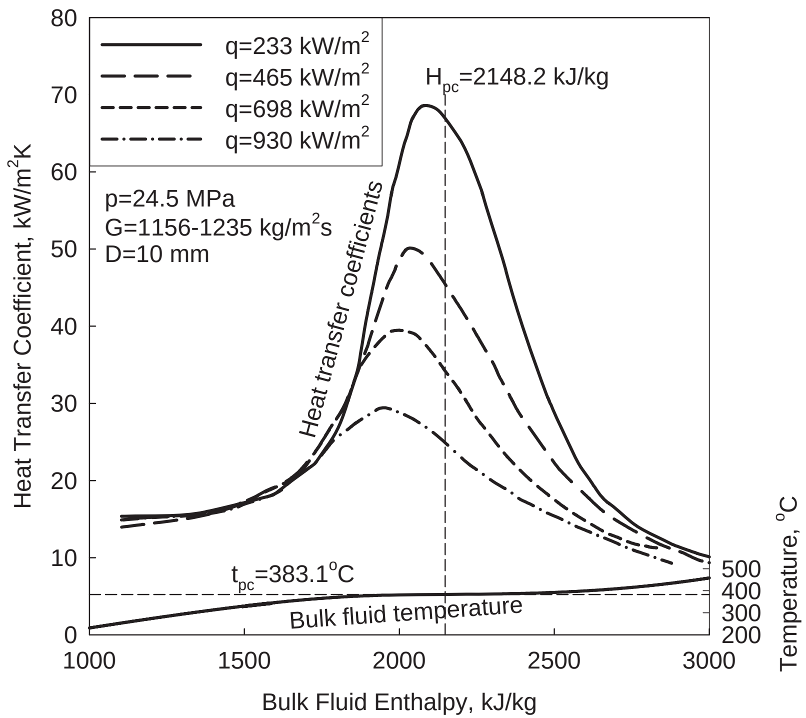
Improvement under force convection, why does it decrease with increasing heat flux

$$h = \frac{q''}{(T_w - T_b)}$$

Improvement



High mass flux low heat flux independent of orientation



Force convection

Several different investigators used this procedure and developed correlations of the form below: Where a,b could be the properties evaluated at the wall, bulk or pseudo critical point.

$$\mathbf{Nu}_{\mathbf{t,\,x}} = C_1\ \mathbf{Re}_{\mathbf{t,\,x}}^{\mathbf{m_1}}\ \mathbf{Pr}_{\mathbf{t,\,x}}^{\mathbf{m_2}} \left(\frac{\rho_t}{\rho_t}\right)_x^{m_3} \left(\frac{\mu_t}{\mu_t}\right)_x^{m_4} \left(\frac{k_t}{k_t}\right)_x^{m_5} \left(\frac{\bar{c}_p}{c_{p,\,t}}\right)_x^{m_6} \left(1 + C_2\,\frac{D_{hy}}{L_h}\right)^{m_7}$$

Last term accounts for entrance effects

Pioro and Duffy (2007) summarized several of the major correlations for forced convection as follows

Reference	Flow geometry	Characteristic parameters in Nu, Re and Pr		m_1	m_2	m_3	m_4	m_5	m_6	m_7
		t, ºC	Length							
McAdams et al. 1950	Annulus		D_{hy}	0.8	0.33	0	0	0	0	1
Bringer, Smith 1957	Tube	t_b, t_{pc} or t_w	D	0.77	0.55 t_w	0	0	0	0	0
Shitsman 1959, 1974	Tube	t_b	D	0.8	0.8 t_b or t_w	0	0	0	0	0
Krasnoshchekov, Protopopov 1959	Tube	t_b	D	~0.8	~0.33	0	0.11	–0.33	0.35	0
Swenson et al. 1965	Tube	t_w	D	0.923	0.613 based on	–0.231	0.231	0	0	0
Kondrat'ev 1969	Tube, annulus	t_b	D_{hy}	0.8	0	0	0	0	0	0
Ornatsky et al. 1970	Tube	t_b	D	0.8	0.8 t_b or t_w	–0.3	0	0	0	0
Ornatsky et al. 1972	Annulus	t_b	D_{hy}	0.8	0.4	0	0	0	0	0
Yamagata et al. 1972	Tube	t_b	D	0.85	0.8 and	0	0	0	0 or n_1	0
Dyadyakin, Popov 1977	Bundle	t_b	D_{hy}	0.8	0.7 based on	–0.45 and	0.2	0	0	1
Kirillov et al. 1990	Tube	t_b	D	~0.8	~0.33 or 0.4	$–n_1$	0	0	n_2	0
Gorban' et al. 1990	Tube	t_b	D	0.9	–0.12	0	0	0	0	0

$$\mathbf{Nu}_{\mathbf{t,\,x}} = C_1\ \mathbf{Re}_{\mathbf{t,\,x}}^{\mathbf{m_1}}\ \mathbf{Pr}_{\mathbf{t,\,x}}^{\mathbf{m_2}} \left(\frac{\rho_t}{\rho_t}\right)_x^{m_3} \left(\frac{\mu_t}{\mu_t}\right)_x^{m_4} \left(\frac{k_t}{k_t}\right)_x^{m_5} \left(\frac{\bar{c}_p}{c_{p,\,t}}\right)_x^{m_6} \left(1 + C_2\,\frac{D_{hy}}{L_h}\right)^{m_7}$$

Recommended correlations

A modified Kranoshechekov et al. correlation proposed by Derek Jackson seems to correlate to the broadest existing data sets the best for single tube and annulus.

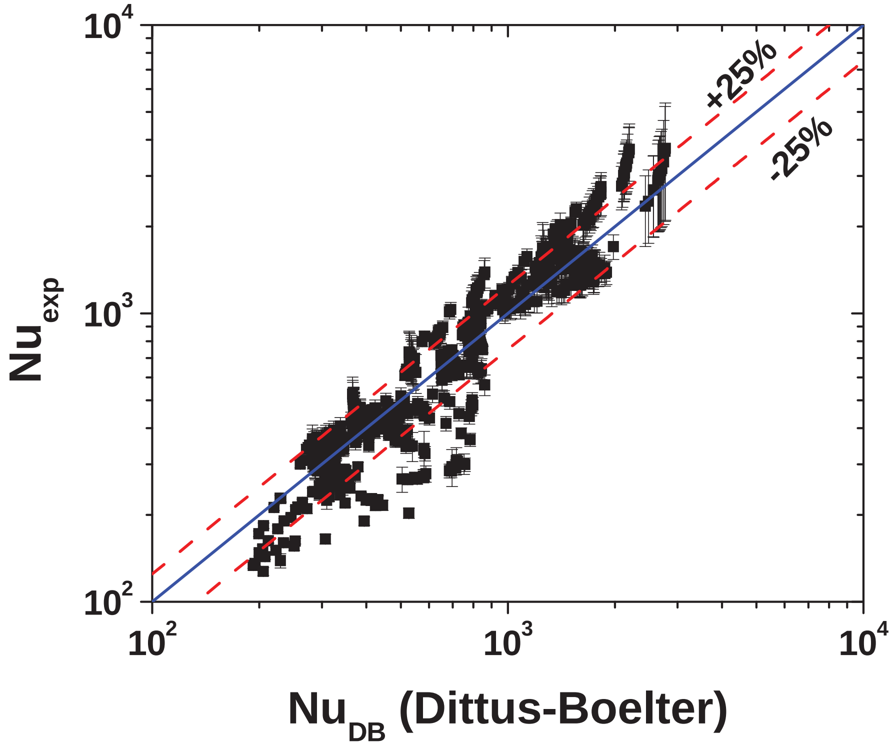
$$Nu_b = 0.0183 Re_b^{0.82} Pr_b^{0.5} \left(\frac{\rho_w}{\rho_b}\right)^{0.3} \left(\frac{\bar{c}_p}{c_{p,b}}\right)^n$$

$$n = 0.4 \qquad \text{for } T_b < T_w < T_{pc} \text{ and for } 1.2T_{pc} < T_b < T_w$$

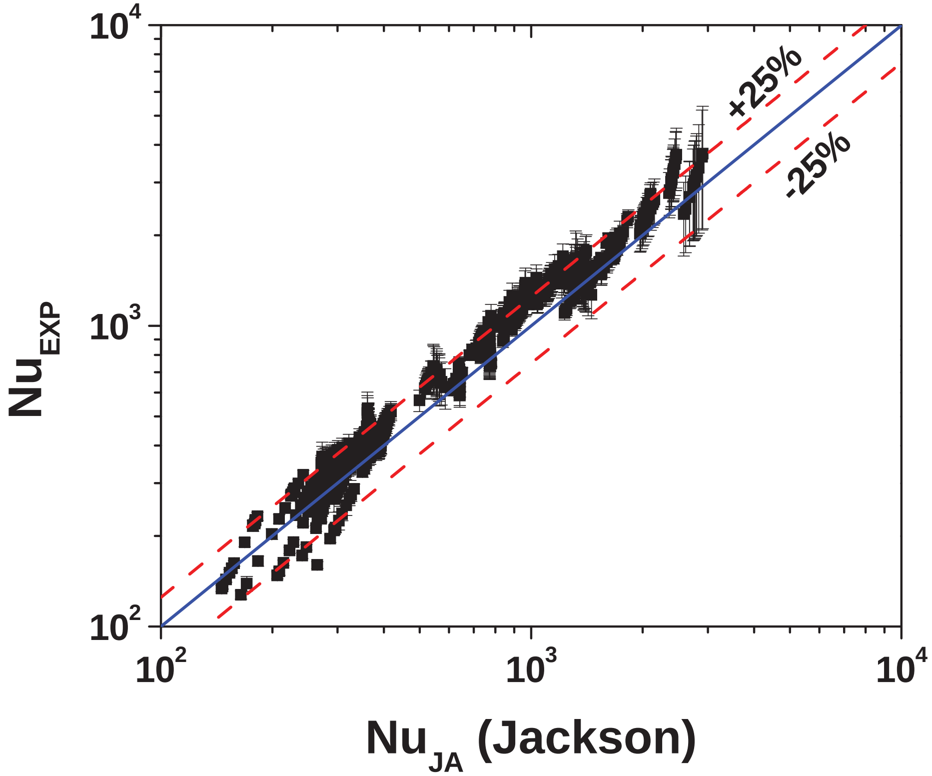
$$n = 0.4 + 0.2\left(\frac{T_w}{T_{pc}} - 1\right) \qquad \text{for } T_b < T_w < T_{pc}$$

$$n = 0.4 + 0.2\left(\frac{T_w}{T_{pc}} - 1\right)\left[1 - 5\left(\frac{T_b}{T_{pc}} - 1\right)\right] \qquad \text{for } T_{pc} < T_b < 1.2T_{pc}$$

Comparison of forced convection correlation with data



$$Nu_{DB,f} = 0.023 \mathrm{Re}_f^{0.8} \mathrm{Pr}_f^{0.4}$$

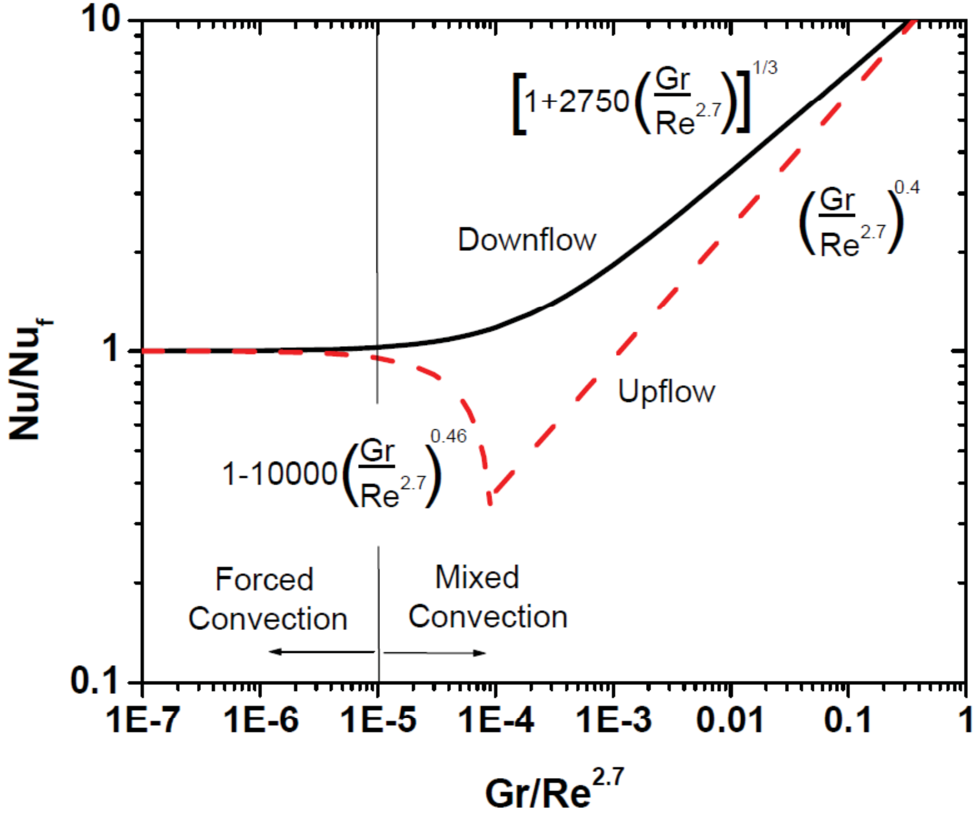


$$Nu_{JA,b} = 0.0183 \mathrm{Re}_b^{0.82} \mathrm{Pr}_b^{0.5} \left(\frac{\rho_w}{\rho_b}\right)^{0.3} \left(\frac{\bar{c}_p}{c_{pb}}\right)^n$$

$$n = f\left(T_b, T_w, T_{pc}\right)$$

Data from Licht et.al 2008

Mixed convection – effect of bouyancy



$$\overline{Gr_b} / \mathrm{Re}_b^{2.7} < 10^{-5}$$

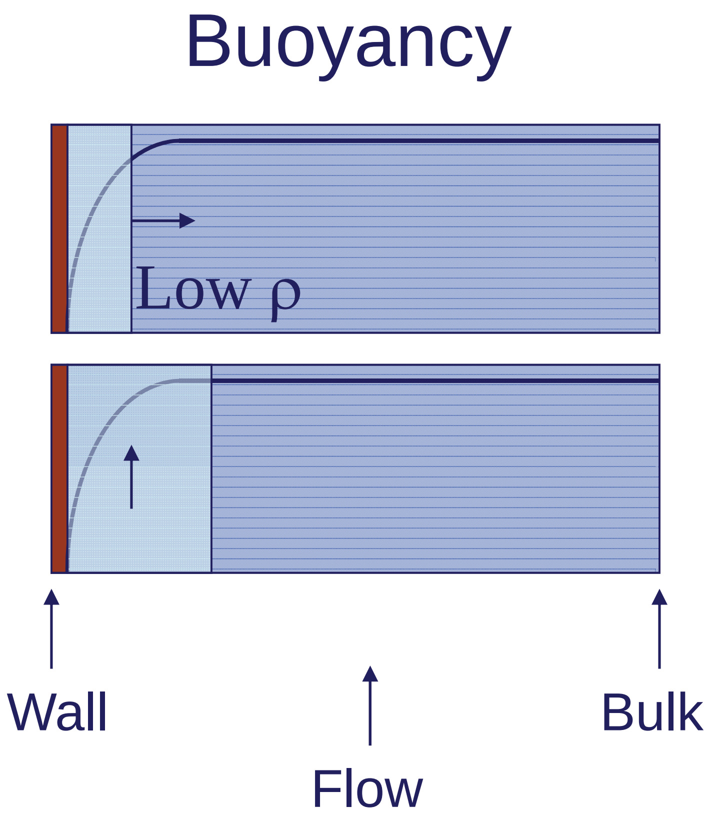
$$\frac{Nu_b}{Nu_{bo}} = \left[1 \pm C_B Bo_b^* \left[\frac{Nu_b}{Nu_{bo}}\right]^{-2}\right]^{0.46}$$

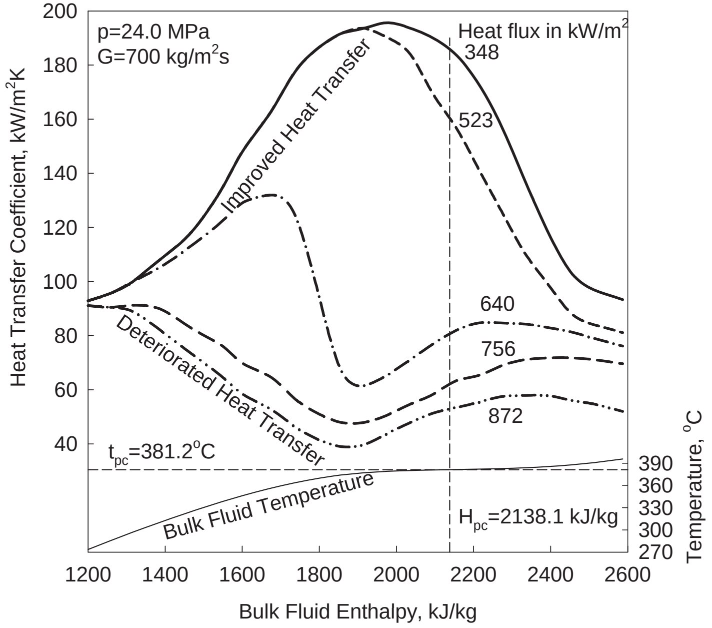
$$Bo_b^* = Gr_b^* / (\mathrm{Re}_b^m \, \mathrm{Pr}^n)$$

$$C_b \sim 10^5$$

Deterioration occurs under mixed convection

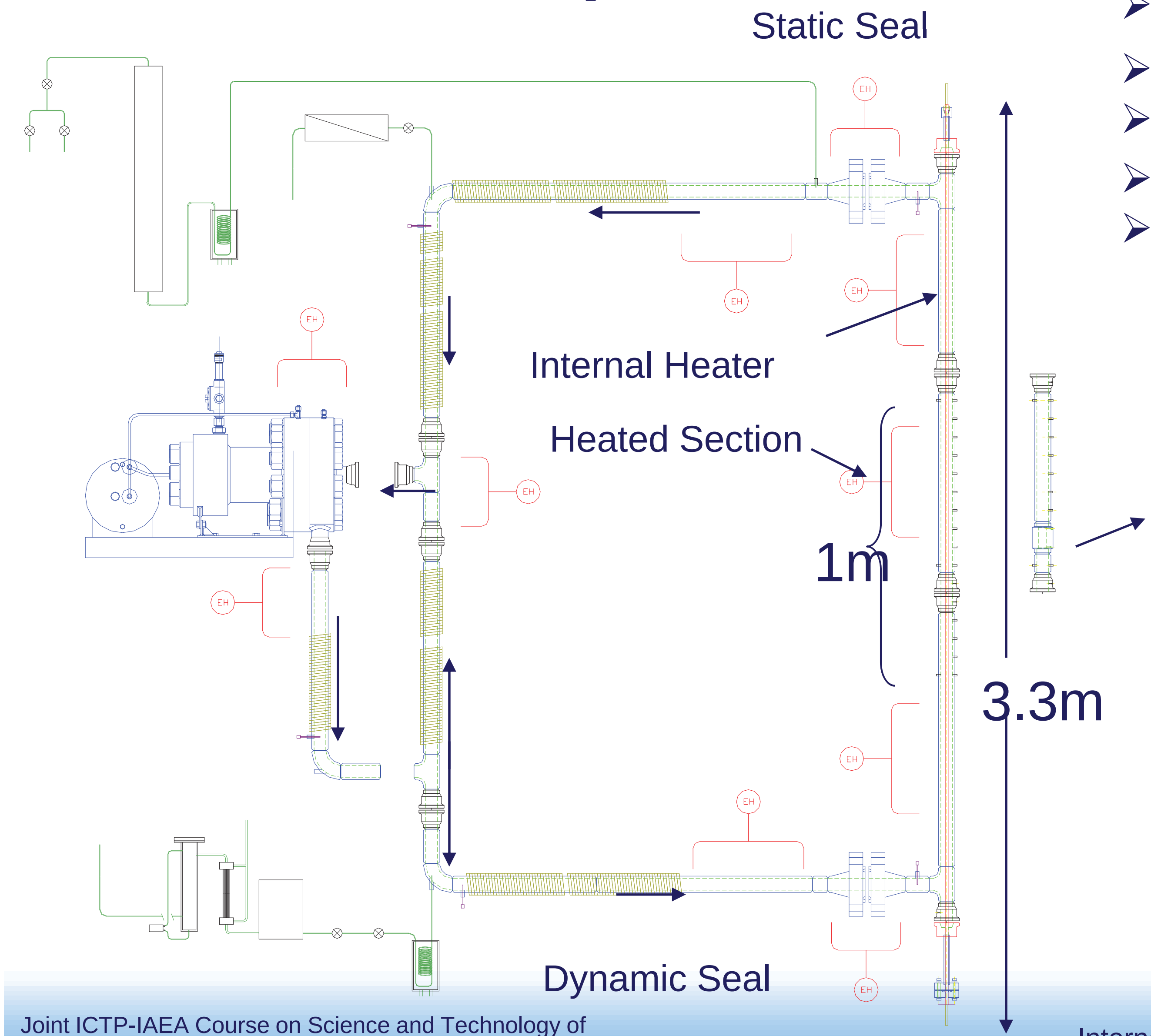
Upward flow low mass flux high heat flux, effect increases with increase heat flux or decrease in mass flux





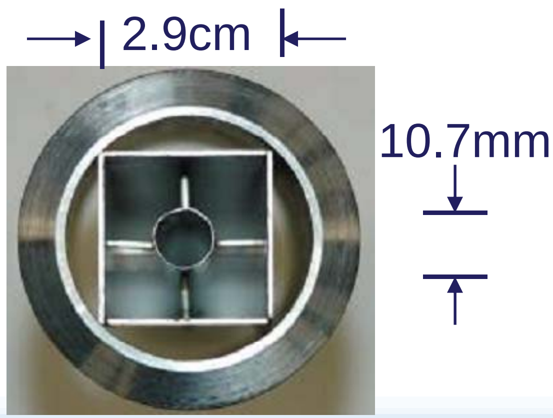
The increase in the buoyancy force in upward flow causes changes in the turbulent boundary layer causing a reduction in the turbulent shear stress (laminar like boundary layer)

Look at a specific experiment to understand the phenomena



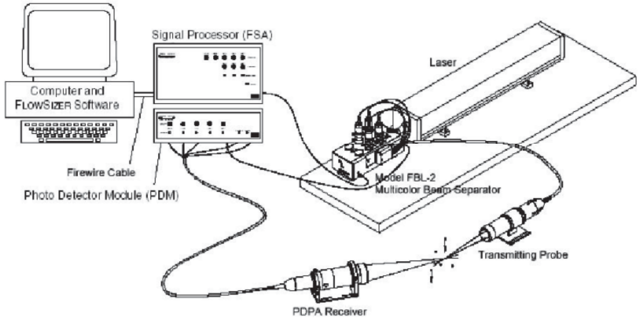
Requirements:

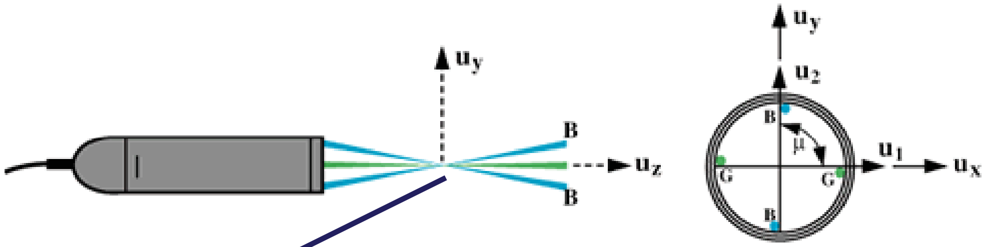
- High P → 25 MPa
- High T → 600 ºC
- Wide G → 2000 kg/m²s
- High Q" → 1.5 MW/m²
- Prototypic Geometry
- Optical Access

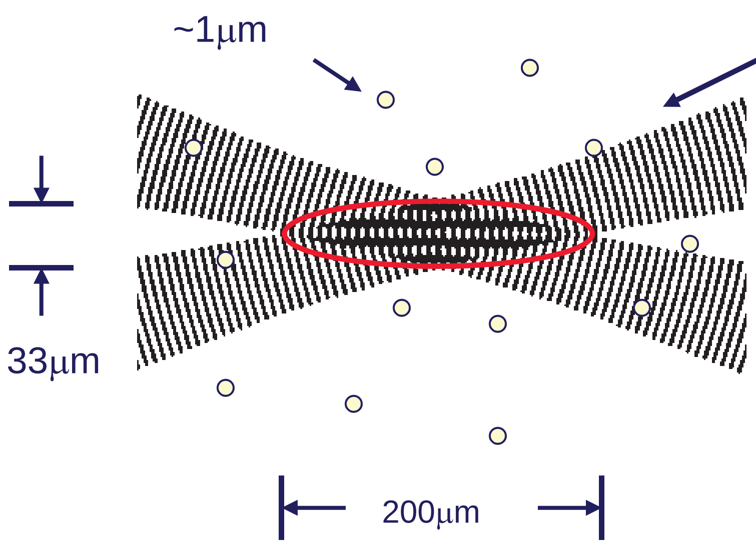


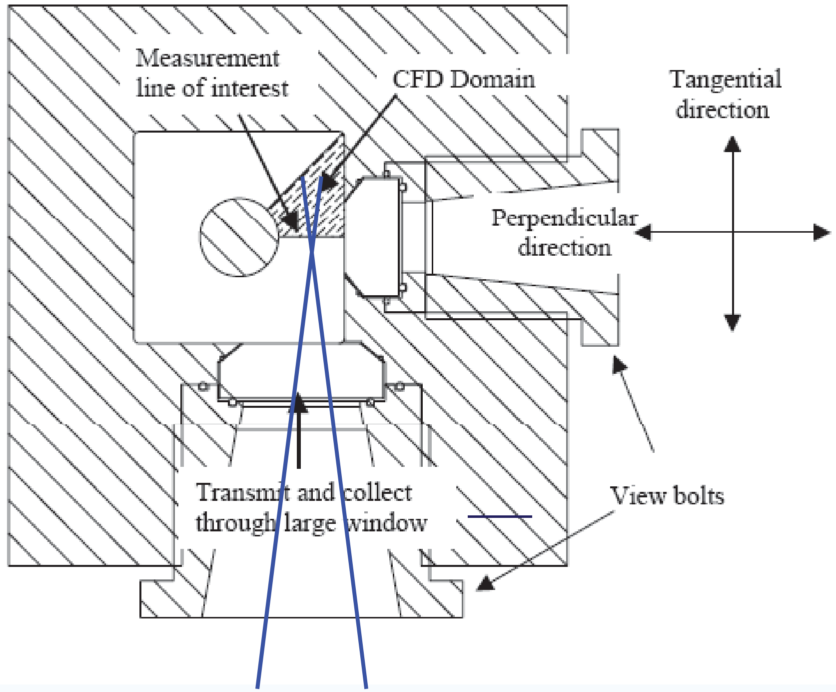
Mean and Turbulent Velocity Measurements

Laser Doppler Velocimetry (LDV) Fluid velocity from light scattered off seeded particles





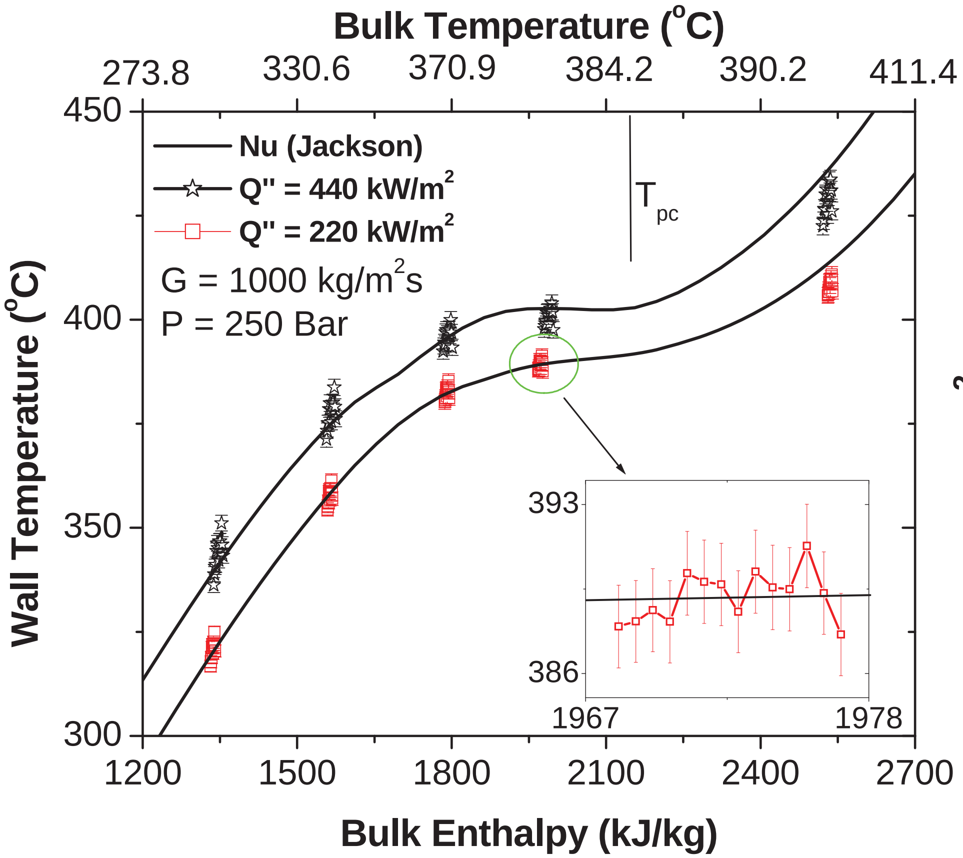


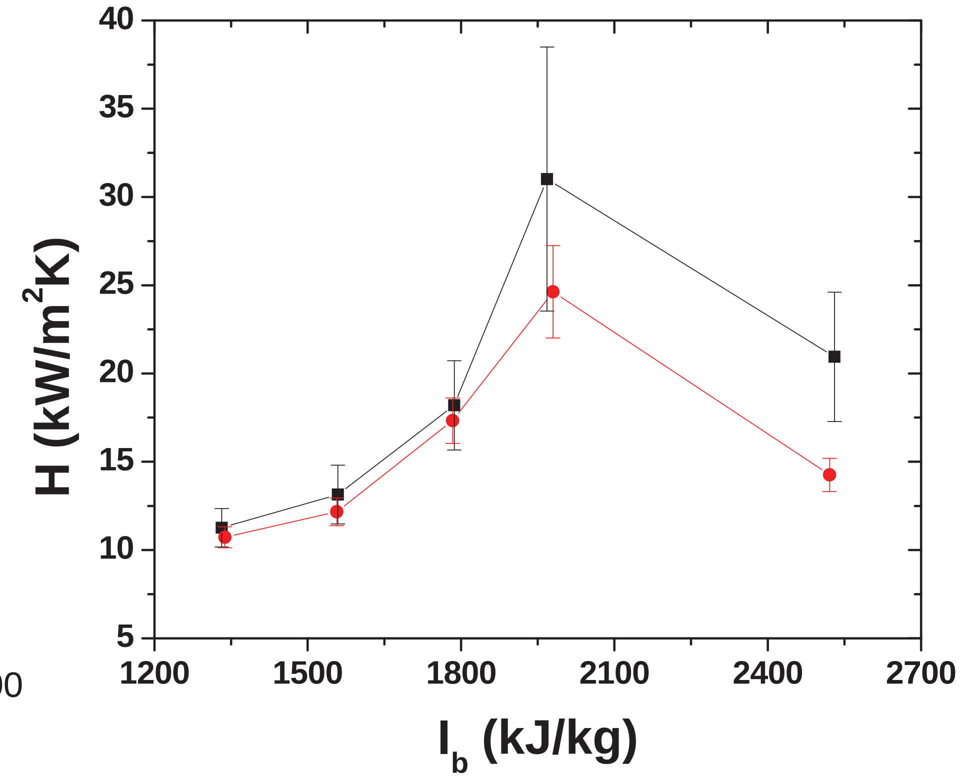


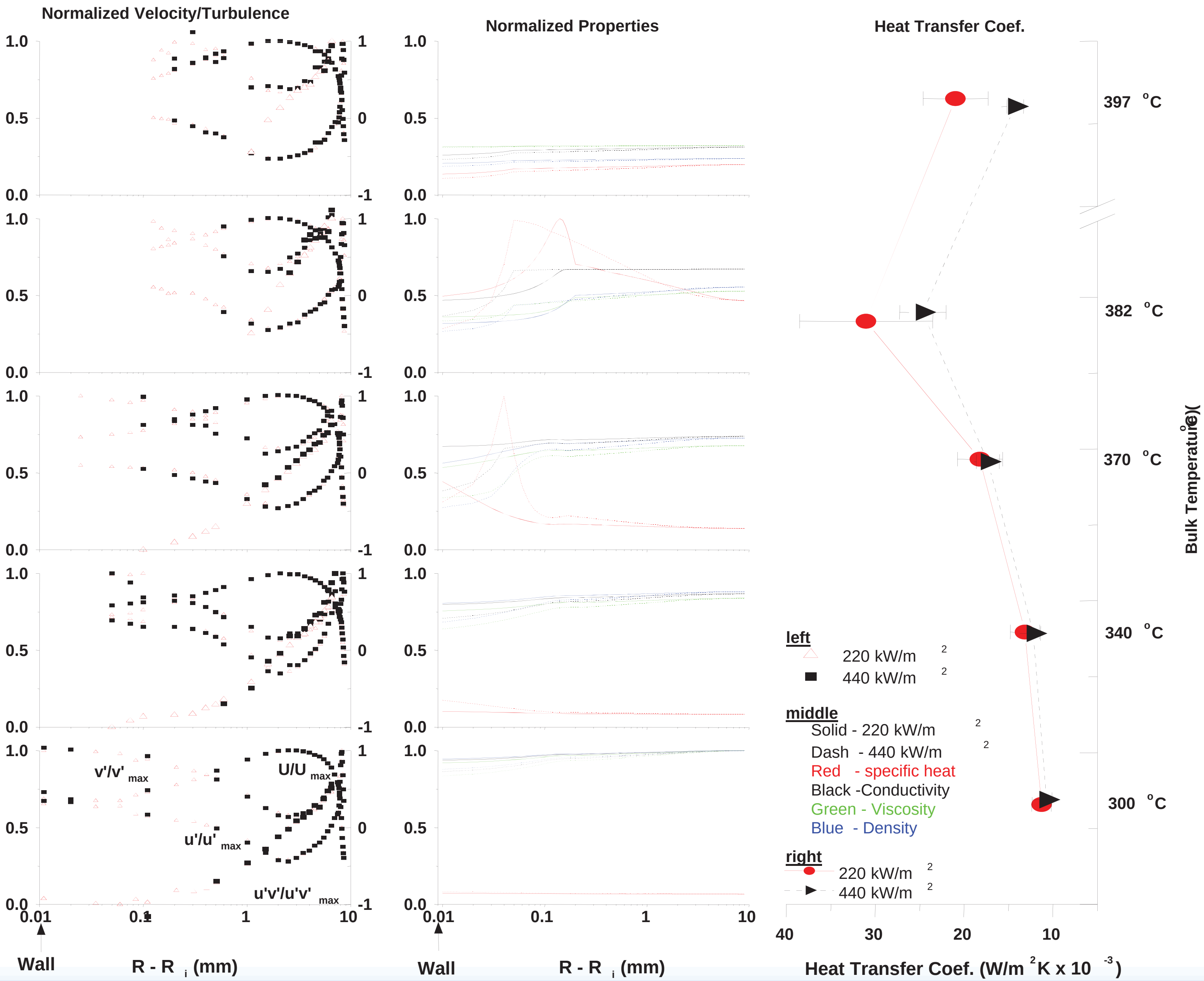
High Mass Velocity: Experimental Conditions

G = 1000 kg/m^{2}s

Square geometry heat transfer data





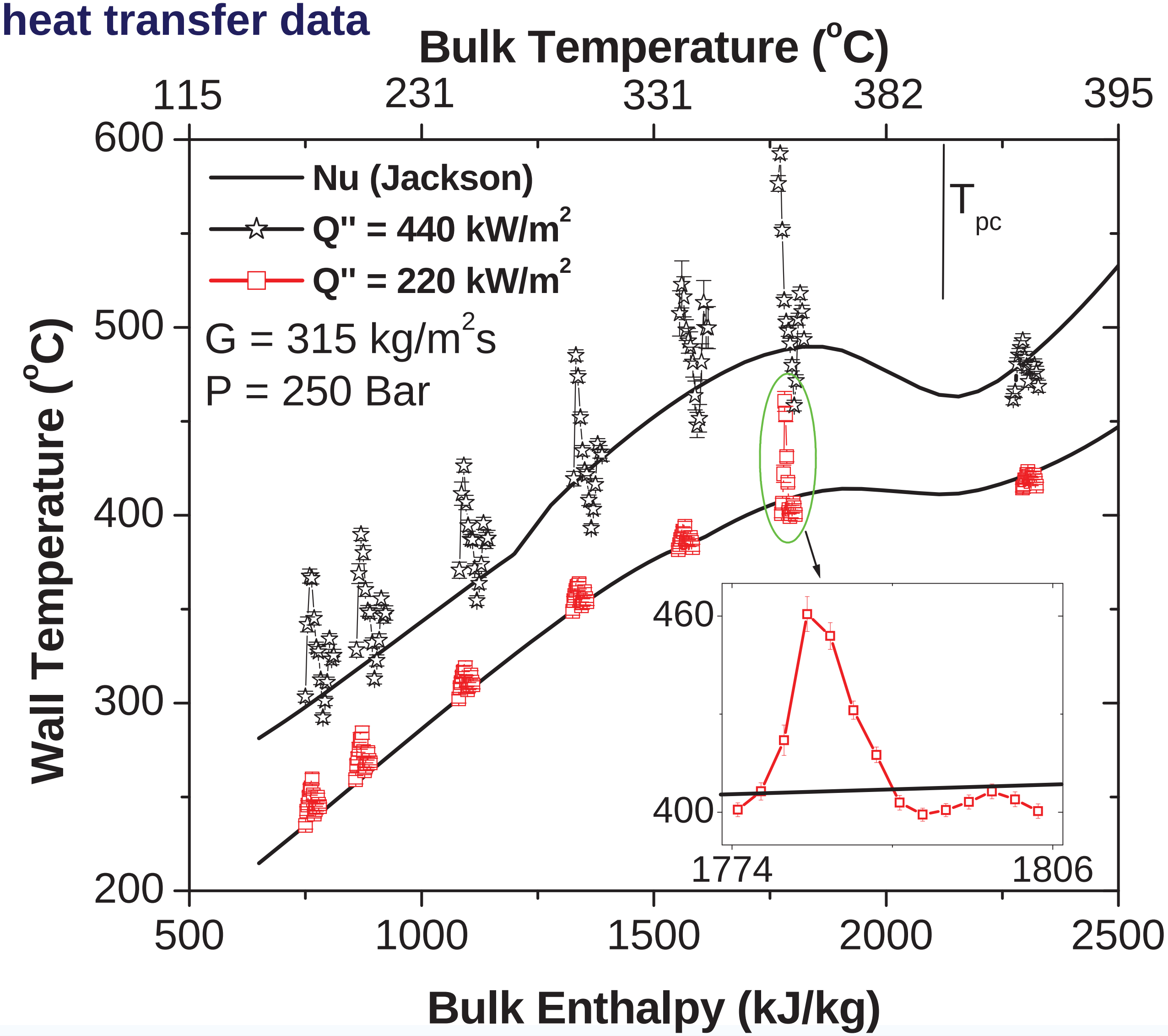


Normalized Velocity/Turbulence
Normalized Properties
Heat Transfer Coef.
v'/v' max
U/U max
u'/u' max
u'v'/u'v' max
Wall
R - R i (mm)
Wall
R - R i (mm)
397 °C
382 °C
370 °C
340 °C
300 °C
Bulk Temperature (°C)
left
220 kW/m2
440 kW/m2
middle
Solid - 220 kW/m2
Dash - 440 kW/m2
Red - specific heat
Black -Conductivity
Green - Viscosity
Blue - Density
right
220 kW/m2
440 kW/m2
Heat Transfer Coef. (W/m2K x 10-3)

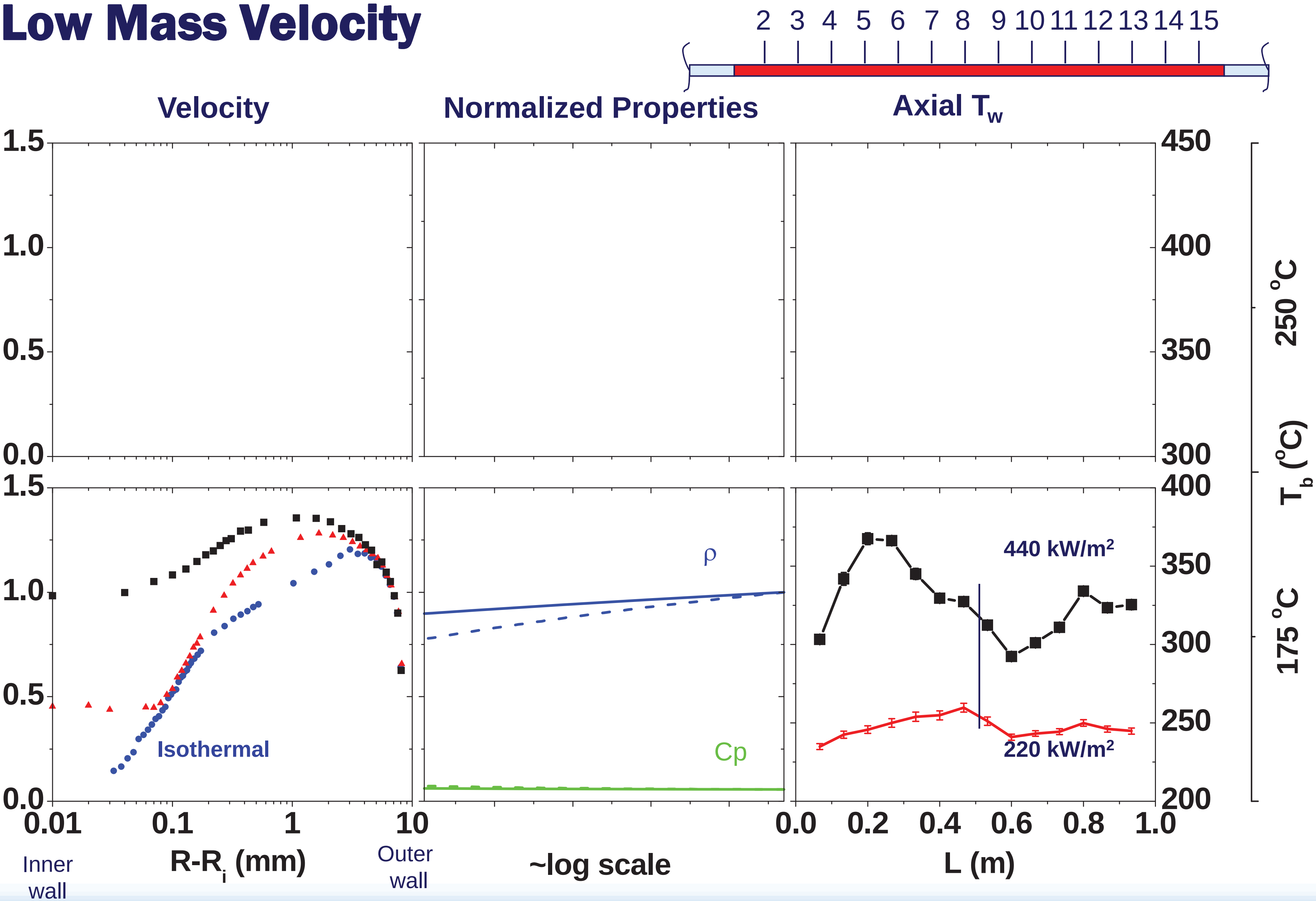
Low Mass Velocity: Experimental Conditions

Square geometry heat transfer data

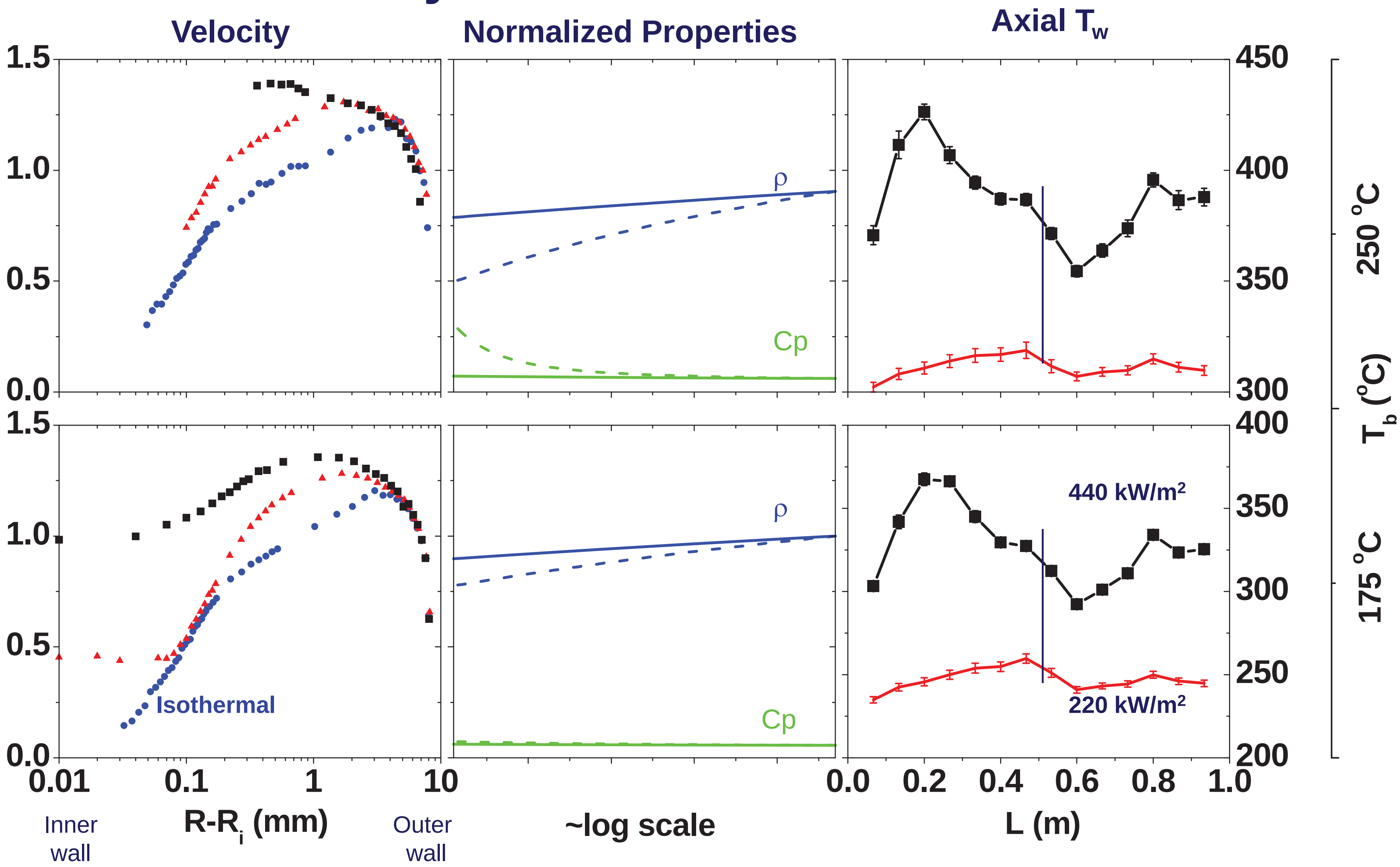
G = 300 kg/m²s



Low Mass Velocity

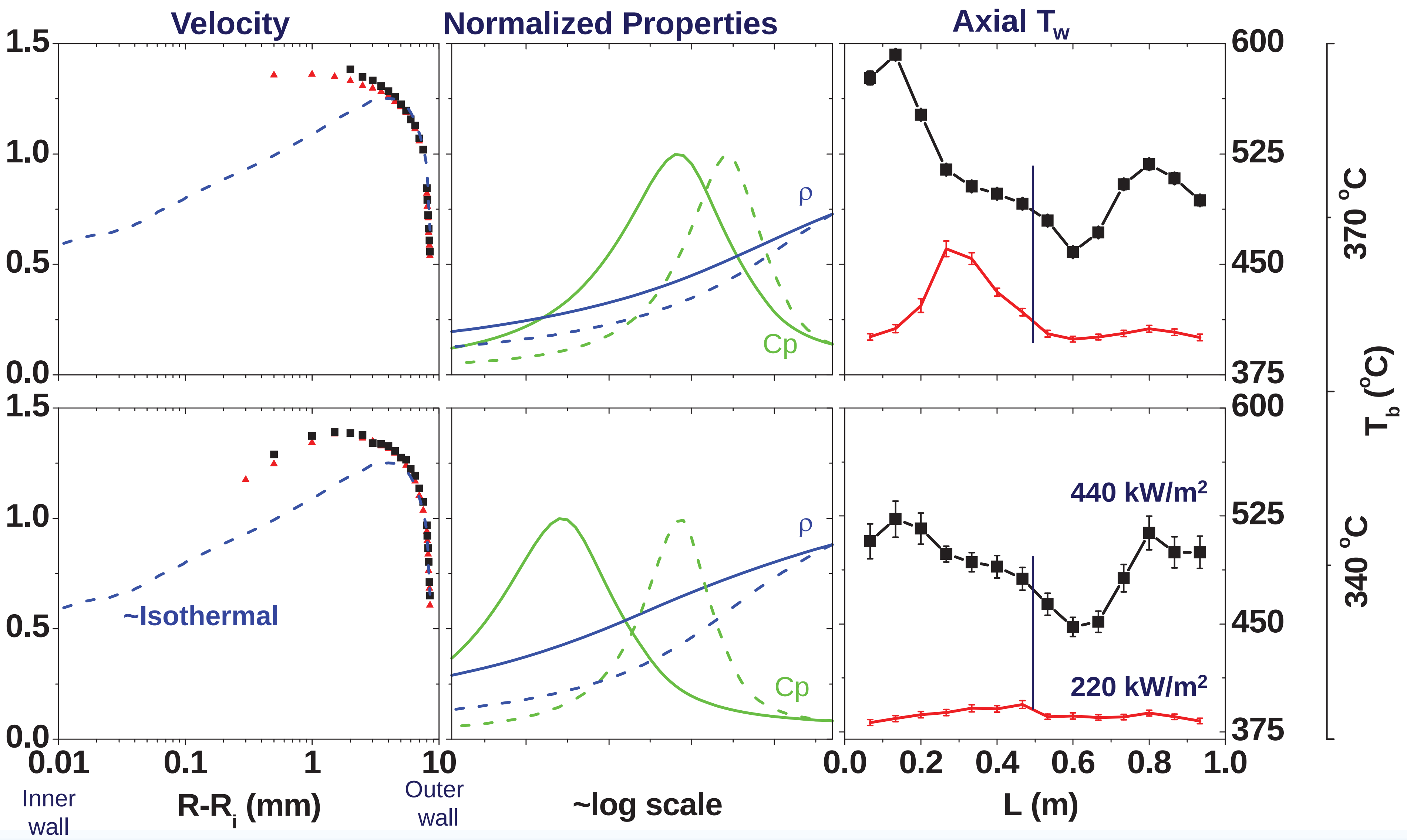


Low Mass Velocity

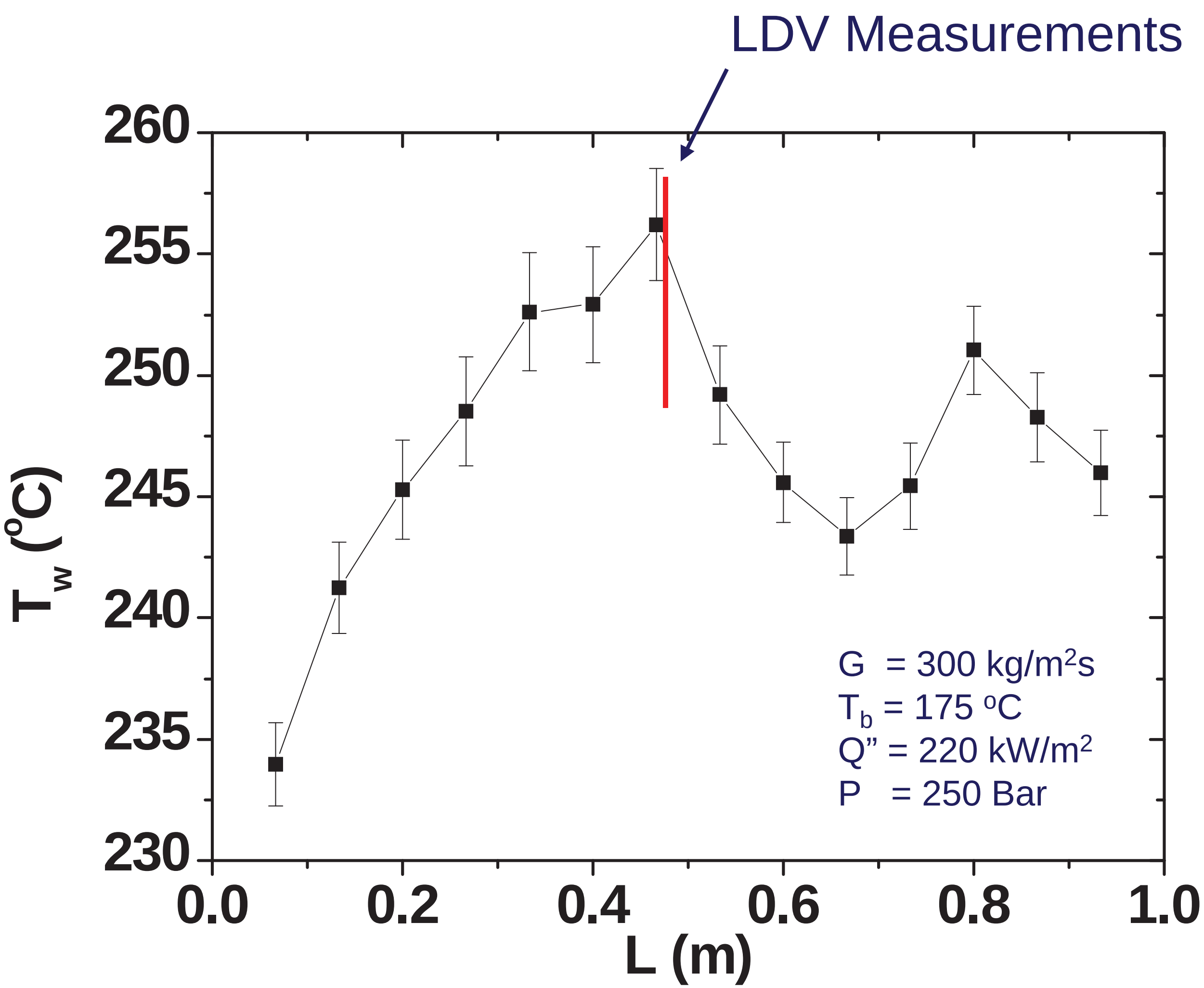


Low Mass Velocity

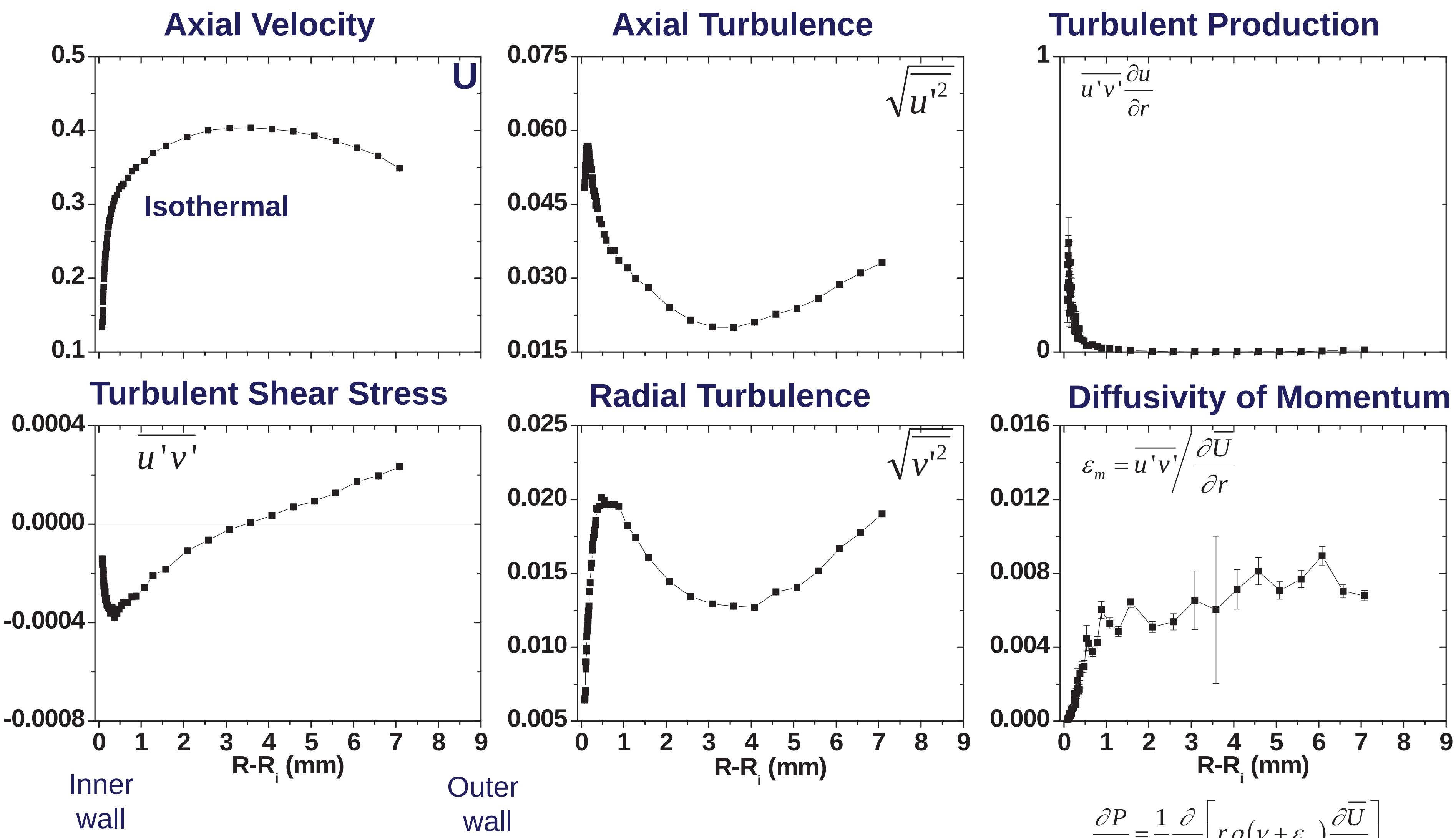
- Velocity/T_w suggest similar effects sub/supercritical



Turbulence measurements in SCF

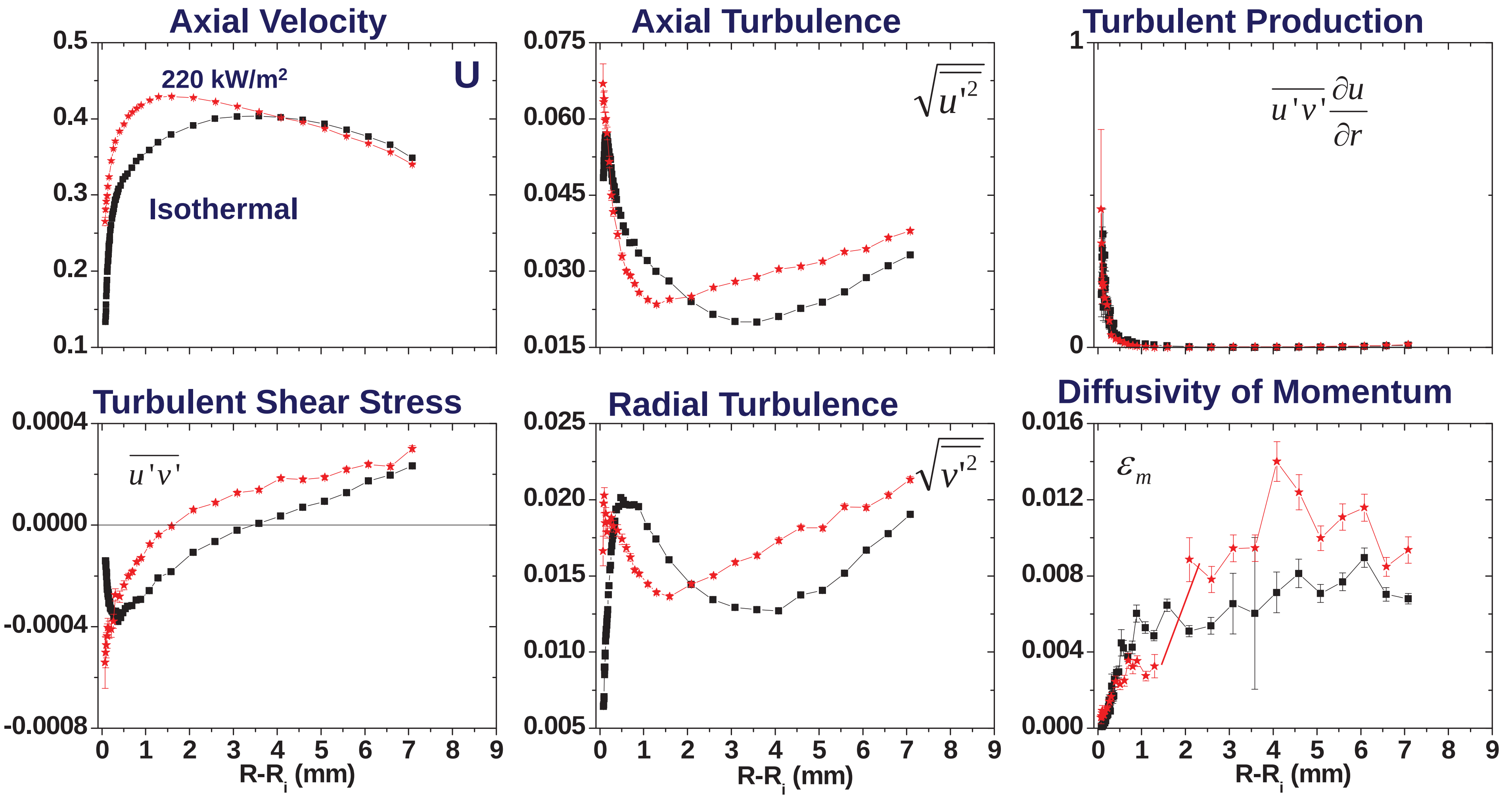


Experimental Turbulence measurements q”=0

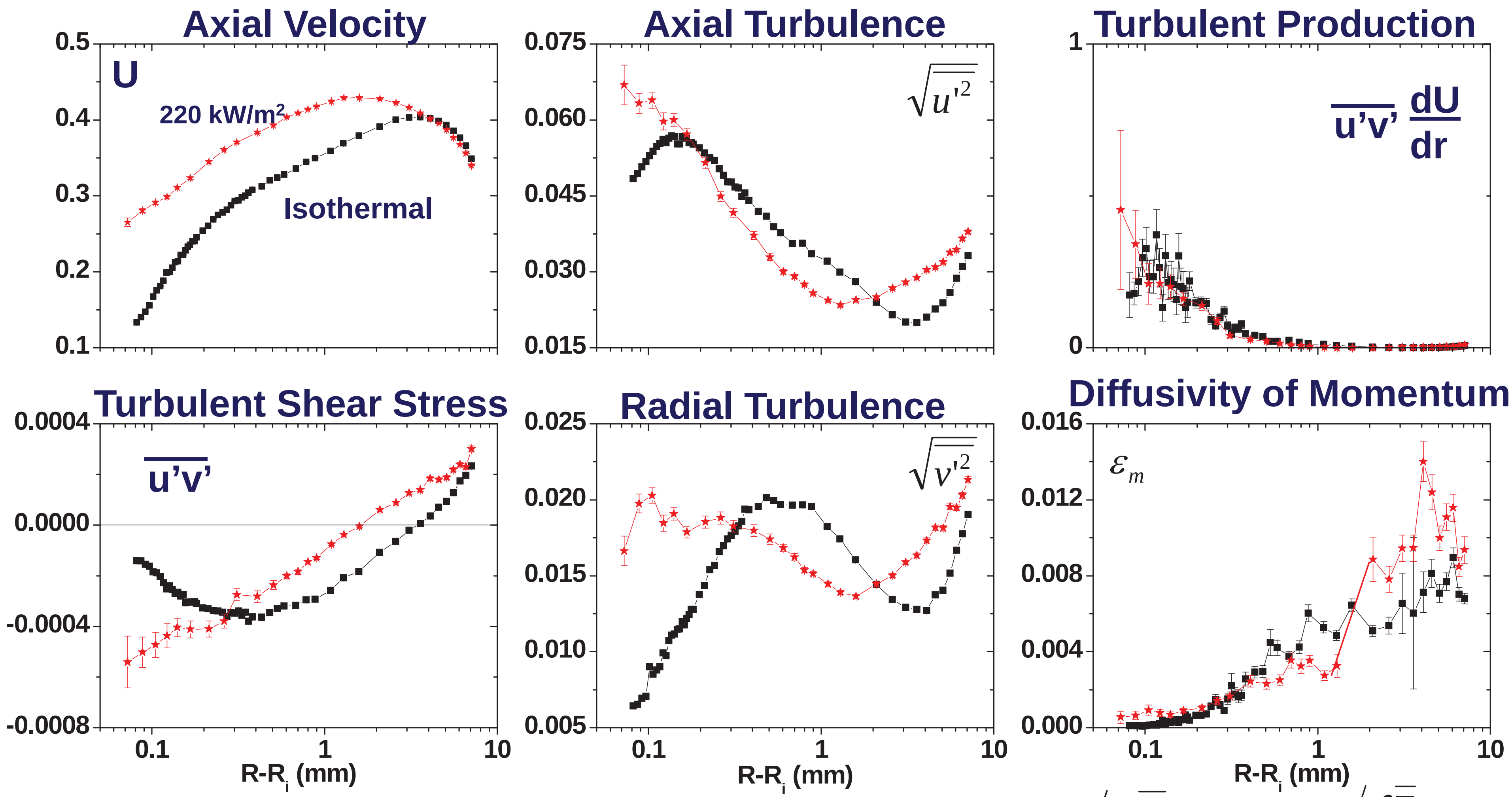


$$\frac{\partial P}{\partial z} = \frac{1}{r}\frac{\partial}{\partial r}\left[r\rho\left(\nu + \varepsilon_m\right)\frac{\partial \overline{U}}{\partial r}\right]$$

Effect of increased heat flux



Experimental Turbulence measurements

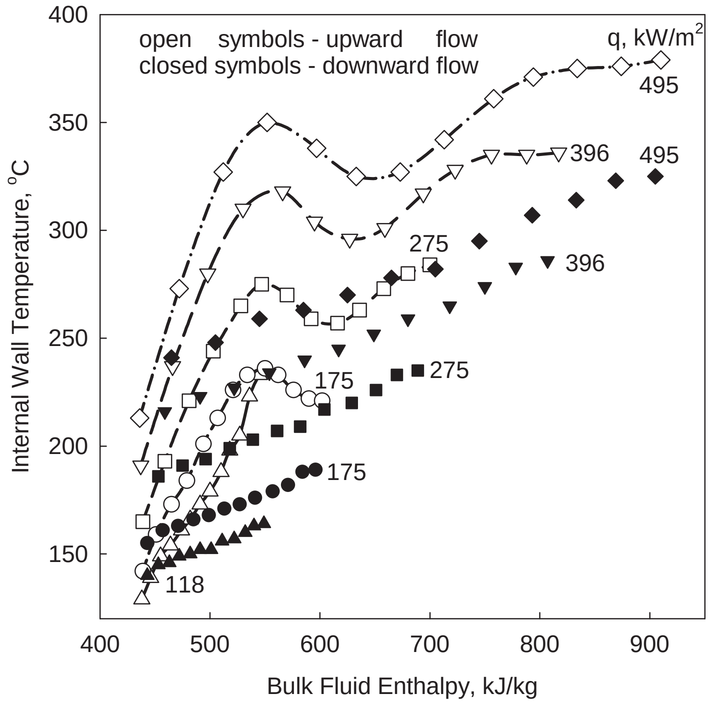


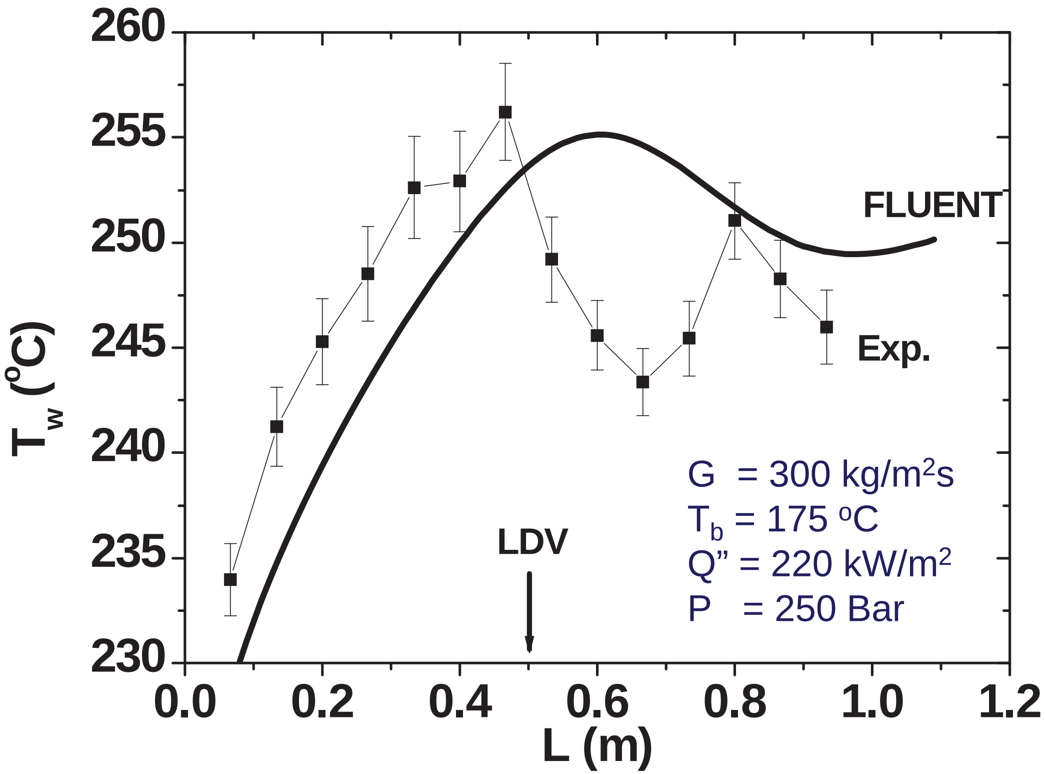
- Increase in very near wall turbulence
- Decrease in turbulence further from wall

$$\Pr_T = \frac{\varepsilon_m}{\varepsilon_H} \approx 1 \qquad \varepsilon_m = \overline{u'v'} \Big/ \frac{\partial \overline{U}}{\partial r} \qquad \varepsilon_H = \overline{v't'} \Big/ \frac{\partial \overline{T}}{\partial r}$$

Mechanism of heat Transfer Deterioration

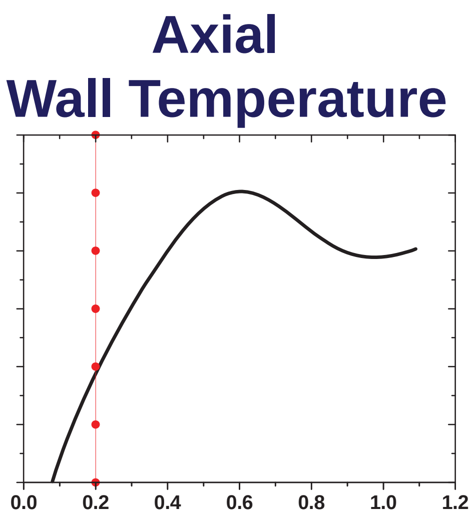
Use CFD to help understand the mechanisms seen in experiments

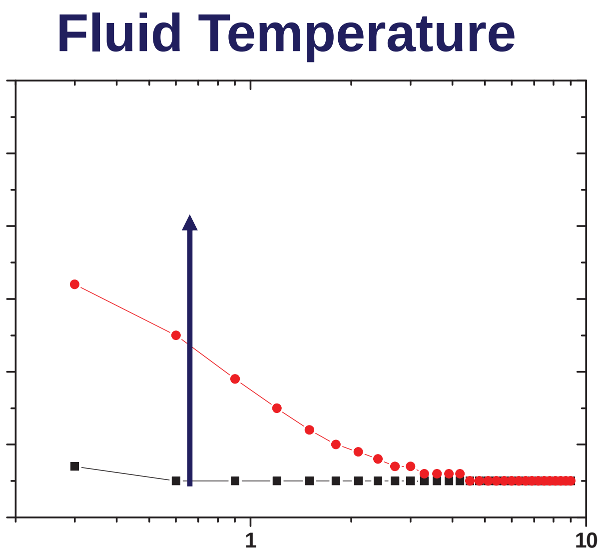


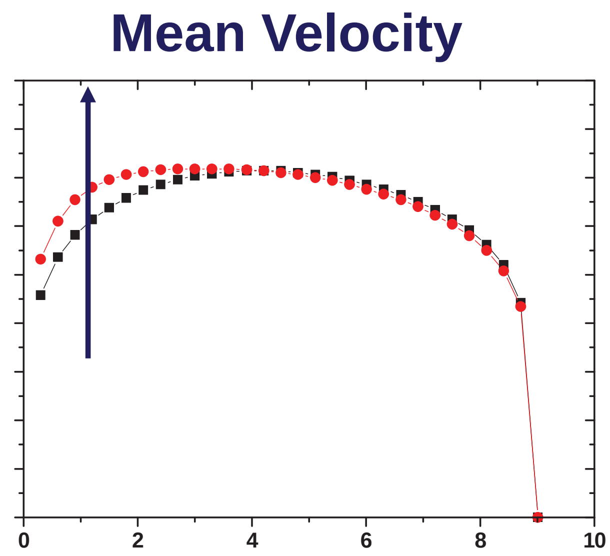


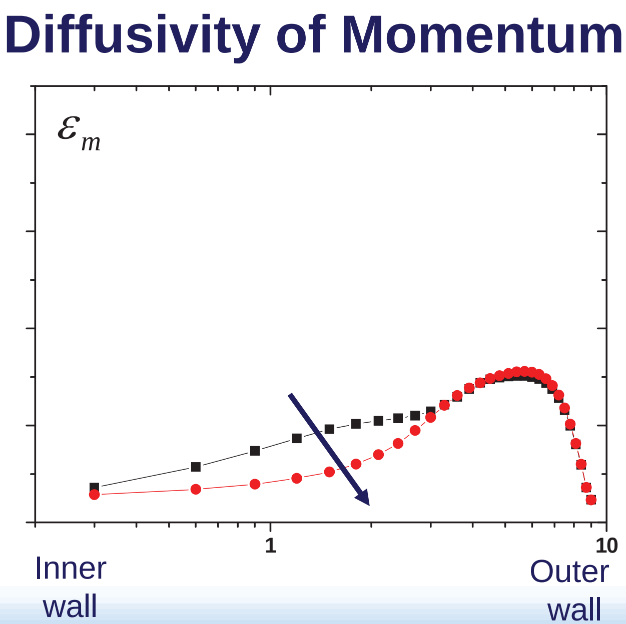
Reynolds Stress Model, Standard Wall Function

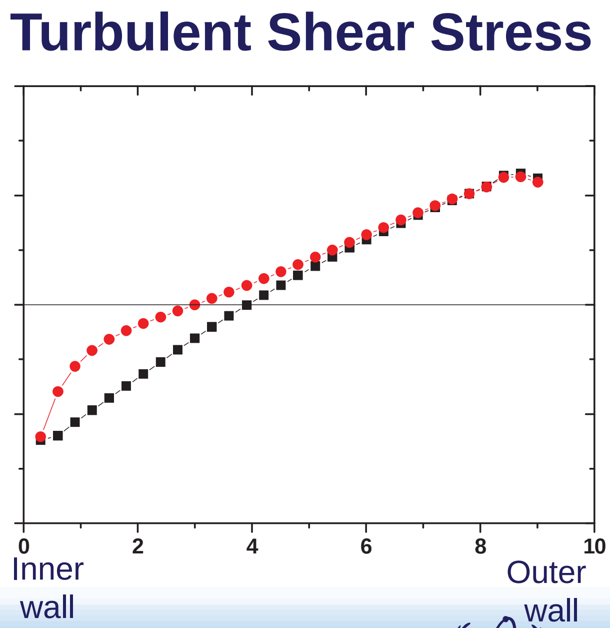
Mechanism of heat transfer deterioration



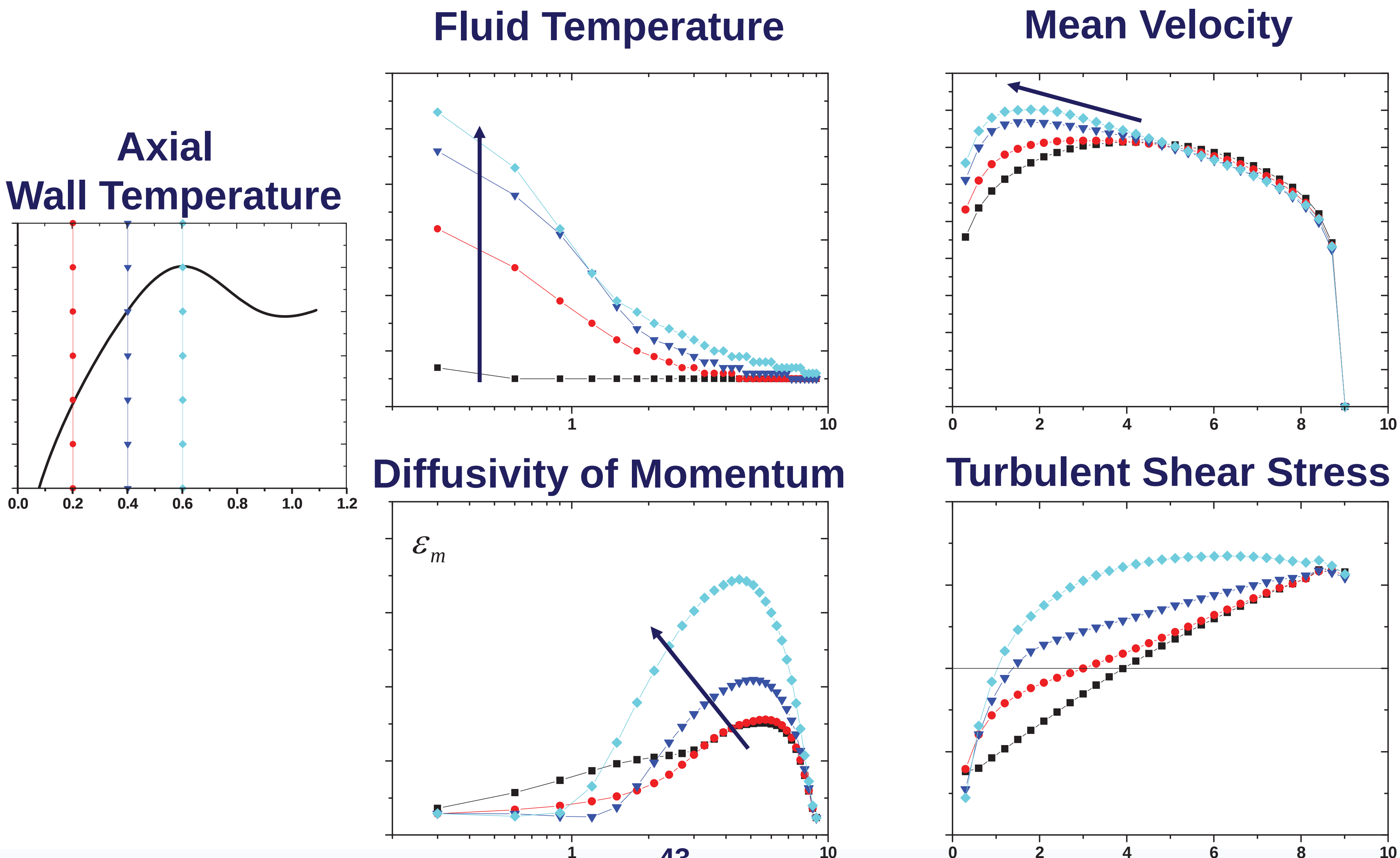






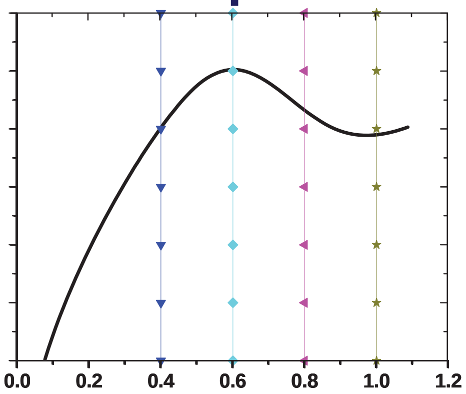


Mechanisms of heat transfer deterioration

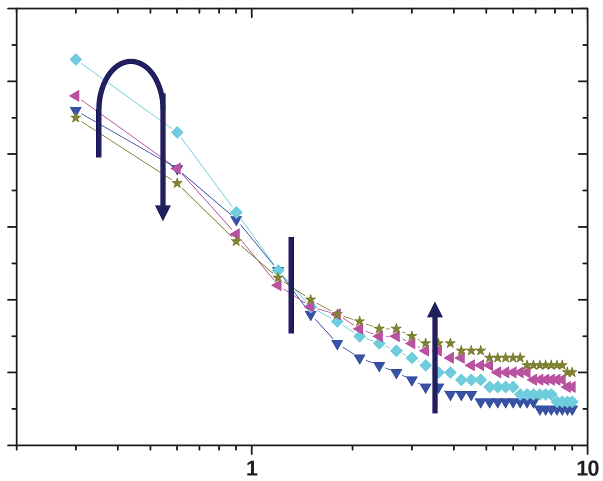


Heat Transfer Recovery

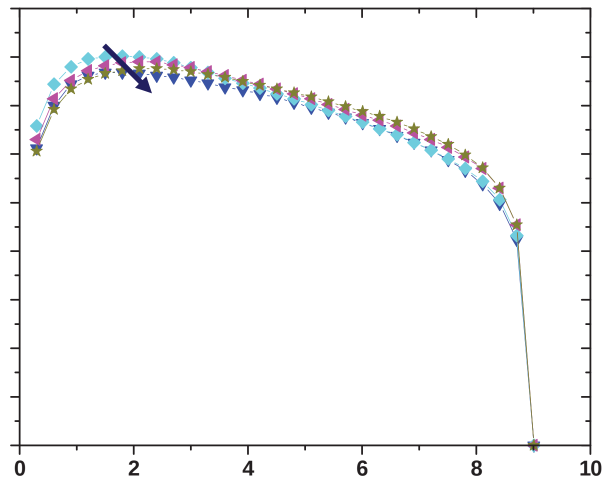
Axial Wall Temperature



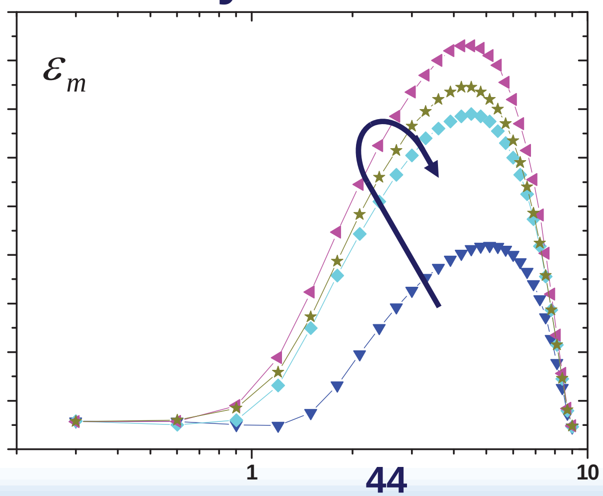
Fluid Temperature



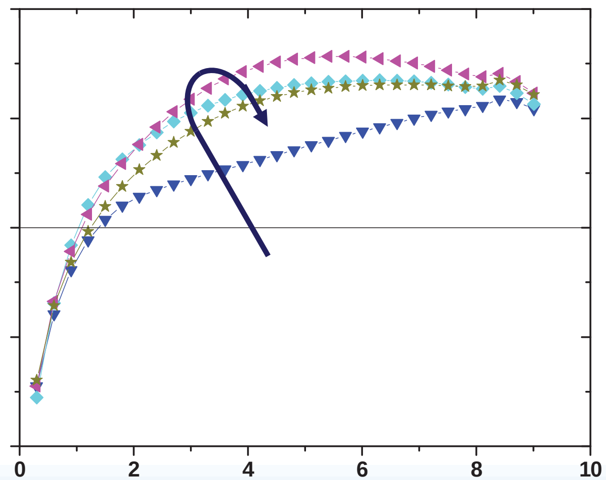
Mean Velocity



Diffusivity of Momentum



Turbulent Shear Stress



Mixed convection

- It is difficult to develop a correlation that can include buoyancy and acceleration effects
- Work is on going in this area

Jackson has recently proposed the following correlation

$$\frac{\mathrm{Nu_b}}{\mathrm{Nu_{b_o}}} = \left[\left|1 \mp \frac{C_B}{F_{TD}} \frac{\mathrm{Gr_b^*}}{\mathrm{Re_b^{3.425}}\,\mathrm{Pr_b^{2n}}} \left(\frac{\overline{\mu}}{\mu_b}\right)\left(\frac{\overline{\rho}}{\rho_b}\right)^{-1/2} \left(\frac{\rho_w}{\rho_b}\right)^{-0.3} \left(\frac{\overline{c}_p}{c_{p_b}}\right)^{-0.4} \left(\frac{\overline{\mathrm{Pr}}}{\mathrm{Pr_b}}\right)^{-0.4} \left(\frac{\overline{\rho\beta}}{\rho_b\beta_b}\right)\left(\frac{\mathrm{Nu_b}}{\mathrm{Nu_{b_o}}}\right)^{-2.1}\right|\right]^{0.46}$$

$$Nu_{b_o} = KF_{TD}\,\mathrm{Re}_b^{0.8}\,\mathrm{Pr}_b^{0.4}\,F_{VP_2}$$ This is the *Nu* for forced convection

$$F_{TD} = 1 + 2.35\,\mathrm{Re}_b^{-0.35}\,\mathrm{Pr}_b^{-0.4}(x/d)^{-0.6}\exp(-0.39\,\mathrm{Re}_b^{-0.1}(x/d))$$ Entrance effect

$$\mathrm{Gr_b^*}(= g\beta_b q_w d^4 / k_b \nu_b^2)$$ Gr based on applied heat flux

A word on CFD

- There are several groups working on CFD applications (SC17)
- The thermal boundary layer for SCF's is typically very thin and it is necessary to resolve very close to the wall $y^{+}<1$
- It is also important to use real thermo-physical properties for the fluid (i.e. call RefProp to get actual properties) This is time consuming – It may be possible to use look-up tables but you need to ensure correct properties.
- k-ω models have been found to give the best result but these were still developed with the assumption of constant properties.
- In general CFD is capable to accurately predict forced flow (improvement) it is more difficult to get deterioration since the properties influence the turbulent boundary layer significantly and most methods RANS rely on constant property equations.

Governing equations for turbulent boundary laye

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}(\rho u_j)$$

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_j u_i) = \frac{\partial}{\partial x_j}(\sigma_{ji}) + X_i$$

$$\frac{\partial}{\partial t}\left[\rho\left(i + \frac{1}{2}u_i u_i\right)\right] + \frac{\partial}{\partial x_j}(\rho u_j(i + \frac{1}{2}u_i u_i)) = \frac{\partial}{\partial x_j}\left(k\frac{\partial T}{\partial x_i}\right) + \frac{\partial}{\partial x_j}(u_i \tau_{ji}) + u_i X_i + S$$

RANS

$$\mathcal{F} = \bar{\mathcal{F}} + \mathcal{F}'$$

$$\bar{\mathcal{F}} = \frac{1}{\Delta t}\int_{t_0}^{t+\Delta t} \mathcal{F}\, dt$$

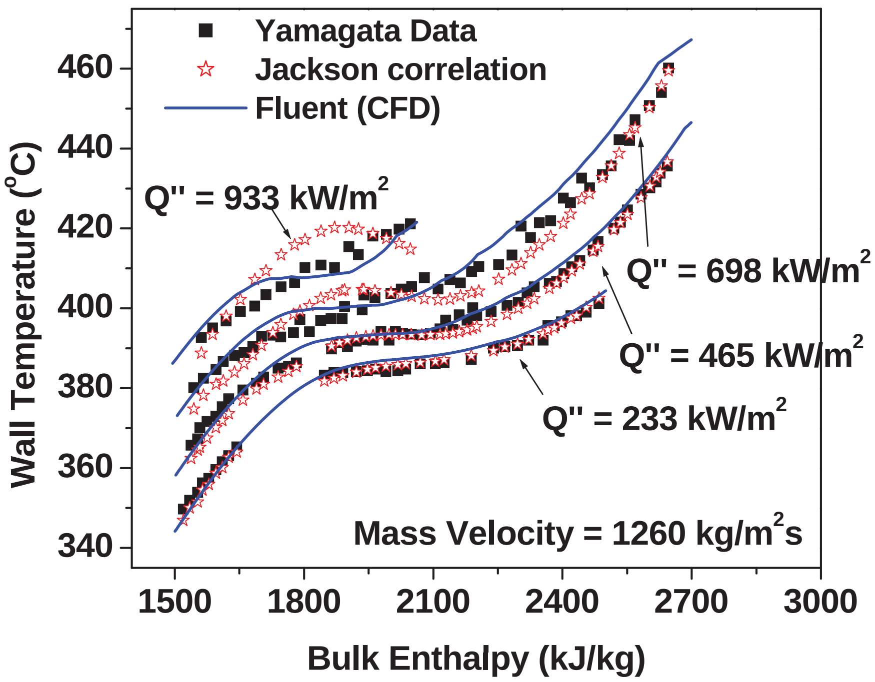
Favre Averaging

$$\tilde{\mathcal{F}} = \frac{\overline{\rho \mathcal{F}}}{\bar{\rho}} = \bar{\mathcal{F}} + \frac{\overline{\rho' \mathcal{F}'}}{\bar{\rho}}$$

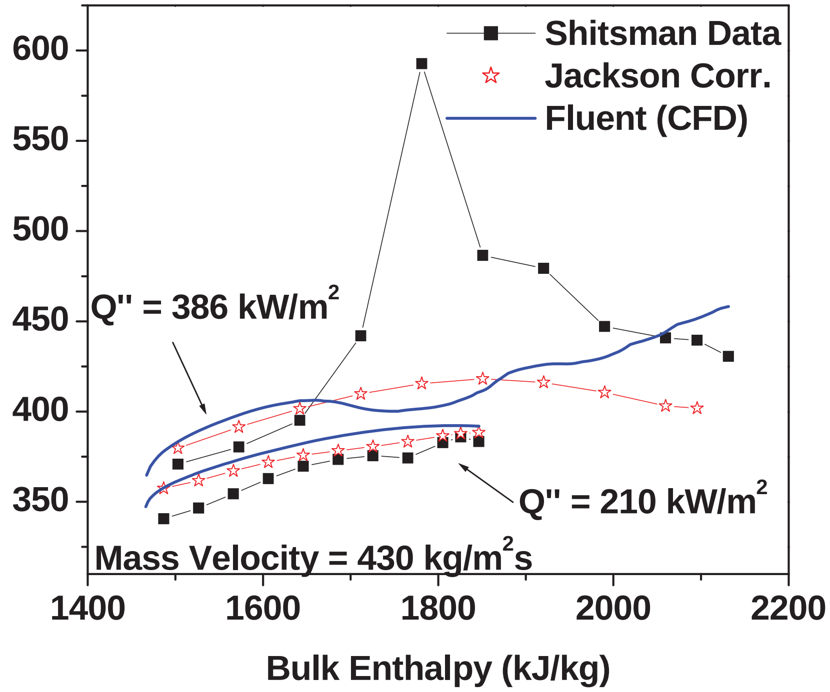
$$\mathcal{F}'' = \mathcal{F} - \frac{\overline{\rho \mathcal{F}}}{\bar{\rho}} = \mathcal{F}' - \frac{\overline{\rho' \mathcal{F}'}}{\bar{\rho}}$$

Comparison of CFD simulations

High mass flux



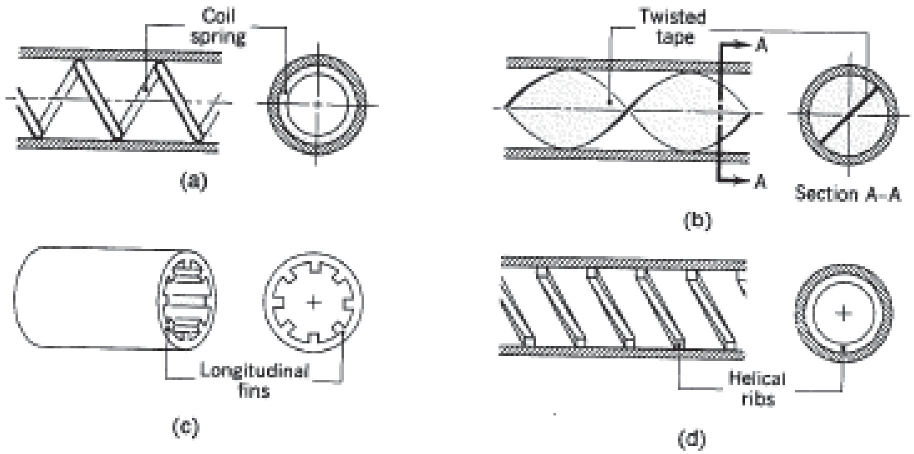
Low mass flux



CFD has trouble with deterioration
There are a lot of “tricks” to improve this but there are issues due how the properties effect the turbulence

Heat transfer enhancement

For more complicated geometry we need CFD or a correlation for the desired geometry



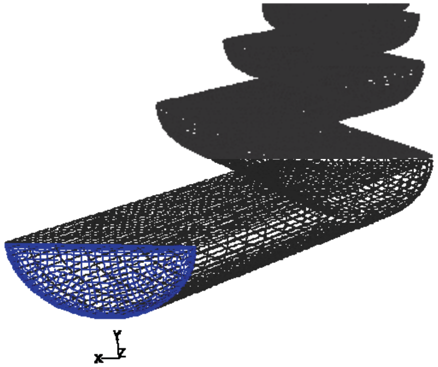
As with normal fluids it is possible to enhance heat transfer in SCF's either for internal tube flow or external flow this can be done with ribs, wire wraps, fins, pins or various other structures.

Bundles and grid-spacers

As we have seen the heat transfer coefficient of SCF's can vary significantly and is highly dependent on the fluid dynamic conditions (buoyancy, acceleration, up-flow, down-flow, horizontal, mass flux, Temperature, Pressure, etc.)
For forced flow high Reynold number flows our current best method is modified Nu correlations

Example of CFD simulation of complex flow

S-CO2 flow in a small zig-zag channel



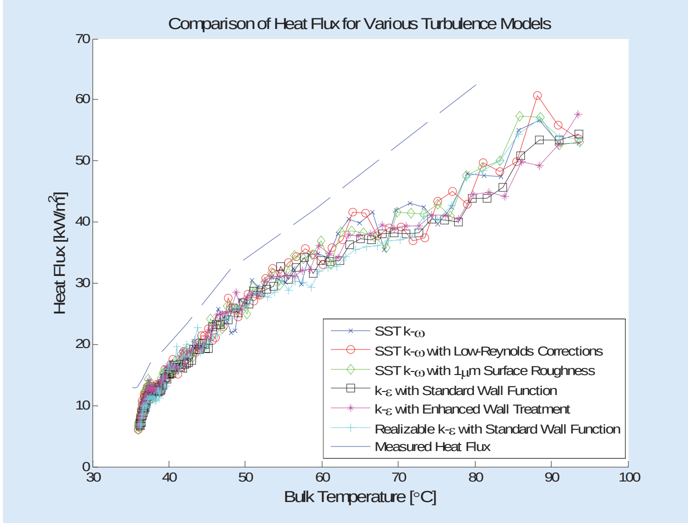
Meshing was performed with hexahedral cells aligned with the predominate flow orientation

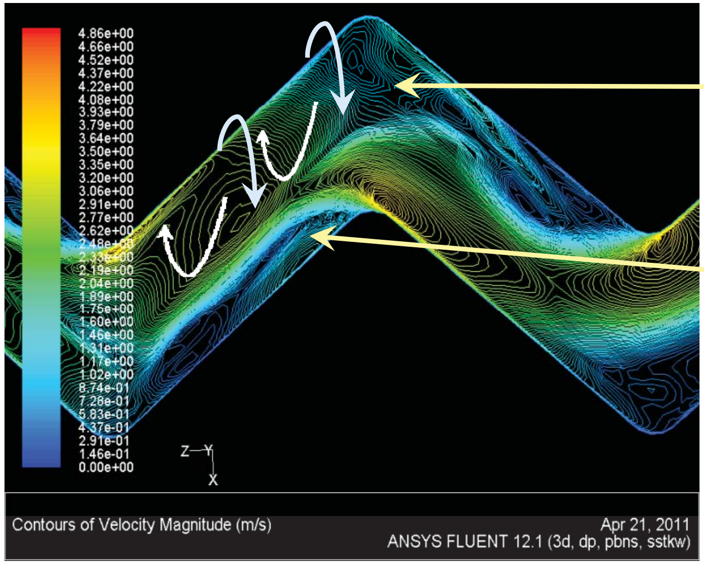
Mesh was inflated at the boundaries

y^+ value around 1.0 for wall adjacent cells (~1 micron in height)

Full-length models typically used 1.0 to 2.5 million cells

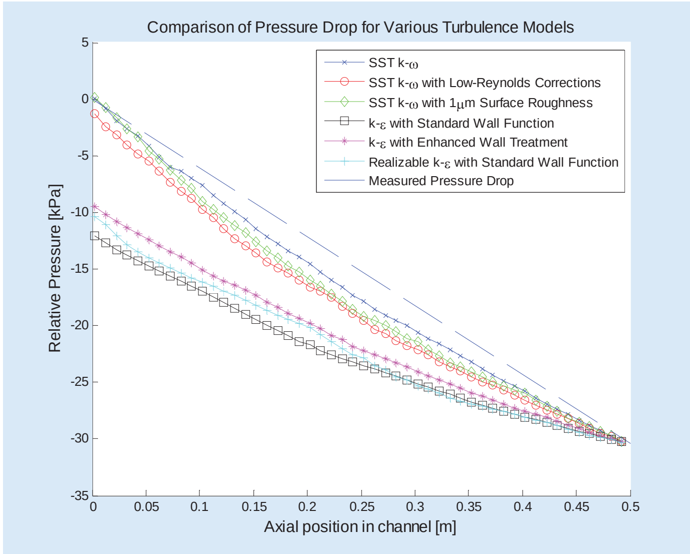
Inlet plenum was also modeled to aid in damping oscillations





Helical flow on the outside of the bends

Large wake regions behind every bend



Bundle data

Reference	p, MPa	t, °C (H in kJ/kg)	q, MW/m²	G, kg/m²s	Flow geometry
Dyadyakin and Popov 1977	24.5	t_b=90–570; H_b=400–3400	<4.7	500–4000	Tight bundle (7 rods (6+1), D_{rod}=5.2 mm, L=0.5 m), each rod has four helical fins (fin height 0.6 mm, thickness 1 mm, helical pitch 400 mm), pressure tube hexagonal in cross section
Silin et al. 1993	23.5; 29.4	H_b=1000–3000	0.18–4.5	350–5000	Vertical full-scale bundles (D_{rod}=4 and 5.6 mm, rod's pitch 5.2 and 7 mm)

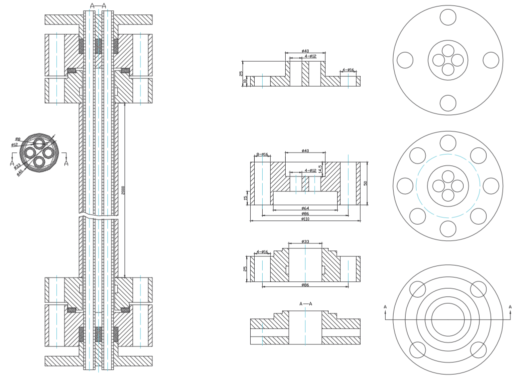
Dyadyakin and Popov 1977 recommend the following correlation for bundles

$$\mathbf{Nu_x} = 0.021\,\mathbf{Re_x^{0.8}}\,\overline{\mathbf{Pr}}_{\mathbf{x}}^{\mathbf{0.7}} \left(\frac{\rho_w}{\rho_b}\right)_x^{0.45} \left(\frac{\mu_b}{\mu_{in}}\right)_x^{0.2} \left(\frac{\rho_b}{\rho_{in}}\right)_x^{0.1} \left(1 + 2.5\frac{D_{hy}}{x}\right)$$

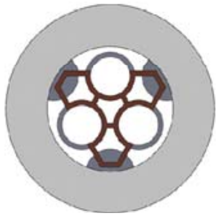
Future Bundle data

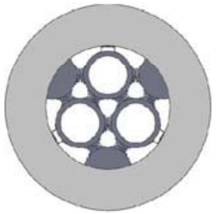
- There has been significant discussion of several groups getting ready to do heat transfer test and a few claiming to be setting up to conduct bundle test, but to date none is available in the open literature
- Xi'an Jiatong university (XJTU) has plans for a 4-rod bundle

Pressure: 23, 25, 28MPa
Mass velocity: 400 - 2000 kg/m^{2}s
Heat fluxes: 200 - 1000 kW/m^2
Inlet fluid temperature: 300 ^{o}C
Outlet fluid temperature: 600 ^{o}C



- AECL and University of Ottawa – Plan to do a three rod bundle with S-CO_2


Endplate


Spacers


Bundle

Where to find Data

The IAEA in cooperation with the OECD/NEA has made a data base of relevant experimental data at the following site; http://www.oecd-nea.org/crp_scwr_ht/ (At this time the data is not open to the public /

- AECL
    - Supercritical carbon dioxide test in a tube
- BARC
    - Supercritical pressure natural circulation experiments with CO_2
- IPPE
    - Experimental data on heat transfer to carbon dioxide under supercritical pressure
- KAERI
    - E1 : Upward flow in eccentric annular channel
    - R1 : Upward flow in concentric annular channel
    - R1D : Downward flow in concentric annular channel
    - T4 : Upward flow in tube with 4.4 mm inner diameter
    - T457 : Upward flow in tube with 4.57 mm inner diameter
    - T457D : Downward flow in tube with 4.57 mm inner diameter
    - T6 : Upward flow in tube with 6.32 mm inner diameter
    - T6D : Downward flow in tube with 6.32 mm inner diameter
    - T6W : Upward flow in tube with 6.32 mm inner diameter (with wire type turbulence generator)
    - T9 : Upward flow in tube with 9.0 mm inner diameter
    - T9D : Downward flow in tube with 9.0 mm inner diameter
- University of Wisconsin - Madison
    - S-CO_2 depressurization
    - S-CO_2 mini-channel heat transfer
    - Annular heat transfer measurements in upflow geometry
    - Fluid flow measurements in supercritical water in upflow geometry
- Shanghai Jiao Tong University
    - Heat Transfer of Supercritical Water

Where to find data (open literature)

- Major Conferences:

  NURETH – International Topical meeting on Nuclear Reactor thermalhydraulics
  ICAPP – International Congress on Advances in Nuclear Power Plants
  ICONE – International Conference on Nuclear Engineering
  International symposium on supercritical-water-cooled Reactors
  Supercritical CO2 power cycle symposium
  International conference GLOBAL
  American Nuclear Society (ANL) International meeting

- Books

  Heat transfer and Hydraulic Resistance at Supercritical Pressures in Power Engineering Applications – Pioro and Duffey

  Super light water Reactors and Super fast Reactors – Oka, Koshizuka, Ishiwateri, Yamaji

Summary

- There are hundreds of publications devoted to forced convective heat transfer under SC pressures most in circular tubes.
- Heat transfer in SCF's are strongly influenced by rapid changes in thermophysical properties
- Heat transfer data observed three modes of heat transfer (Normal, improved and deteriorated)
- There are several correlations that allow the estimation of the heat transfer coefficient . The currently recommended correlation for forced convection is Jackson's correlation which is good to within 20% for simple geometries.

$$Nu_{JA,b} = 0.0183\,\mathrm{Re}_b^{0.82}\,\mathrm{Pr}_b^{0.5}\left(\frac{\rho_w}{\rho_b}\right)^{0.3}\left(\frac{\bar{c}_p}{c_{pb}}\right)^{n} \qquad n = f\left(T_b, T_w, T_{pc}\right)$$

  - Heat transfer is enhanced due to increases in the specific heat
  - If better than 20% is needed a specific correlation for a specific conditions is needed.
  - If no data exists for similar geometry, CFD analysis may be necessary

Summary (cont.)

- It is possible to estimate when mixed convection effect are present.

$$\frac{\bar{Gr}_b}{Re_b^{2.7}} < 10^{-5}$$

  - Under mixed convection conditions in buoyancy and acceleration effects can result in deterioration of heat transfer.
  - This deterioration is difficult to predict (approximate methods for buoyancy and acceleration based on mechanisms have are being developed)
- There are some groups that are also trying to build look-up tables that allow determination of heat transfer coefficients under set geometries.
- There is very little data for bundle geometry, however it is likely that the heat transfer is further improved due to grid structure and there will be little or no deterioration. CFD is necessary for complex geometry
- CFD Techniques are being developed. The k-ω models seem to work best, however the boundary layer must be resolved to $y^+<1$ and real fluid properties need to be implemented. Current work on Farve averaging techniques and modification to turbulence models are under way.
- There is currently a lot of work being conducted in this area and are being input into the IAEA data bank. China, Canada, Japan, EU and US are working on facilities to conduct additional heat transfer and bundle tests.

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…Thank you for your attention!

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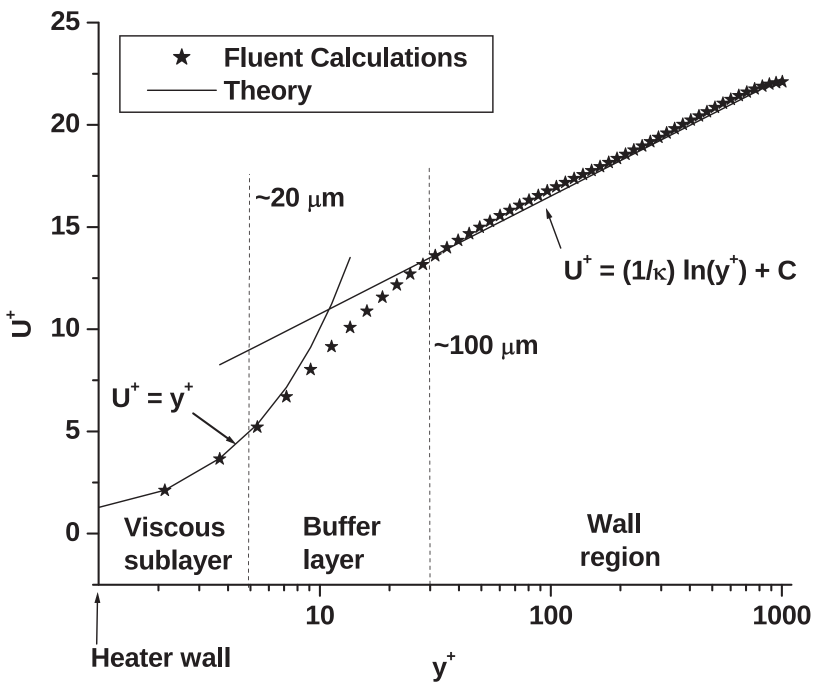
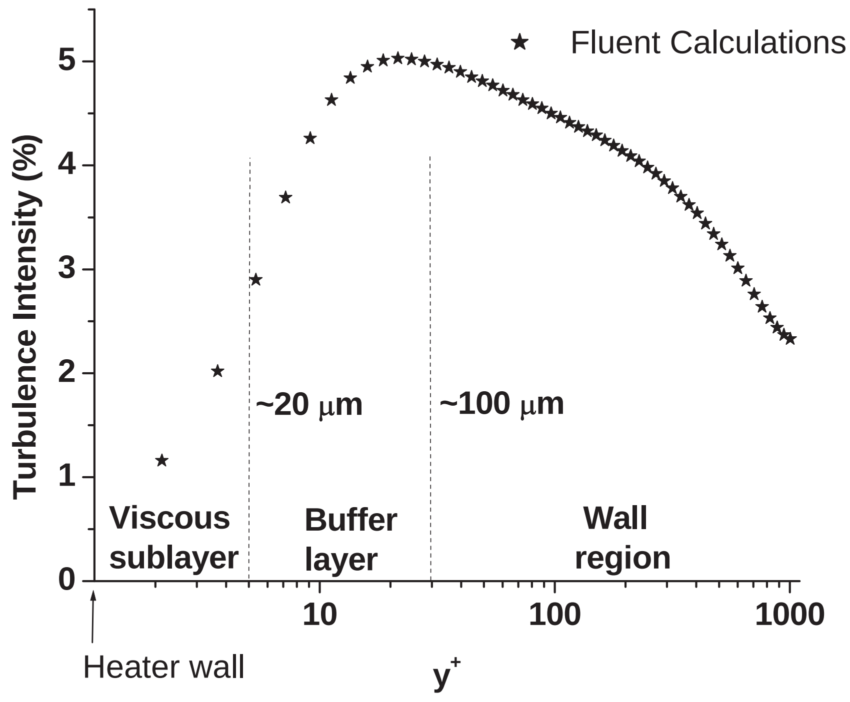
Turbulence measurements





Scaling variables: $U^+ = \dfrac{u}{u_\tau}$ $\quad u_\tau = \sqrt{\tau_w/\rho}$ $\quad y^+ = y\dfrac{u_\tau}{\nu}$ $\quad \tau_w = \mu \left.\dfrac{\partial \bar{u}}{\partial y}\right|_{y=0}$

Previous Work: Shiralkar 1970 – Deterioration *Regardless* of Orientation

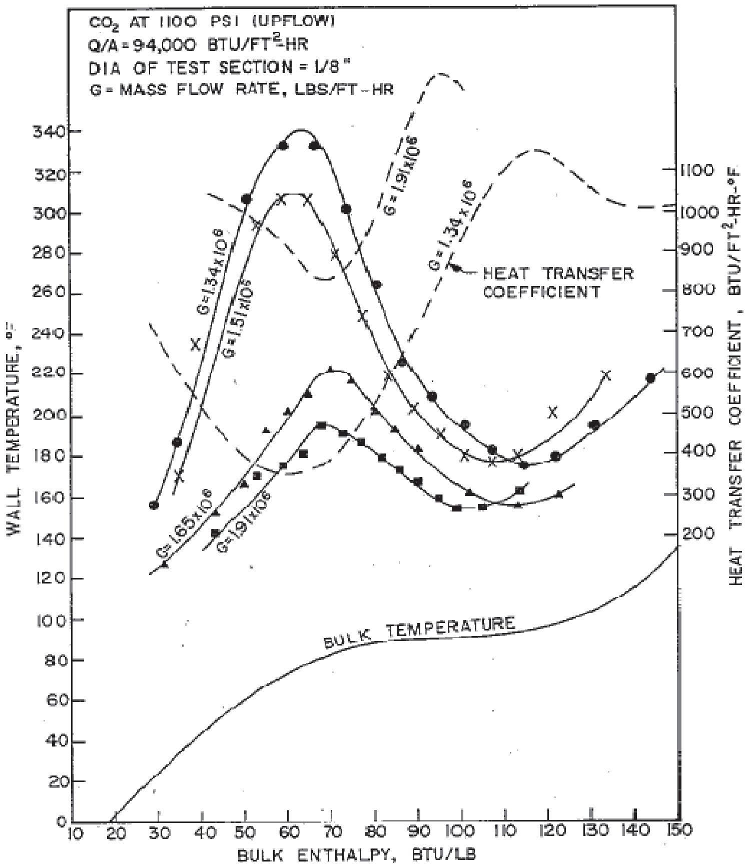


Fig. 3 Variation of the heat transfer coefficient and wall temperature profile for smaller diameter test section

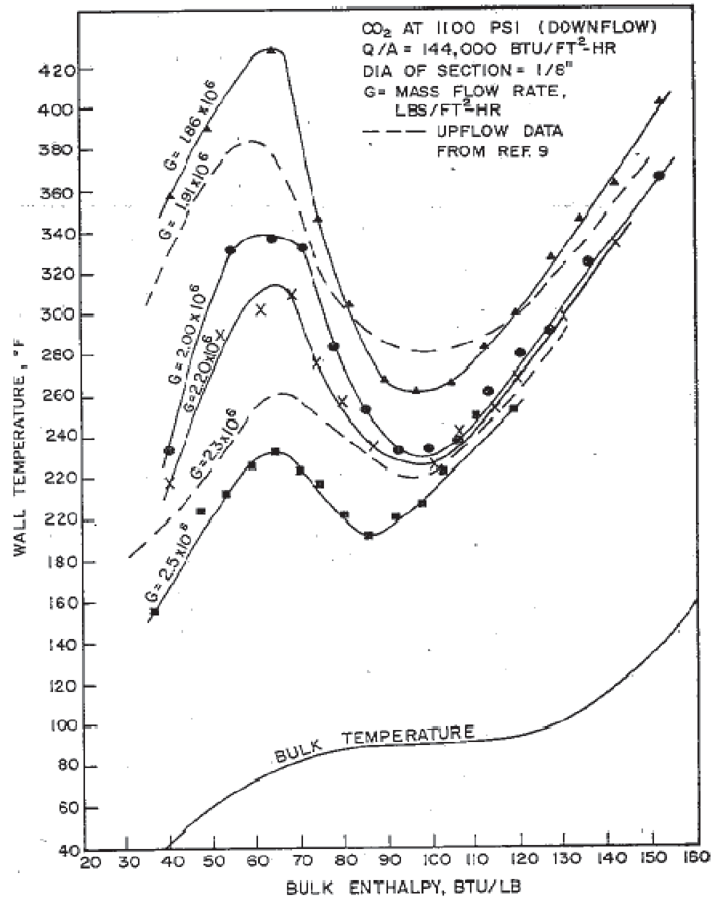


Fig. 4 Comparison between results for upflow and downflow

Dia~6.35/3.2mm
300-450kW/m^2
73e3<Re<400e3

Heat Transfer behavior was independent of flow orientation.