Characterizing Heat Transfer From Impinging Jet Flow

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Group TR9
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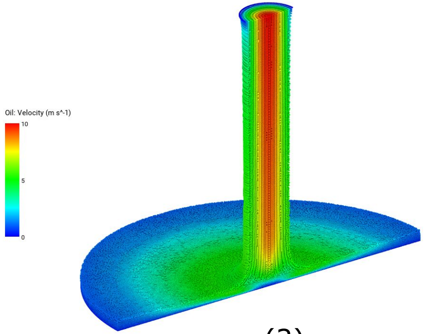
Impinging Jet Flow Paramount in Industry

Industrial Applications:[1]

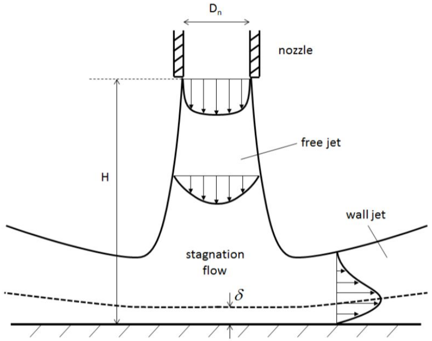
- Temperature control and core cooling in nuclear power plants and foundries
- Temperature regulation in electronic devices



(2)

Objectives:

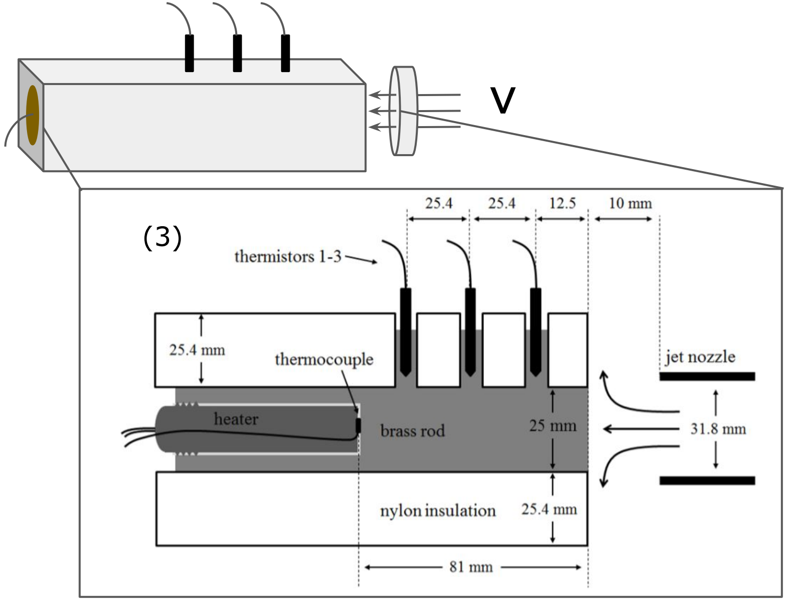
- Determine how the heat-transfer coefficient, h, scales with jet velocity, v
- Compare experimental heat transfer coefficient with theoretical values



(3)

(1) Mills A.F., *Basic Heat and Mass Transfer*; Prentice Hall Inc.: Upper Saddle River, NJ, 1999.
(2) Wendling, L., & Marathe, S.. *Impinging jet benchmark for E-motor cooling applications*. 2010. https://www.fifty2.eu/innovation/impinging-jet-benchmark-for-e-motor-cooling-applications/ (accessed March 6, 2025).
(3) Behesti-Pour, A. Katz. *Heat Transfer (HT) Revised 2025.docx*; University of California, Berkeley: Berkeley, California, 2025.

Experimental Methods Uses Brass Rod Apparatus



(3) $$h = \frac{-k_{\text{brass}}\left(\frac{dT}{dz}\right)}{(T_{\text{surface}} - T_{\text{air}})}$$

<u>Independent Variable</u>

- Jet Velocity

<u>Constants</u>

- Heater Temperature
- Air Temperature
- Fluid & Material Properties

<u>Dependent Variables</u>

- Thermistor Temperatures

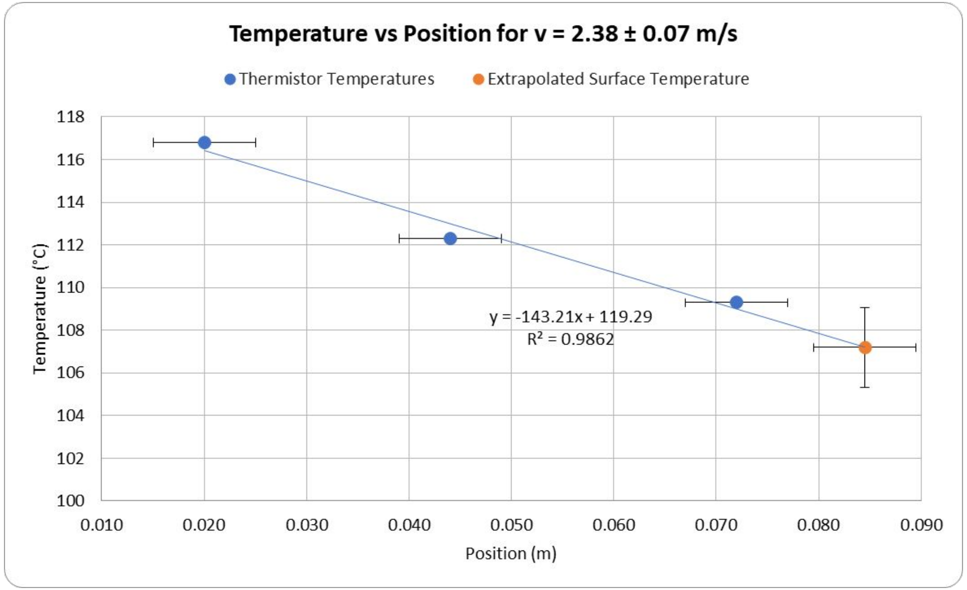
<u>Calculate</u>

- Temperature Gradient (linear)
- Surface Temperature

Experimental Convective Heat Transfer Coefficient

(3) Behesti-Pour, A. Katz. *Heat Transfer (HT) Revised 2025.docx*; University of California, Berkeley: Berkeley, California, 2025.

Determining Surface Temperature and Gradient



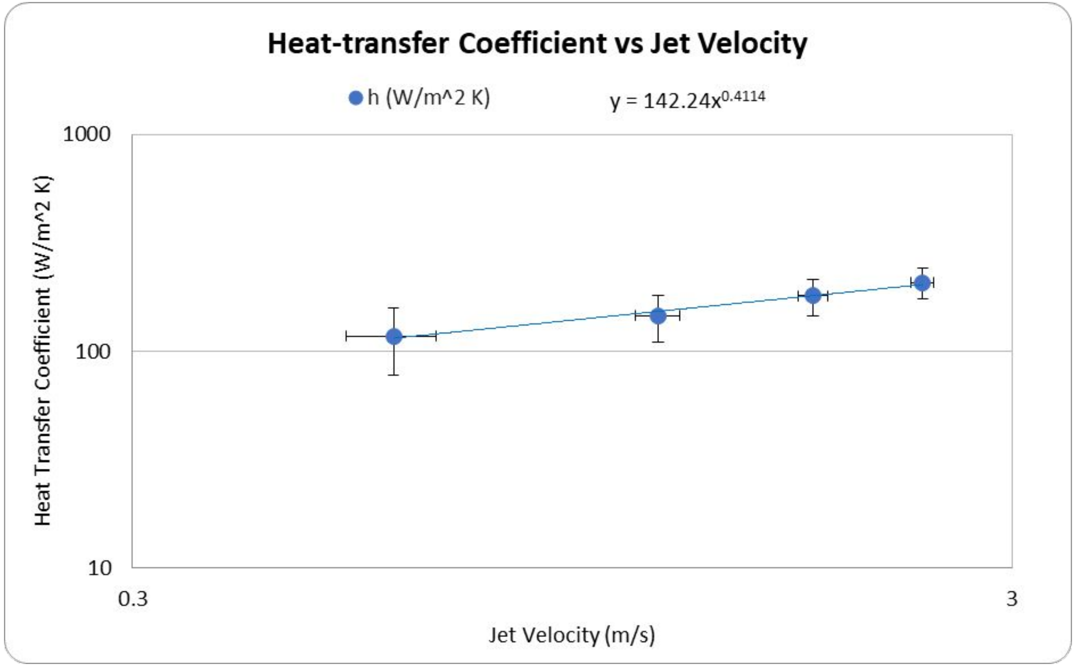
Jet Velocity (m/s)	dT/dz (C/m)	Surface Temperature (C)
2.38 ± 0.07	-140 ± 20	107.2 ± 1.9
1.78 ± 0.07	-130 ± 20	110 ± 2
1.19 ± 0.07	-110 ± 30	112 ± 2
0.59 ± 0.07	-90 ± 30	115 ± 2

Extrapolate to surface, x = 0.085 m

Convective Heat-transfer Coefficient scales with $v^{0.41}$

$$h = \frac{-k_{\text{brass}}\left(\frac{dT}{dz}\right)}{(T_{\text{surface}} - T_{\text{air}})} \quad (3)$$

Jet Velocity (m/s)	h (W/m^2K)
2.38 ± 0.07	210 ± 30
1.78 ± 0.07	180 ± 30
1.19 ± 0.07	150 ± 40
0.59 ± 0.07	120 ± 40



From Experimental Data: $h = 142v^{0.41 \pm 0.04}$

(3) Behesti-Pour, A. Katz. *Heat Transfer (HT) Revised 2025.docx*; University of California, Berkeley: Berkeley, California, 2025.

Theoretical Heat Transfer Coefficient Model Predicts $v^{0.5}$

(4)

$$\frac{Nu}{Re^{1/2}Pr^{1/3}} = a_1\left(\frac{z}{d}\right)^{-0.11}\left(1 - \frac{\left(\frac{r}{d}\right)^2\left(\frac{z}{d}\right)^{-0.2}}{b_1}\right)^{1.2}$$

$$Nu_{(r/D=0)} = a_1 Re^{1/2} Pr^{1/3}\left(\frac{z}{d}\right)^{-0.11}$$

z/d	0.5	0.75	1.0	2.0	3.0	4.0	6.0	8.0
a_1	5.3	5.1	4.6	3.6	3.2	3.2	2.9	2.3

$$Nu = \frac{hD_{\text{nozzle}}}{k_{air}} \qquad Pr_{\text{air}} = \frac{\nu}{\alpha} = \frac{c_p\mu}{k} = 0.707$$

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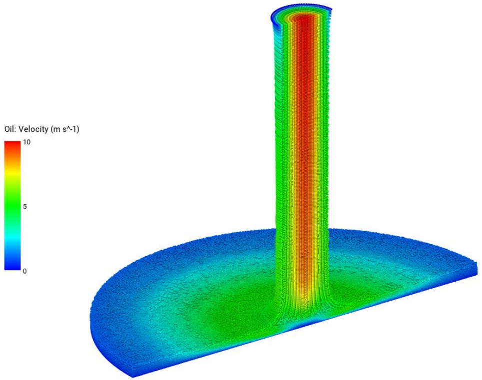


Independent Variable
- Jet Velocity

Constants
- Air Fluid Properties
- z/d
- r/d
- a_1

Dependent Variables
- Nu

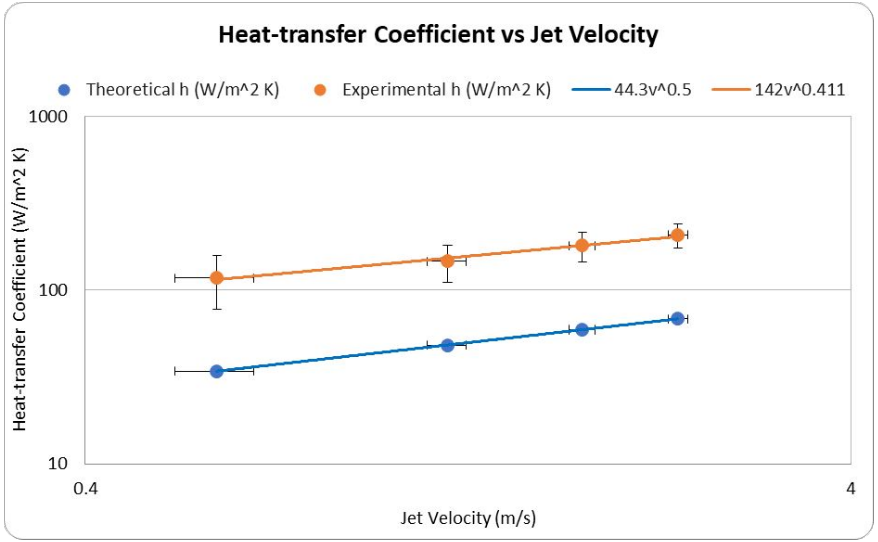
Calculate
- h

Theoretical Convective Heat Transfer Coefficient

(2) Wendling, L., & Marathe, S.. *Impinging jet benchmark for E-motor cooling applications*. 2010. https://www.fifty2.eu/innovation/impinging-jet-benchmark-for-e-motor-cooling-applications/ (accessed March 6, 2025).
(4) Katti, V.; Prabhu, S. V. Experimental Study and Theoretical Analysis of Local Heat Transfer Distribution Between Smooth Flat Surface and Impinging Air Jet From a Circular Straight Pipe Nozzle. *Int. J. Heat Mass Transfer* 2008, *51* (17–18), 4480–4495. DOI: 10.1016/j.ijheatmasstransfer.2007.08.031.

Theoretical Model Underestimates Experimental h

Jet Velocity (m/s)	Predicted h (W/m^2K)
2.38 ± 0.07	68.3 ± 0.3
1.78 ± 0.07	59.1 ± 0.4
1.19 ± 0.07	48.3 ± 0.5
0.59 ± 0.07	34.1 ± 0.7



From Experimental Data: $h = 142v^{0.41\pm0.04}$ **From Theoretical Predictions: $h = 44.3v^{0.5}$**

Thermal boundary layer thickness decreases with v, resulting in increased heat transfer

Radial Heat Transfer Coefficient larger than h

Volumetric Flow Rate (scfm)	U ($W/m^2/K$)	h ($W/m^2/K$)
4	713.31	207.82
3	701.98	180.22
2	689.11	146.10
1	649.49	117.41

$$q_{\text{heater}} = q_{\text{radial}} + q_{\text{axial}}$$

$$q_{\text{heater}} = \frac{P}{A_{\text{heater}}} \qquad q_{\text{axial}} = \frac{T_{\text{surface}} - T_{\infty}}{\frac{L}{k} + \frac{1}{h}}$$

$$U = \frac{q_{\text{heater}} - q_{\text{axial}}}{T_{\text{surface}} - T_{\text{air}}}$$

Possible reasons:

1. Insulation does not fully cover the rod.
2. Alternate conductive paths that bypass insulation.
3. Systematic errors when measuring temperatures.
4. Poor Nylon insulation efficiency at high temperatures.

Axial Heat Flux Becomes More Prominent As Flow Rate Increases

Volumetric Flow Rate (scfm)	Radial Heat Flux (%)	Axial Heat Flux (%)
4	83.22	16.78
3	84.99	15.01
2	86.74	13.26
1	87.79	12.21

Thermal boundary layer thickness decreases with v, resulting in increased heat transfer

Volumetric Flow Rate (scfm)	$q_{axial} + q_{radial}$ (W/m^2)	q_{heater} (W/m^2)	Difference (%)
4	89839.78	91953.92	2.32
3	91062.09	92724.85	1.80
2	87802.59	88950.29	1.29
1	80265.02	81041.26	0.96

The unaccounted energy loss could be due to **radiation** or **conduction losses** from structural supports.

Conclusions

1. Convective heat transfer coefficient h increases as jet velocity v increases with a scaling of $h = 142v^{0.41 \pm 0.04}$.

2. The experimental scaling, $h \propto v^{0.41}$, is slightly lower than the theoretical correlation, $h \propto v^{0.5}$.

3. Radial heat transfer coefficient U is much higher than h, suggesting poor insulation performance. $q_{radial} > q_{axial}$.

Appendix A: Error Analysis

Derived from: $$\delta y = \sqrt{\sum_{i=1}^{N} \left(\frac{\partial f}{\partial x_i}\right)^2 (\delta x_i)^2}$$

$$\Delta(T_{surface} - T_{air}) = \sqrt{(\Delta T_{surface})^2 + (\Delta T_{air})^2}$$

$$\Delta h = h\sqrt{\left(\frac{\Delta(dT/dz)}{(dT/dz)}\right)^2 + \left(\frac{\Delta(T_{surface} - T_{air})}{(T_{surface} - T_{air})}\right)^2}$$

Appendix B: Theoretical Model Concepts

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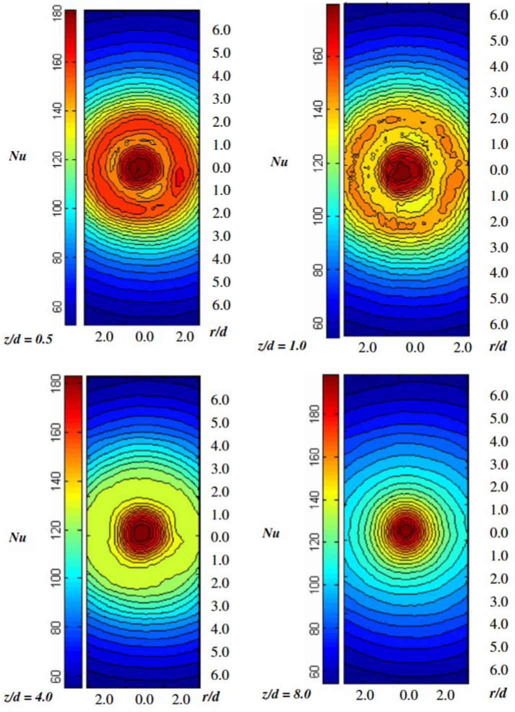

Fig. 6. Distribution of local Nusselt number at $Re = 28{,}000$ for $z/d = 0.5$, 1.0, 4.0 and 8.0.

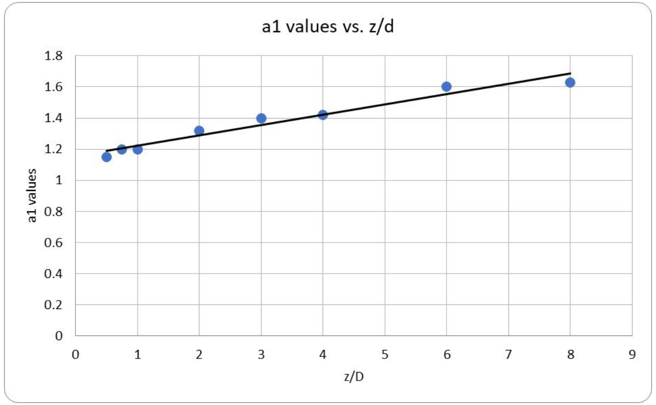

Table 1
Values of constants a_1 and b_1 for different z/d used in Eq. (9)

z/d	0.5	0.75	1.0	2.0	3.0	4.0	6.0	8.0
a_1	1.15	1.2	1.2	1.32	1.4	1.42	1.6	1.63
b_1	5.3	5.1	4.6	3.6	3.2	3.2	2.9	2.3

(4) Katti, V.; Prabhu, S. V. Experimental Study and Theoretical Analysis of Local Heat Transfer Distribution Between Smooth Flat Surface and Impinging Air Jet From a Circular Straight Pipe Nozzle. *Int. J. Heat Mass Transfer* 2008, *51* (17–18), 4480–4495. DOI: 10.1016/j.ijheatmasstransfer.2007.08.031.

Appendix C: Definition of Constants

Variable	Value	Description
k_{brass}	120 W/(m*K)	Thermal Conductivity of Brass
k_{air}	0.025 W/(m*K)	Thermal Conductivity of Air
d	0.0318 m	Diameter of Jet Nozzle
α	$2.062 * 10^{-5}$ kg/m^3	Thermal Diffusivity of Air
μ	$1.66 * 10^{-5}$ Pa*s	Kinematic Viscosity of Air
ρ	1.205 kg/m^3	Density of Air
L	0.081 m	Effective Length Along the Brass Rod
z	0.01 m	Distance Between Jet Nozzle and Brass Rod