A Comparison of Correlations for Heat Transfer from Inclined Pipes

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Abstract

A review of literature on heat transfer coefficients indicated very little work reported for cross-flow pipe arrangement at various angles of inclination. In this study forced airflow at 1.1 m/s and 2.5 m/s across 2 steel pipes of diameters 0.034m and 0.049m were examined with pipe orientation inclined at 30 and 60 degrees to the horizontal position. A comparison of the experimentally determined $N\bar{u}$ and the conventional method using existing correlations for horizontal pipes in cross-flow showed that at 30 degrees inclination, 1.1 m/s, $N\bar{u}$ values were in good agreement. However, there were large differences at 60 degrees inclination, 2.5 m/s. Comparing experimental data with the correlations of Churchill, Zhukaukas, Hilpert, Fand and Morgan showed that for 30 degrees inclination the deviation from experimental $N\bar{u}$ at 1.1 m/s ranged from 2% to 18% and 2% to 8% for the 0.034m and 0.049m pipes, respectively, while at 2.5 m/s the deviation ranged from 12 % to 31% and 20% to 41% for the 0.034m and 0.049m diameter pipes, respectively. At 60 degrees inclination the deviation from experimental $N\bar{u}$ at 1.1 m/s ranged from 19% to 45% and 27 % to 41% for the 0.034m and 0.049m pipes, respectively, while at 2.5 m/s the deviation ranged from 48% to 65% and 29% to 52% for the 0.034m and 0.049m diameter pipes, respectively.

Keywords: Convective heat transfer, Inclined pipes, Heat transfer correlations.

1. INTRODUCTION

In the study of thermodynamics the average heat transfer coefficient, $\bar{h}$, is used in calculating the convection heat transfer between a moving fluid and a solid. This is the single most important factor for evaluating convective heat loss or gain. Knowledge of $\bar{h}$ is necessary for heat transfer design and calculation and is widely used in manufacturing processes, oil and gas flow processes and air-conditioning and refrigeration systems. The heat transfer coefficient is critical for designing and developing better flow process control resulting in reduced energy consumption and enhanced energy conservation. Application of external flow forced convection heat transfer coefficient range from the design of heat exchangers and aircraft bodies to the study of forced convection over pipes.

With the continued increase in design complexity and the modernization of process plant facilities, the study of forced convection over cylindrical bodies has become an important one [1].
By the formulation of correlations, which consist of dimensionless parameters, such as *Nusselt* number (*Nu*), *Reynold's* number (Re) and *Prandtl* number (Pr), for different geometries, the values of $\bar{h}$ can be calculated without having to analyze experimental data in every possible convective heat transfer situation that occurs. Dimensionless numbers are independent of units and contain all of the fluid properties that control the physics of the situation and involve one characteristic length. It is advantageous to present data in the form of dimensionless parameters since it extends the applicability of the data. However, correlations using dimensionless numbers are developed for particular geometries and situations and are applicable within that range. Therefore, it is impractical to use correlations developed for horizontal pipes to determine the $\bar{h}$ for inclined pipes.

2. PRESENTLY USED CORRELATIONS

Presently there are many correlations to predict the heat transfer from heated vertical or horizontal pipes in both forced and natural convection situations. A review of literature on heat transfer coefficients indicated that very little experimental work has been done on inclined pipes in the recent past with little or no conclusive work reported for cross-flow pipe arrangement at various angles of inclination. Generally, for design purposes cross flow correlations for horizontal pipes are being used to determine heat transfer coefficients for inclined orientation. Few correlations exist for inclined pipes with natural convection and none exist for inclined pipes in forced convection flow. Following is a brief overview of the most common correlations that are being used for a horizontal pipe in cross-flow.

2.1 Hilpert

Hilpert [2] was one of the earliest researchers in the area of forced convection from heated pipe surfaces. He developed the correlation:

$$\bar{Nu}_D = \left(\frac{\bar{h}D}{k}\right) = C\,\mathrm{Re}_D^m\,\mathrm{Pr}^{\frac{1}{3}} \tag{1}$$

TABLE I. HILPERT'S CONSTANTS FOR FORCED CONVECTION

Re_D	C	m
0.4-4	0.981	0.33
4-40	0.911	0.385
40-4000	0.683	0.446
4000-400,000	0.193	0.618
400,000-40,000,000	0.027	0.805

where the values of *C* and *m,* are given on Table I.

Hilpert's calculations were done using integrated mean temperature values, not mean film temperature values, and with inaccurate values for the thermophysical properties of air. The thermal conductivity values of air used by Hilpert were lower (2-3%) than the most recent published results [3]. This resulted in the values of *Nusselt* number calculated by the Hilpert correlation to be higher than they should be.

2.2 Fand and Keswani

Fand and Keswani [4, 5] reviewed of the work of Hilpert and recalculated the values of the constants *C* and *m* in equation 1 using more accurate values for the thermophysical properties of air. The constants proposed by Fand and Keswani are given on Table II.

TABLE II. FAND'S CONSTANTS

Re_D	C	m
1-4	-	-
4-35	0.795	0.384
35-5000	0.583	0.471
5000-50000	0.148	0.633
50000-230000	0.0208	0.814

2.3 Zuakaukas

Another correlation proposed by Zukaukas [6] for convective heat transfer over a heated pipe was

$$Nu_f = c\,\mathrm{Re}_f^m\,\mathrm{Pr}_f^{0.37}\left(\mathrm{Pr}_f/\mathrm{Pr}_w\right)^{0.25} \qquad (2)$$

where the values of *c* and *m* are given on Table III. Except for Pr_w, all calculations were done at the mean film temperature.

TABLE III. ZUAKAUKAS' CONSTANTS

Re_D	C	m
1-40	0.76	0.4
$40\text{-}10^3$	0.52	0.5
$10^3\text{-}2(10)^5$	0.26	0.6
$2(10)^5\text{-}10^7$	0.023	0.8

2.4 Churchill and Bernstein

Churchill and Bernstein [7, 8] proposed a single comprehensive equation that covered the entire range of Re_D for which data was available, as well as a wide range of Pr. The equation was recommended for all $Re_D.Pr > 0.2$ and has the form

$$Nu_D = 0.3 + \frac{0.62\,\mathrm{Re}_D^{1/2} + \mathrm{Pr}^{1/3}}{\left[1+\left(0.4/\mathrm{Pr}\right)^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}_D}{282{,}000}\right)^{5/8}\right]^{4/5} \qquad (3)$$

This correlation was based on semi-empirical work and all properties were evaluated at the film temperature.

2.5 Morgan

Morgan [9] conducted an extensive review of literature on convection from a heated pipe and proposed the correlation

$$\overline{Nu}_D = \left(\frac{\bar{h}D}{k}\right) = C\,\mathrm{Re}_D^m\,\mathrm{Pr}^{\frac{1}{3}} \qquad (4)$$

where the values of *C* and *m* are given on Table IV.

TABLE IV. MORGAN'S CONSTANTS

Re_D	C	m
0.0001-0.004	0.437	0.0895
0.004-0.09	0.565	0.136
0.09-1	0.800	0.280
1-35	0.795	0.384
35-5000	0.583	0.471
5000-50000	0.148	0.633
50000-200000	0.0208	0.814

3. EXPERIMENTAL PROCEDURE

3.1 Test Apparatus

A low velocity circular cross-section wind tunnel was designed and built to experimentally determine $\bar{h}$ for circular pipes in cross flow arrangement at varying angles of inclination to the horizontal. The apparatus was designed to accommodate test specimen centrally across the diameter of the wind tunnel as shown in Figure 1(a). The wind tunnel test section was 1.5 m in diameter and 3 m long after which the tunnel was tapered to a diameter of 1 m to accommodate the attachment of a 1 m diameter variable speed extractor fan, Figure 1(b). In this arrangement the wind flowed transversely across the test specimen. The wind tunnel was constructed from 3 mm thick galvanized steel sheet and reinforced with an outer wooden frame.

(a)

(b)

Figure1. Photograph of test apparatus

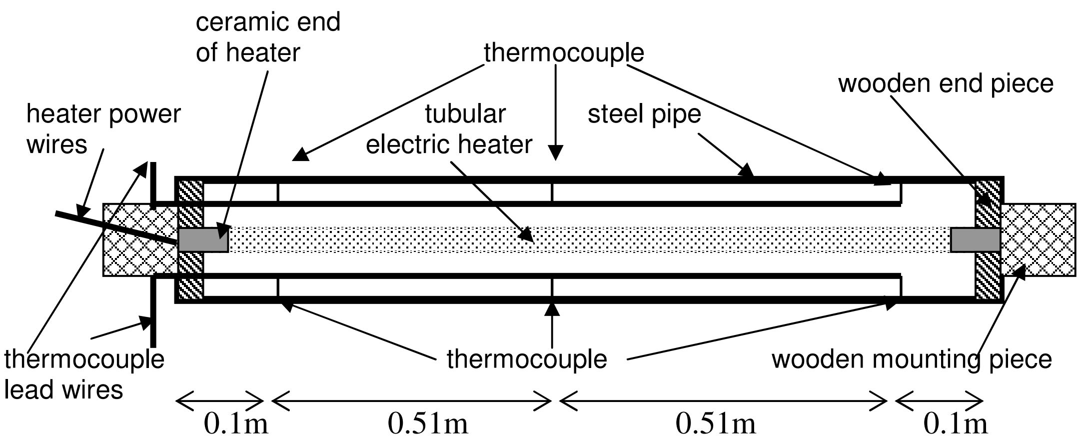

Figure 2. Schematic of test specimen

3.2 Test Specimen

Two test specimens of outer diameter 0.034 m and 0.049 m were prepared from standard schedule 40 steel pipes. The test specimens were 1.22 m long with a 1.07 m long, 0.012 m diameter electric tubular heater centrally located inside the pipes as shown in Figure 2. Six thermocouples were placed at the outer surface of the pipes. The thermocouples lead wires were passed through the annular space and the thermocouple fixed at the pipe outer surface through holes drilled at the appropriate locations. The test pieces were respectively centrally suspended and rigidly fixed with the wooden mounting pieces diagonally across the wind tunnel at a distance of 0.75 m from the leading edge. The circular section test chamber allowed the inclination angle of the test specimens to be varied easily.

3.3 Temperature Measurement

The pipe surface temperature was monitored with the Pico TC-08 data logger via K-type thermocouples. With K-type thermocouples the TC-08 has a resolution of 0.025 ^{o}C and an accuracy of 0.3% over the temperature range –120 ^{o}C to 1050 ^{o}C. The six thermocouples were strategically located on the pipe surface as show in Figure 2. To check for uniform pipe surface temperature and surface temperature stability preliminary heating tests were conducted to verify the test arrangement. Equilibrium conditions were approached within 30 minutes of heating and were verified by subsequently monitoring the six thermocouples at 5 seconds time intervals for twenty minutes. Equilibrium conditions were taken as being established when the variation in temperature readings from the six thermocouples over a twenty-minute period was within 0.75 %. The fluctuation with individual temperature readings were < 0.2 % over the equilibrium twenty-minute period. A plot of one set of temperature readings for the 0.034m and the 0.049m diameter pipes at 75^{o} orientation to the horizontal with no fan (zero air velocity) is shown on Figure 3.

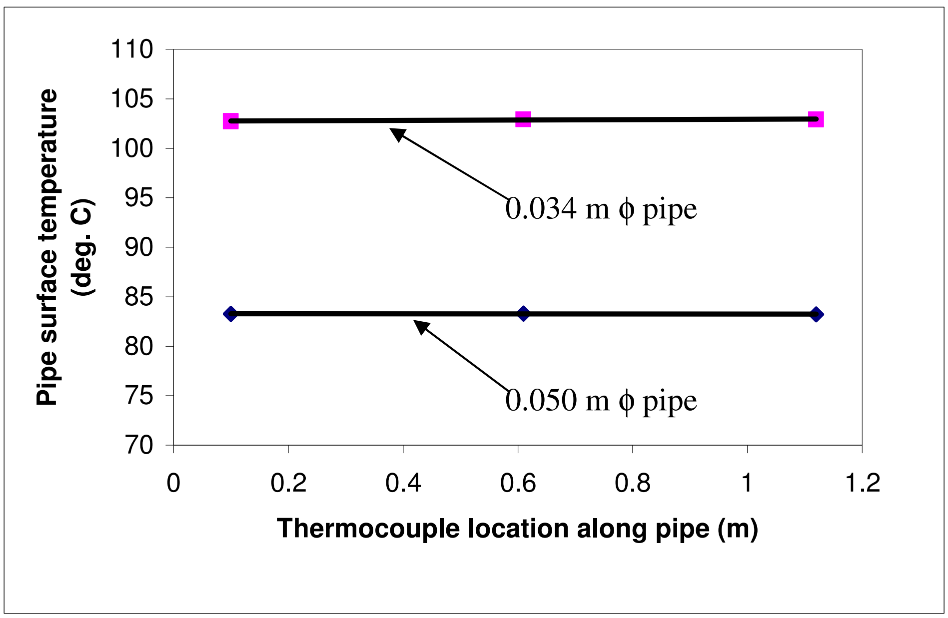

Figure 3. A plot of one set of temperature readings for pipes at 75^{o} orientation to the horizontal with no fan (zero air velocity).

3.4 Test Procedure

The speed of the extractor fan was first adjusted to provide the target air velocity in the wind tunnel. The electric heater was then powered at 90 W. The power was supplied and monitored by the MICROVIP MK1 energy analyzer. The accuracy of the primary measurements (voltage and current) of this instrument is 1%. The apparatus was continuously monitored (temperature readings were recorded at 5 s intervals) with preliminary measurements to determine uniformly heated pipe surface, airflow stability and establishment of equilibrium conditions. After equilibrium, temperature readings were recorder for ten minutes and the average values over this ten-minute period calculated as the experimental result for the test. The power was then switched off and the test pipe allowed to cool to room temperature. This procedure was repeated three times for each test variation and the average of the three test results was calculated and used to determine the heat transfer coefficient, $\bar{h}$.

3.5 Tests Conducted

For the 0.034 m and the 0.049 m diameter pipes tests were conducted at 0, 15, 30, 45, 60, 75 and 90 degrees inclination to the horizontal for air flow velocities of 0.00 m/s, 0.80 m/s, 1.35 m/s and 2.50 m/s, for every case. For each test, after establishing equilibrium conditions, data for the pipe surface temperature, wind tunnel wall temperature, ambient air temperature, wind speed across the test specimen and power to the heater were recorded. The respective experimentally determined heat transfer coefficient, $\bar{h}$, was calculated for every case and the *Nusselt* number, $N\bar{u}$, determined.

4. CALCULATIONS

The measured power input to the heater was taken as the total heat loss from the pipe surface under equilibrium conditions. The radiative heat loss component was calculated and the convective heat loss component was then determined from equation (5). The average heat transfer coefficient, $\bar{h}$, was then calculated from the convective heat transfer component of equation (5).

$$Q_{total} = Q_{conv} + Q_{rad} = \bar{h}\pi DL(T_s - T_\infty) + \varepsilon\sigma\pi DL\left(T_s^4 - T_{surr}^4\right) \qquad (5)$$

The $\bar{h}$ was then used to determine the average *Nusselt* number, $N\bar{u}$, from equation (6).

$$N\bar{u} = \frac{\bar{h}D_o}{k} \qquad (6)$$

Where $N\bar{u}$ = average *Nusselt* number
$\bar{h}$ = average heat transfer coefficient (W/m^{2}K)
k = thermal conductivity of fluid (air) (W/m.K)
D_0 = pipe outer diameter (m)

The $N\bar{u}$ was also calculated for the corresponding test conditions with the commonly used correlations of Hilpert, Fand and Keswani, Zukaukas, Churchill and Bernstein, and Morgan. The calculated results are given on Table V.

4.1 Experimental Uncertainty

The experimental $N\bar{u}$ was calculated from equation (6) using the experimentally determined *h* from equation (5). The value of *h* depends on the measured power (voltage and current) and measured temperature values. From equations (5) and (6) the relationship for the experimentally determined $N\bar{u}$ is given by equation (7).

$$N\bar{u} = \frac{Q_{total} - Q_{rad}}{\pi DL(T_s - T_\infty)}\left(\frac{1}{k}\right)D = \frac{W - \varepsilon\sigma(T_s^4 - T_{surr}^4)}{\pi DL(T_s - T_\infty)}\left(\frac{1}{k}\right) \qquad (7)$$

From the theory of uncertainty analysis [10, 11]the uncertainty in experimentally determined *Nusselt* number, $\frac{\Delta N\bar{u}}{N\bar{u}}$, from the relation in equation (7) is given by equation (8)

$$\frac{\Delta N\overline{u}}{N\overline{u}} = \frac{\Delta W}{W} + 5\frac{\Delta T_s}{T_s} + 4\frac{\Delta T_{surr}}{T_{surr}} + \frac{\Delta T_\infty}{T_\infty} = \frac{\Delta W}{W} + 10\frac{\Delta T}{T} \tag{8}$$

Also, the error associated with the power, ΔW, is

W = Voltage (V) X Current (I)

$$\frac{\Delta W}{W} = \frac{\Delta V}{V} + \frac{\Delta I}{I} \quad = 1\% + 1\%$$

Where:

W	=	Energy Meter Reading (W)
V	=	Voltage (V)
I	=	Current (A)
T_s	=	Surface Temperature (^oC)
T_{surr}	=	Surrounding Temperature (^oC)
T_∞	=	Ambient Air Temperature (^oC)
ΔNu	=	Error associated with Nu
ΔW	=	Error associated with MICROVIP MK1
ΔT	=	Error associated with Pico TC-08

For values of $\frac{\Delta W}{W}$ = 2% and $\frac{\Delta T}{T}$ = 0.3%, the uncertainty in experimentally determined *Nusselt* number is

$$\frac{\Delta N\overline{u}}{N\overline{u}} = \frac{\Delta W}{W} + 10\frac{\Delta T}{T} = 2\% + 10(0.3\%) = 5\%$$

TABLE V. CALCULATED NUSSELT NUMBER
0.034 m diameter pipe

Angle of Inclination from horizontal	Air Speed (m/s)	Experimental Results		Free Correlations			Forced Correlations				
		h Experimental	Nu Experimental	Nu Churchill	Nu Morgan	Nu Fand	Nu Churchill	Nu Zhukaukas	Nu Hilpert	Nu Morgan	Nu Fand
0	0.00	9.82	**11.43**	8.45	9.25	8.98	1.04	0.00	0.00	0.00	0.00
	0.80	13.02	**15.45**	8.08	8.86	8.61	22.21	19.60	15.47	17.83	17.83
	1.35	18.60	**22.57**	7.60	8.36	8.12	29.53	26.86	19.94	23.31	23.31
	2.50	24.55	**30.13**	7.19	7.93	7.70	41.07	38.83	32.22	32.61	32.61
15	0.00	9.44	**11.00**	8.57	9.37	9.10	1.04	0.00	0.00	0.00	0.00
	0.80	12.45	**15.07**	8.15	8.94	8.68	22.54	19.69	15.49	17.85	17.85
	1.35	18.09	**22.15**	7.64	8.40	8.16	29.54	26.90	19.94	23.30	23.30
	2.50	23.65	**29.05**	7.27	8.01	7.78	41.13	38.76	32.24	31.43	31.43
30	0.00	9.21	**10.72**	8.60	9.40	9.13	1.04	0.00	0.00	0.00	0.00
	0.80	11.68	**13.90**	8.33	9.13	8.86	22.25	19.89	15.49	17.86	17.86
	1.35	17.31	**20.99**	7.73	8.50	8.25	29.50	26.97	19.93	23.30	23.30
	2.50	22.34	**27.33**	7.33	8.07	7.84	40.91	38.85	32.07	31.40	31.40
45	0.00	9.21	**10.71**	8.58	9.39	9.12	1.04	0.00	0.00	0.00	0.00
	0.80	11.68	**13.52**	8.41	9.21	8.94	22.28	19.99	15.50	17.87	17.87
	1.35	16.85	**20.42**	7.79	8.56	8.31	29.57	26.97	23.30	23.20	23.20
	2.50	21.42	**26.30**	7.60	8.20	7.97	41.08	39.16	32.23	31.56	31.56
60	0.00	9.15	**10.63**	8.59	9.39	9.12	1.04	0.00	0.00	0.00	0.00
	0.80	11.58	**13.73**	8.33	9.12	8.86	22.20	19.85	15.51	17.82	17.82
	1.35	16.72	**20.51**	7.83	8.60	8.35	29.55	27.12	19.95	23.33	23.33
	2.50	21.22	**26.06**	7.48	8.25	7.99	41.11	39.20	32.17	31.56	31.56
75	0.00	9.12	**10.60**	8.60	9.40	9.13	1.04	0.00	0.00	0.00	0.00
	0.80	11.44	**13.55**	8.33	9.13	8.87	22.15	19.82	15.43	17.78	17.78
	1.35	16.65	**20.24**	7.84	8.62	8.32	29.56	27.16	19.96	23.33	23.33
	2.50	21.10	**25.87**	7.47	8.22	7.95	41.04	39.13	32.18	31.51	31.51
90	0.00	9.09	**10.52**	8.57	9.38	9.11	1.04	0.00	0.00	0.00	0.00
	0.80	11.57	**13.69**	8.30	9.09	8.83	22.15	19.72	15.42	17.77	17.77
	1.35	16.30	**19.83**	7.87	8.64	8.39	29.54	27.14	19.91	23.25	23.25
	2.50	20.85	**25.50**	7.47	8.21	7.98	40.92	39.02	32.06	31.43	31.43

CALCULATED NUSSELT NUMBER
0.049 m diameter pipe

Angle of Inclination from horizontal	Air Speed (m/s)	Experimental Results		Free Correlations			Forced Correlations				
		h Experimental	Nu Experimental	Nu Churchill	Nu Morgan	Nu Fand	Nu Churchill	Nu Zhukaukas	Nu Hilpert	Nu Morgan	Nu Fand
	0.00	9.46	**16.69**	10.99	11.81	11.67	1.04	0.00	0.00	0.00	0.00
	0.80	11.99	**21.45**	10.52	11.35	11.02	27.62	24.85	19.97	21.91	21.91
	1.35	17.59	**32.00**	9.73	10.55	10.25	36.74	34.01	28.22	28.51	28.51
0	2.50	23.24	**42.63**	9.16	9.97	9.68	51.32	59.22	41.79	41.24	41.24
	0.00	9.30	**16.40**	11.01	11.83	11.48	1.04	0.00	0.00	0.00	0.00
	0.80	11.92	**21.33**	10.55	11.37	11.04	27.63	24.87	18.75	21.91	21.91
	1.35	17.17	**31.23**	9.79	10.62	10.31	36.73	34.04	24.12	28.51	28.51
15	2.50	22.39	**41.11**	9.26	10.07	9.78	51.37	49.35	41.84	41.29	41.29
	0.00	9.27	**16.29**	10.98	11.80	11.46	1.04	0.00	0.00	0.00	0.00
	0.80	11.46	**20.40**	10.57	11.39	11.06	27.46	24.74	18.82	21.79	21.79
	1.35	17.09	**30.97**	9.76	10.58	10.28	36.60	33.90	24.05	28.41	28.41
30	2.50	22.12	**40.45**	9.24	10.05	9.76	51.14	49.11	41.62	41.07	41.07
	0.00	9.23	**16.22**	11.01	11.83	11.48	1.04	0.00	0.00	0.00	0.00
	0.80	11.37	**20.24**	10.60	11.43	11.09	27.48	24.78	18.72	21.80	21.80
	1.35	16.90	**30.20**	9.81	10.64	10.33	36.67	34.00	24.09	28.02	28.02
45	2.50	21.76	**39.58**	9.20	10.01	9.72	50.80	48.67	41.03	40.75	40.75
	0.00	9.16	**16.02**	10.93	11.75	11.41	1.04	0.00	0.00	0.00	0.00
	0.80	11.13	**19.69**	10.55	11.38	11.05	27.29	24.58	18.60	21.66	21.66
	1.35	16.48	**29.70**	9.77	10.59	10.28	36.37	33.62	23.91	28.24	28.24
60	2.50	21.21	**38.49**	9.23	10.04	9.75	50.67	48.67	41.18	40.63	40.63
	0.00	9.05	**15.89**	11.01	11.86	11.49	1.04	0.00	0.00	0.00	0.00
	0.80	11.11	**19.69**	10.60	11.42	11.09	27.35	24.67	18.64	21.71	21.71
	1.35	16.32	**29.60**	9.83	10.65	10.34	36.51	33.87	23.99	28.35	28.35
75	2.50	20.81	**37.88**	9.32	10.13	9.84	50.85	48.70	41.35	40.79	40.79
	0.00	8.91	**15.57**	11.01	11.83	11.49	1.04	0.00	0.00	0.00	0.00
	0.80	11.05	**19.57**	10.59	11.42	11.09	27.32	24.63	18.61	21.68	21.68
	1.35	16.25	**29.20**	9.85	10.68	10.37	36.46	33.83	23.96	25.31	25.31
90	2.50	20.39	**37.07**	9.35	10.17	9.87	50.71	48.86	41.28	40.72	40.72

5. DISCUSSION

The test apparatus designed for determination of $\overline{h}$ functioned on the fundamental principle of an energy balance when equilibrium conditions were established. Due to the lack of published data for mixed convective heat loss from inclined pipes comparison of the experimental findings with similar published work was limited. Under the circumstances, the experimentally determined $N\overline{u}$ was compared with the $N\overline{u}$ calculated from the commonly used correlations of Hilpert, Fand and Keswani, Zukaukas, Churchill and Bernstein, and Morgan on Table 5.

For all test conditions the Morgan and Fand correlations yielded the same $N\overline{u}$. A comparison of the experimentally determined $N\overline{u}$ and the conventional method using existing correlations for horizontal pipes in cross-flow showed that at 30^{o} inclination, 1.1 m/s, $N\overline{u}$ values were generally in good agreement. For this condition the 0.034 m and 0.049m diameter pipes showed maximum and minimum deviation of 18% and 1%, and 8% and 2%, respectively. The largest differences occurred with the 60^{o} inclination, 2.5 m/s. For this condition the 0.034 m and 0.049m diameter pipes showed maximum and minimum deviation of 65% and 48%, and 52% and 29%, respectively. For the 30^{o}, 2.5 m/s condition the 0.034 m and 0.049m diameter pipes showed maximum and minimum deviation of 31% and 12%, and 41% and 20%, respectively. For the 60^{o}, 1.1 m/s condition the 0.034 m and 0.049m diameter pipes showed maximum and minimum deviation of 45% and 19%, and 41% and 27%, respectively.

The results on Table 5 indicate that as air velocity increased, the differences between experimental values of $N\overline{u}$ and values obtained from the correlations for horizontal cylinders in cross-flow also increased. The experimental results showed that as the angle of inclination increased, the $N\overline{u}$ decreased, indicating reduced overall heat transfer from the surface as expected. However, the calculated values of $N\overline{u}$ from the published correlations do not show this expected trend. Therefore, the use of correlations developed for forced convection from horizontal pipes to calculate $\overline{h}$ for inclined pipes under forced airflow conditions, especially if the angle of inclination from the horizontal position is large, will result in erroneous results.

6. CONCLUSIONS

- There is an urgent need for the formulation of correlations for convective heat transfer under mixed flow conditions with inclined pipe orientation.
- The study shows that the use of horizontal pipe correlations for calculating heat loss from inclined pipe orientation yields erroneous results of significant magnitude.
- Designers and engineers need to be guided when using horizontal pipe correlations for inclines pipe calculations as there may be significant errors.

7. WORK IN PROGRESS

At present work is being done with pipes oriented at 15^{o}, 30^{o}, 45^{o}, 60^{o}, 75^{o} and 90^{o} to the horizontal. Tests are being conducted at three low speed air velocities, to determine the effect of forced airflow on the heat transfer from the surface of the inclined pipe.

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