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Transient natural convection heat transfer analyses from a horizontal cylinder

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ABSTRACT

Experiments are conducted to analyze the power-on transient natural convection heat transfer around a horizontal cylinder heated in air. The heat flux ranges from 177 W/m^2 to 2426 W/m^2 and Rayleigh number from 4.13×10^4 to 4.35×10^4. The effects of transient heat transfer data on different angular position of the thermocouple (0°, 90°, 180°) are reported. The results indicate that the transient heat transfer around the cylinder is strongly affected by the position of thermocouples. The test rig working condition is verified by comparing the steady-state natural convection heat transfer data with those obtained from the literature. It is observed that the transient heat transfer data obtained at 90° and 180° are higher than that of stagnation point (0°). Finally, based on the transient data, correlations are presented to predict the heat transfer behavior around the heated cylinder.

1. Introduction

The transient natural convection flows over cylinders have wide range of applications in engineering. These kinds of problems are encountered to analyze the heat transfer from tube heating like air conditioning systems, steam heated coils, electronic immersion heaters, etc. However, the analyses of heat transfer around horizontal cylindrical surfaces are becoming very important in present heat transfer analyses. Many studies are reported on steady-state natural convection heat transfer in air and most of the analyses are steady/linear, where the temperature profiles do not vary with time [1–10].

Some studies are available for different experimental and numerical conditions with different geometric profiles [11–26]. Most of the studies are time dependent natural convection heat transfer from vertical flat plate surfaces or enclosures surfaces. The studies by Goldstein and Briggs [11], Genceli [12], Atayilmaz [13], Sadeghipour and Kannani [14] Brown and Riley [15], Sammakia et al. [16], Gupta and Pop [17] are only a few examples of transient studies on cylinders, wires or curved surfaces. Gupta and Pop [17] extended the results of Elliott [18], by analyzing the effects of curvature of cylinder on transient natural convection and stated that the curvature increases the skin friction and heat transfer from the cylinder. Kuehn and Goldstein [1], Merkin [19], and Farouk and Guceri [20], mentioned the steady state heat transfer from horizontal cylinders and ignored the time dependent effects of heat transfer. Parsons and Mulligan [21] reported transient natural convection study from suddenly heated horizontal wire and reported a shape factor correlation to estimate the transient free convection heat transfer.

Wang et al. [22] studied numerically transient natural convection heat transfer from a circular horizontal cylinder. For a suddenly heated thin horizontal wire, Vest and Lawson [23] presented a transient study and obtained good agreement with the experimental

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Nomenclature

A_s	Heat transfer area = πDL, m^2
D	Diameter of heated cylinder, m
h	Heat transfer coefficient, $W/(m^2K)$
g	Acceleration due to gravity, 9.81 m/s^2
I	Current to heated cylinder, A
κ	Thermal conductivity of air, W/(mK)
κ_c	Thermal conductivity of cylinder, W/(mK)
L	Heated length of cylinder, m
Q	Power supplied to cylinder, W
T_a	Ambient air temperature, K
T_f	Film temperature of air, $(T_s + T_a)/2$, K
T_s	Cylinder surface temperature, K
q''	Heat flux, W/m^2

Greek Symbols

μ	Dynamic viscosity, kg/m-s
ρ	Mass density, kg/m^3
θ	Angular position of thermocouple, Deg
β	Thermal expansion co-efficient $[1/T_f]$, 1/K
ν	Kinematic viscosity, m^2/s
α	Thermal diffusivity, m^2/s

Subscripts

D	Diameter
f	Film
in	Input

Exponents

m, n	Exponents of correlation equation
C	Coefficient of correlations

Dimensionless Numbers

Bi	Biot number = hD/k_c
Fo	Fourier number = $\alpha t/D^2$
Nu_D	Nusselt number = $QD/[kA_s(T_s\text{-}T_a)]$
Gr_D	Grashof number = $g\beta(T_s\text{-}T_a)D^3/\nu^2$
Ra_D	Rayleigh number = $Gr_D \times Pr$
Pr	Prandtl number = ν/α

results. Their reported isotherm patterns for different times agreed well with the photographs of Genceli [12]. Kolsi [24] performed a numerical simulation on a fenestration with different internal louvered blinds inclination and investigated the effect of temperature period, amplitude, conductivities ratio and blinds inclination while applying unsteady and periodic temperature to the hot wall. In addition, Kolsi et al. [25] presented the effects of Rayleigh number and temperature difference on flow field pattern in a two-dimensional transient flow and observed the unstable flow at higher Rayleigh number.

It is observed in the literature that some studies are based on transient natural convection heat transfer from thin wire, thin rod, which have some limited applications and cannot meet with the real situation. In fact, the heat transfer situation changes with the circumference of the cylinder; especially near the top of the cylinder where two boundary layers merge to form the buoyant plume without flow separation, which is very important for heat transfer analysis on horizontal cylinders. In addition, insufficient heating produces no penetration in the temperature field. These important issues ignored carefully in the previous studies and there was no correlation reported to predict the heat transfer around the horizontal cylinder during transient operation. Finally, Hussein et al. [26] concluded from the comprehensive review of transient natural convection flow that numerical, experimental and three-dimensional analyses on fluid flow and heat transfer provides a scope for future research. The author reported some analyses for sudden power-on transient natural convection heat transfer and pumping power-off transient forced convection heat transfer from discrete heat sources placed in a vertical rectangular channel [27–29]. Since this basic arrangement has not been studied, and this provides motivation of the present study. However, the present experimental work investigates the cooling of a heated cylinder placed horizontally in a vertical circular duct. For the flow of air, the transient buoyancy driven flow analyzes here in the range of Rayleigh number, $4.13 \times 10^4 < Ra_D < 4.35 \times 10^4$. The flow transient occurs by sudden switch on the heater power that allows the gradual increase of cylinder surface temperature. The results show the variation of average Nusselt number with time dependent Fourier number around the cylinder. The steady-state and time dependent results are compared with the available results of similar conditions. Finally, the correlation equations are reported to predict the heat transfer data at 0°, 90° and 180° of the horizontal cylinder.

2. Experimental analyses

2.1. Set-up

The experimental setup consists of a heated cylinder of outside diameter 10 mm, length 70 mm, mounted at the top of the vertical circular duct of diameter 70 mm. Fig. 1(a) shows a real image of the experimental apparatus, Fig. 1(b) shows the schematic of the test section with the flow direction, heater location in the cylinder, upstream thermocouple position, and Fig. 1(c) shows the section view of the cylinder with thermocouple position. The cylinder, made of copper placed in such a way that the entry effect on the fluid boundary layer is negligible that ensures the fully developed flow before the cylinder. The mounting arrangement of the cylinder designed to minimize loss of heat by conduction to the wall of the duct. The heating element (heater) rated to produce 100 Watts nominally at 24 VDC into the heated cylinder. A film resistive heater is placed inside the cylinder to obtain the uniform surface temperature. A variable voltage transformer uses here to vary the supplied voltage of the heater. The conduction heat transfer toward the circumferential direction depends on the film thickness and it provides a non-uniform convective heat flux. To minimize this variation, the heat transfer rate obtained at each angular position (0°, 90°, 180°). One K-type thermocouple attached to the wall of the heated cylinder to record the surface temperature of the cylinder mid-way along the cylinder. The thermocouple flushed to the

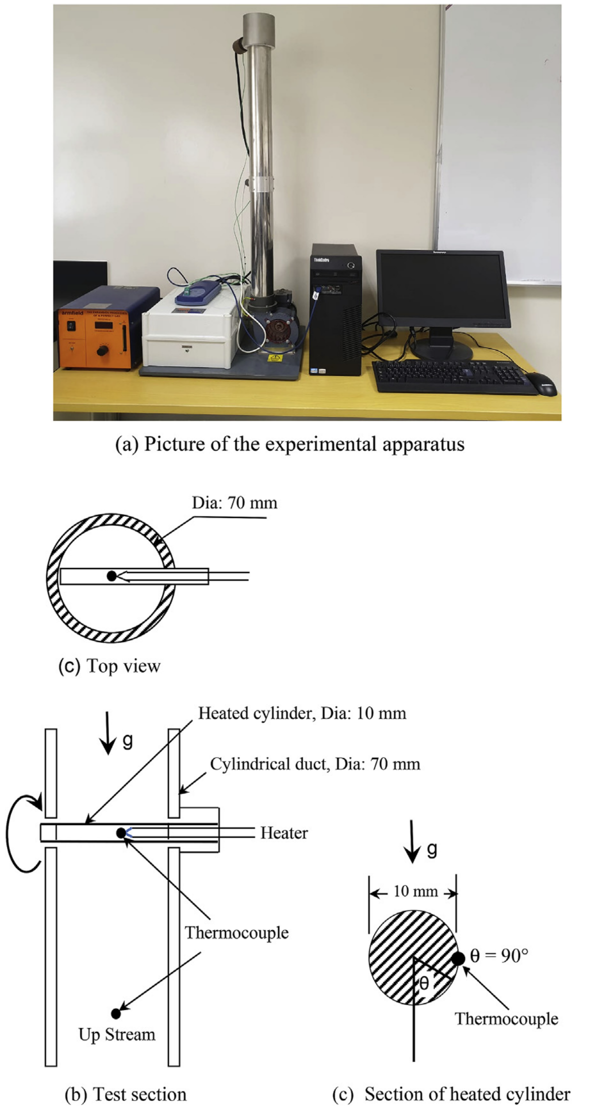

Fig. 1. Picture, schematic diagram of the test section and section of the cylinder.

cylinder surface in such a way that it will not disturb the flow. The thermocouple has the ability to measure the temperature in the range 0–133 °C with resolution, 0.1 °C. The thermocouple is an electrical device consisting of two dissimilar conductors forming electrical junctions at differing temperatures and produces a temperature-dependent voltage due to the thermoelectric effect, and this voltage interprets the temperature. To record the upstream temperature of the cylinder, one more K-type thermocouple fitted at the upstream of the cylinder. The cylinder is mounted in such a way that it can rotate to take the temperature at different angular position, as shown in Fig. 1(c). The thermocouples connect with the PC based data acquisition system to record the steady state and time dependent temperatures. The thermophysical properties of air obtained from the property table of air, based on film temperature and atmospheric pressure [30,31]. Thus, the natural convection parameters, the Nusselt number, the Fourier number, and the Rayleigh number are calculated.

2.2. Procedure

The power supplied to the heated cylinder varied and measured in the data acquisition system. However, the heated horizontal cylinder is tested at different angular positions (θ = 0°, 90°, 180°) of the thermocouples. The voltage regulator, ammeter and digital voltmeter are used to vary the input power of the heater attached to the cylinder. The apparatus is allowed to record the time dependent temperature data after the heater power is switched-on. The applied heater power was defined from the voltage and current flows through the heater. For the specific experimental condition, the heat flux was maintained constant. The heater power was varied as 0.045 Watt to 1.1 Watt for different experimental run. The Biot number varies here from 0.0025 to 0.003 that prevails the isothermal condition on the cylinder. The heated cylinder is then rotated to measure the temperatures at different angular positions, as shown in Fig. 1(b).

2.3. Uncertainty analyses

The uncertainty analyses are followed according to the procedures reported in the literature [32]. The heat loss due to conduction is minimized by the design of the equipment and assumed to be unaffected by the conduction heat transfer. The maximum surface temperature of the cylinder did not exceeded the limit of 120 °C and assumed a negligible heat loss by radiation, only the natural convection heat transfer is considered. The surface of the cylinder coated with heat resistant paint that provides an emissivity close to unity. The data were collected by a single person and by the same measuring instruments under same conditions to verify the reliability of the experimental set-up. The variation of the temperature reading was observed around 2% variations. The uncertainty of the temperature measurements was found less than 1%, provided the uncertainties of q″, Nu_D, Ra_D and Fo as 5.0%, 6.5%, 3.5% and 4.0%, respectively.

3. Results and analyses

3.1. Steady-state results

Firstly, the working conditions of the test rig was verified by steady state analyses before conducting the transient experiments. The steady state data were obtained in 2 h, after setting the heater power. During the steady-state analyses, the total number of test run was 15. The steady state results obtained for heat flux ranging from 20 W/m^2 to 500 W/m^2, the Rayleigh number from 2.1×10^3 to 1.8×10^4, the Prandtl number from 0.726 to 0.729, and the inlet temperature around 24 °C. The steady-state heat transfer data for natural convection are compared with the results of [1–3], shown in Fig. 2. It is seen that the present Nusselt number data are considerably higher of about 15% and they exhibit similar behavior and it is observed that, the present results are qualitatively similar and higher than those of the available data. It is also observed that the present steady state results are agreed well within $\theta < 90^\circ$ and agreed favorably with similar slope at $\theta > 90^\circ$ indicating the region of wider plume that increases with angle.

It can be seen from the circumferential temperature distribution in Fig. 2, at the bottom of the cylinder, stagnation point ($\theta = 0^\circ$), the fluctuation of temperatures is generally high that decreases gradually with θ, and minimum at $\theta = 180^\circ$. These fluctuations create turbulence in the buoyancy flow that leads to increase the heat transfer at $\theta = 0^\circ$, this observation is similar to the numerical observation of Qureshi and Ahmad [6]. Again, the boundary layer is very thin at bottom or stagnation point, $\theta = 0^\circ$, the boundary layer thickness increases from bottom, and reached a thicker layer at the top of the cylinder, $\theta = 180^\circ$, where the flow becomes steady laminar. This laminar flow leads to decrease the temperature gradient and heat transfer at the top of the cylinder. It can also be seen in Fig. 2, the decrease of heat transfer is not very significant up to $\theta = 90^\circ$, after that decreases sharply toward the top of the cylinder. This may be due to the development of buoyant plume above the cylinder, where two boundaries layers' merge. The plume creates insulation that resists bulk fluid flow and reduce the heat transfer. These observations are consistent with the results of similar conditions [1–3].

The comparison of the local and average Nusselt number values with the results of Kuehn & Goldstein [1], Saitoh et al. [2] and Chouikh et al. [3] are shown in Fig. 2 and the numerical data are tabulated in Table 1. It is observed that the present data compare favorably at $\theta < 120^\circ$, however at larger angles, $\theta > 120^\circ$, the data deviates from the prior reports. The discrepancy with the

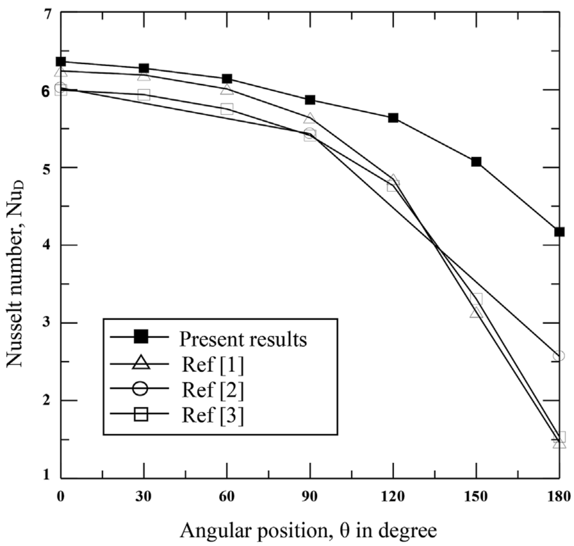

Fig. 2. Nusselt number variation for $Ra_D = 4.13 \times 10^4$.

Table 1
Comparison of present experimental results with the available results.

Ra_D		Local NuD for angular position, θ in degree							Average Nu_D
		0	30	60	90	120	150	180	
Present results	2.1×10^3	4.739	4.729	4.719	4.639	4.328	4.024	3.176	4.34
	5.0×10^3	6.836	6.684	6.396	5.896	5.850	5.037	4.018	5.82
	9.2×10^3	7.512	7.461	7.313	7.072	6.736	6.16	5.321	6.79
	1.3×10^4	7.681	7.534	7.379	7.333	7.079	6.341	5.49	6.97
	1.8×10^4	8.048	8.022	7.922	7.732	7.589	7.036	6.093	7.49
Kuehn & Goldstein [1]	10^4	6.24	6.19	6.01	5.64	4.85	3.14	1.46	4.94
Saitoh et al. [2]	10^4	5.995	5.935	5.750	5.41	4.764	3.308	1.534	4.826
Wang et al. [7]	10^4	6.03	5.98	5.8	5.56	4.87	3.32	1.5	4.86
Chouikh et al. [3]	10^4	6.023	–	–	5.433	–	–	1.539	4.831

reported results may be due to the fineness of the grid spacing, the area of the computation domain as well as the boundary conditions used in the numerical analyses. At the stagnation point, Saitoh et al. [2] observed a large discrepancy due to these reasons. Kuehn & Goldstein [1] used same boundary conditions for the outflow and inflow conditions that provided considerable error for small Rayleigh number as mentioned by Saitoh et al. [2]. Therefore, the present steady state result prevails the favorable test rig condition for transient analyses.

3.2. Transient results

The heater power-on transient natural convection tests were conducted at five different heat fluxes: 177, 500, 987, 1617, and 2426 W/m^2 and the total number of test run for transient operation was 15. At the beginning of the transient period, the temperature of the cylinder increased sharply with time and recorded in 1-s intervals. In the transient period of 190 s, the variation of Nu_D with time dependent Fo, for 177 W/m^2 is shown in Fig. 3, and the variation of average Nu_D with average Ra_D is presented in Fig. 4. The authors present results for low Ra value in the laminar range. It is noted that the curvature effect cannot be neglected at low Ra values. At high Ra value thermal boundary layer thickness decreases and most of the flow comes from the sides instead of bottom that requires high resolution near cylinder as stated by Saitoh et al. [2]. However, the experimental analyses at low Ra values can overcome the problems associated with the available numerical analyses [1–3]. The agreement between our results and those of the reported studies seems favorable and can be a reference to model an array of vertical cylinders.

It can be seen in Fig. 3, the Nu_D decreases with increasing Fo and increases with increasing Ra_D, during the transient period (190 s). A sharp decay of Nu_D is observed at the beginning of transient operation and remains steady with increasing time. The similar heat transfer variations are also obtained for the thermocouple positions of θ = 90° and 180°. This may be due to the small thickness of the thermal boundary layer that increases significantly with increasing time. Parsons and Mulligan [21] also observed this type of variation, at low Rayleigh number. It is also observed that the convection motion is sensible on the cylinder at θ = 90°, because of the high flow velocity at the side of the cylinder or θ = 90°, that leads to increase the convection motion as well as the heat transfer. The

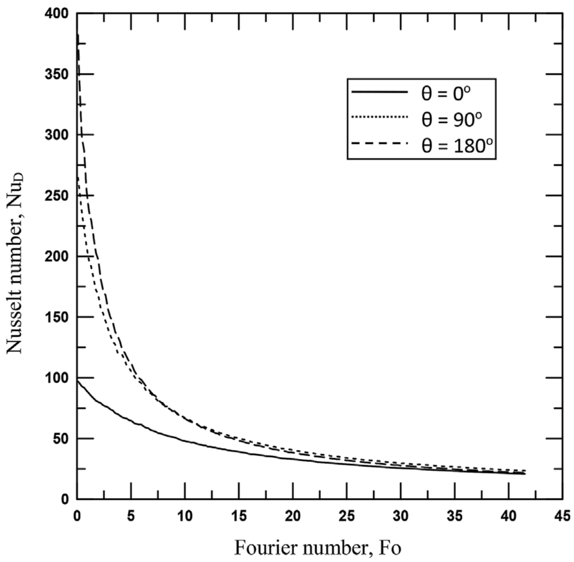

Fig. 3. Variation of Nusselt number with Fourier number at 177 W/m^2.

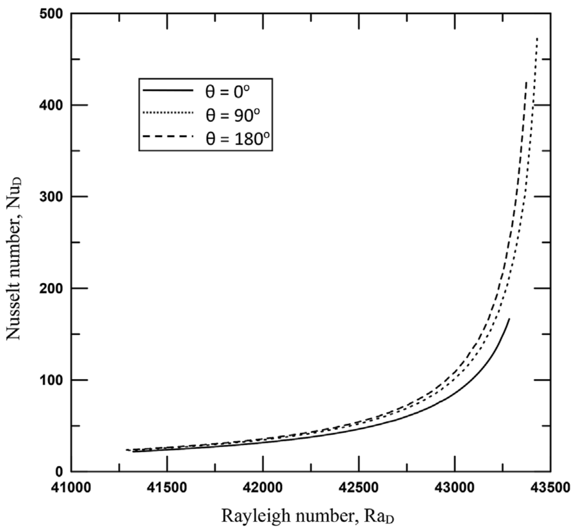

Fig. 4. Dependence of average Nusselt number on average Rayleigh number.

flow pattern is almost similar at 90° and 180°, while the fluid flow at bottom (θ = 0°) and top (θ = 180°) is almost stagnation. It is noted that the beginning of convection flow occurs at 0°, and reaches maximum at 90° when the flow and temperature fields are fully developed. This may be because of the buoyancy driven flow that increases with Ra_D near the surface of the cylinder. The variation of heat transfer is significant up to 20 s as in Fig. 3, and then the heat transfer becomes independent on time. It is also observed in Fig. 4 that at small time (around 20 s), and $Ra_D < 42500$, the Nu_D is independent of time, representing a transient conduction regime. Again, for a short period, the upward fluid motion near the cylinder surface forms circulating flow region around the cylinder. This region grows with time and approaches from bottom to the top of the cylinder. This may be due to the development of isotherms that attach strongly at the bottom of the cylinder and move toward the top of the cylinder with increasing time. However, at the beginning of transient heat transfer, the thermal boundary layer is very thin because of the conduction heat transfer and all the curves coincide with each other, as observes in Fig. 4. The thermal boundary layer grows gradually with time; the Nusselt number starts from very high value and reduces to a minimum, as in Fig. 3. This minimum value represents the transition from the conduction to the free convection heat transfer. At Rayleigh number $Ra_D > 42500$, the free convection regime is steeper than those of conduction regime.

It can also be observed from Fig. 3, the Nu_D variation with Fo, at the bottom (θ = 0°), and at the beginning of transient time, the fluctuation of temperatures at different θ is generally high and decreases with increasing time and angle (θ). From Fig. 3 the difference in heat transfer among the angles are not noticeable when approaching toward steady state condition. For analysis the transient heat transfer regime for 190 s are taken after the heater power-on. In the transient regime, the Nu_D data at 90° and 180° are about 27% and 29% higher than that of 0°, while the maximum Nu_D value obtained at 180°, around 2% higher than that of 90°. It is worth mentioning that the increase of heat transfer is significant in the first 20 s (as in Fig. 3) of transient period and the difference

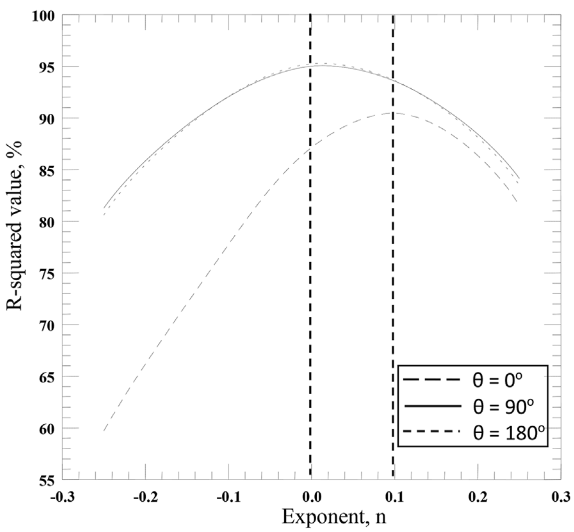

Fig. 5. The R-squared values at different exponents, n.

becomes unnoticeable as the time approaches toward steady state. Again, in Fig. 4 the effect of angles on Nu_D is not significant in $Ra_D < 42500$, while significant effect observes at higher Ra_D. The difference in Nu_D data between 0° and 90° around 27%, and between 90° and 180° around 2% are observed.

The transient heat transfer data are used here for the development of correlation equations at 0°, 90° and 180°. It was observed that transient natural convection heat transfer data influenced strongly by Nusselt number (Nu_D), Rayleigh number (Ra_D) and time dependent Fourier number (Fo) [21,22,27–29]. Thus, the correlation equation can be proposed with the relations of Nusselt number, Rayleigh number, and time dependent Fourier number as:

$$Nu_D = CFo^m(Ra_D)^n \tag{1}$$

To determine the exponent n of Ra_D, the results in Figs. 3 and 4 are re-plotted by modifying the variation of $[Nu_D/Ra_D{}^n]$ with Fo for different values of n, ranging from − 0.25 to + 0.25, and obtained the R-squared values to verify all the linear fit lines. The R-squared values for different n, are plotted in Fig. 5 for θ = 0°, 90° and 180°. It is observed that the maximum R-squared value obtained at n = 0.1 for 0°, and at n = 0 for 90° and 180° that simplifies Eq. (1) as:

$$Nu_D = CFo^m(Ra_D)^{0.1}\ \text{for}\ \theta = 0° \tag{2}$$

$$Nu_D = CFo^m(Ra_D)^0 = CFo^m\ \text{for}\ \theta = 90°\ \text{and}\ 180° \tag{3}$$

The values of C and m, the variation of $[Nu_D/Ra_D{}^n]$ with Fo are plotted again with n = 0.1 for 0° and n = 0 for 90° and 180°, in Fig. 6, where curve lines are the data points and the straight lines are the linear fits. The corresponding R-squared values are also obtained to verify the linear fit lines; the values are tabulated in Table 2. The fineness of the fit lines is ensured, as the R-squared values close to 1. As shown in Figs. 3 and 4, the significant differences in Nusselt number values are obtained at very low Fourier number and at higher Rayleigh number. However, in Fig. 4, the significant variation is observed in the entire Fourier number range. This may be due to the lower Rayleigh number value at 0° than that of 90° or 180°. In literature, as there is no correlation available for transient natural convection heat transfer from horizontal cylinder in air, the correlation Eqns (2) and (3) are compared with the reported correlation for transient heat transfer in discrete heating conditions [27], shown in Fig. 6. It is observed, the present transient data are qualitatively similar to that of the reported transient data, while the slopes of all the fit lines are very similar.

4. Practical significance

The information reported in this paper are of help in better understanding of the time dependent thermal behavior of horizontal cylinder in air, that might be applicable for the design of tube type heat exchange devices.

5. Conclusions

The experiments are conducted to investigate the sudden power-on transient natural convection heat transfer from a horizontal cylinder mounted in a vertical up flow duct. The steady state experimental results of natural convection heat transfer agreed favorably with that of the available results in the literature. During the heater power-on transient operations, the effect of heat transfer on the circumference of the cylinder investigated and observed that the average Nu_D data increased with the angle (θ), of about 27%

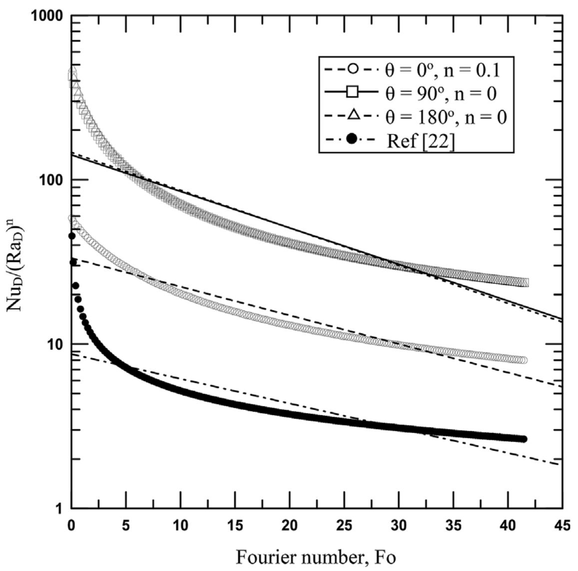

Fig. 6. Linear fit results of the correlations.

Table 2
Linear fit results of correlation equations.

Degree (θ)	n	m	C	R-squared value (%)
0°	0.1	− 0.04	3.506	90.9
90°	0	− 0.051	4.954	85.6
180°	0	− 0.053	4.986	85.2

within 0° and 90°, and 2% within 90° and 180°. All the experimental data represents here by correlation equations. In the correlation equation (Eqn. (1)), the appropriate value of the exponent, n of Ra_D, for θ = 0°, 90°, 180° are obtained, the unique values of the exponents are, n = 0.1 for θ = 0°, and n = 0 for θ = 90°, 180°. Finally, the correlations are shown in Eqns (2) and (3), and the values of C and m are tabulated in Table 2.

Appendix A. Supplementary data

Supplementary data to this article can be found online at https://doi.org/10.1016/j.csite.2019.100422.

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