Heat transfer coefficients for quasi-turbulent and turbulent flow in solar receiver tubes

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Heat Transfer Coefficients for Quasi-turbulent and Turbulent Flow in Solar Receiver Tubes

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Abstract. Several well-known correlations to determine the heat transfer coefficients of quasi-turbulent and turbulent flow in smooth tubes are available in literature. However, when the results of these correlations are compared with each other, the results vary over a considerable range. Therefore, the purpose of this study was to conduct heat transfer and pressure drop experiments in the quasi-turbulent and turbulent flow regimes and to develop an accurate heat transfer correlation. A total of 1 180 experimental data points were collected from careful experiments that were conducted ourselves using two different test section configurations. The first test section configuration consisted of a tube-in-tube test section on which the wall temperatures were obtained either indirectly with the Wilson plot method or by direct surface temperature measurements. The second test section configuration consisted of single tubes being electrically heated at a constant heat flux. Different test sections covering a range of tube diameters from 4 mm to 19 mm and a range of tube lengths from 1 m to 9.5 m, were used. Experiments were conducted from a Reynolds number of 2 445, which corresponded to the start of the quasi-turbulent flow regime, up to 220 800, which was well into the turbulent flow regime. Water, as well as different concentrations of multi-walled carbon nanotubes, were used as the test fluid, which gave a Prandtl number range of 3-10. A new correlation was developed that could estimate 95% of all the experimental data points within 10% and an average deviation of less than 5%.

INTRODUCTION

By 2030, South Africa aims to generate 42% of its electricity from renewable energy sources [1]. To reach this goal, the Department of Energy has prioritized some renewable energy technologies such as concentrating solar power (CSP) [2]. South Africa has some of the highest levels of Direct Normal Irradiance (DNI) in the world, which makes it very suitable for CSP. The overall efficiency of a CSP plant highly depends on the concentrating system and receiver tubes [3], thus it is important that sufficient design information is available to optimize the effectiveness of these tubes.

Turbulent heat transfer in circular tubes has been well researched and documented over the past 100 years and several correlations have been developed. However, there is still a large discrepancy in the agreement of these studies and correlations to each other. When comparing the different correlations, it was found that for a Prandtl number of 7, the Nusselt numbers obtained using the correlations of Petukhov [4] and Gnielinski [5] were within 5%. However, at a Reynolds Number of 10 000, the deviation between the Nusselt numbers obtained using the correlations of Sieder and Tate [6] and Petukhov [4] was more than 50%. This deviation gradually decreased to 40% at a Reynolds number of 200 000.

When analyzing the experimental data that were used to develop these turbulent heat transfer correlations, it was found that various test fluids (thus a wide range of Prandtl numbers) were used, but the experiments were only conducted up to a Reynolds number of 401 600 [7]. Furthermore, in many cases the experiments conducted with the highest Prandtl number fluids were not necessarily in the turbulent flow regime, but in the laminar flow regime (due to the significant increase in pressure drop with increasing Reynolds number when high viscosity fluids are used). To the authors' best knowledge Morris and Whitman [8] conducted turbulent experiments with the maximum Prandtl number of 276 using light motor oil. Therefore, although some correlations are valid up to Reynolds numbers of

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5×10^6 and Prandtl numbers of 10^5, these ranges were obtained by extrapolation and not using actual experimental data points.

Furthermore, it should be noted that in the period of 1922 to 1936, when the majority of the experiments, which formed the basis of the turbulent heat transfer work of many scholars in terms of improvements and refinements, were conducted, the execution of uncertainty analyses was not a requirement in scholarly journals. Therefore, the uncertainties of convective heat transfer equations in smooth tubes, which are widely published in heat transfer textbooks and used for verification and comparison studies today, are in general not readily available. Because the measuring instrumentation available today are more accurate than a century ago, it should be possible to not only conduct more accurate experiments, but also to derive a more accurate correlation with a quantified uncertainty.

The purpose of this study was thus twofold. Firstly, to take accurate heat transfer and pressure drop measurements on a smooth tube in the quasi-turbulent and turbulent flow regimes. Secondly, to develop a new correlation from this experimental data and compare it to the existing correlations and experimental data from literature.

EXPERIMENTAL SET-UP

The experimental set-ups were housed in the Clean Energy Research Group laboratory at the University of Pretoria. The details of the experimental set-ups and different test sections that were used are given in Coetzee [9] and Everts [10] and will only be briefly discussed in this paper. To conduct experiments using different boundary conditions, two different test section configurations were used. The constant surface temperature test section configuration (Fig. 1(a)) consisted of a tube-in-tube heat exchanger, on which the wall temperatures were obtained either indirectly with the Wilson plot method or by direct temperature wall measurements. Furthermore, to obtain experimental data for both heating and cooling conditions, experiments were conducted using either a hot fluid in the inner tube and a cold fluid in the annulus, or a cold fluid in the inner tube and a hot fluid in the annulus. The constant heat flux test section configuration (Fig. 1(b)) consisted of single tubes being electrically heated at a constant heat flux.

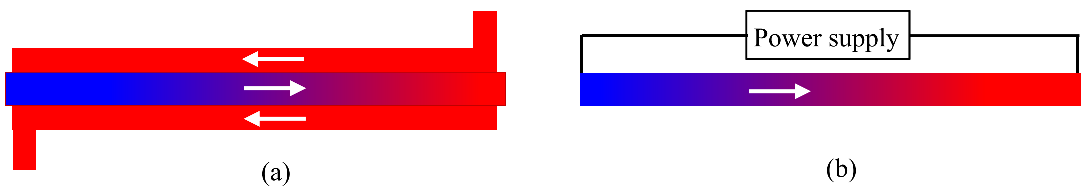

FIGURE 1. Schematic representation of (a) the constant surface temperature test section and (b) the constant heat flux test section

Different test sections covering a range of tube diameters from 4 mm to 19 mm and a range of tube lengths from 1 m to 9.5 m, were used. The surface temperatures were measured using T-type thermocouples at selected axial locations on the test sections. Depending on the test section configuration and heating method, the thermocouples were either soldered or glued onto the test sections. To measure the pressure drop, 30 mm long capillary tubes were silver soldered at each pressure tap station. A hole, which was less than 10% of the test section's inner diameter [11], was drilled through the capillary tube and the tube wall and care was taken to remove all the burrs from the inside of the test section. A bush tap with a quick release coupling was fixed to the capillary tube, and nylon tubing was used to connect the pressure taps to the differential pressure transducers. The test sections were insulated with Armaflex insulation with a thermal conductivity of 0.034 W/m.K.

A total of 1 008 tests were conducted, which covered a Reynolds number range and Prandtl number range of 2 445 – 220 818 and 3.08 – 9.97 respectively. Because the Prandtl numbers of the experimental data of this study were limited to approximately 10, high Prandtl number experimental data from literature were used to evaluate the performance of the correlation developed in this study when high Prandtl number fluids are used.

DATA REDUCTION

Two different methods were used to obtain the surface temperatures: (1) direct temperature measurements (TM) using thermocouples on the test section and (2) the Wilson plot/modified Briggs and Young method (WP) method [12-14]. The data reduction method used for the different test section configurations has been described in detail in references [9, 10, 15-23]. Therefore, only the data reduction method of the main parameters is given in this paper. The bulk fluid temperatures, T_b, were taken as the average of the measured inlet, T_i, and outlet, T_o, temperatures:

$$T_b = \frac{T_i + T_o}{2} \tag{1}$$

The properties of the test fluid (density, ρ, dynamic viscosity, μ, thermal conductivity, k, specific heat, C_p, and Prandtl number, Pr) were determined at the bulk fluid temperature.

The Reynolds numbers, Re, were calculated as

$$Re = \frac{\dot{m}D}{\mu A_c} \tag{2}$$

where $\dot{m}$ is the measured mass flow rate inside the tube, D the inner-tube diameter, μ the dynamic viscosity and A_c the cross-sectional area of the test section ($A_c = \pi/4D^2$).

After the Reynolds numbers and Nusselt numbers were calculated using either surface temperature measurements or the Wilson plot/modified Briggs and Young method, the heat transfer results could also be investigated in terms of the Colburn j-factors. This was to account for the variations in the Prandtl numbers, as well as to investigate the relationship between heat transfer and pressure drop:

$$j = \frac{Nu}{RePr^{\frac{1}{3}}} \tag{3}$$

The friction factors, f, were calculated from the mass flow rate and pressure drop measurements, ΔP, between two pressure taps, which were apart from each other a length L:

$$f = \frac{2\Delta PD}{L\rho V^2} = \frac{\Delta P\rho D^5\pi^2}{8\dot{m}^2 L} \tag{4}$$

In general in this paper, the percentage error of a measurement or calculated value was determined as %error = $|M_{exp} - M_{cor}|/M_{ref} \times 100$. When the experimental set-up and data reduction method were validated, M_{ref} was obtained from existing correlations in literature, M_{cor}. However, when the accuracies of the correlations were determined, M_{ref} was obtained from the experimental data, M_{exp}. The average percentage error was taken as the average of the absolute errors of the data points.

Surface Temperature Measurements

The average surface temperature, T_w, along a tube length, L, measured from the inlet of the test section, was calculated from the local surface temperatures, $T_w(x)$, using the trapezoidal rule:

$$T_w = \frac{1}{L}\int_0^L T_w(x)\,dx \tag{5}$$

The heat transfer coefficients, h, were determined from the following equation, because the heat flux, $\dot{q}$, surface temperature, T_w, and bulk fluid temperature, T_b, were known:

$$h = \frac{\dot{q}}{(T_w - T_b)} \tag{6}$$

The Nusselt numbers, Nu, were determined from the heat transfer coefficients as follows:

$$Nu = \frac{hD}{k} \tag{7}$$

Wilson Plot Method

The Reynolds numbers for the inner tube and annulus were calculated as follows:

$$Re_i = \frac{4\dot{m}_i}{\pi D_{ii}\mu_i} \tag{8}$$

$$Re_o = \frac{4\dot{m}_o}{\pi(D_{oi} - D_{io})\mu_o} \tag{9}$$

For the tube-in-tube test section configurations, the first subscript of the diameter, *D*, refers to the tube and the second subscript refers to the tube surface. For example, D_{oi} indicates the inner surface of the outer tube. By conducting a wide set of experiments at different mass flow rate measurements [9] for the inner stream, the Nusselt number correlations were determined as function of Reynolds number, Prandtl number and viscosity ratio in the format of the Sieder and Tate equations (Eqs. (10) and (11)) by using the modified Wilson plot method as prescribed by Briggs and Young [12].

$$Nu_i = C_i Re_i^{P_i} Pr_i^{\frac{1}{3}} \left(\frac{\mu_i}{\mu_w}\right)^{0.14} \tag{10}$$

$$Nu_o = C_o Re_o^{P_o} Pr_o^{\frac{1}{3}} \left(\frac{\mu_o}{\mu_w}\right)^{0.14} \tag{11}$$

Uncertainty Analysis

The method proposed by Dunn [24] was used to calculate the uncertainties of the parameters obtained in the data reduction. All uncertainties were calculated within the 95% confidence interval. The details of the uncertainty analysis method can be found in references [9, 10, 15-23]. The estimations of the Wilson Plot uncertainties were much more challenging than that of the surface temperature uncertainties as the linear regression analysis used in the Wilson Plot method, needs to be incorporated. The details of the Wilson Plot uncertainty calculations are given in Coetzee [9]. It was found that the Reynolds number uncertainty was less than 3% for all the experimental data, while the maximum friction factor uncertainties in the different test sections varied between 3% and 12%. The Nusselt number uncertainties of the studies that specifically focused on the turbulent flow regime [9, 15] were less than 5%. The other studies [10, 16, 17, 23] that focused more on the laminar and transitional flow regimes, but also conducted limited experiments in the turbulent flow regime had higher Nusselt number uncertainties (with specific reference to Everts [10]).

VALIDATION

The experimental set-up was validated by comparing the turbulent friction factors and Nusselt numbers with the existing correlations in literature and the results are summarized in Table 1 and Table 2. The friction factors correlated very well with the Blasius [25] correlation with an average deviation of only 1.4%. Table 1 also indicates that experimental friction factors correlated very well with the correlations of Filonenko [26] and Fang *et al.* [27], with more than 90% of the data were within 5% of the correlations and the average deviation was less than 2%. The existing friction factor correlations are therefore adequate and very accurate, however, it was necessary to conduct pressure drop experiments in this study, because the friction factors were required to obtain the relationship between pressure drop and heat transfer.

Table 2 indicates that the heat transfer results correlated very well with the correlations of Colburn [28], Gnielinski [5] and Petukhov [4], with average deviations of less than 10%. The correlations of Dittus and Boelter [29], Sieder and Tate [6] and Hausen [30] were not developed for Nusselt numbers in the quasi-turbulent flow regime, which led to increased average deviations. Therefore, although the results in general correlated well with the existing correlations in literature, the validation study confirmed the need for a single correlation that is accurate for Nusselt numbers in both the quasi-turbulent and turbulent flow regimes.

TABLE 1. Performance of the experimental friction factors of this study compared with the most prominent friction factor correlations in literature.

	Data points within error [%] ±5%	±10%	Average deviation [%]
Blasius [25] $f = 0.3125Re^{-0.25}$	100	100	1.4
Petukhov [31] $f = (0.79 \ln Re - 1.64)^{-2}$	78	99	3.1
Filonenko [26] $f = (1.8 \log Re - 1.5)^{-2}$	92	100	2.0
Fang *et al*. [27] $f = 0.25\left[\left(\log \frac{150.39}{Re^{0.98865}}\right) - \frac{152.66}{Re}\right]^{-2}$	96	100	1.7

TABLE 2. Performance of the experimental Nusselt numbers of this study compared with the most prominent heat transfer correlations in literature.

	Data points within error [%] ±10%	±20%	Average deviation [%]
Dittus and Boelter [29] $Nu = 0.023Re^{0.8}Pr^{n}$ n = 0.3 for cooling, n = 0.4 for heating	38	76	14
Colburn [28] $Nu = 0.023Re^{0.8}Pr^{1/3}$	74	99	7.0
Sieder and Tate [6] $Nu = 0.027Re^{0.8}Pr^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}$	27	68	17
Hausen [30] $Nu = 0.037(Re^{0.75} - 180)Pr^{0.42}\left[1 + \left(\frac{D}{L}\right)^{2/3}\right]\left(\frac{\mu}{\mu_w}\right)^{0.14}$	38	89	12
Petukhov [4] $Nu = \left(\frac{f}{8}\right)RePr\left[1.07 + 12.7\sqrt{\frac{f}{8}}(Pr^{2/3} - 1)\right]^{-1}$	72	89	8.5
Gnielinski [5] $Nu = \left(\frac{f}{8}\right)(Re - 1000)Pr\left[1 + \left(\frac{D}{L}\right)^{2/3}\right]\left(\frac{Pr}{Pr_w}\right)^{0.11}\left[1 + 12.7\sqrt{\frac{f}{8}}(Pr^{2/3} - 1)\right]^{-1}$	70	92	8.0

RESULTS

Everts and Meyer [20] investigated the relationship between pressure drop and heat transfer in smooth tubes in all flow regimes and found that a direct relationship between heat transfer and pressure drop existed not only in the laminar and turbulent flow regimes, but also in the transitional and quasi-turbulent flow regimes. This relationship makes it possible to obtain either the friction factors or the heat transfer coefficients when the other variable is available. Figure 2 compares the friction factors and Colburn *j*-factors as a function of Reynolds number. Although the results were obtained in test sections with different tube diameters and boundary conditions, the trends of these two parameters were similar. A difference between the results of the different heat fluxes in the laminar flow regime were found, but the difference between the results of the different heat fluxes, boundary conditions and tube diameters in the quasi-turbulent and turbulent flow regimes was negligible. This is in good agreement with the findings of Everts and Meyer [20] namely that the relationship between heat transfer and pressure drop is a function of Grashof number in the laminar flow regime and a function of Reynolds number in the other flow regimes.

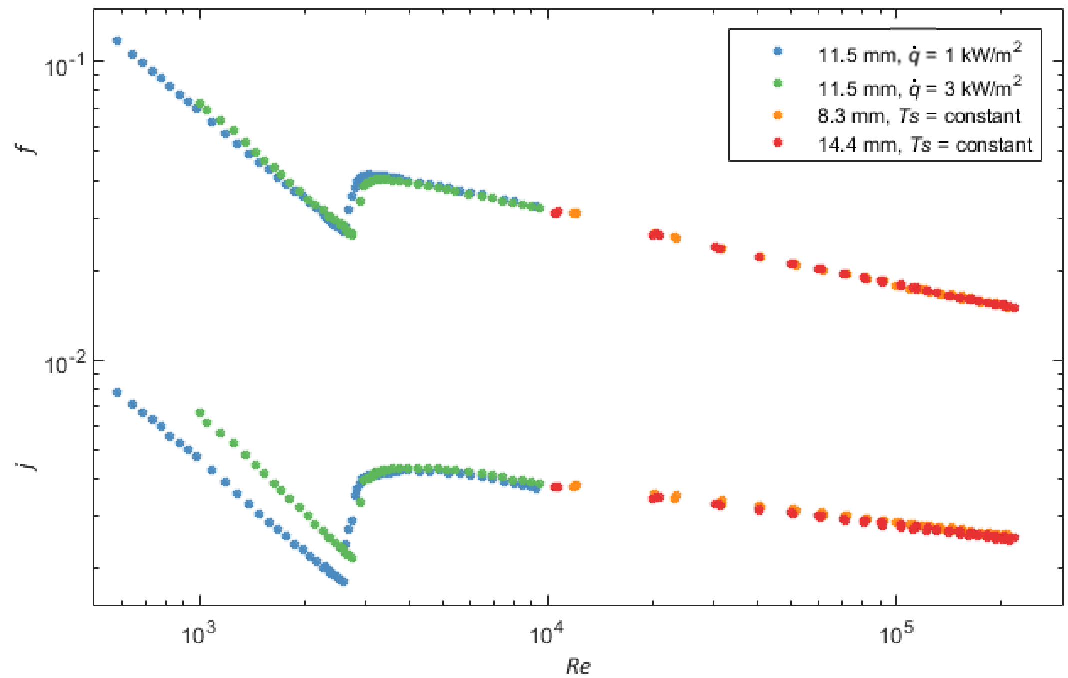

FIGURE 2. Comparison of the pressure drop and heat transfer results in terms of the friction factors and Coburn *j*-factors as function of Reynolds number.

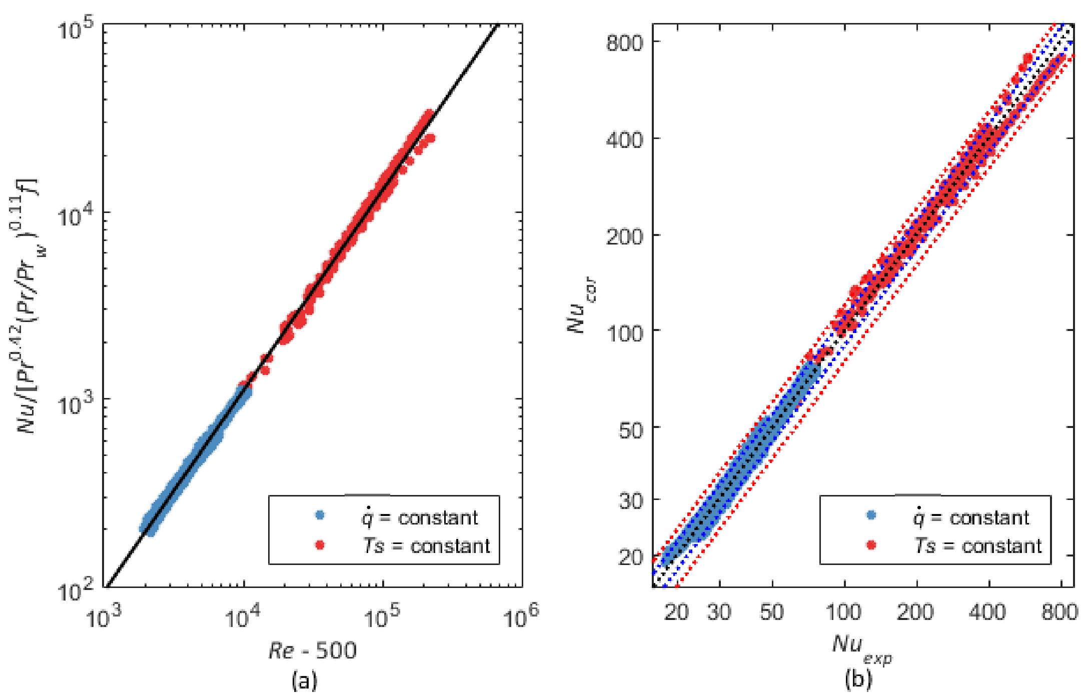

FIGURE 3. Comparison of (a) the experimental heat transfer data of this study in terms of $Nu/[Pr^{0.42}(Pr/Pr_w)^{0.11}f]$ as a function of Re – 500 and (b) deviation between Eq. (12) and the experimental data of this study.

To account for different Prandtl number fluids, variable fluid properties with temperature, and different test section dimensions and configurations, as well as making use of the relationship between heat transfer and pressure drop, the Nusselt numbers were divided by $Pr^{0.42}(Pr/Pr_w)^{0.11}f$ and plotted as a function of $Re - 500$, to specifically account for the quasi-turbulent flow regime. The following versatile Nusselt number correlation for flow in smooth tubes in the quasi-turbulent and turbulent flow regimes, was obtained by doing a power curve fit regression through the data points in Fig. 3(a):

$$Nu = 0.018Re^{-0.25}(Re - 500)^{1.07}Pr^{0.42}\left(\frac{Pr}{Pr_w}\right)^{0.11} \tag{12}$$

The correlation performed very well and Fig. 3(b) indicates that Eq. (12) was able to predict 95% of the data of this study within 10% and the average deviation was only 4.7%. Furthermore, it was able to predict experimental data in literature with a Prandtl number range of 0.47-276 and Reynolds number range of 3 000-401 600 with an average deviation of 14%. More details of how Eq. (12) can be used together with recently developed laminar and transitional correlations, in order to have a single correlation that is valid for all flow regimes, are available in Meyer et al. [32].

CONCLUSIONS

In the past century, several correlations to determine the heat transfer coefficients in smooth tubes in the turbulent flow regime were developed. Unfortunately, when these equations were developed, no uncertainty analyses were conducted. The purpose of this study was thus twofold. Firstly, to take accurate heat transfer and pressure drop measurements on a smooth tube in the quasi-turbulent and turbulent flow regimes. Secondly, to develop a new correlation from this experimental data and compare it to the existing correlations and experimental data from literature. Heat transfer and pressure drop measurements were taken using two different test section configurations. The first configuration consisted of a tube-in-tube heat exchanger to obtain a constant surface temperature boundary condition, for both heating and cooling conditions. The second test section configuration consisted of single tubes being electrically heated at a constant heat flux. Different test sections covering a range of tube diameters from 4 mm to 19 mm and a range of tube lengths from 1 m to 9.5 m, were used. A total of 1 180 experimental data points were collected from careful experiments that were conducted between Reynolds numbers of 2 445 and 220 800.

By making use of the relationship between heat transfer and pressure drop and accounting for different Prandtl number fluids, variable fluid properties with temperature, as well as different test section dimensions and configurations, a versatile Nusselt number correlation for flow in smooth tubes in the quasi-turbulent and turbulent flow regimes, was obtained. The correlation performed very well and was able to predict 95% of the data of this study within 10% and the average deviation was only 4.7%. It can therefore be concluded that this correlation is able to accurately predict the Nusselt numbers in the quasi-turbulent and turbulent flow regimes and can therefore be used to optimize the design of solar receiver tubes.

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REFERENCES

1. O.O. Craig, A.C. Brent, F. Dinter, South African Journal of Industrial Engineering, **28**, 14-27 (2017).
2. DoE, State of renewable energy in South Africa, (2015).
3. H. Benoit, L. Spreafico, D. Gauthier, G. Flamant, Renewable and Sustainably Energy Reviews, **55**, 298-315 (2016).
4. B.S. Petukhov, Advances in Heat Transfer, (Academic Press, New York, 1970), pp. 503-564.
5. V. Gnielinski, International Chemical Engineering, **16**, 359-368 (1976).
6. E.N. Sieder, G.E. Tate, Industrial Engineering Chemistry, **28**, 1429-1435 (1936).
7. J.P. Stone, C.T. Ewing, R.R. Miller, Journal of Chemical & Engineering Data, **7**, 519-525 (1962).
8. F.H. Morris, W.G. Whitman, Industrial and Engineering Chemistry, **20**, 234-240 (1928).
9. N. Coetzee, "Heat transfer coefficients of smooth tubes in the turbulent flow regime", Masters dissertation, University of Pretoria, (2015).

10. M. Everts, "Single-phase mixed convection of developing and fully developed flow in smooth horizontal circular tubes in the laminar, transitional, quasi-turbulent and turbulent flow regimes", PhD thesis, University of Pretoria, (2017).
11. R. Rayle, Influence of orifice geometry on static pressure measurements, (American Society of Mechanical Engineers, 1959).
12. D.E. Briggs, E.H. Young, Chemical Engineering Symposium Series, **92**, 35-45 (1969).
13. R.K. Shah, "Assessment of the modified Wilson Plot techniques for obtaining heat exchanger design data", Proceedings of the 9th International Heat Transfer Conference,(1990), pp. 51-56.
14. E.E. Wilson, ASME Journal of Heat Transfer, **37**, 47-82 (1915).
15. M. Steyn, J.P. Meyer, "Heat transfer coefficients for tubes in the turbulent single phase flow regime with a focus on uncertainty",15th International Heat Transfer Conference (IHTC-15),(2014), pp.
16. J.P. Meyer, T.J. McKrell, K. Grote, Int. J. Heat Mass Transf., **58**, 597-609 (2013).
17. J.P. Meyer, S.M. Abolarin, Int. J. Heat Mass Transf., **117**, 11-29 (2018).
18. J.P. Meyer, M. Everts, Int. J. Heat Mass Transf., **117**, 1251-1273 (2018).
19. M. Everts, J.P. Meyer, Int. J. Heat Mass Transf., **117**, 1331-1351 (2018).
20. M. Everts, J.P. Meyer, Int. J. Heat Mass Transf., **117**, 1231-1250 (2018).
21. M. Everts, J.P. Meyer, Int. J. Heat Mass Transf., **117**, 1274-1290 (2018).
22. J.P. Meyer, M. Everts, A.T.C. Hall, F.A. Mulock Houwer, M. Joubert, L.M.J. Pallent, E.S. Vause, Int. J. Heat Mass Transf., **118**, 257-274 (2018).
23. A.I. Bashir, J.P. Meyer, "Experimental investigation of convective heat transfer in the transitional flow regime of an inclined smooth tube", Proceedings of the 16th International Heat Transfer Conference (IHTC16),(2018), pp. 3127-3133.
24. P.F. Dunn, Measurement and Data Analysis for Engineering and Science, 2nd ed., (CRC Press, United States of America, 2010).
25. F.M. White, Fluid Mechanics, 6th ed., (McGraw-Hill, Singapore, 2009).
26. G.K. Filonenko, Teploenergetika, **1**, 40-44 (1954).
27. X. Fang, Y. Xu, Z. Zhou, Nuclear Engineering and Design, **241**, 897-902 (2011).
28. A.P. Colburn, Transactions of American Institute of Chemical Engineers, **29**, 174-210 (1933).
29. F. Dittus, L.M.K. Boelter, University of California Publications in Engineering, **2**, 443-461 (1930).
30. H. Hausen, Heat transfer in counterflow, parallel-flow, and cross-flow, (McGraw-Hill, New York, 1983).
31. Y.A. Cengel, A.J. Ghajar, Heat and Mass Transfer: Fundamentals and Applications, 5th ed., (McGraw-Hill, 2015).
32. J.P. Meyer, M. Everts, N. Coetzee, K. Grote, M. Steyn, International Communications in Heat and Mass Transfer, **ICHMT4246, https://doi.org/10.1016/j.icheatmasstransfer.2019.03.016, Accepted 1 March 2019**, (2019).