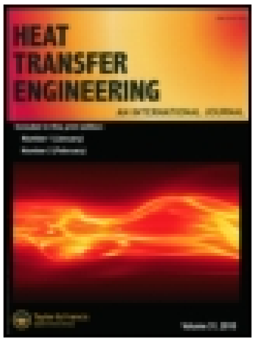

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Calculation of the Heat Transfer Coefficient for Laminar Flow in Pipes in Practical Engineering Applications

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Calculation of the Heat Transfer Coefficient for Laminar Flow in Pipes in Practical Engineering Applications

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ABSTRACT

Paper presents an analysis of existing correlations for laminar liquid flow and heat transfer in tubes based on the published theoretical research and experimental data. Considering the relatively large deviation of existing correlations from the experimental data, the novel correlation for laminar heat transfer in tubes is proposed. The new correlation covers large range of tube diameters ranging from micro–scale level 125.4 μm to conventional diameter 20.8 mm, Graetz numbers up to 6500 and fluid to wall viscosity ratio 0.0048 – 11.7. Correlation ratio of the newly proposed relationship for a total number of 390 experimental runs is 96.6% and standard deviation is 16.2%. Moreover, correlation covers both the hydraulically and thermally undeveloped and developed flows and all cases of boundary conditions that can be met in industrial applications.

INTRODUCTION

Proper design of thermal systems (equipment, pipelines, etc.) is, among many other parameters, significantly dependent on using the suitable correlation for estimation of heat transfer parameters such as heat transfer coefficient or Nusselt number. If inadequate correlation is applied in the design phase it may cause over or under estimation of capacity of heat transfer units (heat exchangers, process furnaces, etc.) or complete process systems.

Heat transfer for laminar flow in pipes/tubes is one of the most frequently discussed problems because contemporary calculation technology allows the usage of computational fluid dynamics (CFD) software packages for solving the fundamental equations for fluid flow and heat transfer. Also, special attention is given to the two cases that can be solved both numerically and analytically: constant wall temperature (t_w = const) and constant heat flux (q = const). A series of papers can be found that theoretically analyze cases of hydrodynamically and thermally developed and undeveloped flows of fluids.

Hydrodynamically fully developed flow occurs when pipe length is greater than a critical value $L > L_{fd,h}$. The limit for the laminar shear stress on the pipe wall that falls within 1% of the value for fully developed flow is $L_{fd,h} / D_i > 0.0575 \cdot \mathrm{Re}$. Thermal entry length $L_{fd,t}$ can arbitrarily be defined as the distance from the entrance section where Nusselt number takes a constant value, to within a specified accuracy. Taking 1% accuracy, it was found that the reduced length of the thermal initial segment is defined as $L_{fd,t}/D_i > 0.055 \cdot \mathrm{Re} \cdot \mathrm{Pr}$. For known values of Re the thermal entry length depends only on Prandtl number. In case of liquid metals when Prandtl number is very small (Pr = 0.005 – 0.05) $L_{fd,t}$ is not greater than few D_i. For gases, when $\mathrm{Pr} \approx 1$, entrance region is $L_{fd,t} \approx 100 \cdot D_i$. For typical Newtonian fluids (water, mineral oils, alcohols, etc.) Prandtl number has greater values (Pr = 1 – 100 or more) and entrance length might be greater than few hundreds or even few thousands of tube diameters.

Both heating and cooling of fluid flowing in pipe produce secondary flow due to difference of fluid density near the wall with respect to bulk density. This is the case of mixed convection (forced convection accompanied by free convection) which is studied in more than a few papers; for example [1] and [2] provide flow maps for evaluating the effect of free convection on local heat transfer coefficient. It can be deducted that in present time the influence of mixed convection on heat transfer is not analyzed or understood completely, and further research is expected in this field.

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In a majority of industrial applications analytically obtained solutions for determination of heat transfer coefficient are not suitable because heat transfer occurs in thermally and hydraulically developing zone:

- tubular heat exchangers (shell–and–tube, double pipe, finned tube, etc.);
- tube bundles or coils immersed in fluid (batch heat exchangers, tanks, etc.);
- pipelines (heat losses for various types of isolated or non–isolated pipes), etc.

In heat exchangers heat transfer at t_w = const. is approximately fulfilled only in cases of one–component constant pressure condensation of hot fluid ($R \to 0$) or boiling of cold fluid ($R \to \infty$) on the outer tube surface. Heat transfer at q = const. is fulfilled in heat exchangers with $R = 1$ which is a very rare case, or when a pipeline is heated with electro–resistant wire. Other cases of cooling and heating of fluids flowing in tubes are not covered by the theoretical analysis.

Moreover, vast quantity of calculations has to be done in a short period of time. Mathematical modeling and solving of practical engineering problems using CFD software requires a relatively long time, so the preference is always given to simple (easy to use) correlations that can provide reliable solutions. Such correlations are presented in handbooks, like [3-5], but without any serious comments and limitations on their usage.

The scope of this research is to analyze the suitability of the existing correlations based on the statistical criteria and published experimental data. Moreover the authors have wanted to obtain the new correlation, in a simple form suitable for engineering calculations, that can be used for estimation of Nusselt number in laminar fluid flow for all previously mentioned cases and which is independent of hydrodynamic and thermal boundary conditions on tube walls.

REVIEW OF EXPERIMENTAL AND THEORETICAL RESEARCH

This brief and chronologically sorted review includes most valuable experimental data and frequently cited equations. More correlations can be found in [6] and other review papers.

Theoretical research

Hydrodynamically and thermally developed flow

There are two analytical solutions for evaluation of the heat transfer coefficient for cases of hydrodynamically ($L_{fd,h} / D_i > 0.0575 \cdot \mathrm{Re}$) and thermally fully developed flow ($L_{fd,t}/D_i > 0.055 \cdot \mathrm{Re} \cdot \mathrm{Pr}$) independent on the Reynolds and Prandtl numbers [7]:

- $\mathrm{Nu}_{fd} = 3.657$ for constant wall temperature (t_w = const);
- $\mathrm{Nu}_{fd} = 4.364$ for constant tube wall heat flux (q = const).

Hydrodynamically developed and thermally undeveloped flow

On the basis of local Nusselt number determined by Petukhov [8], for hydrodynamically developed and thermally undeveloped flow with constant wall temperature, mean Nusselt number can be obtained in the form of

$$\mathrm{Nu} = \begin{cases} 3.657 + 0.2355 \cdot \mathrm{Gz} \cdot \int_0^{\frac{1}{\mathrm{Gz}}} \exp\left(-57.2 \cdot L\right) \cdot L^{-0.488} \cdot \mathrm{d}L & \text{for Gz} \le 10^3 \\ 1.615 \cdot \mathrm{Gz}^{1/3} - 1.7 & \text{for Gz} > 10^4 \end{cases} \quad (1.a)$$

The gap between Gz = 10^3 and Gz = 10^4 can be overcome with simple correlation

$$\mathrm{Nu} = 2.193 \cdot \mathrm{Gz}^{0,295} \quad (1.b)$$

For hydrodynamically developed and thermally undeveloped flow, with constant heat flux, in a similar manner, using the local values from [9], the following equation can be used to calculate the mean Nusselt number

$$\mathrm{Nu}=\begin{cases}4.364+0.263\cdot \mathrm{Gz}\cdot \int_0^{\frac{1}{\mathrm{Gz}}} \exp\left(-41\cdot L\right)\cdot L^{-0,506}\cdot \mathrm{d}L & \text{for } \mathrm{Gz}\leq 667\\ 1.953\cdot \mathrm{Gz}^{1/3}-0.5 & \text{for } 667<\mathrm{Gz}\leq 2\cdot 10^{4}\\ 1.953\cdot \mathrm{Gz}^{1/3}-1.0 & \text{for } \mathrm{Gz}>2\cdot 10^{4}\end{cases} \tag{2}$$

Hydrodynamically and thermally developing flow

For hydrodynamically and thermally developing fluid flow, after theoretical analysis, Stephan [10] gave the correlation for constant wall temperature:

$$\mathrm{Nu}=3.657+\frac{0.0677\cdot \mathrm{Gz}^{1.33}}{1+0.1\cdot \mathrm{Pr}^{0.17}\cdot \mathrm{Gz}^{0.83}} \tag{3}$$

and Hausen [11] provided the following one:

$$\mathrm{Nu}=3.66+\frac{0.19\cdot \mathrm{Gz}^{0.8}}{1+0.117\cdot \mathrm{Gz}^{0.467}} \tag{4}$$

In [8] and [12] authors, for hydrodynamically and thermally developing fluid flow, provide theoretical local values of Nu as a function of tube length. In order to obtain mean Nu the authors have used

$$\mathrm{Nu}=\mathrm{Gz}\cdot \int_0^{\frac{1}{\mathrm{Gz}}} \mathrm{Nu}_{loc}\left(L\right)\cdot \mathrm{d}L \tag{5}$$

and after statistical analysis, they have found the following equations in the form of Stephan’s equation (3):

- for constant wall temperature

$$Nu = 3.657 + \frac{0.06 \cdot Gz^{1.117}}{1 + 0.031 \cdot Pr^{0.08} \cdot Gz^{0.779}} \tag{6}$$

- for constant heat flux

$$Nu = 4.364 + \frac{0.055 \cdot Gz^{1.709}}{1 + 0.046 \cdot Pr^{0.078} \cdot Gz^{1.277}} \tag{7}$$

Shome and Jensen [13] have developed correlations for simultaneously developing laminar heat transfer and fluid flow through the tubes based on the numerically obtained results. The local and average Nusselt numbers and friction factors were correlated in the region very close to the entrance. Correlations cover the range of Pr = 0.7 – 500 and involve the non–dimensional axial distance for q = const and t_w = const as well as intermediary cases. Their analysis neglects effects which could be significant in real flows, such as tube wall axial conduction and property variation.

Experimental research

Morris and Whitman [14] conducted an experimental research on heat transfer and fluid flow through double pipe heat exchanger with water and three different types of petroleum oil. The fluids were heated by steam or cooled by water in similar conditions. The majority of data was gathered for turbulent fluid flow, but there are some results measured for laminar regime. The experimental data are correlated in two stages through diagrams $Nu/Pr^{0.37} = f(Re)$ for cooling and heating of fluids. Besides, no information are provided regarding the influence of the tube length or end effects.

Lawrence and Sherwood [15] made an experimental analysis on determining the influence of the heating length on heat transfer coefficient for tube flow of water heated by the steam. It was concluded that a tube length has negligible effect on heat transfer coefficient. Several equations are proposed to correlate the experimental data. Moreover, the experimental data are compared with correlations proposed by other authors. The following correlation is proposed in this report:

$$\mathrm{Nu} = 450 \cdot \mathrm{Re}^{0.7} \cdot \mathrm{Pr}^{0.5} \tag{8}$$

Kirkbride and McCabe [16] performed the experimental research on heat transfer and fluid flow of water and oil inside a straight tube heated by electric heater. New equation is derived, based also on the data from some other sources, which was also compared with the Nusselt – Grober theory of heat transfer of fluids in viscous flow. Based on the experimental data the following correlation is derived:

$$\mathrm{Nu} = 3.65 + \frac{0.0065}{\mathrm{Gz}} + \frac{0.513}{\mathrm{Gz}^{0.454}} \tag{9}$$

Sherwood et al. [17] performed analysis on the influence of tube length on heat transfer coefficient for oil heated by steam. For the laminar flow regime the experimental data are well correlated with empirical curve of Drew, Hogan and McAdams. Moreover as the oil velocity increases, the sudden rise of the outlet temperature indicates the critical Re. The experimental data were graphically correlated.

Sherwood and Petrie [18] analyzed experimentally the heat transfer and fluid flow of several fluids (water, oil, kerosene, acetone, benzene, alcohol) heated by steam or hot water. The results obtained from data reduction fit well with Dittus – Boelter empirical equation for case of heating:

$$\mathrm{Nu} = 0.024 \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{0.4} \tag{10}$$

Sieder and Tate [19] presented a correlation based on experimental data for heating and cooling of several fluids (three types of petroleum fractions) in horizontal tubes. The Sieder and Tate correlation is:

$$\mathrm{Nu} = 1.86 \cdot \mathrm{Gz}^{1/3} \cdot (\mu / \mu_w)^{0.14} \tag{11}$$

and might be applied for: $\mathrm{Gz}^{1/3} \cdot (\mu / \mu_w)^{0.14} \geq 2$, $0.0044 < \mu / \mu_w < 9.75$ and Pr < 12000.

Analyzing the experimental setup used in [19] it might be concluded that the fluid flow was thermally and hydrodynamically developing. Besides, the correlation was compared with experimental data obtained by other researchers that concluded that Sieder–Tate correlation is useful only for relatively short tubes; for example Kern [20] states that correlation (11) “gave maximum mean deviation of approximately ±12% from Re = 100 to 2100 except for water”.

Kupper [21] performed the experimental measurements of combined free and forced convection of water in horizontal tubes with constant heat flux as the boundary condition. Two new correlations have been proposed based on the obtained experimental data. The first one correlates the 55% of data with deviation of ± 10% in fully developed region. On the other hand the second one has the better accuracy covering the 68% of data with deviation of ± 10%. Moreover it was concluded that at higher Grashof numbers the free convection effects increase the Nusselt number. The following correlation is proposed that covers the 68% of the experimental data:

$$\mathrm{Nu} = 2.41 + 0.082 \cdot (\mathrm{Re} \cdot \mathrm{Ra})^{1/5} \cdot \mathrm{Pr}^{1/3} \quad (12)$$

Lelea et al. [22] made an experimental research on microtube heat transfer and fluid flow with inner diameters in the range D_i = 125.4 μm, D_i = 300 μm and D_i = 500 μm. A distilled water was used as the working fluid and Joule heating was applied on outer wall of the tube (q_w = const.). The experimental results have been compared both with theoretical predictions from literature and results obtained by numerical modeling of the experiment. The experimental results of microtube flow and heat transfer characteristics confirms that, including the entrance effects, the conventional or classical theories are applicable for water flow through microtube of above mentioned sizes. In other words it means that results for Nu confirmed that macro scale theory might be applied also at the micro scale level for the diameter range presented above.

Micro-scale cooling was discussed in [23] through the simulations for hydrodynamically and thermally developing water flow and it was concluded that the variation of physical properties induces radially inward flow that sharpens the axial velocity profile and decreases Nusselt number compared to constant property solution.

Heris et al. [24] conducted the research on experimental apparatus with constant wall temperature: double pipe heat exchanger heated with steam that condensed on the inner tube. The test chamber was 1 m long with copper tube with inner diameter of 5 mm. Test fluid was distilled water and experiments were conducted in range Gz = 11.7 – 20.8.

RESULTS AND DISCUSSION

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The scope of the authors' effort was to derive a single equation that covers all of the available experimental data taken from the literature sources cited in this paper [14]–[22]. The brief look at Gz vs. Nu for measurement given in [18] showed total disagreement with rest of the experimental data so the authors have disregarded the database from [18] in further analysis. Data from [14-17,19,21,22,24] cover the laminar fluid flow heat transfer for:

- constant temperature of the tube wall t_w = const. [17,24]
- constant heat flux q = const. [14,16,22]
- as well as the cooling/heating cases of the fluid in the heat exchangers for $R \neq 1$ [15,19,21].

In aforementioned experiments the liquids (water and various oils) were used in the the following parameters range: Re = 3.35 – 1990, Pr = 4.65 – 12100, μ / μ_w = 0.0048 – 11.7, Gz = 1.4 – 6500, Gr < 30000, L / D_i = 13 – 390 (with the exception of μ_w all physical properties are obtained based on mean fluid temperature). The total number of the experimental regimes is 390 and they can be divided in three groups as follows:

- 169 points for liquids heated under q = const and $R \neq 1$ conditions;
- 91 point from [22] for water heated in microtubes heated under q = const conditions;
- 130 points for liquids heated under t_w = const conditions.

Tube diameter covered by measurements range from $D_{i,min}$ = 125.4 μm used in the microtube experiments up to D_i = 20.8 mm. $D_{i,min}$ is a minimal value of the internal diameter considered in the engineering applications; below this value the phenomena like viscous heating might influence the heat transfer and fluid flow results.

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Table 1 presents the above mentioned correlations and their statistical parameters – theoretically obtained correlations were subjected to experimental data for the first time. Correlations (9) and (10) have poor statistical parameters. Moreover correlation (10) shows significant deviation which confirms the decision to expel the experimental data from [18]. Equations (8) and (12) were omitted from Table 1 because their statistical parameters are rather poor.

Correlation (11), besides its simplicity, provides quite good agreement with the experimental data. Its only shortage is that for small Gz it provides values of Nu smaller than the theoretical minimum Nu_{fd}.

Sieder and Tate [19] have introduced the viscosity ratio $(\mu / \mu_w)^{0.14}$ in order to include the variation of the temperature field in the tube cross–section and Petukhov [8] confirmed their recommendation. For the whole set of experimental data it can be also confirmed by the analysis performed herewith that ratio $(\mu / \mu_w)^{0.14}$ can be used.

After statistical analysis the following correlation is obtained

$$\mathrm{Nu} = \mathrm{Nu}_{fd} + \frac{0.01 \cdot \mathrm{Gz}^{1.7}}{1 + 0.01 \cdot \mathrm{Gz}^{1.3}} \cdot \left(\frac{\mu}{\mu_w} \right)^{0.14} \tag{13}$$

in which Nu_{fd} presents theoretically obtained value for fully developed flow: $\mathrm{Nu}_{fd} = 3.657$ for constant tube wall temperature (exchangers with $R \to 0$ or $R \to \infty$) and $\mathrm{Nu}_{fd} = 4.364$ for constant heat flux (exchangers with $R = 1$). $\mathrm{Nu}_{fd} = 4.364$ should be used as well for cases when $0 < R < \infty$. All physical properties in correlation (13) are obtained at mean fluid temperature, except μ_w, and the range of process and geometric variables was: Re = 3.35 – 1990, Pr = 4.65 – 12100, μ / μ_w = 0.0048 – 11.7, Gz = 1.4 – 6500, Gr < 30000, L / D_i = 13 – 390.

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Since the correlation is based on variety of experiments that cover heating and cooling of liquids in exchangers it can be concluded that correlation (13) can be applied in all other cases with industrial significance ($0 < R < \infty$), as well as for previously mentioned cases.

The correlation (13) is presented in Figure 1 and its statistical parameters are as follows: SD = 16.2%, $maxRE^{-}$ = –60.2%, $maxRE^{+}$ = 46.2%, CR = 96.6%. It has to be noted that the authors have used all of the available experimental data from [14-17,19,21,22,24] without omitting any points (there are only 6 points with maximal positive error greater than ±50%). Moreover, the form of correlation (13) is simple and convenient for both hand and computer calculations, although it has to be said that more complex mathematical models can be applied [25,26].

CONCLUSIONS

Laminar flow occurs in variety of industrial systems and since there are more than a few analytical solutions and experimentally obtained correlations that can be found in the open literature it is worthwhile to analyze well known relations for estimation of heat transfer coefficient along with available experimental data (390 points). The correlation of Sieder and Tate (11) was found to provide good statistical parameters, but for small Graetz numbers calculated values are lesser than theoretical minimum of Nusselt number.

In order to enable easy to use heat transfer coefficient calculations regardless of the stability of fluid flow and direction of the heat flux, the authors have obtained the novel correlation (13) with significantly improved statistical parameters (standard deviation 16.2%). Correlation (13) is based on several well–known data banks obtained experimentally and covers all of the cases of heat transfer that can occur in industrial heat exchangers: constant tube wall temperature ($R \rightarrow 0$ or $R \rightarrow \infty$), constant heat flux (R = 1) and heating/cooling of fluid characterized by $R \neq 1$. Moreover, a large range of the tube diameters, from micro scale level D_i = 125.4 μm to conventional scale tubes D_i = 20.8 mm are covered with correlation (13). The minimum diameter of $D_{i,min}$ = 125.4 might be considered as the lower limit in designing the engineering applications where conventional theories might be applied.

Nomenclature

c_p isobaric specific heat, J/kg K

CR correlation ratio ($CR = \sqrt{1 - \frac{\sum_{i=1}^{n}\left(z_i - z_{c,i}\right)^2}{\sum_{i=1}^{n}\left(z_i - z_{av}\right)^2}}$)

D tube diameter, m

Gz Graetz number ($\mathrm{Gz} = \mathrm{Re} \cdot \mathrm{Pr} \cdot \frac{D_i}{L}$)

h heat transfer coefficient, W/m^{2}K

k thermal conductivity, W/mK

L length, m

m mass flow rate, kg/s

$maxRE^-$ maximal negative error ($maxRE^- = \max\left(\frac{z_{c,i} - z_i}{z_i}\right)$)

$maxRE^{+}$ maximal positive error ($maxRE^{+} = \max\left(\frac{z_i - z_{c,i}}{z_i}\right)$)

n number of experimental data

Nu Nusselt number ($\mathrm{Nu} = \frac{h \cdot D_i}{k}$)

Pr Prandtl number ($\mathrm{Pr} = \frac{c_p \cdot \mu}{k}$)

R ratio of heat capacities of fluids ($R = \frac{(m \cdot c_p)_{cold\ fluid}}{(m \cdot c_p)_{hot\ fluid}}$)

Re Reynolds number ($\mathrm{Re} = \frac{u \cdot D_i \cdot \rho}{\mu}$)

SD standard deviation ($SD = \sqrt{\frac{\sum_{i=1}^{n}\left(\frac{z_i - z_{c,i}}{z_i}\right)^2}{n}}$)

q heat flux, W/m^2

t temperature, °C

u average velocity, m/s

z_{av} average value for n experimental runs ($z_{av} = \frac{\sum_{i=1}^{n} z_i}{n}$)

z_i experimental data

$z_{c,i}$ correlated data

Greek symbols

μ viscosity, Pa·s

ρ density, kg/m^3

Subscripts

av average

fd fully developed

h hydrodynamically

loc local

min minimum

i inner

t thermally

w wall

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Table 1 The statistical data regarding the correlations for laminar flow (n = 390)

Correlation	SD, %	$maxRE^{-}$, %	$maxRE^{+}$, %	CR, %
(1)	28.4	82.4	61.8	78.6
(2)	25.8	108.7	51.9	88.1
(3)	36.8	37.1	73.2	56.9
(4)	27.7	73.0	61.8	77.8
(6)	29.3	68.3	62.5	73.9
(7)	28.5	117.3	51.8	90.0
(9)	64.8	26.8	93.3	0
(10)	98.2	471.3	99.6	0
(11)	22.2	46.5	56.1	95.5
(13)	16.2	60.2	46.2	96.6

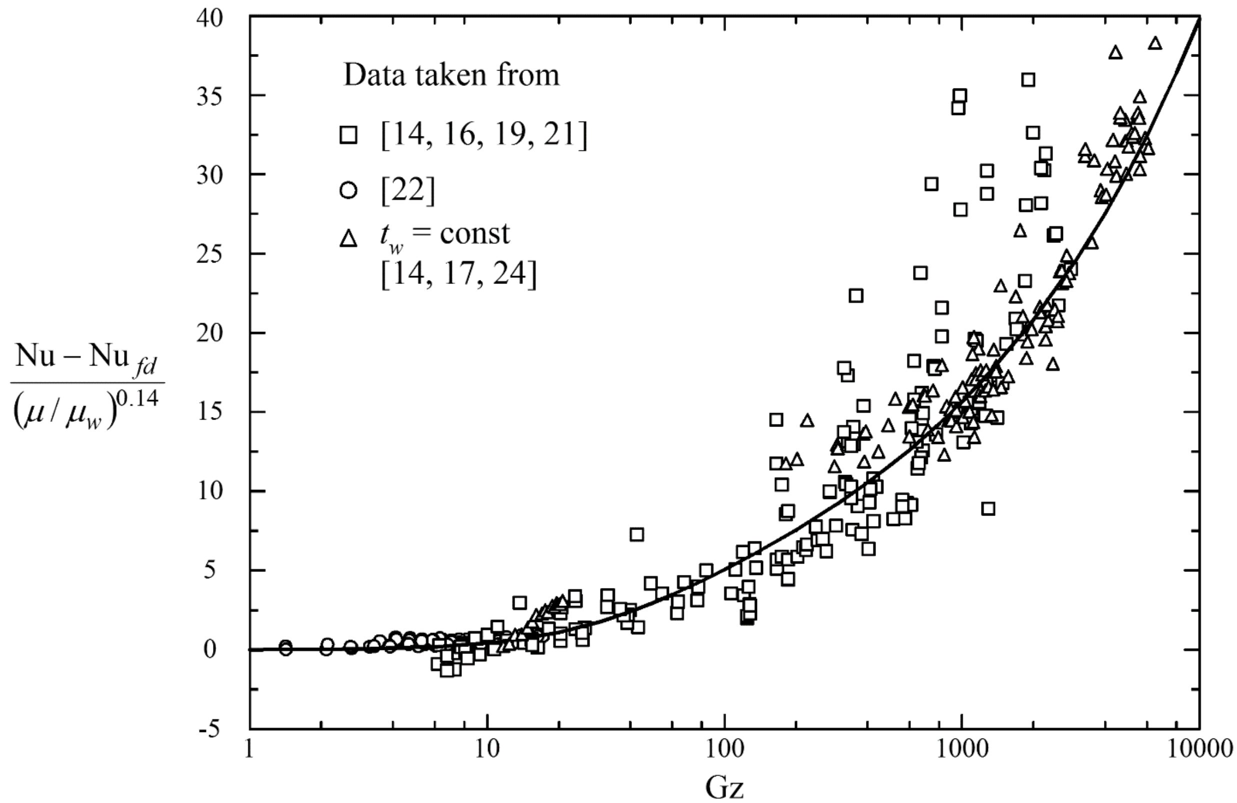

Figure 1 Correlation field of equation (13)

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