



A critical review on various factors affecting the thermohydraulic performance in transition-flow regime

Devendra Kumar Vishwakarma¹ · Suvanjan Bhattacharyya¹ · Mohsen Sharifpur² · Varun Goel³ · Josua P. Meyer⁴

Received: 31 August 2023 / Accepted: 4 April 2024 / Published online: 18 June 2024
© Akadémiai Kiadó, Budapest, Hungary 2024

Abstract

Heat transfer and pressure drop in laminar and turbulent flow have been studied for a long time. However, the thermohydraulic of fluid flow in transition flow is still in the embryonic stage and needs further exploration. Primarily, this article complied all the fragmented research works linked to thermal and flow performance in transition-flow regimes. Several detailed research works pertaining to developing and fully developed transition flow were reported in the past three decades which significantly contribute to the field. It was also found that the transition-flow regime shifted with the change in the operating conditions such as free, forced and mixed convection, developing and fully developed flow, inlet geometries, type of working fluid, channel geometries, roughness, the orientation of test section, etc. Hence, by altering the shape of the entrance, the amount of heat passing through, or the surface of the tube, one could manipulate the range of Reynolds numbers where the transition took place. The temperature of fluids affects their densities, so when heat is applied to the tube wall, it creates temperature differences within the thermal boundary layer. These differences in temperature then cause changes in density and buoyancy due to the force of gravity. Several novel correlations have been developed based on these findings to determine the heat transfer and pressure drop in different flow regimes. In order to develop accurate correlations for heat transfer and pressure drop in the transitional-flow regime, it is necessary to comprehend the factors that impact the beginning and end of this regime. This understanding is crucial for selecting or creating suitable correlations.

Keywords Transition flow · Heat transfer · Forced convection · Mixed convection · Developing and fully developed flow

Introduction

In the year 1883, Reynolds published a research work which was focused on fluid-flow regime and its dependence on the dimensionless parameter later known as the Reynolds number. The conditions which demonstrate the transition of laminar flow to turbulent flow were determined. Slowly, the investigations in the field of flow regimes behavior started getting momentum. The disruption of the flow field while the flow is transitioned from a laminar to a turbulent-flow regime enhances the thermal transport capacity. By disrupting the boundary layer, mixing of fluid can be intensified which leads to the enhancement in heat transfer.

The studies related to heat transfer and pressure drop in the laminar flow have already matured, while hydrothermal behavior in turbulent flow is still being investigated. The thermohydraulics of flow in the transition flow is still in an embryonic stage. The reasons for less research in the field of transition flow are the unpredictable nature of flow as well as very little to no previous works to support the research. Still,

✉ Suvanjan Bhattacharyya
suvanjan.bhattacharyya@pilani.bits-pilani.ac.in

Devendra Kumar Vishwakarma
p20190451@pilani.bits-pilani.ac.in

¹ Department of Mechanical Engineering, Birla Institute of Technology and Science, Pilani, Pilani Campus, Vidya Vihar, Pilani 333 031, Rajasthan, India

² School of Mechanical, Industrial & Aeronautical Engineering, University of the Witwatersrand, Johannesburg 2017, South Africa

³ Department of Mechanical Engineering, National Institute of Technology Hamirpur, Hamirpur 177005, Himachal Pradesh, India

⁴ Department of Mechanical & Mechatronic Engineering, Stellenbosch University, Stellenbosch 7599, South Africa

few research works are available in the area of flow dynamics in transition-flow regimes. As per the authors' knowledge, very few manuscripts are available focusing on the hydrothermal characteristics in the transition-flow regime.

The results obtained from the various research work allow the design and fabrication engineers to incorporate the changes in the existing designs of heat exchangers to get better and improved results. Understanding the thermal and flow behavior in transition-flow regime is very essential for solar, oil and piping industries. Because of high viscosity fluid usage in these industries, the Reynolds number remains restricted to the early turbulent-flow regime so that the pumping losses associated with higher flow rate remain lower. But with the aging of the equipment, the range of flow regimes shifted due to deposition inside the channels. This forced the equipment to work in the transition-flow regime. Hence, most of the time, these devices are working in the transition-flow regime unknowingly. This review work helps these industries by summarizing the results of various researchers working in the field of solar, piping, oil, heat exchanger, power plants, etc. The main intention behind the present review paper is to create a concise document which covers the important findings in the field of transition-flow regime.

In the following subsection, the thermohydraulic characteristics of laminar and turbulent flow will be discussed to create basic knowledge about the physical significance of flow behavior in different flow regimes.

Heat transfer and pressure drop in laminar-flow regime and turbulent-flow regime

In laminar-flow regime, fluid flows in the form of lamina (layers of fluid) parallel to each other without disturbing the adjacent fluid layers. The fluid flows in a parallel layer without any eddies and disruption in the boundary layer

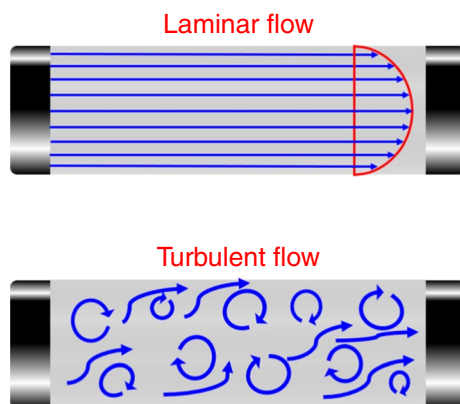


Fig. 1 Schematic representation of flow in laminar and turbulent regime

(Fig. 1). This flow regime is governed by momentum diffusion, i.e., viscous forces dominate over inertia forces. Since the disturbance in this flow regime is minimal and the heat transfer and friction factor remains low. However, transport characteristics in this flow regime can be studied to improve the hydrothermal characteristics by introducing the various types of vortex generators [1], roughness, fins, nanofluids [2–4], electric field, magnetic field, ultrasound [5], etc., in the channel which increase the agitation in the flow field and allows the breaking of boundary layer resulting secondary fluid flow and hence the heat transfer rate can be enhanced [6–9]. Several experimental and numerical works are available in the open literature to support the flow dynamics of the laminar flow. It was found through various reports that the fluid remains in the laminar-flow regime up to a Reynolds number value of 2300 [10–12]. Authors are exploring various combinational methods to enhance thermal transport in the laminar flow. The combined impact of nanofluid and longitudinal vortex generator on the heat transfer attribute in laminar flow has been studied by Asaadi and Abdi [11]. An increase in heat transfer coefficient and pressure drop (friction factor) is reported with an increase in the volumetric concentration of the nanoparticles. This increase in heat transfer coefficient is due to the incorporation of nanoparticles in the base fluid which led to the increase in the effective thermal conductivity as well as the Prandtl number [13]. It was also reported that an increase in the size of nanoparticles has an adverse effect on the heat transfer coefficient [12, 14]. Feng et al. [15] ran experiments to study the influence of the length and twist ratio of twisted tapes on the heat transfer aspects in square mini channels. It was reported that the larger the length of twisted tapes, higher will be the augmentation in heat transfer coefficient. Bain et al. [16] reported on the flow and transport characteristics of nanodiamond-based dielectric oil for cooling applications and found that nanodiamond-based dielectric oil is a promising coolant for transformer applications. Hwang et al. [17] effectuated experiments to evaluate the heat transfer proliferation with Al_2O_3 –water nanofluid in a laminar-flow regime. The usage of nanoparticles in the base fluid enhances the heat transfer coefficient by 8%. Shah and London [18–22] studied the laminar flow for parallel plates, circular tubes, elliptical tubes, triangular tubes and rectangular tubes for different boundary conditions, heat flux, viscous dissipation and momentum diffusion to understand the flow dynamics for different geometries. Bhattacharyya and Saha [23] performed an experimental evaluation to demonstrate the hydrothermal aspect inside a corrugated tube fitted with center-cleared twisted tapes for laminar-flow regime. It was found that combination of corrugated tube along with twisted tapes performed far better than individual techniques acting alone.

From the preceding discussion, it is clear that enhancement of heat transfer in the boundary layer is possible with insertion or modification, usage of high thermal conductivity fluids, nanofluids, etc. Appreciable enhancement in the thermal performance was noted when combined techniques of heat transfer enhancement were used. The disturbance in the boundary layer due to insertions of vortex generators, roughness on the surface of duct, etc., allows turbulence in the flow field which leads to better mixing of fluid elements and enhances thermal transport.

The physical relevance of heat transfer in turbulent-flow regime is completely vice versa in comparison with laminar-flow regime. In turbulent-flow regime, the heat transfer and pressure drop both are very high. This is due to the turbulent nature of fluid flow which allows mixing of the fluid (Fig. 1). The heat transfer appreciation in turbulent-flow regime can be further enhanced with the help of twisted tapes, ribs, fins, roughness, etc. The inserts or roughness enhance the effectiveness of heat transfer, but at the same time increase the drag also which results in a higher pressure drop (friction factor). Heat transfer in this flow region has been studied for the past five or six decades aiming for a higher heat transfer rate. One of the key parameters to evaluate the combined influence of the effective heat transfer aspect along with pressure drop is thermal performance factor (TEF or η). Thermal performance factor (TEF), also sometimes referred to as thermal enhancement factor or performance enhancement coefficient, is the ratio of increase in the Nusselt number (Nu/Nu_0) to the increase in the friction factor (f/f_0). This parameter gives a vivid idea of amplification in heat transfer rate. If the value of thermal performance is above unity, it means for the given conditions heat transfer is more in comparison to the friction factor. The physics of heat transfer enhancement in turbulent-flow regime has been investigated by many authors in the past few decades. For example, Chang et al. [24] conducted experiments to evaluate the thermal and flow performance in a tube fitted with serrated twisted tapes of different twist ratios. Air is taken as the working fluid with Re varied from 5000 to 25,000. It was reported that Nusselt number increases with a decrease in twist ratio. The overall performance of the serrated twisted tape is appreciable when compared with plain twisted tapes. The experimental finding of Eiamsa-ard et al. [25] demonstrates the influence of twisted tapes with center wings and alternate axis on the thermal and flow performance in a turbulent-flow regime. The unique tape geometry splits the two streams of fluid into four streams around the alternate axis. Past alternate axis the fluid streams recombine into two streams. This allows the better mixing of fluid streams and enhances the thermal transport rate. The wings on the surface of tapes also allow furious mixing of fluid by disturbing the thermal boundary layer, leading to higher heat transfer.

Sometimes, heat exchangers are operated in a transition-flow regime unknowingly. This unknown shift in the regime is due to the presence of roughness in the flow channel. Besides this entrance of the channel, usage of twisted tape inserts, corrugation on the surface of channel, amount of heat flux, etc., also significantly control the flow regime shift. Hence, in this work, an effort is made by the author to concise all the available research work related to heat transfer and flow characteristics in the transition-flow regime. This document will provide an easy and thorough summarization of previous research works for single-phase-flow heat transfer and pressure drop in transition-flow regime.

Transition-Flow regime

The transition-flow regime falls between the laminar-flow regime and turbulent-flow regime when fluid passes from the laminar and enters the turbulent-flow regime. This regime is characterized by its unpredictable and chaotic nature of the fluid flow. Generally, plain channel transition occurred between 2000 and 4000 Reynolds numbers in case of internal flow (refer Fig. 2). But it can start as early as 700 Reynolds number and delays up to 10,500 Reynolds number [26, 27] depending upon the disturbances and geometry of the flow channel. The length of the transition regime depends upon various factors such as the inlet geometry of the conduit, surface geometry of conduit, heat flux, type of fluid, etc. (Figs. 3 and 4).

The engineers of heat exchangers generally recommend working in a turbulent-flow regime due to commendatory heat transfer rate. However, with the aging of the heat exchangers mineral deposition takes place on the wall of heat exchanger tubes. Due to this, the flow regime shifted and unknowingly the heat exchangers start operating in the transition-flow regime. Similarly, the usage of the insert also shifts the initiation and termination of the transition-flow regime. These inserts enhance the heat transfer, but at the same time, it may also increase the friction factor. This may increase power consumption. Because of such reasons,

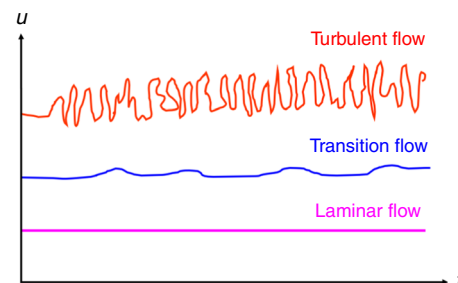


Fig. 2 Velocity trend for laminar (viscous)-, transition- and turbulent-flow regime

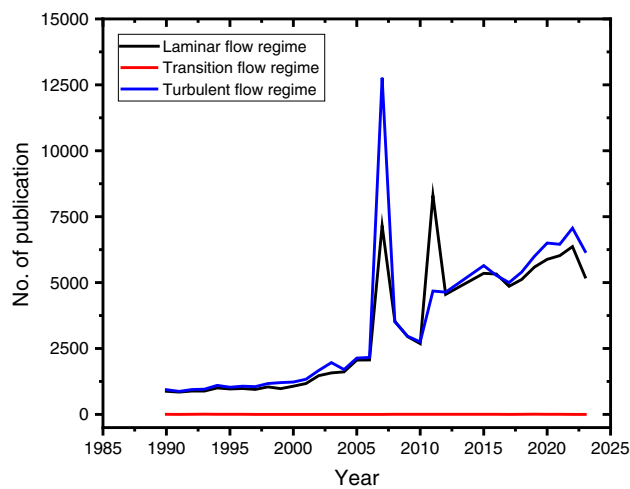
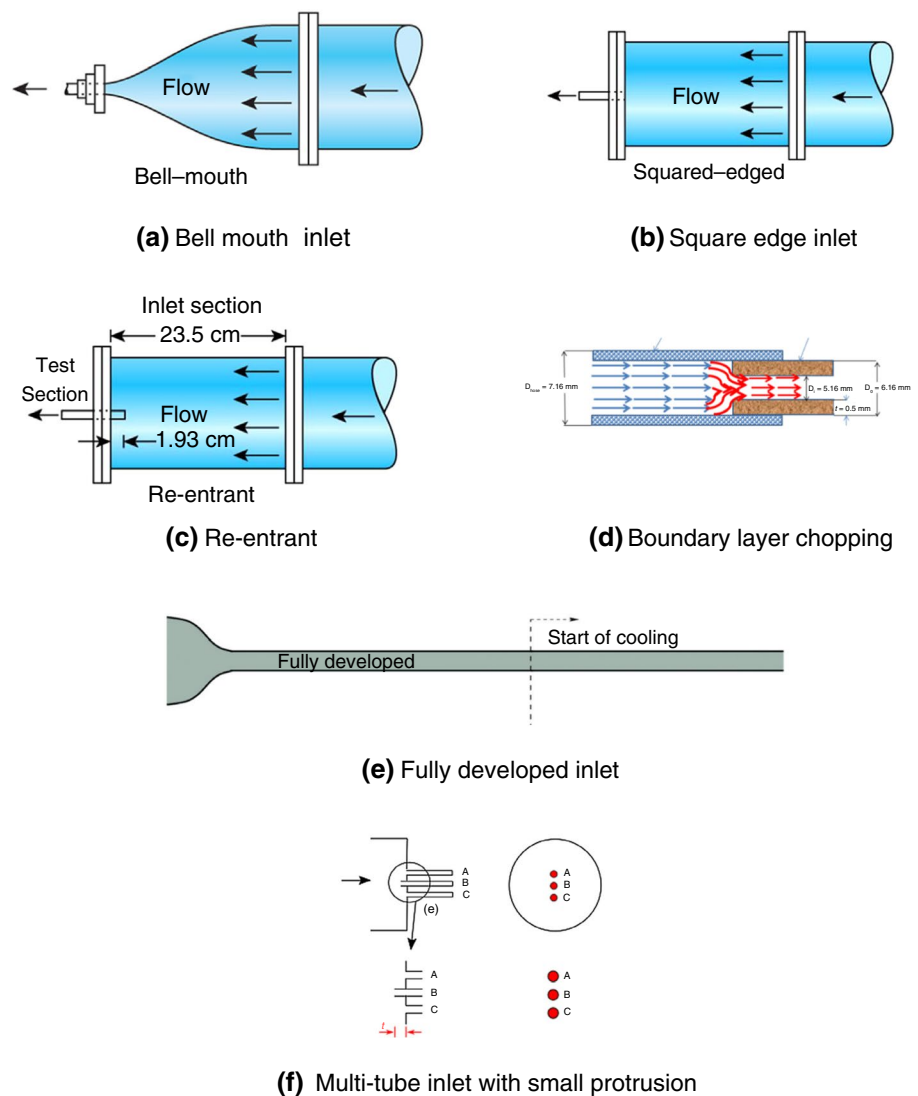


Fig. 3 Year-wise publications in laminar-, transition- and turbulent-flow regime. [<https://www.sciencedirect.com/>, <https://scholar.google.com/>]

researchers need to understand the physics of heat transfer and pressure drop in transition-flow regime. Besides this, the new data obtained from the experimental and numerical investigations help in understanding the flow dynamics. For oil industries, such as refineries, the fluid does not allow to flow at very high speed. This is done to optimize the cost associated with pumping the viscous fluids in the channel. The fluid flow in such industries is generally early turbulent regime. Moreover, this can be further shifted to transition with time due to deposition in the channel. The research in transition-flow regime is still in the early stage with less than a hundred manuscripts published so far. The unpredictable and unstable nature of flow makes this regime very challenging to capture. Figure 3 compares the number of publications in different flow regimes in the field of heat transfer and pressure drop.

Fig. 4 Inlet Geometries **a** Bell mouth inlet **b** Square edge inlet **c** Re-entrant **d** Boundary layer chopping **e** Fully developed inlet **f** Multi-tube with small protrusion [28–31]



Factor influencing the boundaries of transition-flow regime

Several factors influence the heat transfer and friction characteristics in the transition-flow regime. A few of them are discussed below.

Inlet geometry

Inlet geometries greatly influenced the boundaries of transition flow [32]. Most of the researchers investigated the majorly three inlet configurations, namely bell mouth, re-entrant and square edge [28, 33]. Figure 4 shows the schematics of various inlet configurations explored previously for transition regime. Smoother inlets result in a late laminar to turbulent transition, while rough or blunt inlets result in an early transition [28, 29, 33, 34].

Ghajar and Madon [33] and Ghajar and Tam [34] carried out an experimental investigation with three different inlet configurations, namely re-entrant, square edge and bell mouth, for skin friction and heat transfer analysis in the transition-flow regime. The measuring section used for the investigation was 6.10 m in length and 15.8 mm in diameter. To measure the pressure drop, three differential pressure gauges of different capacity were installed over the test section, while to measure the temperature across the test section, thermocouples were installed at 31 stations over the test section. A mixture of water and ethylene glycol was used as the working fluid. A shift in transition boundaries has been reported by the authors for different inlet geometries. It is also vital to note that smoother inlets delayed the transition most, while the blunt inlets result in early transition. Ghajar and Tam [35] carried out experiments to develop a flow regime map for horizontal circular channel for uniform heat flux and three inlets applicable in both developing and fully developed flow. The developed map helps in determining the coordinates for considering the mixed convection. Meyer et al. [36] and Olivier and Meyer [28] performed experiments inside a smooth tube with different inlet profiles for adiabatic and diabatic operating conditions and gathered data for pressure drop and heat transfer. Four inlet configurations, namely hydrodynamically fully developed, bell mouth, re-entrant and square edge, were investigated. Tam and Ghajar [37] reported on the influence of bell mouth configuration on the local heat transfer coefficient in transition-flow regime and turbulent-flow regime. They reported an unusual behavior of local heat transfer coefficient in the transition-flow regime which causes changes in the average heat transfer coefficient. Tam and Ghajar [27] in another investigation reported on the influence of inlet geometry as well as heating on the thermal and flow performance. As expected, re-entrant and square edge inlet show earlier

transition, while the bell mouth inlet shows a delayed transition. Influence of heat flux was also investigated. Increase in heat flux increases the friction factor for a fixed value of Reynolds number. Also, increased heat flux delayed the transition for all the inlet geometries and Reynolds number. Osman et al. [30] investigated a novel inlet geometry “boundary layer chopping inlet” for thermohydraulic characteristic in transition-flow regime and reported early occurrence of transition when inlet geometry is employed. However, termination limit of transition does not vary for the given inlet configuration.

Surface roughness

Numerous technical methods have been developed over time to increase the rate of convective heat transfer from wall surfaces. To boost the heat transfer coefficient from the flow surface by increasing turbulent motion, roughness elements are frequently used in this application. It is preferable that the heat transfer be meliorated with the least amount of drag. It has been observed that surface roughness changes the width of transition-flow regime significantly. The roughness causes disruption in the flow field which in turn results in the shift of the transition-flow regime. Surface roughness includes ribs [38–40], protrusions [41–43], dimples [44–46], surface unevenness [47–49], etc. Meyer and Olivier [50] experimented with an enhanced tube for developing and developed flow to investigate the influence of surface roughness on pressure drop. Four enhanced tubes and three inlet geometries [34] were employed for the experiments. It was found that for a smooth tube, the transition depends upon the type of inlet geometry. Transition starts and ends early for blunt inlet, while it starts and ends later for a smooth inlet. For enhanced tubes, transition begins earlier than the transition of smooth tube. A secondary transition-flow regime was also reported by the investigators between Reynolds number 3000 and 10,000. Meyer and Olivier [51] in the second part of investigation reported on the heat transfer characteristics. It was reported that for adiabatic flow, no changes in the transition-flow regime were reported due to changes in the inlet geometry. Additionally, it seems that the transition area is seldom affected, if at all, by roughness during the heat transfer process. Another reason for lower enhancement was reported obstruction of secondary flow by the roughness of the tube at lower velocity. Everts et al. [52] also reported on the influence of microroughness over the surface of tube on the transition-flow regime. Tubes used for the investigations were having relative roughness close to 0.0003 and 0.0006. The Reynolds number of water varied from 500 to 9800. Three heat fluxes 1, 2 and 3 kW m⁻² were used for heating the test section. The results obtained from the investigation show enhancement in the heat transfer in the laminar-flow regime, while no enhancement is reported

for transition-flow regime and turbulent-flow regime. For isothermal and diabatic conditions, roughness delayed the transition, reduced the width of transition-flow regime and increased the transition gradient. Vicente et al. [53] executed experiments with corrugated tube for mixed convection heat transfer and pressure drop characteristics in laminar-flow regime and transition-flow regime and reported that roughness on the tube surface accelerate the transition below 1300 Reynolds number. In another investigation by Vicente et al. [54], dimpled tube was explored for thermal and flow characteristics and changes associated with surface roughness. Srinivasan and Christensen [55] carried out experiments with spiral tubes and reported early occurrence of transition at Reynolds number approx. 300 and termination at approx. Reynolds number 1000. Besides, a significant improvement in heat transfer and friction factor was also reported when spiral tubes were used. Similar results were also reported by Obot et al. [56]. Olsson and Sundén [57] carried out experimental investigation for solar air heater with various configurations of rib roughness for wide range of Reynolds number. It was found that swirl tube performed better than the other configuration for Reynolds number range between 1000 and 2000. Afterward, the difference is not appreciable. Vicente et al. [58] reported on mixed convection in dimpled tubes (T01–T10) for laminar-flow regime and transition-flow regime. Water and ethylene glycol were used for working fluid. Transition for dimpled tube started at Reynolds number 1400. Both heat transfer and pressure drop were found augmented with dimpled tubes. Zhang [59] presented a numerical study to evaluate the thermal and flow performance inside a cross corrugated heat exchanger in transition-flow regime. K- ω turbulence model was used to model the transition flow. The results were presented in terms of temperature, velocity and turbulence intensity. García et al. [60] carried out experiments with corrugated tube, dimpled tube and wire coils for heat transfer augmentation. They concluded that smooth tube performed best for Reynolds number lower than or equal to 200, while using wire coils for heat transfer enhancement is recommended for Reynolds number range of 200–2000. Beyond Reynolds number 2000, dimpled or corrugated tube performed better. Transition begins earliest for tube with wire coil, followed by the corrugated tube, dimpled tube and smooth tube (in given order). Popov et al. [61] estimated the thermofluidic nature in transition-flow regime. It was reported that roughened tubes performed better over the other in terms of efficiency.

Insertions or turbulators

Turbulators or tape inserts are commonly used tools for the enhancement of heat transfer in heat exchangers also influence the transition-flow regime boundaries. It is very common practice to use tape inserts or turbulators for increased

performance of the thermal system. It is established by the studies of Abolarin et al. [62–64] that tape insertion in a heat exchanger tubes shifts the transition boundaries significantly. They investigated the conventional twisted tapes, clockwise and anticlockwise connected twisted tapes and twisted tapes with conical rings and peripheral U-cuts (refer Fig. 5). Water is used as the working fluid for all the investigations. Meyer and Abolarin [62] experimentally investigated the influence of conventional simple twisted tapes (three twist ratio) and heat flux (three heat fluxes) on the transition-flow regime. A circular test section with an internal diameter 19 mm and length 5.27 m was employed for the experiments. For uniform heat flux, the Colburn j-factor was reported to be highest for the lowest twist ratio, while an increase in twist ratio resulted in decreased j-factor value. Also, transition begins earlier for lower twist ratio. For uniform twist ratio, the transition begins earlier for higher heat fluxes. In another experimental investigation, Abolarin et al. [63] also examine u-cut twisted tapes inside a heat exchanger duct with and without ring inserts. A constant twist ratio tape was used, while the depth of u-cut and ring space was varied. The results obtained from the investigation show the impact of both u-cut and ring space on the transition boundaries. It was revealed that an increase in depth ratio as well as an increase in ring spacing results in early transition. Similar to the above two investigations, Abolarin et al. [64] employed alternating clockwise and anticlockwise twisted tapes joined at 0°, 30° and 60° angle. The experiments were carried out at different heat flux conditions with water as the working fluid. It was revealed from the investigation that both joining angle and heat flux influence the limits of the transition-flow regime. Transition begins and ends earliest for 60° connection angle for all heat fluxes while delayed most for 0° connection angle. Transition begins and ends early for lower heat flux. They also developed several correlations based on their investigation for Nusselt number and friction factor prediction. Chougule and Sahu [61] carried out experiments in a circular tube fitted with helical screw inserts and compared the obtained results for performance in transition-flow regime for two nanofluids. García et al. [65] experimentally evaluated the augmentation of heat transfer and pressure drop in circular tube fitted with wire coil in laminar-flow regime and transition-flow regime. Insertion of wire coil changes the critical Reynolds number for heat transfer and pressure drop. Fiebig [66] carried out experiments to study heat transfer enhancement using various winglets configurations. They identified three methods of heat transfer enhancement, namely boundary layer development on vortex generator, swirl and flow destabilization. It was found that the flow destabilization results in momentous decrement in the critical Reynolds number (300) for the start of transition-flow regime.

Fig. 5 Tape inserts explored by Abolarin et al. [62–64] **a** Simple Twisted tapes **b** U-cut twisted tape with ring inserts **c** Clockwise and counterclockwise tapes joined at different angles

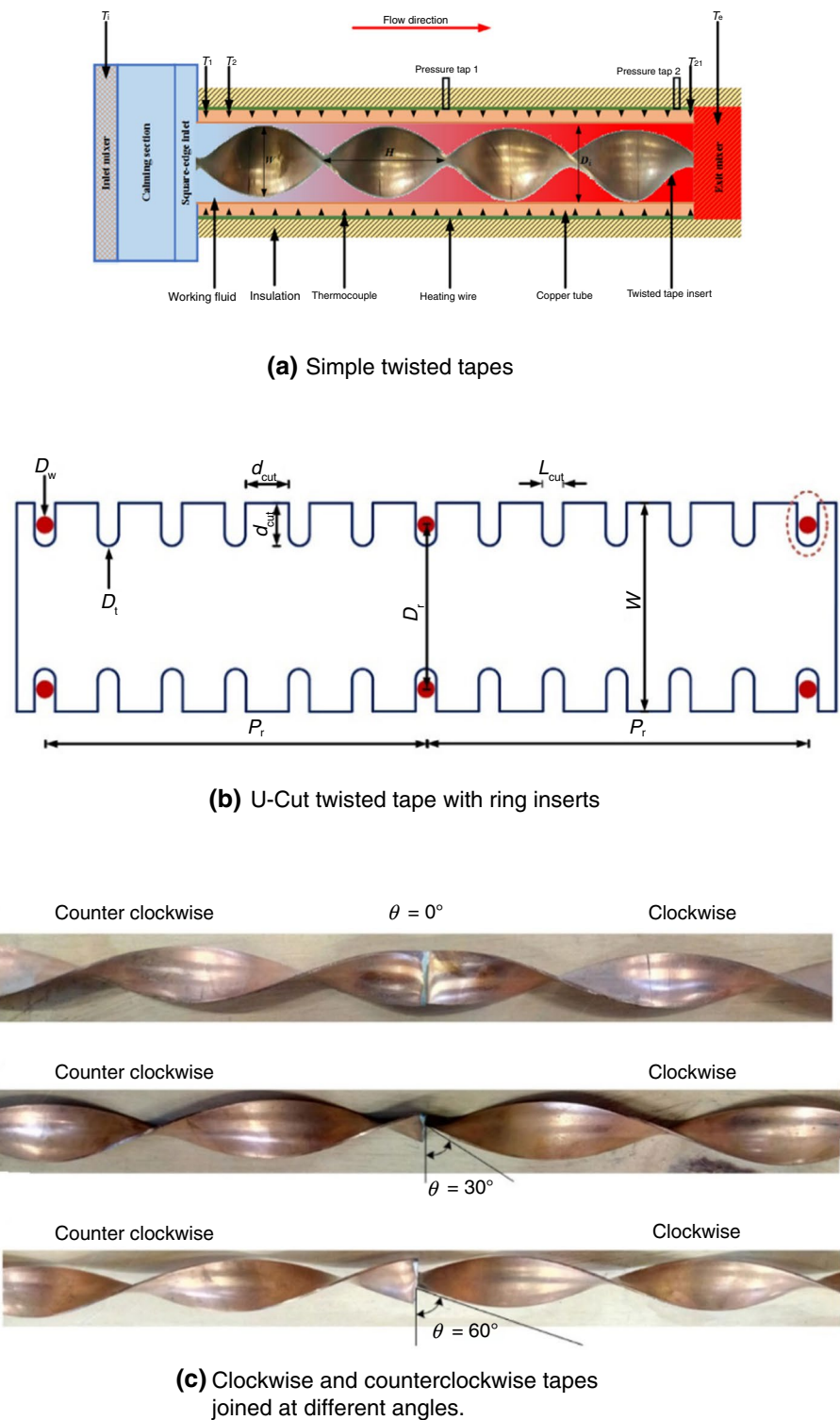


Figure 6 shows the boundaries of transition-flow regime for different tape inserts discussed above. It is clearly depicted from Fig. 5 that transition boundaries vary significantly with the geometry of inserts as well as heat flux. For lower heat flux, transition begins and ends earlier, while for higher heat fluxes, the start and end of transition

is delayed. Similarly, for complex geometries, the transition begins earlier, while for lesser complex geometries it is delayed. One of the very important conclusions made from the above investigations is that transition started and ended way earlier than Reynolds number value of 2300

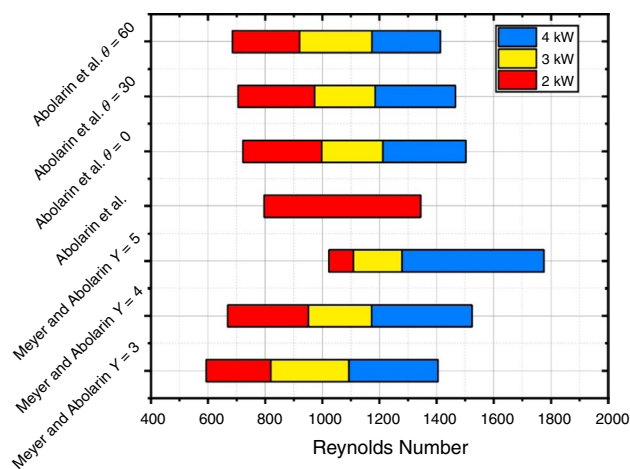


Fig. 6 Transition-flow regime variation for different tapes and heat flux conditions explored by Meyer and Abolarin and Abolarin et al. [62–64]

Zagarola et al. [67] also reported on the thermal performance improvement in laminar-flow regime and transition-flow regime when eddy promoters were employed perpendicular to flow direction. Few other works which moderately discussed the thermal and flow parameters in transition-flow regime included research of Elison and Webb [68], Mikic et al. [69]. García et al. [70] employed hydrogen bubble and PIV techniques to determine the flow mechanism inside a tube fitted with wire coil inserts of three pitches. It was found that the transition for W01 starts at Reynolds number 700, followed by W02 and W03.

From the above discussion, it can be concluded that turbulators enhance the heat transfer efficiency by introducing disruption in the flow field. Along with that, turbulators also cause the shift in the boundaries of transition-flow regime. The higher the complexities in the design of turbulators, earlier the transition begins. This is due to the higher rate of disturbance due to turbulators in the flow field.

Heat flux

The amount of heat energy that flows across a surface is known as heat flux or thermal flux. External heat flux or intensity can greatly influence the flow regimes. The impact of heating on the performance of heat exchangers has been evaluated thoroughly by many researchers in the past. However, due to the uncertain nature of the transition-flow regime, very few studies related to transition-flow regime are available in open literature. This also causes a paucity of data. An extensive investigation of transition-flow regime is the need of the time. If investigated, the research will help in optimizing the heat transfer and pressure drop in the heat exchangers. This will help in saving the cost associated with pumping. Saeedinia et al. [71] experimented with heat

exchanger tube subjected to 6.2 kW m^{-2} to 9.9 kW m^{-2} of heat flux and reported that increasing the heat flux result in enhanced heat transfer for both base fluid and nanofluid. However, the augmentation in heat transfer is more with nanofluid in comparison to the base fluid. Andrade et al. [72] also varied the heat flux from 5.5 to 21.1 kW m^{-2} for internal corrugated tube and revealed relevant heat transfer augmentation in turbulent-flow regime. Ghajar and Tam [34] experimented with a circular heat transfer tube having length approximately 5.5 m which is subjected to heat flux of 4 – 6.70 kW m^{-2} . A combined influence of heat flux and inlet geometry was evaluated and reported a delayed transition when results were compared with no heating condition. Tam and Ghajar [27] reported on the influence of heating on the laminar to turbulent transition-flow regime. The findings showed that, for a constant Reynolds number, the value of the fully developed friction factor rose as the heating rate increased. Heating also causes a delay in the transition for all the inlet geometries. It was found that for 16 kW m^{-2} of heat flux and re-entrant inlet, the transition initiated at $Re = 4100$ and ends at $Re = 5900$, for the square edge the transition begins at $Re = 4500$ and ends at $Re = 6400$, while for bell mouth inlet, the transition initiated at $Re = 7300$ and ends at $Re = 9600$. Martínez et al. [73] used heat flux ranges between 4.5 and 66.4 kW m^{-2} for investigating the heat transfer and pressure drop for heat exchanger tube fitted with a wire coil and reported that the critical Reynolds number goes below 500 for transition. Meyer and Abolarin [62] and Abolarin et al. [64] investigated various heat fluxes, i.e., 1.35 , 2 , 3 and 4 kW m^{-2} for circular channel fitted with a twisted tape of various configurations and reported that greater heat fluxes delayed the transition. Abolarin et al. [63] in another geometry carried out experiments with a uniform heat flux of 1.35 kW m^{-2} in a circular channel fitted with u-cut twisted tapes with and without ring inserts and reported amplification of heat transfer and pressure drop. Meyer and Everts [74] carried out experiments to understand the effect of natural convection on local heat transfer. The experiments were carried out for uniform heat flux. The investigated heat fluxes varied from 0 to 9.5 kW m^{-2} . Higher heat flux results in higher Nusselt value. Duct diameter also plays a significant role in heat transfer enhancement.

Based on the aforementioned investigations, it is evident that heat flux plays a significant role in influencing the transition-flow regime. An increase in applied heat flux led to a delayed transition compared to lower heat flux under similar operating conditions. This effect was accompanied by higher Nusselt numbers and friction factors.

Geometry of test section

Chen et al. [75] employed a grooved channel to augment the thermal performance of the tube in a tube heat

exchanger. It was reported that the grooved channel significantly improves the heat transfer coefficient while decreasing the critical Reynolds number for the transition-flow regime. Osman et al. [76] accomplished experiments in a rectangular tube with Al_2O_3 -water nanofluid (Vol. % 0.3, 0.5 and 1) for heat transfer and pressure drop performance in visco-laminar and transition-flow regime. It was found that transition occurs earlier for a rectangular channel. Besides 54% increment in the convective heat transfer coefficient is reported in the transition-flow regime followed by 11% improvement in the turbulent-flow regime for 1% vol. concentration. Water was employed in a single channel with a rectangular cross section by Dirker et al. [77] employed water as the working fluid in a single channel with a rectangular cross section. They wanted to find out how intake layout affected heat transfer coefficients when frictional elements such as adiabatic and diabatic friction were present. For the bell mouth intake type, a smoother and longer transition was observed, with the adiabatic transition starting at a Reynolds number 1050 and the diabatic transition starting at a Reynolds number 1200. However, the change for the smooth contraction inlet type was more sudden, starting at a Reynolds number 1800 for the adiabatic transition and a Reynolds number 2000 for the diabatic transition. Minhas and Lock [78] numerically studied the fluid flow characteristics for the reflux heat exchanger. It was found that the transition initiated at Reynolds number 875, whereas it is terminated at Reynolds number 1480 for both air and water. Minhas and Lock [79] in another work reported on the forced convection heat transfer and fluid flow in the reflux heat exchanger. An increase in the Nusselt number with an increase in the Reynolds number was reported. Joye et al. [80] experimented with vertical channel for opposing water flow and compared the heat transfer for wide range of Reynolds number. They reported that mixed convection occurs between Reynolds number 4000 and 10,000. Peng and Peterson [81] performed experiments in microchannel (total eight) for forced convection single phase flow of water and methanol mixture. For test section 1, 2 and 3, the transition occurs between Reynolds number 300–700 and a decrease in critical number was found with decrease in the hydraulic diameter of microchannel. For other microchannels the critical Reynolds number for start of turbulent region was found near to Reynolds number 200, while the transition occurs at Reynolds number 70. Besides, the concentration of methanol in water also majorly influences the laminar to turbulent transition.

Hence, it can be concluded that the shape of the test section also has a significant impact on the initiation and conclusion of the transition-flow regime. Diverse shapes can lead to differences in the formation of the boundary layer, which in turn affects the shift from smooth to chaotic flow.

Specific shapes may naturally disrupt the smooth flow, encouraging the transition to turbulence. Engineers and researchers meticulously analyze these geometric aspects to enhance efficiency and forecast the characteristics of fluid flow in different scenarios.

Angular position of test section

Meyer et al. [82] experimented with inclined channel to inspect and evaluate the influence of buoyancy on heat transfer and pressure drop. At heat fluxes of 4–8 kW m^{-2} , the tests were carried out at various inclination angles between 1000 and 6000 Reynolds numbers. It was discovered that decreasing the buoyancy force effects from horizontal flow (0°) to vertical flow (90° flow) lowered the laminar heat transfer coefficients and friction factor for both uphill and downhill flows. It was discovered that when the inclination angles grew from horizontal to vertical, the critical Reynolds numbers at which transition began increased. However, the termination of transition remains unaffected due to variation in the inclination angle. Everts et al. [10] carried out heat transfer and fluid flow experiments in vertical channel to understand the influence of assisting and opposing flow at very Reynolds number. At three distinct heat fluxes of 1, 1.5 and 2 kW m^{-2} , experiments and simulations were carried out for vertically upward and downflow between Reynolds numbers of 180 and 2300. Free-convection effects were reported to no longer be repressed by the fluid's velocity at Reynolds numbers below 250 and 600 for upward and downflow, respectively, and aiding and opposing flow became important. Using both single-phase and multi-phase models, Saha and Paul [83] conducted a computational study to examine the transition-flow behavior of nanofluids flow in an inclined pipe. Alzwayi and Paul [84] reported on the influence of angular inclination and channel width on the heat transfer and pressure drop in transition-flow regime. The results demonstrated how the channel's width and angle affect the average heat transfer coefficient. The findings also demonstrate that the channel's width and inclination have a significant role throughout the transition. Al-Arabi and Sakr [85] examined the local and average natural convection heat transfer from isothermal vertical and inclined plates facing upward to air in both visco-laminar and turbulent zones and suggested empirical correlations to estimate the local and average heat transfer coefficient in the transition-flow regime. Kumar and Premachandran [86] numerically investigated the influence of inclination of fluid domain on the transition-flow regime and reported that the transition is delayed and the transition area moves downstream as the degree of inclination rises.

Angular orientation of heat exchanger tubes also plays a crucial role in determination of transition-flow regime [82]. However, this aspect is not very well studied and

Table 1 Critical Reynolds number and important findings of selective previous works

Authors and year	Type of inlet	Developing flow	Developed flow	Critical Reynolds number (Start)	Critical Reynolds number (End)	Width of transition-flow regime	
<i>Inlet geometry</i>							
Ghajar and Madon [33]	Re-entrant	✗	✓	1980	2600	620	
	Square edge	✗	✓	2070	2840	770	
	Bell mouth	✗	✓	2125	3200	1075	
Ghajar and Tam [34]	Re-entrant	✗	✓	2000	8500	6500	
	Square edge	✗	✓	2400	8800	6400	
	Bell mouth	✗	✓	3800	10,500	6700	
Olivier and Meyer [28]	<i>Adiabatic friction factor 15.88 mm</i>						
	Hydrodynamically fully developed	✗	✓	2212	2600	388	
	Square edge	✓	✗	2717	2850	133	
	Re-entrant	✓	✗	2192	2542	350	
	Bell mouth	✓	✗	7000	7700	700	
	<i>Adiabatic friction factor 19.02 mm</i>						
	Hydrodynamically fully developed	✗	✓	2350	2560	210	
	Square edge	✓	✗	2600	2770	170	
	Re-entrant	✓	✗	2194	2480	286	
	Bell mouth	✓	✗	12,000	17,500	5500	
	Meyer et al. [36]	<i>Adiabatic friction factor</i>					
		Hydrodynamically fully developed	✗	✓	2000	2900	900
		Square edge	✓	✗	2600	3110	510
		Re-entrant	✓	✗	2000	2760	760
		Bell mouth	✓	✗	7000	7800	800
<i>Diabatic friction factor</i>							
Hydrodynamically fully developed		Transition independent of inlet geometry					
Square edge							
Re-entrant							
Bell mouth							
Tam and Ghajar [27]	Re-entrant	✗	✓	2900	3500	600	
	Square edge	✗	✓	3100	3700	800	
	Bell mouth	✗	✓	5100	6100	1000	
Osman et al. [30]	<i>Adiabatic friction factor</i>						
	Boundary layer chopping inlet	✓	✗	2100	3200	1100	
	<i>Diabatic friction factor</i>						
	Boundary layer chopping inlet	✓	✗	2067	3225	1158	
Authors and year	Type of Surface roughness	Developing flow	Developed flow	Important findings			
<i>Surface roughness</i>							
Meyer and Olivier [50]	Internal helical grooves	✓	✓	The kind of inlet employed in the case of smooth tubes had an impact on transition, with the smoothest inlet delaying transition the most			
Meyer and Olivier [51]	Internal helical grooves	✓	✓	The critical Reynolds numbers were unaffected by intake disturbances, contrary to the findings for adiabatic flow in Part I			
Everts et al. [52]	Surface microroughness	✗	✓	The transition began later for low relative surface roughness levels, but it started sooner when the relative surface roughness climbed to moderate and high values			
Vicente et al. [53]	Corrugation	–	–	Corrugation on the surface of test section causes the transition to start as early as Reynolds number 1300			
Vicente et al. [54]	Helical dimples	–	–	Enhancement in heat transfer and pressure drop was reported			

Table 1 (continued)

Authors and year	Type of turbulator or insert	heat flux		Critical Reyn-olds number (Start)	Critical Reyn-olds number (End)	Width of transition-flow regime
<i>Insertions or turbulators</i>						
Meyer and Abolarin [62]	Conventional twisted tapes	y = 3				
		2 kW m ⁻²		593	1125	532
		3 kW m ⁻²		819	1274	455
		4 kW m ⁻²		1093	1404	311
		y = 4				
		2 kW m ⁻²		669	1027	358
		3 kW m ⁻²		951	1418	467
		4 kW m ⁻²		1172	1524	352
		y = 5				
		2 kW m ⁻²		1023	1604	581
		3 kW m ⁻²		1108	1683	575
		4 kW m ⁻²		1278	1775	497
Abolarin et al. [63]	Peripheral u-cut twisted tape with and without ring inserts	1.35 and 2 kW m ⁻²	554	1037	483	
Abolarin et al. [64]	Clockwise and anticlockwise twisted tapes	1.35, 2, 3 and 4 kW m ⁻²	498	818	320	
Authors and year	Heat flux range	Flow type	Important findings			
<i>Heat flux</i>						
Saeedinia et al. [71]	6200–9900 W m ⁻²	Laminar-flow	Increasing the heat flux results in higher enhancement in the heat transfer coefficient			
Andrade et al. [72]	5.5–21.1 kW m ⁻²	Adiabatic and diabatic flow conditions	Compared to smooth tubes, corrugated tubes' friction factor demonstrated a smoother transition-flow regime			
Ghajar and Tam [34]	4–670 kW m ⁻²	Fully developed flow	The boundaries of transition were inlet dependent			
Tam and Ghajar [27]	3, 8 and 16 kW m ⁻²	Fully developed flow	Transition is heating flux dependent. Increase in heat flux causes delay in the start and end of the transition			
Meyer and Everts [74]	0–8 kW m ⁻²	Developing and developed flow	Increase in the Nusselt number and free convection effect was reported with increase in the heat flux and tube diameter			
Authors and year	Cross section of test section	Developing flow	Developed flow	Critical Reynolds number (Start)	Critical Reynolds number (End)	Width of transition-flow regime
<i>Geometry of test section</i>						
Chen et al. [75]	Transversely grooved duct	–	–	1300	6000	4700
Osman et al. [76]	Rectangular	✓	✗	1731	1876	145
Dirker et al. [77]	<i>Adiabatic friction factor</i>					
	Microchannel	✓	✗	1100	1800	700
	<i>Diabatic friction factor</i>					
	Microchannel	✓	✗	1250	2000	750
Authors and year	Angular position	Flow type		Important findings		
<i>Angular position of test section</i>						
Meyer et al. [82]	– 90° to + 90°	Fully developed flow		Critical Reynolds number for start of transition increase with an increase in inclination without affecting the termination critical Reynolds number		
Everts et al. [10]	90°	Fully developed upward and downward laminar flow		Deviation in Nusselt number from Nu = 4.36 was observed. The reason cited was the dominance of free convection effect		

Table 1 (continued)

Authors and year	Angular position	Flow type	Important findings
Saha and Paul [83]	0°, 15°, 30°, 45°, 60° and 75°	Forced convection of nanofluid	With inclination angle, distortion in flow field and turbulent kinetic energy was observed
Alzwayi and Paul [84]	0°, 10°, 30°, 50°, 70°, 80° and 85°	Natural convection of air	Heat transfer coefficient depends upon inclination as well as the width of channel
Al-Arabi and Sakr [85]	0°, 15°, 30°, 40°, 60° and 80°	Natural convection of air	Correlations for local heat transfer were used to estimate the transition regime
Kumar and Premachandran [86]	15°, 30° and 60°	Natural convection of air	Delay in transition with increase in the inclination angle was reported

required further investigation. It was highlighted that heat transfer enhancement is highest for horizontal channel and decreases with an increase in the angular orientation. This is because of the decreased buoyancy forces at inclined position and then the horizontal position.

Others

Yuan et al. [87] performed experiments inside a rectangular channel to assess the influence of rolling motion on heat transfer and flow characteristics to simulate offshore nuclear reactors. The study reveals increment in the heat transfer and friction factor when operated in the transition-flow regime. The fluctuation in the flow rate was quoted as the main cause of enhanced heat transfer. Demirkır and Ertürk [88] investigated the hydrothermal performance of heat exchanger tube carrying graphene–water nanofluid (0.025, 0.1 and 0.2 Vol%) as the working medium. It was found that nanofluid accelerates the transition by $Re = 2315$, while for water the transition begins at $Re = 2475$. Amplification in both heat transfer and pressure drop was reported. Ali et al. [89] performed experiments in a parallel plate solar collector with staggered plated for thermal and flow performance in transition-flow regime. Air is used as the working fluid. In addition to demonstrating that the local Nusselt number is not a unique function of the axial distance, the experimental findings revealed a linear connection between Re and the apparent friction factor. Bhuiyan et al. [90] numerically examined the heat transfer and fluid flow characteristics of a fin-and-tube heat exchanger. This research investigates the impact of geometrical features such fin pitch, longitudinal pitch and transverse pitch of tube spacing. With increasing longitudinal and transverse tube pitch spacing, heat transfer and friction factor are reduced for both laminar- and transition-flow regimes, while the efficiency index increases. Table 1 shows the critical Reynolds number for various cases discussed in this article, while Table 2 summarizes the discussion of this section for easy understanding of parameters and variables used for the investigation purpose. In order

to calculate the heat transfer coefficients in circular pipes for the three potential flow regimes of laminar, transition and turbulent flow, Abraham et al. [91] proposed a theory. For the range $Re > 4800$, it was noted that the experimental results strongly confirmed the predictions given by Gnielinski [92]. However, the experimental results corroborated Churchill's [93] predictions in the range of 2300–3100. Meyer et al. [94] investigated the effect of multi walled carbon nanotubes on hydrothermal performance in transition-flow regime. The experiments were carried out with volume concentrations (vol. conc.) of 0.33%, 0.75% and 1.0% at a constant heat flux of 13 kW m^{-2} . Higher vol. concentration causes the transition to start and ends early when compared with low vol. concentration nanofluid. The pressure drop data showed that when volume concentration increased, the pressure drop also increased, which is a consequence of the viscosity rising. Peixinho [95] investigated in transition-flow regime with Carbopol aqueous solution for thermal and flow performance. It was reported that with shear thinning and yield stress, the critical Reynolds number for transitional flow rises. Stamatoopoulos et al. [96] experimentally evaluated the performance of thermal sensor for gas flow in laminar-, transition- and turbulent-flow regime. Abraham et al. [97] numerically demonstrated the breakdown of laminar flow into transition and turbulent. Celata et al. [98] performed experiments in seven surface roughened microcircular pipes (30–500 μm) for gas flow and reported on the fluid flow behavior in terms of friction factor. For smooth tube, the transition begins at Reynolds number 2000, while for roughened tube it may delays up to 4500. The transition for microtubes having diameter less than 100 μm occurs at almost similar Reynolds numbers with marginal changes. However, for tubes with larger diameter, a delay in the start of transition was observed. Besides it was also found that for micropipes with lesser roughness, the transition was smooth, while for roughened pipes, transition was sharp. Yang et al. [99] also carried out similar nature of investigation with microtubes and reported that if the compressibility of gaseous fluid was taken well into consideration, then microtube

Table 2 Comparison of parameters, geometries of various experiments and their findings

Authors	Test section geometry	Inlet geometry	Working fluid	Developed or developing flow	Prandtl Number	Reynolds number range	Heat flux	Critical Reynolds number	Important findings
Ghajar and Madon [33]	Horizontal seamless 316 stainless steel	Re-entrant, square-edged, bell mouth	Combination of Distilled water and ethylene glycol	Fully developed	–	$500\text{--}15 \times 10^3$	–	Re-entrant = 1980–2600, Square-edged = 2070–2840, Bell mouth = 2125–3200	The growth of the friction factor throughout the pipe and the start and finish of the transition-flow regime are substantially influenced by the kind of inlet geometry
Ghajar and Tam [34]	Horizontal seamless 316 stainless steel	Re-entrant, square-edged, bell mouth	Combination of Distilled water and ethylene glycol	Fully developed	4–158	$280\text{--}49 \times 10^3$	4 to 670 kW m ⁻²	Re-entrant = 2000–8500, Square-edged = 2400–8800, Bell mouth = 3800–10,500	The inlet configuration significantly influences the heat transfer coefficient, as well as the onset and termination of the transition-flow regime. The inlet geometry also responsible for secondary flow in the flow field
Tam and Ghajar [27]	Horizontal seamless 316 stainless steel	Re-entrant, square-edged and bell mouth	Ethylene glycol–water mixtures	Isothermal fully developed	6–30	1×10^3 to 17×10^3	3, 8 and 16 kW m ⁻²	Re-entrant = 2900–3500, Square-edged = 3100–3700, Bell mouth = 5100–6100	With an increase in the heating rate and a given Reynolds number, the value of the fully developed friction factor rose. The bottom and upper limit of the isothermal transition boundaries increased as a result of this rise in friction factor

Table 2 (continued)

Authors	Test section geometry	Inlet geometry	Working fluid	Developed or developing flow	Prandtl Number	Reynolds number range	Heat flux	Critical Reynolds number	Important findings
Karwa et al. [105]	Rib-roughened rectangular duct	Rectangular inlet	Air	Thermally fully developed flow	–	3×10^3 to 2×10^4	1 kW	–	The presence of ribs at one wide wall of the duct results in increases in the Stanton number and friction factor of up to two and three times, respectively, as compared to the smooth duct
Zhang [59]	Cross-Corrugated Triangular Channels	Triangular	–	Fully Developed Flow	–	1×10^2 to 6×10^3	–	–	A complicated flow pattern with flow separation, reattachment and recirculation zones results from the corrugations. This result in higher Nusselt number and friction factor
Dirket et al. [77]	Rectangular channel	Sudden contraction, bell mouth and swirl inlet	Water	–	–	300–2800	–	sudden contraction = 1800–2000, bell mouth = 1800 and swirl inlet = 1500	In all flow regimes, it was discovered that the swirl inlet type changed the behavior of the Nusselt number and friction factor. Due to a drop in fluid viscosity near the wall, a decrease in friction factor was seen with an increase in heat input

Table 2 (continued)

Authors	Test section geometry	Inlet geometry	Working fluid	Developed or developing flow	Prandtl Number	Reynolds number range	Heat flux	Critical Reynolds number	Important findings
Ali et al. [89]	Parallel plate channels with staggered plates	–	Air	–	–	2580–4650	400, 700 and 1000 W m ⁻²	–	The most important configuration element that significantly affects the values of the apparent friction factor and the heat transfer coefficient (j) for shorter plate lengths is the plate thickness
Bhuiyan et al. [90]	Cross flow heat exchanger	–	Air	–	–	4×10^2 – 2×10^3	–	–	The thermal and hydraulic properties of fin-and-tube heat exchangers are significantly influenced by the tube configurations. Staggered tube arrangements create better flow mixing than inline arrangements do for laminar and transition flows, which results in enhanced heat transfer and pressure drop characteristics
Martinez et al. [73]	316L stainless steel tube with a wire coil insert	Circular	Pseudoplastic test fluids	Fully developed flow	150–1900	10–1200	–	–	Results indicate that the wire coils have little impact in the laminar domain, but seamlessly shift to turbulent flow at Re = 500

Table 2 (continued)

Authors	Test section geometry	Inlet geometry	Working fluid	Developed or developing flow	Prandtl Number	Reynolds number range	Heat flux	Critical Reynolds number	Important findings
Zhang et al. [106]	Plain plate-fin and tube	–	Air	–	–	1×10^3 – 5×10^3	–	–	With higher frontal velocities and fin pitches, the velocity oscillations become more pronounced. Heat transport is enhanced and pressure drop is increased by the powerful mixing action of vortices
Meyer et al. [50, 51]	Enhanced circular tube	Square-edged, re-entrant and bell-mouth	Water	Fully developed and developing flow	4.17–5.06	436–21,084	1 598–10 954 W	Starts at 2000 and ending at 3000	It was discovered that transition occurs at lower Reynolds numbers for corrugated tubes than for their smooth tube equivalents. In addition, the type of inlet employed affects transition, just as it does with smooth tubes

Table 2 (continued)

Authors	Test section geometry	Inlet geometry	Working fluid	Developed or developing flow	Prandtl Number	Reynolds number range	Heat flux	Critical Reynolds number	Important findings
Meyer et al. [94]	Straight copper tube	–	MWCNT–water nanofluid	–	–	1×10^3 – 8×10^3	13 kW m^{-2}	–	The nanofluids demonstrated a lower heat transfer coefficient than water when results from several experiments were compared at the same fluid velocity. The pressure drop data showed that when volume concentration increased, the pressure drop also increased, which is a consequence of the viscosity rising
Popov et al. [61]	Rough Tubes	–	Water	–	–	3×10^3 – 1×10^4	–	–	Result indicated enhancement in the heat transfer and pressure drop when tube with spherical protrusion is employed
Tam et al. [107]	Horizontal tube	square-edged and re-entrant inlets	Mixtures of ethylene glycol and water	Entrance and fully developed flow	4.3–46.3	8×10^2 – 22×10^3	4.3– 8.9 kW m^{-2}	Square-edged: 2222–3588, Re-entrant: 2032–3031	The start and end of transition were postponed by heating, which also reduced the laminar and transition friction factor in the entry and fully developed flow region
Garcia et al. [65]	316 L stainless steel tube with a wire coil insert	–	Water, propylene glycol USP grade and a mixture of propylene glycol and water	Fully developed flow	200–700	10–2500	–	1000–1300	Insertion of wire coil changes the critical Reynolds number for heat transfer and pressure drop

Table 2 (continued)

Authors	Test section geometry	Inlet geometry	Working fluid	Developed or developing flow	Prandtl Number	Reynolds number range	Heat flux	Critical Reynolds number	Important findings
Ghajar et al. [108]	Stainless steel tubes	–	distilled water	–	–	700–5500	–	1300 < Re < 1700, 1500 < Re < 4000	It has been discovered that decreasing tube diameters and increasing relative roughness have an impact on friction factor. This also led to early onset of transition in microtube
Meyer et al. [36]	Smooth tube	square-edged, re-entrant and a bell mouth inlet	water	–	4–6	1×10^3 to 2×10^4	–	square-edged = 2600, bell mouth inlet = 7000	The transition from laminar to turbulent flow was substantially influenced by the intake profile, according to adiabatic studies, and it may take up to 7,000 Reynolds numbers for the shift to occur
Meyer and Preez [109]	Counterflow tube-in-tube	–	water	–	–	1×10^3 to 5×10^3	–	–	Results for heat transfer and pressure drop were presented

Table 2 (continued)

Authors	Test section geometry	Inlet geometry	Working fluid	Developed or developing flow	Prandtl Number	Reynolds number range	Heat flux	Critical Reynolds number	Important findings
Oliver et al. [28]	Smooth tubes with	Hydrodynamically fully developed, re-entrant and bell mouth	Distilled water		4–6	1×10^3 – 2×10^4	4–61 kW m ⁻²	2100–2900	Adiabatic findings demonstrated that the intake profile greatly influenced the transition from laminar to turbulent flow, delaying it to Reynolds numbers as high as 12,000. Results for the diabatic heat transfer and friction factor indicated that the transition was independent of the inlet geometry and took place at a Reynolds number of around 2100
Naik and Sundar [110]	Circular Tube with Helical Inserts	–	water/propylene glycol-based CuO nanofluid	–	–	25×10^2 to 1×10^4	–	–	Enhancement in the pressure drop and heat transfer is reported
Bertsche et al. [111]	Circular pipe	–	water–glycol mixture	–	7–41	5×10^2 – 23×10^3	–	2300–4000	Improvement in the heat transfer and pressure drop was reported
Ndenguma et al. [112, 113]	Tube-in-tube heat exchanger	–	water	hydrodynamically fully develop flow	–	100–14000	–	350 and 500 for a cooled annulus and between 430 and 510 for a heated annulus	In comparison to the transition based on friction factor, the flow regime transition based on the Nusselt number began sooner

Table 2 (continued)

Authors	Test section geometry	Inlet geometry	Working fluid	Developed or developing flow	Prandtl Number	Reynolds number range	Heat flux	Critical Reynolds number	Important findings
Everts and Meyer [114]	Smooth horizontal tube	square-edged inlet	water	Developing and fully developed flow		$7 \times 10^2 - 1 \times 10^4$	$0 - 3 \text{ kW m}^{-2}$	2540	A relationship between heat transfer and pressure drop is confirmed experimentally, and the boundary of transition is same for both heat transfer and pressure drop
Everts and Meyer [115]	smooth circular test section	square-edged inlet	water	Developing and fully developed flow	3–7	500–10000	$0 - 3 \text{ kW m}^{-2}$	2361	
Chen et al. [75]	transversely grooved tube using rectangular channel	circular	molten salt	-	11–27	$3 \times 10^2 - 6 \times 10^4$	-	1300	
Osman et al. [76]	rectangular channel	rectangular	aluminum oxide–water nanofluid	-	-	$2 \times 10^2 - 7 \times 10^3$	current of 0.75 A and voltage of 125 V, 200 W power supply	1731 for the 0.3 vol% nanofluid, 1723 for the 0.5 vol% nanofluid, and 1705 for the 1.0 vol% nanofluid	
Meyer and Abolarin [26, 62–64]	Smooth circular tube with and without turbulators	Square-edged inlet, re-entrant inlet	water	Fully developed flow	2.9–6.7, 3.23–6.5	400–11400, 300–11400, 315–11,404, 653 to 10 553	1.35, 2, 3 and 4 kW m^{-2}	3053–3331, 3164–3452, 3266–3565	
Bashir et al. [116]	Circular tube	Square-edged and re-entrant inlets	Water	Hydro dynamically fully developed inlet	-	1×10^3 to 6×10^3	4, 6 and 8 kW m^{-2}	2300 and 2280 for the empty and original flow-calming section, respectively	
Bashir et al. [117]	circular tube	Circular inlet	Water	Developing and fully developed flow	3.5 and 8.1	$4 \times 10^2 - 6 \times 10^3$	$1 - 8 \text{ kW m}^{-2}$	-	

Table 2 (continued)

Authors	Test section geometry	Inlet geometry	Working fluid	Developed or developing flow	Prandtl Number	Reynolds number range	Heat flux	Critical Reynolds number	Important findings
Meyer et al. [82]	Smooth inclined tubes	Circular inlet	Water	Developing and fully developed flow	3–7	1×10^3 – 6×10^3	4 – $8 \text{ kW}\cdot\text{m}^{-2}$	for horizontal flow ($\theta=0^\circ$), 2733–3070	It was discovered that when the inclination angles grew from horizontal to vertical, the critical Reynolds numbers at which transition began increased

behaves similar to conventional tube for heat transfer and fluid flow characteristics. Meyer et al. [100] experimented to understand the flow and heat transfer characteristics of multiwalled carbon nanotubes in transition-flow regime. It was found that transition initiated earlier for 1% volume concentration of nanotubes while it delays at lower concentrations. Varnaseri and Peyghambarzadeh [101] reported on the influence of polyacrylamide on the friction factor and thermal performance. The inclusion of the drag-reducing polymer lowers heat transfer, but supplying a steady heat flux to the pipe minimizes the polymer's influence on the percentage of heat transfer reduction, according to the results. Briclot et al. [102] reported on the thermal and flow performance of Al_2O_3 -water nanofluid in the transition-flow regime. It was reported that mass percentage does not impact the critical Reynolds number and nanofluid acts as a single-phase fluid. Liu et al. [103] experimented with printed circuit heat exchanger for large range of Reynolds numbers. The working fluid used for the experimentation was supercritical carbon dioxide. According to the results, the essential Reynolds values for the semi-circular channel are 2100 and 2700, respectively, to transition from laminar- to turbulent-flow regime. Jamali and Toghraie [104] studied the thermal performance characteristics of nanofluid computationally for different inlets. Their results indicate that transition for square edge inlets starts later than that of smooth tube.

The studies reviewed offer valuable insights into heat transfer and flow characteristics across various fluid types, configurations and flow regimes. They highlight enhanced heat transfer and flow fluctuations in offshore nuclear reactors, accelerated transitions with nanofluids and the impact of geometrical features on heat-exchanger performance. Theoretical models for calculating heat transfer coefficients and experimental investigations into carbon nanotubes' effects on thermal performance provide further understanding. Additionally, exploration of shear thinning and yield stress effects on transitional flow, surface roughness influence on fluid behavior and nanofluid behavior in thermal systems are discussed. These findings contribute to a broader understanding of thermal system optimization and design considerations across diverse applications.

Correlation for heat transfer and pressure drop

Correlations are developed using the data obtained from the experimental investigation for predicting the values. These correlations also show the dependence of different parameters used in the investigation on the Nusselt number and friction factor. Several factors influence heat transfer and pressure drop which includes geometrical parameters, velocity of flow, Reynolds number, Prandtl number, etc. Various

Table 3 Correlations for Nusselt number and friction factor proposed by various researchers

S.N	Author and year of publication	Nusselt number	Friction factor
1	Ghajar and Madon [33]	–	For re-entrant inlet $C_{f,ir} = -9.88 * 10^{-3} + 1.15 * 10^{-5} Re - 1.29 * 10^{-9} Re^2$ For square edge inlet $C_{f,ir} = -2.56 * 10^{-2} + 2.49 * 10^{-5} Re - 4.25 * 10^{-9} Re^2$ For bell mouth inlet $C_{f,ir} = -8.03 * 10^{-3} + 1.05 * 10^{-5} Re - 1.47 * 10^{-9} Re^2$ –
2	Ghajar and Tam [34]	$Nu_{tr} = \{Nu_l + \exp[(a - Re)/b] + Nu_l^c\}^c$ For re-entrant inlet $a = 1766$, $b = 276$, and $c = -0.955$ For square edge inlet $a = 2617$, $b = 207$, and $c = -0.905$ For bell mouth inlet $a = 6628$, $b = 237$, and $c = -0.980$ –	$C_f = \left[1 + \left(\frac{Re}{a} \right)^b \right]^c \left(\frac{h_w}{\mu_w} \right)^m$ For re-entrant inlet $a = 5840$, $b = -0.0145$, and $c = -6.23$ For square edge inlet $a = 4230$, $b = -0.16$, and $c = -6.57$ For bell mouth inlet $a = 5340$, $b = -0.0990$, and $c = -6.32$ for $5 \leq e^+ < 20$ $R = 1.66e^{-0.0078\phi} (W/H)^{-0.4} \left(\frac{p}{e} \right)^{2.695} * \exp \left[-0.762 \{ \ln(p/e) \}^2 \right] (e^+)^{-0.075}$ for $20 \leq e^+ \leq 60$ $R = 81.325e^{-0.0078\phi} (W/H)^{-0.4} \left(\frac{p}{e} \right)^{2.695} * \exp \left[-0.762 \{ \ln(p/e) \}^2 \right]$ $f_a = 5.76(e/d)^{0.95} (p/d)^{-1.21} Re^{-0.217}$ $f_a = 9.35(p/e)^{-1.16} Re^{-0.217}$ –
3	Tam and Ghajar [27]	–	–
4	Karwa et al. [105]	for $7 \leq e^+ < 20$ $g = 103.77e^{-0.006\phi} (W/H)^{0.5} \left(\frac{p}{e} \right)^{-2.56} * \exp \left[0.7343 \{ \ln(p/e) \}^2 \right] (e^+)^{-0.31}$ for $20 \leq e^+ \leq 60$ $g = 32.26e^{-0.006\phi} (W/H)^{0.5} \left(\frac{p}{e} \right)^{-2.56} * \exp \left[0.7343 \{ \ln(p/e) \}^2 \right] (e^+)^{0.08}$	–
5	Gracia et al. [118]	$Nu_a = 0.303(e/d)^{0.12} (p/d)^{-0.377} Re^{0.72} Pr^{0.37}$ $Nu_a = 0.132(p/d)^{-0.372} Re^{0.72} Pr^{0.37}$	–
6	Gracia et al. [65]	$Nu = 0.104 Re^{-1.01} (Re = 2 * 10^2 - 10^3, Pr = 200 - 650)$ $Nu = 0.100 Re^{-0.99} (Re = 2.5 * 10^2 - 1.1 * 10^3, Pr = 350 - 650)$ $Nu = 0.077 Re^{-1.03} (Re = 5 * 10^2 - 1.1 * 10^3, Pr = 350 - 600)$	–
7	Meyer and Olivier [50]	–	$f_{te} = 4 \left(\frac{16}{Re_{cr}} \right)^{0.94} \exp \left(0.57 \frac{Re}{Re_{cr}} \right) \left(\frac{\rho}{90} \right)^{0.37} \times \left(\frac{e^2}{\rho D_e} \right)^{0.028} \left(\frac{p_L}{D_e} \right)^{-0.009} \left(\frac{e}{D_e} \right)^{0.06}$ –
8	Meyer and Olivier [51]	$Nu_{te} = [Nu_{Le}^7 + Nu_{Te_{38}}^7]^{1/7}$	–

Table 3 (continued)

S.N	Author and year of publication	Nusselt number	Friction factor
9	Tam et al. [107]	–	$C_{f,ir,iso} = \left(\frac{16}{Re}\right) \left\{ \left[1 + (0.0049 Re^{0.75})^{a_1} \right]^{\frac{1}{a}} + b \right\}$ • Re-entrant: $a = 0.52, b = -3.47$ for $2026 < Re < 3257.3 < x/D < 200$ • Square-edged: $a = 0.50, b = -4.0$ for $2111 < Re < 4141.3 < x/D < 200$ $f_{Reg} = 0.18 Re^{-0.18} (1 + \varphi)^{0.23} \left(1 + \frac{H}{D_c} \right)^{-0.09} \left(\frac{D}{D_c} \right)^{0.3}$
10	Naik and Sundar [110]	$Nu_{Reg} = 0.32 Re^{0.45} Pr^{0.48} (1 + \varphi)^{0.32} \left(1 + \frac{H}{D_c} \right)^{-0.3} \times \left(\frac{D}{D_c} \right)^{1.27}$	–
11	Taler [119]	$Nu_{n,q,3} = 0.924 Pr^{1/3} \left(Re \frac{d_h}{L} \right)^{1/2}$ $Nu_{n,T,3} = \left(\frac{2}{1+22Pr} \right)^{1/6} \left(Re Pr \frac{d_h}{L} \right)^{1/2}$	–
12	Ndenguma et al. [112]	$Nu = C_1 \tau^n \text{ where } C_1 = B_1 \left(\frac{GrPr}{Re} \right)^{z_1} \text{ and } n = B_2 \left(\frac{GrPr}{Re} \right)^{z_2}$	$f_d = f_{iso} C_2 \left(\frac{\mu_n}{\mu_w} \right) \tau^m$ where $C_2 = E_1 (Gr^{\mu} Pr^{\nu})^{z_2}$ $m = E_2 (Gr^{\mu} Pr^{\nu}) + E_3$ $f_{iso} = C_{iso}(\lambda) Re^{-m(i)}$ $f_d = f_{iso} \lambda^2 C_d \left(\frac{d_h}{\mu_w} \right)$ For heated annuli: $C_d = 46.6 \times 10^6 (Gr^{0.01} Pr^{2.9})^{-3.12}$ $z = 3.8 \times 10^{-3} (Gr^{0.01} Pr^{2.9}) - 0.88$ For cooled annuli: $C_d = 10.95 (Gr^{0.01} Pr^{2.9})^{0.19}$ $z = -0.14 \times 10^{-3} (Gr^{0.01} Pr^{2.9}) - 0.45$ $f = \left(\frac{3.74 Re - 8.066 \times 10^3}{Re - 2.320 \times 10^3} \right) \frac{Nu}{Re Pr^{0.087}}$
13	Ndenguma et al. [113]	$Nu = C \lambda^{-n}$ Where, $\lambda = aL/D_h$ For heated annuli: $C = 134 \left(\frac{GrPr}{Re} \right)^{0.401}$ $n = 0.32 \left(\frac{GrPr}{Re} \right)^{0.147}$ For cooled annuli: $C = 1183 \left(\frac{GrPr}{Re} \right)^{0.28} n = 0.496 \left(\frac{GrPr}{Re} \right)^{0.122}$	–
14	Everts and Meyer [115]	$Nu = (0.00108 Re - 2.49) Gr^{-0.04} Pr^2$	–
15	Chen et al. [75]	$[Nu - 0.3]^{3.1142} = \left[\frac{0.62 Re^{1/2} Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \right]^{2.76} + \left[\frac{0.001168 Re Pr^{1/3}}{[1 + (0.4/Pr)^{2/3}]^{1/4}} \right]^{4.9847}$	–
16	Meyer and Everts [74]	$Nu = 4.36 + (Nu_1^6 + Nu_2^6)^{1/6}$ $Nu_1 = (0.33 Gz^{0.54} - 0.84) Pr^{-0.2} Nu_2 = (0.207 Gz^{0.305} - 1.19) Pr^{0.5} Gz^{-0.08}$ $Nu = 4.36 + (Nu_1^6 + Nu_2^6)^{1/6}$ $Nu_1 = (0.33 Gz^{0.54} - 0.84) Pr^{-0.2} Nu_2 = (0.202 Gz^{0.254} - 1.23) Pr^{0.45} Gz^{-0.06}$	–
17	Mortean and Mantelli [120]	$Nu_{tran} = \left[(Nu_{lam})^6 + (Nu_{urb})^6 \right]^{1/6}$	–
18	Meyer et al. [121]	$Nu = (0.017 Re - 30.3) Pr^{0.33} Gr^{-0.08}$	–
19	Bashir et al. [117]	$Nu_{FC} = 4.36 + 5.36 \times 10^{-9} Re^{2.39}$	$f_{FC} = \frac{Nu_{FC} (15.88 - 0.0014 Re)}{Re}$

correlations developed in the past for predicting the heat transfer and pressure drop in the transition-flow regime are summarized in the tabulated format in Table 3 for the convenience of the future reader of this article.

Discussion

Understanding the thermohydraulics of fluid flow in the transition-flow regime is becoming a necessity. This brings sudden momentum in the field of heat transfer and pressure drop in the transition-flow regime. Although the field is in the early stage of research, the effort made by researchers helps in the better understanding of the heat transfer and flow dynamics in the transition-flow regime. The research group of Ghajar initially begins to investigate the thermohydraulic of fluid flow in transition-flow regime with a motive to gather the data for the better realization of the physics. The group carried out a series of experiments and collected data for different operation conditions such as inlet geometry, uniform and variable heat flux, developed and developing flow, etc. Following the path of Ghajar and co-worker and understanding the gap in the research, Meyer and co-workers further investigate the transition-flow regime for different fluids, heat flux, channel geometries, channel inclination, turbulators, etc. The results obtained by Meyer and co-workers are remarkable in the field of heat transfer and fluid flow. The correlations developed by Meyer and co-workers are very much appreciated and challenge the previous correlations. The use of nanofluids and turbulators was also first investigated in this group which shed light on the flow dynamics. The clear shift in the transition-flow regime with changes in the geometry, flow condition, position, etc., was clearly notified and explained the detail. One of the main findings is the change in the value of Nusselt number for forced-convection heat transfer with an increase in Reynolds number which otherwise considered constant. This finding challenges the existing correlations and brings the need for changes in them.

After carefully reviewing all the articles related to transition flow heat transfer and pressure drop, it can be said that transition depends on lots of factors. Geometry of inlet, type of fluid, heat flux, turbulators, orientation of test channel, flow directions, etc., are the factors influencing the onset and cessation of the transition. The widely accepted range of transition is between 2100 and 2300 Reynolds numbers. However, after the research of almost three decades by several research groups, it can be said that start and end of transition changes with change in the operating conditions. It can start as early as $Re = 600$ and can be delayed up to $Re = 10,000$.

Therefore, it can be said that understanding the thermohydraulics of fluid flow in the transition-flow regime

is crucial for advancing heat transfer and pressure drop knowledge in the transition-flow regime. Research efforts by various groups, such as Ghajar and Meyer, contribute significantly to the understanding of heat transfer and flow dynamics in transition-flow regimes. Factors such as inlet geometry, fluid type, heat flux and channel orientation play significant roles in influencing the onset and cessation of transition. The research highlights that the transition range can vary depending on operating conditions, challenging the widely accepted transition range of 2100–2300 Reynolds numbers.

In addition, it is also pointed out that while some research suggests that transition typically occurs within the range of 2100–2300 Reynolds numbers, other findings indicate that transition can start as early as $Re = 600$ and be delayed up to $Re = 10,000$, suggesting a broader variability in transition onset and end points. The research by Meyer and co-workers challenges existing correlations, indicating a need for updating and revising them to accommodate the observed variations in transition behavior. There may be differing opinions on the specific factors contributing to transition onset and cessation, with some emphasizing the importance of inlet geometry, fluid type and heat flux, while others may prioritize other factors such as channel inclination or flow direction. Despite the agreement on the significance of understanding transition-flow regimes, there may be differing perspectives on the implications and applications of this knowledge, depending on the context and objectives of the research.

Conclusions

The prime intent of present review work is to summarize all the research works which studied the heat transfer and fluid flow characteristics in single-phase transition flow and compile them together to create a coherent body for ease for future readers and researchers. After reviewing various articles on developing and fully developed flow, it can be said that factors such as the geometry of inlet, type of fluid, heat flux, turbulators, orientation of test channel, flow directions, etc., remarkably influence the onset and cessation of the transition in the flow field. Due to the higher velocity of the fluid, the laminar–turbulent transition happens quicker throughout the tube length as the Reynolds number rises. When opposed to forced convection circumstances, free or natural convection effects first disrupt the fluctuations inside the test section, resulting in a delayed laminar–turbulent transition. The laminar–turbulent transition throughout the tube length occurs faster as a result of the improved mixing, and the magnitude of the Nusselt numbers escalates. The properties of developing

and fully developed flow, particularly in the transition-flow regime, differ substantially in terms of heat transmission and pressure drop.

Scope for future works

Investigations in the transition-flow regime are still in the early stage and need to be explored extensively for improved understanding of fluid flow and heat transfer in this intricate flow regime. This flow regime is known for its unsteady, chaotic and unrealistic flow behavior. The following recommendations for future work are outlined below:

1. The existing studies have predominantly focused on constant heat flux conditions. It is recommended to investigate the impact of constant wall temperature on the transition-flow regime for both heating and cooling scenarios.
2. Research on transition-flow regimes with turbulators remains limited, necessitating thorough investigations with various turbulator designs and inlet obstructions.
3. The influence of channel cross section has been omitted in previous literature and should be incorporated in future research.
4. The impact of orientation and flow direction on the transition-flow regime with inserted turbulators should be investigated in future studies.

References

1. Sharaf MA, Marzouk SA, Aljabr A, Almehmadi FA, Kaood A, Alqaed S. Heat transfer enhancement in a double-pipe helical heat exchanger using spring wire insert and nanofluid. *J Therm Anal Calorim*. 2024. <https://doi.org/10.1007/s10973-024-12992-1>.
2. Selvam L, Aruna M, Hossain I, Venkatesh R, Karthigairajan M, Prabakaran S, Mohanavel V, Seikh AH, Kalam MA. Impact of hybrid nanofluid on thermal behavior of flat-plate solar collector: performance study. *J Therm Anal Calorim*. 2024. <https://doi.org/10.1007/s10973-024-12994-z>.
3. Wadi A, Fatlawi A, Niazmand H. Thermal analysis of hybrid nanofluids inside a microchannel heat exchanger for electronic cooling. *J Therm Anal Calorim*. 2024. <https://doi.org/10.1007/s10973-024-12991-2>.
4. Varma A, Nidhul K. Thermo-hydraulic and entropy generation analysis of nanofluid flow with variable properties in various duct cross sections: a 3-D two-phase approach. *J Therm Anal Calorim*. 2024. <https://doi.org/10.1007/s10973-024-12940-z>.
5. Sedighi S, Abdollahi A, Homayoon A, Isfahani M. Experimental investigation of ultrasonic effect on natural and forced heat transfer in heat sinks. *J Therm Anal Calorim*. 2024. <https://doi.org/10.1007/s10973-024-12880-8>.
6. Sheikholeslami M, Farshad SA. Numerical simulation of effect of non-uniform solar irradiation on nanofluid turbulent flow. *Int Commun Heat Mass Transf*. 2021. <https://doi.org/10.1016/j.icheatmasstransfer.2021.105648>.
7. Sheikholeslami M, Jafaryar M. Thermal assessment of solar concentrated system with utilizing CNT nanoparticles and complicated helical turbulator. *Int J Therm Sci*. 2023. <https://doi.org/10.1016/j.ijthermalsci.2022.108015>.
8. Sheikholeslami M, Said Z, Jafaryar M. Hydrothermal analysis for a parabolic solar unit with wavy absorber pipe and nanofluid. *Renew Energy*. 2022. <https://doi.org/10.1016/j.renene.2022.02.086>.
9. Sheikholeslami M. Numerical investigation of solar system equipped with innovative turbulator and hybrid nanofluid. *Sol Energy Mater Sol Cells*. 2022. <https://doi.org/10.1016/j.solmat.2022.111786>.
10. Everts M, Bhattacharyya S, Bashir AI, Meyer JP. Heat transfer characteristics of assisting and opposing laminar flow through a vertical circular tube at low Reynolds numbers. *Appl Therm Eng*. 2020;179: 115696. <https://doi.org/10.1016/j.appltherm.2020.115696>.
11. Asaadi S, Abdi H. Numerical investigation of laminar flow and heat transfer in a channel using combined nanofluids and novel longitudinal vortex generators. *J Therm Anal Calorim*. 2020. <https://doi.org/10.1007/s10973-020-09795-5>.
12. Ebrahimnia-Bajestan E, Niazmand H, Duangthongsuk W, Wongwises S. Numerical investigation of effective parameters in convective heat transfer of nanofluids flowing under a laminar flow regime. *Int J Heat Mass Transf*. 2011;54:4376–88. <https://doi.org/10.1016/j.ijheatmasstransfer.2011.05.006>.
13. Almohammadi H, Vatan SN, Esmaeilzadeh E, Motezaker A, Nokhosteen A. Experimental investigation of convective heat transfer and pressure drop of Al_2O_3 /water nanofluid in laminar flow regime inside a circular tube. *World Acad Sci Eng Technol*. 2012;68:2208–13.
14. Heyhat MM, Kowsary F, Rashidi AM, Momenpour MH, Amrollahi A. Experimental investigation of laminar convective heat transfer and pressure drop of water-based Al_2O_3 nanofluids in fully developed flow regime. *Exp Therm Fluid Sci*. 2013;44:483–9. <https://doi.org/10.1016/j.expthermflusci.2012.08.009>.
15. Feng Z, Ai X, Wu P, Lin Q, Huang Z. Experimental investigation of laminar flow and heat transfer characteristics in square minichannels with twisted tapes. *Int J Heat Mass Transf*. 2020;158: 119947. <https://doi.org/10.1016/j.ijheatmasstransfer.2020.119947>.
16. Bain A, Languri E, Davidson J, Kerns D, Costa L. Experimental evaluation of dielectric oil functionalized nanodiamond suspension under laminar flow regime. *Intersociety conference thermal thermomechanical phenomena electronic systems ITherm*. 2020:276–282. <https://doi.org/10.1109/ITherm45881.2020.9190606>.
17. Hwang KS, Jang SP, Choi SUS. Flow and convective heat transfer characteristics of water-based Al_2O_3 nanofluids in fully developed laminar flow regime. *Int J Heat Mass Transf*. 2009;52:193–9. <https://doi.org/10.1016/j.ijheatmasstransfer.2008.06.032>.
18. Shah RK, London AL. circular duct, laminar flow forced convection ducts. 1978;16:78–152. <https://doi.org/10.1016/b978-0-12-020051-1.50010-3>.
19. Shah RK, London AL. Rectangular ducts, laminar flow forced convection ducts. 1978:196–222. <https://doi.org/10.1016/b978-0-12-020051-1.50012-7>.
20. Shah RK, London AL. Elliptical ducts, laminar flow forced convection ducts. 1978;247:247–252. <https://doi.org/10.1016/b978-0-12-020051-1.50014-0>.
21. Shah RK, London AL. Parallel plates, laminar flow forced convection ducts. 1978;2:153–195. <https://doi.org/10.1016/b978-0-12-020051-1.50011-5>.

22. Shah RK, London AL. Triangular ducts, laminar flow forced convection ducts. 1978;223–246. <https://doi.org/10.1016/b978-0-12-020051-1.50013-9>.
23. Bhattacharyya S, Saha SK. Thermohydraulics of laminar flow through a circular tube having integral helical rib roughness and fitted with centre-cleared twisted-tape. *Exp Therm Fluid Sci*. 2012;42:154–62. <https://doi.org/10.1016/j.expthermflusci.2012.05.002>.
24. Chang SW, Jan YJ, Liou JS. Turbulent heat transfer and pressure drop in tube fitted with serrated twisted tape. *Int J Therm Sci*. 2007;46:506–18. <https://doi.org/10.1016/j.ijthermalsci.2006.07.009>.
25. Eiamsa-ard S, Wongcharee K, Eiamsa-ard P, Thianpong C. Thermohydraulic investigation of turbulent flow through a round tube equipped with twisted tapes consisting of centre wings and alternate-axes. *Exp Therm Fluid Sci*. 2010;34:1151–61. <https://doi.org/10.1016/j.expthermflusci.2010.04.004>.
26. Abolarin SM. Heat transfer and pressure drop characteristics in the transitional flow regime of twisted tape insert, 2018. <https://doi.org/10.1615/ihct.16.her.023213>.
27. Tam LM, Ghajar AJ. Effect of inlet geometry and heating on the fully developed friction factor in the transition region of a horizontal tube. *Exp Therm Fluid Sci*. 1997;15:52–64. [https://doi.org/10.1016/S0894-1777\(97\)00035-6](https://doi.org/10.1016/S0894-1777(97)00035-6).
28. Olivier JA, Meyer JP. Single-phase heat transfer and pressure drop of the cooling of water inside smooth tubes for transitional flow with different inlet geometries (RP-1280). *HVAC&R Res*. 2010;16:471–96. <https://doi.org/10.1080/10789669.2010.10390916>.
29. Ghajar AJ. Heat transfer and pressure drop in the transition region of smooth horizontal circular tubes with different inlet configurations. In: *Advanced Heat Transfer*. Elsevier Ltd; 2019. pp. 1–53. <https://doi.org/10.1016/bs.aiht.2019.05.001>.
30. Osman S, Sharifpur M, Meyer JP. The effect of chopping the boundary layer at the inlet on the transition heat transfer and pressure drop characteristics in smooth horizontal tube. In: *14th International conference heat transfer fluid mechanics thermodynamics*. Wicklow, Ireland; 2019.
31. Meyer JP, Everts M, Hall ATC, Mullock-Houwer FA, Joubert M, Pallent LMJ, Vause ES. Inlet tube spacing and protrusion inlet effects on multiple circular tubes in the laminar, transitional and turbulent flow regimes. *Int J Heat Mass Transf*. 2018;118:257–74. <https://doi.org/10.1016/j.ijheatmasstransfer.2017.10.125>.
32. Nagendra HR. Interaction of free and forced convection in horizontal tubes in the transition regime. *J Fluid Mech*. 1973;57:269. <https://doi.org/10.1017/S0022112073001151>.
33. Ghajar AJ, Madon KF. Pressure drop measurements in the transition region for a circular tube with three different inlet configurations. *Exp Therm Fluid Sci*. 1992;5:129–35. [https://doi.org/10.1016/0894-1777\(92\)90062-A](https://doi.org/10.1016/0894-1777(92)90062-A).
34. Ghajar AJ, Tam L-M. Heat transfer measurements and correlations in the transition region for a circular tube with three different inlet configurations. *Exp Therm Fluid Sci*. 1994;8:79–90. [https://doi.org/10.1016/0894-1777\(94\)90075-2](https://doi.org/10.1016/0894-1777(94)90075-2).
35. Ghajar AJ, Tam LM. Flow regime map for a horizontal pipe with uniform wall heat flux and three inlet configurations. *Exp Therm Fluid Sci*. 1995;10:287–97. [https://doi.org/10.1016/0894-1777\(94\)00107-J](https://doi.org/10.1016/0894-1777(94)00107-J).
36. Meyer JP, Liebenberg L, Olivier JA. Pressure drop inside a smooth tube with different inlet geometries in the transitional flow regime for water cooled at a constant wall temperature. In: Janusz S, Szmyd, editors. *7th world conference experimental heat transfer fluid mechanics thermodynamics*. AGH University of Science and Technology Press, Kraków, 2009, Krakow; 2009. pp. 1265–1272.
37. Tam LM, Ghajar AJ. The unusual behavior of local heat transfer coefficient in a circular tube with a bell-mouth inlet. *Exp Therm Fluid Sci*. 1998;16:187–94. [https://doi.org/10.1016/S0894-1777\(97\)10019-X](https://doi.org/10.1016/S0894-1777(97)10019-X).
38. Xiao H, Liu Z, Liu W. Turbulent heat transfer enhancement in the mini-channel by enhancing the original flow pattern with v-ribs. *Int J Heat Mass Transf*. 2020;163: 120378. <https://doi.org/10.1016/j.ijheatmasstransfer.2020.120378>.
39. Maithani R, Saini JS. Heat transfer and friction factor correlations for a solar air heater duct roughened artificially with V-ribs with symmetrical gaps. *Exp Therm Fluid Sci*. 2016;70:220–7. <https://doi.org/10.1016/j.expthermflusci.2015.09.010>.
40. Nuntadusit C, Wae-hayee M, Bunyajitradulya A, Eiamsa-ard S. Thermal visualization on surface with transverse perforated ribs. *Int Commun Heat Mass Transf*. 2012;39:634–9. <https://doi.org/10.1016/j.icheatmasstransfer.2012.03.001>.
41. Sparrow EM, Gorman JM, Friend KS, Abraham JP. Flow regime determination for finned heat exchanger surfaces with dimples/protrusions. *Numer Heat Transf Part A Appl*. 2013;63:245–56. <https://doi.org/10.1080/10407782.2013.730450>.
42. Yadav S, Kaushal M, Varun, Siddhartha, Nusselt number and friction factor correlations for solar air heater duct having protrusions as roughness elements on absorber plate. *Exp Therm Fluid Sci*. 2013;44:34–41. <https://doi.org/10.1016/j.expthermflusci.2012.05.011>.
43. Kumar S, Verma SK. Heat transfer and fluid flow analysis of sinusoidal protrusion rib in solar air heater. *Int J Therm Sci*. 2022;172: 107323. <https://doi.org/10.1016/j.ijthermalsci.2021.107323>.
44. Huang Z, Yu GL, Li ZY, Tao WQ. Numerical study on heat transfer enhancement in a receiver tube of parabolic trough solar collector with dimples, protrusions and helical fins. *Energy Procedia*. 2015;69:1306–16. <https://doi.org/10.1016/j.egypro.2015.03.149>.
45. Goel V, Kumar R, Bhattacharyya S, Tyagi VV, Abusorrah AM. A comprehensive parametric investigation of hemispherical cavities on thermal performance and flow-dynamics in the triangular-duct solar-assisted air-heater. *Renew Energy*. 2021. <https://doi.org/10.1016/j.renene.2021.04.006>.
46. Wang Y, He YL, Lei YG, Zhang J. Heat transfer and hydrodynamics analysis of a novel dimpled tube. *Exp Therm Fluid Sci*. 2010;34:1273–81. <https://doi.org/10.1016/j.expthermflusci.2010.05.008>.
47. Patil PM, Doddagoudar SH, Hiremath PS. Impacts of surface roughness on mixed convection nanofluid flow with liquid hydrogen/nitrogen diffusion. *Int J Numer Methods Heat Fluid Flow*. 2019;29:2146–74. <https://doi.org/10.1108/HFF-11-2018-0703>.
48. Tran N, Sajjad U, Lin R, Wang CC. Effects of surface inclination and type of surface roughness on the nucleate boiling heat transfer performance of HFE-7200 dielectric fluid. *Int J Heat Mass Transf*. 2020;147: 119015. <https://doi.org/10.1016/j.ijheatmasstransfer.2019.119015>.
49. Lu H, Xu M, Gong L, Duan X, Chai JC. Effects of surface roughness in microchannel with passive heat transfer enhancement structures. *Int J Heat Mass Transf*. 2020;148: 119070. <https://doi.org/10.1016/j.ijheatmasstransfer.2019.119070>.
50. Meyer JP, Olivier JA. Transitional flow inside enhanced tubes for fully developed and developing flow with different types of inlet disturbances: part I—adiabatic pressure drops. *Int J Heat Mass Transf*. 2011;54:1587–97. <https://doi.org/10.1016/j.ijheatmasstransfer.2010.11.027>.
51. Meyer JP, Olivier JA. Transitional flow inside enhanced tubes for fully developed and developing flow with different types of inlet disturbances: part II—heat transfer. *Int J Heat Mass Transf*.

- 2011;54:1598–607. <https://doi.org/10.1016/j.ijheatmasstransfer.2010.11.026>.
52. Everts M, Robbertse P, Spitholt B. The effects of surface roughness on fully developed laminar and transitional flow friction factors and heat transfer coefficients in horizontal circular tubes. *Int J Heat Mass Transf.* 2022;189: 122724. <https://doi.org/10.1016/j.ijheatmasstransfer.2022.122724>.
 53. Vicente PG, García A, Viedma A. Mixed convection heat transfer and isothermal pressure drop in corrugated tubes for laminar and transition flow. *Int Commun Heat Mass Transf.* 2004;31:651–62. [https://doi.org/10.1016/S0735-1933\(04\)00052-1](https://doi.org/10.1016/S0735-1933(04)00052-1).
 54. Vicente PG, García A, Viedma A. Heat transfer and pressure drop for low Reynolds turbulent flow in helically dimpled tubes. *Int J Heat Mass Transf.* 2002;45:543–53. [https://doi.org/10.1016/S0017-9310\(01\)00170-3](https://doi.org/10.1016/S0017-9310(01)00170-3).
 55. Srinivasan V, Christensen RN. Experimental investigation of heat transfer and pressure drop characteristics of flow through spirally fluted tubes. *Exp Therm Fluid Sci.* 1992;5:820–7. [https://doi.org/10.1016/0894-1777\(92\)90126-P](https://doi.org/10.1016/0894-1777(92)90126-P).
 56. Obot NT, Esen EB, Snell KH, Rabas TJ. Pressure drop and heat transfer characteristics for air flow through spirally fluted tubes. *Int Commun Heat Mass Transf.* 1992;19:41–50. [https://doi.org/10.1016/0735-1933\(92\)90062-M](https://doi.org/10.1016/0735-1933(92)90062-M).
 57. Olsson C-O, Sundén B. Experimental study of flow and heat transfer in rib-roughened rectangular channels. *Exp Therm Fluid Sci.* 1998;16:349–65. [https://doi.org/10.1016/S0894-1777\(97\)10034-6](https://doi.org/10.1016/S0894-1777(97)10034-6).
 58. Vicente PG, García A, Viedma A. Experimental study of mixed convection and pressure drop in helically dimpled tubes for laminar and transition flow. *Int J Heat Mass Transf.* 2002;45:5091–105. [https://doi.org/10.1016/S0017-9310\(02\)00215-6](https://doi.org/10.1016/S0017-9310(02)00215-6).
 59. Zhang L-Z. Numerical study of periodically fully developed flow and heat transfer in cross-corrugated triangular channels in transitional flow regime. *Numer Heat Transf Part A Appl.* 2005;48:387–405. <https://doi.org/10.1080/10407780590957314>.
 60. García A, Solano JP, Vicente PG, Viedma A. The influence of artificial roughness shape on heat transfer enhancement: corrugated tubes, dimpled tubes and wire coils. *Appl Therm Eng.* 2012;35:196–201. <https://doi.org/10.1016/j.applthermaleng.2011.10.030>.
 61. Popov IA, Shchelchikov AV, Yarkaez MZ. Thermal and hydraulic characteristics of discretely rough tubes at transient flow regimes. *Russ Aeronaut (Iz VUZ).* 2013;56:83–9. <https://doi.org/10.3103/S1068799813010121>.
 62. Meyer JP, Abolarin SM. Heat transfer and pressure drop in the transitional flow regime for a smooth circular tube with twisted tape inserts and a square-edged inlet. *Int J Heat Mass Transf.* 2018;117:11–29. <https://doi.org/10.1016/j.ijheatmasstransfer.2017.09.103>.
 63. Abolarin SM, Everts M, Meyer JP. The influence of peripheral u-cut twisted tapes and ring inserts on the heat transfer and pressure drop characteristics in the transitional flow regime. *Int J Heat Mass Transf.* 2019;132:970–84. <https://doi.org/10.1016/j.ijheatmasstransfer.2018.12.051>.
 64. Abolarin SM, Everts M, Meyer JP. Heat transfer and pressure drop characteristics of alternating clockwise and counter clockwise twisted tape inserts in the transitional flow regime. *Int J Heat Mass Transf.* 2019;133:203–17. <https://doi.org/10.1016/j.ijheatmasstransfer.2018.12.107>.
 65. García A, Solano JP, Vicente PG, Viedma A. Enhancement of laminar and transitional flow heat transfer in tubes by means of wire coil inserts. *Int J Heat Mass Transf.* 2007;50:3176–89. <https://doi.org/10.1016/j.ijheatmasstransfer.2007.01.015>.
 66. Fiebig M. Embedded vortices in internal flow: heat transfer and pressure loss enhancement. *Int J Heat Fluid Flow.* 1995;16:376–88. [https://doi.org/10.1016/0142-727X\(95\)00043-P](https://doi.org/10.1016/0142-727X(95)00043-P).
 67. Zagarola MV, Smits AJ, Karniadakis GE. Heat transfer enhancement in a transitional channel flow. *J Wind Eng Ind Aerodyn.* 1993;49:257–67. [https://doi.org/10.1016/0167-6105\(93\)90021-F](https://doi.org/10.1016/0167-6105(93)90021-F).
 68. Elison B, Webb BW. Local heat transfer to impinging liquid jets in the initially laminar, transitional, and turbulent regimes. *Int J Heat Mass Transf.* 1994;37:1207–16. [https://doi.org/10.1016/0017-9310\(94\)90206-2](https://doi.org/10.1016/0017-9310(94)90206-2).
 69. Mikic BB, Vujisic LB, Kapat J. Turbulent transition and maintenance of turbulence; implication to heat transfer augmentation. *Int J Heat Mass Transf.* 1994;37:425–31. [https://doi.org/10.1016/0017-9310\(94\)90042-6](https://doi.org/10.1016/0017-9310(94)90042-6).
 70. García A, Solano JP, Vicente PG, Viedma A. Flow pattern assessment in tubes with wire coil inserts in laminar and transition regimes. *Int J Heat Fluid Flow.* 2007;28:516–25. <https://doi.org/10.1016/j.ijheatfluidflow.2006.07.001>.
 71. Saeedinia M, Akhavan-Behabadi MA, Nasr M. Experimental study on heat transfer and pressure drop of nanofluid flow in a horizontal coiled wire inserted tube under constant heat flux. *Exp Therm Fluid Sci.* 2012;36:158–68. <https://doi.org/10.1016/j.expthermflusci.2011.09.009>.
 72. Andrade F, Moita AS, Nikulin A, Moreira ALN, Santos H. Experimental investigation on heat transfer and pressure drop of internal flow in corrugated tubes. *Int J Heat Mass Transf.* 2019;140:940–55. <https://doi.org/10.1016/j.ijheatmasstransfer.2019.06.025>.
 73. Martínez DS, García A, Solano JP, Viedma A. Heat transfer enhancement of laminar and transitional Newtonian and non-Newtonian flows in tubes with wire coil inserts. *Int J Heat Mass Transf.* 2014;76:540–8. <https://doi.org/10.1016/j.ijheatmasstransfer.2014.04.060>.
 74. Meyer JP, Everts M. Single-phase mixed convection of developing and fully developed flow in smooth horizontal circular tubes in the laminar and transitional flow regimes. *Int J Heat Mass Transf.* 2018;117:1251–73. <https://doi.org/10.1016/j.ijheatmasstransfer.2017.10.070>.
 75. Chen YS, Tian J, Fu Y, Tang ZF, Zhu HH, Wang NX. Experimental study of heat transfer enhancement for molten salt with transversely grooved tube heat exchanger in laminar-transition-turbulent regimes. *Appl Therm Eng.* 2018;132:95–101. <https://doi.org/10.1016/j.applthermaleng.2017.12.054>.
 76. Osman S, Sharifpur M, Meyer JP. Experimental investigation of convection heat transfer in the transition flow regime of aluminium oxide-water nanofluids in a rectangular channel. *Int J Heat Mass Transf.* 2019;133:895–902. <https://doi.org/10.1016/j.ijheatmasstransfer.2018.12.169>.
 77. Dirker J, Meyer JP, Garach DV. Inlet flow effects in micro-channels in the laminar and transitional regimes on single-phase heat transfer coefficients and friction factors. *Int J Heat Mass Transf.* 2014;77:612–26. <https://doi.org/10.1016/j.ijheatmasstransfer.2014.05.048>.
 78. Minhas H, Lock GSH. Laminar-turbulent transition in a bayonet tube. *Int J Heat Fluid Flow.* 1996;17:102–7. [https://doi.org/10.1016/0142-727X\(95\)00078-5](https://doi.org/10.1016/0142-727X(95)00078-5).
 79. Minhas H, Lock GSH. Forced convection in an air-filled bayonet tube during the laminar-turbulent transition. *Int J Heat Mass Transf.* 1997;40:1885–94. [https://doi.org/10.1016/S0017-9310\(96\)00225-6](https://doi.org/10.1016/S0017-9310(96)00225-6).
 80. Joye DD. Comparison of aiding and opposing mixed convection heat transfer in a vertical tube with Grashof number variation. *Int J Heat Fluid Flow.* 1996;17:96–101. [https://doi.org/10.1016/0142-727X\(95\)00077-4](https://doi.org/10.1016/0142-727X(95)00077-4).
 81. Peng XF, Peterson GP. Forced convection heat transfer of single-phase binary mixtures through microchannels. *Exp Therm Fluid Sci.* 1996;12:98–104. [https://doi.org/10.1016/0894-1777\(95\)00079-8](https://doi.org/10.1016/0894-1777(95)00079-8).

82. Meyer JP, Bashir AI, Everts M. Single-phase mixed convective heat transfer and pressure drop in the laminar and transitional flow regimes in smooth inclined tubes heated at a constant heat flux. *Exp Therm Fluid Sci.* 2019;109: 109890. <https://doi.org/10.1016/j.expthermflusci.2019.109890>.
83. Saha G, Paul MC. Transition of nanofluids flow in an inclined heated pipe. *Int Commun Heat Mass Transf.* 2017;82:49–62. <https://doi.org/10.1016/j.icheatmasstransfer.2017.02.017>.
84. Alzwayi AS, Paul MC. Transition of free convection flow inside an inclined parallel walled channel: effects of inclination angle and width of the channel. *Int J Heat Mass Transf.* 2014;68:194–202. <https://doi.org/10.1016/j.jheatmasstransfer.2013.09.015>.
85. Al-Arabi M, Sakr B. Natural convection heat transfer from inclined isothermal plates. *Int J Heat Mass Transf.* 1988;31:559–66. [https://doi.org/10.1016/0017-9310\(88\)90037-3](https://doi.org/10.1016/0017-9310(88)90037-3).
86. Kumar D, Premachandran B. Investigation of transition in a natural convection flow in an inclined parallel Plate Channel. *Int Commun Heat Mass Transf.* 2022;130: 105768. <https://doi.org/10.1016/j.icheatmasstransfer.2021.105768>.
87. Yuan H, Tan S, Zhuang N, Lan S. Flow and heat transfer in laminar-turbulent transitional flow regime under rolling motion. *Ann Nucl Energy.* 2016;87:527–36. <https://doi.org/10.1016/j.anucene.2015.10.009>.
88. Demirkır Ç, Ertürk H. Convective heat transfer and pressure drop characteristics of graphene-water nanofluids in transitional flow. *Int Commun Heat Mass Transf.* 2021;121: 105092. <https://doi.org/10.1016/j.icheatmasstransfer.2020.105092>.
89. Ali AHH, Ahmed M, Abdel-Gaied SM. Investigation of heat transfer and fluid flow in transitional regime inside a channel with staggered plates heated by radiation for PV/T system. *Energy.* 2013;59:255–64. <https://doi.org/10.1016/j.energy.2013.06.047>.
90. Bhuiyan AA, Amin MR, Islam AKMS. Three-dimensional performance analysis of plain fin tube heat exchangers in transitional regime. *Appl Therm Eng.* 2013;50:445–54. <https://doi.org/10.1016/j.applthermaleng.2012.07.034>.
91. Abraham JP, Sparrow EM, Tong JCK. Heat transfer in all pipe flow regimes: laminar, transitional/intermittent, and turbulent. *Int J Heat Mass Transf.* 2009;52:557–63. <https://doi.org/10.1016/j.jheatmasstransfer.2008.07.009>.
92. Gnielinski V. New equations for heat and mass transfer in turbulent pipe and channel flow. *Int Chem Eng.* 1976;16:359–67.
93. Churchill SW. Comprehensive correlating equations for heat, mass and momentum transfer in fully developed flow in smooth tubes. *Ind Eng Chem Fundam.* 1977. <https://doi.org/10.1021/i160061a021>.
94. Meyer JP, McKrell TJ, Grote K. The influence of multi-walled carbon nanotubes on single-phase heat transfer and pressure drop characteristics in the transitional flow regime of smooth tubes. *Int J Heat Mass Transf.* 2013;58:597–609. <https://doi.org/10.1016/j.jheatmasstransfer.2012.11.074>.
95. Peixinho J, Desaubry C, Lebouché M. Heat transfer of a non-Newtonian fluid (Carbopol aqueous solution) in transitional pipe flow. *Int J Heat Mass Transf.* 2008;51:198–209. <https://doi.org/10.1016/j.jheatmasstransfer.2007.04.012>.
96. Stamatopoulos C, Petropoulos A, Mathioulakis DS, Kaltsas G. Study of an integrated thermal sensor in different operational modes, under laminar, transitional and turbulent flow regimes. *Exp Therm Fluid Sci.* 2008;32:1687–93. <https://doi.org/10.1016/j.expthermflusci.2008.06.003>.
97. Abraham JP, Sparrow EM, Tong JCK. Breakdown of laminar pipe flow into transitional intermittency and subsequent attainment of fully developed intermittent or turbulent flow. *Numer Heat Transf Part B Fundam.* 2008;54:103–15. <https://doi.org/10.1080/10407790802156178>.
98. Celata GP, Lorenzini M, Morini GL, Zummo G. Friction factor in micropipe gas flow under laminar, transition and turbulent flow regime. *Int J Heat Fluid Flow.* 2009;30:814–22. <https://doi.org/10.1016/j.jheatfluidflow.2009.02.015>.
99. Yang CY, Chen CW, Lin TY, Kandlikar SG. Heat transfer and friction characteristics of air flow in microtubes. *Exp Therm Fluid Sci.* 2012;37:12–8. <https://doi.org/10.1016/j.expthermflusci.2011.09.003>.
100. Meyer JP, Grote K, McKrell T. Heat transfer characteristics of multi-walled carbon nanotubes in the transitional flow regime. In: *Proceedings of the 3rd international forum heat transfer*. Nagasaki, Japan; 2016. pp. 9–11.
101. Varnaseri M, Peyghambarzadeh SM. The effect of polyacrylamide drag reducing agent on friction factor and heat transfer coefficient in laminar, transition and turbulent flow regimes in circular pipes with different diameters. *Int J Heat Mass Transf.* 2020;154:119815. <https://doi.org/10.1016/j.jheatmasstransfer.2020.119815>.
102. Briclot A, Henry JF, Popa C, Nguyen CT, Fohanno S. Experimental investigation of the heat and fluid flow of an Al_2O_3 -water nanofluid in the laminar-turbulent transition region. *Int J Therm Sci.* 2020;158:106546. <https://doi.org/10.1016/j.ijthermalsci.2020.106546>.
103. Liu SH, Huang YP, Wang JF, Liu RL. Experimental study on transitional flow in straight channels of printed circuit heat exchanger. *Appl Therm Eng.* 2020;181:115950. <https://doi.org/10.1016/j.applthermaleng.2020.115950>.
104. Jamali M, Toghraie D. Investigation of heat transfer characteristics in the developing and the developed flow of nanofluid inside a tube with different entrances in the transition regime. *J Therm Anal Calorim.* 2020;139:685–99. <https://doi.org/10.1007/s10973-019-08380-9>.
105. Karwa R, Solanki SC, Saini JS. Heat transfer coefficient and friction factor correlations for the transitional flow regime in rib-roughened rectangular ducts. *Int J Heat Mass Transf.* 1999;42:1597–615. [https://doi.org/10.1016/S0017-9310\(98\)00252-X](https://doi.org/10.1016/S0017-9310(98)00252-X).
106. Zhang LZ, Zhong WC, Chen JM, Zhou JR. Fluid flow and heat transfer in plate-fin and tube heat exchangers in a transitional flow regime. *Numer Heat Transf Part A Appl.* 2011;60:766–84. <https://doi.org/10.1080/10407782.2011.627802>.
107. Tam HK, Tam LM, Ghajar AJ. Effect of inlet geometries and heating on the entrance and fully-developed friction factors in the laminar and transition regions of a horizontal tube. *Exp Therm Fluid Sci.* 2013;44:680–96. <https://doi.org/10.1016/j.expthermflusci.2012.09.008>.
108. Ghajar AJ, Tang CC, Cook WL. Experimental investigation of friction factor in the transition region for water flow in minitubes and microtubes. *Heat Transf Eng.* 2010;31:646–57. <https://doi.org/10.1080/01457630903466613>.
109. Meyer JP, Du Preez A. Heat transfer and pressure drop characteristics of water in the transitional flow regime of horizontal smooth tubes at constant wall temperature. In: *2010 14th international heat transfer conference*, Vol. 2. ASMEDC; 2010. pp. 437–444. <https://doi.org/10.1115/IHTC14-22340>.
110. Naik MT, Sundar LS. Heat transfer and friction factor with water/propylene glycol-based CuO nanofluid in circular tube with helical inserts under transition flow regime. *Heat Transf Eng.* 2014;35:53–62. <https://doi.org/10.1080/01457632.2013.810451>.
111. Bertsche D, Knipper P, Wetzel T. Experimental investigation on heat transfer in laminar, transitional and turbulent circular pipe flow. *Int J Heat Mass Transf.* 2016;95:1008–18. <https://doi.org/10.1016/j.jheatmasstransfer.2016.01.009>.
112. Ndenguma DD, Dirker J, Meyer JP. Transitional flow regime heat transfer and pressure drop in an annulus with non-uniform wall

- temperatures. *Int J Heat Mass Transf.* 2017;108:2239–52. <https://doi.org/10.1016/j.ijheatmasstransfer.2017.01.022>.
113. Ndenguma DD, Dirker J, Meyer JP. Heat transfer and pressure drop in annuli with approximately uniform internal wall temperatures in the transitional flow regime. *Int J Heat Mass Transf.* 2017;111:429–41. <https://doi.org/10.1016/j.ijheatmasstransfer.2017.02.064>.
 114. Everts M, Meyer JP. Heat transfer of developing and fully developed flow in smooth horizontal tubes in the transitional flow regime. *Int J Heat Mass Transf.* 2018;117:1331–51. <https://doi.org/10.1016/j.ijheatmasstransfer.2017.10.071>.
 115. Everts M, Meyer JP. Relationship between pressure drop and heat transfer of developing and fully developed flow in smooth horizontal circular tubes in the laminar, transitional, quasi-turbulent and turbulent flow regimes. *Int J Heat Mass Transf.* 2018;117:1231–50. <https://doi.org/10.1016/j.ijheatmasstransfer.2017.10.072>.
 116. Bashir AI, Everts M, Meyer JP. Influence of inlet contraction ratios on the heat transfer and pressure drop characteristics of single-phase flow in smooth circular tubes in the transitional flow regime. *Exp Therm Fluid Sci.* 2019;109: 109892. <https://doi.org/10.1016/j.expthermflusci.2019.109892>.
 117. Bashir AI, Everts M, Bennacer R, Meyer JP. Single-phase forced convection heat transfer and pressure drop in circular tubes in the laminar and transitional flow regimes. *Exp Therm Fluid Sci.* 2019;109: 109891. <https://doi.org/10.1016/j.expthermflusci.2019.109891>.
 118. García A, Vicente PG, Viedma A. Experimental study of heat transfer enhancement with wire coil inserts in laminar-transition-turbulent regimes at different Prandtl numbers. *Int J Heat Mass Transf.* 2005;48:4640–51. <https://doi.org/10.1016/j.ijheatmasstransfer.2005.04.024>.
 119. Taler D. A new heat transfer correlation for transition and turbulent fluid flow in tubes. *Int J Therm Sci.* 2016;108:108–22. <https://doi.org/10.1016/j.ijthermalsci.2016.04.022>.
 120. Morteau MVV, Mantelli MBH. Nusselt number correlation for compact heat exchangers in transition regimes. *Appl Therm Eng.* 2019;151:514–22. <https://doi.org/10.1016/j.applthermaleng.2019.02.017>.
 121. Meyer JP, Everts M, Coetzee N, Grote K, Steyn M. Heat transfer coefficients of laminar, transitional, quasi-turbulent and turbulent flow in circular tubes. *Int Commun Heat Mass Transf.* 2019;105:84–106. <https://doi.org/10.1016/j.icheatmasstransfer.2019.03.016>.

Publisher's Note Springer Nature remains neutral with regard to jurisdictional claims in published maps and institutional affiliations.

Springer Nature or its licensor (e.g. a society or other partner) holds exclusive rights to this article under a publishing agreement with the author(s) or other rightsholder(s); author self-archiving of the accepted manuscript version of this article is solely governed by the terms of such publishing agreement and applicable law.