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Alternative Internal Configurations for Enhancing Heat Transfer in Telecommunication Cabinets

Antony Jobby, Mehdi Khatamifar and Wenxian Lin *

College of Science & Engineering, James Cook University, Townsville, QLD 4811, Australia; antony.jobby@my.jcu.edu.au (A.J.); mehdi.khatamifar2@jcu.edu.au (M.K.)
* Correspondence: wenxian.lin@jcu.edu.au

Abstract: Telecommunication systems have become a critical part of society which enables connectivity to many essential and trivial services. Consequently, telecommunication equipment is housed in cabinets to protect the electronics from a variety of hazards; one of which is temperature-related failure. Current practices use a notable amount of power for the thermal management of telecommunication cabinets which can be reduced by considering alternative methods of cooling. In this paper, experiments were carried out to investigate the effectiveness of different internal mounting configurations of electronic components on the thermal performance of a telecommunication cabinet. The investigation tested inclinations (0–90°), different staggered offsets (0–50 mm), changing stream-wise spacing (29–108 mm), and fan speed (with a Reynolds number in the range of 1604 to 5539). The experimental study revealed that heat transfer was enhanced by 9.99% by altering component inclination to 90°, 25.90% by increasing stream-wise spacing from 29 mm to 108 mm , and 36.02% by increasing the Reynolds number from 1604 to 5539. However, the staggered arrangement of internal components decreased Nu by 3.26% for the natural convection condition but increased by 5.69% for the forced convection condition over the tested range and increasing the centre offset of the staggered components with respect to the cabinet did not influence Nu in any significant manner. Natural convection and forced convection also had notable influence on the heat transfer rate. Hence it was seen that alternative internal configurations positively influence heat transfer in telecommunication cabinets for the cases studied.

Keywords: telecommunication cabinet; angle; spacing; staggered; heat transfer; convection

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1. Introduction

Around the world, increased connectivity and the use of smartphones have led to the rise of telecommunication infrastructure [1]. The need for high-speed telecommunication and high bit rate services result in a large amount of heat being generated in telecommunication cabinets [2]. The development of smaller and improved microelectronic components further necessitate the development of efficient cooling techniques [3,4]. Therefore, the development of technology is a major factor that is driving the need for cooling capabilities in telecommunication cabinets [5]. Additionally telecommunication networks are vital in many sectors, such as government, healthcare, education, finance, etc. [6], and as a result influence society to a high degree. Hence, effective thermal management of electrical components is necessary to meet current demand and facilitate future growth.

Telecommunication cabinets are fully enclosed storage units that house electronics primarily used for telecommunication applications, such as switching and signal-processing [7]. Telecommunication cabinets have two fundamental functions; to protect electronics from environmental hazards (such as solar degradation, humidity, dust, or debris) and to provide a suitable operating condition (such as the prescribed temperature of the electronics) [8]. The operating condition is a key factor that affects the performance and life of electronic devices. Amongst several factors, the most common failure mode of electronic components is associated with temperature [7]. High temperatures are known to cause thermal failures in chips which include mechanical stresses, temperature-induced de-bonding, and fractures [9]. Furthermore, the rate of electronic component failure has been reported to almost doubles with every 10 °C increase above the prescribed operating temperature [10–13]. Thus, heat transfer becomes an important aspect to realise value and life from electronics [6,14]. Consequently, different approaches are used for thermal management in telecommunication cabinets to uphold the reliability of the system.

Although telecommunication cabinets that provide climate control features are very useful for electrical devices, it is also important to consider the type of cooling used. Power usage, financial cost, and environmental factors become pertinent depending on the thermal management technique. Telecommunication cabinets achieve thermal management by passive, active, or a combination of passive and active cooling techniques. Passive cooling methods refer to approaches that do not require the input of power, whereas active cooling methods refer to those that require the input of power. It has been reported that for actively cooled cabinets, approximately 30% of the total power consumption is attributed to cooling [5,10,15]. Power usage is an important metric that contributes to the overall operating cost of the system. Therefore, the potential to reduce the power requirement and in turn cost is possible by utilising passive cooling methods. In general, using both passive and active cooling methods can reduce power consumption from 28% to 15% and in certain cases, no power is needed [7,10]. Furthermore, it has been reported that electric power consumption has a direct or indirect contribution to greenhouse gas emissions and thus climate change [15].

Passive cooling provides various avenues to achieve cooling without the need for external energy. In outdoor telecommunication cabinets solar radiation can increase the internal temperature by 20%. However, it has been reported that applying a coat of paint on the external surface of the cabinet can reduce the internal temperature by 25%. The inclusion of a radiation shield and a double-walled enclosure (that has air circulation), can also provide a 25% improvement in the internal temperature [8]. The energy storage capabilities of various substances offer another solution in the form of phase change materials (PCMs). PCMs are substances that can absorb (or release) thermal energy by undergoing a phase transformation [16]. PCMs are used to absorb thermal energy when peak temperatures occur and release the stored energy once the temperature drops. Typically, PCMs should be used when cyclic thermal loads are present and are not suited for continuous or ongoing thermal loads. PCMs can be especially useful to offset the effects of rapid temperature fluctuations as a result of peak loads [17]. Passive closed-loop thermosyphons are energy transfer devices that can transport thermal energy relatively long distances from a heat source to a heat sink [2]. Depending on the specific configuration, a cabinet equipped with a thermosyphon can operate with almost two times greater power than a cabinet without a thermosyphon [5]. Moreover, heat pipes are considered as one of the most efficient passive heat transfer methods. Heat pipes work based on phase change and vapour diffusion processes [18]. Other methods are also used in telecommunication cabinets for thermal management.

Active heat transfer mechanisms require energy to achieve cooling and include the use of air conditioners and fans. The use of fans is an effective way to increase the air flow and thus the convective heat transfer rate of an object [19]. Active cooling systems are also used in applications that require the internal temperature of a telecommunication cabinet to be lower than the ambient temperature. To achieve sub-ambient temperatures, air-conditioners are typically used [7]. The positioning of the inlet and outlet of the air-conditioning system also affects cooling. Ideally, the air supply should be located in the lower position of an enclosure and the air-return outlet should be placed at a higher position. The specific configuration follows the natural process of warm air rising [20]. Additional methods to achieve sub-ambient temperatures utilise thermoelectric devices such as the thermoelectric cooler (TEC) which holds tremendous potential [21,22]. Other active cooling method are also used for various purposes.

Passive cooling methods do not need to be backed-up (which is needed for active systems [17]), and together with low operating costs, passive techniques are more desirable in some electronic cooling applications [23]. It is widely recognised that the orientation and position of a body affect heat dissipation, particularly convective heat transfer. For a cube (bounded by a surface), it was reported that the geometric factors affect the convective heat transfer by as much as 58% [24]. Thus, geometric alteration is a feasible method to enhance heat transfer.

Previous studies have investigated the effect of inclination on heat transfer [3,19,25]. For an array of blocks tested experimentally in a wind tunnel, it was reported that the orientation (angle) has a significant effect on heat transfer. Testing the angle of attack from 0 to 45° revealed that the highest Nusselt number (Nu)—the non-dimensionalised convective heat transfer coefficient—corresponded to the highest tested angle. It was stated that increasing the angle of the blocks encourages the flow of fluid to transition from a laminar to a turbulent regime sooner which promotes heat transfer. In the conventional orientation (or the 0° angle case), recirculation could occur between the block and limit heat transfer [3,19]. The concept was further substantiated as a numerical investigation also showed that altering the angle of incidence between a rectangular block and the fluid stream improved heat transfer. Once again the effect was attributed to the turbulence that was generated as a result of inclining the block [25]. Therefore, inclining blocks/components is a useful parameter to enhance heat transfer. An alternative method to incline individual components is to incline the entire cavity or enclosure.

Although numerous studies have been conducted for tilted cavities [26–32], all studies are not directly comparable due to the different positions of the heat sources or the cavity shape. Previous studies included those conducted on fully enclosed and open rectangular cavities as well as 'T'-shaped cavities.

Certain studies have been conducted with fully enclosed designs which include a source for heating and cooling. Fully enclosed cavities with inclination angles in the range of 0–90° were reported. One numerical study consisted of placing a vertically situated heating unit in the centre of a rectangular cavity. It was reported that Nu increased until an angle of 22.5° from a horizontal datum [26]. The effect of the inclination of a cavity from a 2D (two-dimensional) rectangular heat source placed on the lower wall, with the top wall cooled and all other walls adiabatic, was also investigated numerically. It was reported that the average Nu changed with the inclination and reached a maximum at 30° from the horizontal [27]. Both numerical studies used air as the working fluid. Separately, two similar experimental investigations also tested the inclination angle in an enclosed cavity. It was reported that Nu increases approximately from 0° to 60° and then falls from 60° to 90° with the maximum value observed at 60°. The working fluids were water [28] and aluminium-oxide-water nanofluid [29].

Other studies focused on cavities where fluid was allowed to pass through the enclosure from an inlet to an outlet. A numerical study on a vertical channel that varied the inclination from the vertical position to the nearly horizontal position, in the range of 0–80°, reported that the local and average heat transfer reduced as the channel orientation changed from the vertical to the horizontal orientation [30]. A numerical and experimental investigation reported the effect of inclination angle on natural convection heat transfer (in the range of 0–90° with steps of 15°). The investigation showed that increasing inclination, or orienting the channel so that fluid can flow vertically up, enhanced heat transfer. Interestingly, it was reported that after increasing the inclination to be more than 60°, there was a reduced effect on heat transfer. It was concluded that the strength of the buoyancy force in the vertical direction decreases as the angle of inclination increases [31]. A numerical investigation focused on the effect of inclination in a 'T'-shaped cavity where the working fluid was allowed to pass through the domain. It was reported that orienting the cavity so that the fluid flows vertically and then inclining the cavity towards the horizontal, decreased heat transfer. In other words, orienting the cavity from the vertical towards the horizontal reduces heat transfer [32]. Hence, it can be seen that the inclination of cavities for heat transfer is most prominent within a certain range; however, in general shows an overall improvement.

A staggered arrangement involves offsetting the centre of components so that the fluid has space to pass through the arrangement more easily than in-line components. Experimental investigations have shown that staggered arrangements yield better heat transfer rates in comparison to in-line arrangements [33,34]. Therefore, the incorporation of staggered arrangements in telecommunication cabinets is an option to improve heat transfer.

Altering the stream-wise spacing between components is another method to dissipate heat through the provision of additional space between components in the direction which the fluid flows. Numerical simulation of blocks mounted in a horizontal channel with mixed convection cooling showed that stream-wise block separation has a considerable impact on enhancing the heat transfer [6]. An experimental study in a horizontal wind tunnel also reported that stream-wise block spacing improves heat transfer. It was suggested that the increased space reduces flow by-pass and recirculation to improve heat transfer [3]. Furthermore, a numerical simulation of blocks placed on one side of a vertical channel, with turbulent natural convection cooling, found that increasing the stream-wise spacing reduced the temperature of the blocks [35]. Thus, increasing stream-wise spacing has a positive impact on heat dissipation in both vertical and horizontal flows.

Span-wise spacing is the method of increasing the distance between components (in the direction perpendicular to the flow), to improve heat transfer. In a numerical study the spacing between components that were attached to the lower wall of a rectangular cavity was varied. The blocks were heated and the effect of natural convection heat transfer was observed. It was reported that the increased span-wise spacing increased Nu [36]. In other numerical studies that considered the spacing of components with a 'T'-shaped gap also found that increasing span-wise spacing increased heat transfer [23,37]. The effect of varying geometric features of two blocks mounted on each sidewall of a vertical channel was investigated numerically to observe the influence on heat transfer. It was reported that increasing the width of the two adjacent blocks (thereby reducing the span-wise space), drastically reduced the mass flow rate and heat transfer, particularly when half the channel was blocked [38]. An experimental study compared the effect of both stream- and span-wise spacing, showing that both parameters have a similar effect on heat transfer; however, stream-wise spacing was slightly more effective in improving heat transfer than span-wise spacing [19]. Hence, both stream- and span-wise spacing remain very useful passive cooling mechanisms for heat-generating blocks such as those used in telecommunication cabinets.

The use of fans is a common method to enhance heat transfer and is consequently used in many applications including telecommunication cabinets. Many studies have established that heat transfer increases as a function of fluid velocity, where Nu increases as the Reynolds number (Re) is increased [39]. For instance, an experimental study comparing the effects of Re, inclining block (relative to the flow), varying stream- and span-wise spacing in a wind tunnel showed that Re has the most prominent effect in enhancing heat transfer. It was further reported that enhanced heat transfer occurs as a result of a turbulent or disordered flow pattern [19].

The prospect of adapting various passive configuration-based heat transfer methods is an option to improve heat transfer in telecommunication cabinets by altering the inclination of components, stream-wise spacing and staggered positioning. Considering the effect of fan speed will also show how the heat transfer of different configurations is affected by the air flow rate. Previous studies have investigated the principles relating to heat transfer from component inclination, stream-wise spacing, staggered positioning, and air flow rate [3,19]. However, no studies, to the best knowledge of the authors, have investigated the same parameters experimentally in a telecommunication cabinet setup with heat-generating components that are not wall-bounded. The internal construct of a telecommunication cabinet setup poses certain constraints on the flow pattern and the obtained results are expected to be more aligned with heat transfer effects that are observed in practice. Moreover, reliable and accurate comparisons can be deduced when all the parameters are investigated in a single experimental setup in comparison to results from different studies.

The objective of the present study is to evaluate the effectiveness of alternative configurations of internal components and an operating condition to enhance heat transfer inside a telecommunication cabinet. The investigated configurations included component inclination (0–90°), staggered arrangement (with a centre-offset from 0 mm to 50 mm), and stream-wise spacing (between 29 mm and 108 mm). The effect of airflow velocity, from natural convection (with the fan speed at 0 m/s) to 1.57 m/s, was also investigated to see if an optimal fan speed range exists to achieve the most effective cooling without wasting power. The outcome of the investigation is expected to show which parameters have the capacity to improve heat transfer in the tested telecommunication cabinet setup. Additionally, the results will highlight which parameters can be further investigated by manufacturers and researchers to reduce energy consumption related to cooling in telecommunication cabinets. Hence, present cabinet arrangements can be altered if needed and future cabinets can be designed in light of the results.

2. Experimental Methodology

The overarching focus was to investigate all the parameters (component inclination, staggered arrangement, stream-wise spacing, and fan speed), in the same cabinet and conditions. An illustration of the parameters is shown in Figure 1. The main data were collected in the form of temperature and subsequently analysed to extract quantitative results.

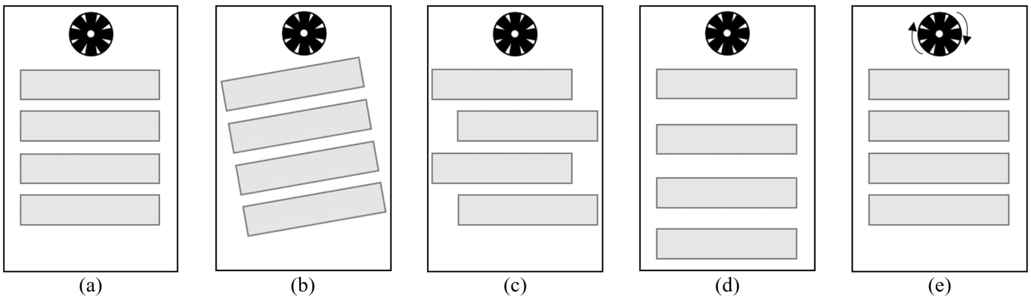


Figure 1. Configurations under investigation: (**a**) control/reference, (**b**) inclined, (**c**) staggered, (**d**) stream-wise spacing, and (**e**) changing fan speed.

2.1. Experimental Setup

The experimental setup included a decommissioned telecommunication cabinet, simulated electronic components and a fan. The temperatures of the system were measured at various locations inside the telecommunication cabinet and subsequently used to quantify the heat transfer capacity of different configurations. A schematic of the combined system is shown in Figure 2. The cabinet provided the boundary for the experiment and ensured that the air passed through the inlet/outlet to limit heat loss by other avenues.

A Patchpak by Modempak telecommunication cabinet was used for the investigation which measured 865 mm × 450 mm × 535 mm externally. Modifications were necessary to use the cabinet for testing which involved repainting, installing the component mounting mechanism (which was comprised of the lateral support bars/chains), and adding additional ventilation inlets (as shown in Figure 2). A 85 CFM (cubic feet per minute) Sunon 2123HBT fan was also fitted on the wooden back panel of the cabinet to induce air flow. The fan speed was controlled using a Shanghai Shouni Electronic Technology SNT-4000W thyristor electric voltage regulator connected to a power outlet. A funnel-shaped plastic section was attached to the back of the fan to serve as a flow mixer, as shown in Figure 3. The flow mixer was used to mix the outgoing air so that the outlet temperature measurement was an actual representation of the air that passed out of the cabinet [19].

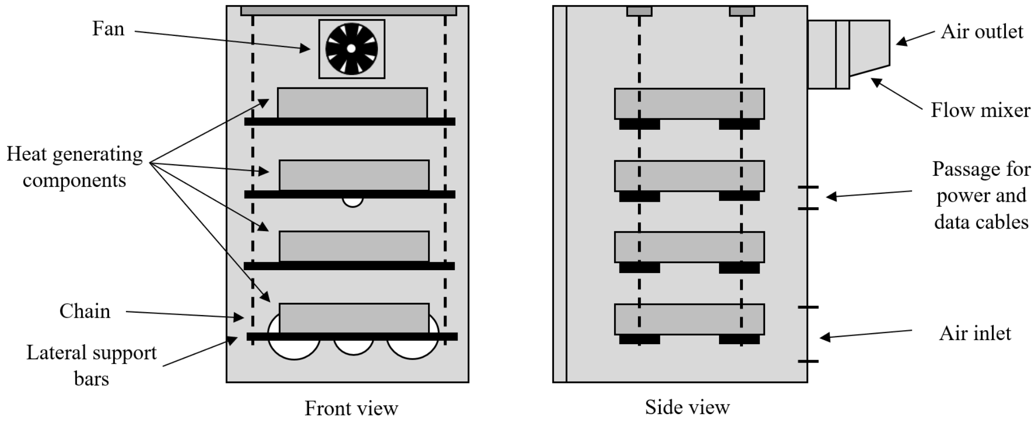
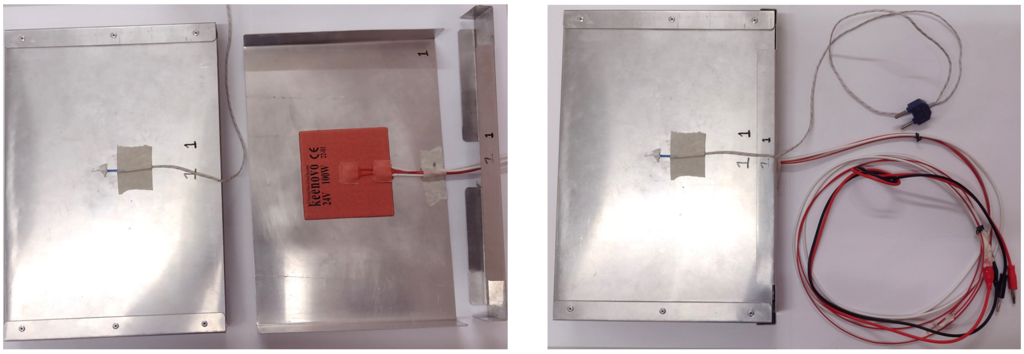


Figure 2. Schematic of the cabinet.

Figure 3. Flow mixer.

In telecommunication cabinets used in practice, heat is generated from electronics as a result of normal operation. For the experiment, the heat generation was emulated using four 24 volt, 100 watt, 100 mm × 100 mm Keenovo silicon heaters inside custom-built aluminium rectangular boxes. Each heater was affixed to a 1 mm thick removable aluminium plate (using 3M adhesive), which was then placed inside an aluminium box with a removable lid. The combination of the heater, removable aluminium plate, box and lid will be herein referred to collectively as a component or heat-generating component. The dismantled and assembled heat-generating component is shown in Figure 4. In total, four heat-generating component were used. Each component measured 332 mm × 233 mm × 47 mm. The power was supplied to the heaters from two Powertech MP-3084 power supplies (which was capable of powering one component each) and a GW Instek GPS-3303 3CH laboratory power supply (which was capable of powering two components).



(a) (b)

Figure 4. A heat-generating component: (**a**) dismantled, and (**b**) assembled.

The specialised component-mounting mechanism was also custom-built to accommodate all the parameters that were tested. The mounting subsystem consisted of four chains that were hung from rails at the top of the cabinet. Each chain could be translated laterally, one at each corner of the cabinet. Two lateral support bars were threaded and suspended by the chains using nuts and bolts. In total eight lateral support bars were suspended on the chains to form four levels. A component was placed on each level that could be manipulated in different ways to accommodate the test conditions. For example, one side of the lateral support bar could be secured higher than the other side, thereby creating an angled space for the components to sit. The position of the lateral support bars could also be changed to account for different stream-wise spacing requirements. The need to stagger or offset the components was met by manufacturing the components with less width than the lateral support bars so that the components could be translated. The control/reference setup is shown in Figure 5.

Figure 5. Control configuration.

Data were quantitatively collected during the experiments using a variety of instruments. The temperature at specific locations of the cabinet was measured using thermocouples and recorded using a data logger in conjunction with a computer. The schematic of the experimental setup is shown in Figure 6. A total of eight Mastech P3400 k-type thermocouples were used. Two thermocouples were affixed to the air inlet and two thermocouples were placed at the outlet. The remaining four thermocouples were affixed on the top surface of the four components. Duratech NM2014 heatsink plaster (which is a heatsink compound), was applied to ensure the thermocouples had good thermal contact with the surface of the aluminium boxes. The temperature of each component or aluminium box was therefore measured by a single thermocouple. The thermocouples were subsequently connected to a Campbell Scientific CR1000X data logger. The data logger was used to collect the temperature of the thermocouples at a frequency of 1 s and save the averaged temperature every 1 min. The inclination angle was measured using a Stanley fatmax angle spirit level. The airflow velocity from the outlet was measured using a Lutron AM-4214SD hot wire anemometer. Velocity measurements were taken three times for each fan speed setting and averaged to find the air velocity at the outlet. The experimental setup is shown in Figure 7.

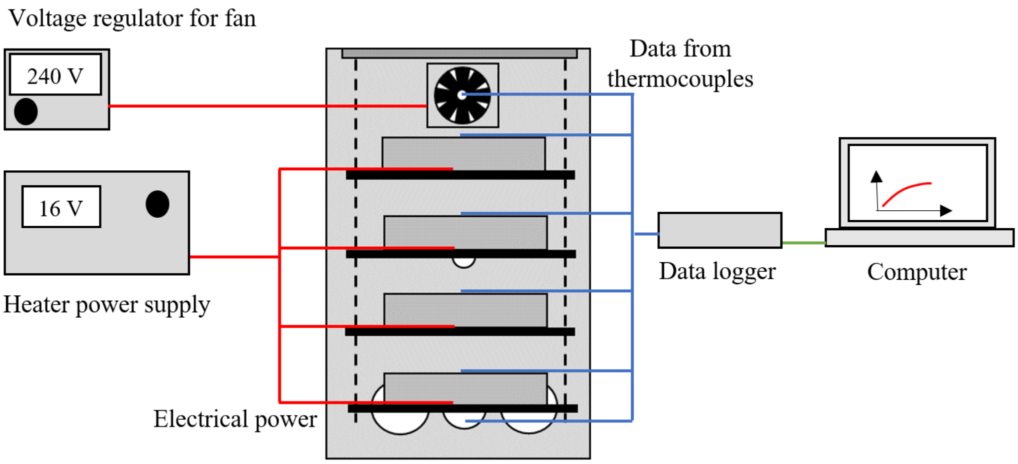


Figure 6. Schematic of the experimental setup with power supply and data acquisition.

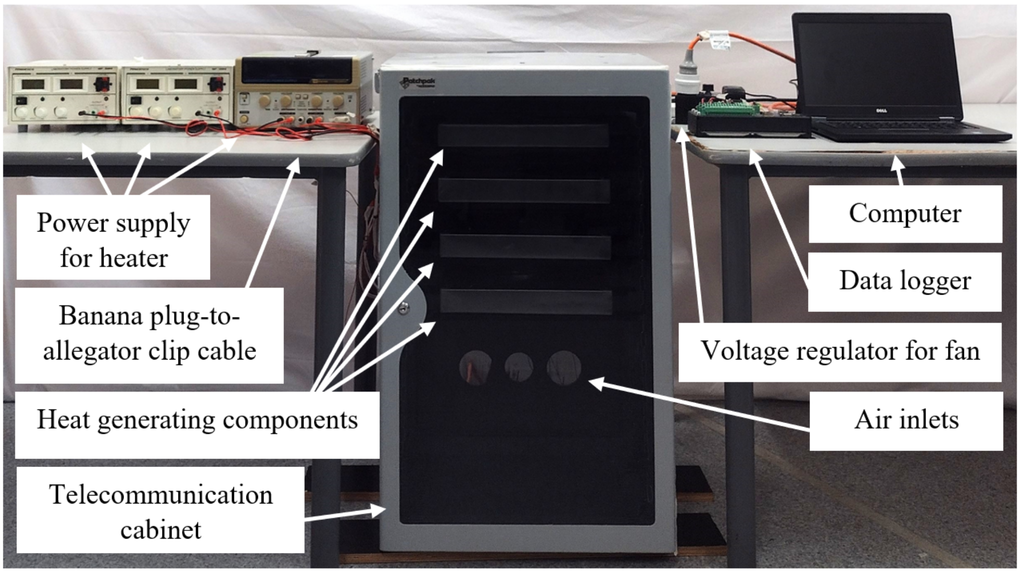


Figure 7. Experimental setup.

To measure the steady-state temperature of different parameters a steady-state criterion was established as a means to stipulate when the system had reached a steady-state condition. The steady-state criterion was based on a relative temperature change so that the meaningful response of the setup could be captured without consuming excessive experimental time. The criterion for the steady-state was proposed as less than 1 °C change in 5 min for all the temperature measurements. Thus, if the temperature did not change by more than 1 °C in 5 min the trial was deemed to have reached steady-state. During the experiments steady-state was reached after approximately 40 min with the proposed criterion. Figure 8 shows a sample of the transient temperature data with inclination of 0°, staggered offset distance of 25 mm, stream-wise spacing of 62 mm and fan speed that corresponded to an outlet velocity of 1.01 m/s. The first vertical red dashed line shows the point where the test was deemed to have reached steady-state according to the criterion, and the second vertical line shows the point when the test was terminated. Initially the temperature changed by 150% from 0 to 45 min, and only by a further 5% when tested from 45 to 90 min. Hence, the proposed steady-state criterion was deemed appropriate and was used for the experiment.

In the experimental procedure each test was conducted three times to ensure repeatability and identify errors. The range and increments of the tested parameters are presented in Table 1. Once the specific arrangement was setup inside the cabinet, the data logger was initiated to collect data. The power supplies were turned on to 16 volts (which corresponded to 2.47–2.57 amperes) and the fan was also set to the appropriate level depending on the test. LoggerNet software was used to view the real-time temperature of all the thermocouples on a computer. Thereafter the temperature and time were observed for each thermocouple. Once the change in temperature over 5 min was less than 1 °C, for all the thermocouples, the test was stopped and the data was saved. All the tests were conducted in an air-conditioned office space.

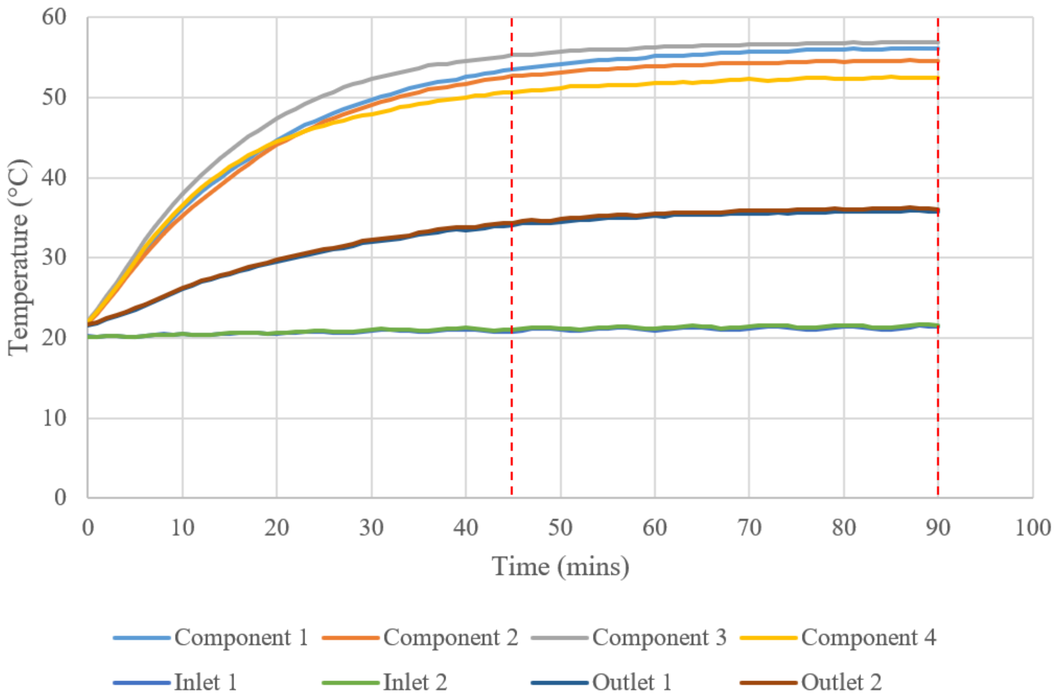


Figure 8. Sample of transient data for the case with inclination of 0°, staggered offset distance of 25 mm, stream-wise spacing of 62 mm and fan speed with an outlet velocity of 1.01 m/s.

Table 1. Range and increments of the tested parameters.

Parameter	Range	Increments
Inclination (degree)	0–90	0, 5, 10, 15, 30, 60, 90
Stream-wise spacing (mm)	29–108	29, 62, 83, 108
Staggered (offset, mm)	0–50	0, 15, 25, 50
Fan (outlet air velocity, m/s)	0.46–1.57	0.46, 0.49, 1.01, 1.57

2.2. Data Analysis

The data obtained from the tests were analysed to evaluate the heat transfer capacity of the tested conditions. Firstly, the energy balance of the system was assessed, where energy was supplied in the form of electric power to the heaters which was then dissipated by heat transfer. Heat transfer occurs in three ways which include conduction, convection, and radiation [40]. Therefore, a thermal energy balance can be generated by considering the heat-generating and -transferring mechanisms. The energy balance of the telecommunication cabinet system is as follows,

$$Q_{Conv} = Q_{Elec} - Q_{Cond} - Q_{Rad}, \tag{1}$$

where Q_{Elec}, which is the the product of the voltage and current supplied to the heaters, represents the heat generated by the heaters, and Q_{Cond}, Q_{Conv}, and Q_{Rad} represent the heat loss as a result of conduction, convection and radiation, respectively. All the heat generated by the heater was assumed to be dissipated by convection, conduction, and radiation only, and all other forms of heat transfer into the system were assumed to be negligible. Subtracting the conductive and radiative heat transfer terms shows how much heat is removed via convective heat transfer, as shown in Equation (1) [41].

The conductive heat loss of the system was assumed to be negligible as the heat transfer path for conductive heat loss was deemed inadequate for sufficient heat transfer. Radiation heat losses were calculated with the assumption that the heat-generating silicon pads were 'grey' (suggesting that the properties are independent of wavelength) and 'diffuse' (properties are independent of direction) [40,42]. The heat transfer by radiation was estimated to be 13.23% of the supplied power using a resistance-network calculation. The remaining heat was assumed to be transferred by convection.

The primary mode of heat transfer was assumed to be from convection and the non-dimensional convective heat transfer coefficient (Nu) was used to characterise the heat transfer of each parameter. Q_{Conv} was calculated based on Equation (1), with the convective heat transfer coefficient h calculated by,

$$h = \frac{Q_{Conv}}{A_s[T_s - 0.5(T_{in} + T_{out})]}, \tag{2}$$

where A_s is the combined surface area of all the heat-generating components, which was calculated based on the manufactured dimensions of the aluminium components, T_s represents the sum of all the individual temperatures of the components, and T_{in} and T_{out} refer to the averaged inlet/outlet temperatures, respectively, [3,19,41].

Equation (3) shows the formula to calculate Nu. The characteristic length was denoted by L_c and was taken to be the largest dimension of one aluminium component, which was 335 mm. The thermal conductivity, k, of the fluid (air), was obtained from published values at the film temperature (evaluated as the average inlet and outlet air temperatures of the cabinet) [40,41].

$$Nu = \frac{hL_c}{k} \tag{3}$$

The Reynolds number (Re) was calculated using Equation (4), in which D represents the diameter of the air outlet of the telecommunication cabinet, which was 56 mm, V_f is the exiting fluid velocity, and ν is the kinematic viscosity evaluated at the fluid temperature.

$$Re = \frac{DV_f}{\nu} \tag{4}$$

Table 2 presents the maximum uncertainty of the main parameters of this study. An error analysis was conducted and found that the overall maximum probable errors calculated for Nu of the inclination, staggered, stream-wise spacing and fan speed configurations and conditions were 4.46, 4.34, 4.34 and 4.45%, respectively.

Table 2. The maximum uncertainties of the main parameters.

Parameter	Maximum Uncertainty
Temperature	±3 °C
Voltage	±0.05 V
Current	±0.005 A
Length	±0.005 m
Velocity	±0.01 m/s

3. Results and Discussion

To investigate the heat transfer in the telecommunication cabinets, the effects of the component inclination (0–90°), staggered arrangement (with a centre-offset of 0–50 mm), stream-wise spacing (29–108 mm), and airflow rate (Re between 1604 and 5539) were analysed. The following sections present a brief synopsis on the effects of different internal configuration and airflow rate on heat transfer.

3.1. Component Inclination

Component inclination refers to inclining the angle of internal telecommunication components with respect to the horizontal. The aim was to promote air passage between individual components with a favoured air inlet (lowered side) and air outlet (elevated side) in order to reduce stagnation. Auxiliary components such as the fan, heaters, power packs, and data logger were kept the same; however, when the 30° and 60° tests were conducted, 100 mm steel spacers were used to ensure that the stream-wise spacing was kept the same between the components. It was assumed that the presence of the spacers did not affect the heat transfer as the 30° case was also tested without spacers (for an outlet velocity of 1.01 m/s) and a negligible difference was observed.

Figure 9 shows the effect on Nu of the inclination (in the range of 0–90°) at three different conditions of; outlet fan off (natural convection condition), and outlet fan on (forced convection condition) with outlet velocities of approximately 1.01 m/s and 1.57 m/s.

It was observed that the inclination had noticeable effect on Nu, which can be approximated empirically by quadratic correlations with the experimental results. The results further demonstrate that airflow rate significantly affects the heat transfer as it resulted in three distinct heat transfer correlations. The quadratic correlations had the coefficients of regression of $R^2 = 0.95$ for the natural convection case, and 0.98 and 0.96 for the induced outlet air velocity of 1.01 m/s and 1.57 m/s cases, respectively. It should be noted that the zero angle data point was excluded to obtain a better fit for the induced outlet air velocity of 1.57 m/s case, as the horizontal case (at zero angle), involves distinctively different flow and heat transfer behaviours.

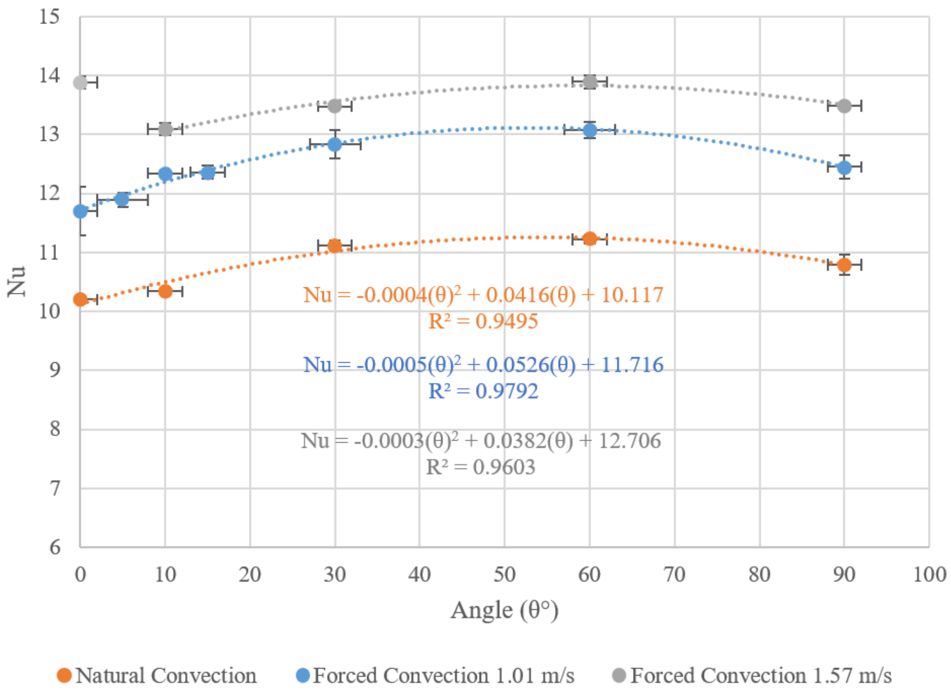


Figure 9. Effect of component inclination on Nu.

In general, it is seen that increasing the inclination corresponds to an improvement in the Nu. This finding is in agreement with other studies that reported an increase in heat transfer when the inclination angle is increased with respect to the flow direction [3,19,25]. However, the parabolic trend suggests that the maximum heat transfer occurs at a specific angle. It can be seen that the maximum heat transfer occurs between 30 and 60°. For all cases the highest Nu values were captured at an angle of 60°. With the measured data an improvement of 9.99%, 11.79%, and 0.09% was recorded for the natural convection, convection at 1.01 m/s and 1.57 m/s outlet flow velocities, respectively. The increase in Nu as a function of the angle for the inclination below 60° is likely associated with the improvement in the air passage path between components. In the traditional rack-wise horizontal mounting configuration, heat dissipated from the topmost component is carried away by the air outlet. However, the warmer air of lower components is mostly entrapped between the gap of adjacent components. The angled configuration promotes air passage in the gap between components. Air enters the gap between the components from the lowered side and once heated up by the components, the air is expected to rise and exist from the raised side, as shown in Figure 10.

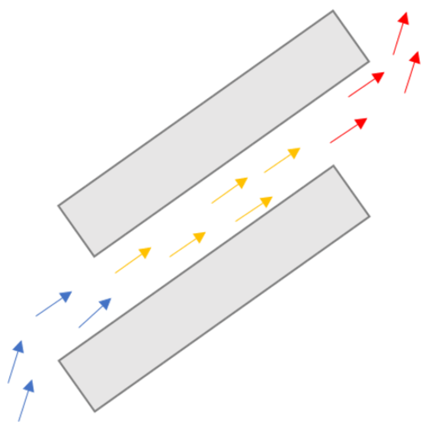
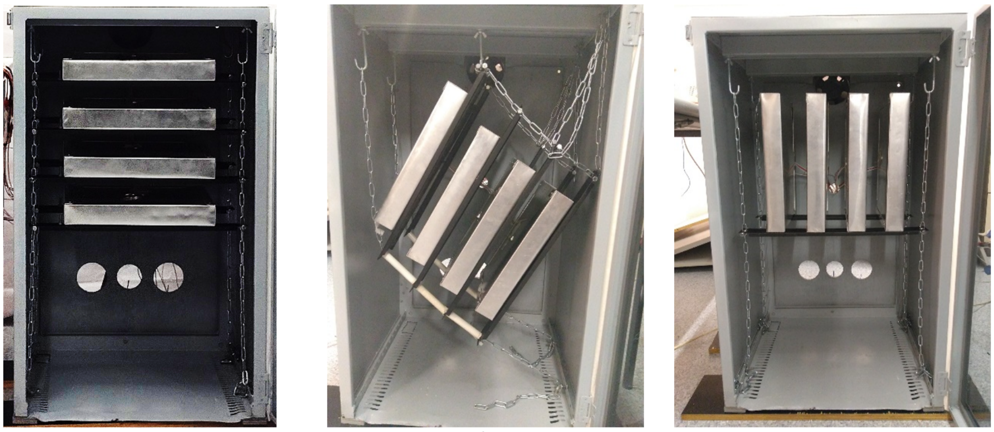


Figure 10. Heat transfer enhancement from component inclination.

The second mechanism that would have influenced the heat transfer is assumed to be associated with the vacant space between the components and the side walls of the cabinets. When the components are set to the 0° and 90° cases there is a specific distance to the cabinet walls. Therefore, a certain portion of the air flows through the sides without passing through space between the components. However, when the components are inclined the vacant space on the sides is notably less, as shown in Figure 11. Consequently, the area for the fluid to by-pass the components is reduced and more incoming cool air is forced to pass through the gap between the components in the central region of the cabinet. Hence it is assumed that reducing flow by-pass through the space between the edge of the components and the side walls led to rise in Nu, especially for the 60° case.



(a) (b) (c)

Figure 11. Inclination angles of (**a**) 0°, (**b**) 60°, and (**c**) 90°.

The Nu–inclination correlation implies that the heat transfer in telecommunications cabinets is controllable by the inclination of the components and Re. The relations also show that heat transfer in telecommunication cabinets is enhanced by approximately 10% for the natural convection case and outlet air velocity of 1.01 m/s conditions without the addition of extra cooling power. Therefore, inclining components inside telecommunication cabinets offers a possibility to save energy and cost by promoting improved passive heat transfer.

3.2. Staggered

The staggered component arrangement refers to translating the individual components in the horizontal direction so that the centre of each component is offset from the centre of the cabinet. Each component was translated to alternating sides to have a staggered configuration. The staggered configuration was tested to observe if staggered components offer an improved heat transfer mechanism in comparison to the standard in-line arrangement. The relation between (staggered) offset distance (x/W) and Nu in the range of 0–50 mm is shown in Figure 12. The offset distances (x) have been non-dimensionalised by dividing with the internal cabinet width of $W = 530$ mm.

The results from Figure 12 suggests that a staggered arrangement does not significantly influence Nu. The offset distances of 0–0.09 mm/mm (which corresponds to 0–50 mm) were tested for natural and forced convection conditions. The outlet air velocity for the forced convection case was set to 1.01 m/s. An approximately linear correlation was fitted to the experimental data; however, no meaningful trends were evident as shown by the R^2 values of 0.5 and 0.4 for the natural/forced convection conditions, respectively.

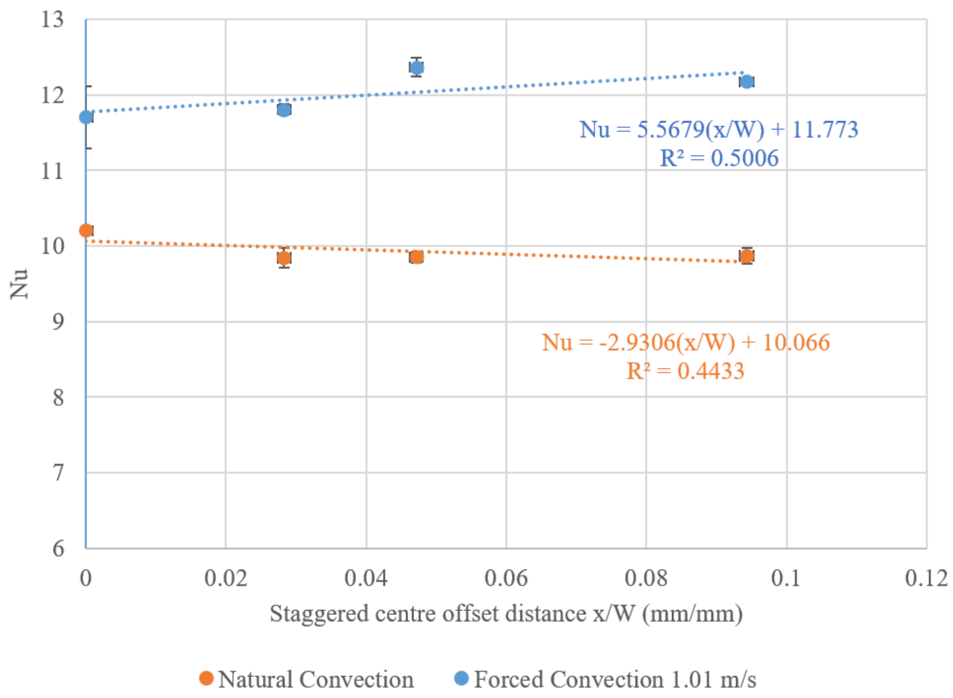


Figure 12. Effect of staggered arrangement on Nu.

Staggered or offset components do not greatly enhance the heat transfer inside the telecommunication cabinet. However, the results indicate that Nu is influenced by forced air flow as Nu was increased for all the increments that were tested for forced convection. The reason for negligible improvement maybe associated with the flow pattern arising from the specific cabinet used for testing. Nevertheless some studies have reported improvement in heat transfer for the staggered component arrangement [33,34]. The intention of the offset components was to initiate improved air passage through the central region of the cabinet as air would ideally enter from the bottom, flow between all the components and finally exit at the top. However, both the inlet and outlet of the tested cabinet were located on the back wall. Therefore, translating the components in the horizontal direction may not have affected the overall flow pattern enough to improve heat transfer. Nevertheless, the small improvements obtained for the offset components (tested with the air inlet and outlet located on the back wall), may have been associated with the increase in the unblocked surface of the components. When the components are offset less of the component's top surface is shielded by the adjacent component above (as the component above has been moved in the opposite direction). Therefore, warm air on the exposed/unblocked surface of the components can move up more freely than warm air in the shielded portion of the component.

The heat transfer of telecommunication cabinets with a similar location of the air inlet and outlet location as cabinet used in this study cannot be significantly enhanced by using staggered or offset arrangements. However, it may be feasible for other cabinet arrangements that have different air inlet or outlet locations (particularly if the inlet is located on the floor of the cabinet and outlet located at the top).

3.3. Stream-Wise Spacing

Stream-wise spacing refers to increasing the space between each component in the stream-wise direction. For the specific cabinet conditions tested, the stream-wise spacing corresponded to an increase in the vertical space between each component. The intention was to increase air passage between components due to the large unobstructed space as well as to reduce boundary layer effects. The effect of component stream-wise spacing, in the range of 29–108 mm, on Nu is presented in Figure 13. The stream-wise space (x), has been non-dimensionalised using the internal height of the cabinet which was $H = 860$ mm.

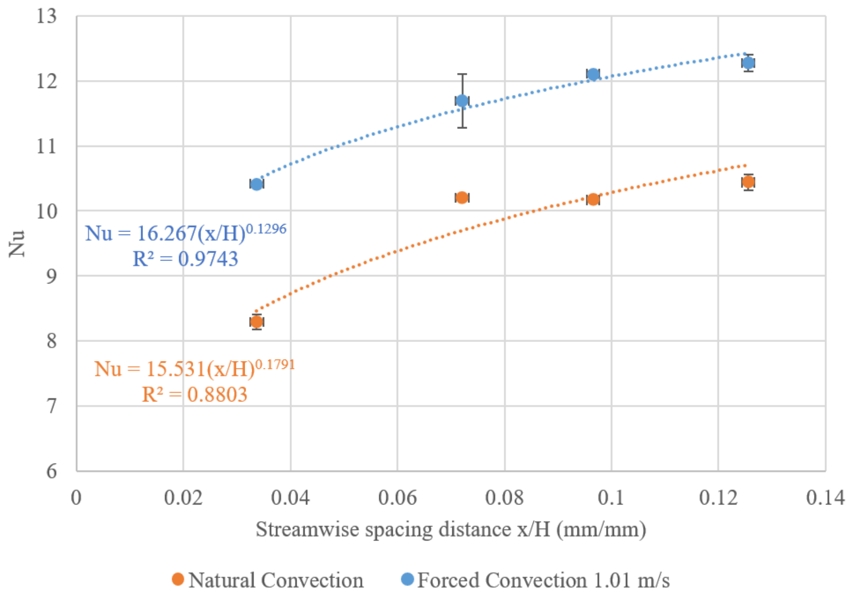


Figure 13. Effect of stream-wise spacing on Nu.

Each stream-wise spacing was conducted for natural and forced convection (outlet velocity of 1.01 m/s) conditions. A power function was fitted to both natural and forced convection datasets. It was found that stream-wise spacing of internal components had a significant effect on the heat transfer in telecommunication cabinets, especially at the low increments that were tested. For the natural convection case an enhancement of 23.03% can be seen between 0.03 and 0.07 mm/mm. Thereafter the improvement diminished. For the forced convection case, an initial improvement of 12.26% was recorded from 0.03 and 0.07 mm/mm which also diminished at higher spacings. Therefore, heat transfer increases rapidly; however, the effect is less pronounced as the distance is increased beyond a certain threshold. In effect, a total improvement of 25.9% was seen for the natural convection case and 17.8% for the forced convection case between the maximum and minimum spacings tested. This observation is in agreement with other published research [3,6,19,35]. The control/reference stream-wise spacing was set as 0.07 mm/mm (and not the smallest value tested 0.03 mm/mm) as it was assumed that in practice there would some stream-wise spacing. Consequently, the maximum enhancement from the control/reference case was 2.33% for natural convection and 4.94% when the spacing was tested under forced convection (at 1.01 m/s).

The space or gap between adjacent components significantly affects how much air can pass between adjacent components. When the space between components is minimal only a limited amount of air passes through, which limits the amount of cool air that enters and warm air that can flow away. However, when the space between components is increased the air flow rate also increases, thus promoting improved heat transfer. The results, (as per Figure 13), show that Nu increases with increasing space. The diminishing rate of improvement from one increment to the next, is likely associated with other geometric constraints. It is expected that air initially enters through the gap between components (at the back), moves over the surfaces, and exits through the side or front gaps between the components. Increasing the vertical gap beyond a certain value may not have any notable effect on heat transfer as the overall width/depth of the cabinet remains the same and may not allow for more air to pass through the cabinet. This observation suggests that stream-wise spacings offer a solution to improve heat transfer in telecommunication cabinets, especially in the critical range.

3.4. Fan Speed

The fan speed tests were used to observe the effect of increasing the air flow rate inside the telecommunication cabinet. The increased flow rate can draw in and push out more air to enhance cooling in comparison to the natural convection arrangement (where the fan was off) [19,39]. The results of Nu against four different fan speeds are presented in Figure 14. Tests were conducted in the range of 0.46–1.57 m/s which corresponds to Re in the range of 1604–5539. During the fan speed tests the inclination was kept to 0°, staggered or offset distance was 0 mm and the stream-wise spacing was 62 mm (or 0.07 mm/mm).

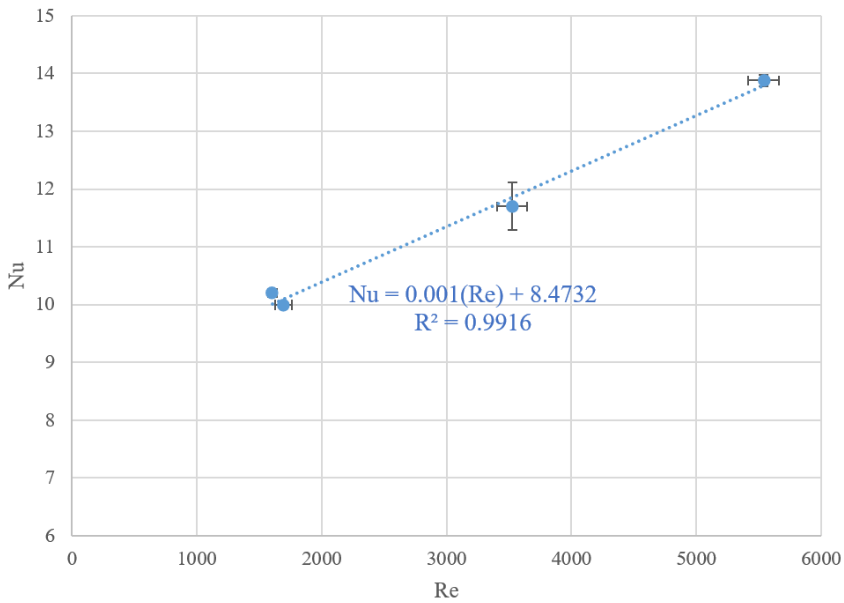


Figure 14. Effect of Re on Nu.

A clear linear trend was evident between the rate of heat transfer and the rate of air flow over the four tested cases. When the air is constantly refreshed by cooler air entering the cabinet, through the inlet and warmer air expelled, heat transfer takes place more efficiently and the cooler air comes into contact with the components and removes the heat. The notable improvement of 36% is evidence from the strong trend (as per Figure 14). No evidence was found relating to an optimal range of fan speed that would yield the most efficient cooling. In the range of tested fan speeds, Nu did not diminish.

The maximum Nu enhancements from inclining internal components, using stream-wise spacing, staggered arrangement, and fan speed are summarised in Table 3. All improvements were calculated with respect to the control/reference case. The control/reference case has an inclination of 0°, staggered offset of 0 mm/mm and stream-wise space of 0.07 mm/mm. The results from the control/reference case with natural and forced convection (with an outlet velocity of approximately 1.01 m/s or Re = 3526) are presented separately in Table 3.

The enhancement data showed that most of the tested configurations improved the heat transfer in comparison to the control/reference arrangement. However, certain parameters show much higher improvement while others are marginal.

Table 3. Maximum Nu enhancement from tested configurations with respect to control/reference case.

Parameter	Control with Natural Convection (%)	Control with Forced Convection 1.01 m/s (%)
Inclination	9.99	11.79
Staggered (offset)	−3.26	5.69
Stream-wise spacing	2.33 [a]	4.94 [b]
Fan (outlet air velocity)	36.02	18.64

[a] For the natural convection tests, an improvement of 25.9% between the smallest and largest tested stream-wise spacing was observed. [b] For the forced convection tests, an improvement of 17.8% between the smallest and largest tested stream-wise spacing was observed.

4. Conclusions

The effect of different configurational arrangements such as component inclination (0–90°), staggered arrangement (with a centre-offset of 0–50 mm), and stream-wise spacing (29–108 mm) as well as the effect of airflow rate ($Re = 1604$ to 5539) on a telecommunication cabinet was studied. The key findings of the present study can be summarised as follows:

1. The inclination of internal components increased Nu by 9.99% for the natural convection and 11.79% for the forced convection condition over the tested range. An approximately quadratic correlation was observed between Nu and the inclination, where the maximum was observed to occur between 30 and 60°.
2. The staggered arrangement of internal components decreased Nu by 3.26% for the natural convection condition but increased by 5.69% for the forced convection condition over the tested range. From the tested range and arrangement, increasing the centre offset of the staggered components with respect to the cabinet did not influence Nu in any significant manner.
3. The improvement from stream-wise spacing was 2.33% and 4.94% for the natural and forced convection conditions, respectively, when compared to the control/reference arrangement. Increases of 25.90% and 17.80% in Nu were observed for the natural convection and forced convection conditions, respectively, when the initial and maximum stream-wise spacing increments were considered. Furthermore, it was identified that increasing the stream-wise spacing beyond a certain threshold had a diminishing effect.
4. Fan speed had the most pronounced effect on heat transfer, where Nu increased by 36% over the tested range. In all cases (inclination, staggered and stream-wise spacing), the fan speed was observed to improve Nu significantly.

The outcomes of the investigation provide insight into which parameters influence the heat transfer in a telecommunication cabinet with the inlet/outlet located on the back wall. Moreover, the results of the investigation can be used by manufacturers to compliment the design of future telecommunication products. Further investigations could potentially focus on the effect of different air inlet/outlet locations and the use of computational fluid dynamics (CFD) simulations to understand the intricate flow patterns.

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Abbreviations

The following abbreviations are used in this manuscript:

A_s	Combined surface area of all the heat-generating components, m^2
D	Diameter of the cabinet air outlet, m
h	Convective heat transfer coefficient, W/m^2·K
H	Height of the cabinet, m
k	Thermal conductivity, W/m·K
L_c	Characteristic length, m
Nu	Nusselt number
Q_{Elec}	Electrical power, W
Q_{Cond}	Conduction heat transfer rate, W

Q_{Conv}	Convection heat transfer rate, W
Q_{Rad}	Radiation heat transfer rate, W
Re	Reynolds number
T_{in}	Averaged cabinet inlet temperature, K
T_{out}	Averaged cabinet outlet temperature, K
T_s	Summation of all surface temperatures of the components, K
V_f	Fluid velocity, m/s
W	Width of the cabinet, m
x/H	Non-dimensionalised distance with respect to the height of the cabinet
x/W	Non-dimensionalised distance with respect to the width of the cabinet

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