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Determination of Natural Convective Heat Transfer Coefficient for Plain Knitted Fabric via CFD Modeling

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Abstract
In this study, the natural convective heat transfer coefficient for plain knitted fabric is estimated by numerical modeling, and the applicability of commercial software using the finite volume method (FVM) to textile problems was investigated. For this purpose, the convective heat transfer coefficient of plain knitted fabric made from cotton was first measured experimentally with the help of the in-house set-up developed. Then a plain weft knitted fabric model was created using CATIA, and Gambit and Fluent software was employed for numerical analyses. Finally experimental results were compared with those obtained from the numerical method. Based on the results, it was concluded that the numerical modeling simulated the related experimental study well. It was also stated that the numerical model developed has the potential to obtain the heat convection coefficient of the material considered for different conditions such as environment temperature etc.

Key words: *finite volume method (FVM), convective heat transfer coefficient, knitted fabric.*

Introduction

The coupled heat and moisture transfer in textile fabrics has been widely recognised as being a very important issue for understanding the dynamic thermal comfort of clothing during wear [1 - 5].

Chen et al [6] measured clothing thermal insulation with very low perspiration and very heavy perspiration using a novel perspiring fabric thermal manikin. The results showed that clothing thermal insulation during heavy perspiration is significantly less than that with low perspiration. The differences vary from 2 - 8% depending on the increase in moisture accumulation within the clothing.

Hatch et al [7] confirmed that fabric structural features, not component fibres, are the most important controllers of thermal dissipation in the presence of moisture diffusion. Moreover heat transfer is highly related to fabric thickness, bulk density and the air volume fraction. Thermal transfer from a simulated sweating skin surface is strongly correlated with fabric porosity and air permeability. The measurement of transient heat transfer from the fabric surface to a contacting warm body suggested that the cotton knit would generate a cooler feeling [8].

Yoo et al [9] showed that as the thickness of the air layer increases, the buffering index increases; but an unnecessarily large air gap does not increase the buffering capacity efficiently. As the openness increases, the effect of fabric on the microclimate decreases gradually, losing its effect at 60% and approaching the value of nude skin.
Gibson [10] implies that air permeability becomes particularly important in the case of an air space between the fabric and sweating skin simulant surface.

Li et al [11] showed that the heat transfer process, which is influenced by fabric thickness and porosity, significantly impacts moisture transport processes.

Holcombe and Hoschke [12] showed that a relationship exists between the thermal conductivity of the fabric and the thermal conductivities of air and fibre, together with the packing factor of the construction. The entrapped air is by far the greatest determinant of fabric conductivity, and within the range of typical textile fibre conductivities, the contribution of fibre is relatively small.

Fan and Cheng [13] found that most of the changes in the temperature distribution within the fibrous battings sandwiched by the inner and outer layers of thin covering fabrics take place within 30 minutes of exposure to cold conditions. The distributions of temperature and water content within the battings are affected by the moisture absorption properties of fibres and the density or porosity of the batting. The effect of the permeability of the covering fabric on heat loss and water content distribution is relatively small, especially when the batting is a dense or less permeable type. Inner fibrous battings with higher fibre contents, fine fibres, greater fibre emissivity, higher air permeability, a lower disperse coefficient of surface free water and a lower moisture absorption rate cause less condensation and moisture absorption, which is beneficial to thermal comfort during and after exercising in cold weather conditions [14].

The aim of Purvis and Tunstall’s [15] study was to examine the effects of two different sock types on foot skin temperature and to investigate any impact on whole body thermoregulation and energy expenditure. The choice of running sock does not appear to have a thermal or physiological impact during exercise. The ergonomic sock was perceived to be cooler and was preferred, which suggests

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that subjective perceptions may be more important than objective measurements when selecting a sock for wear during prolonged exercise.

Hossain et. al [16] investigated the airflow and heat transfer through fibrous webs via a mathematical model developed and interpreted by commercial software - Fluent. They also derived a thermal energy equation incorporating the heat of fibre fusion in a fibrous web. The model solved continuity, momentum and energy equations based on the porous media concept and was applied for different process settings and web properties, such as airflow velocity, web thickness and porosity. In this study, Fluent was employed in order to obtain local values of air velocity, temperature and the melt fraction of fibres. As a result of using the model developed, they provided a very important parameter; the time required to melt fibres throughout the web at different process conditions and web properties, which can be used potentially in the design of through-air bonding process. Ci and Wang [17] analysed the heat transfer process through kapok insulating material at different temperatures against wind speed with a theoretical model of heat flow combined with conduction, convection and radiation. The finite element method was used for the study. They performed an experiment in an artificial climate chamber and obtained good agreement between theoretical and experimental results. They also used the finite element method for the study and concluded that transient heat loss from the human body did not change with the wind speed when the fabrics came into contact with the human body.

V. K. Kothari and D. Bhattacharjee [18, 19] worked on an exhaustive exercise to predict the thermal resistance of woven fabrics with the help of a first principles based model and application of computational fluid dynamics (CFD) to simulate convective heat transfer. In the first part of the study, they developed a mathematical model to predict conductive and radiative heat transfer through fabrics and validated the model with thermal resistance values obtained from Alambeta. According to the results obtained in Part 1, they concluded that the thermal resistance of woven fabrics can be calculated using a lumped model. It was also stated that the sum of conductive and radiative heat transfer based on the mathematical model developed gives good prediction of thermal resistance values when compared with existing experimental data. In the second part of the work, however, simulation of convective heat transfer in CFD was performed to find the coefficients of convective heat transfer, and the values were validated with actual results obtained from experiments. It was also shown that there was good agreement between the mathematical model, CFD analysis and the experimental values obtained.

Figure 1. *Places of thermo receptors in the hot unit.*

Since there are very few studies concerning the application of numerical analysis to textile fabrics, this study was also performed to investigate the applicability of numerical modeling based on the finite volume method (FVM) for knitted fabrics.

Experimental Details

In this study, plain jersey socks were produced from Ne 30/1 cotton ring yarn on a Nagata D210 double cylinder hosiery machine of 176 needles. Unlike ordinary socks, nylon and elasthane was not utilised in the production for eliminating the effect of these fibres on the heat transfer behaviour of the samples. After the samples had been produced, they were conditioned under standard atmospheric conditions (T = 21 ± 1 °C and RH = 65 ± 2%) for one week.

Sock ironing plates of the type used in the textile industry were manufactured for the in-house set-up, with some modifications, in order to measure the natural thermal convectivity of the sock samples [20]. As may be seen in ***Figure 1***, four thermocouples were attached to both the lower and upper parts of the ironing plate. An electric heater was placed between these parts before welding in order to construct a hot plate. The surface temperatures of the hot plate were held at between 35 °C and 40 °C.

The plate was covered by the sock and positioned in a box (h = 80 cm, w = = 60 cm, d = 50 cm) in order to block the environmental effects (i.e. radiation from sunlight, air circulation existing in the laboratory). Furthermore, to measure the outer temperature, 20 thermocouples were located in the box. In **Figure 2** a schematic of the system can be seen. Where T_H is the temperature of the sample, and T_{wall} is the temperature of the box wall.

DC power was used to feed the electrical heater in order to protect the thermocouples from electrical inductance. Temperature data were collected by a computer-controlled data acquisition system [20].

Modelling and finite volume analysis

Modelling

A fabric model was developed using CATIA V5R16. Due to the fact that fabric properties were continuous unless the fabric structure and/or tightness were changed [21], it was assumed that modelling one repeat of the structure was adequate to represent the whole sample. For modelling purposes, photos of the structure were initially taken using a high-resolution digital camera, and both course and wale spacing values were measured using these photos. To have the third dimension of the fabric, thickness values were also measured in accordance with the relevant

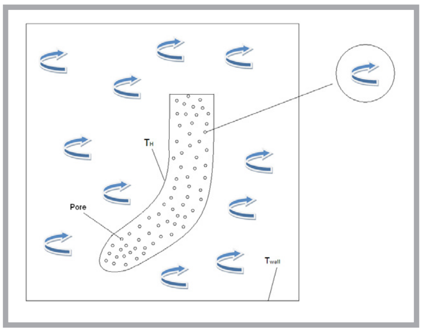

Figure 2. *Position of sock during convection.*

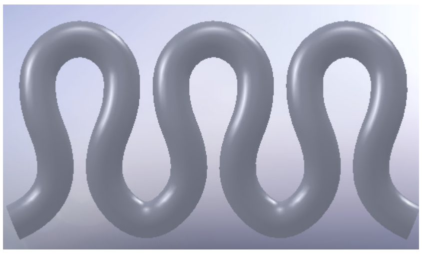

Figure 3. *Technical face of the fabric structure.*

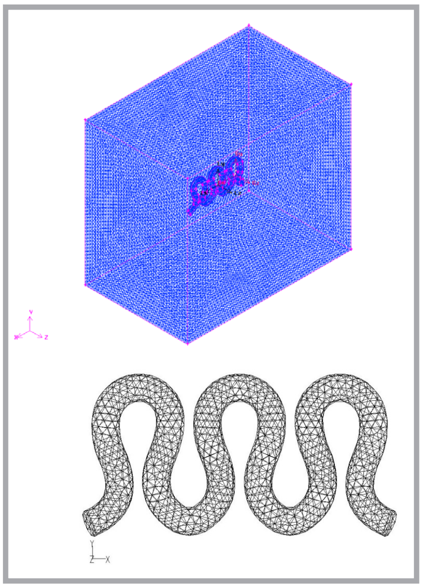

Figure 4. *Numerical domain for the air (top) and mesh structure of the solid model (botton).*

standard (BS 2544), and then the data were entered into CATIA V5R16 [22].

Numerical analysis of a knitted fabric, on the other hand, is a difficult problem. This is partly because the multi-scale nature of the fabric (macro-scale), which is composed of yarns (meso-scale) and fibres (micro-scale) in turn, leads to a complicated geometry that is hard to identify [24]. Moreover, defining the geometry to which the mesh is applied presents a quite complicated task which needs to be solved. Consequently for simplification purposes, the assumptions given below were made:

1. The plain knitted fabric is made of frictionless, inextensible, incompressible and naturally straight yarn, which can be considered as homogeneous elastic rods.
2. The fabric is formed of planar loop structures. All loops within the fabric keep an identical configuration.
3. No plastic deformation takes place in the yarn when the fibres make contact with each other, and their cross section is circular as well as constant throughout all of the knitted part [24]. Even though the section shape changes along the yarn, variations of yarn thickness and width are small enough to be neglected [23].
4. Contact resistance between the sample and heating unit is neglected, and thus the temperature of both is assumed to be the same.
5. In natural convection, heat is transferred by means of fluid recirculation due to the density variation caused by the temperature difference between two mediums. In the case of the present study, these are the knitted fabric sample (higher temperature source) and the air filling the box (lower temperature source). Thus it can be concluded that a great amount of heat is transferred by the air that fills the box, surrounding the fabric sample. The remaining amount of heat transfer occurs by means of the recirculation of the still air pores in the fabric [19]. However, once the amount of air in the pores is comparable with that of air surrounding the sample (filling the box), it can be concluded that the recirculation of air existing in these pores does not contribute to the heat transfer as much as the surrounding air does. Referring to the explanation given, the following assumption can be made for the numerical domain: Since the heat transfer is dominated by natural convection, using either a whole fabric sample or just a piece of it in the computations does not give any significant difference in the results, although the former one contains relatively more still air pockets. Thus, the numerical domain was constructed where a one course three wide fabric was employed to simulate the whole sample. Despite the micro scale of the numerical domain constructed using the assumption given above; conservation equations for the macro scale were concluded to be applicable for the numerical analysis, according to existing literature [22].

The fabric structure, which is presented in ***Figure 3***, was modeled accordingly.

Governing equations and numerical analysis

The governing equations utilised in the numerical analysis are based on the Boussinesq approximation for natural convection. Assuming the fluid is incompressible, the continuity and momentum equations are expressed as the following, respectively.

$$\nabla V = 0 \tag{1}$$

$$\rho \frac{dV}{dt} = \rho g - \nabla p + \nabla \cdot \left(\mu \nabla V \right) \tag{2}$$

$$\rho = \rho_\infty + \frac{\partial \rho}{\partial T}\left(T - T_\infty \right) \tag{3}$$

$$\beta = -\frac{1}{\rho_\infty}\left(\frac{\partial \rho}{\partial T} \right)_p \tag{4}$$

By combining ***Equations 4, 3 & 2***, the momentum equation for natural convection becomes,

$$\rho \frac{dV}{dt} = \left(-\nabla p + \rho_\infty g \right) + {}_\infty g \quad T \quad T_\infty$$
$$- \rho_\infty g \beta \left(T - T_\infty \right) + \nabla \cdot \left(\mu \nabla V \right) \tag{5}$$

The general energy equation is also written below.

$$\rho c_p \frac{dV}{dt} = \nabla \cdot \left(k \nabla T \right) + q''' + \nabla V \tau \tag{6}$$

A fabric model was obtained in CATIA V5R16 and was interpreted by Gambit and Fluent in order to create a numerical model and mesh structure, as well as to perform numerical analysis, respectively. The numerical model was constructed in such a way that the sample hanged in a box, like in the experimental set-up. With the help of the fabric modeling assumption given above (see also ***Figures 3*** & ***4***), different node densities are provided in order to obtain the grid independency and to avoid divergence problems. Furthermore, since the structure was simulated via a single fabric over the repeat of the structure, the dimensions of the box containing the numerical model were scaled down by using the surface area ratio between the whole structure (used in the experiment) and the sample model considered. Accordingly, a numerical domain and mesh structure were created via Gambit, presented in ***Figure 4***.

As a result of the parametric study on grid independency, 111669 nodes were found to be adequate for the numerical analysis. Regarding the boundary conditions in the experimental set-up, a constant temperature of '37.6 ± 0.5 °C' and '23.3 ± 0.5 °C' were given to the sample and box walls, respectively. Since the sample was hung in a box and natural convection was the leading parameter for heat transfer, the gravity effect along the –y direction was also considered in the simulations. Thermo physical properties at the arithmetical average value of the temperature boundary conditions mentioned above (i.e. 30.42 °C) were taken for air filling

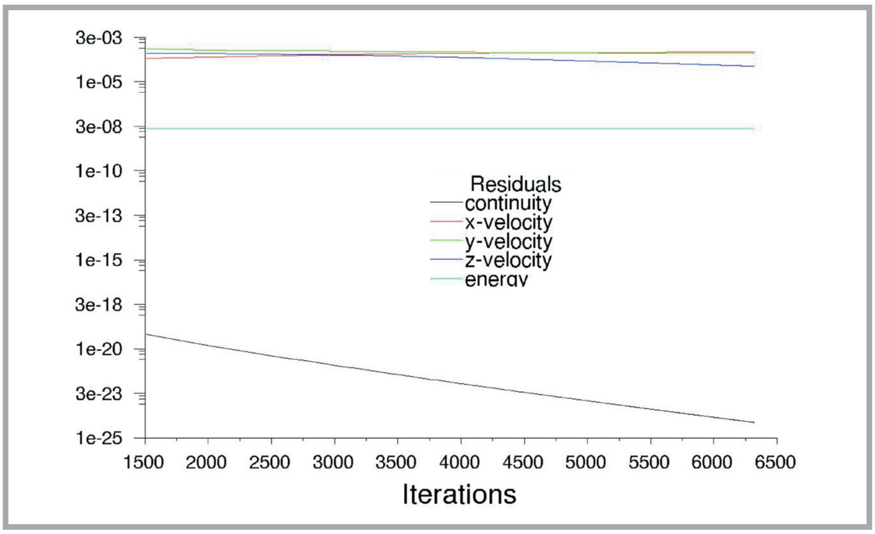

***Figure 5.** Residual values for the convergent case.*

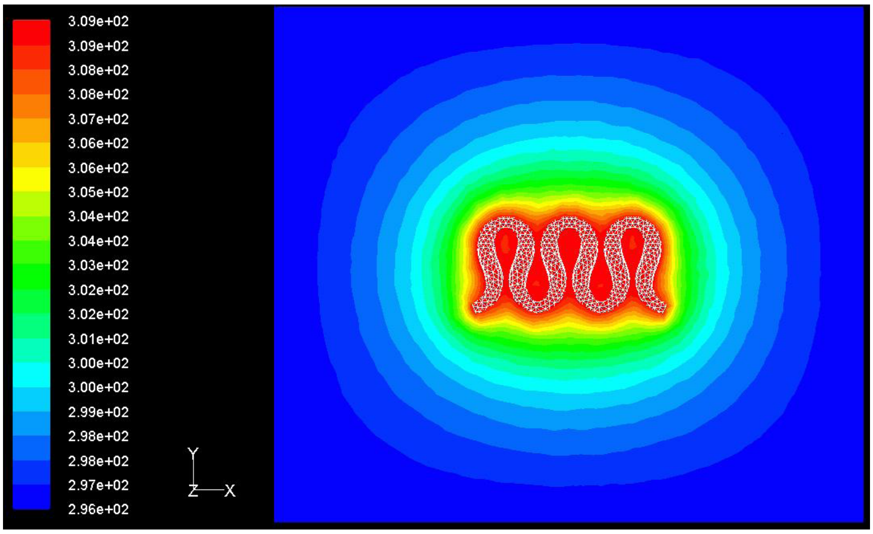

***Figure 6.** Temperature distribution at z=0 plane in the box.*

the box, while those of the material of the structure i.e. cotton - which are not available in the material library of Fluent - were taken from literature [22, 25]. All values employed are listed in ***Table 1***. In the numerical analysis, air was assumed to be an ideal gas, gradients were evaluated via the Green-Gauss Cell Based method, and the SIMPLE algorithm was employed for pressure-velocity coupling. As spatial discritisation, the second order upwind scheme was used for momentum and energy, while PRESTO! was utilised for pressure. The average of the temperature boundary conditions (i.e. 30.42 °C) was taken as the operating temperature for Boussinesq parameters. The 'WALL' type boundary condition was employed for the box wall and sample wall.

***Table 1.** Thermo physical properties of air and cotton [22, 25].*

Properties	Air	Cotton
density - ρ, kg/m^3	1.225	1520
specific heat - c_p, J/kgK	1006.43	1210
heat conduction coefficient - k, W/mK	0.0242	0.0364
dynamic viscosity - μ, kg/ms	1.7894×10^{-5}	-

Numerical iteration was kept running until the heat flux density value calculated from the sample remained the same, taken as the convergence criteria which indicated that the numerical analysis was stabilised. Residual values for the convergent case can be seen in ***Figure 5***.

Results and discussion

The experimental system explained above was heated for 3 hours until it reached thermal equilibrium, and 64 measurements were made after having reached a steady state. To calculate the heat convection coefficient of the cotton samples, the following formula was derived;

$$q'' = \frac{\dot{Q}}{A_S} \tag{7}$$

$$q'' = h\left(T_H - T_{Air}\right) \tag{8}$$

where;

q'' - heat flux density in W/m^2
$\dot{Q}$ - heat transfer rate in W
A_S - area of sample in m^2
h - Convective heat transfer coefficient in W/m^{2}K
T_H - temperature of the sample in K
T_{Air} - outer temperature in K.

At the end of the experimental study, it was found that the natural heat convection coefficient of the cotton sample was 13.1 W/m^{2}K. The uncertainty value for the heat convection coefficient was calculated to be 11%.

The temperature distribution at the 'z = 0' plane (middle of the box), obtained from numerical simulation, is shown in ***Figure 6***. Since the steady state condition was simulated and the natural heat transfer in the box was bidirectional, isothermal regions of circular shape were obtained, as may be seen in **Figure 6**. All temperatures given in the figure are in K.

According to the temperature values given as the boundary conditions, the heat flux density from the structure to the surrounding medium was obtained as a result of the numerical calculation. By substituting temperature values and the numerical heat flux density in ***Equation 8***, the numerical heat convection coefficient for the model was estimated to be 13.2 W/m^{2}K. A comparative study of these results showed that there was good agreement between numerical and experimental values of the natural convective heat transfer coefficient. Thus, it can be concluded that numerical analysis by means of the finite volume method may be utilised to simulate the heat transfer properties of knitted fabrics.

Conclusions

The main objective of the paper was to investigate the applicability of commercial software performing finite volume analysis to textile problems, in particular to the heat transfer behavior of plain knitted cotton fabrics. For this purpose, experimental research was conducted in order to investigate the natural convective heat transfer coefficient of cotton plain jersey knitted fabric for given conditions. And then a plain weft knitted fabric was modelled using CATIA V5R16 and analysed via FLUENT.

Finally, good agreement between numerical and experimental values, with an uncertainty value of 11%, of the heat transfer coefficient for plain knitted cotton fabrics was obtained. The results also suggested that the study offers a valuable reference point for further studies on FVM applications to knitted structures.

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