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THERMALHYDRAULIC ANALYSIS OF FOUR GEOMETRICAL DESIGN PARAMETERS IN RIB-ROUGHENED CHANNELS

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The present article reports RANS simulations of the flow and heat transfer in a 2-dimensional rib-roughened passage. The effects of four different geometrical factors including rib profile, rib pitch-to-height ratio, rib height, and rib width are investigated. The Reynolds number, based on the channel bulk velocity and hydraulic diameter, is 30,000. Two low-Reynolds-number linear EVMs, namely the Menter k-ω-SST model and a variant of Durbin’s $v^2-f$ formulation, are examined. All computations are undertaken using the commercial CFD code STAR-CD. In comparison with experimental data, it emerges that the $v^2-f$ model generally returns more accurate results than the k-ω-SST closure.

1. INTRODUCTION

Rib-roughened surfaces are commonly used to enhance convective heat transfer by the promotion of higher turbulence levels. The drawback to such roughening is an increase in frictional and form drag and consequently much effort has been devoted to the optimization of roughness designs. The principal application of the present work is to the rib-roughened fuel pins of the UK fleet of Advanced Gas-cooled Reactors (AGRs). Results presented below are for a 2D-plane passage (Fig. 1), this study forming a precursor to the simulation of the more complex reactor geometry.¹

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¹see, for example, A. Keshmiri, Three-Dimensional Simulation of Simplified Advanced Gas-Cooled Reactor Fuel Elements, J. Nuclear Engineering Design (under review).

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There have been numerous studies of the flow dynamics and heat transfer characteristics of various rough surfaces. However, a full understanding of the detached flow physics remains elusive. This is largely due to experimental difficulties and high turbulence intensities that render measurement techniques inaccurate [1].

Experimental studies of flow over rib-roughened surfaces date back at least as far as 1950s. One of the early experiments on these types of surfaces were carried out by Webb et al. [2] who investigated pipe flow with repeated rectangular ribs, focusing

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**NOMENCLATURE**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>cross-sectional area of the channel</td>
</tr>
<tr>
<td>b</td>
<td>rib width</td>
</tr>
<tr>
<td>c_f</td>
<td>local friction coefficient</td>
</tr>
<tr>
<td>c_p</td>
<td>specific heat coefficient at constant pressure</td>
</tr>
<tr>
<td>C_p</td>
<td>pressure coefficient, ((p_s - P_{\text{ref}})/((0.5pU_b^2)))</td>
</tr>
<tr>
<td>D_e</td>
<td>hydraulic diameter, (4A/P)</td>
</tr>
<tr>
<td>H</td>
<td>channel height</td>
</tr>
<tr>
<td>k</td>
<td>height of the rib or turbulent kinetic energy</td>
</tr>
<tr>
<td>k_0</td>
<td>original rib height</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number, (qD_e/\lambda(T_w - T_b))</td>
</tr>
<tr>
<td>p</td>
<td>pressure</td>
</tr>
<tr>
<td>P</td>
<td>pitch, wetted perimeter</td>
</tr>
<tr>
<td>P_k</td>
<td>rate of shear production of (k)</td>
</tr>
<tr>
<td>p_s</td>
<td>static pressure at the wall</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number, (c_p\mu/\lambda)</td>
</tr>
<tr>
<td>(\bar{q})</td>
<td>wall heat flux</td>
</tr>
<tr>
<td>Re</td>
<td>Reynolds number, (U_bD_e/v)</td>
</tr>
<tr>
<td>T</td>
<td>temperature</td>
</tr>
<tr>
<td>T_i</td>
<td>turbulent timescale</td>
</tr>
<tr>
<td>U_i, u_i</td>
<td>mean, fluctuating velocity components in Cartesian tensors</td>
</tr>
<tr>
<td>U_c</td>
<td>friction velocity, ((c_w/p)^{1/2})</td>
</tr>
<tr>
<td>x,y</td>
<td>streamwise and wall-normal coordinates</td>
</tr>
<tr>
<td>(y^+)</td>
<td>dimensionless distance from the wall, (yU_b/\nu)</td>
</tr>
<tr>
<td>(\epsilon)</td>
<td>dissipation rate of (k)</td>
</tr>
<tr>
<td>(\eta)</td>
<td>efficiency index</td>
</tr>
<tr>
<td>(\lambda)</td>
<td>thermal conductivity</td>
</tr>
<tr>
<td>(\mu)</td>
<td>dynamic viscosity</td>
</tr>
<tr>
<td>(\mu_t)</td>
<td>turbulent viscosity</td>
</tr>
<tr>
<td>(\nu)</td>
<td>kinematic viscosity, (\mu/\rho)</td>
</tr>
<tr>
<td>(\rho)</td>
<td>density</td>
</tr>
<tr>
<td>(\tau_w)</td>
<td>wall shear stress</td>
</tr>
<tr>
<td>(\omega)</td>
<td>dissipation rate per unit of kinetic energy, (e/C_mk)</td>
</tr>
</tbody>
</table>

**Subscripts**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
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<tr>
<td>b</td>
<td>bulk</td>
</tr>
<tr>
<td>ref</td>
<td>reference</td>
</tr>
<tr>
<td>t</td>
<td>turbulent</td>
</tr>
<tr>
<td>w</td>
<td>wall</td>
</tr>
</tbody>
</table>

**Acronyms**

- AGR: Advanced Gas-cooled Reactors
- CFD: Computational Fluid Dynamics
- MSRP: Multi-Start Rib Profile
- RANS: Reynolds-Averaged Navier-Stokes
- SRP: Square Rib Profile
- SST: Shear Stress Transport

Additional symbols are defined in the text.

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**Figure 1.** Schematic diagram of a rib-roughened surface.
upon the effects of geometrical factors on flow resistance and heat transfer (Reynolds and Prandtl number effects were also studied). Han et al. [3] later conducted experiments on a rectangular channel with rib-roughened upper and lower walls. The work of Han et al. included an examination of the effect of rib cross-section on the flow thermal-hydraulics: it was established that rib profile had a significant effect on the flow resistance, but only a very limited effect on heat transfer. In the past four decades, there have been several other experimental studies on flow over ribbed (and other roughened) surfaces. A recent review is given by Jiménez [4]. Some of the more recent experimental works on rib-roughened channels were reported by Park et al. [5], Liou et al. [6, 7], Okamoto et al. [8], Taslim and Wadsworth [9] and Rau et al. [10]. In these experiments, the effects of various factors including rib pitch-to-height ratio ($P/k$), rib-to-channel height ratio ($k/H$), rib shape and Reynolds number were tested [7, 11]. In addition, Pirie [12] carried out full size tests using pressurised CO$_2$ at Re $\approx 10^6$ to measure the pressure drop of a fuel element containing multi-start fuel pins (see section 3.2. below).

In the CFD simulations reported here, the focus is mainly on the data of Rau et al. [10] who employed two geometrically (and dynamically) similar square cross-section test sections: a smaller one for heat transfer measurements and a larger version designed to give good resolution of the flow field. Air was the working fluid, and the Reynolds number based on the bulk velocity and equivalent diameter was fixed at Re $= 30,000$. In both cross-sections square ribs could be mounted on the lower surface only (1s, Figure 1), or on both the lower and upper surfaces (2s). In all cases, a large blockage ratio was imposed ($k/H = 0.1$) and the surfaces may be considered to have a high degree of roughness. Rau et al. reported 1s channel results for $P/k = 6, 9$ and 12, while $P/k$ was set to 9 in the 2s section.

Additionally, there also have been numerous attempts to numerically simulate straight and inclined ribs in stationary and rotating passages and U-bends; the most widespread techniques adopted are based on solution of the Reynolds-averaged Navier-Stokes (RANS) equations. In this approach, the Reynolds stresses are computed using a supplementary turbulence model. The choice of turbulence model plays a critical role in determining the accuracy of the simulations. In the present study, two turbulence models of the low-Reynolds-number type are employed. A number of models of this general class have been used by the authors and their colleagues in the computation of smooth surface buoyancy-affected, or mixed convection flows [13–15]. Other numerical investigations of rib-roughened surfaces were carried out by Acharya et al. [16], Liou et al., [11], Iacovides and Raisee [17, 18], Bredberg and Davidson [19], Manceau et al., [20], Ooi et al. [21], amongst others. In addition, Iaccarino et al. [22], Kim and Kim [23], Ryu et al. [24], and Kamali and Binesh [25] have investigated the effects of various geometrical factors on mean flow and heat transfer using RANS, and Leonardi et al. [26] and Nagano et al. [27] using direct numerical simulation (DNS).

The objective of the present work is to study the effects of four different geometrical design parameters which have direct application to the current design of AGR fuel elements. The design parameters tested here include two rib profiles, three rib pitch-to-height ratios ($P/k$), four rib height-to-channel height ratios ($k/H$), and three rib width-to-height ratios ($b/k$) (see Table 1 for details).
2. PHYSICAL AND NUMERICAL FORMULATIONS

2.1. Mean Flow Equations

The mean flow equations for steady and incompressible forced convection flow are written as follows:

Continuity

$$\frac{\partial U_j}{\partial x_j} = 0 \quad (1)$$

Momentum

$$\frac{DU_j}{Dt} = -\frac{1}{\rho} \frac{\partial p}{\partial x_j} + \frac{\partial}{\partial x_j} \left( \nu + \nu_t \frac{\partial U_j}{\partial x_j} \right) \quad (2)$$

Energy

$$\frac{DT}{Dt} = \left[ \left( \frac{\nu}{Pr} + \frac{\nu_t}{\sigma_t} \right) \frac{\partial T}{\partial x_j} \right] \quad (3)$$

where, following standard modeling practice, e.g., Launder and Sharma [28], the turbulent Prandtl number is set to a constant value $\sigma_t = 0.9$.

2.2. Numerical Procedures

The commercial code STAR-CD version 4.02 [29] is used to generate results using the $k$-$\omega$-SST model of Menter [30] (LRN $k$-$\omega$-SST model) and a variant of Durbin’s $v^2$-$f$ model due to Iaccarino [31] ($v^2$-$f$ model). In earlier works by the authors and their colleagues, the turbulence models in STAR-CD have been validated against in-house and industrial codes with the same turbulence models [13, 14].

All fluid properties are assumed to be constant. The momentum and turbulence transport equations are discretized using second-order central differencing and first-order upwind differencing schemes, respectively. The energy equation is discretized using the monotone advection and reconstruction scheme (MARS) [29].

The reader is referred to www.CFDtm.org for results of some of the cross-code validation tests carried out by the authors and their colleagues.
The SIMPLE algorithm is adopted for pressure-velocity correction. Various sensitivity tests have been applied to ensure the numerical reliability of the computations.

### 2.3. Geometry and Grid

In the present work, it is assumed that the flow over the centre-line of the 3-D channel studied by Rau et al. [10] can be represented by a 2-D configuration with relatively good accuracy [32, 33], resulting in significant savings in computation power and time required, i.e., suitable for carrying out parametric studies.

In total, 10 different 2-D grids were used in this study to test the effects of all four design parameters. Some details of the grids used here are listed in Table 2, and the schematic of one of the grids (mesh number 2) is shown in Figure 2. Despite their differences, all the grids have some common features; they all consist of 2-dimensional channels, the lower walls of which are roughened by square ribs of height $k$. The computational domain is of length $2P$, i.e., it includes two ribs. Streamwise periodicity is assumed and cyclic (periodic) boundary conditions are applied at the inlet and outlet planes, which not only reduces the grid size, but also reduces the uncertainty in the results associated with approximate inlet boundary conditions. Since low-Reynolds-number turbulence models are employed, the grids were generated so as to be very fine near the wall (the wall-adjacent cell typically extends only to $y^+ \leq 0.5$).

In the 1s case, the domain is of height $H$ (Figure 3a), whereas for the 2s channel symmetry permits the use of a domain of height $H/2$ (Figure 3b). The thermal boundary conditions at both the lower and upper walls of the 1s case consist of the same uniform wall heat flux. As noted above, the upper boundary of the 2s domain is a symmetry plane.

The hydraulic diameter ($D_e = 4 \times \text{[flow area]} / \text{[wetted perimeter]}$) of the present 2-D channels is 0.1 m. Thus, the rib height to channel hydraulic diameter ratio is $k/D_e = 0.05$, and the blockage ratio ($k/H$) is 10% (except for the case where the effects of changing $k/H$ ratio is studied).

The Reynolds number based on hydraulic diameter is fixed at $Re = 30,000$. The Prandtl number is set to 0.71. There is an unavoidable discrepancy between the

<table>
<thead>
<tr>
<th>Mesh no.</th>
<th>Design parameters</th>
<th>No. of rough walls</th>
<th>$P/k$</th>
<th>$k/H$</th>
<th>$b/k$</th>
<th>Rib profile</th>
<th>No. of cells</th>
</tr>
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<td>1</td>
<td>$P/k$ ratio</td>
<td>1</td>
<td>6</td>
<td>0.1</td>
<td>1</td>
<td>Square</td>
<td>133,000</td>
</tr>
<tr>
<td>2</td>
<td>$P/k$ ratio, Rib shape</td>
<td>1</td>
<td>9</td>
<td>0.1</td>
<td>1</td>
<td>Square</td>
<td>161,000</td>
</tr>
<tr>
<td>3</td>
<td>$P/k$ ratio, $k/H$ ratio, $b/k$ ratio</td>
<td>2</td>
<td>9</td>
<td>0.1</td>
<td>1</td>
<td>Square</td>
<td>111,000</td>
</tr>
<tr>
<td>4</td>
<td>$P/k$ ratio</td>
<td>1</td>
<td>12</td>
<td>0.1</td>
<td>1</td>
<td>Square</td>
<td>189,000</td>
</tr>
<tr>
<td>5</td>
<td>Rib shape</td>
<td>1</td>
<td>9</td>
<td>0.1</td>
<td>1</td>
<td>Multi-start</td>
<td>112,000</td>
</tr>
<tr>
<td>6</td>
<td>$k/H$ ratio</td>
<td>2</td>
<td>9</td>
<td>0.090</td>
<td>1</td>
<td>Square</td>
<td>111,000</td>
</tr>
<tr>
<td>7</td>
<td>$k/H$ ratio</td>
<td>2</td>
<td>9</td>
<td>0.075</td>
<td>1</td>
<td>Square</td>
<td>111,000</td>
</tr>
<tr>
<td>8</td>
<td>$k/H$ ratio</td>
<td>2</td>
<td>9</td>
<td>0.050</td>
<td>1</td>
<td>Square</td>
<td>103,400</td>
</tr>
<tr>
<td>9</td>
<td>$b/k$ ratio</td>
<td>2</td>
<td>9</td>
<td>0.1</td>
<td>1.5</td>
<td>Square</td>
<td>107,000</td>
</tr>
<tr>
<td>10</td>
<td>$b/k$ ratio</td>
<td>2</td>
<td>9</td>
<td>0.1</td>
<td>0.5</td>
<td>Square</td>
<td>113,000</td>
</tr>
</tbody>
</table>
square cross-sectioned experimental channel and the present two-dimensional simulations, namely that the equivalent diameter of the square channel is equal to \( H \), whereas \( D_e = 2H \) in the 2-D case. In order to overcome this difficulty, at least partially, the bulk velocity is set to one-half of the experimental value (and hence the Reynolds numbers based on equivalent diameter are equal). The rib height, \( k \) is maintained at the experimental value and therefore the blockage ratio, \( k/H \), rather than \( k/D_e \), is matched.

Following the findings of Manceau et al. [20] and Iaccarino et al. [22], the ribs are assumed to be insulated in the present simulations. Manceau et al. [20] and Iaccarino et al. [22] tested two different types of thermal boundary condition at the ribs. In the first type, the fluid problem was coupled with the conduction problem in the

![Figure 2. Schematic of mesh number 2 (in Table 2) used for STAR-CD computations (color figure available online).](image)

![Figure 3. Computational domains used in the present work. (a) 1s configuration and (b) 2s configuration (color figure available online).](image)
rib. In the second type, the constant heat flux which was imposed at the lower face of the rib was equally distributed on the other faces. Both boundary conditions produced identical Nusselt number distributions, except on the rib faces and in the vicinity of the lower corners of the rib. Similar observations were also reported by Iaccarino et al. [22].

2.4. Eddy-Viscosity Turbulence Models

Two turbulence models, both of the eddy viscosity type, are examined in the present study: A variant of Durbin’s $v^2$-$f$ model due to Iaccarino [31] and the $k$-$\omega$-SST model of Menter [30]. While both models are originally based on the standard EVM of Launder and Spalding [34], they embody quite different modifications to that scheme. Note that the $k$-$\varepsilon$ model itself has not been tested here since it has shown to return relatively poor results for the present case (and over-predicts the heat transfer levels) [18, 22].

Despite differences in the governing equations, the $v^2$-$f$ and $k$-$\omega$-SST models share common features in that they are both of the low-Reynolds-number type, i.e., the transport equations are integrated over the entire flow domain, there being no need to employ wall functions in wall-adjacent regions. More detailed description of both turbulence models are as follows.

2.4.1. $v^2$-$f$ model. The first model to be considered here is the $v^2$-$f$ model (or simply $v^2$-$f$) which was first proposed by Durbin [35] ($v$ is the wall-normal fluctuating velocity). Damping in the $v^2$-$f$ scheme is related to the diminution of the $v^2/k$ that occurs as a wall is approached.

$$\nu_t = C_m \frac{v^2}{k} T_s$$

(4)

The large-scale turbulence timescale is truncated on the Kolmogorov scale.

$$T_s = \max \left[ \frac{k}{\varepsilon}, C_k T_s \left( \frac{v}{\varepsilon} \right)^{0.5} \right]$$

(5)

The form of the $v^2$-$f$ model implemented in STAR-CD is that given by Iaccarino [31].

In addition to two transport equations for $k$ and $\varepsilon$, an elliptic equation for the redistribution term in the $v^2$ equation, $f_{22}$, is included in this model to account for near-wall and non-local effects.

The governing equations of the $v^2$-$f$ model used by STAR-CD are given in Iaccarino [31] as follows.

$$\frac{Dk}{Dt} = P_k + \frac{\partial}{\partial x_j} \left[ \left( \nu + \frac{\nu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] - \varepsilon$$

(6)

$$\frac{D\varepsilon}{Dt} = \frac{C_{\varepsilon 1}}{T_s} P_k + \frac{\partial}{\partial x_j} \left[ \left( \nu + \frac{\nu_t}{\sigma_k} \right) \frac{\partial \varepsilon}{\partial x_j} \right] - \frac{C_{\varepsilon 2} \varepsilon}{T_s}$$

(7)
\[
\frac{Dv^2}{Dt} = \frac{\partial}{\partial x_j} \left[ (v + \frac{v_t}{\sigma_k}) \frac{\partial v^2}{\partial x_j} \right] + k f_{22} - 6v^2 \frac{\varepsilon}{k} \tag{8}
\]

\[
L^2 \nabla^2 f_{22} - f_{22} = \left( 1 - \frac{C_1}{T_s} \right) \left( \frac{2}{3} - \frac{v^2}{k} \right) - C_2 \frac{P_k}{k} - 5 \frac{v^2}{k T_s} \tag{9}
\]

where \( \nabla^2 \equiv \nabla \cdot \nabla \) is the Laplacian operator and

\[
L = C_L \max \left( k^{3/2}/\varepsilon, C_\eta (\nu^3/\varepsilon)^{1/4} \right) \tag{10}
\]

\[
C_{e1}^c = 1 + 0.045 \sqrt{k/v^2} \tag{11}
\]

The coefficients of the \( v^2-f \) model in STAR-CD are given in Table 3.

2.4.2. \( k-\omega \)-SST model. The \( k-\omega \)-SST model [30] is another EVM to be considered in the present work. In this model, \( \omega (= \varepsilon/C_\mu k) \) represents the large-scale turbulence frequency. The \( \omega \)-transport equation is obtained by formally manipulating the \( k \)- and \( \varepsilon \)-equations, although in so doing a cross-diffusion term arises in the \( \omega \)-equation. That term is omitted in the standard \( k-\varepsilon \) model [36], and therefore there is not a strict equivalence between the \( k-\varepsilon \) and \( k-\omega \) formulations.

Advantages of both the \( k-\varepsilon \) and \( k-\omega \) models are combined in the shear stress transport (SST) model of Menter [30]. In its initial formulation, through a blending function this model effectively uses a LRN formulation of the \( k-\omega \) model in the boundary layer and a version of the \( k-\varepsilon \) model in the free shear layer. This is based on the observations that the \( k-\varepsilon \) model is much less sensitive to the free-stream value of \( \varepsilon \) than the \( k-\omega \) model is to \( \omega \).

Apart from this unique feature, the main differences between the standard \( k-\omega \) model and the SST model are as follows.

- The SST model includes a damped cross-diffusion derivative term, as well as a blending function, in the \( \omega \)-transport equation.
- The definition of the turbulent viscosity in the SST was modified to improve the prediction of the turbulent shear stress.
- The coefficients of the model were modified to improve the overall performance of the model.

Note that the functions and coefficients of the \( k-\omega \)-SST model can be found in many text books and papers including Pope [37] and Menter [30] and therefore are not included here.

<table>
<thead>
<tr>
<th>( C_\mu )</th>
<th>( \sigma_k )</th>
<th>( \sigma_\varepsilon )</th>
<th>( C_{e1} )</th>
<th>( C_{e2} )</th>
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<tr>
<td>0.22</td>
<td>1.0</td>
<td>1.3</td>
<td>1.4</td>
<td>1.9</td>
</tr>
<tr>
<td>( C_1 )</td>
<td>( C_2 )</td>
<td>( C_L )</td>
<td>( C_\eta )</td>
<td>( C_{kT} )</td>
</tr>
<tr>
<td>1.4</td>
<td>0.3</td>
<td>0.23</td>
<td>70.0</td>
<td>6.0</td>
</tr>
</tbody>
</table>
3. RESULTS AND DISCUSSION

3.1. Preliminary Remarks

In the results presented in this section, local Nusselt number is defined as

\[ \text{Nu} = \frac{\dot{q} D_e}{\lambda (T_w - T_b)} \]  \hspace{1cm} (12)

where \( \dot{q} \) is the heat flux, \( D_e \) the channel hydraulic diameter, \( \lambda \) the fluid conductivity, and \( T_w \) and \( T_b \) represent the wall and bulk temperatures, respectively. In the present work, the average Nusselt number, \( \text{Nu}_{av} \), is defined as the Nusselt number averaged over the first near-wall cell within the gap between the ribs.

Following Rau et al. [10], all Nusselt number distributions for the ribbed duct calculations are normalized by the value associated with a smooth passage (the Dittus-Boelter equation).

\[ \text{Nu}_0 = 0.023 \text{ Re}^{0.8} \text{ Pr}^{0.4} \]  \hspace{1cm} (13)

While in a smooth channel the friction coefficient can be directly linked to the shear stress at the wall, this is not true for a ribbed channel. In the present work, the friction coefficient \( c_f \) is defined as

\[ c_f = \frac{\Delta p D_e}{2 \rho U_b^2 L} \]  \hspace{1cm} (14)

where \( \Delta p \) is the pressure drop over the whole domain, \( \rho \) is fluid density, \( U_b \) is the bulk velocity, and \( L \) is the axial length of the domain and \( L = 2P \) (see Figure 1). The average friction coefficient \( c_{fav} \) is obtained from the pressure drop across the whole domain. Thus, the average friction coefficient is the sum of two components: form drag and wall drag. In the present rib-roughened channel simulations, the rib form drag is the dominant contribution to the average friction coefficient \( c_{fav} \).

All friction coefficients are normalized using the value for a smooth tube (the Blasius equation).

\[ c_{f0} = 0.079 \text{ Re}^{-0.25} \]  \hspace{1cm} (15)

It is also useful to define the efficiency index \( \eta \) in order to represent the overall thermalhydraulic performance of ribbed channels.

\[ \eta = \left( \frac{\text{Nu}/\text{Nu}_0}{(c_f/c_{f0})^{1/3}} \right) \]  \hspace{1cm} (16)

Other researchers including Han et al. [38], Taslim and Wadsworth [9], and Kim and Kim [23] have also used this efficiency index to evaluate the overall performance of their test cases. In the present work, the efficiency index is calculated for all design parameters (except the rib profile) to indicate the optimum configuration for each case.
The pressure coefficient is defined here as

\[ C_p = \frac{p - p_{\text{ref}}}{0.5 \rho U_b^2} \]

(17)

where \( p \) is the static pressure on the wall and \( p_{\text{ref}} \) is a reference pressure. Note that \( C_p \) in all cases is offset to the experimental value at \( x/k = 0.5 \) (the trailing edge of the upstream rib, see Figure 1).

The effects of all four geometrical parameters will next be discussed separately. It is worth noting that there are other parameters that affect the choice of a ribbed surface for a nuclear plant. For instance, steel (absorber) content should be minimised by keeping rib material to a minimum. Ribbing also affects the strength of the fuel pin to the internal pressure. In addition, sometimes the criterion is to keep the pressure drop the same as that in an existing design. Also there are cluster parameters that can be optimised such as mixing of coolant between sub-channels. The discussions of these parameters, however, are beyond the scope of the present work and are therefore not discussed any further.

3.2. Effects of Rib Profile

The majority of the experimental and numerical works on rib-roughened channels reported in the literature have been carried out for ribs with a square cross-section (profile). However, in the current design of AGRs, fuel pins have a different rib profile, called the multi-start rib design (Figure 4). In Table 4, real dimensions of the multi-start design are compared with the dimensions of the square rib profile design adopted by Rau et al. [10].

In this section, the effects of changing the rib profile are examined by comparing the square rib profile (SRP) to the multi-start rib profile (MSRP). These two rib profiles, however, have different dimensions and since the results reported by Rau et al. are for square ribs, the rib height of the MSRP is made the same as the SRP (i.e., \( k = 5 \text{ mm} \)) and \( w \) and \( b \) are scaled by the ratio of rib heights. The transformed dimensions of the present SRP and MSRP geometries are listed in Table 5. Mesh numbers 2 and 5 in Table 2 are used here for SRP and MSRP cases, respectively.

![Figure 4. Schematic of the square and multi-start rib profile designs.](image)
In Figure 5a it can be seen that both models suggest that there is little effect on the Nusselt number distribution in-between the two ribs, which is consistent with the results of Han et al. [3] who found that changes to the rib cross-section had only a very limited effect on heat transfer levels. In fact, Han et al. showed that the influence of the rib profile on heat transfer completely disappeared for the range $Re = 10,000–30,000$, where the flow was in a completely rough region. In addition, the experiments of Liou and Hwang [39] on three different rib profiles, namely triangular, semi-circular and square revealed that these profiles have comparable thermal performances, while the square ribs had the highest average friction factor.

Figures 5b and c, respectively, show the normalized streamwise velocity distributions at one-tenth of the rib height and the wall-normal velocity distributions at rib height. Again it is seen that both models return similar results for the SRP and MSRP cases. In Figure 5b, the maximum difference between the two cases occurs in the recirculation regions, where both models return slightly higher streamwise velocity magnitude for the MSRP design.

In Figure 5d, while the $v^2-f$ model returns very similar results for both SRP and MSRP cases, the $k$-o-SST model returns slightly higher $C_p$ for the MSRP case near the upstream face of the rib. It is worth noting that in Figure 5d the data of Rau et al. does not extend all the way to the upstream face of the second rib, but if one assumes a linear extrapolation (since all simulations show a quasi-linear distributions) then

<table>
<thead>
<tr>
<th>Table 4. Real dimensions of square and multi-start rib profile designs (all dimensions are in mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Properties</td>
</tr>
<tr>
<td>Reference</td>
</tr>
<tr>
<td>Rib height ($k$)</td>
</tr>
<tr>
<td>Rib width ($w$)</td>
</tr>
<tr>
<td>Rib base ($b$)</td>
</tr>
<tr>
<td>Radius of the lower corner ($R_1$)</td>
</tr>
<tr>
<td>Radius of the upper corner ($R_2$)</td>
</tr>
<tr>
<td>Pitch ($P$)</td>
</tr>
<tr>
<td>Pitch-to-rib height ratio ($P/k$)</td>
</tr>
</tbody>
</table>

*Rib width at half rib height.
**Exact value depending upon whether pitch is measured normal to the ribs or in the axial direction.
the predicted pressure drop by the $v^2-f$ model would appear closer to the data compared to the $k-\omega$-SST model.

Velocity profiles at two wall-normal planes shown in Figures 5e and 5f, indicate that both models return nearly identical results for both SRP and MSRP cases. The results in Figure 5e are in good agreement with the data as, to a somewhat lesser degree, is the velocity profile in Figure 5f. In the latter figure, both models severely over-predict the size of the separation bubble (and recirculation length).

Figures 6a and 6b show broadly similar streamlines for both rib profiles, although the recirculation length of the SRP case is slightly larger than that of the
In addition, the counter-rotating separation bubbles at the corners are slightly larger for the SRP, clearly due to its sharper corners. Larger separation bubbles are predicted by the \( k-\omega \)-SST model in comparison to the \( v^2-f \) model. The over-prediction of the separation bubble and reattachment length by the \( k-\omega \)-SST model is due to a turbulent viscosity limiter that exists in the \( k-\omega \)-SST model, which acts to limit the shear stress when production exceeds dissipation by an order of magnitude, resulting in larger separation bubbles [30, 40].

### 3.3. Effects of Pitch-to-Rib Height Ratio (\( P/k \))

In this section, the effects of varying the rib pitch-to-height ratio (\( P/k \)) is investigated. The results presented here are for \( 1s \) channels and three \( P/k \) ratios of 6, 9, and 12 (mesh numbers 1, 2, and 4 in Table 2), and comparison is made against the experimental data of Rau et al. [10]. Computations are undertaken using both the \( k-\omega \)-SST and \( v^2-f \) models. For comparison, the helical ribs of AGR fuel pins have a pitch-to-height ratio \( P/k \approx 5.5-6.5 \), the exact value depending upon whether pitch is measured normal to the ribs or in the axial direction [41].

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Figure 6. Streamlines for \( P/k = 9; 1s \) and different rib profiles. (a) Using the \( k-\omega \)-SST model and (b) the \( v^2-f \) model.
3.3.1. **k-ω-SST model.** Figure 7a compares heat transfer levels for various $P/k$ ratios using the $k$-$\omega$-SST model. The results in this figure indicate that the model broadly under-predicts the heat transfer levels, except for $P/k = 6$, where the model returns the most accurate results. In general, in Figure 7a the largest discrepancies between the predictions and the data are near the ribs. This is due to the $k$-$\omega$-SST model predicting relatively large counter-rotating vortices near rib corners (Figure 7b), except for $P/k = 6$, where only one vortex is evident downstream of the first rib. The reattachment length returned by the $k$-$\omega$-SST model is also not in agreement with the data, i.e., another reason for under-predicting the heat transfer levels.

Figure 7c shows the streamwise velocity distributions at $y/k = 0.1$. Large discrepancies are evident especially for $P/k = 9$ where the $k$-$\omega$-SST model indicates that the flow remains reversed at this elevation in the entire inter-rib cavity which is in contrast with the data of Rau et al. For $P/k = 12$, the recirculation length is over-predicted by about 50% compared to the data. This in turn results in an under-prediction of the velocity magnitude in the recovery region. It can be seen that within the primary recirculation region, $U/U_b$ is negative and there are two negative

**Figure 7.** Distributions for $P/k = 6$, 9, and 12 using the $k$-$\omega$-SST model. (a) Nusselt number, (b) velocity streamlines, (c) streamwise velocity at $y/k = 0.1$, and (d) wall-normal velocity at $y/k = 1$ (color figure available online).
peaks within the inter-rib cavity for $P/k = 9$ and 12. For $P/k = 6$ the peaks are more difficult to distinguish.

In Figure 7d it is evident that except for $P/k = 12$, the wall-normal velocity distributions at $y/k = 1$ are in relatively good agreement with the data. In addition, Rau et al. [10] noted that the maximum wall-normal velocity was slightly higher for $P/k = 9$ compared to 6 and 12, while the $k$-$\omega$-SST model indicates that by increasing the $P/k$ ratio, the maximum wall-normal velocity increases too, as shown in the inset of Figure 7d.

3.3.2. $v^{2}$-$f$ model. Figure 8a compares heat transfer levels for various $P/k$ ratios using the $v^{2}$-$f$ model. In general, it is seen that the model tends to overestimate levels of heat transfer. The $v^{2}$-$f$ model returns the least accurate results for $P/k = 6$, where it over-predicts the maximum heat transfer level by approximately 50%. The most accurate results are for $P/k = 9$.

Streamlines for all three $P/k$ ratios are shown in Figure 8b. This figure indicates that there is no reattachment point for $P/k = 6$, a result that is in agreement with the data (Figure 8c). The recirculation lengths for $P/k = 9$ and 12, are nearly the same; however, the model over-predicts the measurements of Rau et al. [10].

Figure 8. Distributions for $P/k = 6$, 9, and 12 using the $v^{2}$-$f$ model. (a) Nusselt number, (b) velocity streamlines, (c) streamwise velocity at $y/k = 0.1$, and (d) wall-normal velocity at $y/k = 1$ (color figure available online).
Figure 8c shows normalized streamwise velocity distributions at $y/k = 0.1$. The $v^2$-f model clearly underestimates the velocity magnitude in the separation region before the downstream rib. Consistent with Figure 8b, there is no reattachment of the flow for $P/k = 6$. From Figure 8c, it is also clear that the magnitude of the streamwise velocity increases for higher $P/k$ ratios as there is a longer space between the reattachment point and the second rib, over which the flow can recover (i.e., higher flow renewal for higher $P/k$ ratios). This, in turn, results in higher impingement force on the front face of the downstream rib.

Results for the wall-normal velocity at rib-height are presented in Figure 8d. The results of the $v^2$-f model are in close agreement with the data except for $P/k = 12$, where the model under-predicts the magnitude of the downward normal velocity in the recovery region. Ooi et al. [21] also found similar results for $P/k = 12$ and argued that this under-prediction of the downward velocity leads to an under-prediction of the impingement strength which in turn results in under-predicting the heat transfer from the floor between the two ribs. This argument, however, is not true in the present results (see heat transfer levels in Figure 8a). In addition, in contrast to the results of Rau et al. (but consistent with the predictions of the $k$-$\omega$-SST model), the $v^2$-f model shows that the maximum wall-normal velocity increases with the $P/k$ ratio (inset of Figure 8d).

Table 6 lists the values of normalized average heat transfer and friction coefficient as well as the efficiency index (see Eq. (16)) obtained by both models for different $P/k$ ratios. It can be seen that from the predictions of the $k$-$\omega$-SST model, $P/k = 9$ is the configuration with the maximum efficiency index, while the $v^2$-f model predicts $P/k = 6$ to have the maximum $\eta$. Note that consistent with the findings of Rau et al., three other earlier experimental works of similar flow problem have found that heat transfer levels are maximized for $P/k = 8.5 – 10$ [6, 8, 9]. (Note, however, that due to shortcomings of the $k$-$\omega$-SST model in the present simulations, the correct prediction of the $P/k$ ratio with the maximum efficiency index should perhaps be assumed to be fortuitous.)

### 3.4. Effects of Rib Height ($k/H$)

All the computations presented so far have been carried out for a constant rib height (for 1s channels: $k/H = 0.1$ or $k/D_c = 0.05$). In this section, the aim is to see the effects of reducing the rib height on the mean flow and heat transfer. This analysis has relevance to one of the current problems that all AGRs are facing due to carbon

<table>
<thead>
<tr>
<th>$P/k$ ratio</th>
<th>Method</th>
<th>$\text{Nu}_{av}/\text{Nu}_0$</th>
<th>$c_{f\text{av}}/c_{f0}$</th>
<th>$\eta = (\text{Nu}<em>{av}/\text{Nu}<em>0)/(c</em>{f\text{av}}/c</em>{f0})^{1/3}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>6</td>
<td>$k$-$\omega$-SST model</td>
<td>1.634</td>
<td>2.931</td>
<td>1.141</td>
</tr>
<tr>
<td>9</td>
<td>$k$-$\omega$-SST model</td>
<td>2.384</td>
<td>4.045</td>
<td>1.496</td>
</tr>
<tr>
<td>12</td>
<td>$k$-$\omega$-SST model</td>
<td>1.905</td>
<td>4.168</td>
<td>1.184</td>
</tr>
<tr>
<td>6</td>
<td>$v^2$-f model</td>
<td>2.426</td>
<td>3.949</td>
<td>1.535</td>
</tr>
<tr>
<td>9</td>
<td>$v^2$-f model</td>
<td>2.248</td>
<td>4.329</td>
<td>1.379</td>
</tr>
<tr>
<td>12</td>
<td>$v^2$-f model</td>
<td>2.446</td>
<td>4.084</td>
<td>1.530</td>
</tr>
</tbody>
</table>
particle deposition on the fuel pins. This deposition results in an overall reduction of the effective rib height. Based on the data provided by British Energy, this rib height reduction in AGRs due to carbon deposition is typically between 1.2–14.6% \[42\].

In addition, the present authors estimate on the basis of design data provided by Fairbairn \[41\] that \(k/D_e\) ratio in a typical AGR fuel channel is an order of magnitude less than that in the experiments of Rau et al. \[10\], where \(k/D_e = 0.1\). Nevertheless, the ratio of \(k\) to \(D_e\) in AGRs is still sufficiently high to raise the possibility that there will be significant restructuring of the logarithmic layer. (Also, the distance between adjacent fuel pins is somewhat less than \(D_e\); consequently, normalization of \(k\) by inter-pin distance will indicate a greater degree of relative roughness.)

In this section, four \(k/H\) ratios are investigated: a) 0.100, b) 0.090, c) 0.075, and d) 0.050 (which, respectively, correspond to 0, 10, 25, and 50% rib height reduction). Computations are undertaken using four different grids with \(P/k = 9; 2s\) (mesh numbers 3, 6, 7, and 8 in Table 2). Only the results of the \(v^2-f\) model are presented here.

In Figure 9a, it is evident that reducing the rib height results in a decrease in heat transfer levels. By reducing the rib height by 10%, the average \(\text{Nu}/\text{Nu}_0\) decreases by 3.8%, while reduction of the rib height by 25% decreases the average \(\text{Nu}/\text{Nu}_0\) by 11.5%. From Figure 9a it can also be seen that the shape of the Nusselt

\[\text{Figure 9.} \text{ Distributions for } P/k = 9; 2s \text{ and different rib heights using the } v^2-f \text{ model. } (a) \text{ Nusselt number, } (b) \text{ streamwise velocity at } y/k = 0.1, (c) \text{ wall-normal velocity at } y/k = 1, \text{ and } (d) \text{ pressure coefficient (color figure available online).}\]
number distribution becomes rather skewed towards the upstream rib by reducing the rib height.

The streamwise velocity at the $y/k = 0.1$ plane is shown in Figure 9b. It can be seen that the smaller the rib height, the shorter the recirculation length becomes. The maximum positive velocity is also higher for shorter ribs, i.e., higher flow renewal within the cavity. It is also seen that the size of the primary recirculation bubble downstream of the first rib is very different for various rib heights, although the size of the second separation vortex upstream of the second rib remains nearly unchanged.

Figure 9c shows the magnitude of the wall-normal velocity at rib height. The figure shows that different rib heights have similar distributions, except for $k/H = 0.05$, which has the smallest downward velocity magnitude within $3 < x/k < 6$.

Figure 9d shows the pressure coefficient distributions. All the distributions are offset to the same value at $x/k = 0.5$. This figure shows the pressure coefficient near the second rib decreases by decreasing the rib height since the size of the recirculation bubble (and hence turbulence) is smaller for shorter ribs (see Figure 9b).

Table 7 lists the values of average $Nu/Nu_0$, $c_f/c_{f0}$, and the efficiency index $\eta$ for each rib height. Consistent with the correlation of Ravigururajan and Bergles [43] and the DNS data of Nagano et al. [27], the $v^2$-$f$ model shows that normalized average heat transfer decreases by reducing the rib height. From the table, it is also seen that the friction coefficient decreases by reducing the rib height which is again consistent with the DNS data of Nagano et al. [27]. However, the $v^2$-$f$ model predicts a reduction of 64% in the friction coefficient when the rib height is reduced by 50%, while for the same rib height reduction, the DNS data of Nagano et al. [27] returned a drop of only 27%. The inconsistencies between the present simulations and those of Nagano et al. [27] could be associated with the differences in the Reynolds number, $P/k$ ratio, and number of roughened walls in the simulated channel. In Table 7, the maximum value of $\eta$ is obtained for $k/H = 0.05$. Similarly, Nagano et al. [27] concluded that ribs with smaller heights have better overall thermalhydraulic performances.

Finally, Figure 10 shows the effects of rib height on normalized heat transfer rate, $Nu/Nu_{k0}$ and comparison has been made against the experimental data of Pirie [12] (Pirie’s data are for $Re = 8 \times 10^5$). $Nu_{k0}$ corresponds to the value of Nusselt number for the original (reference) rib height, $k_0$. ($k_0 = 5$ mm in the present work, while $k_0 = 0.4191$ mm in the data of Pirie [12].) The figure shows that there is good agreement between the present results obtained using the $v^2$-$f$ model and the data.

### 3.5. Effects of Rib Width ($b/k$)

In this section, the effects of rib width on the mean flow and heat transfer are examined. Three different rib width-to-height ratios, $b/k$ are tested: a) 0.5, b) 1.0, c) 2.0.

<table>
<thead>
<tr>
<th>$k/H$ ratio</th>
<th>Turbulence model</th>
<th>$Nu_{av}/Nu_0$</th>
<th>$c_f/av/c_{f0}$</th>
<th>$\eta = (Nu_{av}/Nu_0)/(c_f/av/c_{f0})^{1/3}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.100</td>
<td>$v^2$-$f$ model</td>
<td>2.687</td>
<td>12.350</td>
<td>1.163</td>
</tr>
<tr>
<td>0.090</td>
<td>$v^2$-$f$ model</td>
<td>2.585</td>
<td>10.523</td>
<td>1.179</td>
</tr>
<tr>
<td>0.075</td>
<td>$v^2$-$f$ model</td>
<td>2.379</td>
<td>7.841</td>
<td>1.198</td>
</tr>
<tr>
<td>0.050</td>
<td>$v^2$-$f$ model</td>
<td>1.968</td>
<td>4.396</td>
<td>1.201</td>
</tr>
</tbody>
</table>
Figure 10. Effects of rib height on heat transfer using the $v^2-f$ model compared against the data of Pirie [13] (color figure available online).

Figure 11. Distributions for $P/k = 9; 2s$ and different rib widths using the $v^2-f$ model; (a) Nusselt number, (b) streamwise velocity at $y/k = 0.1$, (c) wall-normal velocity at $y/k = 1$, and (d) pressure coefficient (color figure available online).
and c) 1.5. Three grids with \( P/k = 9; 2s \) (mesh numbers 3, 9, and 10 in Table 2) are used. Only the results of the \( v^2-f \) model are presented here.

Figure 11a compares heat transfer levels and it is seen that the highest rate of heat transfer is predicted for the case with the minimum width, i.e., \( b/k = 0.5 \), mainly because in this case, the inter-rib cavity between the two ribs is wider, which in turn leads to higher rate of flow renewal in the cavity.

Streamwise velocity distributions at \( y/k = 0.1 \) plane are shown in Figure 11b. The results indicate that while similar trends are found for all three \( b/k \) ratios, the case with the minimum rib width has the highest velocity magnitude in both the recirculation and recovery regions. Note that in Figure 11b, decreasing the rib width results in an increase in the intensity of the recirculation region while the size of the recirculation bubble upstream of the rib (represented by the second negative peak) is the same for all three cases.

In Figure 11c it can be seen that by changing the rib width, wall-normal velocity distributions at rib-height are affected only marginally in the regions near the ribs. In fact, these distributions are nearly identical over a wide range (1.5 < \( x/k < 6.5 \)) within the cavity.

As was noted earlier, by decreasing the rib width heat transfer enhances (Figure 11a). This, however, is achieved with the penalty of having higher friction (form drag) as a result of higher pressure difference between upstream and downstream regions of the ribs, as shown in Figure 11d. (As before, the distribution in the figure are offset to the same value at \( x/k = 0.5 \).)

Table 8 lists the values of average \( \text{Nu}/\text{Nu}_0, c_f/c_{f0} \) and \( \eta \) for different \( b/k \) ratios. The table indicates that increasing the rib width results in a decrease in both heat transfer and friction. Wilkie [44] also found a similar decreasing trend for the normalized friction coefficient. The table also shows that the case with the smallest rib width has the maximum value of \( \eta \).

### 4. CONCLUSIONS

Numerical simulations of the flow and heat transfer in 2-dimensional rib-roughened ducts have been performed using a variant of Durbin’s \( v^2-f \) formulation and the Menter \( k-o\)-SST model. The effects of four different geometrical factors including rib shape, \( P/k, k/H \), and \( b/k \) ratios were investigated. Comparison was made against the experimental data of Rau et al. [10]. All simulations were undertaken using a commercial CFD package, STAR-CD. The following conclusions could be derived from the present simulations.

<table>
<thead>
<tr>
<th>( b/k ) ratio</th>
<th>Turbulence model</th>
<th>( \text{Nu}_{av}/\text{Nu}_0 )</th>
<th>( c_{fav}/c_{f0} )</th>
<th>( \eta = (\text{Nu}<em>{av}/\text{Nu}<em>0)/(c</em>{fav}/c</em>{f0})^{1/3} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.5</td>
<td>( v^2-f ) model</td>
<td>2.877</td>
<td>14.116</td>
<td>1.190</td>
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<tr>
<td>1.0</td>
<td>( v^2-f ) model</td>
<td>2.687</td>
<td>12.350</td>
<td>1.163</td>
</tr>
<tr>
<td>1.5</td>
<td>( v^2-f ) model</td>
<td>2.561</td>
<td>11.411</td>
<td>1.137</td>
</tr>
</tbody>
</table>
In comparison with the experimental data, it emerges that both the $v^2$-f and $k$-$\omega$-SST models performed relatively well in capturing the general trend of the data; however, the $v^2$-f model generally returned more accurate results, particularly for heat transfer.

Computations using the $v^2$-f and $k$-$\omega$-SST models indicated that there is little difference between the dynamic and thermal performance of channels with square ribs and those with AGR rib profiles, i.e., multi-start rib profile.

Configurations with rib pitch-to-height ratios ($P/k$) of 6, 9, and 12 (where $k$ was kept constant) were studied using the $v^2$-f and $k$-$\omega$-SST closures. The results of the $v^2$-f model showed that $P/k = 6$ had the highest value of efficiency index $\eta$ while the $k$-$\omega$-SST model indicated that the maximum value of $\eta$ occurs at $P/k = 9$.

The effects of rib height were examined using the $v^2$-f model. The rib height-to-channel height ratios ($k/H$) of 0.1, 0.09, 0.075, and 0.05 were tested. The results showed that the average Nusselt number and friction coefficient decrease with reducing the rib height. The configuration with the shortest rib height ($k/H = 0.05$) was found to have the highest value of efficiency index.

The effects of rib width were examined by testing the rib width-to-height ratios ($b/k$) of 0.5, 1.0, and 1.5. The computations using the $v^2$-f model showed that the average Nusselt number and friction coefficient decrease by increasing the rib width. It was found that the configuration with $b/k = 0.5$ gives the highest value of efficiency index.

REFERENCES


