16èmes Journées Internationales de Thermique (JITH 2013)
Marrakech (Maroc), du 13 au 15 Novembre, 2013

Friction factor and heat transfer optimization in a channel with graded size ribs

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Abstract: It is easy to increase the thermal transfer intensity in the presence of baffles on the walls of the channels, but one of the problems which always decrease the applicability of this technique is the friction factors. This paper presents a novel design of the baffled channel, which is here called decreased/graded baffle-design. The baffles are decreased or graded according the channel length, this geometry characterized by tows ratios: graded baffle ratio (GBR) and decreased baffle ratio (DBR) are varied from 0 to 0.08. The turbulent flow and heat transfer characteristics are numerically investigated by the software Fluent in a Reynolds number range from 10^4 to 2 x 10^4. The numerical results show that the decreased/graded baffle design can significantly reduce friction factors, for the channel at DBR = 0.08 the friction factors decrease from 4-8%, this reduction of friction factors decreased the thermal transfer by 5% at maximum value, consequently.

Keywords:

Friction factors, Degraded baffle, graded baffle, heat transfer. ribs.

1. Introduction

Nowadays, the energy cost is very high; this problem carries out to seek techniques for improvement the thermal systems performances in deferent industrial applications. Within this framework, baffles, ribs or obstacles to take a fundamental role for the improvement of the thermal transfer rate in much of thermal systems like the heat exchangers, the solar collectors, the gas turbines blades and the car radiators and other fluids mechanics applications such as the water desalination.

For the studies in this technique, the researchers base in their experimental or numerical works on deferent geometrical parameters as the shape, the size,

the spacing of baffle, the blockage ratio, the attack angle and space porous. They have characterized the effect of baffles on the thermal transfer rate and the pressure losses for a laminar or turbulent flow by the heat exchange coefficient or the Nusselt number and the friction coefficient. The baffles shape is a necessary parameter for the generation of vortex. For this problem, the several works proposed several shapes, for example, the V baffle shape. [1-6], W. [7], Z. [8] and diamond shape proposed by Somchai Sripattanapipat and al. [9]. All these shapes increase the thermal transfer rate but created catastrophic pressure losses. For the angle of attack, Prashanta Dutta and al. [10] carried out an experimental study on the characteristics of local thermal transfers and the losses of pressure affected in a rectangular channel with inclined, plate and perforated baffles .They noted that the thermal transfer rate depends on the position, the orientation and the geometry of the second baffle. Pongjet Promvonge and al. [11] proposed inclined baffles by 45°, for different values of blockage ratios (BR).They concluded that the slope of baffles by this angle in a square channel with BR=0.05-0 increase the thermal transfer rate from 150 to 850%, and these improvements associated dramatic losses of pressure tends from 2 to 70 times compared to the smooth channel. Pongjet Promvonge. [12] and Pongjet Promvonge and al. [13] led to a combination between the angle of attack, and V baffle shape, they noted that this combination increases the thermal transfer rate and associated losses of pressure for deferent values of blockage ratio. In same subject, Pongjet Promvonge and al. [14] proposed in their experimental investigations of the delta baffles shapes for thermal performance improvement of a solar air heater. The surfaces roughness in the baffled channels is present in deferent experimental and numerical studies. For example, D.N. Ryu and al. [15], R. Kamali and al. [16], and Pongjet Promvonge and al. [17] proposed several shapes for increases the thermal transfer rate at minimum pressure losses. The angle of orientation is present in the literature for the improvement of thermal transfer rate. Ahmet Tandiroglu and al. [18] presented in their experimental study three orientations angles 45°, 90° and 180° with three values of diameter ratio H/D=1,2 and 3, they concluded that the 9035 type presents better performances for the forced convection. In same

subject, Ahmet Tandiroglu presented empirical formulas for calculate the Nusselt numbers and the friction coefficients. [19]. the same author, in another article tested the entropy generation for the deferent baffles types. [20].
The porous baffle was underlined by several researchers because the perforated space to enable us to agitate the flow stagnated behind of baffles. [21, 22] What increase thermal transfer intensity and eliminate the lowers heat transfer areas (LHTA). [23, 24]. All the studies in literature, proposed a baffles designs increase the thermal transfer intensity significantly, but this works does not correct the major problems of this techniques: the formation LHTA and the pressure losses. This article presents a new design of baffles at decreased/graded height according to the channel length; this design can significantly reduce the friction factors in a channel.

2. Flow configuration and mathematical formulation

2.1. Baffle geometry

The system studied in this work is a channel of horizontal plan with a baffle series of decreased/graded height according to the length of the channel, L placed in a staggered array on the upper and lower channel walls as shown in fig. 1.This reduction/gradation defined by the decreased baffle ratio, DBR as shown in the fig.1a and the gradation of baffle ratio, GBR as shown in the fig.1b. For the results comparison, one bases on the simple baffles geometry (without reduction/gradation of baffle) with known dimensions, where e the baffles thickness fixed with 0.02H, b is the baffle height, H fixed to 0.1 m, is the height of channel and b/H is known as the blockage ratio, BR. The distance between the baffles is set to s in which s/H is defined as the spacing ratio, S fixed to 1. These dimensions to allow studies the effect of DBR which varied between 0 to 0.08 and of GBR which varied between 0 to 0.08 where Dhb and Ghb are known.

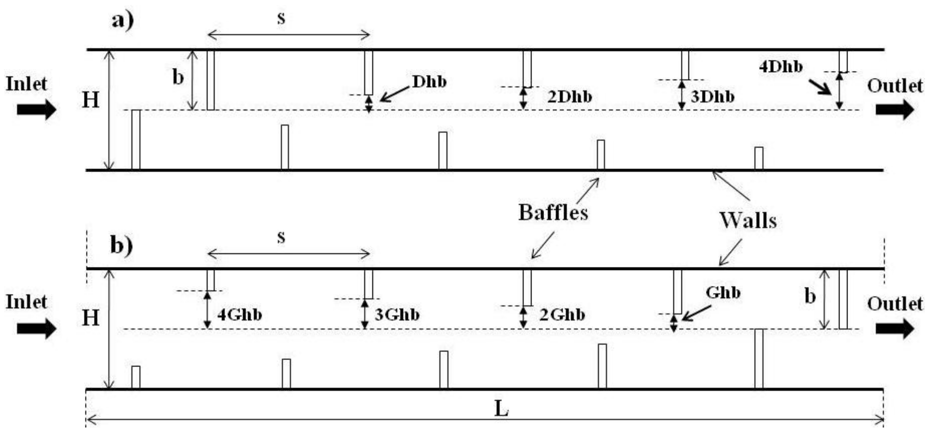


Figure 1. Channel and baffle geometry.

2.2. Mathematical modeling

The numerical model for fluid flow and heat transfer in the tow dimensional channel based on the above assumptions, the duct flow is governed by the Reynolds averaged Navier–Stokes (RANS) equations and the energy equation:

Continuity Equation

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \qquad (1)$$

Momentum equation

$$\frac{\partial}{\partial x_i}(\rho u_i u_j) = \frac{\partial}{\rho x_i}\left[\mu\left(\frac{\partial u_i}{\partial x_j} - \rho\overline{u_i' u_j'}\right)\right] - \frac{\partial P}{\partial x_i} \qquad (2)$$

Where u' is a fluctuating component of velocity.

Energy equation

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\rho x_j}\left((\Gamma + \Gamma_t)\frac{\partial T}{\partial x_j}\right) \qquad (3)$$

Where Γ is the molecular thermal diffusivity, as given by $\Gamma = \mu/Pr$, Γ_t is the turbulent thermal diffusivity, as given by $\Gamma_t = \mu_t/Pr_t$. The Reynolds-averaged approach to turbulence modeling requires that the Reynolds stresses $-\rho\overline{u_i'u_j'}$ in Eq. (2) need to be modeled. The Boussinesq hypothesis relates the Reynolds stresses to the mean velocity gradients as seen in the equation below:

$$-\rho\overline{u_i'u_j'} = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_i}{\partial x_i}\right) - \frac{2}{3}\left(\rho k + \mu_t\frac{\partial u_i}{\partial x_j}\right)\delta_{ij} \qquad (4)$$

Where k is the turbulent kinetic energy, as defined by $k = \frac{1}{2}\overline{u_i'u_i'}$, and δ_{ij} is the Kronecker delta.

The RNG-based k-ε turbulence model is derived from the instantaneous Navier-Stokes equations, using a mathematical technique called "renormalization group" (RNG) methods. The steady state transport equations are expressed as:

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\Gamma + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + G_k - \rho\varepsilon \qquad (5)$$

$$\frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\Gamma + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_{1\varepsilon}\frac{\varepsilon}{k}G_k - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \qquad (6)$$

Where G_k is the rate of generation of the k-ε model $C_{1\varepsilon}$ and $C_{2\varepsilon}$ are constants. μ_t is the turbulent viscosity, as defined by $\mu_t = \rho C_\mu k^2/\varepsilon$, C_μ is a constant and set to 0.0845, derived using the RNG theory.

All the governing equations were discretized by the QUICK numerical scheme, decoupling with the SIMPLE algorithm and solved using a finite volume method. For closure of the equations, the RNG k-ε model was used in the present study. The solutions were converged when the normalized residual values were less than 10^5 for all variables but less than 10^7 only for the energy equation.

The friction factor, *f* is computed by pressure drop, ΔP across the length of the channel, L as

$$f = \frac{2}{(L/D)}\frac{\Delta P}{\rho U^2} \qquad (7)$$

For the thermal study of behavior of baffles one based on the number of local Nusselt, is given by

$$Nu_x = \frac{h_x.D}{k_f} \qquad (8)$$

And the area-average Nusselt number can be obtained by

$$Nu = \frac{1}{L}\int Nu_{(x)}\partial x \qquad (9)$$

The thermal enhancement factor η is defined by

$$\eta = (Nu/Nu_0)/(f/f_0)^{1/3} \qquad (10)$$

Where Nu_0 and f_0 stand for Nusselt number and friction factor for the smooth channel, respectively.

2.3. Boundary conditions

The working fluid in all cases is air. The inlet temperature of air is considered to be uniform at 300 K (Pr = 0.7). The Reynolds number varies from 10000 to 20000 at the inlet. Impermeable boundary and no-slip wall conditions have been implemented over the channel wall as well as the baffle. The constant temperature of the bottom and upper plates is maintained at 330 K while the baffle is assumed at adiabatic wall conditions.

2.4. Grid generation

The different computational domains are resolved by quadrilateral meshes. For this channels flow, however, regular grid was applied throughout the domain. Grid independent solution is obtained by comparing the solution for different grid levels. It is found that the difference in heat transfer coefficient between the results of grid system of about 25.600, 36.100 and 40.200 is less than 2%. Considering both convergent time and solution precision, the grid system of 25.600 was adopted for the computational models.

3 Results and discussion

3.1. Verification of results for smooth channel

The verification of results in this work based on the Nusselt number and friction factor obtained from the present smooth channel are, respectively, compared with Dittus-Boelter and Blasius correlations.

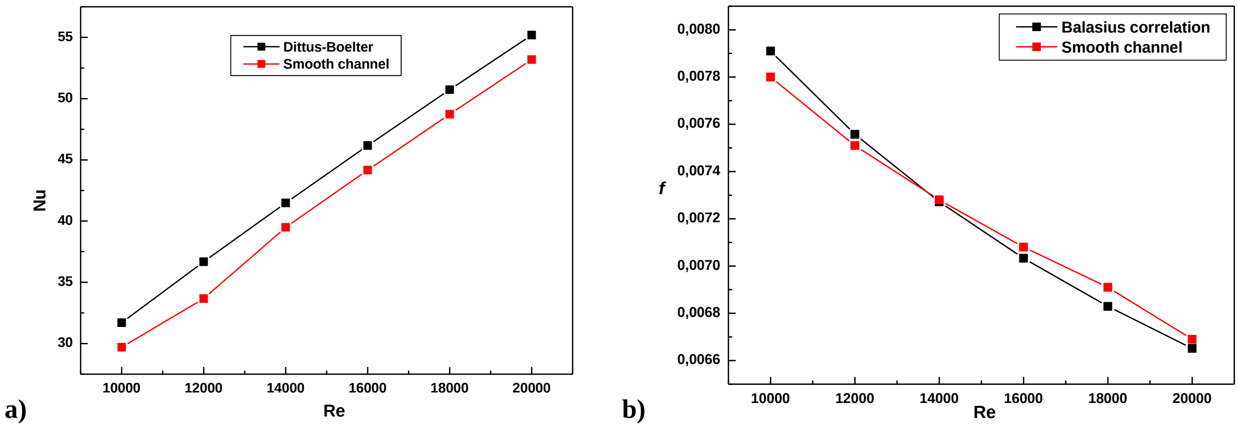


Figure 2. Verification of (a) Nusselt number and (b) friction factor for smooth channel.

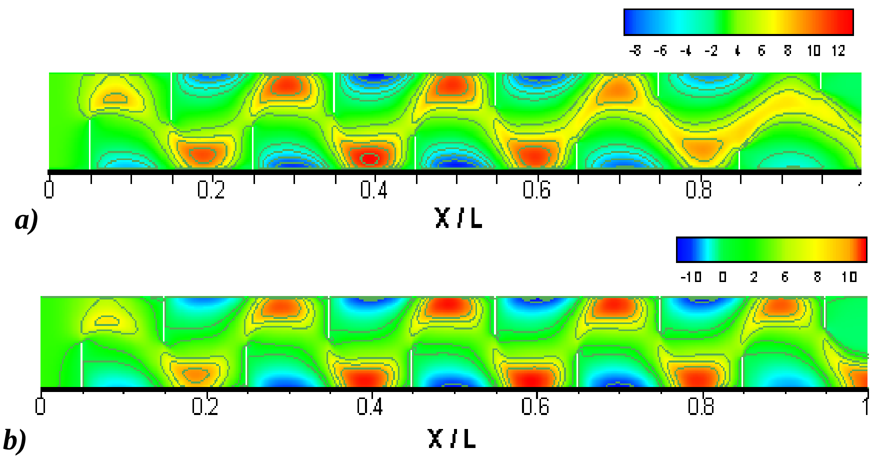
$$Nu = 0.023\, Re_{D_h}^{0.8}\, Pr^{0.4} \quad \text{For Re} \geq 10\,000 \qquad (11)$$

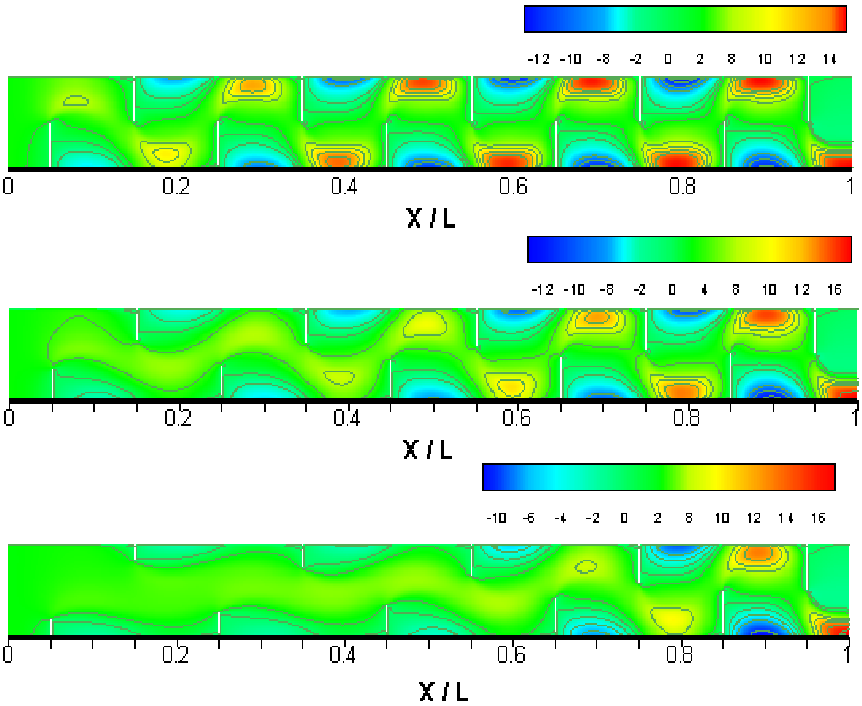
$$f = 0.0791\, Re^{-1/4} \quad \text{For } 4000 < \text{Re} < 10^5 \qquad (12)$$

Fig. 2a and b show, respectively, a comparison of Nusselt number and friction factor obtained from the present study with those from Eqs. (11) and (12). In the figures, the present results reasonably agree well within ±2.9% and ±5.5% for both friction factor correlation of Blasius and Nusselt number correlation of Dittus-Boelter, respectively.

3.2 Flow structure

The Fig. 3 presents the axial velocity contours for DBR = 0.08, DBR = 0.04, DBR/GBR = 0, GBR = 0.04 and GBR = 0.08 at Re = 18000 and S=1, this figure shows the aerodynamics nature of the flow in the presence of baffles with decreased/graded height. In the passing of a baffle, the flow separates in three zones: a zone of flow recirculation behind of baffles where the vortex appears clearly. The second zone located in opposite of the first zone where axial velocity is very high, and the third zone located between the first and the second zone where the flow passes by a mean velocity. These aerodynamic phenomena depend on the baffles size and the reduction/gradation baffles direction. Consequently, this figure shows that the rise of DBR/GBR decrease the formation of the vortex recirculation zones.





c)

d)

e)

Figure 3. X-Velocity contours for different baffled channels,
(a) DBR = 0.08; (b) DBR = 0.04; (c) DBR, GBR = 0; (d) GBR = 0.04; (e) GBR = 0.08

3.3. Pressure losses

Fig. 4 presents the variation of the normalized friction factor, f/f_0 with Reynolds number values for various GBR and DBR. In the figure, the use of decreased/graded height baffle leads to considerable decrease in friction factor in comparison with the simple baffles (GBR, DBR = 0). The DBR values present better increases for friction factor in comparison with GBR values. The DBR = 0.08 present minimum value of friction factor is found to be about 4–8% low the simple baffle channel depending on the Reynolds number values.

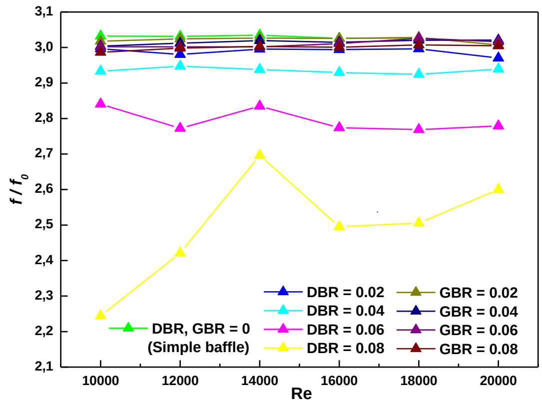


Figure 4. Variation of f / f_0 with Reynolds number for various baffled channels.

3.4. Heat transfer

Fig. 5 present axial variation of Nu_x for the lower channel wall with various decreased/graded height baffles at Re=18 000. The following phenomena can be observed:

- The weak thermal transfer rate observed on the basis of baffle, in the behind of baffles in particular, caused by the formation of hot pockets in these zones.
- The zones which present better thermal transfer rate are the reattachments zones located between successive two baffles.
- The simple baffle presents better rate of thermal Transfer, and baffles it in the case of GBR=0.08 presents the weak thermal Transfer rate.

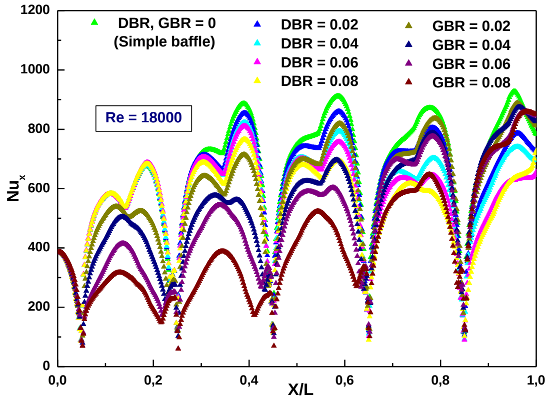


Figure 5. Axial variation of Nu_X along lower channel wall for various baffled channels, Re = 18000.

Fig. 6 presents the variation of the average Nu/Nu_0 ratio with Reynolds number for different decreased/graded height baffles ratios at BR = 0.5, for simple baffle and S = 1.

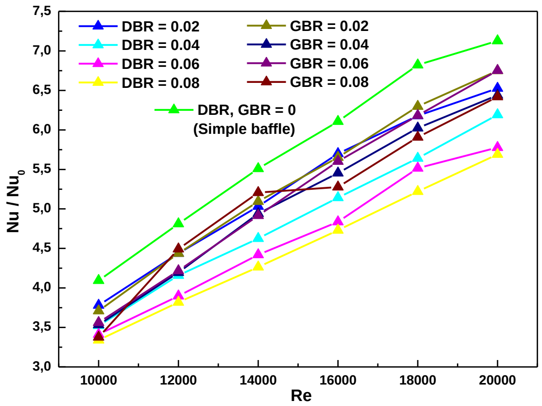


Fig. 6: Variation of Nu / Nu_0 with Reynolds number for various baffled channels.

This figure show that the Nu/Nu_0 value tends to increase with the rise of Reynolds number values for all GBR and DBR values. The use of simple baffle presents better thermal transfer rate, and the baffles it in the case of GBR = 0.02 presents less transfer thermal rate by 5% for the comparison with the simple baffle. The baffles presents minimum thermal transfer rate is DBR = 0.08 tends to 16% for the comparison with the simple.

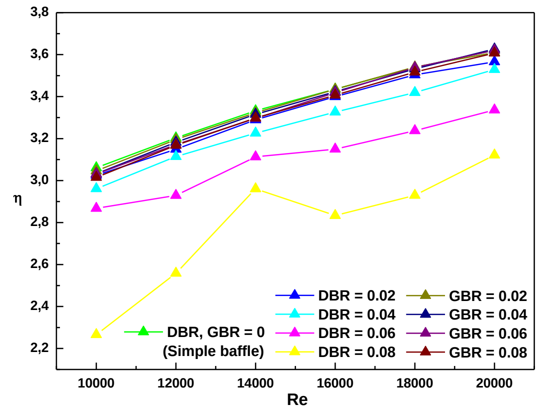


Fig.7: Thermal enhancement factor for various baffled channels.

Fig. 7 shows the variation of the thermal enhancement factor (η) with Reynolds number for different graded/decreased baffle ratios. The enhancement factor tends to increase with the rise of Reynolds number values for all cases, and for S = 1. It is interesting to note that the simple baffle, GRB = 0.02 present the maximum enhancement factors (η) at same value about to 3.6 at the highest value of Reynolds number.

4. Conclusions

Turbulent flow and heat transfer characteristics in a bi-dimensional channel fitted with graded/decreased height baffles, have been investigated numerically. The results of this study shows that the DBR = 0.08 present minimum value of friction factor is found to be about 4–8% low the simple baffled channel depending on the Reynolds number values. However, the friction factor reduction is associated with weak diminution of heat transfer tends to 5% at maximum value compared with the simple baffle.

Nomenclature

b	Simple baffle Height, (m)
BR	Blockage ratio, (=b/D)
D	Hydraulic diameter, (m)
DBR	decreased baffle ratio,(= Dhb/H)
Dhb	decreased height baffle, (m)
f	Friction factor
h	Heat transfer coefficient, (W m^2 K^{-1})
H	Channel Height, (m)
GBR	graded baffle ratio.(= Ghb/H)
Ghb	graded height baffle, (m)
G_K	Turbulent kinetic energy production
k	Turbulent kinetic energy, (m^2 s^{-2})
k_f	Thermal conductivity, W m^{-1} K^{-1}
L	Channel Length, (m)
Nu	Nusselt Number
Re	Reynolds number, $R_e = \rho.u.D_h/\mu$
S	Spacing ratio, (S = s/D)
T	Temperature , ^{0}K
u	Mean velocity at the channel, (m s^{-1})
u_i	Velocity in x_i -direction, (m s^{-1})
u_j	Velocity in y_i -direction, (m s^{-1})

Greek letter

ε	Turbulent energy dissipation
μ	Dynamic viscosity, kg s^{-1} m^{-1}
η	Thermal enhancement factor
ρ	Density, kg m^{-3}

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