Numerical study of turbulent forced convection in a vertical pipe with unsteady flow

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ABSTRACT

A number of low Reynolds number k-ε turbulence models have been used to simulate turbulent forced convection in a pipe with linearly increasing and decreasing flowrates. Under such conditions, the temperature field is influenced by delays in the response of the turbulence field and by the transient nature of the thermal diffusion and convection processes. Results from three typical models are presented here and compared with experimental data. The general features of the response of the temperature field observed in the experiment are reproduced quite well with all the models. However, as far as the detailed development of the temperature field within the near-wall region is concerned, clear differences in performance of the models are evident.

INTRODUCTION

Standard k-ε turbulence models are not directly applicable in the near-wall region of boundary layer flows, where the effects of the damping of turbulence by the wall are important. Semi-empirical wall functions based on knowledge of near-wall shear flow behaviour under simple, well documented conditions, are usually used to bridge this region. Problems are encountered when flow is affected in one way or another by complicating factors. Examples are unsteady flows, buoyancy-influenced flows and separated flows with re-attachment reversal, etc., where simple wall functions do not describe correctly what is happening in the wall region. Low Reynolds number (LRN) k-ε turbulence models have been developed in order to enable the flow in the near-wall region to be directly modelled. Studies utilising this approach have reported with increasing frequency during the last decade or so. Work on the development of such models has been aided by the availability of results from direct numerical simulation studies. A variety of versions of low Reynolds number k-ε turbulence models have been proposed to date. However, few of them have been validated against experimental data for flows involving complicating factors such as those mentioned above and, in particular, for non-steady turbulent flow and heat transfer.

The availability very recently of detailed measurements of wall and fluid temperature in an unsteady turbulent pipe flow from the experimental study of Buyukalaca[1] has

provided an opportunity to test the applicability of LRN k-ε models to such flows and to compare the behaviour of the various models. Very little can be found in the literature on computational studies of forced convection in unsteady turbulent flows, although some studies have been reported for isothermal unsteady flows. An up-to-date review of this topic has been carried out by Ismael[2]. In the present study, we have employed several different LRN k-ε models to simulate the experiments of Buyukalaca. Some results produced by three typical models, namely, Launder and Sharma (LS)[3], Lam and Bremhorst (LB)[4] and Shih and Hsu (SH)[5], are presented and discussed here. The restriction to results from these three models was dictated by the limitations of space. A more comprehensive coverage is given in Jackson and He[6].

THE PROBLEM AND MATHEMATICAL FORMULATION

The experiments simulated here were conducted with water flowing downwards in a uniformly heated vertical pipe of internal diameter 48.34mm, wall thickness 1.29mm and length 3m, which was covered on its outer surface by low conductivity thermal insulation material and preceded by a long unheated flow development section. Care was taken in the design of the experiment to ensure that influences of buoyancy were negligible. During an experiment, the flowrate was increased or decreased linearly with time from one level to another during a prescribed time period with the rate of heat generation in the pipe wall kept constant. The temperature of the wall and the fluid (at chosen radial positions) was measured as a function of time using thermocouples at certain axial locations (2.25m and 2.625m for the wall and fluid respectively) downstream of the start of the heated section. An experiment was normally repeated in an identical manner for 50 times and the mean temperature of the wall and fluid was obtained using the ensemble averaging approach.

In the mathematical formulation, the experimental situation has been approximated using the following assumptions: a) the flow is incompressible and fully developed on entering the heated part of the test section; b) the heat generation within the pipe wall material is uniform in both the axial and circumferential directions (no assumptions concerning wall temperature or heat flux are necessary since the wall is included in the computing domain); c) there is no heat transfer from the outer surface of the pipe; d) no influences of variable properties or buoyancy are present.

Under the above assumptions, the governing equation can be written in cylindrical-polar coordinates as:
Momentum equation:

$$\frac{\partial U}{\partial t}=-\frac{1}{\rho}\frac{\partial P}{\partial x}+\frac{1}{r}\frac{\partial}{\partial r}\left(r(\nu+\nu_t)\frac{\partial U}{\partial r}\right) \tag{1}$$

Energy equation:

$$\frac{\partial T}{\partial t}+\frac{\partial (UT)}{\partial x}=\frac{1}{r}\frac{\partial}{\partial r}\left(r(\frac{\nu}{Pr}+\frac{\nu_t}{\sigma_T})\frac{\partial T}{\partial r}\right) \tag{2}$$

The LRN k-ε models can be specified in a general form as:
Constitutive equation:

$$\nu_t=C_\mu f_\mu \frac{k^2}{\tilde{\varepsilon}} \tag{3}$$

k equation:

$$\frac{\partial k}{\partial t}=\nu_t\left(\frac{\partial U}{\partial r}\right)^2+\frac{1}{r}\frac{\partial}{\partial r}\left[r\left(\nu+\frac{\nu_t}{\sigma_k}\right)\frac{\partial k}{\partial r}\right]-\varepsilon+\Pi \tag{4}$$

ε equation:

$$\frac{\partial \varepsilon}{\partial t}=\frac{1}{r}\frac{\partial}{\partial r}\left[r\left(\nu+\frac{\nu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial r}\right]+C_{\varepsilon 1}f_1\frac{\varepsilon}{k}\nu_t\left(\frac{\partial U}{\partial r}\right)^2-C_{\varepsilon 2}f_2\frac{\varepsilon^2}{k}+E \tag{5}$$

The model constants and damping functions for the three models are tabulated in Tables 1 to 3. The various parameters used in the models are defined as:

$$\tilde{\varepsilon}=\varepsilon-D,\ Re_t=\frac{k^2}{\nu\varepsilon},\ Re_y=\frac{\sqrt{k}\,y^+}{\nu},\ y^+=\frac{u_\tau y}{\nu},\ Re_U=\frac{U^4}{\nu\varepsilon}.$$

Table 1. Model Constants

Model	C_μ	C_1	C_2	σ_k	σ_ε
LS	0.09	1.44	1.92	1.0	1.3
LB	0.09	1.44	1.92	1.0	1.3
SH	0.09	1.5	2.0	1.3	1.3

Table 2. Damping Functions

Model	f_μ	f_1	f_2	f_3
LS	$\exp\left(-3.5/(1+Re_t/50)^2\right)$	1	$1-0.3\exp(-Re_t^2)$	1
LB	$[1-\exp(-0.0165Re_y)]^2\left(1+\frac{20.5}{Re_t}\right)$	$1+\left(\frac{0.05}{f_\mu}\right)^3$	$1-\exp(-Re_t^2)$	1
SH	$1-\exp(-a_1Re_u^{1/4}-a_2Re_u^{1/2}-a_3Re_u)$ *where* a_1=5e-3, a_2=7e-5, a_3=8e-7	1	$\left[1-0.22\exp(-\frac{Re_t^2}{36})\right]\frac{\tilde{\varepsilon}}{\varepsilon}$	1

Table 3. Extra Terms

Model	D	E	Π
LS	$2\nu(\partial\sqrt{k}/\partial y)^2$	$2\nu\nu_t(\partial^2U/\partial y^2)^2$	0
LB	0	0	0
SH	$\varepsilon\exp(-Re_t^{1/2})$	$2\nu\nu_t\left(\frac{\partial^2U}{\partial y^2}\right)^2$	$\frac{1}{r}\frac{\partial}{\partial y}\left(r\frac{0.01}{f_\mu^2}\frac{\nu_t}{\sigma_k}\frac{\partial k}{\partial y}\right)$

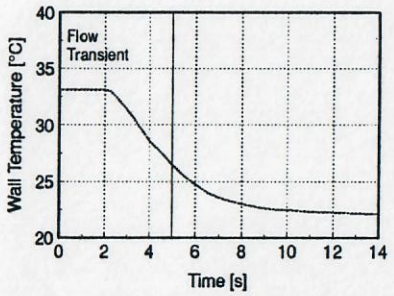


Figure 1 Time variation of outer wall temperature in a 5 s ramp-up experiment

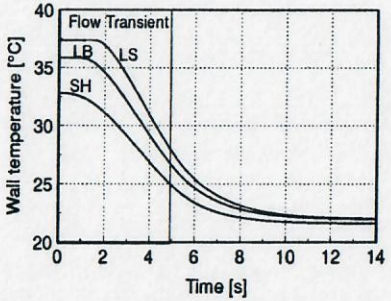


Figure 2 Predictions of time variation of outer wall temperature in a 5 s ramp-up flow using various models

RESULTS

Figure 1 shows the variation of outer wall temperature during a 5 second ramp-up flow variation measured in an experiment. The flowrate range was from 0.278×10^3 to 2.22×10^3 kg/s. The heat generation rate in the pipe wall was 1.33 kW per meter length and the inlet temperature was kept at 19.1°C. It can be seen that the outer wall temperature remained unchanged until about 2 seconds had elapsed, after which it started to fall. It continued to reduce after the flow transient had ended and reached a new steady state after about 14 seconds, i.e., 9 seconds after the end of the imposed flow transient. The predictions of outer wall temperature variation under the same conditions using three turbulence models (LS, LB and SH) are shown in Figure 2. It should be noted that the predictions of the wall temperate using the models differ even under the steady state conditions. At t=0, the start of the transient, the SH model gives a wall temperature of 32.8°C, which is close to the experimental value of 33.1°C. The predictions of the LS and LB models at t=0 are about 5 and 3 degrees higher than the experimental value respectively. However, by comparing Figures 1 and 2, it can be seen that in terms of reproducing the main features of response of the wall temperature to the flow transient (and, in particular, the initial delay), the LS model simulation gives results closest to observed behaviour. This model gives a 2 second initial delay, which is very close to that observed experimentally. The LB model simulation gives an initial delay of just over 1 second and the SH model simulation gives no obvious delay. The variation of wall temperature predicted in the later stages of the transient is similar for each of the models. When the final steady state conditions is eventually reached, the three models again give different values.

Figure 3a shows the variation of local fluid temperature at 5 radial positions measured during a 5 second ramp-up test. It is clear that the pattern of the temperature variation with time at the different radial positions is similar. Fluid temperature decreases gradually after the start of the flow transient until a point (a local minimum) is reached where it starts to increase for a short while and reaches a local peak value, after which it then falls again. The time at which the temperature reaches its local peak value t_p depends on the radial position. The further the position is away from the wall, the longer is the time t_p. Hereafter, we will refer the above phenomenon as a 'perturbation'

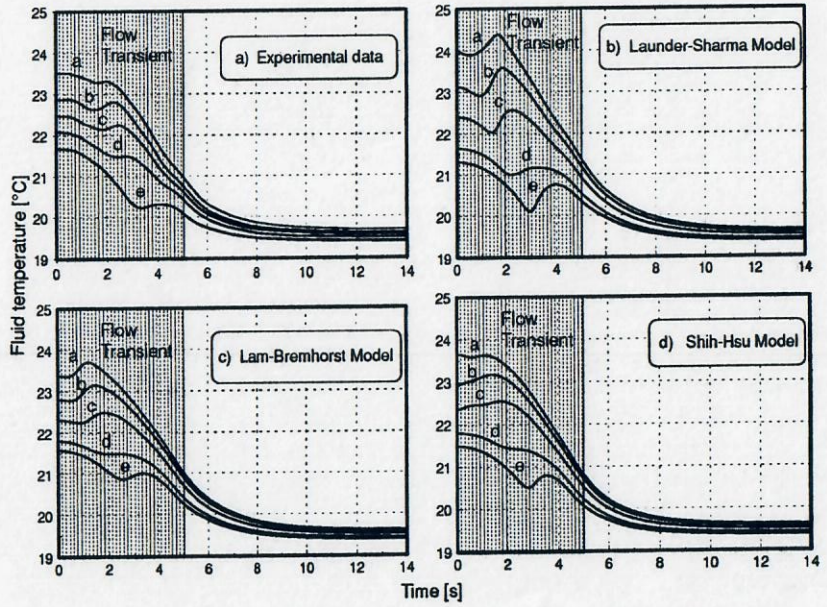


Figure 3 Variations of fluid temperature in a 5 s ramp-up flow - Comparisons between experiment and modelling

and t_p as the 'perturbation time'. The fluid temperature reached its final steady state value approximately 14 seconds after the start of the transient. The predictions of variation of fluid temperature for the same conditions using the three models are shown in Figures 3b to 3d. As can be seen, the predictions again differ under steady state conditions and the LS model gives values which are higher than experiment. It can be seen from the figures that the models all predict the observed general pattern of the fluid temperature variation to some extent. The main features, including the local minimum and peak, and the dependence on radial position of the time at which they occur, are reproduced. However, the detailed behaviour varies significantly from model to model. All the models tend to predict values of t_p smaller than that observed. The LS model gives values of t_p closest to that of the data. As an example, let us look at the predictions of temperature at the pipe centre. In Figure 3a, the measured local peak can be seen at t=4.5 seconds. As can be seen from Figures 3b to 3d, the LS model gives a value of 4.0 seconds whereas the LB and SH models give values of about 3.5 seconds. The predictions of the SH model seem to be generally the poorest. As can be seen from Figure 3d, the temperature at radial positions b and c increased without any initial delay in that case. Detailed discrepancies between the observed behaviours and that exhibited by LS and LB model simulations are clearly evident. On balance, taking all aspects of the comparisons with experiment into consideration, it can be said that the LS model gives the best overall simulation but it is only marginally better than the LB model and the observed behaviour could be said to be somewhat between that of the two simulations.

DISCUSSION

The response of the temperature of the wall and the fluid during and after a flow transient can be associated with a number of mechanisms. These can be distinguished as being turbulence dependent (mainly due to the delayed response of the turbulence field in transient flows - see Jackson and He[7]) and turbulence independent (dependent on the time of the residence of the fluid in the heated pipe in relation to that of the imposed flow transient, the storage or release of heat from the wall and delays in the diffusion of heat across the wall and across the near-wall fluid - see Buyukalaca[1]). Those features of the response related to turbulence independent mechanisms can be reliably simulated in the numerical studies and should be much the same in the predictions of each of the models. However, those features which are related to turbulence dependent mechanisms will be simulated differently by each turbulence model. Consequently, we expect a better prediction of temperature field if the model is able to give a better description of the turbulence during transient flow and a worse prediction if the description of the turbulence is worse. As we have already seen, the predictions of the temperature field using the LS model are marginally the best and the predictions using SH model are the least satisfactory. This is not surprising in view of the results of our earlier study of the modelling of unsteady flows (Jackson and He[6]). In that study, the LS model was found to give a fairly good prediction of the response of the turbulence field to ramp change of flow rate (in particular that in the wall region which is particularly important as far as heat transfer is concerned). The performance of the LB model was not as good in this respect and that of the SH was the poorest.

Now let us look at the response of outer wall temperature in some detail. The delay cannot be attributed solely or even mainly to the transient heat conduction within the wall, since the time scale associated with that is only a small fraction of a second. Rather, it can be associated with the delayed response of the turbulence in the near wall region and the process of molecular diffusion of heat across the viscous sublayer. Predictions of the variation of the temperature at a greater number of radial positions using the LS model are shown in Figure 4. The evolution of the response of temperature across the viscous sublayer and the wall is apparent. Starting from the radial position r=23.26mm, the response extends gradually towards both the wall and the core. Towards the wall, the delay of the response increases and is the greatest at the wall (approximately 2 seconds).

The response of fluid temperature in the core region of the flow can be mainly associated with two factors, namely, the residence time in the heated pipe and the response of the turbulence field. On imposing a ramp-up flow transient, the fluid temperature begins to fall immediately due to the decrease of the residence time. The influence of turbulence is somewhat different. Following the increase of the flowrate, the response of turbulence starts in the wall region and gradually extends to the core. Associated with the increase of turbulence, the temperature increases locally (more heat is diffused to this region from the region of higher fluid temperature near the wall). A perturbation on temperature variation results accordingly and this perturbation propagates away from the wall. In the study of Jackson and He[6], all the three models for which results are presented here were found to be able to simulate the propagation process in isothermal flow simulations quite well. Consistent with this, in the present study, these models have each been found to predict the general pattern of the

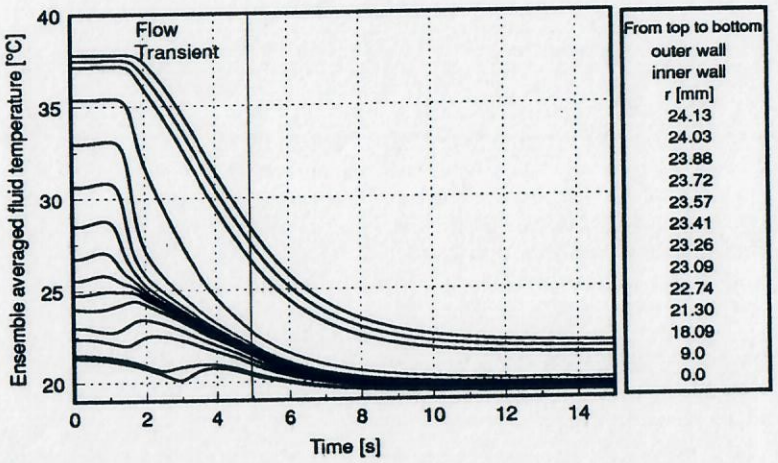


Figure 4 Prediction of time variation of wall and fluid temperature in a 5 second ramp-up flow using the Launder-Sharma model

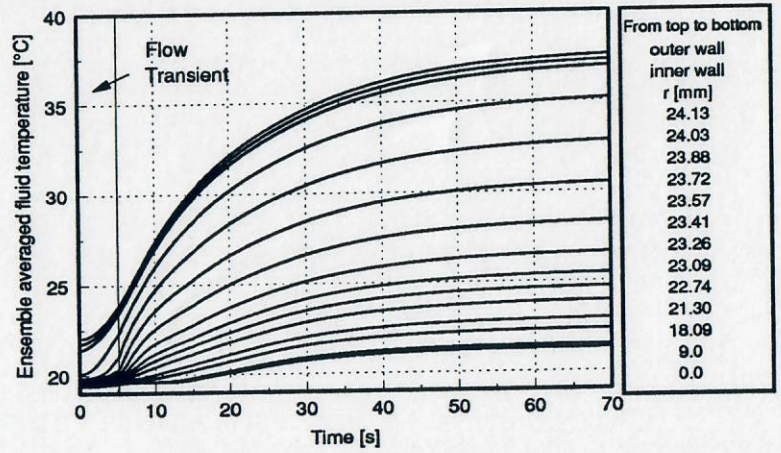


Figure 5 **Predictions of time variation of wall and fluid temperature in a 5 second ramp-down flow using the Launder-Sharma model**

temperature development reasonably well. The fact that the predictions of the perturbation time using different models were different can also be understood. In the experimental study of He[8], a delay in the response of turbulence in the wall region was observed. The actual delay in turbulence response at all other radial positions was found to be due to a combination of this delay and the propagation time mentioned above. In the numerical study of Jackson and He[6], the LS model prediction of the delay in turbulence response was found to be very close to that observed experimentally. The LB model predicted a smaller delay and the SH model essentially gave no delay. This can therefore explain why the LS model predicts longer

perturbation times than the other two models.

Before closing, it is of interest to consider the prediction of the response of the temperature field in a 5 second ramp-down transient using the LS model shown in Figure 5. The test conditions were the same as those of the ramp-up tests apart from that the flowrate was decreased from 2.22×10^3 to 0.278×10^3 kg/s. Two points can be made from the results. Firstly, a small perturbation can be seen in the early stage of the variation of the temperature, which seemed to occur rather later than in the corresponding ramp-up case (about 5 seconds after the flow transient ended). Secondly, the variation of the temperature extends for a very long time. The fluid temperature still varies even 65 seconds after the end of the 5 second flow transient. Both features are consistent with experimental observations.

CONCLUSIONS

The general features of the response of the temperature field in transient turbulent flow are reproduced quite well by the models tested. However, as far as the detailed development of the temperature field within the near-wall region is concerned, the performance of the models does vary from model to model. The Launder-Sharma model is generally found to perform best in simulating transient flow heat transfer experiment considered here. However, it does not reproduce experimentally observed behaviour exactly and it could no doubt be modified to refine its performance.

REFERENCES

1. **Buyukalaca, O.** 'Studies of Convective Heat Transfer to Water in Steady and Unsteady Pipe Flow' Ph.D. Thesis, University of Manchester, 1993
2. **Ismael, J.O.** 'Theoretical Studies of Unsteady Turbulent Pipe Flow' Ph.D. Thesis, University of Manchester, 1993
3. **Launder, B.E. and Sharma, B.I.** 'Application of Energy Dissipation Model of Turbulence to the Calculation of Flow Near a Spinning Disc' Let. Heat Mass Transfer, Vol. 1, pp131 to 138, 1974
4. **Lam, C.K.G. and Bremhorst, K.** 'A Modified Form of the k-ε Model for Predicting Wall Turbulence' ASME Transactions, Journal of Fluids Engineering, Vol. 103, pp 456 to 460, 1981
5. **Shih, T.H. and Hsu, A.T.** 'An Improved k-ε Model for Near-Wall Turbulence' AIAA Paper, AIAA-91-0611, 1991
6. **Jackson, J.D. and He, S.** 'Computational Study of Fluid Flow and Heat Transfer in Transient Turbulent Pipe Flow' Progress Report on PWR-related Research, Nuclear Electric Contract SX/XX-831, 1993a
7. **Jackson, J.D. and He, S.** 'Turbulence Propagation in Transient Turbulent Shear Flows' in *Near Wall Turbulent Flows* (Eds. R.M.C. So, C.G. Speziale and B.E. Launder), pp 939 to 948, Elsevier, 1993b
8. **He, S.** 'On Transient Turbulent Pipe Flow' Ph.D. Thesis, University of Manchester, 1992

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PREFACE

This book contains the edited versions of the papers presented at the Third International Conference on Advanced Computational Methods in Heat Transfer held in Southampton, UK, in 1994. Previous conferences in this series were held in Milan, Italy, 1992 and Portsmouth, UK, 1990. The main objective of the series of conferences is to provide a forum for discussion of advanced topics in heat transfer modelling which include different heat transfer mechanisms and their coupling, nonlinearities, phase change and other complex phenomena. The majority of participants are engineers and scientists actively involved in developing new numerical and computational algorithms, as well as in applying numerical techniques for solving problems of industrial interest.

The conference has attracted contributions from many different countries, covering a multitude of topics which have been broadly divided into the following sections: Heat Conduction; Natural/Forced Convection; Phase Change; Casting, Forming and Welding Processes; Fire and Combustion; Turbulent Heat Transfer; Coupled Thermal/Electrical or Electromagnetic Problems; Heat Transfer in Porous Media; Numerical and Computational Techniques; Industrial and Scientific Applications. All papers have been reproduced from material submitted by the authors, and their content is a reflection of their research work.

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