A CONTIRIBUTION TO THE STUDY OF FLOW FIELD AND HEAT TRANSFER ON A REARWARD - FACING STEP

دراسة مشتركة لسريان المائع والحرارة على الحافة الخلفية لوجه سلمة

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الخلاصة

خصائص السريان للمائع وانتقال الحرارة للحافة الخلفية لسلمة قد بحثت هنا نظرياً وتجريبياً، كما تمت دراسة تأثير ارتفاع السلمة على شكل ومعدل انتقال الحرارة للحافة الخليفة للسلمة مع قيم مختلفة لرقم رينولدز. نقطة اعادة التصادم بين المائع والسطح حدث ووجد أن أقصى معامل انتقال الحرارة يقع عندها، حجم منطقة السريان ومعامل انتقال الحرارة يعتمدان بدرجة كبيرة على رقم رينولدز. كما لوحظ أن معامل انتقال الحرارة ينخفض بسرعة بعيداً عن الحافة الخلفية للسلمة ويقترب شكل السريان ليصبح كما لو كان سريان ذو طبقة حدية.

ABSTRACT

The flow and convective heat transfer characteristics over a backward-facing step were investigated numerically and experimentally. Two different step-heights were studied for different Reynolds numbers. The point of reattachment of the flow behind the step proved to be the point of maximum heat transfer coefficient. The size of tle separated region as well as the Nusselt number depended greatly on the Reynolds number. Downstream of reattachment, the heat transfer coefficient declined rapidly and followed gradually the contour of the turbulent boundary layer. The results of the investigation were compared to others from the literature with satisfaction.

NOMNCLATURE

A	Van Driest's constant (= 2.06 for smooth wall)
C_1......C4	are the coefficients in approximated turbulence transport equations.
CD	Empirical constant (= 0.22 for the k -ε model)
cp	Constant pressure specific heat of air
Cp	Pressure coefficient
E	Function of wall roughness (= 9.0 for smooth wall)
H	Height of step.
h	Local heat transfer coefficient
K	Turbulent kinetic energy (U_i U_i/2)
k	Thermal conductivity
l	Turbulence mixing length
Pr, Pr_t	Laminar and turbulent Prandtl number
P , Pω	Static pressure, atmospheric free stream pressure.
q"	Heat flux

T	Temperature
t	Thickness of wall
Re	Reynolds number based on height of step (U.H/ý)
Nu	Nusselt number based on height of step (h.H/k)
U,V	Mean components of velocity in x-direction and y-direction.
U_∞	Free stream velocity
u', v'	Fluctuating velocity components in x-and y-directions
x,y	Cartesian coordinates parallel to the wall and normal to it.
ε	Rate of dissipation of turbulence energy
λ	Von Karman's constant
μ, μt, μe	Molecular, turbulent and effective viscosity
v	Kinematic viscosity
ρ	Density
ϕ	A generalized dependent variable
υk	Prandtl number for kinetic energy
$\upsilon\varepsilon$	Prandtl Number for dissipation.
ψ	Stream function

Subscript

f	denotes fluid (air)
i , j	denote cartesian coordinates directions
n	general subscript
p	value adjacent to the wall (second node in y-direction)
w	wall value
∞	free stream value

INTRODUCTION

Separated flows are observed in many engineering applications such as airfoils, diffusers, turbomachines and civil engineering structures. Such flow separations produce significant performance losses, and have been of great interest to engineers and researchers.

The problem of separated flows, besides being of great importanoe due to this wide occurrence, is basic problem in turbulent flow and heat transfer. Its comprehensive understanding is essential for the turbulent behaviour of boundary layers in general. The physics of the separation phenomenon can be simply described as resulting in stagnant trapped vortices in the flow, which could enhance or inhibit the convective heat transfer process. The rate of heat transfer depends also on the size of the vortex, which in turn is controlled by the geometry and flow characteristics.

Turbulent heat transfer for rearward-facing steps has been treated with great interest in the literature recently. Gooray et. al. [1] studied the problem of heat transfer for rearward-facing steps and sudden pipe expansions. Their study consisted of computed distributions of heat transfer coefficients for several Reynolds numbers, particularly for low ranges. Celenligil et. al. [2] gave a numerical solution for a two-dimensional turbulent separated flow using a Reynolds stress closure model. Moss et. al. [3] published some experimental results on mean velocity and Reynolds stresses in regions of recirculation flows. Amano and Goel [4] applied a Reynolds stress turbulence closure model for studying the flow in a pipe with sudden expansion. Their results showed an advantage of the Reynolds stress model over the k-ε model by high Reynolds numbers. The author [5] investigated experimentally the flow characteristics and heat transfer coefficients near separation on a flat plate boundary layer in the action of severe adverse pressure gradients and also its influence on the laminar sublayer [6]. Kim et. al. [7] and [8] studied the mechanism of a reattaching turbulent shear layer over a backward-facing step. Moore [9] published some experimental results on the reattachment of a laminar boundary layer separating from a rearward-facing step. Baughn et. al. [10] used a new technique to measure the heat transfer coefficient downstream of an abrupt expansion in a circular channel with a constant heat flux.

In the present study three significant flow configurations are considered: boundary layer over a flat plate, separation and reattachment behind a backward-facing step and a free shear layer with streamline curvature. The study includes experimental investigation and numerical calculations for flow characteristics and heat transfer. The flow field is considered two-dimensional, turbulent and incompressible.

EXPERIMENTAL STUDY

The experiments were conducted in a low speed low turbulence open type wind tunnel for the purpose of measuring heat transfer coefficents and pressure gradients. The test section is a square of side length 460 mm. The flat plate is 1.2 m long and 20 mm thick and made from plexiglass. has a 10 mm step down, backward-facing the flow, at a distance of 800 mm from the leading edge. Two side walls along the test section help to keep the flow two-dimensional. The leading edge of the flat plate is flushed very smoothly with the converging exit part of the wind tunnel to keep any turbulent disturbances to a minimum. To eliminate the side walls blockage effect and keep a good portion of undisturbed free stream around the centerline, the corners of the entrance portion of the test section were fitted with fillets of 135^{o} on both bottom sides. Figure 1 shows the test section with some details.

The lower side of the plate is heated with a 2.0 kW blower type heater. The hot air hits a row of vanes and circulates in a chamber beneath the test section and then escapes through side holes. A thermo-regulator keeps the temperature in the chamber constant to an average of (55 ± 0.5) ^{o}C.

Along the centerline of the test section, 1 mm diameter holes serve for static pressure measurements. The temperature is measured by 0.5 mm diameter copper-constantan thermo-couples which are embedded in the surface of the plate. The temperature of the hot lower side.of the plate is averaged from the readings of three embedded thermo-couples. The temperatures were measured and recorded with a 30-channel digital recorder, the pressure measurements were taken by a micro-manometer.

To study the influence of the Reynolds number on the flow field and heat transfer characteristics, two different runs were conducted. The Reynolds number had a range of

$$Re = \frac{U_\infty . H}{v} = 1.9 \times 10^4 \ldots\ldots\ldots 14.57 \times 10^4$$

The maximum temperature of the lower side was kept constant (about 55°C) for all runs. The room temperature was between 20°and 24°C.

STATEMENT OF UNCERTAINTY

The uncertainty of the present data is mainly in the measurement of the pressure and temperaturs. For the temperature measurements the digital reorder had a sensitivity of 1/15.5°C. The temperature range was about 55°C. The thermo- regulator controlled the heating to exatly ± 0.5°C. This gives an estimation of the error as a maximum of 1.8%.Both the thermo-regulator and the recorder were accurately maintained every six months.

The pressure was measured with a micro-manometer which was developed at the Technical University of Berlin. It has a sensitivity of 0.008 mm water head, and could be handled easily and very accurately. This value is smaller than the minimumof the static pressure difference measured along the surface of the plate by a factor of 10.

Due to the separation and recirculation of the flow the readings were subjected to strong turbulent fluctuations. But the repeated experiments and the reproduction of the points and curves confirmed the consistency of the data. The geometric positions are accurate to within ± 0.10 mm.

ANALYTICAL STUDY

Analytical studies of turbulent flows are usually of complicated and not definite nature. The problem becomes even more complicated and analysis of the governing equations more cumbersome, with flow separations. In the present work a computational method was employed to investigate the heat transfer coefficient for a flow over a backstep. The flow is considered turbulent, incompressible, recirculating, two-dimensional and steady. The governing equations for heat transfer and fluid flow include those for stream funotion ψ, vorticity ω, turbulent kinetic energy K, turbulent dissipation rate ω and temperature T, which are expressed in equations (l) to (5) respectively:

$$- \frac{\delta}{\delta x}\left(\frac{\delta \psi}{\delta y}\right) - \frac{\delta}{\delta y}\left(\frac{\delta \psi}{\delta y}\right) - \rho\omega = 0 \qquad (1)$$

$$\frac{\delta}{\delta x}\left(\omega \frac{\delta \psi}{\delta y}\right) - \frac{\delta}{\delta y}\left(\omega \frac{\delta \psi}{\delta x}\right) - \frac{\delta}{\delta x}\left(\mu e \frac{\delta \varpi}{\delta x}\right) - \frac{\delta}{\delta y}\left(\mu e \frac{\delta \varpi}{\delta x}\right) = 0 \qquad (2)$$

$$\frac{\delta}{\delta x}\left(K \frac{\delta \psi}{\delta y}\right) - \frac{\delta}{\delta y}\left(K \frac{\delta \psi}{\delta x}\right) - \frac{\delta}{\delta x}\left[\left(\mu + \frac{\mu t}{\theta k}\right)\frac{\delta k}{\theta x} - \frac{\delta}{\delta y}\left[\left(\mu + \frac{\mu t}{\upsilon k}\right)\frac{\delta k}{\delta y}\right]\right. \qquad (3)$$

$$- [C_3 \mu_t G - C4 \rho\varepsilon] = 0$$

$$\frac{\delta}{\delta x}\left(\varepsilon\frac{\delta\psi}{\delta y}\right)-\frac{\delta}{\delta y}\left[\varepsilon\frac{\delta\psi}{\delta x}-\frac{\delta}{\delta x}\left[\left(\mu+\frac{\mu t}{\upsilon\varepsilon}\right)\frac{\delta\varepsilon}{\delta x}-\frac{\delta}{\delta y}\left[\left(\mu+\frac{\mu t}{\upsilon\varepsilon}\right)\frac{\delta\varepsilon}{\delta y}\right.\right.\right.$$ (4

$$-\left[\frac{C_1\varepsilon\,\mu t\,G}{K}-\frac{C_2\rho\varepsilon\,2}{K}\right]=0$$

$$\frac{\delta}{\delta x}\left(T\frac{\delta\psi}{\delta y}\right)-\frac{\delta}{\delta y}\left(T\frac{\delta\psi}{\delta x}\right)-\frac{\delta}{\delta x}\left[\left(\frac{\mu}{Pr}+\frac{\mu_t}{Pr_t}\right)\frac{\delta T}{\delta x}\right]-\frac{\delta}{\delta y}\left[\left(\frac{\mu}{Pr}+\frac{\mu_t}{Pr_t}\right)\frac{\delta T}{\delta y}\right]=0$$ (5

where

$$G=2\left[\left(\frac{\delta u}{\delta x}\right)^2+\left(\frac{\delta v}{\delta y}\right)^2\right]+\left(\frac{\delta u}{\delta y}+\frac{\delta v}{\delta x}\right)^2$$ (6

The velocity components u and v are related to the stream function by

$$\rho u=\frac{\delta\psi}{\delta y}$$ (7

$$\rho v=\frac{\delta\psi}{\delta x}$$ (8

The turbulence viscosity is related to the turbulence energy and dissipation rate by the following equation:

$$\mu_t=\frac{C_\mu\ \rho K^2}{\varepsilon}$$ (9

where $C\mu$ is an empirical constant.

The simultaneous solutions of equations (1 - 5) give prediction of momentum transport in the general turbulent flow problem. These equation possess much similarity in form, a fact which is helpful in programming the solution procedure. The general form of the governing equation is

$$a\left[\frac{\delta}{\delta x}\left(\phi\frac{\delta\psi}{\delta y}\right)-\frac{\delta}{\delta y}\left(\phi\frac{\delta\psi}{\delta x}\right)\right]-\frac{\delta}{\delta x}\left[b\frac{\delta(c\phi)}{\delta x}\right]-\frac{\delta}{\delta y}\left[b\frac{\delta(c\phi)}{\delta y}\right]+d=0$$ (10

The functions a, b, c, and d are given in the following table:

ϕ	a	b	c	d (Source Term)
ω	1	$(\mu + \mu_t)$	1	0
ψ	0	1	1	$-\rho\,\omega$
T	1	$(\frac{\mu}{P_r} + \frac{\mu_t}{P_{rt}})$	1	0
K	1	$(\mu + \frac{\mu_t}{\theta k})$	1	$-[C_3\,\mu_t\,G - C_4\,\rho\varepsilon$
ε	1	$(\mu + \frac{\mu_t}{\theta\varepsilon})$	1	$-[\frac{C_3\,\mu_t\,G}{K} + \frac{C_2\,\rho\varepsilon^2}{K}]$

$$\frac{(T_p - T_w)\,Cp\,\rho\,C_\mu^{1/4}\,Kp^{1/2}}{h\,(T_w - T_\alpha)} = \frac{P_{rt}}{\lambda}\ln\left[\frac{E\,y_p\,C\mu^{1/4}\,Kp^{1/2}}{Y}\right] + Pr_t\,\frac{\pi/4}{\sin(\pi/4)}\left(\frac{4}{\lambda}\right)^{1/2} \quad (12)$$

$$*\left(\frac{Pr}{Pr_t} - 1\right)\left(\frac{Pr}{Pr_t}\right)^{1/4}$$

This wall function is taken from Launder and Spalding [12]. Here C_P is the constant pressure specific heat, λ is the Von karmen constant, A is Van Driest's constant and E is a function of the wall roughness. In the present work for the K-ε model and a smooth wall $\lambda = 0.433$, A = 26.0 and E = 9.0.

With the values obtained for C_1 and C_2 and with the above values from the wall function the empirical constant for the turbulent viscosity is $C\mu = 0.009$. Verifying the values of C_n in the empirical formula given by [12]

$$\frac{n2}{\delta\varepsilon}\;\frac{C_1\,C\mu^{1/2}}{\lambda^2} + \frac{C_1\,C\mu^{1/2}}{\lambda^2} = 0 \quad (13)$$

is very satisfactory.

Equation (12) is the most widely used wall function, and can be applied for many practical purposes. One of its merits is to economize the computer work, and the other merit is to reproduce identically the logarithmic law of the wall. A two layer and a three-layer wall function models were suggested by Amano [14]. The computations in these models are based on detailed treatment of the equation near the wall, and the local variations of turbulent quantities were evaluated through the viscous sublayer and the overlap layer regions respectively. These models seem to be advantageous for flows with higher Reynolds numbers.

The background of the computer program employed in this work is described at some length in reference [11]. In this problem a variable grid system with 21 lines in the x- and y- directrons is used. The non-uniform nodal distance was increasing in size away from the wall and step corners. Figure 2 shows schematically the two dimensional backstep geometry. The employment of non-uniform nodal distance was very advantageous, since the density of the mesh increases near the wall and results in significant saving in computing time. Extreme care was given to the program to make sure that the heat transfer coefficients were independent of the grid. The boundary conditions for the problem are illustrated in Fig. 3.

The program was run on a UNIVAC 1100 computer. It took about 12 minutes to convergence, which was achieved after 500 iterations to attain maximum flux error of 0.005. The programs were then repeated increasing the number of iterations to 1000 and thus attaining a mass flux error of less than 0.001.

RESULTS

While the investigation was primarily concerned with the computation of the flow field and heat transfer, preliminary measurements of surface temperature and static pressure behind the step were taken to help the prediction model, and precisely choose the constants. It is important to emphasize that the results presented here include those obtained by the computational model as well as by experimental techniques. It is also worth mentioning that extensive care was given to the design of the eperimental apparatus with the aim of having a relatively wide range of undisturbed two-dimensional flow in the test section and to make sure that the separation occured only at the beginning of the step. Likewise the computational program was repeated many times and compared to experimental values, to ensure best choice of constants and accurate results.

The pressure coefficient was calculated according to

$$Cp = \frac{P - P_{\infty}}{1/2\ P\ U^2_{\infty}} \tag{14}$$

The experimental results are shown in Fig. 4 together with other results from the literature [3] and [15]. The influence of the Reynolds number is very obvious. With increasing Re the reattachment point ($C_p = O$) is pushed further downstream. This agrees completely with the other results. and they all show the same trend. Figures 5 and 6 show patterns of the streamlines and isothermal lines for two different step heights and different Reynolds numbers. At larger Reynolds numbers the point of reattachment is pushed further downstream, and the region of separation becomes wider, which is in full agreement with the previous statement. Behind the reattachment point the flow returns very slowly to the structure of an ordinary turbulent boundary layer.

The range of the Reynolds number applied in these experiments is for a turbulent flow. This was verified by examining the plot of the velocity profiles around the region of the step. This fact indicates that the case studied is most likely turbulent, where the flow is turbulent at both separation and reattachment.

Figures 7 and 8 show the velocity profiles at various stations along the test section for two different step heights. The region of recirculating flow is clearly observed by negative values of U behind the step, also the reattachment point and the pattern of the boundary layer after reattachment. In the separation region the velocity profiles possess two distinct zero points.

The heat transfer coefficient was calculated through equation (11), while the measurement of the temperatures yielded where k is the thermal conductivity of plexiglass (0.19 W/m^{o}K) and

$$h = \frac{K\,(T_i - Tw)}{t\,(T_W - T_\infty)} \tag{15}$$

T_i denotes the temperature of the lower heated surface of the wall. The Nusselt number was then determined with the step height as the reference length. The results were plotted in Fig. 9 and one may observe a fair agreement between computational and expermental results. Other results from the literature [l] are also included. The influence of Reynolds number, also calculated with the step height as reference length, is very clear on the values of the Nusselt number. The maximum heat transfer occurs at the point of reattachment, and the maximum values are related to the Reynolds number.

At higher values of Re the Nusselt number is als higher and the reattachment point slides further downstream. This behavior of the heat transfer gives two suggestions which seem to be unclear in the literature. The first is that the maximum value of the Nusselt number is pushed downstream at higher Reynolds numbers, and this agrees with [1], [10] and [15]. The second is that the peak of the Nusselt number is raised to higher values at higher Reynolds numbers; and this point is in agreement values at higher Reynolds numbers, and this point is in agreement with results from [l] and [15], but disagrees with those from [10].

The turbulent kinetie energy K was calculated according to equation (3) and normalized with the free stream velooity to K/U^2_α . The variation of K over the test section is shown in Fig. 10 and 11 for two step heights. By comparing the results from figures 10, 11 and those of 7, 8 we conclude that the position of the maximum values of turbulent kinetic energy K/U^2_α coincides with that of the zero velocity values. This fact can be understood by physical aonsideration of the flow in the recirculating region. The zero velocity line separates two distinct, but rather fluctuating, counter currents. It is on this line that the maximum shear stresses are acting, and the maximum kinetic energy available.

CONCLUSION

In this work the separated flow behind a backward-facing step was investigated. This subect, besides being of great interest to thermal-fluid sciences, due to its many engineering applications, is a basic problem in heat transfer. The investigation was done theoretically by applying the K-ε computational model. Experiments were also conducted in the wind tunnel to measure heat transfer coefficients and pressure gradients. The results were compared to other existing results in the literature.

One of the significant results of the present investigation is the influence of the Reynolds number on the size of the separation region behind the step, and the position of the reattachment point of the flow. By inoreasing the Reynolds number the reattachment point is pushed downstream thus increasing the size of the separated region. This result is proved experimentally and numerically and is physcially acceptable.

The local heat transfer coefficient increases also by increasing Reynolds number, and reaches a peak value at the reattachment point, and then declines further downstream. The locus of the maxima of the heat transfer curves depends on the Reynolds number, a point which seems to be unclear in the literature. The isothermal lines are shifted upwards in the region of separation, while behind the reattachment point they follow the contour lines of the ordinary boundary layer.

By physical analysis of the flow field and detailed study of the computational work alongside with the experimental results, new values of the flow characteristics constants C_1 - C_4 were used. These values made a good matching between theory and experiment. It is suggested that other separated turbulent fluid flow problems may be handled in a similar manner. Nevertheless more experiments are needed in this and similar area.

Generally speaking the computational technique applied in this work was very satisfactory for the backward-facing step. However, due to the inherent limitations of the K-ε model further research is needed.

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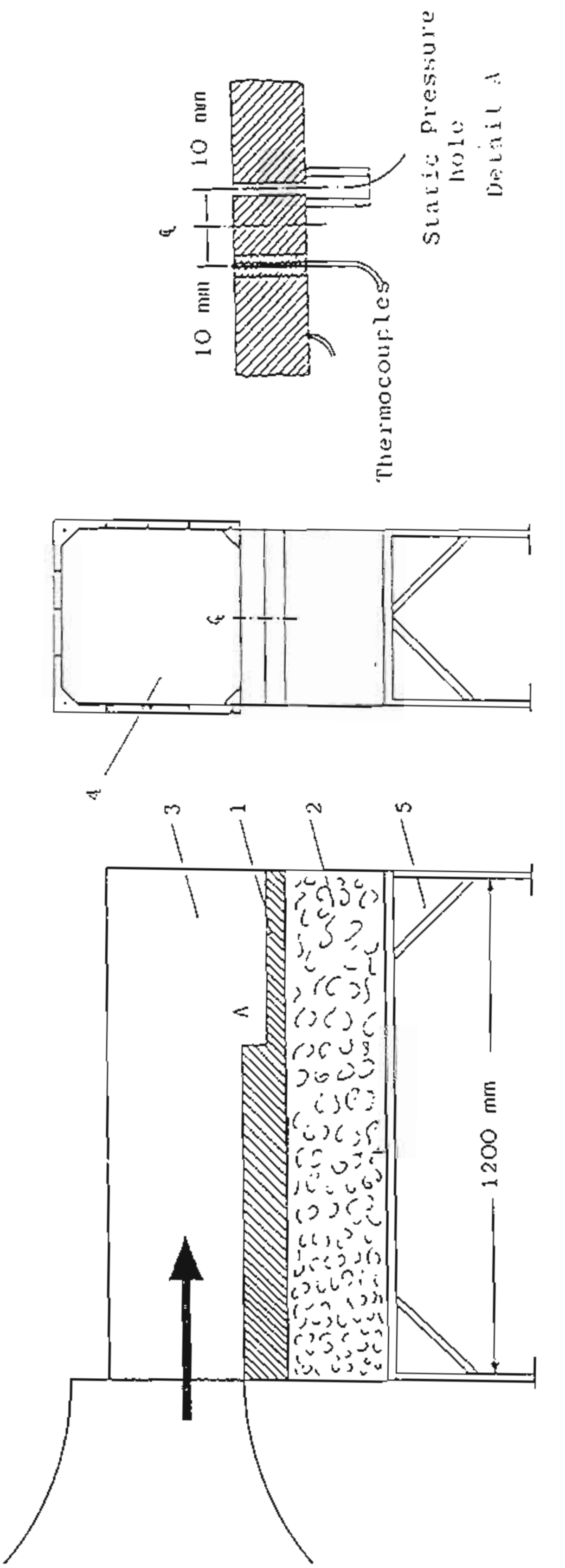

1. Flat plate with step
2. Hot air
3. Side walls of test section
4. Side view of test section
5. Stand

Fig. 1 Test Section

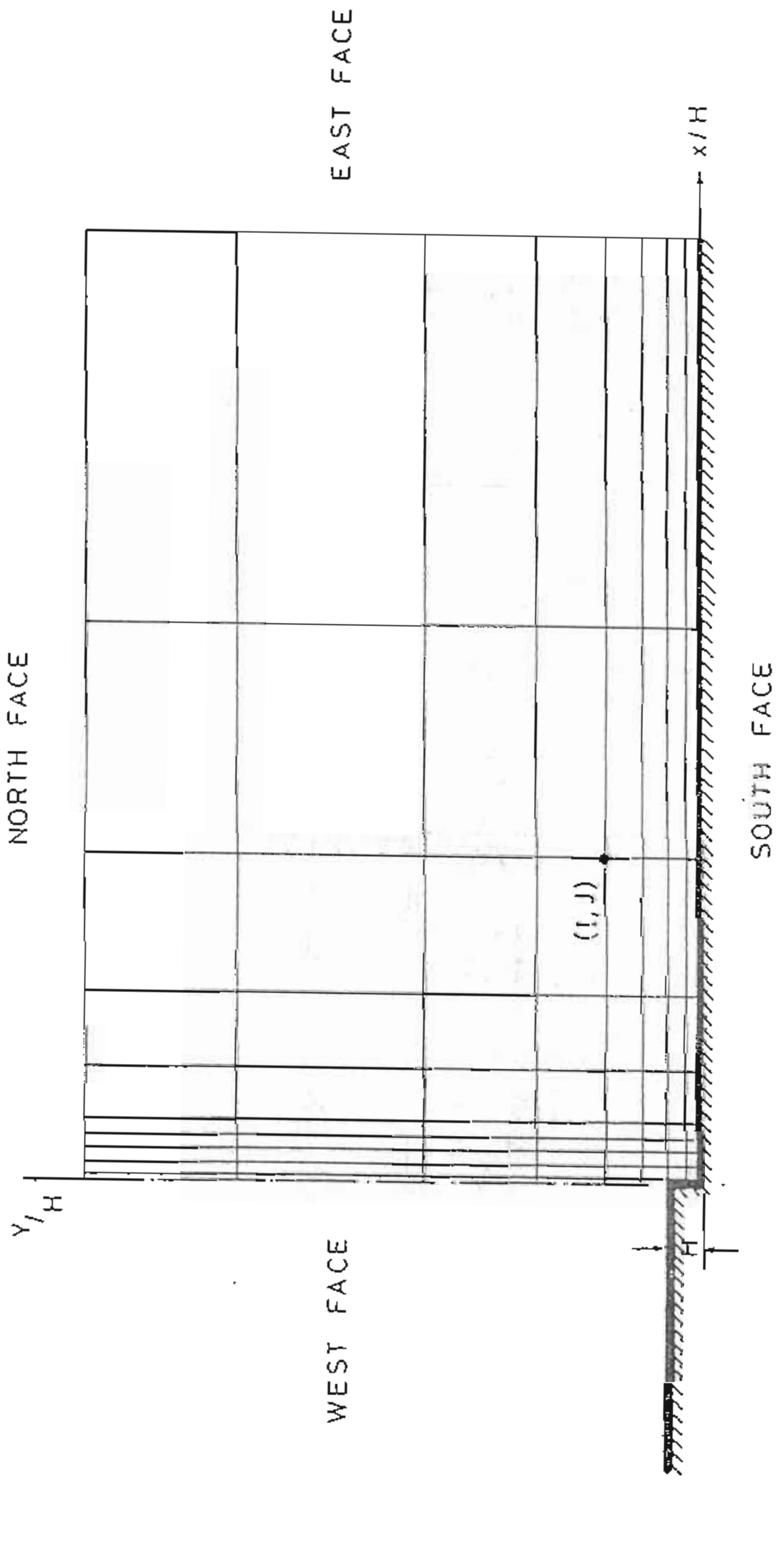

Fig. 2 The two dimensional backstep geometry.

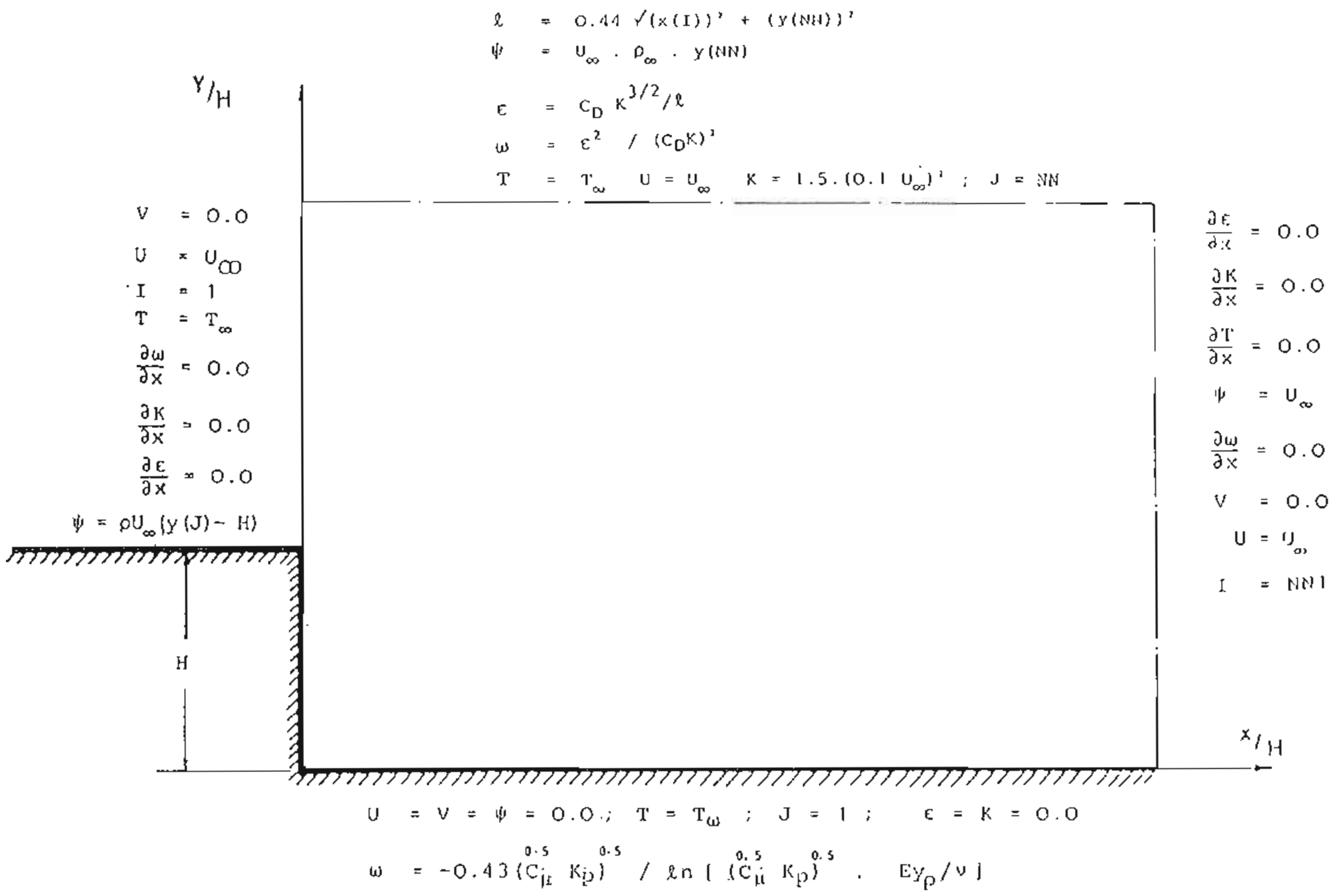

Fig. 3 The boundary conditions.

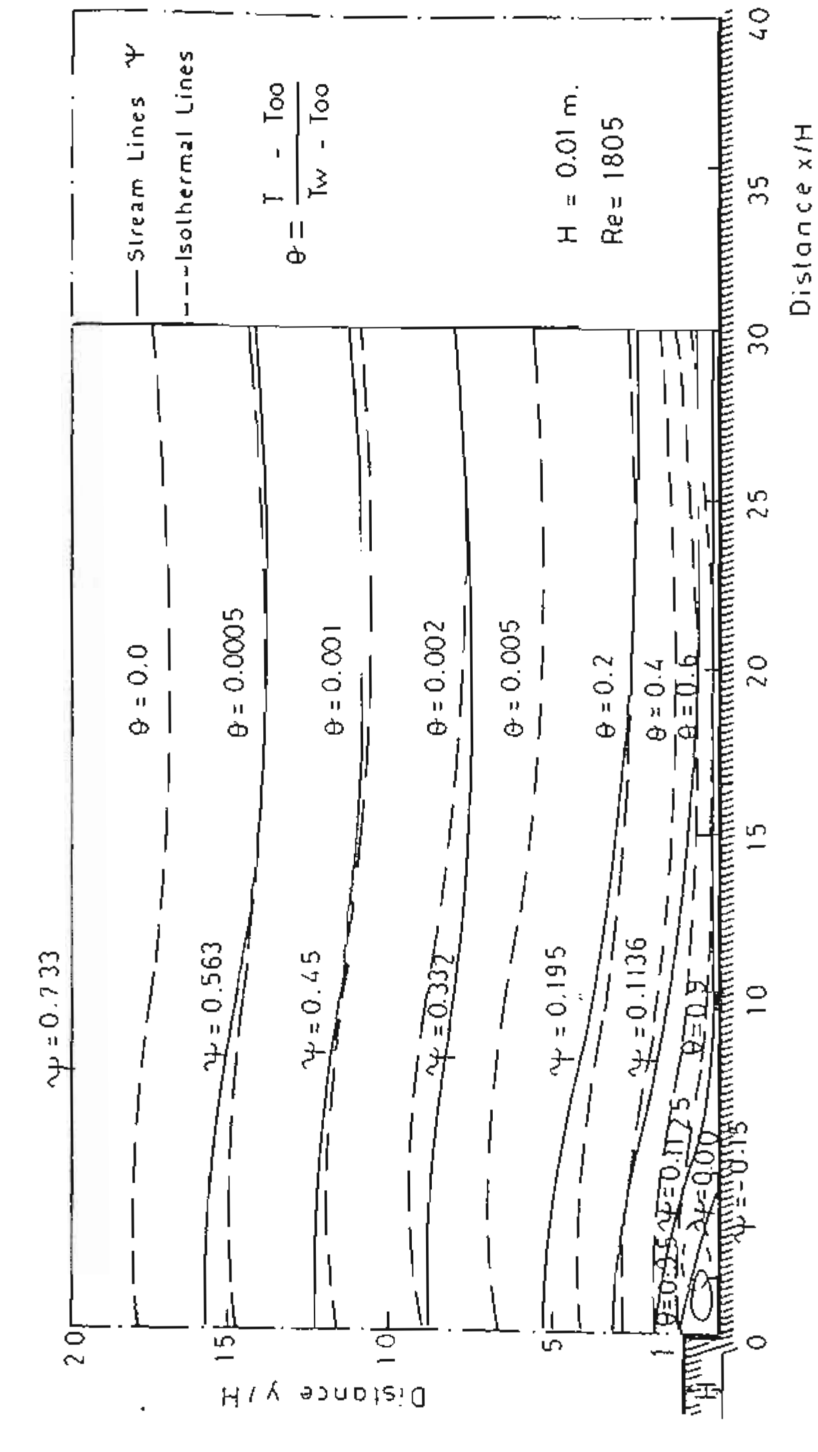

Fig. 4 The experimental results.

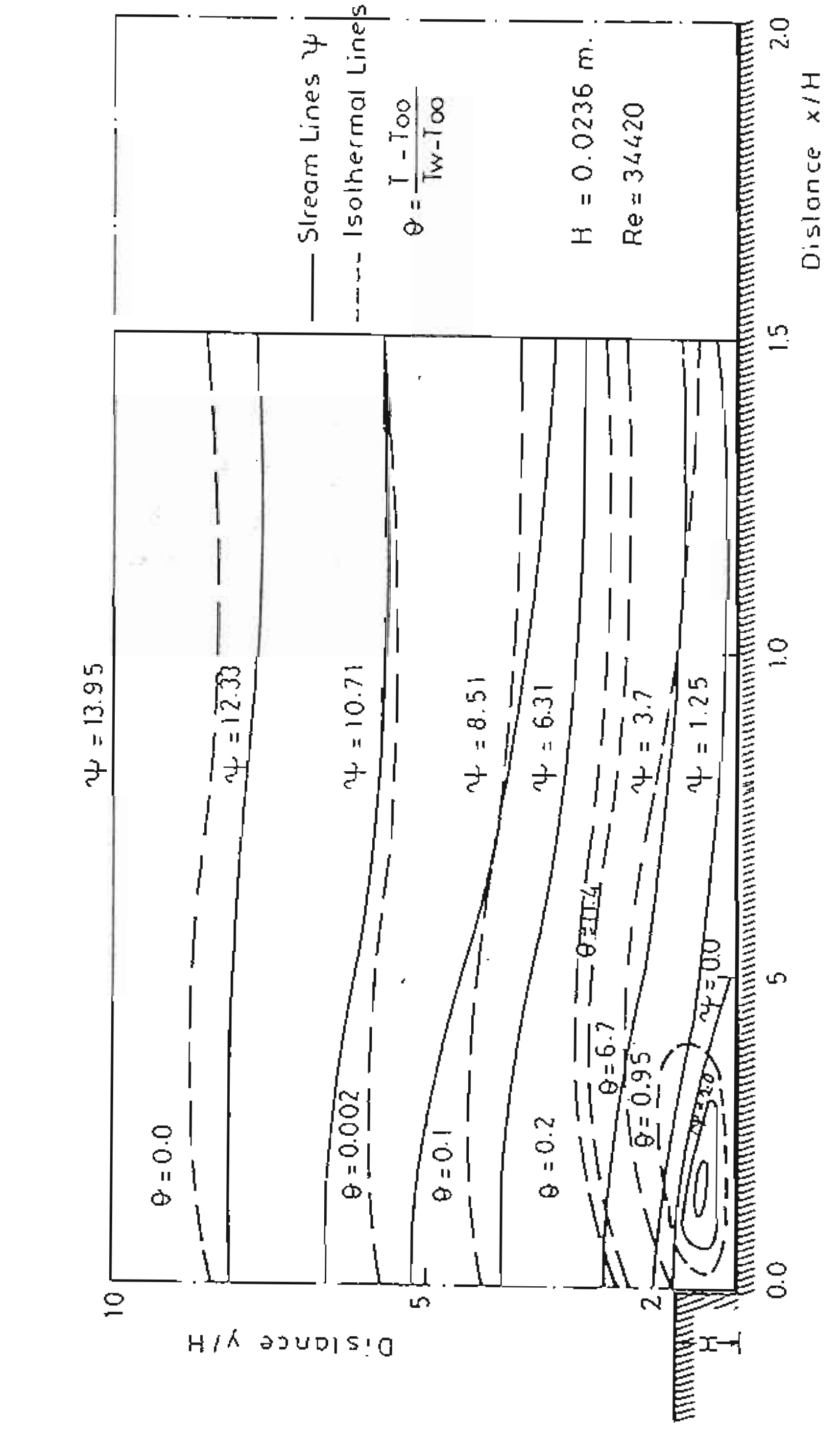

Fig. 5 The streamlines and isothermal lines for two different step heights and different Reynolds numbers.

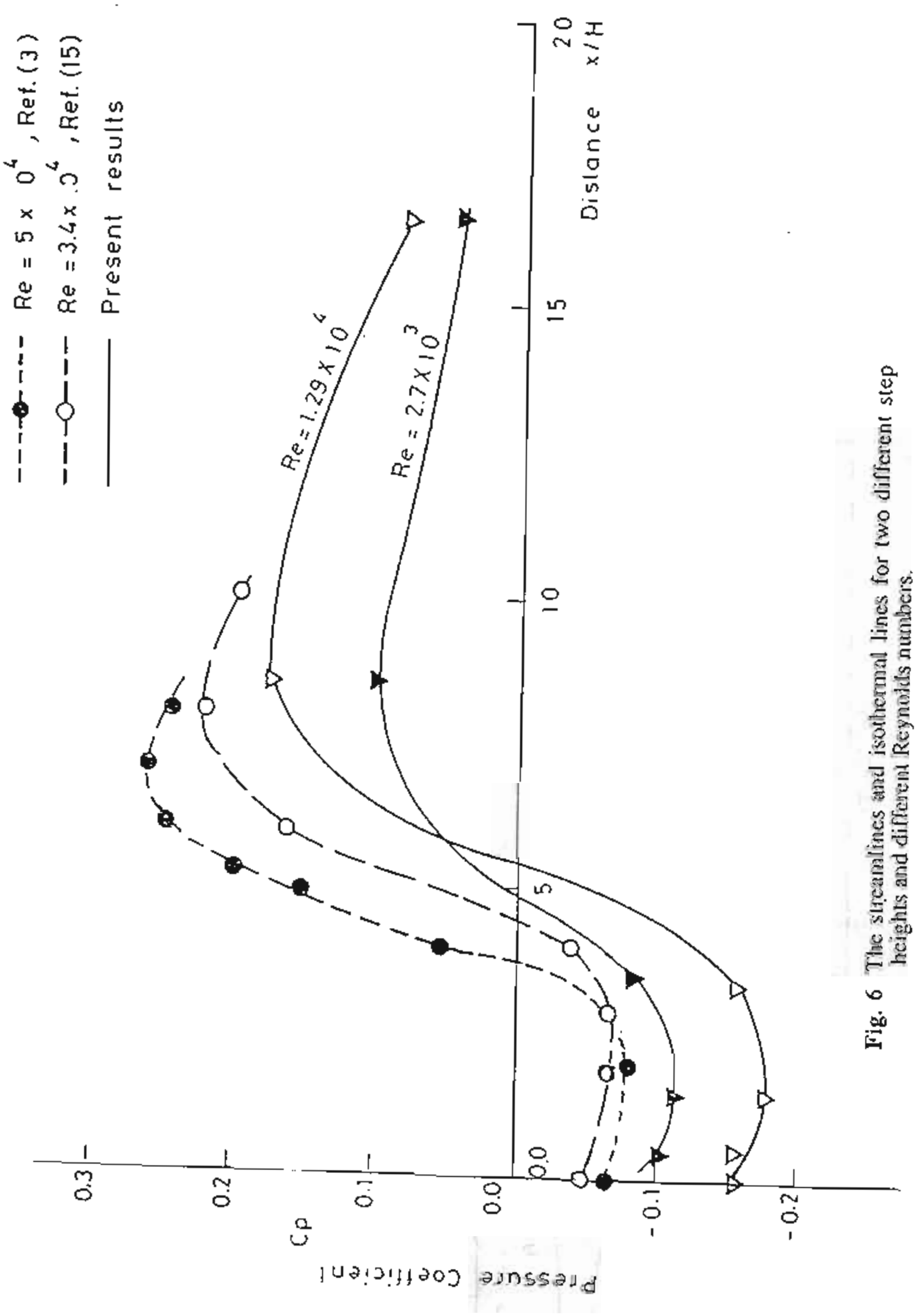

Fig. 6 The streamlines and isothermal lines for two different step heights and different Reynolds numbers.

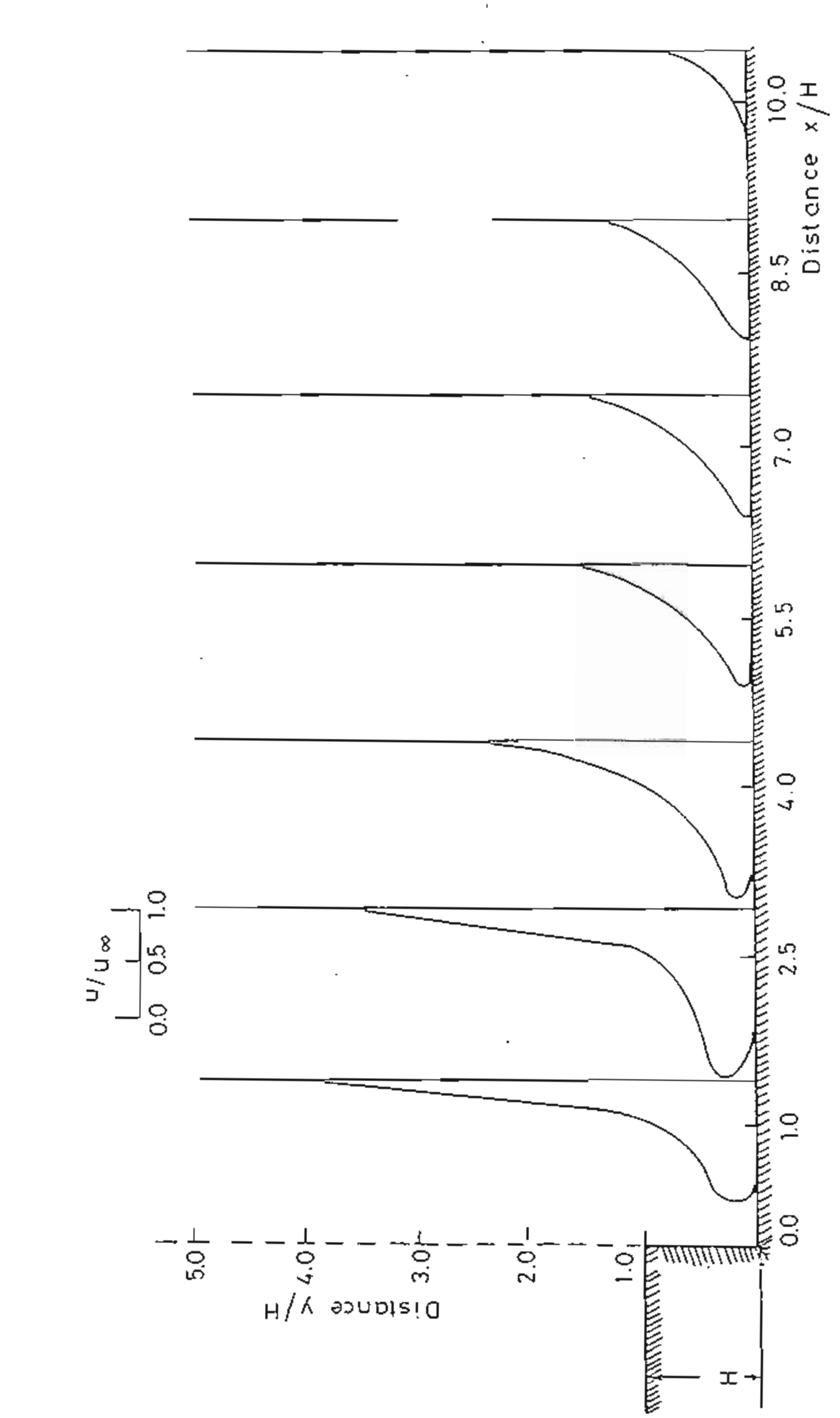

Fig. 7 The Velocity profiles at Various stations along the test section For H = 1.0.

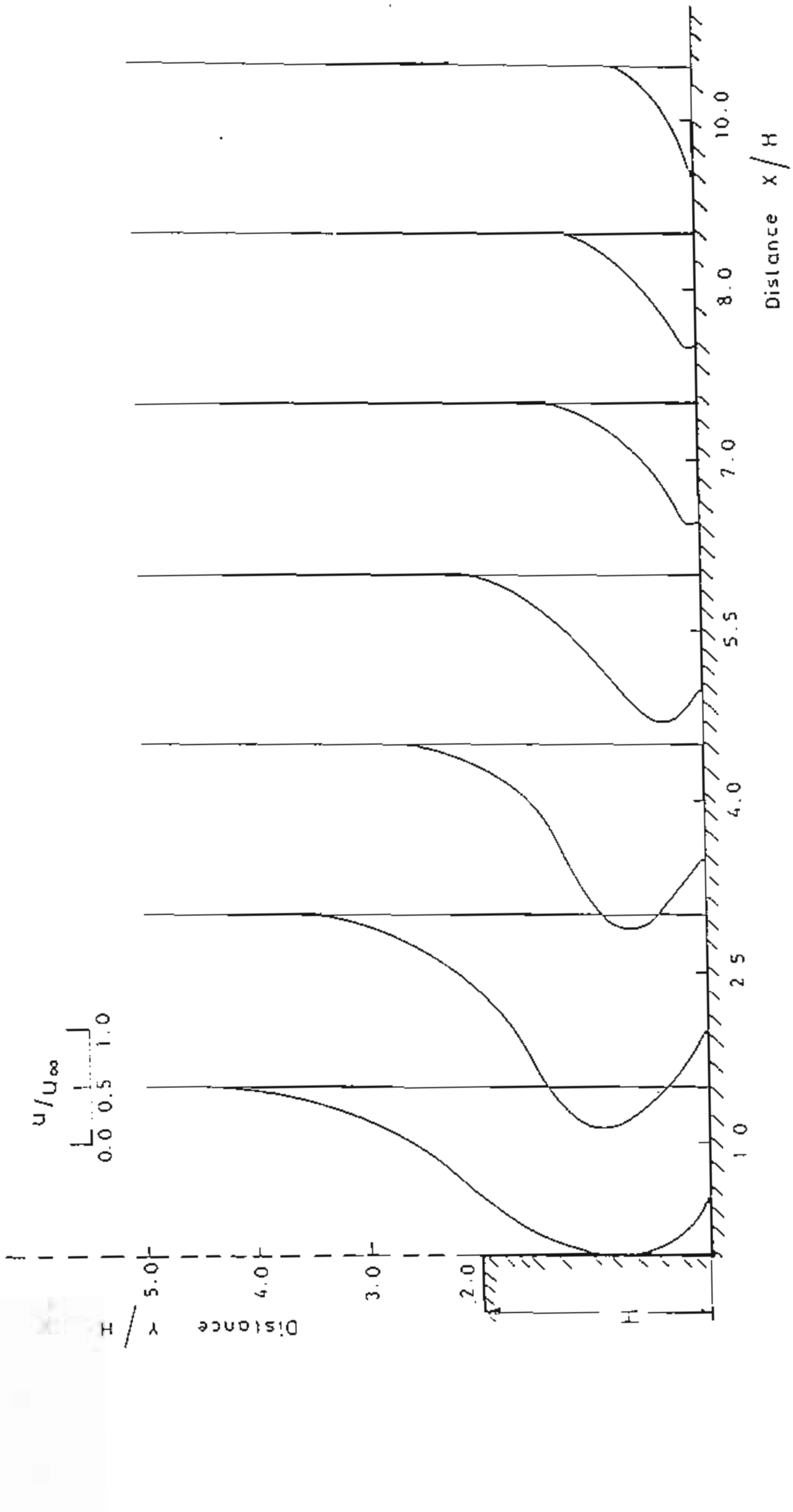

Fig. 8 The Velocity profiles at Various stations along the test section For H = 2.0.

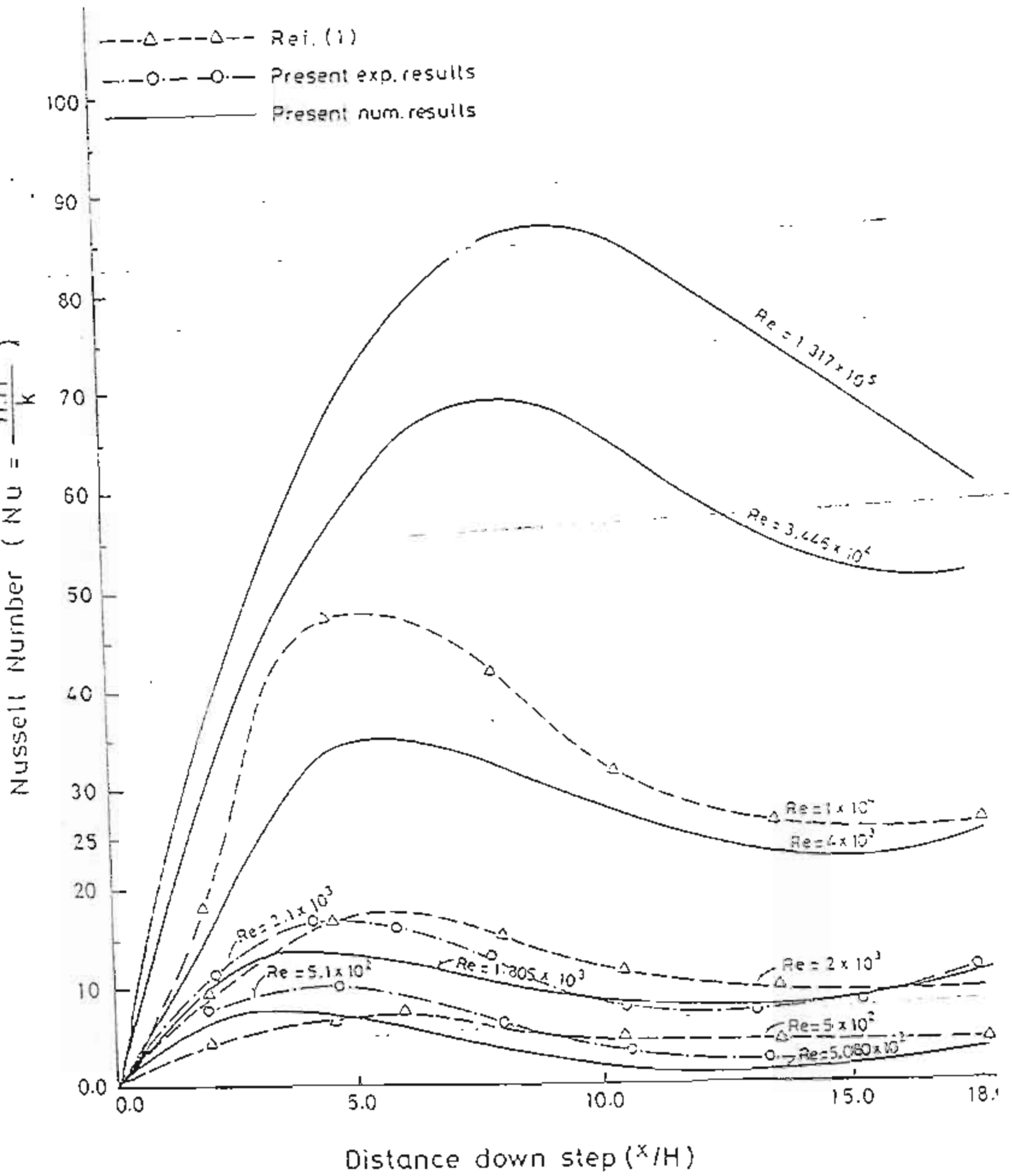

Fig. 9 Exepremental and resrlts Numencal.

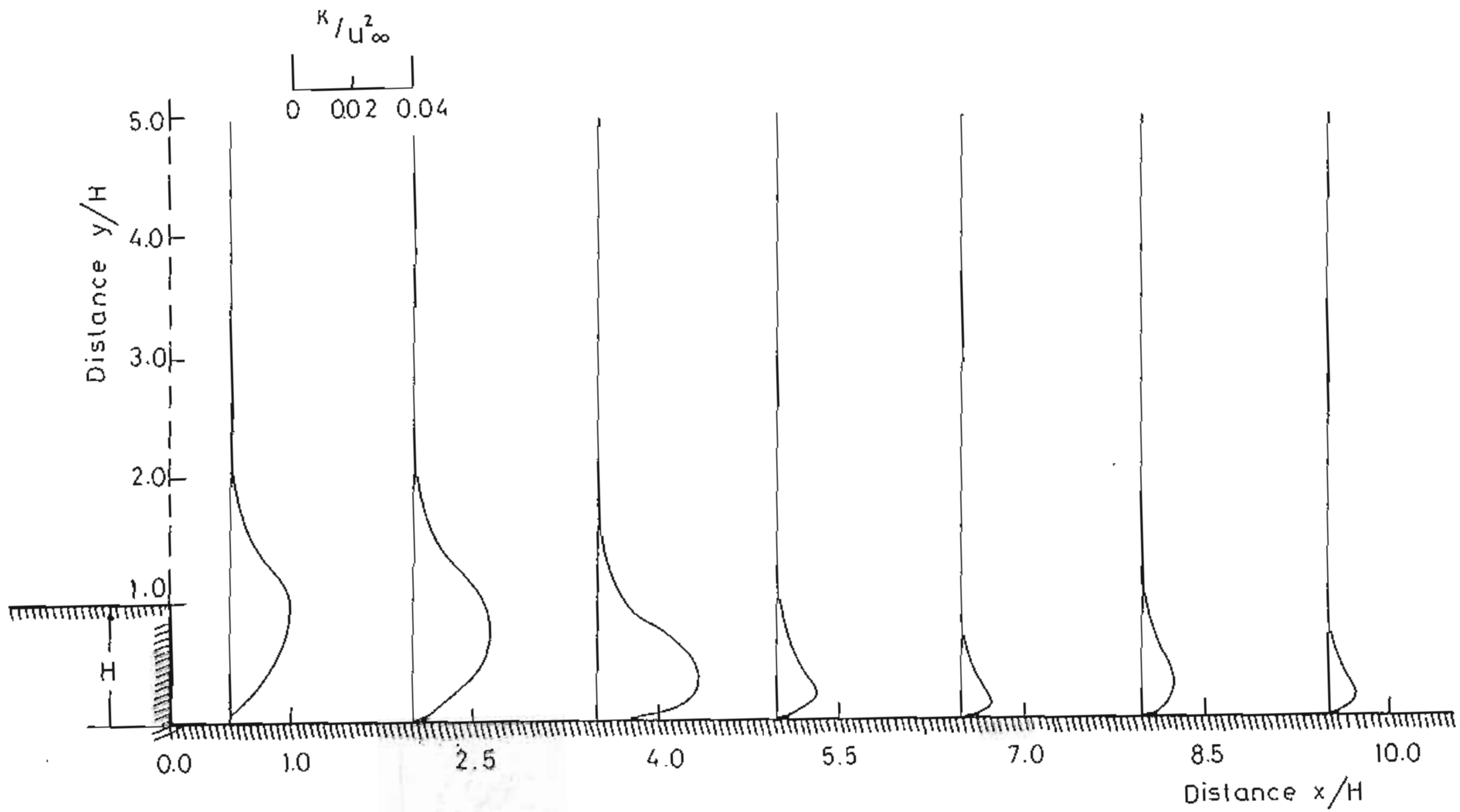

Fig. 10 The effect of the Variation of (K) over the test section for H = 1.0.

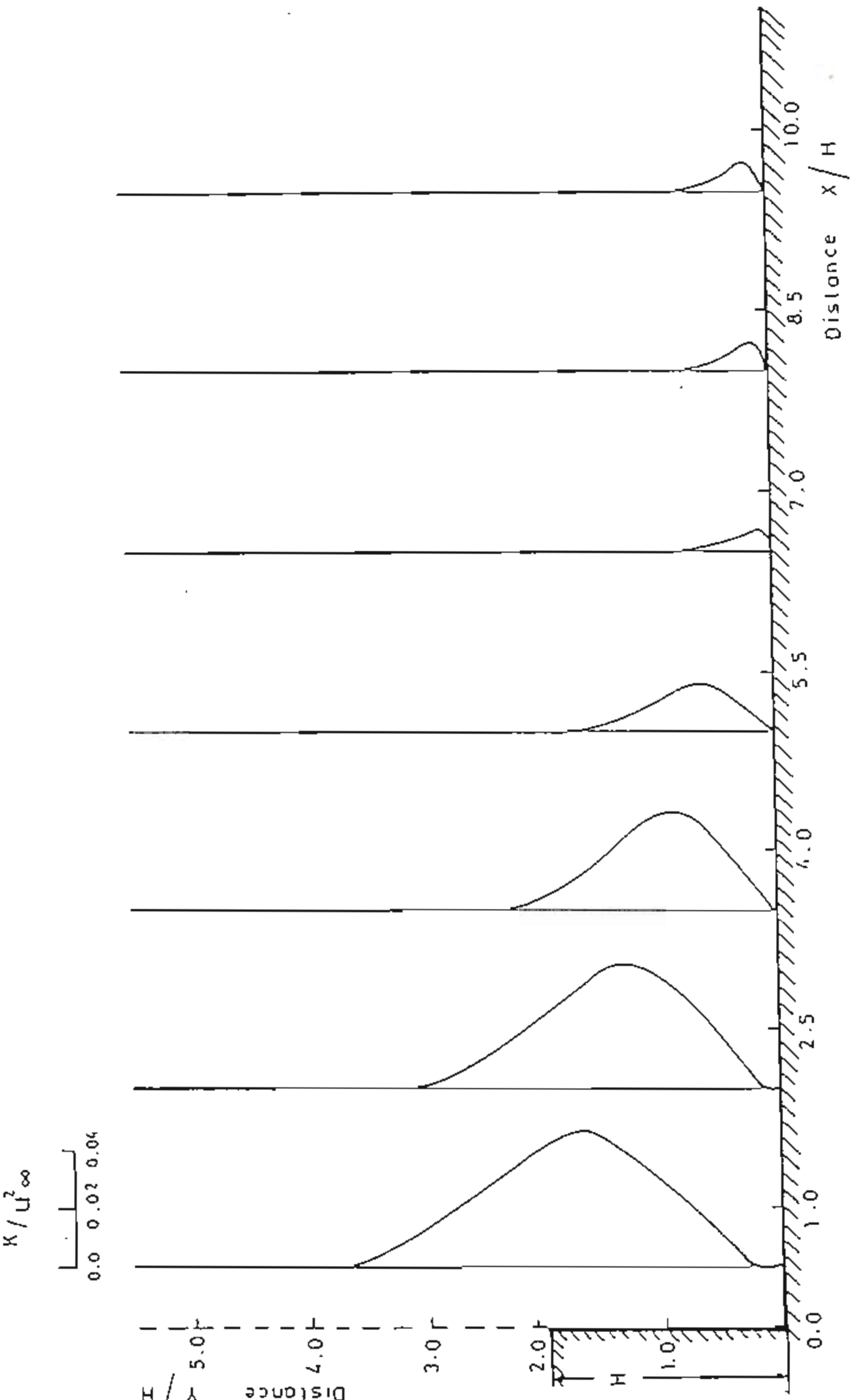

Fig. 11 The effect of the Variation of (K) over the test section for H = 2.0.