

BİLİM VE TEKİNİĞE KATKILAR

TRANSIENT CONJUGATED HEAT TRANSFER
WITH THERMALLY DEVELOPING LAMINAR PIPE FLOW

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Abstract: This paper presents an analysis for an unsteady conjugated heat transfer problem in thermally developing laminar pipe flow, involving two-dimensional wall and fluid axial conduction. The problem is solved numerically by a finite-difference method for a thick-walled, two-regional pipe with a step change in the constant outside temperature of the heated downstream section, which is initially isothermal. A parametric study is done to analyze the effects of four defining parameters namely, Peclet number, wall-to-fluid thermal conductivity ratio, wall-to-fluid thermal diffusivity ratio and wall thickness to inner radius ratio.

(Key words: Conjugated heat transfer, thermally developing flow, finite difference method)

BORULARDA LAMİNER AKIŞ ISIL GELİŞME BÖLGESİ
GEÇİCİ REJİM BİRLEŞİK ISI TRANSFERİ

Özet: Bu çalışmada, borularda laminer akış ısı gelişme bölgesi geçici rejim birleşik ısı transferi, iki boyutlu yüzey ve eksenel akışkan iletimi gözönüne alınarak incelendi. Problem, kalın cidarlı, iki bölge ve başlangıçta eşsıcaklıklı olan bir borunun, sabit dış yüzey sıcaklığında, ısıtılan alt akış bölgesinde meydana gelen basamak değişikliği sınır şartı altında sonlu farklar yöntemiyle sayısal olarak çözüldü. Problemi tanımlayan dört boyutsuz değişkenin; Peclet sayısı, yüzey-akışkan ısı iletkenlik katsayısı oranı, yüzey-akışkan ısı yayılım katsayısı oranı ve cidar kalınlığı-iç yarıçap oranı; etkileri incelendi.

(Anahtar kelimeler: Birleşik ısı geçişi, ısı olarak gelişmekte olan akış, sonlu farklar yöntemi)

NOMENCLATURE

a	Constant of discretization equation (Eq. 6)	δr	Radial position difference
c_p	Specific heat at constant pressure	δx	Axial position difference
d	Wall thickness	Δr	Radial step size
k	Thermal conductivity	Δt	Time step increment
Nu	Nusselt number	Δx	Axial step size
Pe	Peclet number	ρ	Density
q	Heat flux	Subscripts	
r	Radial coordinate	b	Bulk
t	Time	f	Fluid
T	Temperature	i,j	At nodal point i,j
u	Velocity	m	Mean
x	Axial coordinate	r	Radial
α	Thermal diffusivity	s	Solid

*Yazar daha önceki sayılarımızda tanıtılmıştır.

sf	Solid to fluid ratio
w	At wall-fluid interface
x	Axial

Superscripts

'	Dimensionless quantity
o	At previous time step

1. INTRODUCTION

Analysis of unsteady conjugated heat transfer is important for control of heat exchangers during startup, shutdown or any change in the operating conditions. The transient behavior of heat transfer in laminar internal flow has been investigated by many authors under step or periodic change of either boundary or fluid inlet conditions. Early investigators have considered extremely thin walls where wall conduction effects are neglected and conditions at the outer surface of the pipe or duct prevail along the inner surface. In problems that are referred to as conjugated, the thermal boundary conditions along the solid-fluid interface are not known a priori, and the energy equations should be solved under the conditions of continuity in the temperature and/or heat flux.

Conjugated heat transfer problems in laminar duct flow under steady state conditions have been studied for several geometries and for different boundary conditions since 1980's and a summary of the literature survey for the related subject may be found in [1].

The first investigator on unsteady conjugated problem was Succes [2] who studied heat transfer to slug flow between parallel plates with time varying inlet fluid or duct wall temperatures. Krishan [3] worked for fully developed pipe flow with a step change in heat flux or outer surface temperature. Succes and Sawant [4] suggested an improved analytical method for conjugated transient heat transfer problem of a parallel plate duct with periodically varying inlet fluid temperature. Succes [5,6] extended the same problem with duct walls exposed to suddenly changing ambient fluid temperature. Cotta, Mikhailov and Özişik [7] studied for slug flow inside parallel plate channels and in circular ducts and solved the problem analytically for periodic variation of inlet temperature. Lin and Kuo [8] considered a step change in uniform wall heat flux over a finite length of a long circular duct and

solved the problem numerically for the thermal entrance region. The same problem with constant outside surface temperature again for a finite heated length of the pipe was studied by Yan, Tsay and Lin [9]. Analytical methods was used by Travelho and Santos [10] for a parallel plate duct with slug flow and varying inlet temperature and by Olec, Elias, Wacholder and Kaizerman [11] in fully developed pipe flow. Recently, numerical methods were used for thick-walled pipes, considering two-dimensional wall conduction by Schutte, Rahman and Faghri [12] with simultaneously or thermally developing laminar flow under constant wall heat flux; by Lee and Yan [13] with fully developed flow under constant outside surface temperature and by Yan [14] for thermally developing channel flow with convection from the ambient. In all these final investigations the related boundary conditions were applied to a finite length of the pipe or channel.

2. ANALYSIS

In the present problem consideration is given to laminar pipe flow in an infinitely long pipe ($-\infty < x < +\infty$) which has a finite wall thickness, d . A schematic diagram of the problem is shown in Fig.1. At the far upstream ($x = -\infty$) the fluid entering the pipe has a uniform temperature, T_0 , which is equal to the initial temperature of the system. The upstream side of the pipe ($x < 0$) is sufficiently long so that the flow is hydrodynamically developed at the beginning of the heating section. At time $t=0$, the temperature of the outer surface of the heated section ($x > 0$) of the wall is suddenly raised to a new value T_1 and remain constant thereafter until the system to reach the steady state conditions. All physical properties of the fluid and wall are constant and viscous dissipation in the flow is neglected.

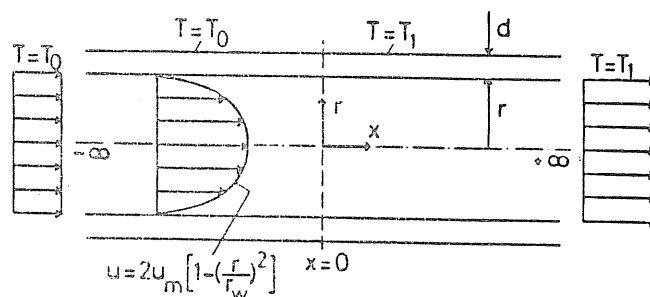


Figure 1. Schematic diagram of the problem.

The above described problem may be

formulated in non-dimensional form as follows:

In the wall region, the differential equation is

$$\frac{1}{\alpha_{sf}} \frac{\partial T'_s}{\partial t'} = \frac{1}{r'} \frac{\partial}{\partial r'} \left[r' \frac{\partial T'_s}{\partial r'} \right] + \frac{1}{Pe^2} \frac{\partial^2 T'_s}{\partial x'^2} \quad (1a)$$

the initial and the boundary conditions are

$$\text{At } t' = 0 \quad T'_s = 0 \quad (1b)$$

$$\text{At } x' = -\infty \quad T'_s = 0 \quad (1c)$$

$$\text{At } x' = +\infty \quad \frac{\partial T'_s}{\partial x'} = 0 \quad (T'_s = 1 \text{ at steady state}) \quad (1d)$$

$$\text{At } r' = 1 + d' \text{ for } x' < 0 \quad T'_s = 0 \quad (1e)$$

$$\text{At } r' = 1 + d' \text{ for } x' \geq 0 \quad T'_s = 1 \quad (1f)$$

$$\text{At } r' = 1 \quad T'_s = T'_f \text{ and } \frac{\partial T'_s}{\partial r'} = \frac{1}{k_{sf}} \frac{\partial T'_f}{\partial r'} \quad (1g, h)$$

In the fluid region, the differential equation is

$$\frac{\partial T'_f}{\partial t'} + (1 - r'^2) \frac{\partial T'_f}{\partial x'} = \frac{1}{r'} \frac{\partial}{\partial r'} \left[r' \frac{\partial T'_f}{\partial r'} \right] + \frac{1}{Pe^2} \frac{\partial^2 T'_f}{\partial x'^2} \quad (2a)$$

the initial and the boundary conditions are

$$\text{At } t' = 0 \quad T'_f = 0 \quad (2b)$$

$$\text{At } x' = -\infty \quad T'_f = 0 \quad (2c)$$

$$\text{At } x' = +\infty \quad \frac{\partial T'_f}{\partial x'} = 0 \quad (T'_f = 1 \text{ at steady state}) \quad (2d)$$

$$\text{At } r' = 0 \quad \frac{\partial T'_f}{\partial r'} = 0 \quad (2e)$$

$$\text{At } r' = 1 \quad T'_f = T'_s \text{ and } \frac{\partial T'_f}{\partial r'} = k_{sf} \frac{\partial T'_s}{\partial r'} \quad (2f, g)$$

Non-dimensional parameters of the problem are defined as

$$T' = \frac{T - T_0}{T_1 - T_0}, \quad x' = \frac{x}{r_w Pe}, \quad r' = \frac{r}{r_w},$$

$$d' = \frac{d}{r_w}, \quad t' = \frac{t \alpha_f}{r_w^2}, \quad k_{sf} = \frac{k_s}{k_f},$$

$$\alpha_{sf} = \frac{\alpha_s}{\alpha_f}, \quad Pe = \frac{2 u_m r_w \rho_f c_{pf}}{k_f}$$

Fluid bulk temperatures, T'_b , interfacial wall heat flux, q_w and local Nusselt numbers, Nu are variables of engineering interest and may be computed as follows:

$$T'_b = 4 \int_0^1 r' (1 - r'^2) T' dr' \quad (3)$$

$$q_w = - \left. \frac{\partial T'_f}{\partial r'} \right|_{r'=1} \quad (4)$$

$$Nu = \frac{2 q_w}{T'_w - T'_b} \quad (5)$$

3. SOLUTION METHODOLOGY

The system of Eqs. (1) and (2) are solved simultaneously by a numerical finite-difference approach. Eq. (1a) and conductive terms in Eq. (2a) are discretized by central-difference schemes and convective terms in equation (2a) are discretized by an exact method defined in Ref.[15]. For the transient terms in the equations, a fully-implicit formulation in time is applied to assure stability in solutions. The following discretization equation is obtained for a nodal point (i,j) in the fluid region.

$$a_{i,j} T'_{i,j} = a_{i+1,j} T'_{i+1,j} + a_{i-1,j} T'_{i-1,j} + a_{i,j+1} T'_{i,j+1} + a_{i,j-1} T'_{i,j-1} + a_{i,j}^o T'^o_{i,j} \quad (6a)$$

where

$$a_{i+1,j} = \frac{(r'_j - r'^3_j) (\Delta r')_j}{\exp[Pe^2 (1 - r'^2_j) (\delta x')_{i+1}] - 1} \quad (6b)$$

$$a_{i-1,j} = \frac{(r'_j - r'^3_j) \exp[Pe^2 (1 - r'^2_j) (\delta x')_{i-1}] (\Delta r')_j}{\exp[Pe^2 (1 - r'^2_j) (\delta x')_{i-1}] - 1} \quad (6c)$$

$$a_{i,j+1} = \left[\frac{r'_j}{(\delta r')_{j+1}} + 0.5 \right] (\Delta x')_i \quad (6d)$$

$$a_{i,j-1} = \left[\frac{r'_j}{(\delta r')_{j-1}} - 0.5 \right] (\Delta x')_i \quad (6e)$$

$$a_{i,j}^o = \frac{r'_j (\Delta x')_i (\Delta r')_j}{\Delta t'} \quad (6f)$$

$$a_{i,j} = a_{i+1,j} + a_{i-1,j} + a_{i,j+1} + a_{i,j-1} + a_{i,j}^o \quad (6g)$$

Temperature distributions at any time step was found by the line by line method [16] visiting the points both in the solid and in the fluid region

from outer surface to axis of the pipe and from upstream to downstream. A consecutive procedure is applied in the solutions by transferring information between the solid and the fluid side using the matching conditions of the problem. For a typical run, Eq. (2f) and therefore the previously calculated temperature distribution at the wall-fluid interface is used as a boundary condition for the fluid region. Iteration is then continued in the solid side by condition (1h) and so the interfacial heat flux values are used to transfer information from the fluid to the solid side.

The grids are laid both in the solid and in the fluid side and contracted in radial direction near the interface in both regions. Axial grids are also contracted in the vicinity of the beginning of the heating (or cooling) zone where the temperature step occurs and linearly stretched by taking the axial step size of a grid as 1.5 times as the previous grid, increasing both in downstream and upstream directions. The minimum step size used in axial direction is 0.0001 while in radial direction it is $d'/8$. The axial distances both for the downstream and the upstream sides, where conditions (1c), (1d), (2c) and (2d) apply, are estimated by a previous steady state solution of the same problem [1] and checked by trial runs with coarse grid systems. In the solutions satisfactory results were obtained by using 12 grid spacings in radial direction (4 in the solid and 8 in the fluid side) and from 35 to 45 grid spacings in axial direction. The number of axial stations and therefore the axial length of the computational region depends on the parameters of the problem and mainly on the Peclet number.

A non-uniform time step is also used to speed up the solutions and to ensure accuracy. The first time step is taken to be 0.0001 and increased by 10% in the subsequent steps.

Accuracy tests were done by increasing the number of grid points, by decreasing the time steps, by changing the locations of the grid points and by reversing the direction of the traversing and sweeping of the points during iterations. The differences between the computed values for any cases were not considerable.

Another indication of the accuracy of the method is the asymptotic time-independent results of the runs. These were compared with the results of Ref. [1] and fairly good agreement was seen.

4. RESULTS AND DISCUSSIONS

Inspection of the analysis shows that the results of the transient conjugated problem depend on four non-dimensional parameters; namely the Peclet number, Pe , the wall thickness ratio, d' , the wall-to-fluid conductivity ratio, k_{sf} and thermal diffusivity ratio, α_{sf} . Computations were made for several combinations of these parameters: $Pe=1, 5$ and 20 ; $d'=1.02, 1.1$ and 1.3 ; $k_{sf}=1, 10$ and 100 and $\alpha_{sf}=0.1, 1$ and 10 . These values were selected as appropriate for problems of engineering interest and from the range that all the presumed effects of the defined problem, i.e. two-dimensional wall and fluid axial conduction are in a meaningful level.

Local Nusselt number as traditionally considered in the presentation of the convection heat transfer results is not a convenient tool for the conjugate problems, [17], since it contains three unknowns in its definition. However, local interfacial heat flux gives more useful information. The results, therefore, presented by local interfacial heat flux values but some results were also given for fluid bulk and inner wall temperature distribution in order to better understand some of the transient behavior of the conjugated problem.

Fig. 2 shows the local interfacial heat flux values for a typical run for $Pe=5$, $d'=0.1$, $k_{sf}=10$ and $\alpha_{sf}=1$ at several time steps. Almost the same behavior is observed for the other runs with different combinations of the parameter values.

As can be seen from the figure, there is a substantial amount of heat flux occurring in the upstream region due to the penetration of heat opposite to the direction of flow, resulting from axial wall and fluid conduction. At early transient period heat flux is from wall to fluid, but after a certain time the situation is reversed. At initial times the wall axial conduction is more rapid and inner wall temperatures are higher than the adjacent fluid temperatures. When time increases, the fluid axial conduction increases and the fluid temperatures in the vicinity of the wall, where convection vanishes, are higher than the inner wall temperatures, and therefore negative interfacial heat flux values are obtained. Since the outer temperature of the wall in the unheated upstream side of the pipe is still kept constant for $t > 0$ with its initial value of T_0 , the pipe wall loses heat outside, resulting shorter penetration lengths in the wall than in the fluid region, leading to reverse heat

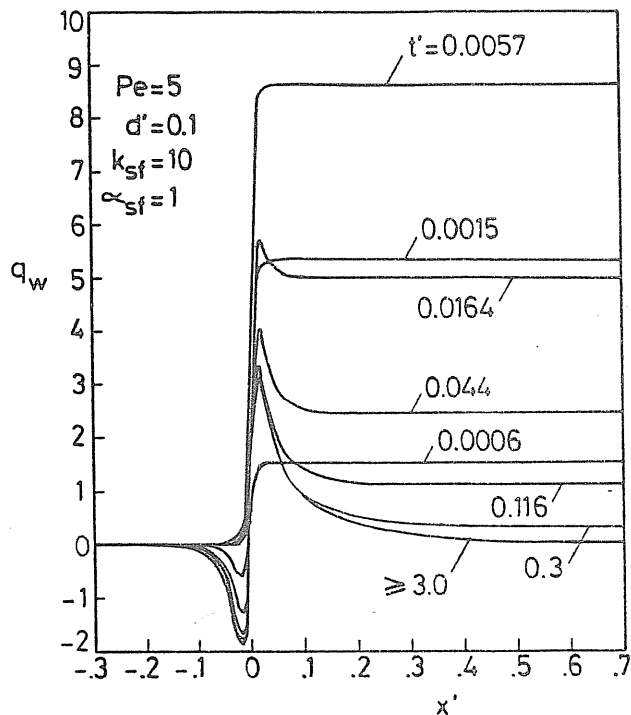


Figure 2. Transient axial distributions of interfacial heat flux.

flux from the fluid to the wall. The amount of heat flux in the upstream section decreases with increasing upstream distance and increases with time.

In the downstream side of the pipe the curves rise to a maximum value and then decrease, and at early times attain a constant value. At the beginning of the transient period the radial wall conduction is dominant which results higher rates of increase in the inner wall temperatures than in the fluid temperatures. As time goes on, heat transfer by convection is more influenced, therefore the value of peak decreases and the downstream uniformity in heat flux values is disturbed. This trend continues until the system reaches to its final steady state.

Figures (3) and (4) show the axial distribution of the fluid bulk and inner wall temperatures respectively for the same run. During the early transient period the downstream part of the curves for both T'_b and T'_w are mainly flat as in q_w curves, and the values are rather smaller in the upstream portion. As time increasing, the increase in both fluid and wall axial conduction causes an increase in both fluid bulk and interfacial temperatures.

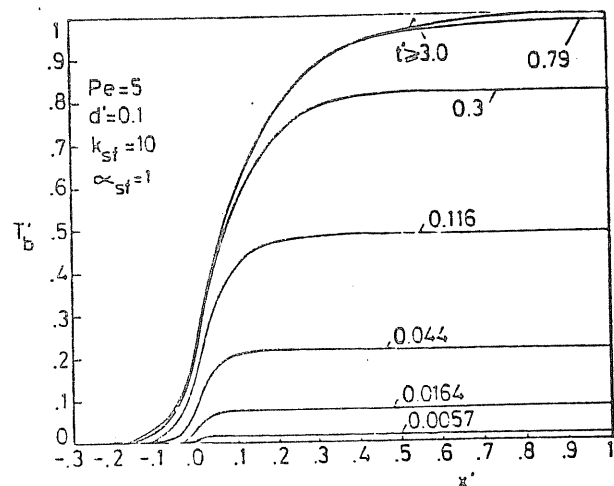


Figure 3. Transient axial distributions of fluid bulk temperatures.

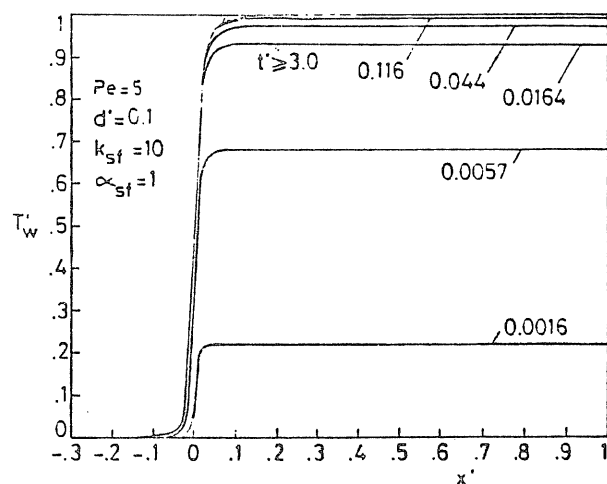


Figure 4. Transient axial distributions of inner wall temperatures.

As can be seen from the figures, at early times inner wall temperatures are higher than fluid bulk temperatures and heat flux is from the wall to the fluid in the upstream portion of the pipe. As time elapses, the penetration of heat backward through the upstream side in the fluid region is more influenced than in the solid side, resulting higher fluid bulk temperatures compared to the inner wall temperatures. This explains why negative heat flux values are obtained in the upstream section of the pipe. At steady state, both fluid bulk and inner wall temperatures reach their asymptotic

value of 1 in the fully developed region.

In order to investigate the effect of pipe wall thickness Fig. 5 is drawn for axial distributions of unsteady interfacial heat flux for different d' values at three different time steps. The thermal resistance and heat capacity of the system are smaller for thinner walls and heat supplied from the outer surface is easily transferred to the fluid. By this reason, for the early transient period, the values of interfacial heat flux are higher for smaller wall thickness. After this early period, when convective heat transfer is becoming more influenced than conduction in the wall, with an increasing effect for thin walled pipes, q_w values are lower for smaller d' values and the curves cross at some distance downstream of the heated section. Since thick walls exhibit more axial conduction, the extent and the magnitude of the reverse interfacial heat flux are smaller and it begins later in the upstream section. The time required to reach the steady state is not much affected by the wall thickness and slightly increased for $d'=0.3$.

Fig. 6 shows the effect of wall-to-fluid thermal conductivity ratio on axial distributions of unsteady interfacial heat flux. Inspection of this figure reveals that the length of penetration and the amount of reverse heat flux in the upstream section are greater with large k_{sf} . Since large k_{sf} means small thermal resistance in the pipe wall, interfacial temperatures and therefore heat flux values are larger in the heated section. The time required to reach the steady state is longer for small k_{sf} values, since small k_{sf} corresponds to small α_{sf} , which means large thermal capacity of the pipe wall.

In Fig. 7 comparative results for interfacial heat flux at some instants of time for different values of Peclet number are given. In the upstream section of the pipe both the extent and the magnitude of the reverse heat flux increased with decreasing Peclet number. This is because of greater amounts of axial fluid conduction and penetration of heat backward through the upstream side in the fluid region for small Peclet numbers. The extent of postheating in the downstream section is also increased with decreasing Peclet number, since convective effect is low and therefore the development length is increasing. The degree of peak is smaller and the drop-off is more gradual in heat flux values for decreasing Peclet numbers.

Finally the effect of diffusivity ratio, α_{sf} , on interfacial heat flux is shown in Fig. 8. At early time period the values of q_w is higher for large α_{sf} . This is because large α_{sf} means small thermal capacity of the pipe wall and the response of q_w is faster. When time elapses, convective heat transfer is dominant over radial pipe conduction with an increasing effect of decreasing α_{sf} . This is due to the fact that, a lower value of α_{sf} means a higher value of k_f in a relative sense. Therefore after a certain time period q_w values are greater for small values of α_{sf} . The development of the curves is faster at initial transient period for the cases of large α_{sf} , but the time to reach the steady state is the same for all cases. The final shape of q_w curves is independent of diffusivity ratio as expected.

One of the most interesting and surprising outcome deduced from the present investigation is that; although the unsteady behavior of conjugated heat transfer is considerably affected by the parameters, especially for early and intermediate periods, the time for the system to reach the steady state is not much affected by a single parameter. The cumulative effect of all four parameters, however is higher but not so high as expected.

Two extreme cases may be considered for comparison. The final time to reach the steady state for the run with $Pe=1$, $d'=0.3$, $k_{sf}=1$ and $\alpha_{sf}=0.1$ is 8.55 while it is 2.72 for the run with $Pe=20$, $d'=0.02$, $k_{sf}=100$ and $\alpha_{sf}=10$. This fact is probably due to the boundary conditions of the present problem which are different than those in References [12], [13] and [14], in which the heated portion of the tube is bounded in a finite length and the outer wall is insulated outside of this region. This result indicates that the thermal inertia of the system is mainly dependent on the flow conditions rather than to be on the wall characteristics.

5. CONCLUSIONS

In the present paper, a transient conjugated heat transfer problem for thermally developing laminar pipe flow is considered, which takes into account radial and axial conduction both in the wall and in the fluid region. The problem is solved numerically by a finite difference method and a parametric study was done to investigate the effects of four defining parameters, Pe , d' , k_{sf} and α_{sf} . The results may be summarized as follows:

1. Considerable amount of heat is

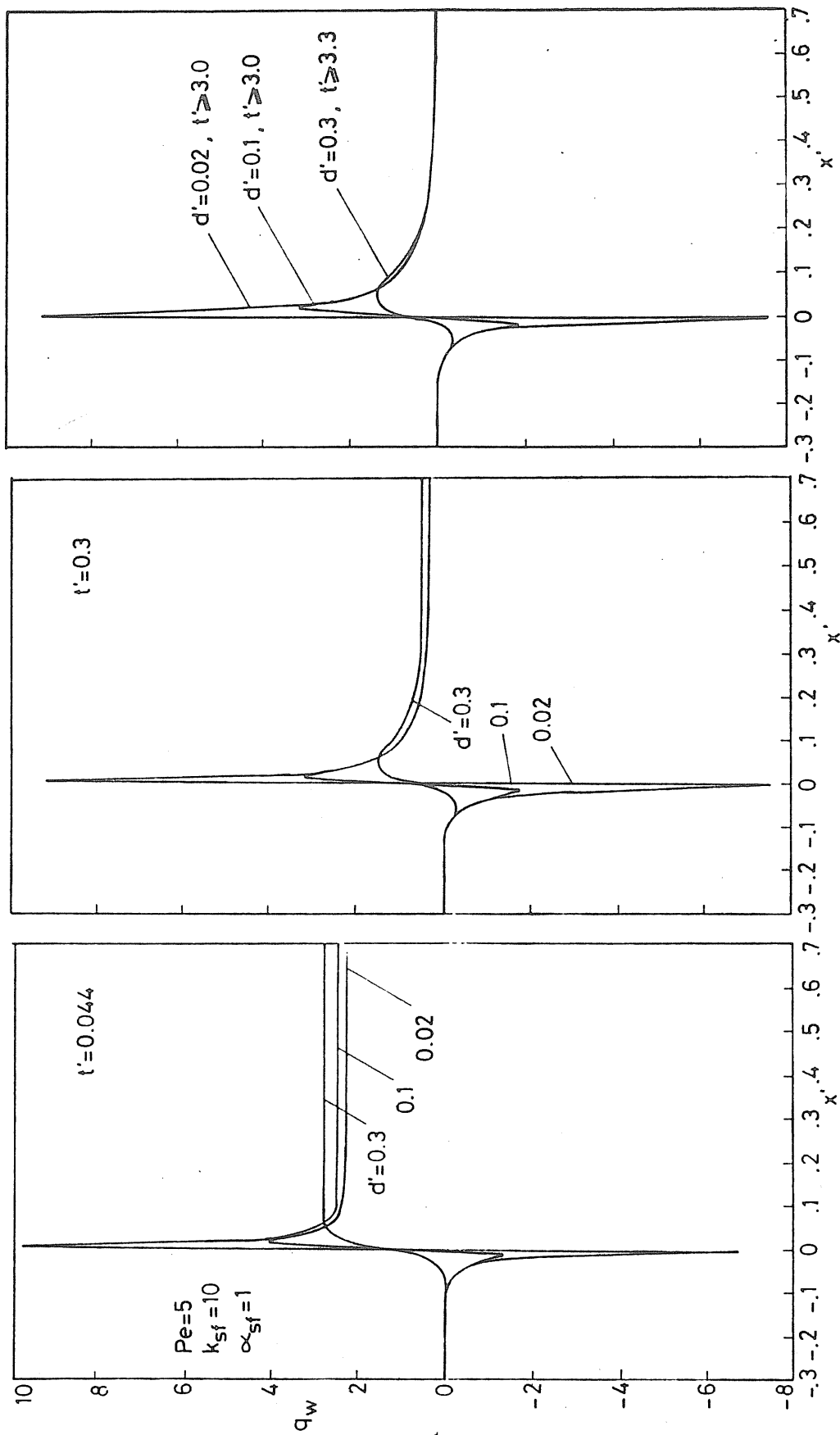


Figure 5. Effect of thickness ratio on interfacial heat flux.

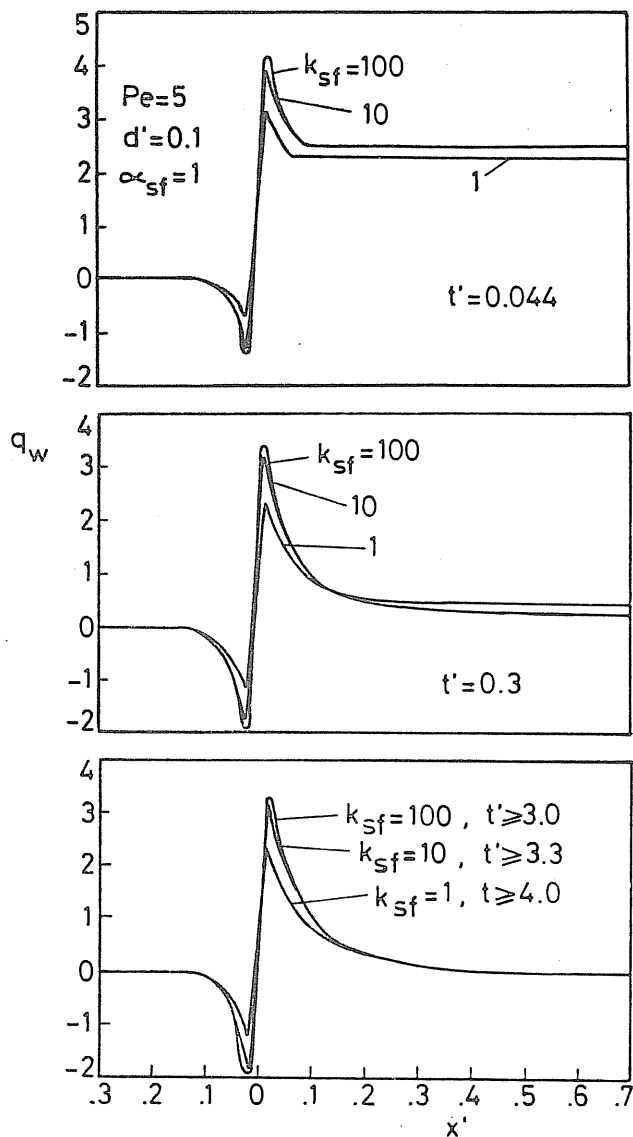


Figure 6. Effect of conductivity ratio on interfacial heat flux.

conducted through the upstream side both in the wall and in the fluid region, and this causes preheating of the fluid before entering to the heated section. Due to the outer boundary temperature of the unheated part of the pipe, the wall loses energy to the outside, and the length of backward penetration of heat by axial conduction may be greater in the fluid region. This causes negative interfacial heat flux, i.e. from fluid to the wall in the upstream section of the pipe.

2. In the downstream side of the pipe, interfacial heat flux values rises to a maximum and then decrease. At early transient period they attain a constant value, and as time goes on, heat flux values decrease monotonically until the system to reach the steady state.

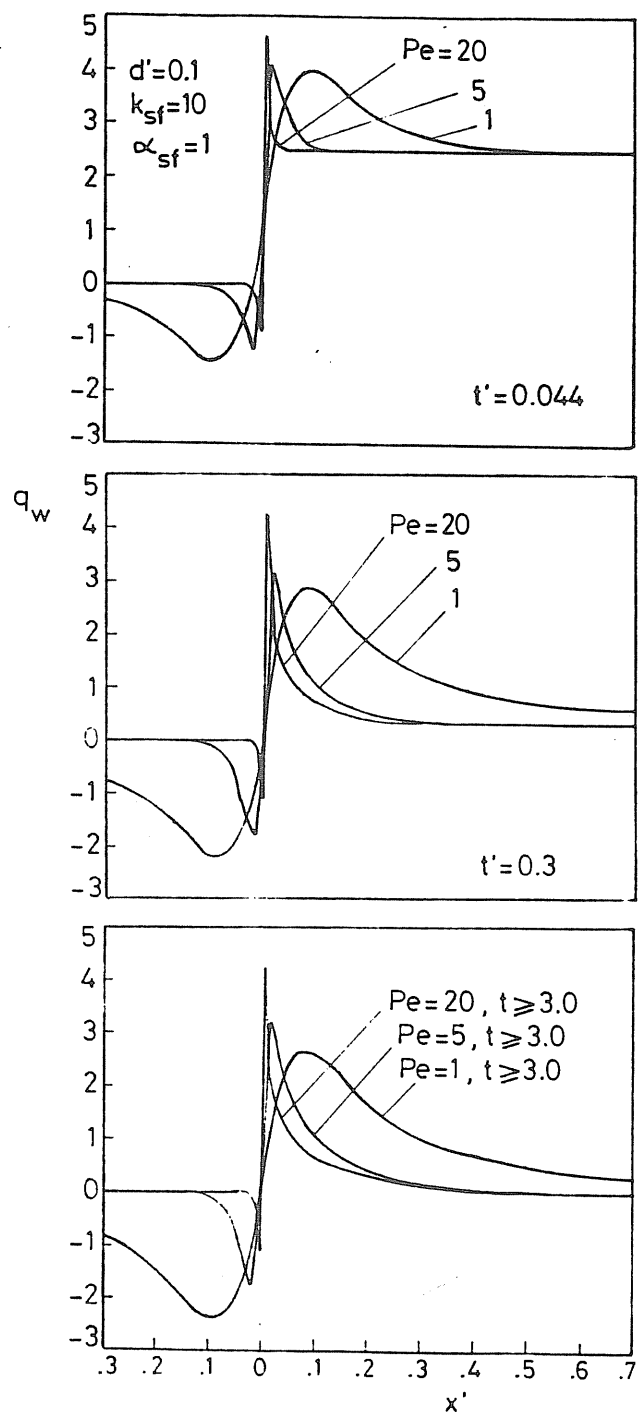


Figure 7. Effect of Peclet number on interfacial heat flux.

3. The effect of wall conduction on heat transfer is more pronounced for high values of d' , k_{sf} and α_{sf} , and for small values of Pe . The effect of the parameters is more pronounced in the early and intermediate transient periods and the time to reach the steady state does not change much with the parameters.

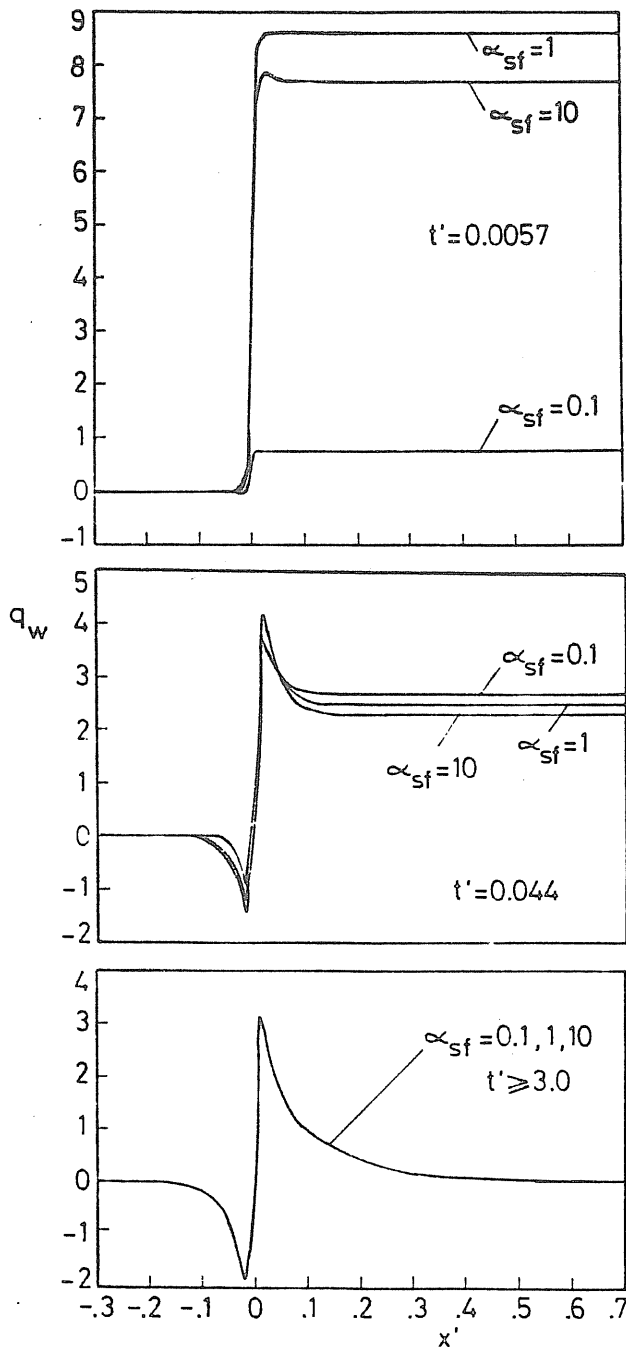


Figure 8. Effect of diffusivity ratio on interfacial heat flux.

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PAMUKKALE ÜNİVERSİTESİ

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