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HEAT TRANSFER IN TUBE BUNDLES OF HEAT EXCHANGERS WITH FLOW BAFFLES INDUCED FORCED MIXING#

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Nuclear Depratment

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ABSTRACT

Thermal analysis of shell-and-tube heat exchangers is being investigated through geometric modeling of the unit configuration in addition to considering the heat transfer processes taking place within the tube bundle. The governing equations that characterize the heat transfer from the shell side fluid to the tube side fluid across the heat transfer tubewalls are indicated. The equations account for the heat transfer due to molecular conduction, turbulent thermal diffusion, and forced fluid mixing among various shell side fluid channels. The analysis, though general in principle, is being applied to the Clinch River Breeder Reactor Plant - Intermediate Heat Exchanger, which utilizes flow baffles appropriately designed for induced forced fluid mixing in the tube bundle. The results of the analysis are presented in terms of the fluid and tube wall temperature distributions of a non-baffled and baffled tube bundle geometry. The former case yields axial flow in the main bundle region while the latter is associated with axial/cross flow in the bundle. The radial components of the axial / cross flow yield the necessary fluid mixing that results in reducing the thermal unbalance among the heat transfer to the allowable limits. The effect of flow maldistribution, present on the tube or shell sides of the heat exchangers, in altering the temperature field of tube bundles is also noted.

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INTRODUCTION

Detailed knowledge of the fluid and thermal fields of large size heat exchangers, such as those used in nuclear applications, is of paramount importance in the thermal/hydraulic and the thermal/structural unit design analyses [1]. This is becoming essential for the purpose of ensuring that these units achieve their performance rating requirements with a high degree of reliability within their design lives. For the flow fields in nuclear heat exchangers, this has been determined by flow development testing of the prototype and model configurations [2], and by detailed and overall flow distribution analysis [3-5]. Thermal testing of large size heat exchangers, on the other hand, has been hindered by the lack of availability of large capacity testing facilities, and, therefore, testing has been limited to a single tube or a cluster of heat transfer tubes. Although, these tests provide valuable information on tube performance, they do not establish the detailed temperature field within the unit, because this depends on the tube bundle configuration and on the flow distribution within the bundle. Thermal computerized analysis of heat transfer in tube bundles, coupled with its comparison with experimental data on operating heat exchangers, would provide the necessary tools to confirm the design analysis of such units.

The present paper deals with the thermal performance characteristics of shell-and-tube heat exchangers currently in use in nuclear applications. Emphasis is on investigating, in detail, the heat transfer processes taking place within the tube bundles, rather than employing the Lumped Type Formulation commonly used in heat exchanger design. Of special interest in the present analysis is the examination of the effects of the following geometric and physical parameters on the heat exchanger thermal/hydraulic design:

1. The overall configuration of the tube bundle including the influence of the entrance and exit regions in establishing the temperature field.

2. The role of the flow baffles (sometimes denoted as support plates) in altering the flow distribution in the tube bundle, and consequently the temperature field and the heat exchanger thermal performance.
3. The influence of the physical heat transfer processes of molecular heat conduction, of turbulent thermal diffusion, and of forced fluid mixing in equalizing the lateral temperature distribution on the shell side, and accordingly in minimizing the temperature unbalance among the heat transfer tubes.
4. The degree to which any flow maldistribution present on the tube or the shell side alters the temperature field of heat exchangers.

Consideration of these parameters in heat exchanger design would provide complete maps of the temperature fields of both the working fluids and of the heat transfer tubes. Knowledge of the temperature distributions of both the shell and the tube side fluids will ensure that the unit satisfies its thermal performance requirements. On the other hand, values of the material temperatures of the heat transfer tubes are required in the thermal/structural analysis of heat exchangers. This is to ensure that none of the tubes is excessively loaded due to any thermal unbalance in the tube bundle, a situation which has to be avoided in the general design of heat exchangers partioularly for those employing straight heat transfer tubes as the CRBRP-IHX[1].

This paper presents the equations which govern the heat transfer from the shell side fluid to the tube side fluid across the heat transfer tube walls. On the shell side, the tube bundle is divided into several fluid channels, where thermal interaction between various channels is accounted for through the inclusion of molecular conduction, turbulent thermal diffusion and forced fluid mixing. The present heat transfer analysis of tube bundles follows, to a large extent, the techniques established in dealing with the thermal/hydraulics of rod bundles of nuclear reactors [6-8], with modifications for the presence of the tube side flow replacing the specified fuel rod heat fluxes. The thermal analysis is applied to the CRBRP-IHX tube bundle geometry shown in Fig. 1, where the flow field resulting from the presence of the baffle plates in the tube bundle is taken from a previous publication by the authors[4]. Representative results for the temperature distribution in the tube bundle are given here, with a comparison between the results of a baffled and non-baffled tube bundle geometry.

ANALYSIS AND THERMAL MODELLING

A mathematical model for the analysis of heat transfer in tube bundles of heat exchangers would have to include the heat exchange from the shell side fluid to the tube side fluid across the tube wall thicknesses, in addition to accounting for the energy interchange among various regions of the shell side flow. Such a model would have to simulate both the geometric configuration of the unit and the thermal processes within the bundle, and the results should be applicable to the unit design. Figure 2 shows the thermal simulation model of the CRBRP-IHX where the shell side fluid enters the unit through the primary inlet nozzle into the inlet plenum. It then moves upwards in the inlet plenum, and enters the tube bundle. In the entrance region, the shell side flow moves inwards towards the inner shroud while it is being divided into 12 fluid streams moving downwards in the tube bundle. The placement of perforated overlapping flow baffles in the direction of motion of the downward flow, causes the shell side flow to move in an axial/cross flow combination dictated by the design of the baffles [4]. The tube side fluid moves through the downcomer towards the tube side plenum where it is divided among the heat transfer tubes. In the present modeling analysis, the number of tubes located within each of the 12 channel boundaries (as shown in Fig. 3) are lumped into an equivalent tube side channel with its boundary heat exchange surface area identical to that of the tubes. With the shell side fluid streams transferring their convective heat to the tube side fluid in the tube bundle, they subsequently merge into a single stream that leaves the exit region of the tube bundle.

Governing Equations

Under steady-state conditions, the equations which characterize the heat exchange within the shell and tube side fluid channels and across the heat transfer tubewalls are written in finite difference forms as,

Shell Side Fluid Nodes

$$W_{s,i-1}\, C_s(T_{s,i-1} - T_{s,i})_j + w_{ij} C_s(T_{s,j-1} - T_{s,i})_j =$$

$$(UA)_s(T_s - T_w)\Big|_{i,j} + \sum_{j-1,j+1}\left(\frac{S}{\Delta y_j} K_s + w'_{ij} C_s\right)(T_{s,i} - T_{s,j})_j \tag{1}$$

where

$$w'_{ij} = 0.0018\, \frac{S\mu}{c}\, Re_i^{0.9} \left(\frac{c}{d}\right)^{-0.4} \cdot \bar{\psi} \tag{2}$$

$$U_s = \left[(1/h_o) + \frac{t d_o}{2K_w d_m}\right]^{-1} \tag{3}$$

The subscripts j and i refer to the channel number and to the node number within each of the channels. The first and second terms on the left side of equation (1) represent the convective heat to the shell side fluid node from the axial and from the radial direction, respectively. On the right side of equation (1), the first term represents the heat transfer from the shell side coolant to the tube side coolant across the tube wall thickness, while the second term corresponds to the heat transfer by molecular conduction and turbulent thermal diffusion among the shell side fluid nodes in the radial direction. Heat transfer by axial conduction among the nodes has been neglected. The functional dependence of the turbulent diffusion cross flow on fluid velocity and tube arrangement was taken from the experimental data correlation of reference [8], and is being adjusted to account for the difference in the values of momentum and thermal diffusion by the turbulence diffusivity factor $\bar{\psi}$ given in reference [9].

Tube Side Fluid Nodes

$$W_{t,j+1}C_t(T_{t,i} - T_{t,i+1})_j = (UA)_t(T_w - T_t)\Big|_{i,j} \tag{4}$$

where

$$U_t = \left[(1/h_i) + \frac{td_i}{2K_w d_m}\right]^{-1} \tag{5}$$

Equation (4) gives the energy balance on the tube side fluid nodes, where the increase in the convective heat of the node is due to the thermal energy transferred from the tube wall to the tube side coolant. In both equations (1) and (4), the difference in the convective heat at the entrance and exit locations of a specific node is being related to the temperature difference between this node and the upstream node with the flow rate taken as the upstream flow.

Tubewall Material Nodes

Neglecting axial conduction along the walls of the heat transfer tubes, the energy conservation equation for the tube wall nodes under steady-state conditions is,

$$(UA)_s(T_s - T_w)_{i,j} = (UA)_t(T_w - T_t)_{i,j} \tag{6}$$

where U_s and U_t are given in equations (3) and (5). A_s and A_t are the nodal surface areas of the tubes based on tube outer and inner diameters, respectively. Values of the heat transfer coefficients on the tube and shell sides are taken from reference[9]for the case of sodium coolant fluids.

METHOD OF SOLUTION

Equations (1), (4) and (6) constitute a set of linear equations for the fluid and material nodal temperatures. A thermal resistance circuit approach was used to solve these equations by use of the Cinda computer code [10]. In this code, the conductances connecting each node to its neighboring nodes could be included either explicitly or evaluated by subroutines furnished by the user, and incorporated within the code. The code offers the advantage of using one or two-way conductance. Referring to Fig. 2, node "0" on the shell side is influenced by one-way conductances due to forced convection from its neighboring nodes "1" and "4". Two way conductances influence node "0" through molecular conduction and turbulent diffusion from nodes "3" and "4" and through heat transfer to the tube wall of node "2". In present analysis, the two-way conductanced due to turbulent diffusion and heat transfer to and from the tube walls, as given by equations (2), (3) and (5), were evaluated by subroutines tailored specifically to the IHX geometry. In both the entrance and exit regions of the tube bundle, pure cross flow was used for the shell side nodes with no conduction or turbulent diffusion included. As the flow leaves the entrance region, it is taken as uniformally distributed among the tubes. Depending on the presence or absence of the flow baffles, the flow continues to move downwards in an axial or axial/cross flow pattern accordingly. The effect of incorporating perforated flow baffles within the tube bundle is to create radial mixing that result from alternating the axial components of the flow from the maximum values at the baffle windows to their minimum values through the baffles. Table 1 lists the overall thermal/hydraulic parameters of the CRBRP-IHX that are being used in the thermal analysis program. The distribution of the tubes among various channels and the total flow rates per channel are shown in Table II, for both cases of axial and axial/cross flow arrangements. The influence of flow maldistribution on the thermal field of heat exchangers is being investigated through an assumed linear flow maldistribution on both the tube and shell sides which has the effect of maximizing the thermal unbalance among the tubes. Linear flow maldistributions, characterized by flow variation of -5% and +5% of the mean flow in the tube and shell side outer channels, respectively, were used in the present analysis. The flow variation in various other channels under this linear flow maldistribution study is given in Table II.

RESULTS

Figure 4 shows the shell side fluid temperature distribution across the tube bundle at three different axial locations: the entrance region, middle of the bundle, and the exit region. As noted in the figure, the shell side fluid temperature drops as the flow crosses the tubes in the entrance region and moves inwards towards the inner shroud. With the shell side fluid temperature distribution established in the entrance region among various channels, the flow moves downwards in the main bundle, exchanging heat with its neighboring channels while transferring its convected heat to the tube side fluid. In the exit region, the coolant streams, after mixing with the shell side stream of the outer channel, merge into one where it eventually leaves the tube bundle. Figure 4 shows also the effect of molecular conduction and turbulent diffusion among various channels on the shell side fluid temperature. This effect is shown to be of significance only in the regions where large temperature differences occur such as near the inner and outer shrouds. On the other hand, the effect of forced fluid mixing (axial/cross flow) is to alter the temperature distribution in the main bundle. Figures 5 and 6 show the effect of flow maldistribution on the temperature of the shell side and tube side fluid coolants. For the case of axial flow in Fig. 5, the effect of tube side flow maldistribution, which is associated with low flow rates in the outer channels and high flow rates in the inner channels, is to increase the thermal nonuniformity of both the tube and shell side coolants. This situation is being further aggrevated by an additional flow maldistribution on the shell side where high flow rates are assumed to take place in the outer shell side fluid channels and low flow rates in the inner channels. The results of Fig. 6, for the case of axial/corss flow on the shell side, show how forced mixing influences the temperature distribution of the coolants with the effect of the tube side flow maldistirbution being minimized.

In both Figs. 5 and 6, the tube side nodal temperatures seem to follow the corresponding shell side temperatures, as large values of the heat transfer coefficients have been used. Fig. 7 shows the effect of forced fluid mixing on the tubewall mean temperature distribution within the bundle, and as noted maximum temperature differentials of $\pm$ 6^{o}C from the mean temperature of the tubes have been achieved with axial/cross flow arrangement in the tube bundle.

DISCUSSION AND CONCLUSION

The results of the present analysis indicate the significant effect of tube bundle configuration on the thermal field of heat exchangers. With the shell side fluid temperature being established in the entrance region of the tube bundle, the nonuniform temperature distribution of the shell side flow continues to persist in the main bundle region, causing a severe thermal unbalance among the heat transfer tubes. The effect of molecular conduction and turbulent thermal diffusion among various fluid channels on the temperature distribution was found to be slight, and only of significance near the edges of the tube bundle. Flow maldistribution on the tube or the shell sides tends to aggrevate further the thermal unbalance among the heat transfer tubes. Forced fluid mixing, on the other hand, was found to be an effective mechanism in reducing this thermal unbalance among the tubes. A case that uses this forced fluid mixing is the CRBRP-IHX where axial/cross flow is being created in its bundle through the design of its flow baffles.

NOMENCLATURE

A	=	Surface area of the heat transfer tubes per node, $(A = \pi d . \Delta x . N_j)$
c	=	Minimum spacing between the heat transfer tubes, $(c = p - d_o)$
C	=	Specific heat
d_o	=	Outer tube diameter
d_i	=	Inner tube diameter
d_m	=	Mean tube diameter
D_e	=	Equivalent diameter of shell side flow
h_t	=	Tube side heat transfer coefficient
h_s	=	Shell side heat transfer coefficient
K	=	Thermal conductivity
N_j	=	Number of tubes per channel
p	=	Tube pitch in shell side triangular arrangement
Re	=	Reynolds number based on mean axial flow per node, $(Re = \frac{\bar{V} D_e}{\nu})$
S	=	Radial free area between shell side fluid channels per node
t	=	Wall thickness of heat transfer tubes
T	=	Temperature
U	=	Thermal conductance per unit area given by equations (3) and (6)
$\bar{V}$	=	Mean axial shell side velocity
$w_{i,j}$	=	Radial mass flow rate
$w'_{i,j}$	=	Turbulent diffusion cross flow given by equation (2)
W	=	Axial mass flow rate
Δx_i	=	Length of each node in the axial direction
Δy_j	=	Centroid radial distance between two adjacent nodes
ν	=	Kinematic viscosity
$\bar{\psi}$	=	Mean value of the turbulent diffusivity ratio

<u>Subscript</u>

i = Node number within each channel

j = Channel number

s = Shell side coolant

t = Tube side coolant

w = Tubewall

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Table I. Design Parameters of CRBRP-IHX
Used in the Thermal Modeling Analysis

Shell Side Flow, Kg/s	1741.3
Tube Side Flow, Kg/s	1610.2
Shell Side Inlet Temp., °C	535.0
Tube Side Inlet Temp., °C	343.9
Total Number of Heat Transfer Tubes	2850
Tube Outside Diameter, mm	22.22
Tube Wall Thickness, mm	1.143 +7.5%
Tube Arrangement	Triangular Pattern
Pitch to Outside Diameter Ratio	1.5
Number of Baffle Plates	18
Axial Length of Entrance Region, m	0.540
Axial Length of Exit Region, m	0.539
Axial Spacing Between Two Baffles, m	0.379
Centroid Radial Distance Between Two Adjacent Channels, m	0.054

Table II. Distribution of the Heat Transfer Tubes and Flow Rates Among Various Channels

*Parameter	Channel Number "J"											
	1	2	3	4	5	6	7	8	9	10	11	12
N_j	4.24	27.77	31.78	34.81	37.84	40.86	43.41	46.44	49.94	52.97	56.00	27.53
W_t, Kg/s	2.40	15.69	17.95	19.67	21.38	23.08	24.53	26.24	28.22	29.92	31.64	15.55
W_s(For Axial Flow), Kg/s	2.59	16.97	19.42	21.27	23.12	24.97	26.53	28.37	30.51	32.36	34.21	16.82
W_s(For Axial/Cross Flow):												
1. At Outer Baffle Location, Kg/s	10.98	21.88	27.37	33.35	37.82	14.88	17.78	20.46	23.28	25.78	27.33	16.23
2. At Inner Baffle Location, Kg/s	3.23	15.44	16.64	17.73	18.42	18.29	17.66	17.19	16.46	49.17	48.57	38.34
— w(Cross Flow Rate), Kg/s	7.75	14.19	24.92	40.54	59.94	56.53	56.65	59.92	66.74	43.35	22.11	
Tube Side Flow Deviation, %	5.18	4.44	3.7	2.96	2.22	1.48	0.74	0	-1.25	-2.5	-3.75	-5.0
Shell Side Flow Deviation, %	-5.18	-4.44	-3.7	-2.96	-2.22	-1.48	-0.74	0	1.25	2.5	3.75	5.0

*Based on one degree radian pie-section

LIST OF FIGURES

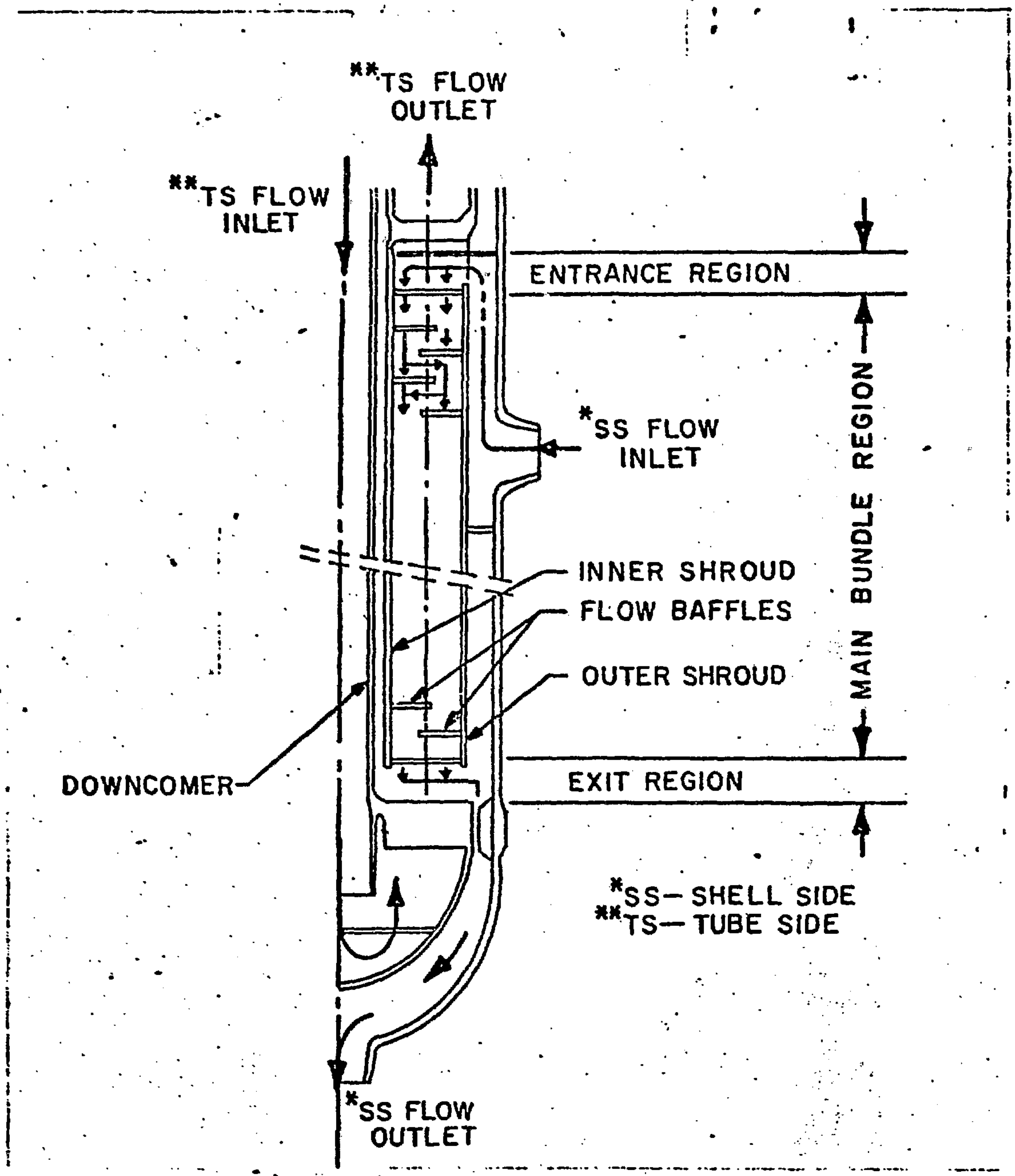

**TS FLOW
OUTLET
**TS FLOW
INLET
ENTRANCE REGION
*SS FLOW
INLET
MAIN BUNDLE REGION
INNER SHROUD
FLOW BAFFLES
OUTER SHROUD
DOWNCOMER
EXIT REGION
*SS— SHELL SIDE
**TS— TUBE SIDE
*SS FLOW
OUTLET

Fig. 1

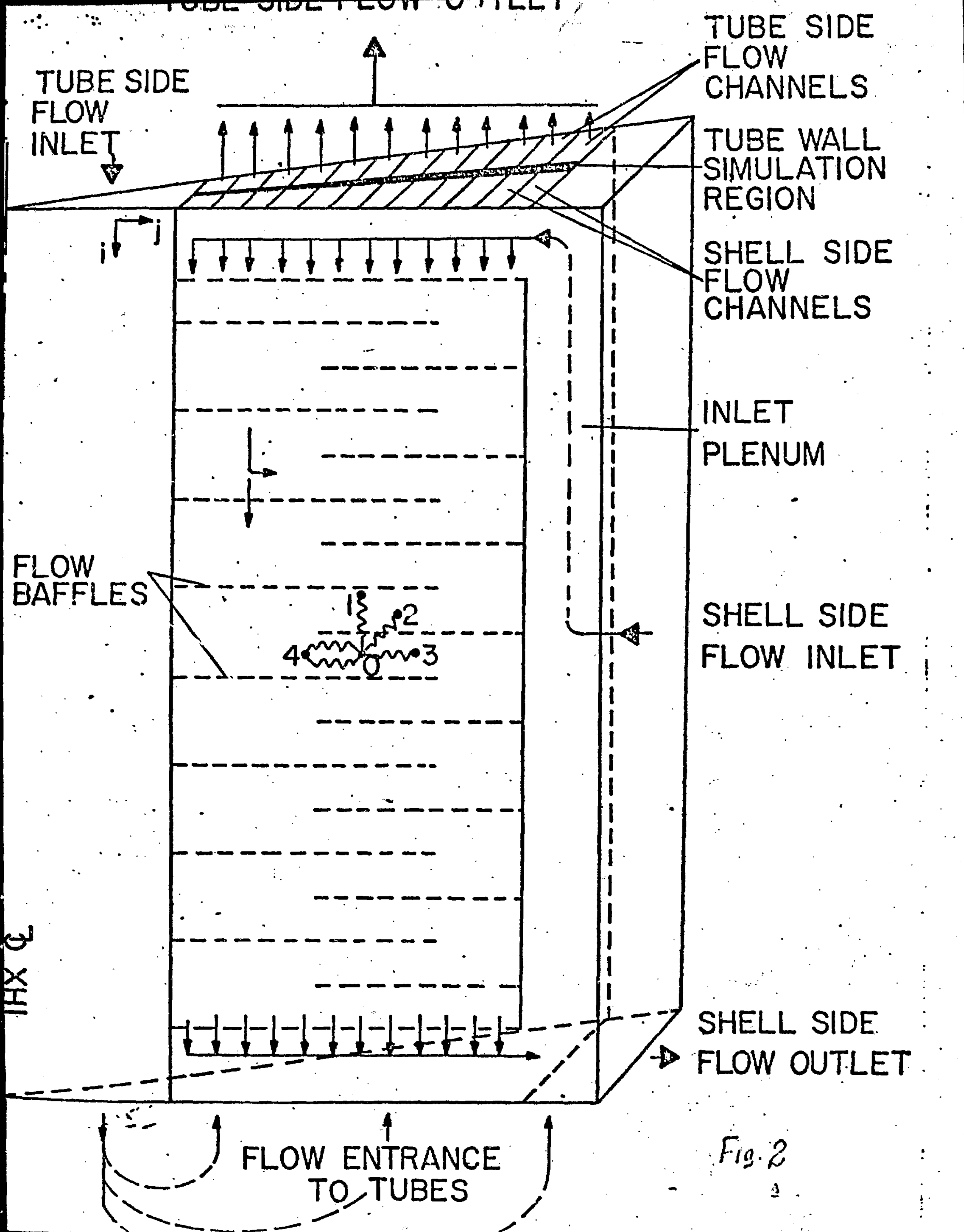

TUBE SIDE FLOW OUTLET
TUBE SIDE FLOW INLET
TUBE SIDE FLOW CHANNELS
TUBE WALL SIMULATION REGION
SHELL SIDE FLOW CHANNELS
INLET PLENUM
FLOW BAFFLES
SHELL SIDE FLOW INLET
SHELL SIDE FLOW OUTLET
FLOW ENTRANCE TO TUBES
IHX ℄
i
j
0
1
2
3
4

Fig. 2

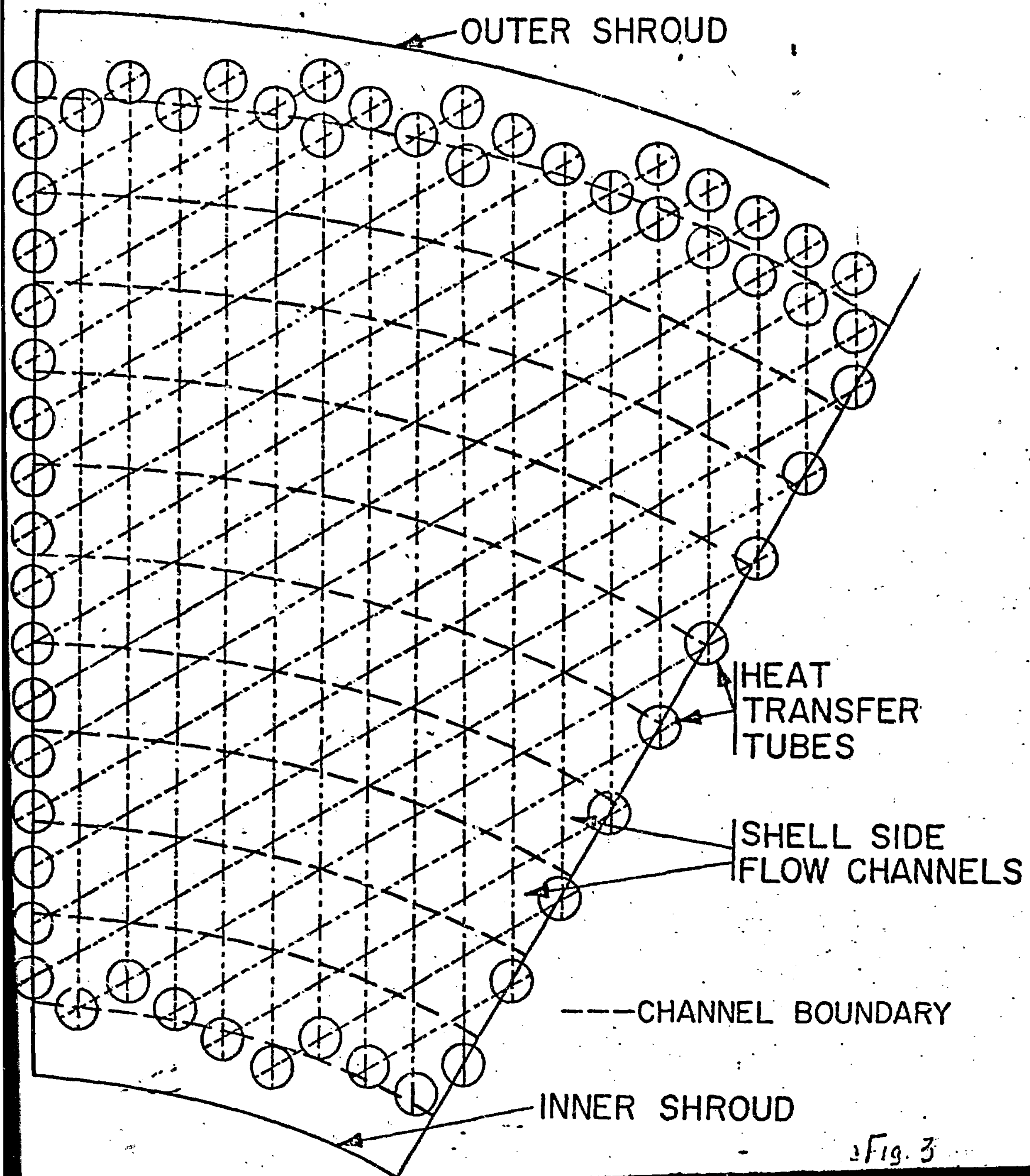

OUTER SHROUD
HEAT TRANSFER TUBES
SHELL SIDE FLOW CHANNELS
CHANNEL BOUNDARY
INNER SHROUD

Fig. 3

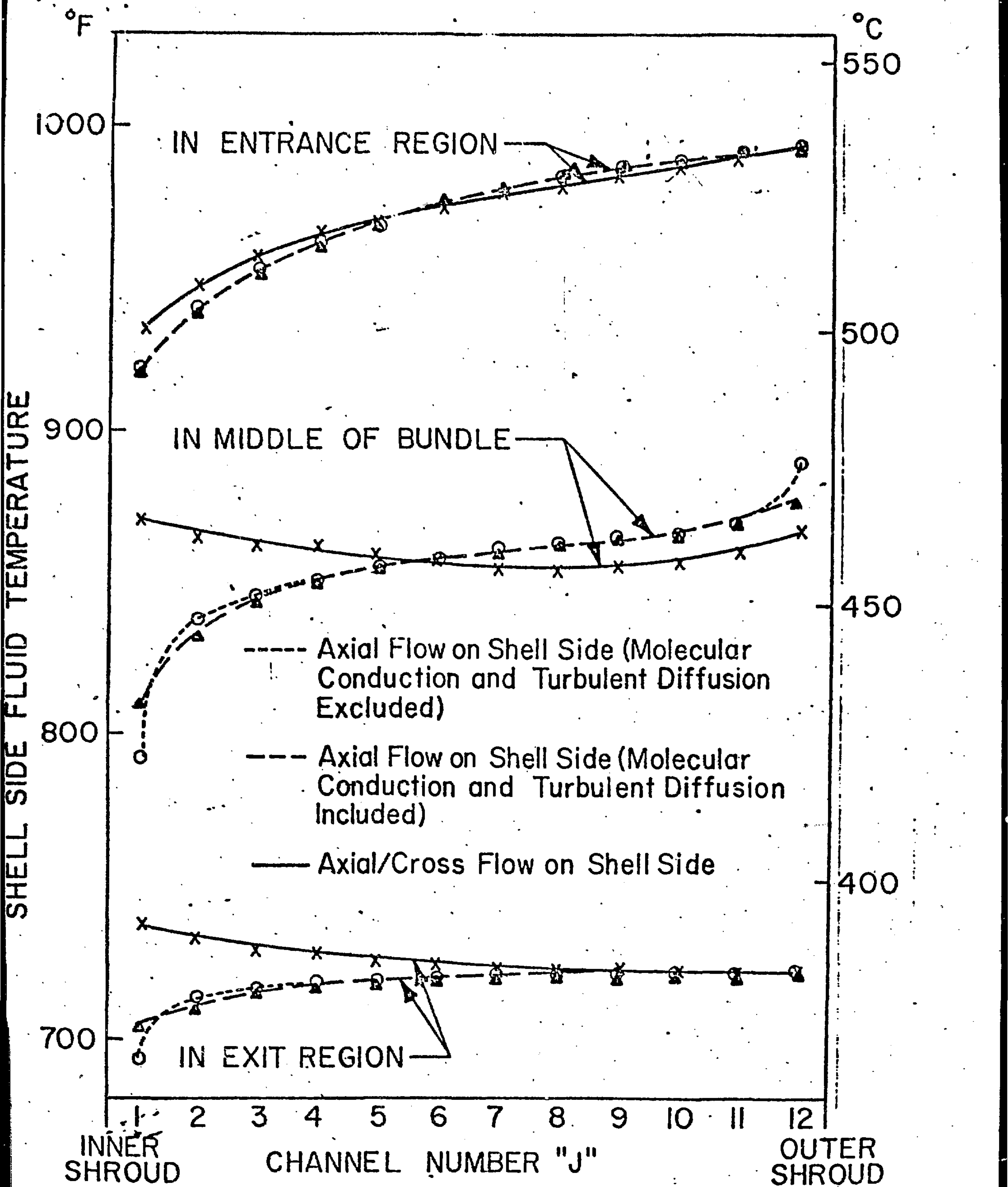
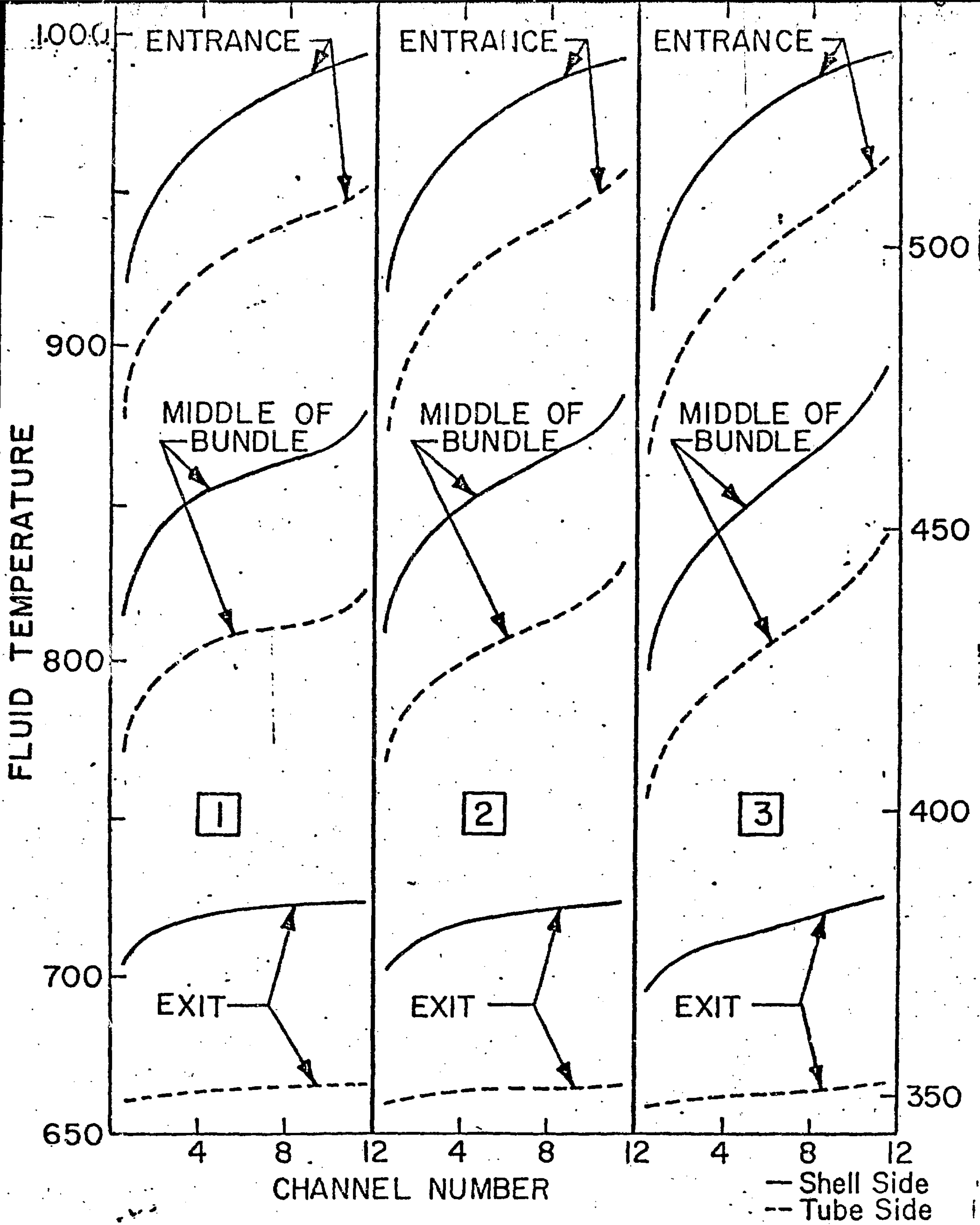
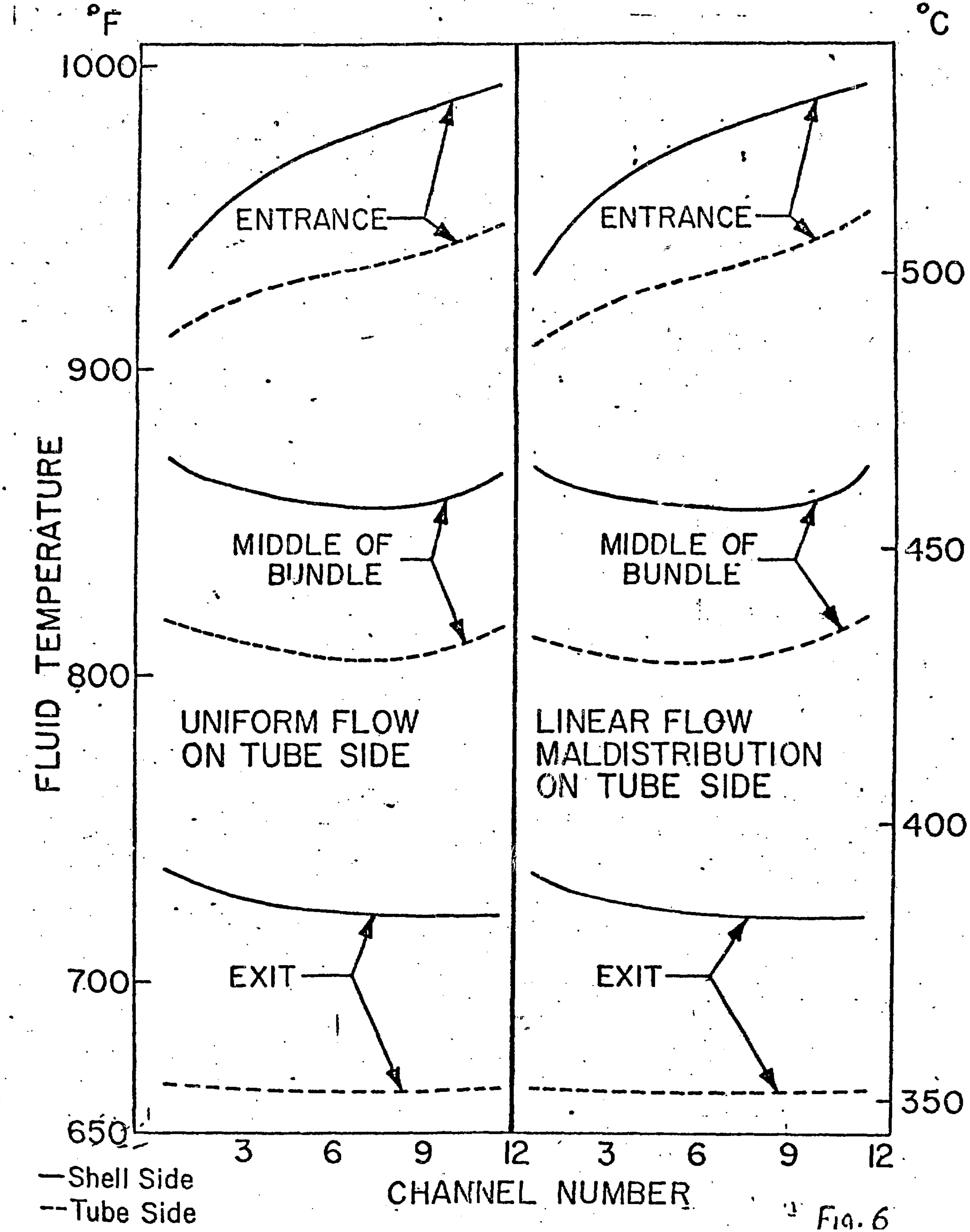

°F
°C
550
1000
IN ENTRANCE REGION
500
900
IN MIDDLE OF BUNDLE
450
SHELL SIDE FLUID TEMPERATURE
Axial Flow on Shell Side (Molecular Conduction and Turbulent Diffusion Excluded)
800
Axial Flow on Shell Side (Molecular Conduction and Turbulent Diffusion Included)
Axial/Cross Flow on Shell Side
400
700
IN EXIT REGION
1
2
3
4
5
6
7
8
9
10
11
12
INNER SHROUD
CHANNEL NUMBER "J"
OUTER SHROUD

Fig. 4



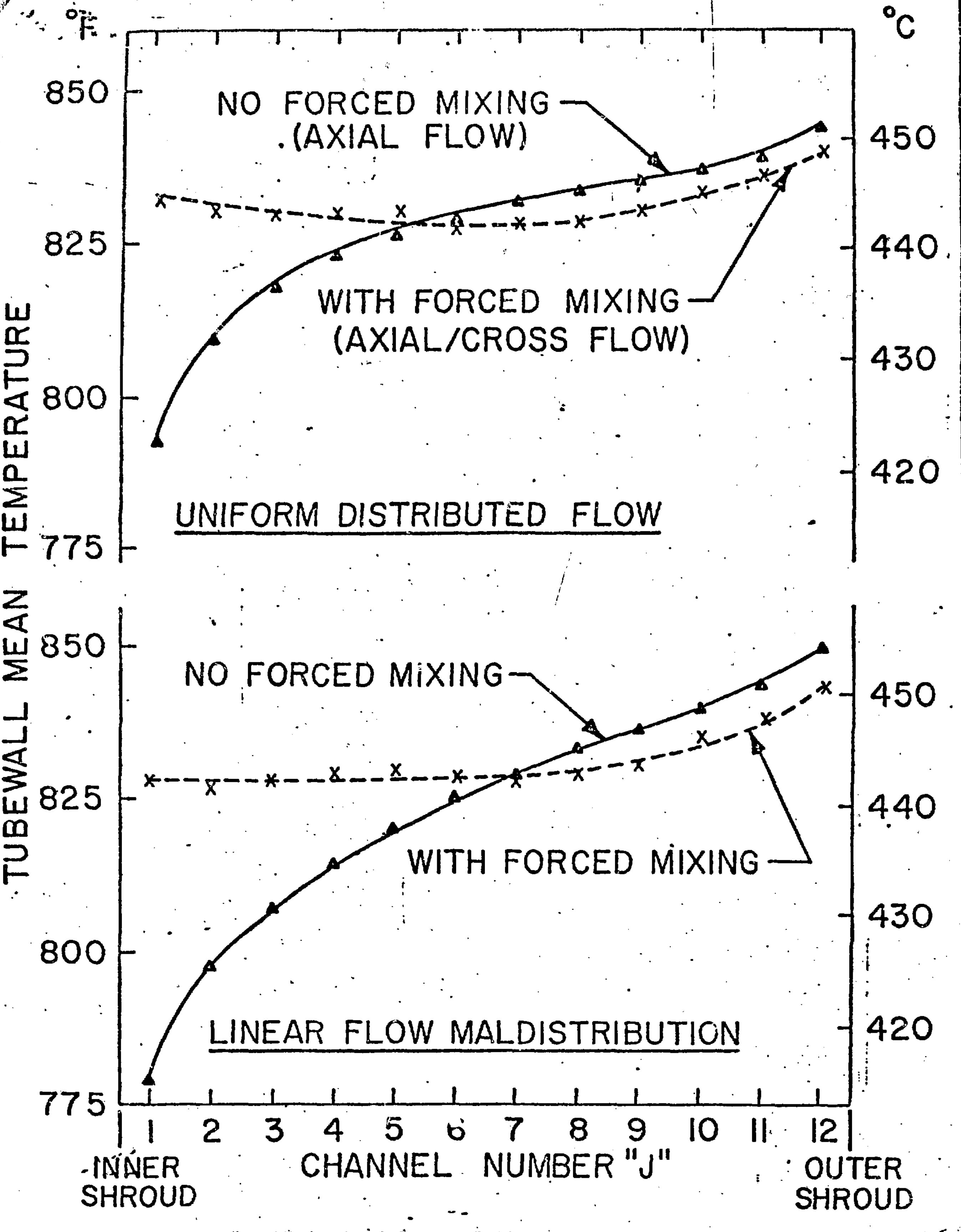
1 Uniform Flow on Tube and Shell Sides
2 Linear Flow Maldistribution on Tube Side
3 Linear Flow Maldistribution on Tube and Shell Sides

Fig. 5



°F
°C
1000
900
800
700
650
500
450
400
350
FLUID TEMPERATURE
ENTRANCE
MIDDLE OF BUNDLE
EXIT
UNIFORM FLOW ON TUBE SIDE
LINEAR FLOW MALDISTRIBUTION ON TUBE SIDE
3
6
9
12
CHANNEL NUMBER
—Shell Side
--Tube Side

Fig. 6



°F
°C
TUBEWALL MEAN TEMPERATURE
850
825
800
775
450
440
430
420
NO FORCED MIXING
(AXIAL FLOW)
WITH FORCED MIXING
(AXIAL/CROSS FLOW)
UNIFORM DISTRIBUTED FLOW
NO FORCED MIXING
WITH FORCED MIXING
LINEAR FLOW MALDISTRIBUTION
1 2 3 4 5 6 7 8 9 10 11 12
INNER SHROUD
CHANNEL NUMBER "J"
OUTER SHROUD

Fig. 7