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TMI-2 ACCIDENT: POSTULATED HEAT TRANSFER MECHANISMS AND AVAILABLE DATA BASE

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ABSTRACT

In light of the TMI-2 nuclear reactor accident, transient and small LOCA events have been identified as areas of some of the most urgent research needs in light water reactor safety. Of particular interest is the development of the capability to simulate a wide range of postulated transient and accident conditions in order to gain insight into measures that can be taken to improve reactor safety. Advanced computer codes are needed for analyzing a variety of transient and small LOCA events which may lead to a partial uncovering of the reactor core. Of special importance is the cooling of a severely damaged core and a knowledge about the primary coolant system when operating under transient and natural circulation conditions. This is necessary for examining various failure conditions in order to investigate aspects of plant design and safety system operation which may require further regulatory attention.

Computer codes which have or are being developed for predicting the thermal-hydraulic behavior of a reactor system require good understanding of the heat transfer and fluid friction characteristics. However, the existence of the data base for the description of post-CHF heat transfer especially at very low flow rates and/or natural circulation conditions is uncertain. For this reason, the open literature has been critically reviewed to determine the available data base. Special attention was focused on low coolant flow and/or high core temperature conditions. Combined (forced and natural) convection, natural convection, combined convection and radiation, and natural convection circulation in the presence of an inert gas are reviewed. The understanding of the basic heat transfer and fluid friction processes which are essential for realistic modeling of the thermal-hydraulics of a light water nuclear reactor under transient or accident (e.g. TMI-2) conditions are discussed. A number of critical problem areas where the needed data base does not exist or is inadequate are identified.

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LIST OF SYMBOLS

c_p	Specific heat at constant pressure
D	Tube diameter
D_e	Equivalent (hydraulic) diameter
f	Friction factor
G	Mass flux
g	Gravitational constant
Gr_L	Grashof number, $g\beta\Delta TL^3/\nu^2$
Gr_x	Local Grashof number, $g\beta\Delta Tx^3/\nu^2$
Gz	Graetz number, $\rho u_m(\pi D^2/4)c_p/kL$
h	Heat transfer coefficient
k	Thermal conductivity
L	System length
Nu_L	Nusselt number, hL/k
Nu_x	Local Nusselt number, hx/k
Pr	Prandtl number, $\mu c_p/k$
p	Pressure
q	Heat flux
Ra	Rayleigh number, $Gr \cdot Pr$
Re	Reynolds number, $\rho u_m D/\mu$
u	Axial velocity
T	Temperature
α	Thermal diffusivity
β	Thermal expansion coefficient
ν	Kinematic viscosity
μ	Dynamic viscosity
σ	Stefan-Boltzmann constant

<u>Subscripts</u>

b	Refers to bulk conditions
f	Refers to film conditions
L	Refers to total length L as a characteristic dimension
m	Refers to mean (average) conditions
x	Refers to local value

1. INTRODUCTION

1.1 Background

The safe operating record of the nuclear power plants all over the world to date is indicative of a built-in conservation in the design and operation of these power plants. From licensing considerations the reactor design is required to satisfy the safety requirements arising out of "Transients" that are classified in four categories. These are [1]:

1. Normal Operation and Operational Transients:

   These include start-up, hot and cold shut-downs, and normal refueling phenomena. Provisions for these are covered by margins in the design.

2. Faults of Moderate Frequency:

   Examples of such faults include misalignment in control rods, etc., which are also covered by design margins, or at best, require a temporary shut down.

3. Infrequent Faults:

   Faults in this category include small loss of reactor coolant, minor secondary system pipe break, etc. Such transients are safeguarded against by provisions in the cooling system. At the most a few fuel rods may need to be replaced but without warranting a long shut down.

4. Loss of Coolant Accident (LOCA):

   LOCA is the most severe form of transients that a reactor design has been required to cope with. The thermal-hydraulic phenomena related to LOCA have been postulated, and to date, form the basis of nuclear reactor safety analysis.

In light of the TMI-2 nuclear reactor accident, transient and small LOCA events have been identified as areas of some of the most urgent research needs in light water reactor safety. Of particular interest is the development of the capability to simulate a wide range of postulated transient and accident conditions in order to gain insight into measures that can be taken to improve reactor safety. Some of the more specific research needs include:

- Improvement of "best estimate" transient and small LOCA thermal hydraulics codes.
- Development of models to predict the cooling of partly uncovered cores.
- Development of models to predict the cooling of severely damaged cores.
- Development of models for predicting the primary coolant system behavior under natural circulation (including the effects of noncondensable gases), transient, and small LOCA conditions.

Advanced computer codes are needed for analyzing a variety of transient and small LOCA events which may lead to a partial uncovering of the reactor core. Of special importance is the cooling of a severely damaged core and a basic knowledge about the behavior of the core and the primary coolant system when operating under transient and natural circulation conditions. This is necessaary for examining various failure conditions in order to investigate aspects of plant design and safety system operation which may require further regulatory attention.

Predicting the surface temperature of a fuel element of a PWR core during a transient or a LOCA is a difficult problem since it requires the simultaneous prediction of the local fluid conditions (which depend on the local heat flux) and the local heat flux (which depends on the local fluid conditions). Computer codes which have been developed for predicting core temperature [2 - 4] require a good knowledge of the heat transfer and fluid friction characteristics. Numerous assumptions are made in constructing the models (such as the use of correlations which may be inapplicable to describe heat transfer) because of the lack of sufficient data base. The data base requiring particularly close scrutiny is the one which is used for calculating the CHF (e.g. DNB for PWR, dryout for BWR or core uncovery during LOCA), particularly at transient conditions. Also, the data base for the description of post-CHF heat transfer especially at very low flow rates or under natural convection conditions appears to be inadequate.

The primary objective of this review is twofold:

1. Identify the fundamental heat transfer data base essential for constructing computer codes required for the purposes mentioned above.
2. Suggest areas of basic heat transfer studies directly relevant to water reactor safety that are needed to fill the gap in the data base.

However, before proceeding with these tasks, it is thought desirable to recount the postulations of LOCA with relative details. It would be also seem that the accident at Three Mile Island Unit No. 2 was different from a postulated LOCA although the consequential damages have been equally severe.

1.2 LOCA

In the design condition heat transfer from the fuel rods to the primary coolant is maintained in single phase (liquid) turbulent regime. The coolant (water) in a PWR is pressurized to a subcooled state where the outside temperature of the fuel rod is below the corresponding saturation temperature. Typically, (e.g. TMI) the operating pressure of the primary coolant may be 2200 psia (T_{sat} = 649°F), whereas the cladding temperature of the fuel rod is 600°F.

From a heat transfer point of view the lesser degrees of severity in the first three categories of transients imply that the loss of coolant shall not be to the extent that there shall be a departure from nucleate boiling condition (from single phase turbulent). Sufficiently high heat transfer rate under nucleate boiling condition shall not allow permanent damage to the fuel rods.

On the contrary, loss of primary coolant in a LOCA is expected to be severe (causing departure from nucleate boiling) and thermally induced damage to the fuel rods can be expected due to inadequate heat transfer. However, the cladding temperature should not be allowed to exceed 2000°F since the generation of a significant amount of hydrogen (by reaction between zircaloy and steam) could take place [5]. This upper limit is supposed to be contained by different (emergency) core cooling systems that the manufacturers are required to comply with.

Configurations of the core and the primary coolant circuit are indicated in Figure 1. A LOCA may be caused by a large rupture in either the hot or cold leg. The loss of coolant in excess of supply by primary feed pumps will result in loss of pressure and subcooling, leading to steam formation. The sequence of events during a LOCA and the emergency core cooling process are indicated in Figure 2.

It would have been appropriate to discuss at this stage the heat transfer regimes associated with a LOCA. We, however, prefer to defer this until a brief review of the TMI accident is made. Indeed, a LOCA is a postulation, whereas the TMI incident was reality.

1.3 TMI Incident

The sequence of events at TMI Unit No. 2 originated with the failure of main feed water (secondary) pumps for the steam generator [6]. With the reduction in secondary flow rate, the heat transfer from primary coolant was reduced,

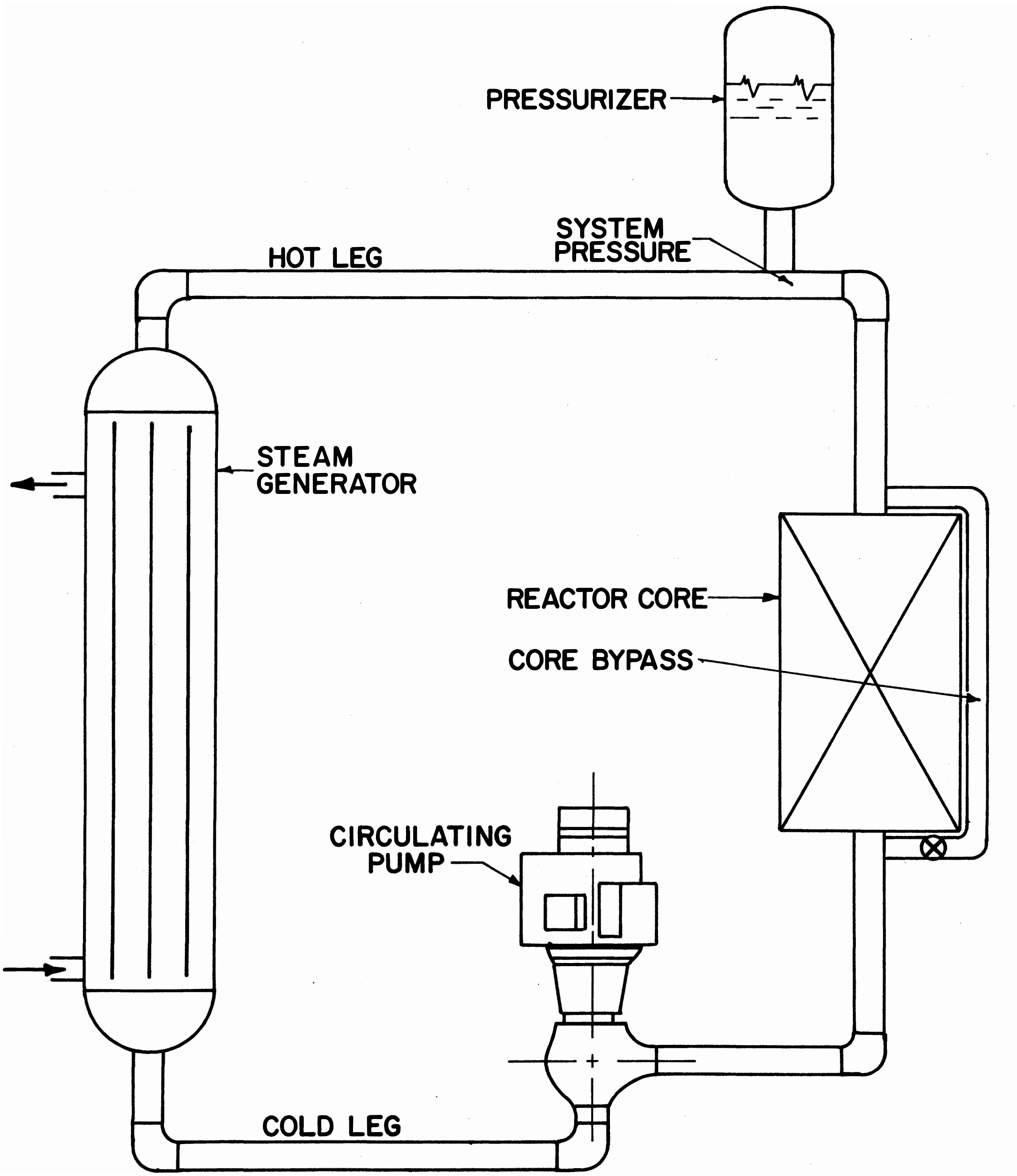


Figure 1. CADDS loop model

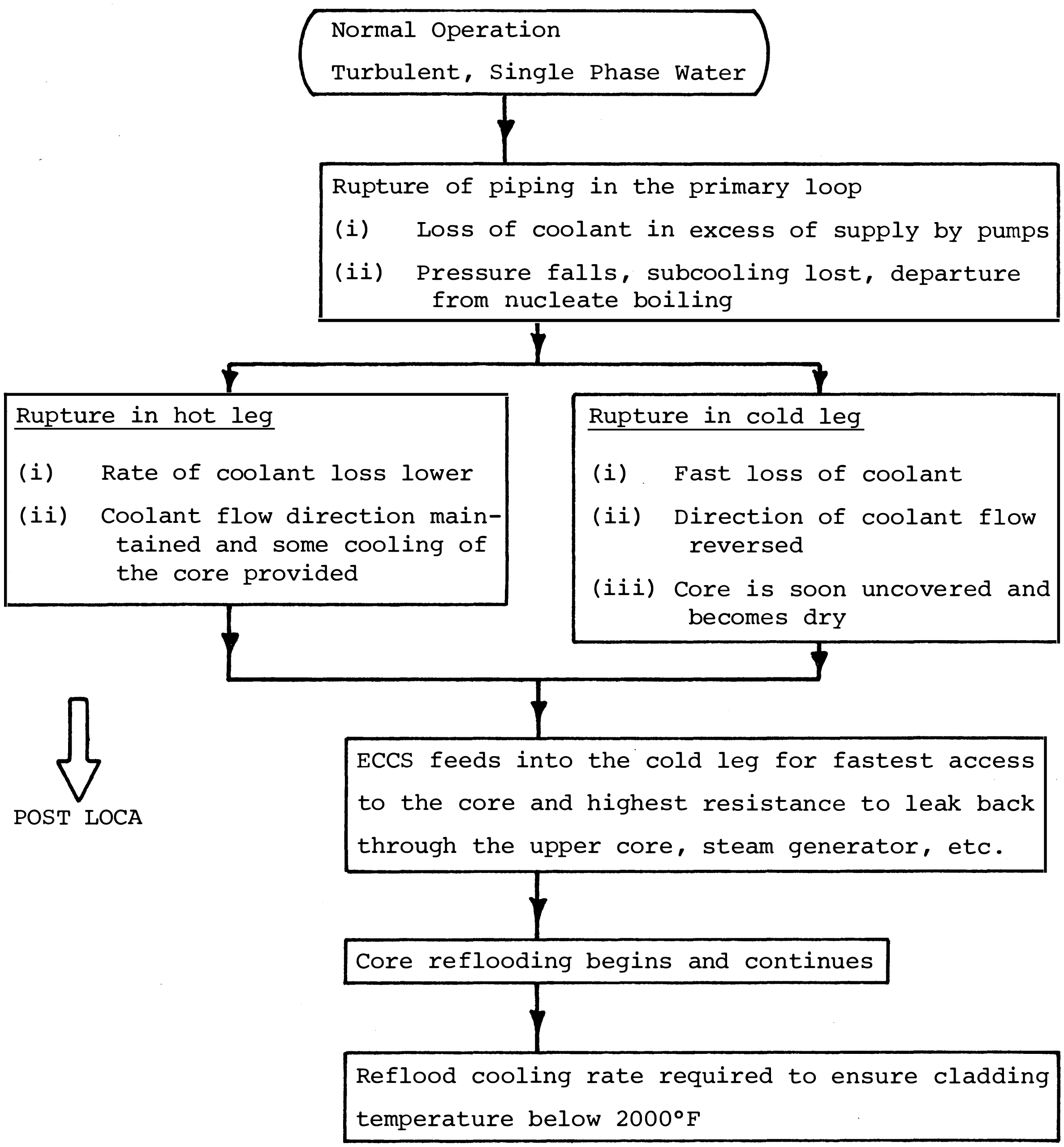


Figure 2. Postulated sequence of incidents in a LOCA.

and as a result the fuel rod temperature increased causing loss of subcooling. Increased specific volume of liquid and possible steam formation resulted in reactor pressure rise during the first 8 seconds (Figure 3), with the reactor generating full power. The reactor tripped due to over-pressurization.

Although in principle, the decreasing pressure phenomenon after 8 seconds at TMI is similar to a postulated LOCA, the details differ. In particular, it was not a loss of primary coolant that triggered the accident, as is postulated in a LOCA. The nonavailability of auxiliary feed water in the steam generator due to the inadvertently closed valves is the primary reason for the progress of the accident in the early phase. Later, it is the nonclosure of the Power Operated Relief Valve (PORV) of the Pressurizer (that continued the loss of system pressure leading to the formation of steam and blockage of the primary coolant loops by a steam-water mixture) that augmented the system failure.

1.4 Heat Transfer Mechanisms

We intend to postulate in the following section the mechanisms and modes of heat transfer in the reactor system that control the events like the one that happened at TMI and those of a LOCA.

1.4.1 Initial Pressurization (TMI)

Before the failure of the secondary water system (steam generator), the reactor was operating at the design condition for which the rod to coolant heat transfer is estimated to be in turbulent flow regime of a subcooled liquid. With the loss of secondary fluid, it is the rise in primary coolant temperature that resulted in decreased heat removal from the fuel rods; the flow rate (primary coolant), however, was maintained.

It is noted from steam tables that a fluid temperature rise of 30°F in the vicinity of 650°F (corresponding to saturation around 2200 psia) causes an

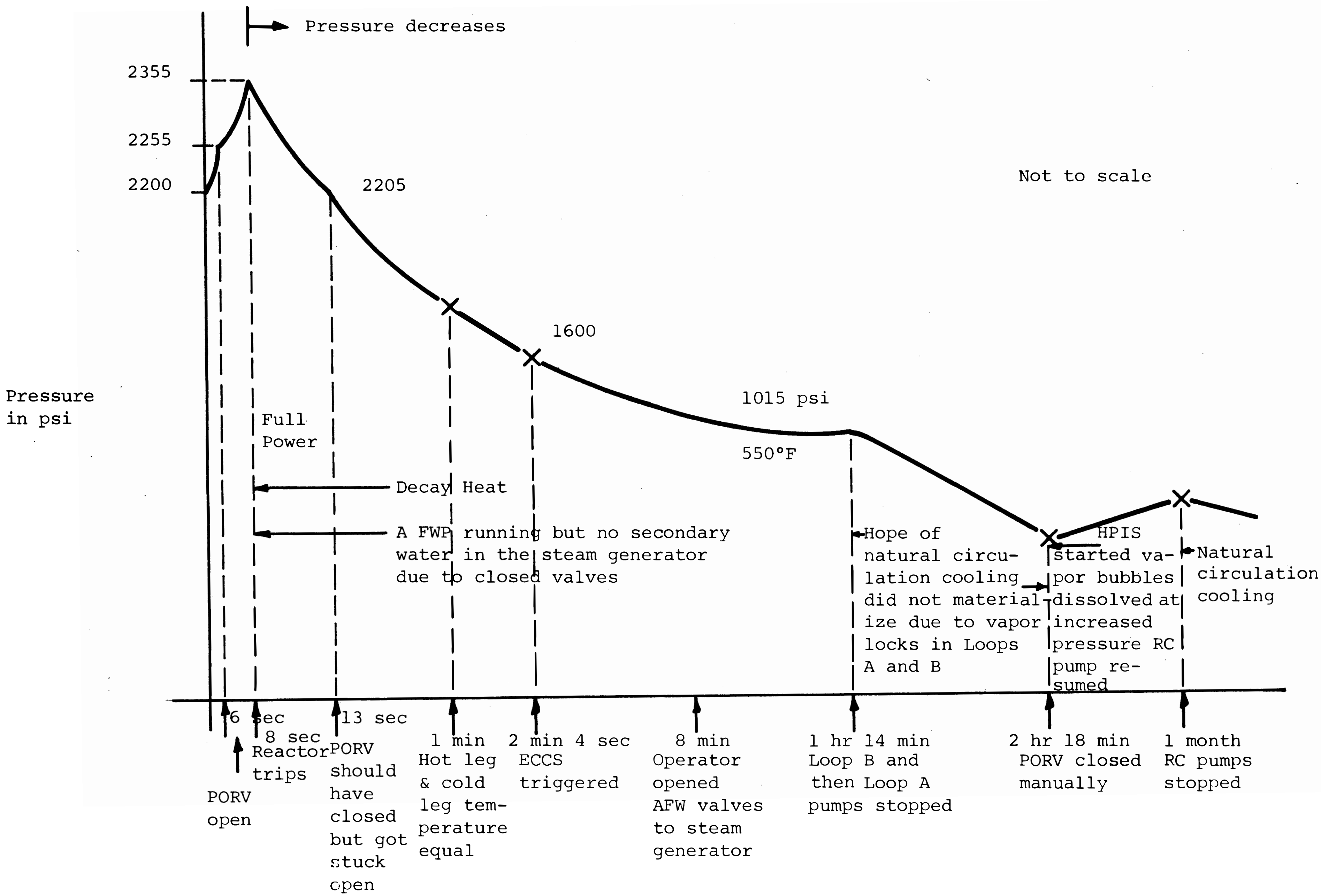


Figure 3. Sequence of incidents at TMI, Unit 2.

increase of 10 percent in the specific volume of water. This increase in specific volume can contribute significantly to the observed pressure rise. From a heat transfer point of view two effects need to be considered:

1. Buoyancy induced convection, associated with implied change in specified volume. Such natural convection is superimposed on the prevalent forced convection.
2. Variation in thermophysical properties, the reference temperature for their evaluations, and incorporation of such in the heat transfer correlations used.

In writing down the above two effects, it is presumed that the quantity of steam formed during this period was not significant, due to the preceding degree of subcooling, increasing pressure and short time elapsed. Experimental studies are needed to verify this presumption.

At full power the heat flux at the fuel rod is extremely high. The thermal response, and temperature-time history of a fuel rod at such high flux conditions, to postulated variations in heat transfer coefficient, are necessary for a system evaluation.

1.4.2 Stuck-Open PORV and Primary Coolant Pumps Running (TMI); Hot-Leg LOCA

In a LOCA due to a rupture in the hot-leg of the primary coolant circuit, the coolant flow direction and some degree of core cooling are maintained. The coolant loss is less compared to a cold-leg rupture. In the TMI accident similar events took place due to the stuck-open PORV.

With the reduction of coolant flow rate, buoyancy effects become more prominant. Until this time (1 hr., 14 min. into the accident) the main pumps for primary coolant were running, though under a higher back pressure due to steam blockage in the suction side. It is thus likely that most of the water within the core was in the liquid phase. There could have been some steam formation in the top portion, but this is not likely to have been a large quantity since "steam-binding" was not strong enough to block leakage through the PORV.

The above leads one to hypothesize that the heat transfer mechanisms were as indicated below:

1. Buoyancy influenced low Reynolds number convective heat transfer to water in the greater part (lower) of the core (now decaying thermally).
2. Heat transfer to steam-water mixture in the upper part of the core.
3. Reduced quantity of steam-water mixture (partial loss due to high leakage in the PORV) moving up through the hot leg and evaporating due to absorption of residual heat from the piping. Heat transfer to this steam-water mixture is by combined forced and free convection.
4. Partial loss of latent heat of primary coolant by conduction to stagnant secondary water in the steam generator. The hypothesis that the heat loss from the primary coolant at this stage and zone was in the form of latent heat is based on the observation that after 1 minute into the accident the hot-leg and cold-leg temperatures were equal.

After the ECCS was triggered, the coolant inventory in the core would have improved and the rate of steam formation decreased.

1.4.3 Stuck-Open PORV, Primary Coolant Pumps Stopped (TMI)

The main coolant pumps in loops A and B were stopped after 1 hr and 14 min due to vapor-lock and excessive vibration. With only partial coolant supply through HPIS and higher loss through stuck-open PORV, the system pressure decreased rapidly, much of the core uncovered and heat removal was primarily by steam generation. The core probably sustained the greatest damage during this period. The release of volatile fission products and generation of H_2 by the reaction of steam with the zirconium cladding indicates that the cladding temperature could have exceeded 2000°F.

The state of the coolant at different levels of the core and the mode of heat transfer are postulated in Figure 4.

A significant fraction of water supplied by the HPIS is likely to be evaporated at the free surface by absorption of fuel rod decay-heat (including net radiation absorption due to exchange between fuel elements and the free surface), assisted by reduced pressure due to leakage through the stuck-open PORV. The net rise of water level in the core thus depends on the difference between HPIS injection rate and mass lost through PORV. The steam formed inside the reactor vessel may also have exerted a pressure on the water front, as in the case of reflooding following a LOCA (steam-binding).

If the water level rise was fast enough, it is likely that the difference between the rod surface temperature and saturation temperature would not be too high, and nucleate boiling with net evaporation might have taken place. The thermal zones in such a situation are thus examined:

1. Zone I in Figure 4 indicates the region of high heat transfer rates due to nucleate boiling. Below the free surface, liquid water would be gaining heat by convection (at low Reynolds number) from the fuel rods, at a rate proportional to the HPIS injection flow rate. The free surface would also receive heat by radiation from hotter portion of the fuel rods. Two pressure related phenomena may take place in Zone I:
   (i) Due to conversion of water to steam in a limited space (gap between a set of rods), i.e. due to sudden and large increase in specific volume (from water to steam), there may be a sharp decrease in pressure. There is also a possibility that choked flow of the steam-water mixture may occur in this zone.
   (ii) The second phenomenon is what we generally call the buoyant effect. Steam formed at the interface may be heated due to

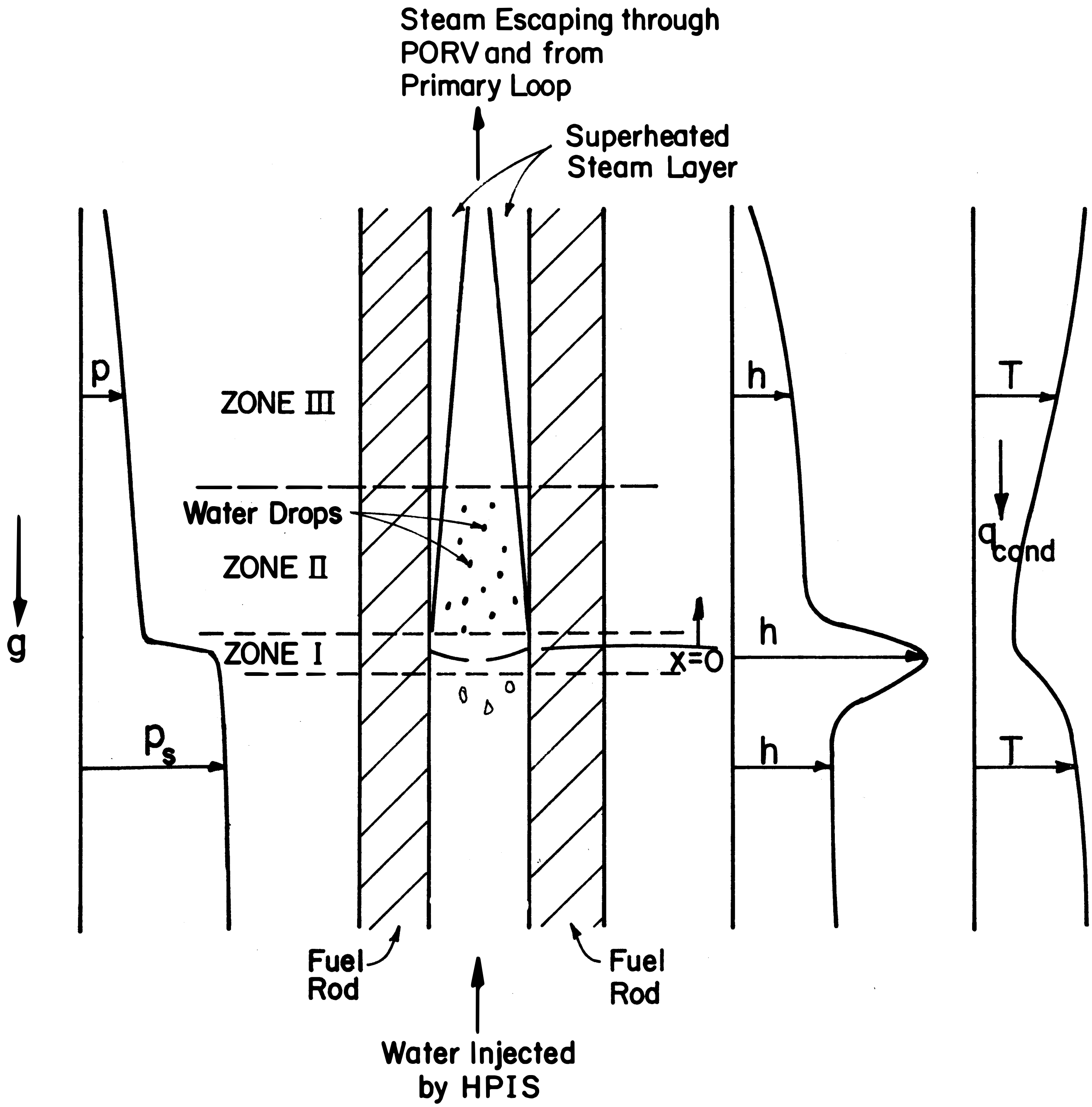


Figure 4. Thermal zones in a partially uncovered core.

contact with the fuel rod resulting in temperature induced buoyancy force. In other words, at $x = 0$ there is a sudden acceleration brought about by the change in specific volume. For $x > 0$ the driving force is that due to buoyancy. Both conservation of mass and energy have to be considered under the high heat flux condition at the liquid-vapor interface, Zone I.

2. Steam emanating from the interface may tend to be superheated in layers close to the rod surfaces. The core may still contain some water droplets. In Zone II heat transfer is expected to be buoyancy dominated with dispersed flow. Some of the liquid drops may occasionally contact the rod surface causing localized and temporary increase of heat transfer.

3. In the region above Zone II, the steam layer may be considerably superheated. Heat transfer is considered to be by simultaneous radiation and combined convection. Radiation exchange between rods, rods and steam, rods and vapor bubbles of Zones I and II, and between rods and the liquid free surface may take place; the relative magnitude depending on the respective "view-factors".

Heat transfer rates in Zone I are expected to decrease with height, and the rod temperature may exceed the limit for onset of steam-zirconium reaction, in which case noncondensible gases such as H_2 and He may be released in the form of local bubbles. From a natural convection point of view, such bubbles could provide additional resistance to heat flow. Furthermore, at high temperatures, the fuel rods may undergo thermal expansions causing local distortions of the flow path. This will be reflected in residence time and heat transfer rate to the escaping steam.

1.4.4 PORV Closed, All pumps Stopped

When the decay heat became comparable (after about a month) to the frictional heat generated by the circulation pumps, the pumps were stopped and the core left to cool by natural circulation. By this time the central portion of the core had been considerably (practically 100%) blocked [7] and the coolant circulation is considered to take place through this peripheral zone. The heat transfer mechanisms between the rods and the coolant are considered to be as indicated in subsection 1.4.3.

There can be temperature difference between rods located at the central region and those at the periphery due to:

(i) Cooling of peripheral rods by the naturally circulating coolant; the central portion is blocked.

(ii) Power density of peripheral rods may have been lower to start with.

(iii) Peripheral rods can radiate heat to the reactor vessel, apart from loss to the coolant.

The effect of such temperature difference would cause heat exchange in a radial direction by radiation between rods, and by conduction from blocked central core to the peripheral rods through debris and water film. It is necessary to evaluate the buoyancy force required to overcome the different resistances offered by blockages and friction in the coolant loop.

1.5 Scope of This Review

In the preceding subsection we have postulated the heat transfer mechanisms that come into play during an accident arising out of disruptions in the primary coolant flow (as in LOCA) and initiated by a high temperature rise of the primary coolant due to failure of the secondary feedwater system (as was the case at TMI).

We may briefly recall these mechanisms as:

(a) Basic Modes of Heat Transfer

1. Forced convection turbulent heat transfer to water.
2. Low Reynolds number flow combined with buoyancy in water.
3. Nucleate boiling and departure from it.
4. Water-vapor interface heat and mass transfer in presence of high heat flux.
5. Buoyancy induced heat transfer to a vapor with an initial acceleration due to sudden change in specific volume.
6. Combined free and forced convective heat transfer to a dispersed mixture of vapor and bubbles.
7. Free convection and radiation exchange between rods and steam, with and without noncondensible gas bubbles.
8. Radiation exchange between rods.

These above modes of heat transfer have to be evaluated under the following geometrical influences.

(b) Geometical Influences

1. Rods in vertical position arranged in rectangular, square and triangular arrays.
2. Passages blocked by buckled and eccentric rods.
3. Blocked central core with flow passage between reactor shell and peripheral rods.

The heat flux along a fuel rod is determined from nuclear considerations. For example, the flux may be cosinusoidal during normal operation; becoming almost constant during decay following reactor trip. During an accident, however, the rod will experience time-varying heat transfer rate due to different states of the fluid. This will result in axial temperature gradients in any given rod.

Furthermore, since the primary coolant is expected always to flow in a closed loop, whether in normal operation or during an accident, the thermal and hydraulic characteristics of the system would control the core cooling rate.

Two system effects which also influence the core cooling rate are:

(c) System Effects

1. Effects of temperature gradient along the length of a rod.
2. Buoyancy force vis-a-vis resistance to circulation of a two phase fluid, either in single or differing phase or in simultaneous two phase conditions, through different limbs and components of the coolant circuit.

2. RELEVANT DATA BASE FOR LIGHT WATER REACTOR CORE HEAT TRANSFER

Having postulated the heat transfer mechanisms that are probable to control the safety of a pressurized water nuclear reactor, we now proceed to evaluate the literature on such heat transfer modes.

2.1 Forced Convection

The emphasis of the review was on combined (natural and forced) and on natural convection and not on forced convection heat transfer in a PWR core under normal operating conditions. Therefore, the following subsection is included only for the sake of completeness and is not intended to be a thorough review. References will be cited in the report where additional literature sources concerned with forced convection heat transfer to water, steam and boiling of water can be found.

2.1.1 Water

The basic design condition of a PWR is generally based on fully developed turbulent flow of water through a pipe using the Dittus-Boelter equation,

$$Nu_b = 0.023\ Re_b^{0.8} Pr_b^{0.4}, \tag{1}$$

The physical properties are evaluated at the fluid bulk temperature T_b. The other versions of the above correlations are the Colburn j-factor,

$$j = \left(\frac{h}{c_p G}\right) Pr_f^{2/3} = 0.023\ Re_f^{-0.2} \tag{2}$$

with the properties evaluated at film temperature $T_f = (T_w + T_b)/2$ and the Sieder-Tate equation

$$Nu_f = 0.0027 Re^{0.8} Pr^{0.33} \left(\frac{\mu_b}{\mu_w}\right)^{0.14} \tag{3}$$

It is important that some of the BEESTimate heat transfer codes [2] also use the Sieder-Tate relationship. Although all of the above three equations predict nearly equal heat transfer rate through a circular pipe at $Pr \simeq 1$ and moderate temperature differences, it is known that the results vary by ± 30 to 40% when Pr is different from unity [7, p. 129].

The Dittus-Boelter equation overpredicts heat transfer coefficients for flow of air through an isosceles triangular duct [8, p. 7-122, Fig. 104]. At Re = 10,000 the Nusselt number predicted by the Dittus-Boelter equation is twice the measured value.

Experimental results and correlations for turbulent forced convection through rod clusters are given in Table 38 of the Heat Transfer Handbook [8]. It is seen that only in a seven rod cluster delta array, pitch to diameter P/D ratio of 1.015, Pr = 0.7 and Reynolds numbers between 10,000 and 60,000, is the heat transfer coefficient predicted by Dittus-Boelter equation. In general, the rod cluster results depend upon P/D ratio, Pr, Re and the arrangement; the reported variation is of the order of 30% from the values calculated from the Dittus-Boelter equation. At low Reynolds number in water flow and closely spaced rods ($P/D \simeq 1$) the heat transfer coefficient is less than 50% of value predicted by Eq. (1) [Subbotin et al., see Ref. 8, p. 7-134]. Nusselt number increases with increasing P/D ratio.

The available rod cluster results do not contain information about the effects of grid spacers, heat flux variation along the rod, bowing of the rods in the course of use, etc.

Practically no experimental information exists for heat transfer and fluid friction in the entrance region of a rod cluster. Information for simple geometries, such as a tube, is also meager. Heat transfer in the entrance region depends on the type of entry, L/D ratio, and separate or simultaneous development of hydrodynamic and thermal buoyancy layers, apart from the other factors that control the fully developed heat transfer.

Some experimental results for pipe flow are correlated as [8, pp. 7-36]:

$$\frac{Nu_m}{Nu_\infty} = 1 + (\frac{C}{4D}) \tag{4}$$

where Nu_m is the mean Nusselt number over the length L and Nu_∞ is the fully developed value. The coefficient C depends upon the factors mentioned.

In general, it is known that the entrance length is influenced by the obstructions and geometrical considerations at entry. An understanding of the effects of spacer grids, wire wrap, turbulent level, departure from validity of hydraulic diameter concept, on heat transfer in the entrance region of a rod cluster is needed.

The rod assembly under the influence of fluid flow could be subjected to flow induced vibrations. It is well known that vibrations often enhance convective heat transfer. The magnitude of such influence on rod cluster has to be known so that its reduction at low flow rates, under accident condition, can be estimated.

Detailed discussion of forced convection heat transfer in ducts [8-10] and of design correlations used [11, 12] can be found in the literature.

2.1.2 Steam-Water Mixture

Heat transfer to the two-phase steam-water mixture related to nuclear reactor applications is possibly one of the most intensely studied topics. Bjornand and Griffith [6] have recently discussed the scheme and correlations used for the two phase regime in the BEEST code. Comparative results of different correlations and effects of upflow and downflow are summarized. Basically, the code predicts an estimated heat flux value given the (i) wall temperature, (ii) mass flux, (iii) pressure and (iv) void ratio of the fluid. It is, however, not clear as to how one would postulate the values of the input variables under an accident condition. Apparently these have to be evaluated from possible (though limited) measurements and analysis of the system dynamics.

Transition boiling data for water under forced convection conditions is given in Reference 10. Attention is also drawn to the degree of dependability of such correlations.

The correlations used for two-phase forced convection are generally derived from experiments with tube flow. It is necessary that such correlations be reviewed with regard to their applications for the rod-cluster geometry.

It is reported [14] that critical heat fluxes in rod cluster arrangement varies considerably with the design of grid-spacers on their pitch along the rod. More detailed studies on these aspects are necessary.

Fundamentals and design correlations for boiling and two-phase flow heat transfer are available in textbooks [15, 16]. Up-to-date critical reviews of heat transfer to steam-water mixtures have been prepared [17-20] and need not be repeated here.

2.1.3 Steam and Gases

Forced convection heat transfer data from gases, including steam, are typically presented [11] by correlations which are valid over a limited Prandtl number range, $0.3 < Pr < 1.5$. So far as the superheated steam is concerned, treatment as a simple gas is adequate. However, heat transfer coefficients for moist and nearly saturated steam could be different. Unfortunately, not very much information is available for such conditions.

For superheated steam flowing through a smooth tube, the following correlation is suggested by Babcock & Wilcox [21, pp. 12-6]:

$$h = \frac{0.0266}{D^{0.2}} G^{0.8}(c_p \mu^{0.2}) \tag{5}$$

where G is the mass flow rate per unit area. Written in dimensionless form Eq. (5) becomes

$$Nu = \frac{hD}{k} = 0.0266\ Re^{0.8} Pr \tag{6}$$

The effect of physical property variations, in terms of the factor $c_p \mu^{0.2}$, with pressure and temperature is incorporated through a correction chart. For example, at 100 psi and 600°F, the value of h is 27% of the value predicted by Eq. (5).

Groeneveld [22] has summarized the available experimental results and of his own measurements for a seven rod cluster inside a horizontal shell. The rod array is finite grid triangular, and the minimum heat transfer coefficient in superheated steam (for P/D = 1.052) is given by

$$Nu_f = \frac{hD_e}{k_f} = 0.0078\ Re_f^{0.8774}\ Pr_f^{0.6112}\ (L/D_e)^{-0.0328} \tag{7}$$

He postulates that at the minimum heat transfer rate, giving rise to maximum temperature, bowing of the rods is to be expected. The effect of rod bowing is to reduce the heat transfer rate by 40%, estimated conservatively. The L/D_e term takes into account the effect of the entrance region.

Larsen et al. [23] have carried out both analytical and experimental investigations of heat transfer in a 12.7 mm ID tube to steam and nitrogen. While nitrogen does not participate in radiative transfer, both convective and radiative transport of heat from wall to steam takes place. The following range of parameters were covered for nitrogen:

Wall temperature, T_w: 260 to 1040°C

Bulk temperature, T_b: 27 to 150°C

Reynolds number, Re_b: 2,000 to 10,000

and for water vapor:

Wall temperature, T_w: 200 to 1100°C

Bulk temperature, T_b: 150 to 595°C

Reynolds number, Re_s: 420 to 33,000

The heat transfer results for nitrogen attributed to convection alone were correlated by

$$Nu = 0.021\ Re_b^{0.8} Pr_b^{0.4} (T_w/T_b)^{-0.5} [1 + (L/D)^{-0.7}] \quad (8)$$

For estimating the radiative component of heat transfer within water vapor, the convective contribution, Eq. (8), was subtracted from the total. Thus, the radiative component was estimated to be 5 to 22% of the total. However, the total heat transfer was well estimated by the Dittus-Boelter equation, Eq. (1), without regard to radiative-convective interaction. One would, therefore, suspect, conversely, that at lower temperatures, where radiation effects are negligible, Eq. (1) shall overpredict the actual heat transfer.

More recently, Chiba and Greif [24] have reported that in heat transfer to superheated steam in turbulent flow through a 2 inch tube, the experimental results are 20% higher at a wall temperature of 2650°F than that predicted by a pure convection model. The convection contribution is apparently estimated using the Dittus-Boelter equation. The findings of Chiba and Greif are in disagreement with those of Larsen et al. in the sense that the former predict higher (total) heat transfer than the latter even when the wall temperature should indicate the reverse.

Sutherland and Kays [25] carried out analytical and experimental investigations of heat transfer in triangular rod array with air at Reynolds number range from 7000 to 200,000. For P/D = 1, rods touching, they corroborate the trend of Subbotin et al., for water (predicting heat transfer coefficients smaller than the D-B equation). The Nusselt number determined from the D-B correlation is higher by 30% for P/D = 1.15 and 1.25. Friction factors differ from Blasius pipe results by the same order and similar magnitude as the Nusselt number.

Furber et al. [26] present experimental results for carbon dioxide flowing through annuli of radius ratio 1.47 and 1.96 in the Reynolds number range of 1×10^5 to 6×10^5. Both entrance region and assymptotic fully developed region results are presented for wall temperature up to 1000°C with $T_w/T_b = 3$. While the entrance region data are useful for the reactor safety analysis, we cite here the assymptotic fully developed values, for comparison with other data:

$$Nu = 0.024\ Re^{0.8} Pr^{0.4} \quad \text{for} \quad D_1/D_2 = 1.47$$

and (9)

$$Nu = 0.026\ Re^{0.8} Pr^{0.4} \quad \text{for} \quad D_1/D_2 = 1.96$$

The authors indicate that T_w/T_b corrections to property values are not necessary at $Re > 10^5$. An apparent exclusion of radiation heat transfer has been made in arriving at Eq. (9). Although the geometry was vertical, no mention is made of free convection effects, implying that at such flow rates the Gr number dependence is not strong.

The Nusselt numbers predicted by the different correlations mentioned above are summarized in Table 1 for Pr = 0.7 and $Re = 10^4$ for the sake of comparison.

Clearly, the Babcock and Wilcox value is too conservative when $c_p\mu^{0.2}$ differs from 1. More important is the fact that Nusselt numbers for rod arrays could be different from pipe results. Secondly, the values for finite [22] and infinite rod arrays [25] also differ widely.

Table 1

Comparison of Nusselt Numbers for Forced Convection Heat Transfer Predicted by Different Correlations: Pr = 0.7, Re = 10,000, $T_b \simeq 300°C$

Authors	Geometry	Fluid	Pr	Re	$T_{w_{max}}$	$(T_w/T_b)_{max}$	Nu
Dittus-Boelter [7]	Pipe	Many					31.6
Babcock & Wilcox [21]	Pipe	Steam	$c_p\mu^{0.2}$=0.27	(100 psi, T_b=600°F)			7.97
Groeneveld [22]	7 rod delta finite array P/D = 1.052	Steam					20.27
Larsen et al. [23]	Pipe	N_2	0.68-0.70	2000-10,000	1040°C	2	28.76
		Steam	0.87-1.09	420-33,000	1100°C	2	31.6
Chiba & Greif [24]	Pipe	Steam (adding 20% to convection value)	0.61-0.68 (calculated backward)	15,300 and 22,100	650°F	≃1.3	38
Sutherland & Kays [25]	6 rod-delta infinite array	Air	0.7	7000 to 200,000			40 P/D=1.15 & 1.25 15 P/D=1.05
Furber et al. [26]	Annulus	CO_2		10^5 to 6×10^5	1000°C	3	33 D_1/D_2=1.47 35.72 D_1/D_2=1.96

In summary, forced convective heat transfer for particular rod arrangement of interest should be determined; specificially for the operating and accident conditions flow rate and temperature.

2.2 Combined Forced and Natural Convection

At the accident condition, as we have already stated, the flow rate is expected to be reduced to an extent that buoyancy effects can control to a significant degree the heat removal from the rods. Buoyancy, as it will be seen from the following discussions, has opposite effects on heat transfer for laminar and for turbulent flow and whether the natural convection is aiding or opposing the flow. Under certain thermal boundary conditions, buoyancy also causes considerable delay in establishing fully developed conditions, at least in laminar flow. Because of opposing influence on laminar and turbulent flows, we propose to arrange the available information on the basis of flow regimes, rather than on fluid conditions hitherto adopted.

2.2.1 Internal Flow with Buoyancy

2.2.1.1 Laminar Flow

Combined natural and forced convection heat transfer in single vertical tubes has been studied [27-31] and found that stability of flow depends primarily on the shape of the velocity and only secondary on the value of the Reynolds number. For upflow heating the flow first becomes unstable when the velocity profile develops points of inflection. Transition to an unsteady flow involves the gradual growth of small disturbances, and therefore it is quite possible to have unstable flows without observing transition. For downflow heating the flow instability is associated with separation at the wall. Transition to an unsteady flow is sudden and therefore occurs shortly after an unstable flow occurs. Over the range of Re of 80 to 4,800 transition occurs for values of Gr/Re of 78 to 56 for downflow.

Experimental results [31] show marked differences between the upflow and downflow Nusselt numbers measured in a uniformly heated vertical pipe. For upflow heating Nu is larger than what would exist for steady parabolic (laminar) flow. However, except for the data at low Reynolds numbers the Nusselt numbers are not too different from those predicted for distorted laminar flow. For downflow heating the measured values of Nusselt numbers are much greater than those predicted for distorted laminar flow. Increases in Gr/Re and Re cause Nu to increase; however, the effect of Re appears to be much greater for downflow than for upflow heating.

A detailed experimental study on combined free and forced convection heat transfer with water in up and downflow through a vertical tube was reported by Hallman [27]. The tube wall was at uniform heat flux condition and the primary emphasis was on the entrance region. Although it has been aptly shown that heat transfer is increased in upflow and decreased in downflow, compared to pure forced convection, the questions of predicting the variation of heat transfer with Gr or GrPr/Re, entry length and transition to turbulence have not been answered precisely.

Lawrence and Chato [32] studied analytically, and verified salient results experimentally, upward developing flow through a vertical tube; taking into account nonlinear variation of density and viscosity of water. For a constant heat flux boundary condition, velocity and temperature profiles were not developed even at $x/D \simeq 170$, $Re = 210$ and $Gr/Re = 54.3$. The heat transfer rate was higher than for forced convection entry region results, and only by allowing for nonlinear property variation, could their analysis corroborate their experimental results.

Zeldin and Schmidt [33] carried out numerical solutions for developing laminar combined convection constant property upflow through a vertical tube for an isothermal wall boundary condition. Uniform and fully developed (parabolic) entry velocity profiles were considered. It is seen from numerical results and as corroborated by experimental data for air that the convex (parabolic) and uniform velocity profiles develop a small distance away from the walls as the flow proceeds downstream; the profiles, however, lose the concavity further downstream, finally becoming fully developed at $x/r_o = Re \cdot Pr$; for the experiments performed this is at $x/D = 126$. For $Gr/Re = 30$, the heat transfer is marginally higher than forced convection value. The more important finding of the study is that the velocity gradient at the wall is increased due to buoyancy effect. This would mean higher fluid friction, which is not discussed. Temperature profiles and reasons for increase (though marginal) of Nu are also not discussed.

Recent numerical and experimental study of Greif [34] in air and argon, flowing upward in a vertical tube with an imposed constant heat flux, indicates fully developed condition at $x/D = 108$, and the laminar heat transfer results are 16% higher than the pure forced convection, $Nu = 4.364$. In view of Greif's finding of fully developed conditions, Lawrence and Chato [32] conclusion that the flow is nondeveloped can only be attributed to physical property variation of water. Greif's temperature profile data and his own comment that buoyancy causes a flatter temperature profile, however, does not explain the increase in heat transfer for laminar flow. By scrutinizing his figures, it is seen that a larger value of Nu could be due primarily to a higher bulk temperature which is influenced by change in velocity profile that has not been discussed in the paper. The 16% increase for $Gr/Re \simeq 0.038$ seems too high compared to the results of Zeldin and Schmidt [33] (granting that the latter results were for a different thermal boundary condition). Greif's findings are also higher compared to the data reported by Maitra and Subba Raju [35], who, for fully developed upward flow of water through an annulus, found that the threshold value for increase in Nu was $Ra = 10^3$ (corresponding to $Gr \cdot Pr/Re = 0.8$).

Classically, the laminar combined convection through a vertical tube used to be calculated from the Martinelli-Boelter relationship [7, p. 233],

$$Nu = 1.75 \, F_1 [Gz \pm 0.0722 \, (\frac{D}{L} Gr \cdot Pr)^{0.75} \, F_2]^{1/3} \tag{10}$$

The positive and negative signs in Eq. (10) are respectively for vertical upflow and downflow. So far as the vertical upflow is concerned, the results [32 -34] qualitatively through not quantitatively corroborate Martinnelli-Boelter correlations.

Recently, however, the experimental data of Mullin and Gerhard [36] tend to contradict Eq. (10), indicating that in downflow of water through a heated isothermal tube the heat transfer coefficient was slightly higher than the upflow. As a compromise, they suggest that the positive sign should be used for both up and downflows, making no difference between the two.

Kemeny and Somers [37] attempted to measure the variation of friction factor in laminar and in combined upward flow of water and oil through a vertical tube with a uniform heat flux boundary condition. From pressure drop measurements they indicate an increase in the laminar friction factor of about two to three fold over the Poiseulle result. The flow was in the entrance region, yet the authors do not account for momentum variation in the measured pressure gradient. As such, the implied variation in friction factor cannot be accepted quantitatively.

Heat transfer in a uniformly heated annulus both in buoyancy assisted and opposed flow has been studied by Sherwin [38]. The analytical results showed that Nu increased with Gr/Re for upflow and decreased with downflow. For a uniformly heated inner surface the experimental data for downflow shoed that for $Gr/Re > 150$ there was an increase in Nusselt number and thus indicate that flow reversal has occurred near the heated surface and the flow is unsteady.

Shervin]39] had reported the effect of natural convection on laminar flow heat transfer in a system consisting of three parallel, vertical, annular channels, each having the same nominal size and uniform heat input at the inner surface. The heat transfer results were found to have a wide spread about the average, but the average itself showed only small variation over a range of Gr/Re tested. The results of the investigation show that single channel results can be misleading when applied to a multi-channel system, especially when natural convection effects are present in comparatively low speed flows.

The above discussions of available results on laminr combined forced and free convection heat transfer point conclusively to the fact that heat transfer rate is increased due to buoyancy in a vertical upward internal flow. Information on almost all other parameters are inconclusive. These are: (1) threshold value of Gr/Re, (2)extent of variation in Nu with Gr/Re, (3)explanation for change in Nu, (4)length of entrance region for different GrPr/Re, (5) value of critical Reynolds number fordifferent GrPr/Re, (6)variation in friction factor, and (7)effects on all of the above parameters for different thermal boundary conditions. Some of the salient available information is summarized in Table 2.

2.2.1.2 Turbulent Flow

While the general feature of the laminar combined convection investigations discussed in the foregoing section was in association with the entrance region, most of the available information on turbulent combined convection pertains to the fully developed condition. This is not to imply that the question of entry length in the latter regime of flow and heat transfer has been answered quantitatively. We would also like to note that direction of flow appears to have an opposing effect on heat transfer in turbulent flow compared to that in laminar flow.

Table 2

Summary of Laminar Combined Forced and Natural Convection Heat Transfer Results in Channels

Authors	Geometry Fluid	Nature of Treatment Theoretical	Nature of Treatment Experimental	Effect on Forced Convection Parameters Nu Upflow	Nu Downflow	f Upflow	f Downflow	Entrance Effects	Gr/Re Range	Thermal Boundary Conditions	Remarks
Hallman [27]	Vertical tube Water		✓	+	-				≃ 0.5	q_w=const	
Hanratty et al. [28]	Vertical tube	✓	✓							T_w=const	Flow visualization experiments only
Sheele & Hanratty [31]	Vertical tube	✓	✓	+	+			Flow & heat transfer fully developed, L/D = 762	< 700	q_w=const	
Lawrence & Chato [32]	Vertical tube	✓	✓	+ (not emphasized)				Flow & temperature not developed even at x/D ≃ 170	50-450	q_w=const	Increase in Nu value is not emphasized; reasons for increase not discussed
Zeldin & Schmidt [33]	Vertical tube Air	✓	✓	+ (marginal)				Fully developed at x/D ≃ 126	30	T_w=const	
Greif [34]	Vertical tube Air or Argon	✓	✓	+ (16%)				Fully developed at x/D = 108	0.038	q_w=const	Temperature gradient contradicts the increase in Nu, in the absence of discussion on velocity profile & rod bundle temperature
Mullin & Gerhard [36]	Vertical tube Water		✓	+	+				(GrPrD/L) ≃5.5×10^5 Re not given	T_w=const	Heat transfer rate is identical in upflow and downflow
Shervin [38]	Vertical Annulus, R_o/R_i=3	✓	✓	+				L/D_ℓ=120 calming L/D_ℓ = 96 heated	< 700	q_w=const at inner surface	
Shervin [39]	Bundle of Vertical Annuli	✓	✓	+					< 30	q_w=const at inner surface	Wide range from channel to channel

Alferov et al. [40] made heat transfer measurements in upward flow of water in a vertical tube up to $L/D \simeq 180$, beyond a 100 D unheated length required for hydrodynamic development. The pressure range was varied from 150 to 300 atm; temperature lower than saturation in sub-critical region and limited to 350°C in the supercritical region. The heat transfer rate deteriorated with free convection. In a very unorthodox manner, disregarding property variation, they quantified the heat transfer variation in a plot of $(Nu/Nu_{forced})_{minimum}$ vs. (Nu_{forced}/Nu_{free}). The pure forced and free convection Nusselt numbers were estimated using Dittus-Boelter and McAdams [8, Eq. (7-9)] equations, respectively. The experimental results were well correlated, heat transfer decreasing with increased value of (Nu_{forced}/Nu_{free}), going through a minimum equal to about 40% of the pure forced convection value at $(Nu_{forced}/Nu_{free}) = 1$ for values of this parameter higher than 2, pure forced convection results are obtained.

Herbert and Sterns [41] also observed a cup-like behavior of Nusselt number variation with Reynolds number in the range of 6,000 to 20,000 in upflow of water for Grashof numbers of the order of 10^7. The minimum was about 50% of the value predicted by Eq. (1) at Re ≃ 15,000, levelling up to about 90% at Re > 30,000. With downward flow the heat transfer rate was higher reaching to 1.4 times the result calculated from Eq. (1) at Re ≃ 6,000. The Grashof number was of the same order of magnitude. In downflow the experimental results did not exhibit cup-like behavior and asymptotically approached the Nusselt number predicted from the Dittus-Boelter equation at about Re = 20,000. Note that lower Nu numbers were predicted for downward laminar flow, on the contrary.

Salient heat transfer results are summarized in Figure 7. The Reynolds numbers are limited to the order of 10,000, so as to be relevant to the reactor accident condition. Note that while in laminar flow (Table 2) the appropriate parameter was Gr•Pr/Re, in turbulent flow the parameter is $Gr \cdot Pr/Re^2$.

Greif [34] on the other hand, found that the heat transfer rate in turbulent upward flow of air and argon at Re = 10,000 to 19,000 and Gr = 70 to 80 was practically the same as predicted by the Dittus-Boelter correlation. This would be expected, since the Gr number is low. Recall, however, that Greif observed a 16% rise in heat transfer coefficient for laminar flow.

Easby's [42] experimental results in downward flow of nitrogen through a vertical tube at low turbulent Reynolds number ($2{,}000 < Re < 10{,}000$) also indicate rise in heat transfer rate due to buoyancy effects. However, there is no quantitative agreement between Easby and Herbert and Sterns [41]. For example, at $Gr \cdot Pr/Re^2 \simeq 1$, Easby's nitrogen data is 45% higher than Herbert and Sterns [42], who at this value of $Gr \cdot Pr/Re^2$ predict nearly or slightly lower Nusselt number than the Dittus-Boelter equation. Common sense would, however, indicate that for water, Herbert and Sterns would have found higher values.

Easby [42] further found that the friction factor decreases from pure convective (McAdams) values in buoyancy affected downward flow. He recommends the following correlations for f and Nu in downward flow:

$$f/f_{iso} \simeq 1 - 5.1\ Gr/Re^2 \tag{11}$$

and

$$Nu/Nu_{D-B} = 1 + 8.9\ Gr/Re^2 \tag{12}$$

The variations of Nusselt number and friction factor predicted by the above correlations are plotted in Figures 7 and 8.

The difference between the two results [41, 42] imply that $Gr \cdot Pr/Re^2$ is not the only relevant parameter for correlating the turbulent combined convection heat transfer.

Since it has not been categorically suggested that buoyancy causes variations of different magnitude depending on the wall thermal boundary conditions (for example, the uniform heat loss or isothermal) in turbulent flow, the same can be assumed for combined convection.

Carr et al. [43] measured variation in friction factor for upward flow of air in a vertical tube with Gr varying from 1.11 to 2.54×10^4 while the Reynolds number was around 5,000 for most of the data. The friction factor at first decreases with Gr (Figure 8) going through a minimum nearly equal to 86% of McAdams' value for pure forced convection and rising again to higher than pure convection result. In the light of Easby's [30] finding for downward flow, the increase in friction factor with $Gr \cdot Pr/Re^2$ for $(Gr \cdot Pr/Re^2) > 0.00025$ can be considered as complementary. But this would mean that the decreasing friction factor for small $Gr \cdot Pr/Re^2$ is contradictory or is a different phenomenon.

Connor and Carr [44] have summarized combined convection heat transfer results in air, water and mercury in upward flow through a vertical tube in the following manner:

$$\frac{Nu}{Nu_T} = 8.84 \left[\frac{Gr_A}{Re^2}\right]^{0.263} \tag{13}$$

where $\quad Nu_T = 0.021\ Re^{0.8} Pr^{0.4} \quad$ and

$$Gr_A = \frac{Gr \cdot Nu}{4Pr \cdot Re}$$

For a nominal value of Re = 10,000 and Pr = 0.7, the parameter $Gr \cdot Pr/Re^2$ becomes

$$\frac{GrPr}{Re^2} = 678.25 \left(\frac{Nu_T}{Nu}\right) \left(\frac{Gr_A}{Re^2}\right) \tag{14}$$

Using Eqs. (13) and (14) some heat transfer results have been calculated for the range of validity of Connor-Carr correlations, as indicated in their paper [44]. These results are presented in Figure 7. Clearly, Connor and Carr [44] predict similar value of Nusselt number at $GrPr/Re^2$ that are order of magnitude away from Herbert and Sterns [41] results. It may be of interest to mention that although Connor and Carr have referred to Herbert and Sterns paper [41], the latter results were not correlated in [44].

Axcell and Hall [45] have reported mixed convection turbulent heat transfer to air flowing down a heated pipe. The results are as before higher than predicted by the Dittus-Boelter equation. Their correlation

$$\frac{Nu}{Nu_D} = [1 + 4500\ GrRe^{-21/8} Pr^{-1/2}]^{0.31} \qquad (15)$$

seems to corroborate the trend of experimental results, but quantitatively underpredicts by about a factor of two.

Convective heat transfer is controlled by the distributions of velocity and temperature. In the presence of buoyancy the two profiles are coupled. While discussing laminar results [34] it was indicated that variation in velocity profile, rather than of temperature could qualitatively explain the change in heat transfer rate. A similar explanation would be applicable for turbulent flow [40, 44, 45]. If this were to be viewed in the light of the Reynolds analogy for momentum and heat transfer, an increasing heat transfer should be associated with increasing friction factor. However, this is contradicted by Easby's [42] measurements (Figures 7 and 8) for downflow.

A reduction in pressure loss in a downward flow could be explained by treating buoyancy such as to introduce an adverse pressure gradient. While this could explain Easby's [42] friction results, it would fail for Carr et al.'s [43] decreasing f observation in upward flow (Figure 8).

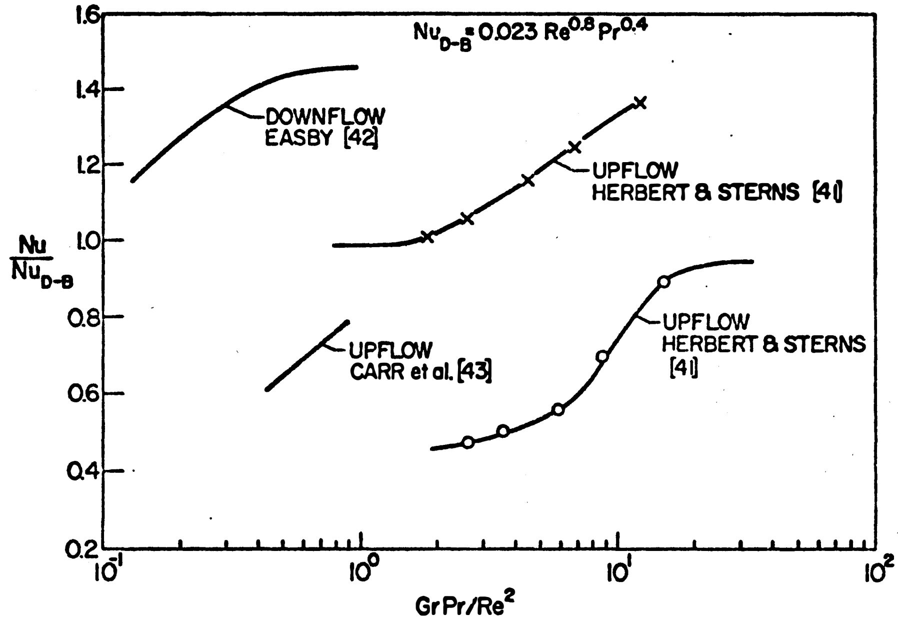


Figure 5. Comparison of heat transfer correlations for combined convection in a vertical pipe.

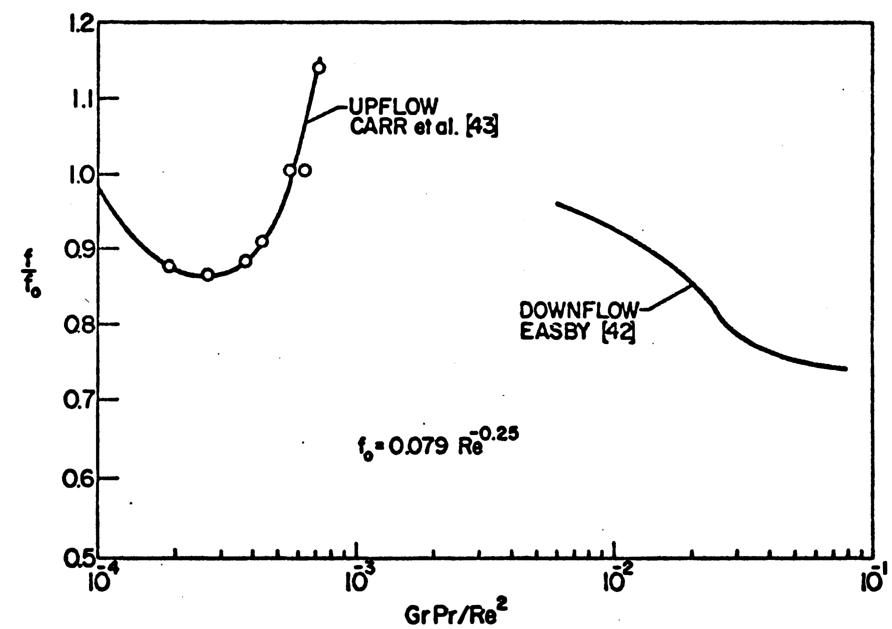


Figure 6. Comparison of friction factor correlations for combined convection in a vertical pipe.

Hall [46] postulates that small buoyancy force causes a thickening of laminar sublayer near the wall, causing thereby an area averaged reduction in turbulent intensity in a duct. This could explain reduction in turbulent transport. With increasing buoyancy effect the wall shear could become zero and change sign. According to Hall, further increase in buoyancy would increase the turbulence level. This would mean a return to higher level of transport rates.

Hall's hypothesis implies a "cup" type plot of heat transfer and friction factor at the same $Gr \cdot Pr/Re^2$, with buoyancy effect. Heat transfer [40, 41] and friction factor [43] results considered independently, could support Hall's postulation. But taken together as in Easby's [42], or consideration of flow directions, or for a range of $Gr \cdot Pr/Re^2$, the postulation is not satisfactory.

2.2.2 Combined Natural and Forced Convection in Rod Bundles

There have been fewer works relating to the geometry of present interest, namely the rod cluster. Almost all of these studies are numerical, and have not been corroborated by experiments.

Igbal et al. [47] considered laminar fully developed, combined convection heat transfer through triangular and square arrays for upward flow, with uniform wall heat flux and uniform wall temperature boundary conditions. The numerical results indicate that an increase in heat transfer over the forced convection value is noticeable at Ra > 100. With increasing P/D ratio the buoyancy influence is reduced, so is the difference for the two thermal boundary conditions and for the square and triangular arrangements.

In a later paper Igbal et al. [48] report numerical results for the conjugate problem of combined convective flow through rectangular ducts of different aspect ratios. The wall heat flux is constant, whereas the temperature variation is estimated using the coupled conduction (along the duct) and convection (in the fluid) boundary condition. The threshold Rayleigh number is, as before for the rod cluster, around 100. Large free convection effect seems to dampen the wall temperature variation.

A numerical study by Ramm and Johannsen [49] of combined convection in vertical hexagonal rod bundle is useful, for it demonstrates that the velocity and temperature profiles can be strongly influenced by a variation of heat flux at different rods in a given assembly. At a given radial variation of heat flux inner to outer radius ("Power Skew"), increasing the Rayleigh number (from 10^{-3} to 5) causes an inversion of the velocity profile (see Figure 5, Power Skew 1.1) in the rod subchannel. At a given Ra, increasing the "Power Skew" changes the velocity profile in the same direction. Buoyancy effect causes a flattening of the temperature profile; increasing the "Power Skew" increases temperature gradient (Figure 5).

Ramm and Johannsen observe that at certain "Power Skew" and buoyancy conditions, the rod surface temperature at some radial and angular locations is lower than the bulk temperature of the fluid. This would mean that heat could flow from fluid to rod; a phenomenon termed as negative Nusselt number [50]. From such observations, Ramm and Johannsen voice a fundamental question as to whether the concept of Nusselt number should be used to estimate heat transfer in a real fuel bundle. In general, upflow causes higher heat transfer than forced convection; downflow results in lower values.

Yang [50] performed a numerical study of fully developed upflow through rod clusters in square and triangular arrangements and combined convection. The threshold Grashof number is suggested to be 100. In both arrangements the Nusselt number increased monotically with increase of P/D ratio and of Gr/Re for a given spacing to diameter ratio. The pressure drop (friction factor), however, shows an interesting variation, particularly for the triangular array. At low Gr/Re values, pressure drop continuously decreases with increased spacing P/D. When Gr/Re is increased, at first pressure loss decreases with P/D, goes through a minimum and then again increases. Apparently, the velocity profile at higher P/D ratios undergoes inversion with increasing Gr/Re. Figures 3 and 7 of Yang are reproduced as Figure 8 of this report for illustration. Recall that Ramm

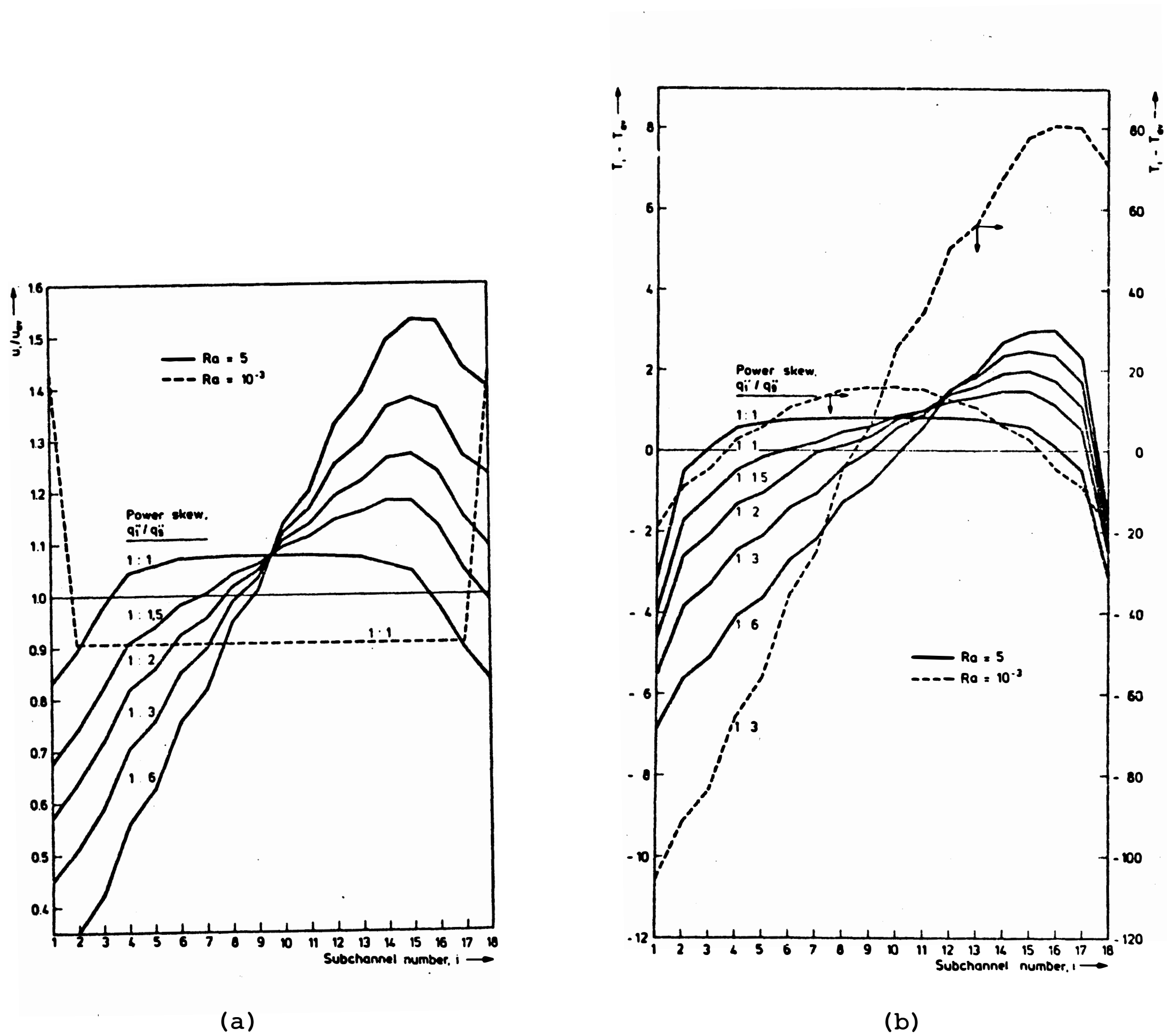


Figure 7. Effect of power skew and Rayleigh number for 61-rod subassembly velocity (a) and temperature (b) distributions: P/D = W/R = 1.077 (From Reference 49).

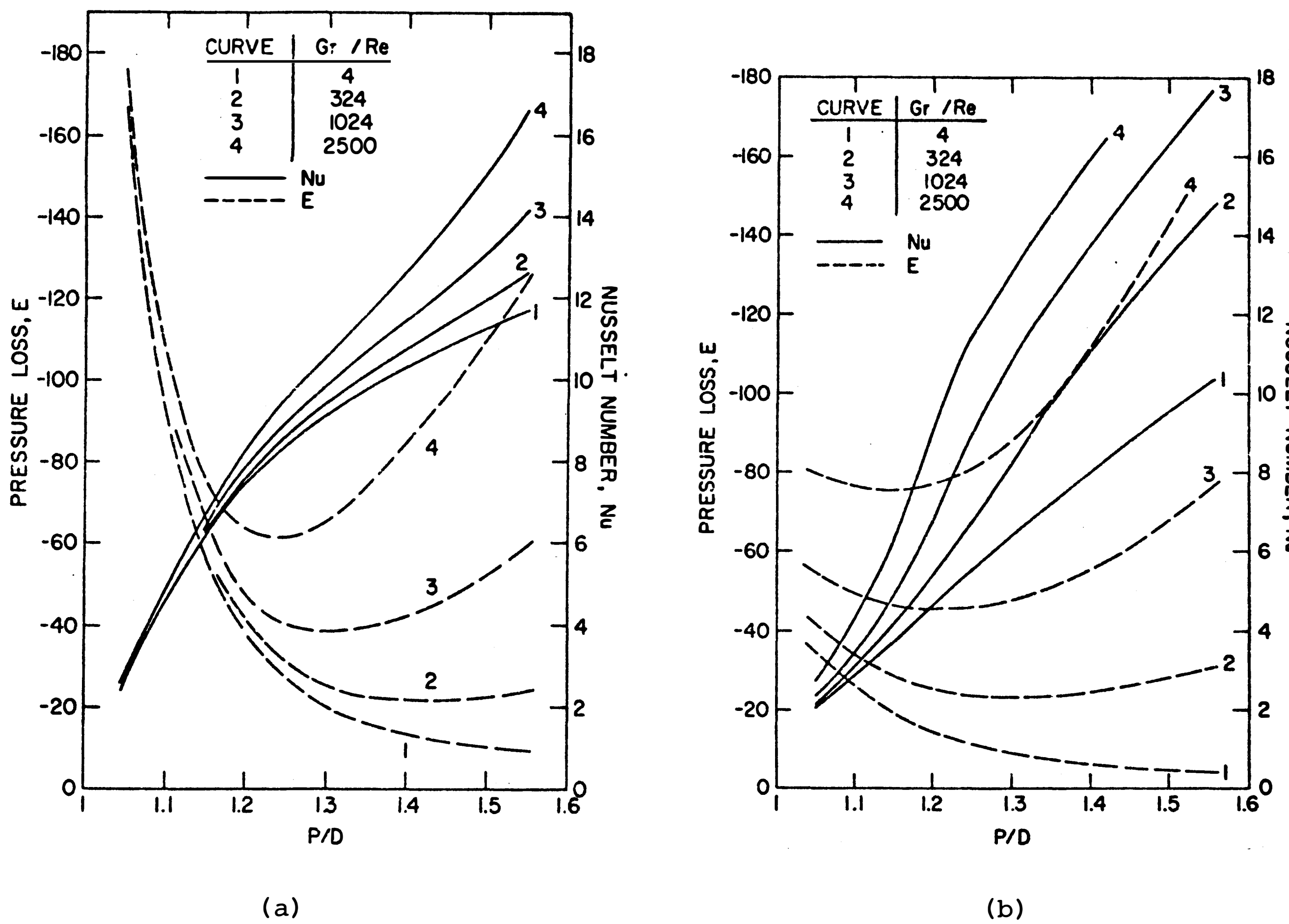


Figure 8. Predicted Nusselt numbers and pressure losses for combined convection in infinite rod arrays: (a) triangular array and (b) square array (From Reference 50).

and Johannsen [49] also spoke of inversion of velocity profile with Rayleigh number, and predicted inversion even at P/D = 1.077.

In a later paper Yang [51] analyzed downward combined convective flow and heat transfer through square and triangular arrays. Again, the threshold value of Gr/Re was about 100. With increase in buoyancy, both heat transfer and pressure loss were found to decrease. The influence of buoyancy was to reduce wall sheer stress, finally making it zero and leading to separation.

Before concluding this section, we would like to refer to a recent experimental study [52] on combined convection of water through a 2 $\times$ 6 rectangular rod bundle in the upward flow. The maximum Reynolds and Grashof numbers were in the range of 1000 and 4×10^6, respectively, and the radial power variation from rod to rod was at a maximum of 2:1. The results indicate that velocity and temperature profiles were considerably influenced by both "Power Skew" and buoyancy effects and suggest that the buoyancy effect is immediately noticeable if the power distribution is slightly different from uniform. Although the COBRA code [53] predicts the trend in variation of temperature and velocity profiles, quantitatively disagreement to the extent of 40 percent exists. No information in terms of overall friction factor or Nusselt number are given in the paper.

2.2.3 Combined Free and Forced Convection in External Flow

Combined free and forced convection heat transfer in external flow over vertical plates and cylinders has received some attention. Most of the work is analytical with the buoyancy force either assisting or opposing the main flow. Since external flow results are not directly applicable to nuclear reactor core cooling under low flow conditions, only attention is brought to some of the more recent studies for the sake of background and are not discussed in detail.

Combined forced and free convection from an isothermal or uniformly heated vertical plate has been studied analytically [54-57] and an extensive review is available [58]. Churchill [58] has correlated the local Nusselt number by an empirical equation of the form,

$$\left[\frac{Nu_x f_F(Pr)}{A_F Re_x^{1/2} Pr^{1/3}}\right]^3 = 1 + \left[\frac{A_N Ra_x^{1/4} f_F(Pr)}{A_F Re_x^{1/2} Pr^{1/3} f_N(Pr)}\right]^3 \tag{16}$$

with

$$f_F(Pr) = [1 + (C_F/Pr)^{2/3}]^{1/4} \tag{16a}$$

$$f_N(Pr) = [1 + (C_N/Pr)^{9/16}]^{4/9} \tag{16b}$$

The constants A_F, A_N, C_F and C_N are all tabulated by Churchill. The experimental data for vertical plates scatter widely but rather randomly about Eq. (16). Petrova et al. [59] have obtained the following empirical equation for the average Nusselt number,

$$Nu_L/Re_L^{1/2} = 0.68[(0.952 + Pr)^{1/2}(Gr_L/Re_L^2)^{1/2} + Pr]^{1/2} \tag{17}$$

with the height of the plate L taken as the characteristic dimension. Although Eq. (17) is suggested for a Prandtl number of approximately unity, there is no indication about the range of validity of the equation.

Contrary to the good agreement between analytical and experimental results for the heat transfer coefficients, the results for the velocity profile show considerable discrepancy. A comparison of velocity and temperature profiles reported by Gryzagoridis [60] is given in Figure 9. The agreement between predictions and data is excellent for $Gr_x/Re_x^2 < 0.5$, but at larger values of Gr_x/Re_x^2 the data consistently tend to fall below the computed curves [57]. Regardless of the discrepancy between data and predictions, the available results show that as $Gr_x/Re_x^{1/2}$ increases there occurs a significant

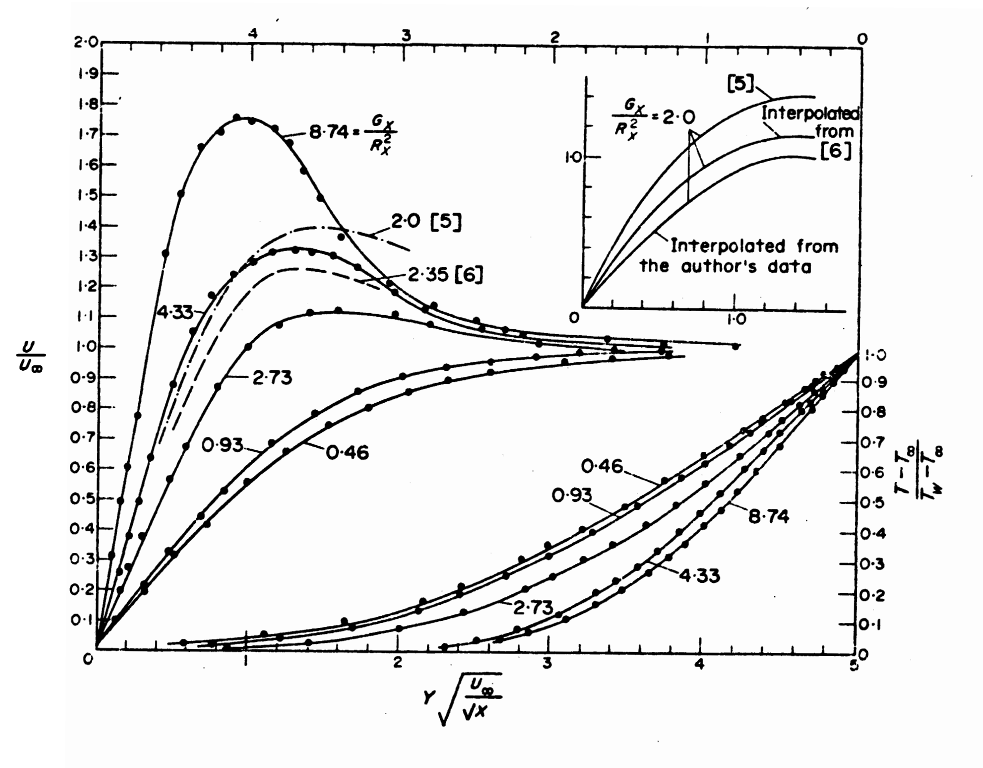


Figure 9. Temperature and velocity profiles from a vertical plate in free convection with assisting external flow (from Reference 60).

departure of the velocity profile in the vicinity of the wall as compared to the profile for forced convection.

Combined forced and free convection from vertical isothermal [61, 62], and uniformly heated [62, 63] cylinders has been studied analytically. The problem is relatively more complex than for flow along a vertical flat plate due to the nonsimilarity arising from the transverse curvature. The available results show that for buoyancy assisted and opposed flow, free convection increases both the friction factor and heat transfer. There is little [61] experimental verification of analyses. The analytical results reported for combined convection along vertical cylinders (and also plates) are for laminar flow only.

2.2.4 Summary of Combined Natural and Forced Convection Results

The uncertainties and opposite trends in combined convective momentum and heat transfer for different flow regimes and directions (particularly for internal flow and in rod bundles) could make the use of available data base extremely vulnerable to errors for a reactor safety code. We summarize here the effects of buoyancy on fluid friction and heat transfer in laminar and turbulent flow.

1. Critical Reynolds Number

   The effects of buoyancy on heat transfer for laminar and turbulent flows are opposite for a given direction of main flow. For example, in upflow (for a hot leg LOCA) laminar heat transfer is enhanced while the turbulent is reduced. The parameters for which the flow can be considered to be natural, combined or forced have not been established.

2. Entrance Region

   The need to know the extent of the entrance region is especially important for laminar flow in order to account for the Graetz number effects. Correlations are not available for determining the length of the entrance region in terms of dimensionless parameters and conditions relevant to the problem.

3. Relative Magnitude of Grashof and Reynolds Numbers

The correlating parameter for laminar flow is GrPr/Re, whereas that for turbulent is $GrPr/Re^2$. It is therefore important to know the relative magnitude of GrPr/Re or $GrPr/Re^2$. According to Hall's hypothesis [46] and measurements [40, 41], the heat transfer coefficient for turbulent flow may first decrease, and then increase as the parameter $GrPr/Re^2$ is increased. It becomes crucial to determine on which part of the Nu vs. $GrPr/Re^2$ curve the system is operating.

4. Effect of Power Skew

Apart from the buoyancy, power skew induces radial (secondary) flow and the interaction of these two effects and their combined influence on heat transfer and fluid friction has not been studied and is not known.

2.3 Natural Convection

Combined forced and natural convection is expected to be the primary mode of heat transfer when, following the reactor shutdown, a fewer number of pumps are used to maintain circulation. After about one month of shutdown, the decay heat may fall below the frictional heat added by pumps to coolant. At this stage, it is logical to stop the pumps and leave the core to cool by natural convection circulation.

In the initial period the core temperature would be high leading to turbulent natural convection ($Gr > 10^9$). On the other hand, after further elapse of time laminar natural convection would control the heat removal when the core has cooled down to lower temperature. The critical value of the Rayleigh number for the onset of turbulence has not been established either by experiments or analyses. The transition zone is even more uncertain. The commonly used criteria for natural convection from a vertical plate are:

$$Ra_c \simeq 10^9 \sim 10^{10} \text{ for uniform wall temperature}$$

and

$$Ra_c^* \simeq 10^{13} \sim 10^{14} \text{ for uniform heat flux}$$

The large majority of available information on laminar natural convection is analytical and/or numerical work, mostly in external flow. We will briefly refer to those, primarily for the sake of completeness.

2.3.1 Internal Natural Convection

Laminar natural convection in the entrance region of a vertical tube has been solved by Davis and Perona [64] using finite difference scheme. In isothermal tube flow the hydrodynamic development length, defined by attainment of 90% of the fully developed (Poiseulle) velocity profile, corresponds to

$$Gr^+ = \frac{g(T_w - T_\infty)R^4}{T_\infty \ell \nu^2} = 4 \tag{18}$$

where R is the tube radius, T_∞ is the ambient temperature and ℓ is the length measured from tube inlet. Their heat transfer calculations compared well with available experimental data for air, near tube entry, and was correlated by

$$Nu_r = 0.61(Gr^+ Pr)^{1/4} \tag{19}$$

where the Nusselt number is based on the radius as the characteristic length. Experimental results for air together with finite difference solution for a vertical tube with a constant heat flux boundary condition have been reported by Dyer [65]. At short distances from entry the correlation is given as

$$Nu = 0.67\ Ra^{0.2} \tag{20}$$

and near fully developed condition as

$$Nu = (Ra/8)^{0.5} \tag{21}$$

Natural convection laminar flow and heat transfer in vertical channels formed by two parallel walls at equal [66, 67-69], unequal [66, 70-72] uniform wall temperatures as well as uniform wall heat flux (asymmetric) [70, 73, 74] boundary conditions has received considerable analytical and experimental attention. A more complete list of earlier studies are cited in the papers referred to above. The local Nusselt numbers have been found to depend on the velocity profile at the inlet [74]. At the inlet, the local heat transfer could be approximated by the results for free convection heat transfer from a vertical plate. In general, a satisfactory agreement has been obtained between analytical predictions and experimental data for the average heat transfer for both air [66, 69, 71] and for water [74]. Figure 10 is included here as evidence of the good agreement between analysis and data.

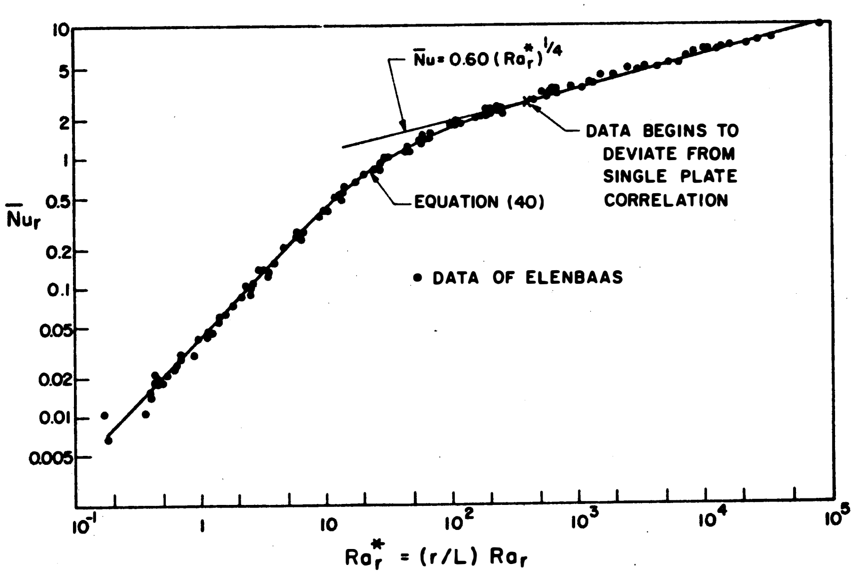


Figure 10. A comparison between the derived correlation and the measurements of Elenbaas for average heat transfer between parallel isothermal plates (From Reference 66).

The analytical results of Kettleborough [67] show that the local Nusselt number reaches a minimum at some distance along the channel. The minimum does not remain fixed at a given location but varies during the transient. Similar type of behavior has been noted for transient natural convection in cavities and along a vertical plate.

Davis and Perona [75] have carried out numerical and experimental investigations of laminar natural convection heat transfer in rod bundles. Agreement within 25% was obtained between numerical and experimental results for heat transfer in air at P/D = 1.68. The fully developed Nusselt number is nearly 6, compared to forced convection value of 4.364 for a tube with a uniform wall heat flux. Numerically, they predict a decrease in the heat transfer rate as P/D is increased.

Dutton and Welty [76] have reported experimental results for mercury with rod clusters of P/D = 1.1, 1.3 and 1.5. The trend of decreasing Nusselt number with increasing P/D, as reported in [75], is confirmed, but with a difference. Such decrease is observed at values of Gr^* below 10^4; nearly unaffected (with change in P/D) in the neighborhood of $Gr^* = 10^8$; but increased (with increase in P/D) beyond 10^9, the results for a pipe being the upper limit.

Recently an experimental study has been reported [77] where heat transfer from an array of four cylinders, in vertical and horizontal orientations, to its cubical enclosure through fluid media are given. The fluids used were air, water, glycerin and a silicone oil covering a Prandtl number range from 0.7 to 3.1×10^4. The results are correlated by the following equations:

Vertical:
$$Nu_L = 0.262\ Ra_L^{0.268} Pr^{0.028} \tag{22}$$

Horizontal:
$$Nu_L = 0.498\ Ra_L^{0.245} Pr^{-0.002} \tag{23}$$

where Nu_L and Ra_L are based on length of the cylinder as the characteristic dimension. The average deviation of the data from Eqs. (22) and (23) was reported to be less than 5 and 7 percent, respectively.

The available analyses for laminar natural convection cannot be applied to wet or saturated steam as the variations in physical properties have not been adequately taken into account. Practically, no experimental information exists on heat transfer with steam in rod bundles.

The question of flow resistance in natural convection through an internal geometry has not been addressed by Davis and Perona [75]. They have estimated the volume flow rate through a rod bundle by integrating the laminar velocity profile. Whereas the calculated and measured heat transfer rate agreed, the measured flow rate was too low due to the resistance offered by numerous spacer grids. There has been no suggestion on how to estimate such resistances under natural convection flow conditions. Some recent approaches in analyzing natural circulation loops, to be discussed later, treat flow resistance by equivalent forced convection friction factor. The relation between the Grashof and Reynolds numbers in an open, vertical tube under natural convection conditions has been established empirically [78]. It was confirmed that the effects of a calming (developing) section length on the relations between Gr and Re are similar to those on the heat transfer relations between Rayleigh number and Nusselt number reported by Dyer [65].

2.3.2 External Natural Convection

2.3.2.1 Laminar Flow

In flow of constant property fluids over a vertical surface, Squire [79, p. 528] developed an approximate solution assuming equal values for the thickness of hydrodynamic and thermal boundary layers. The results have been corroborated experimentally in air by Schmidt and Beckmann [79, p. 527] and the general trend has been summarized by Eckert and Drake [79, p. 528] as

$$Nu = 0.508\, Pr^{1/4}(0.952+Pr)^{-1/4} Ra^{1/4} \tag{24}$$

When the plate is inclined, the vertical component of gravitational acceleration ($g \sin \alpha$) is used to define Grashof number in Eq. (24).

Ostrach [80] has generated numerical solutions of the governing differential equations for Prandtl numbers in the range of 0.1 to 100 assuming constant physical properties, and the results have been experimentally corroborated in air. Much of the available information has been summarized by Ede [81]. Methods of supposition, local similarity [82] and nonsimilarity [83] have been used to deal with non-uniform thermal boundary conditions. Empirical correlations for laminar free convection heat transfer from a vertical plate with uniform temperature and uniform heat flux boundary conditions have been constructed by Churchill and Ozoe [84]. These correlations were obtained from available analytical results and are appropriate for the complete ($0 < Pr < \infty$) Prandtl number range. There has been but very limited experimental confirmation of the correlations proposed [84]. For gases with temperature dependent viscosity and thermal conductivity, Sparrow and Gregg [85] have suggested use of constant transport property results with properties evaluated at a reference temperature,

$$T_r = T_w - 0.38(T_w - T_\infty) \tag{25}$$

where T_w is the isothermal wall temperature and T_∞ is the free stream value.

The earlier experimental free convection heat transfer data for laminar flow over a vertical isothermal plate has been presented by McAdams [7, p. 172] in terms of the correlation,

$$Nu_L = 0.59(Gr_L \cdot Pr)^{1/4} \tag{26}$$

The local heat transfer coefficient for laminar flow along a uniformly heated vertical plate was correlated by Vliet [86] in a form of an empirical relation,

$$Nu_x = 0.60(Gr^* \cdot Pr)^{1/5} , \quad 10^5 < Gr_x{}^*Pr < 10^{11} \tag{27}$$

where

$$Gr^* = Gr_x \cdot Nu_x = \left[\frac{g\beta(T_w - T_\infty)x^3}{\nu^2}\right]\left[\frac{q_w x}{k(T_w - T_\infty)}\right]$$

The experimental results were obtained with water as the test fluid.

The effect of a nonisothermal wall on natural convection has been investigated. Solutions for laminar natural convection boundary layers on nonuniformly heated [87] and on nonisothermal vertical flat plates have been developed by Kao et al. [88], Merkin [89], and Kelleher and Yang [90]. An approximate method for analyzing both external and internal free convection heat transfer problems has been developed by Raithby and Hollands [66], and good agreement was obtained between predictions and available data. In each case the wall thermal boundary conditions must be specified <u>a priori</u> and fluid response determined. Unfortunately, this is not the physical situation when heat is generated in the solid, as in the case of nuclear reactor fuel element. A solution to the conjugate problem of laminar free convection from a conducting, heat generating vertical flat plate was obtained by Kelleher and Yang [91].

Martynenko et al. [92] have compared the results of different analyses and experiments for external flow over isothermal vertical cylinders, for constant property fluids in the Prandtl number range of 0.01 to 100. They recommend the following correlations for average heat transfer in laminar flow over a cylinder of diameter D and length L,

$$\frac{\overline{Nu}_L}{Ra_L^{1/4}} = 0.53\ Pr^{0.05} + 0.68(Ra_L^{-1/4}\ \frac{L}{D}), \quad \text{for } Ra_L^{-1/4}(L/D) < 2 \qquad (28a)$$

and

$$\frac{\overline{Nu}_L}{Ra_L^{1/4}} = 0.87(Ra_L^{-1/4}\ \frac{L}{D})^{0.87}, \quad \text{for } Ra_L^{-1/4}(L/D) > 2 \qquad (28b)$$

Comparison of local and average heat transfer results for slender cylinders show [92-94] that the Nusselt number correlations for free convection from an isothermal vertical plate cannot be used to predict heat transfer for a vertical cylinder.

Fahidy [95] suggested an upper limit at which, for 5% error, the effect of curvature on heat transfer may be neglected. This limit is $Ra_L^{-1/4}(L/D) < 0.039$. Thus, the average heat transfer coefficient for $Ra_L^{-1/4}(L/D) < 2$ may be calculated from Eq. (28a), which is derived from numerical data. For thin wires, $Ra_L^{-1/4}(L/D) > 2$, Eq. (28b) gives the best agreement with experiment.

For laminar free convection from a vertical, uniformly heated cylinder Nagendra et al. [96] suggest the following empirical equations:

$$Nu_D = 0.93(Ra_D \frac{D}{L})^{0.05} \quad \text{for wires} \quad (Ra_D \frac{D}{L} > 10^4) \tag{29a}$$

$$Nu_D = 1.37(Ra_D \frac{D}{L})^{0.16} \quad \text{for long cylinders} \quad (0.05 < Ra_D \frac{D}{L} < 10^4) \tag{29b}$$

$$Nu_D = 0.6(Ra_D \frac{D}{L})^{0.25} \quad \text{for short cylinders} \quad (Ra_D \frac{D}{L} < 0.05 \tag{29c}$$

These correlations are compared with experimental data and with correlations for isothermal cylinders. No experimental work appears to be available for nonuniformly heated cylinders.

2.3.2.2 Turbulent Natural Convection

The BEEST [2] and other codes for analysis of heat transfer in reactor core use the free convection correlations for constant property fluid in external flow [7, p. 181]. This correlation is based on measurements of Jakob and coworkers [97] in the early thirties, for free convection from isothermal vertical cylinders and plates,

$$\frac{hL}{k_f} = 0.13(Gr_{L,f}(Pr_f)^{1/3} \text{ for } 10^9 < Gr_{L,f}Pr_f < 10^{13} \tag{30}$$

The subscript f refers to evaluation of properties at the average film temperature. The index of characteristic length L in Eq. (30) is such that the heat transfer coefficient is independent of geometry, and hence the correlation can be applied to different geometries in external flow.

Eckert and Jackson [98, pp. 322-325] carried out an analysis using the available information on turbulent free convection and arrived at the following correlation,

$$Nu = 0.0295\ Pr^{7/15}Gr^{2/5}(1 + 0.494\ Pr^{2/3})^{-2/5} \tag{31}$$

In the range of Grashof number from 10^{10} to 10^{12}, the correlation was in good agreement with experimental values of Jakob [97] and Saunders [79, p. 535] in air, as well as with measurements of Fujii et al. [99] in water and oil.

Bayley [100] has suggested the following correlations, on the basis of an analysis corroborating several experimental results,

$$Nu = 0.10(GrPr)^{0.33} \text{ for } 2\times10^9 < GrPr < 10^{12} \tag{32a}$$

and

$$Nu = 0.183(Gr\ Pr)^{0.31} \text{ for up to } GrPr = 10^{15} \tag{32b}$$

Protopopov and Sharma [101] correlated the results for carbon dioxide near the critical region, taking into account the physical property variation and expressing the dimensionless parameters in terms of the integrated average values over the temperature range of interest. They found the following correlations to be applicable,

$$\overline{Nu} = 0.53(\overline{Gr}\ \overline{Pr})^{1/4} \tag{33a}$$

in the laminar range and

$$\overline{Nu} = 0.135(\overline{Gr}\ \overline{Pr})^{1/3} \tag{33b}$$

in the turbulent range. The correlation of Kato et al. [102],

$$\overline{Nu} = 0.138\ Gr^{0.36}(Pr^{0.175} - 0.55) \tag{34}$$

has been applied to experimental results for air and ethylene glycol.

The correlations referred to in Eqs. (30) to (33) are for external flow and as such are not applicable to rod cluster geometry, although they have been used in the two codes (TRAC, BEEST) mentioned earlier. Even for the external geometries the correlations predict widely differing Nusselt numbers at different Prandtl and Grashof numbers. Some results are presented in Table 3 and Figures 11 and 12 for Pr = 0.7 and 1, respectively, to indicate the degree of divergence. Wide variations are noted. For example, at Pr = 0.72 and Gr = 10^{12}, Eckert and Jackson's correlation predicts a Nusselt number which is over two times larger than that calculated from the Bayley [100] correlation. Ede [103] has compared the available experimental data for plates and large cylinders with correlations. There is considerable scatter in the data and it is difficult to state that any one is better than another. Clearly, more reliable correlations are needed, particularly at large Grashof numbers. Note that no experimental

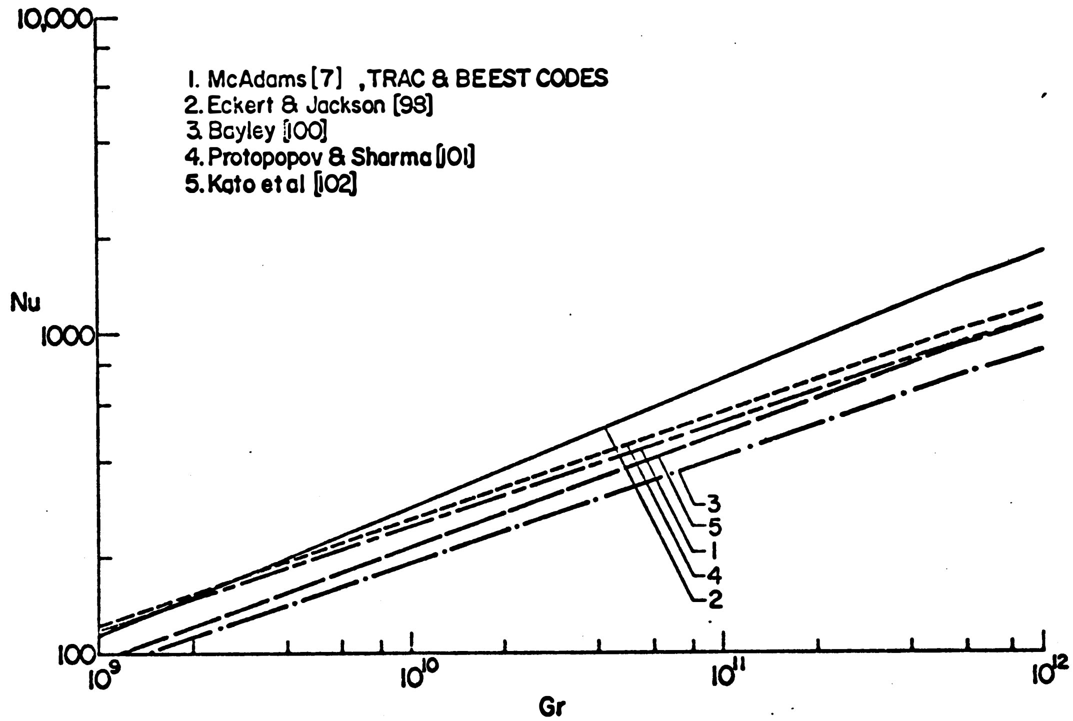


Figure 11. Comparison of average turbulent free convection heat transfer from an isothermal vertical plate: Pr = 0.72.

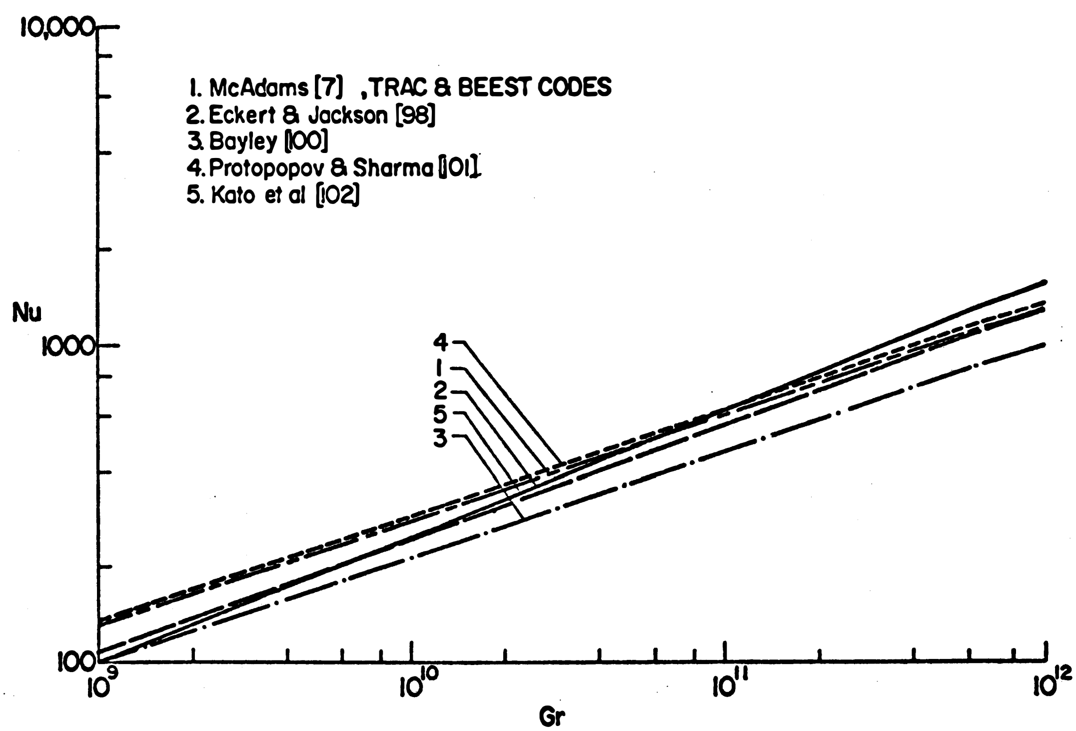


Figure 12. Comparison of average turbulent free convection heat transfer from an isothermal vertical plate: Pr = 1.0.

turbulent natural convection data are available for steam.

The conclusions reached above have been substantiated independently by a very recent publication [104]. Turbulent natural convection heat transfer to air from a vertical plate at large Grashof numbers has been reported by Siebers [104]. The simplified continuity, momentum and energy equations for a two-dimensional model were solved numerically using computer code STAN-5. Figure 12 gives a comparison of the numerical predictions of heat transfer by STAN-5 to the experimental work of Saunders [105] and Warner and Arpaci [106]. The results show that in both the laminar ($Gr_x < 5\times10^9$) and turbulent ($Gr_x > 5\times10^9$) flow ranges the numerical predictions of local heat transfer agree reasonably well with best fit curves for the experimental results. Figure 13 is a plot of local Nusselt vs. local Grashof number predicted by STAN-5 [104] for surfaces at 30°C and 530°C. Also shown are extrapolations of the analytical correlations of Eckert and Jackson [98] and Bayley [100] and the experimental correlation of McAdams [7, p. 181] at lower Grashof numbers. The figure points out several facts. First, the high temperature has a large effect on the heat transfer in the boundary layer (note that radiation has not been included in the Nusselt number). There is about a 50% difference in heat transfer between the predictions for the low and high temperatures of STAN-5. Second, an extrapolation of McAdams' correlation from the range of Grashof numbers in which the experiments were conducted appeasr to be conservative in predicting heat transfer. This correlation is one of the most commonly used in reactor heat transfer codes [2]. Finally, the analytical results of Eckert and Jackson [98] fall somewhere in between the high and low temperature results.

A detailed study of turbulent free convection flow and heat transfer along a uniformly heated vertical plate has been reported by Vliet and Lin [107]. The local heat transfer coefficients were correlated by the empirical equation

Table 3

Free Convection Nusselt Number in External Flow -

Predictions of Different Correlations

Authors	Correlations	Nu Numbers for Pr=0.72				Nu Numbers for Pr=1			
		Gr=10^9	10^{10}	10^{11}	10^{12}	10^9	10^{10}	10^{11}	10^{12}
McAdams, Jakob, BEEST, TRAC [7, 97, 2]	$\frac{h\,L}{k_f} = 0.12\left[\frac{L^3\rho_f^2 g\beta_f\Delta T}{\mu_f^2} \times \left(\frac{c_p\mu}{k}\right)_f\right]^{1/3}$	116	251	540	1165	130	278	603	1300
Eckert and Jackson [98]	$Nu=0.0295\ Pr^{7/15}Gr^{2/5}\ (1+0.0494Pr^{2/3})^{-2/5}$	115	278	726	1823	100	250	634	1577
Bayley [100]	$Nu=0.10(GrPr)^{0.33}$	89	190	413	890	100	214	464	1000
Protopopov and Sharma [101]	$Nu=0.135(\overline{Gr}\ \overline{Pr})^{1/3}$	121	259	562	1211	135	289	626	1350
Kato et al. [102]	$Nu=0.138\ Gr^{0.36} \times (Pr^{0.175} - 0.55)$	94	215	492	1128	108	247	565	1295

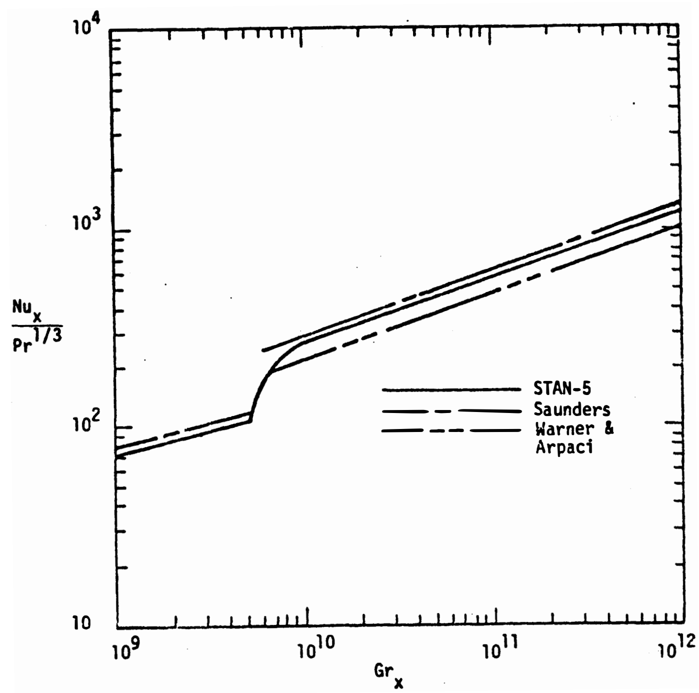


Figure 13. Comparison of local free convection heat transfer predictions with experimental data (From Reference 104).

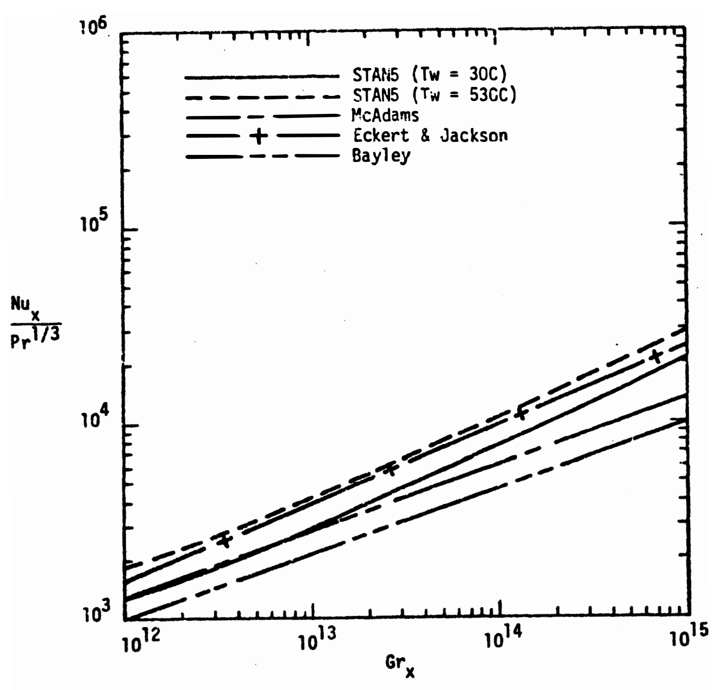


Figure 14. Comparison of local free convection heat transfer results at high Grashof numbers (From Reference 104).

$$Nu_x = 0.568(Gr_x^*\cdot Pr)^{0.22},\ 2\times10^{13} < Ra^* < 10^{16} \tag{35}$$

This equation indicates that the heat transfer coefficient decreases slightly (as $x^{0.12}$) with the distance along the plate. The results are in fair agreement with the analysis of Bayley [100]. The constant-wall-temperature data of other investigators when plotted in the form of Nu_x versus $Gr_x^*\cdot Pr$ show quite good agreement with the extension of their data to lower $Gr_x^*\cdot Pr$, but exhibit much earlier transition in terms of $Gr_x^*\cdot Pr$.

Average heat transfer coefficient for air on vertical cylinders maintained at uniform temperature have been reviewed by Ede [103]. Despite the considerable amount of scatter it is clear that the effect of reducing D/L is to increase Nu_L at a given value of Ra_L. There is insufficient information to enable any conclusions to be drawn concerning the effect of D/L on the values of Rayleigh or Grashof numbers at which turbulence starts.

2.3.3 Natural Convection: Summary

The works on natural convection just discussed are for uniform temperature and heat flux boundary conditions. Little analytical [66] and no experimental data are available on natural convection heat transfer from nonisothermal and/or nonuniformly heated surfaces of the type that would be relevant to nuclear reactor core cooling. The effect of surface roughness elements on fluid friction and heat transfer under natural convection conditions have received little attention [108]. This is relevant to the nuclear reactor core because spacers are placed on the fuel rods. Roughness elements influence the transition from laminar to turbulent flow, and a comprehensive, up-to-date discussion of the basic processes involved is available [109].

No information could be found on internal turbulent natural convection with through-flow. Theoretical and experimental results are available for recirculating flow in cavities [110], but the information is not applicable either to the reactor vessel or the primary coolant loop. There is a need for fluid friction and heat transfer data with water, steam and water-steam mixtures for internal natural convection flow through channels and in rod bundles. The experimental study of Fujii et al. [99] was in a sense for internal geometry (e.g. annulus) with through flow. However, the geometric dimensions were such (80 mm ID to 385 mm OD) that the presence of external cylinder was not felt and hydrodynamically the flow was external.

Even in external flow, there are significant differences between predictions of available correlations and experimental results with nearly constant property fluids (air). Available results are limited to about $Gr = 10^{12}$, and no experimental data exist for steam. It was postulated that at high core temperature reaction of steam and zirconium could release hydrogen, helium and other gases which would interact with flow and alter natural convection heat transfer. Relatively large bubbles of such gases could cause partial blockage of the passage. Information with such conditions simulated in laboratory models are necessary.

2.4 Combined Natural Convection and Radiation

With increasing core temperature, in Zone III and above (Figure 3), radiative heat exchange is expected to occur simultaneously with natural convection. Depending on the temperature level and its variation along the fuel rods, radiation may even predominate over convection. Radiation heat exchange will take place between

- rods and superheated steam
- rods and liquid droplets in steam
- rods and liquid-vapor interface
- rods and upper plenum

and

- rods to rods to can wall to reactor vessel as a result of power skew.

As a consequence, radiation needs to be accounted in nuclear reactor core heat transfer calculations under accident conditions when the core is partially or totally uncovered. Convection and radiation will occur in parallel and the two modes of heat transfer may be simply additive or interaction of convection and radiation may have to be accounted for estimating the total heat transfer rate. In this subsection the state-of-the-art of modeling combined natural convection and radiation heat transfer will be discussed.

2.4.1 Modeling of Radiation Heat Transfer in Rod Cluster

One of the major problems encountered in the mathematical treatment of fluid flow and heat transfer in rod cluster is the simulation of the cluster geometry. So far as radiation heat transfer is concerned, the question of radiation exchange between the rods, rods and shell etc. arises for both analytical and experimental evaluations. Radiation exchange between rods, in a radial direction, can take place due to power-skew even at normal operating condition. When the core is partially uncovered, there can be strong variation of temperature along the length of a rod, due to exposition of differing heat transfer rates (Figure 3), depending on the extent of the uncovering, and radiation exchange in axial direction is expected to occur. We first consider radiation heat transfer in a rod bundle in the absence of an absorbing and emitting medium between the rods.

If the rods are separated by a radiation nonparticipating medium, radiation heat exchange between the rods in the radial and axial directions can be calculated using well established zonal methods of radiant heat transfer analysis [111, 112]. There are no conceptual difficulties in predicting radiant heat exchange if the radiative characteristics of materials are known [113-117]; however, evaluation of configuration (view, angle) factors and a simultaneous solution of a large number of nonlinear algebraic equations can become very tedious and use of Monte Carlo methods [112] may be preferred. The procedure for evaluating infinitely long, parallel rods are available [119], but the height of the core uncovered is expected to be finite and this will necessitate developing models for evaluating radiation heat exchange, between finite length rods and steam surrounding them.

Since radial exchange may exist at all conditions, reference is made to Reilly et al. [116] who have suggested a two-rod model for evaluating a configuration factor in delta or rectangular arrays of infinitely long rods. Two imaginary surfaces, both tangents to each of the two rods, and the convex half surfaces of the two facing rods form a closed cell. The surfaces are considered as gray and to radiate diffusely. Exchange with the fluid is neglected; radiation flux at a rod surface is used as boundary condition for the solution of transient heat conduction in the rods.

Yao et al. [117] have proposed grouping rods at one temperature as one surface, although they are physicall separated. Their model assumes three surface zones for the entire core (rods and enclosures), since the surfaces are on the average at three different temperatures. In order to estimate the configuration factor of a rod with respect to others in a bundle, they consider the presence of only those rods which are largely "visible". This leads to accuracies of 97 percent for P/D = 1.3 and 90 percnet for P/D = 1.6, and are considered acceptable.

Anderson and Tien [118] have suggested anistropic transport model, in addition (though independently) to the surface grouping scheme of Reference 115. They point out that when the surface is large (one "large surface" is made up of several rods at nearly equal temperature in a reactor core) the incident radiation will not be uniform over the surface, whereas the emitted radiation can be considered as being diffuse (due to a circumferentially uniform rod temperture). Consequently, the "view factor" for a rod for reflection should be different from the one for emission. This is the anisotropicity. A fraction μ of the incident radiation is reflected towards the origin and $(1 - \mu)$ uniformly in other directions.

For a typical LOCA and core uncovering in BWR's and PWR's, radiation heat transfer in a fuel rod bundle can be important because not only the channel but also the reactor vessel walls act as heat sinks. Available calculations show that anisotropy has a significant effect on the uniformity of the clad surface temperature [114]. It was found that the conventional model with uniform temperature overestimated heat transfer by about 30% in a typical situation or correspondingly underestimated the temperature rise. The fact that the power skew is present not only in a subassembly but in the entire reactor may be particularly significant. Neither the effect of the power skew nor of the longitudinal temperature variation along the fuel rods on radiation heat transfer and temperature in the core have been investigated.

2.4.2 Interaction of Convection and Radiation

Cooling of a partly uncovered core is by simultaneous convection and radiation, and the presence of radiation is expected to influence convective heat transfer. Here, we briefly review the very meager literature which is concerned with heat transfer by natural convection and radiation to a nonparticipating gas in external and internal flow as well as to a radiating gas surrounding a vertical plate. The problems considered are not directly applicable to the cooling of a nuclear reactor core, and the discussion is intended primarily as a background.

Heat transfer from a heat generating vertical cylinder by free convection to a transparent fluid and by radiation to the surroundings has been studied analytically [120]. The results obtained show that radiation equalizes the surface temperature of the cylinder. In addition, it was found that radiation reduces and curvature increases heat dissipation from the cylinder for some values of relevant dimensionless parameters. As a consequence of the interaction between free convection and radiation a minimum results in the heat transfer coefficient at some distance away from the leading end of the cylinder. Use of the results obtained for isothermal vertical plate to predict convective heat transfer from a vertical cylinder would underestimate heat dissipation [120].

Interaction of radiation and developing laminar natural convection heat transfer to a transparent fluid (e.g. air) in a vertical parallel plate channel with asymmetric heating has been performed by Carpenter et al. [121]. Consideration of radiation leads to five dimensionless parameters (heat flux ratio, Rayleigh number, aspect ratio, emissivity and radiation number) which affect wall temperature and heat transfer performance. Radiation to the inlet-exit and the cooler opposing entrance wall significantly alters the convective heat transfer by reducing the maximum wall temperature by as much as 50 percent. The fully-developed flow solution reported for natural convection in the absence of radiation [70] and could not be obtained with radiation present. Finally, under certain conditions, the local Nusselt number actually becomes negative, indicating a large radiative heat loss to the exit of the channel and subsequent heating of the wall by the high local fluid temperature.

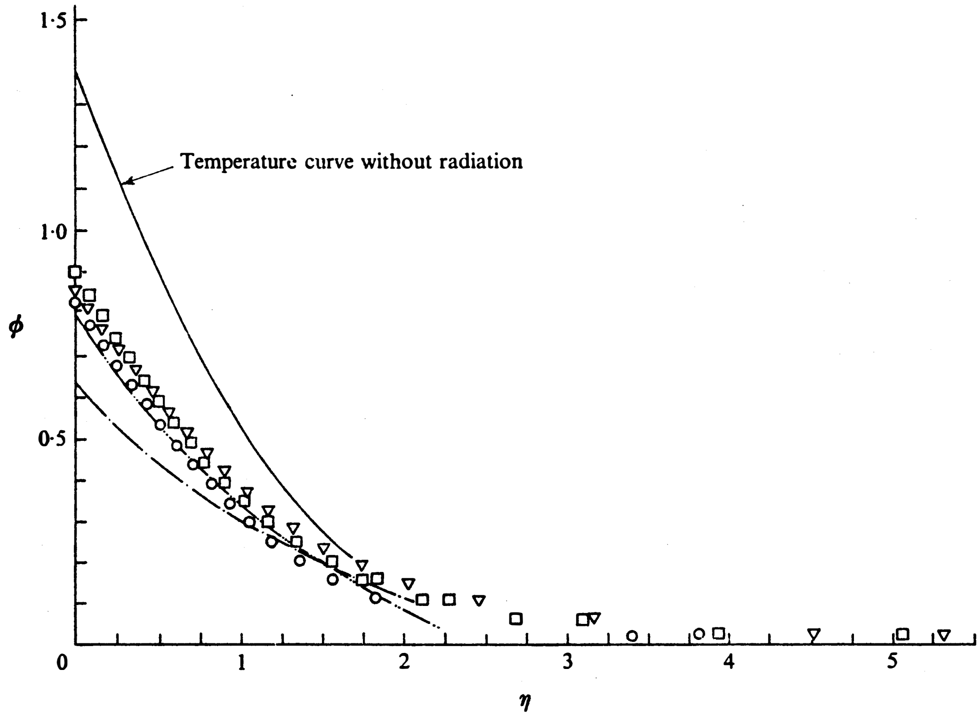
Combined natural convection and radiation heat transfer from a vertical surface suspended in an infrared radiating gas (CO_2, H_2O, NH_3, etc.) of infinite extent has received some analytical [122-128] and experimental [123, 126, 128] attention. Radiative transfer has been treated as one-dimensional and both gray [122-127] and nongray [128] models have been used. The results obtained are inconclusive. Some investigators have obtained that interaction between natural convection and radiation in a participating gas increases the convective heat transfer at the wall, whereas others have found that the interaction is negligible. Both England and Emery [124] and Hasegawa et al. [128] have concluded from their experimental studies that convective heat transfer may be predicted through standard non-radiative gas heat transfer correlations and the total heat transfer can be determined with sufficient accuracy by addition of the convective and radiative contributions without considering mutual interaction. The experimental data in these two studies were obtained with carbon dioxide (CO_2) as the radiating gas and the "free stream" temperature maintained close to the ambient temperature. Anderson and Gebhart [126], on the other hand, found that ammonia causes a strong interaction between convection and radiation. The measured temperature profiles in the gas for a wall emissivity of 0.8 are shown in Figure 15. The presence of a radiating gas (NH_3) increased the convective heat transfer by as much as 40% for the conditions used in the experiments. It is also shown that the experimental temperature distributions agree very well with theoretical predictions obtained by treating the convection and radiation processes as independent and superimposed.

2.4.3 Natural Convection and Radiation Heat Transfer in Rod Bundles Containing Steam and Steam-Water Mixture

Sun et al. [113] have discussed combined convection and radiation heat exchange between rods and the steam-water (droplet) mixture, as applicable to the spray cooling of an ECCS. The modes of heat transfer considered are: convection to the vapor, radiation from the walls to the vapor, from the walls to the droplet, and vapor-droplet heat transfer. Both vapor and the droplets are considered to be optically thin. Heat transfer from the droplet to its surrounding superheated steam is assumed as in forced convection heat transfer from a sphere, e.g.

$$Nu = 0.37\ Re^{0.60} \tag{36}$$

The relative velocity of vapor past the droplet is estimated by trial and error from a force balance between the droplet weight and drag force. The droplet diameter and the spray concentration have been taken as independent variables.



Temperature distribution in ammonia gas, $\epsilon = 0{\cdot}8$, Δt_c based upon q_w, average $Pr = 0{\cdot}902$, ambient temperature for all runs: $T_0 = 299{\cdot}7\,°K$.

p (atm)	Δt (°C)	ζ	ξ	γ	Experimental data points	Present theoretical solution
2·03	32·2	0·0131	0·131	0·238	▽	—·—
4·10	13·43	0·0193	0·113	0·122	□	—··—
6·03	7·12				○	

Figure 15. Comparison of predicted and measured temperature distributions in the natural convection boundary layer along a vertical plate (From Reference 126).

Smaller droplets are more effective as radiation heat sinks [117]. By comparing FLECHT Code for heat transfer coefficient Sun et al. [113] postulate that average droplet size in a rod bundle is 0.228 cm.

Although the procedures outlined in this paper are useful for heat transfer estimations, it is to be noted that salient input information such as droplet-size, etc. are derived from available computer codes whose data base itself is in question.

Droplets of water and steam (vapor) present in the core at accident condition are considered as absorbing media [117]. Semi-gray models are used in the sense that absorption characteristics at rod average temperature and emissivity of vapor and droplets at two phase temperature have been assumed. The rod surfaces are considered to be gray. Numerical calculations of Yao et al. [117] indicate that cooling of the hot rods is strongly influenced by radiation absorption of droplets when the droplet size is small. At a given void fraction, the thimble rods are heated by surface radiation from hot rods when the droplet size is large. Cooling of the thimble rods with droplets acting as radiation sink takes place when the droplets are small.

Numerical calculations for a seven-rod bundle [118] indicate that good accu- (with reference to the "exact" solution obtained by considering the rods separately) and saving in computation time are achieved taking $\mu = 0.5$ and two surfaces of reflection per rod. Anderson and Tien [118] consider parallel plate model (in simplification to the rod bundle geometry) and two flux radiation components for heat exchange with vapor droplets. Surprisingly, the simple radiation model together with assumed convection heat transfer coefficients predicts the postulated LOCA cladding temperature through a CORECOOL computer code.

The infrared radiation characteristics of steam have been reasonably well established, however, the presence of water droplets results in scattering. The uncertainty in the droplet size distribution and the complexity of the rod bundle geometry introduces complexity into radiation transfer calculations. More detailed radiation analyses modelling cooling an uncovered reactor core and experimental verification of the models are needed to establish procedures for calculating radiation heat transfer.

The flow of vapor and droplet mixture is sometimes referred to as dispersed flow. Liquid droplets in dispersed flow can cause localized cooling of the heated wall provided the wall temperature is above saturation to a limited extent (Leidenfrost phenomena). Several measurements [129] indicate that such cooling at atmospheric pressure practically reduces to zero if the surface temperature is above 450 K.

Ganic and Rohsenow [130] have suggested a correlation for estimating heat transfer in dispersed flow:

$$q = q_d + q_v + q_r \tag{37}$$

The subscripts d, v and r, respectively, refer to droplet, convection to vapor, and radiation to vapor and drop from the wall. The individual contributions are given by

$$q_d = M_w h_{fg} \exp[1 - (T_w/T_s)^2] \tag{38a}$$

$$q_v = 0.023(k_g/D)\,Re^{0.8} Pr^{0.4} (T_w - T_{sat}) \tag{38b}$$

$$q_r = (F_{w\ell} + F_{wv})\,\sigma\,(T_w^4 - T_{sat}^4) \tag{38c}$$

where M_w is the mass flux of liquid drops and h_{fg} is the latent heat of vaporization. Equation (38b) is similar to the Dittus-Boelter relation. The droplet mass flux rate in Eq. (38a), and radiation view factors, in Eq. (38c), are estimated in the manner outlined by Sun et al. [117]. The total heat flux expressed by Eq. (37) has been found to correlate well with the experimental results in liquid nitrogen, especially for (T_w - T_{sat}) higher than 50°K.

The behavior of liquid drops suspended in a fluid stream and heat addition to dispersed flow is important to many technologies, including nuclear reactor safety, and an up-to-date review of relevant references is given by Ganic and Rohsenow [130, 131]. The review of literature indicates that the trajectories of the droplets near and the deposition of particles on the heated surface is not completely understood and fundamental heat transfer studies at conditions (e.g., rod bundle, temperature level, heat flux, natural convection, etc.) modeling cooling of a partly uncovered nuclear reactor core have not been performed.

2.5 Condensation

Condensation of steam is discussed briefly here because the process can occur during an accident and influence heat removal from the core [5, 6]. For example, the steam leaving the liquid-vapor interface (Figure 3) can condense in the cooler (upper) part of the core or in the plenum, and the condensate can run down along the fuel elements. Also, because of changes in system pressure and heat removal in the steam generator of a PWR, condensation may occur in the generator and the liquid return by gravity flow to the reactor vessel and the core.

A complete discussion of the fundamentals of condensation heat transfer available in Collier's book [15, Chapt. 10] and an up-to-date review of more recent contributions have been presented [132]. A couple of recent studies [133, 134] should be mentioned for the sake of completeness. Blangetti and Schlünder [133] have observed that the local Nusselt number for condensation of steam flowing down a vertical tube is proportional to $Re_f^{0.8}$ (for Re_f > 900), where

Re_f is based on the mass flow rate per unit perimeter. Miropolsky et al. [134] have reported extensive experimental results for condensation heat transfer from steam over a wide range of pressure and mass velocity through vertical tubes and annuli. Since the experimental parameters were rather arbitrarily varied and no nondimensionalization has been attempted, it is difficult to present a comprehensive picture. In general, it is noted that heat transfer coefficient decreases as the pressure is increased. At a mass flux of 10 kg/m^2s and p = 4×10^5 N/m^2, the condensation heat transfer from near saturated steam is about 50 times the pure forced convection heat transfer coefficient for flow through an 18 mm diameter vertical tube. However, the presence of small amounts of noncondensible gases can drastically reduce the condensation heat transfer coefficient [8, Chapt. 12; 15, Chap. 10; 9, Chap. 15; 135]. The fundamentals are reasonably well understood, but no experimental and analytical studies relevant to nuclear reactor core heat transfer have been reported.

Noncondensable gases such as hydrogen, helium and other gases generated by reaction of steam with zirconium cladding and/or released after the cladding had ruptured are expected to be present in the core, plenum, piping and steam generator [5, 6]. The steam-noncondensible gas mixture in the primary loop cannot be vented into the atmosphere, as is done in fossil-fired power plants for removing air from the condenser and the inert gases would continue to accumulate in the system (e.g. steam generator) and affect heat removal from the core.

Minkowycz and Sparrow [136] have performed analysis of laminar film condensation in the presence of inert gas. They showed that marked decrease in heat transfer can take place due to the presence of very small amount of noncondensibles. For example, 0.5 percent by mass of air in steam causes 50 percent reduction in heat transfer. The noncondensable gas provides a high diffusive resistance to the condensing vapor which results in loss of partial pressure of vapor at the condensing surface. This is attributed to be the main reason for the reduction in heat transfer. Their condensation heat transfer coefficients [136, 137] for stagnant (adjacent to a vertical surface) and forced steam-air mixtures are summarized in Figure 15. Mori and Hijikita [138] have shown analytically that the Nusselt number, for condensation heat transfer on a vertical plate in the presence of noncondensible gas, asymptotically approaches those for free convection and film condensation at both extremes.

Based on approximate analysis of condensation of steam on a vertical wall in the presence of inert gases, Madejski [139] had proposed the following empirical equation,

$$\frac{h}{h_o} = c\left[\frac{1 - (p_i/p)}{c + (1 - c)(p_i/p)}\right] \tag{39}$$

for the reduction in the condensation heat transfer coefficient. In this equation h and h_o are the condensation heat transfer coefficients in the presence and absence of inert gas, respectively; c = 0.012 and p_i/p is the ratio of the inert gas pressure to the total pressure. Reasonably good agreement has been obtained between Eq. (35) and available experimental data [139]. Also, the trends indicated in Figure 15 are consistent with those predicted by Eq. (35). It should be noted that the parameter c is not expected to be constant, and in the first approximation it should at least depend on the condensation Nusselt number in the absence of an inert gas [139].

A study by Borishanskiy et al. [138] reports experimental condensation heat transfer results for pure steam and steam-inert gas (nitrogen) mixtures in a vertical tube at low (0.8 to 3.0 MPa) pressures. The inlet velocities ranged from 2 to 50 m/s; the discharge steam quality was less than 0.5, and the volume fraction of the noncondensible gas was less than 2.5 percent. Empirical correlations are recommended in the paper for the condensation heat transfer in steam and steam-inert gas mixtures.

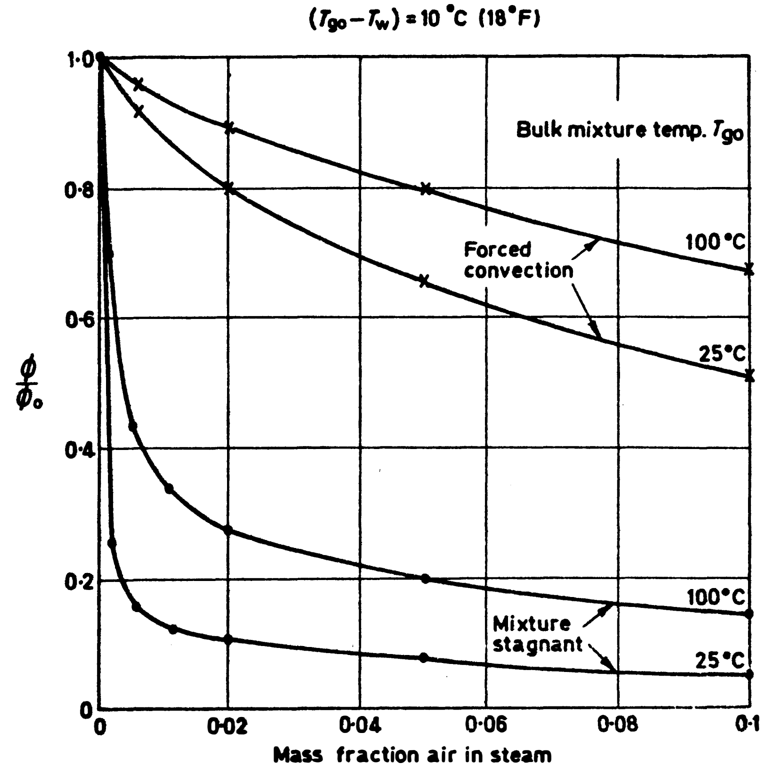


Figure 16. The effect of inert gas (air) on the condensation heat transfer [$\phi = h(T_{gi} - T_w)$] of steam (From Reference 15).

Since it is the diffusion of the condensable vapor through the vapor-inlet gas mixture that controls the heat transfer rate, the available information [139-141] (primarily analytical) cannot be extended to the TMI-2 reactor, the core of which is expected to contain hydrogen and helium. The experimental data on hand are for steam-air or steam-nitrogen mixtures. Furthermore, the effects of pressure and superheat need to be estimated.

3. DATA BASE FOR NATURAL CONVECTION CIRCULATION IN LOOPS

The proper design of many technologically important systems requires an understanding of the thermal-hydraulic behavior in which the circulation is caused by the density differences between the heated and cooled sections. Such systems include natural circulation boilers, nuclear reactor under accident or decay heat removal conditions, evaporators, thermosyphons for turbine blade, rotating electrical machinery, transformer, etc. cooling, passive solar energy collection and storage systems, and others. The buoyancy driven natural convection flows in closed and open loops consisting of single and multiple-parallel channels are quite complex. Problems such as stability of the loop is not completely understood and a realistic capability to simulate the dynamics of such a complex system as a nuclear reactor has not been developed. Even computer models for relatively simple laboratory loops have not predicted a priori system performance without adjusting model parameters on the basis of experimental data obtained with the system. In this review emphasis will be on the basic fluid friction and heat transfer data base needed for modeling natural circulation systems, analytical and experimental modeling studies using simple laboratory systems, and dynamic simulation of natural circulation in primary coolant nuclear reactor loops.

3.1 Fluid Friction and Heat Transfer Parameters

The primary coolant loop shown in Figure 1 consists of the reactor core, hot leg piping, steam generator primary side, cold leg and the coolant pump. In the post accident period, the primary coolant, either single phase or two phase, will circulate in the coolant loop due to the difference in densities between the two vertical legs. The net buoyancy force is balanced by friction and related pressure losses in different parts of the loop. Only if the buoyancy is sufficiently large to overcome the total losses in the complete loop will circulation continue. If the buoyancy force is lower then only localized recirculation may occur. The accurate knowledge of buoyancy induced pressure gradient vis-a-vis the frictional pressure drop of different components in the coolant loop is thus obviously very critical. In formulating models for the coolant loop and for the evaluation of the system dynamics of a reactor, generally the one-dimensional friction factor and heat transfer coefficients are used.

The discussions in earlier sections have been related to heat transfer mechanisms from which the heat transfer coefficients under different conditions can be evaluated. What follows now is a review of the available information and practice for combined heat transfer and flow resistance (friction factor) under natural convection conditions. The forced flow results are reasonably established [7]; in any event, the reserve capacity of the pump can take care of the uncertainties under forced flow conditions.

As already discussed in Sections 2.2 and 2.3, for both combined and natural convection, heat transfer affects the velocity and and temperature profiles very strongly. This is clearly indicated by velocity and temperature profiles calculated by some investigators [142-144]. Marked deviations occur already in combined flow at $Gr/Re \simeq 4$. The overall quantities such as the fluid friction factor and the Nusselt number, however, are less sensitive. For fully developed laminar combined convection in a tube the behavior of these two quantities (see Figures 17 and 18) is very similar. For example, the analysis of Brown [142, 143] yields the following relations:

$$f\ Re = 16 \quad \text{for} \quad Gr/Re < 8.4 \tag{40a}$$

$$f\ Re \simeq 5.8(Gr/Re)^{0.475} \quad \text{for} \quad Gr/Re > 8.4 \text{ and } \partial\rho/\partial z < 0 \tag{40b}$$

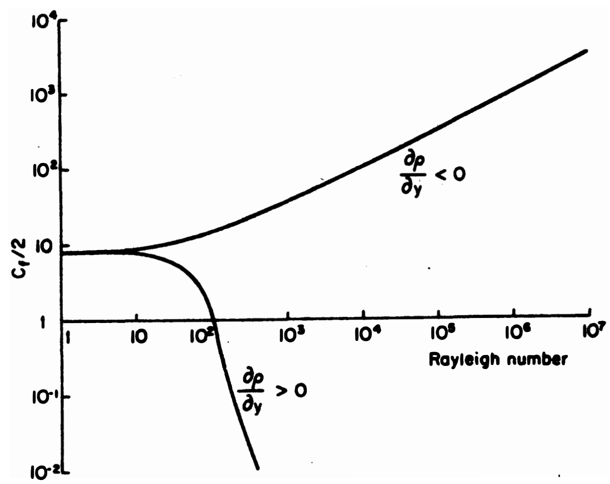


Figure 17. Variation of the friction factor constant, c_f, with Rayleigh number (From Reference 144).

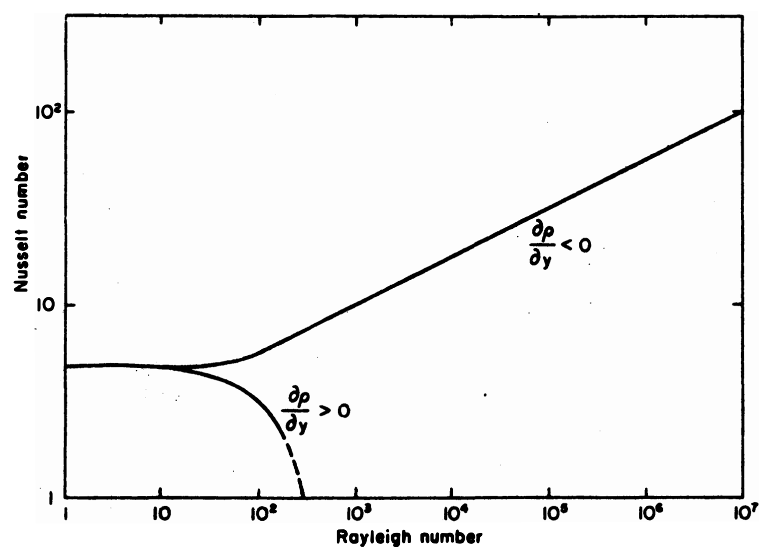


Figure 18. Variation of Nusselt number, based on fluid mean to wall temperature difference, as a function of Rayleigh number (From Reference 144).

and

$$Nu = 4.73 \quad \text{for} \quad Gr/Re < 12.8 \tag{41a}$$

$$Nu \simeq 2.5(Gr/Re)^{0.25} \quad \text{for} \quad Gr/Re > 12.8 \text{ and } \partial\rho/\partial z < 0 \tag{41b}$$

Equation (40) lies just above the upper range of the friction factor data observed with oil and water [145]. The measured temperature profiles [142] and Nusselt numbers [142, 32] agreed well with Eq. (41) for moderate heat input rates.

For both liquids and gases the friction factor f in combined and natural convection (in a narrow parameter range) can be approximated by a simple function,

$$f = C_f/Re^s \tag{42}$$

where the coefficient C_f and s depend on the Reynolds and Rayleigh (or Grashof) numbers as well as wall to bulk temperature ratio, T_w/T_b. The friction factor for natural convection circulation in a loop was taken [146-147] as f = 16/Re. This result is for laminar, fully developed Hagen-Poisseulle flow in a smooth circular tube. Extension of Poisseulle result to natural convection and use of the relation in modelling fluid friction in natural circulation loops, even under laminar flow conditions, is questionable. There is experimental evidence [148-151] that the fluid friction is significantly higher for natural than forced convection. Apart from the consideration of laminar flow, there are questions regarding the length required to reach fully developed condition under natural circulation conditions and the channel ("passage") geometry.

It is reported [148] that clear transition to turbulent regime is not observed in flow through wire wrapped rod bundles. The laminar friction factor is found to vary as 21/Re instead of 12/Re for pipes [151]. For turbulent condition, the friction factor in buoyancy induced flow is about two times the value predicted by the McAdams or Blasius correlations. Agarwal et al. [151] use the following flow dependent expression for f,

$$f = \frac{0.0050}{4} \{1 + [20000(\varepsilon/D_e) + (10^6/Re)]^{1/3}\} \tag{43}$$

where f is the fanning friction factor and ε is the surface roughness.

Conventionally, natural convection transport rates are expressed in terms of Grashof number. It is thus necessary to know a relationship between Gr and equivalent Re so as to apply the forced convection results. Kokugan et al. [78] have presented an analysis with experimental support, for such interrelationships, so far as laminar momentum transport is concerned. They considered natural convection flow through a pipe of heated length L_H at temperature T_H, followed by a calming length L_o at T_o; both the sections are of diameter D. From an energy balance on the system and assumption that for laminar Poisseulle flow, f = 16/Re, they obtained the result,

$$Gr_H = (7/4)Re_D^{\,2}(\rho_p/\rho_H) + 32Re_D[(L_H + L_o)/D](\nu_H/\nu_o) \tag{44}$$

when the physical property variation between the cold and hot sections could be neglected, Eq. (44) reduces to

$$Gr_H = (7/4)Re_o^{\,2} + 32\ Re_o(L_H + L_o)/D \tag{45}$$

The objective in describing the derivation of Kokugan et al. [78] was to indicate the procedure adopted for interrelating natural and forced convection parameters. They found experimentally that

$$Gr_o = 6.3\ Re_o^{\ 2} + 32\ Re_o(L_H + L_o)/D \tag{46}$$

where $Gr_o = (D^2 L_H g\beta_o \Delta T/\nu_o^{\ 2})$. Note that the characteristic dimension in Gr_o is a combination of diameter and length, or the cylinder volume. Comparison of theoretical and experimental results indicate confirmation of Poiseulle friction factor result for natural convection, although the entry and exit loss coefficients are different. Relations similar to those given by Eq. (46) should be established for different geometries and flow conditions so that forced convection information, if known, can be extended to natural convection.

Chato [152] has suggested a numerical scheme for estimating natural convection flow distribution in a system of vertical channels connected in parallel. However, the basic input parameters are extrapolated from forced convection results. Validity of friction factors, entrance and other pressure loss coefficients need verification for natural circulation systems because the velocity profiles are expected to be different. Correlations are needed not only for natural but also combined convection at low flow rates where buoyancy effects may be important.

An extensive review of heat transfer for natural convection boiling of different fluids in tubes, from plates and from cylinders has recently been prepared by Stephan and Abdelsalm [153] and need not be repeated. Empirical correlations for four different groups of fluids (water, hydrocarbons, cryogenic fluids and refrigerants) and employing different sets of dimensionless numbers for each group of substances has been recommended on the basis of regression analysis of the data.

3.2 Natural Convection Circulation in Simple Loops

During the last few years there has been resurgence of interest in natural convection circulation in simple single phase loops. This is in response to a need for solving practical problems and is evidence by recent publications [146, 147, 154-157]. All of the works cited above, except References 147 and 149, are analytical and are primarily concerned with stability and thermal performance of the system.

Creveling et al. [149] have performed an experimental study of natural convection in a toroidal loop and report what appear to be the only fluid friction and heat transfer data. Their results are shown in Figure 19 and 20. The results of Figure 18 show that there is a transition between the laminar and turbulent regimes as evidenced by a change in slope. Note also that the experimentally determined values of f are significantly greater than those predicted by standard correlations for forced flow in a tube. Damerell and Schoenhals [147] performed experiments in a toroidal loop in which the heated (lower) and cooled (upper) portions of the loop were displaced from a horizontal plane. They observed that the flow inside the toroid could become unstable for certain range of heat addition rates and orientation angles. The thermal performance of the loop was predicted from a steady-state momentum balance between wall shear stress and buoyancy. By considering the flow to be fully developed and using laminar pipe flow fluid friction factor ($f = 64/Re$), the circulation velocity was estimated for given thermal conditions in the heated zone as a function of the angular displacement. The trends in experimental data were found to be similar with the predictions, but values of observed flow velocity are significantly lower than the correspondingly analytically predicted values in the range of angles between 0 and 60 degrees.

Natural convection circulation in multiple parallel vertical channels in single-phase [144, 152] and two-phase [158, 159] flow systems has been studied. A three channel configuration was investigated in detail both analytically and experimentally [152]. The results disclosed the existence of metastable regime

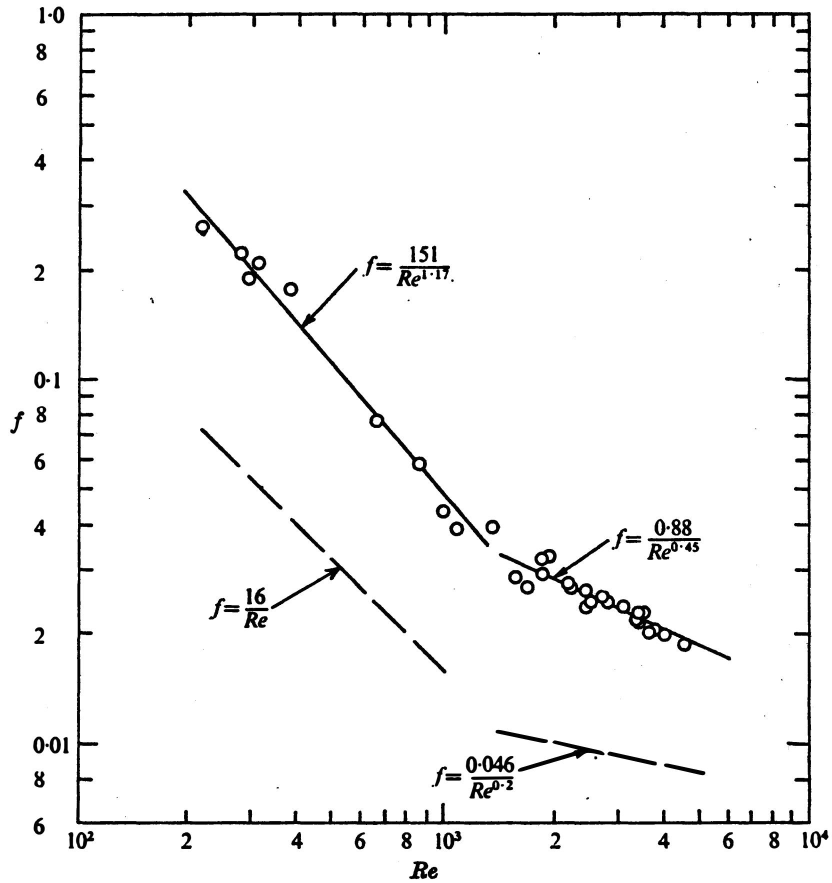


Figure 19. Effect of Reynolds number on friction factor (From Reference 149).

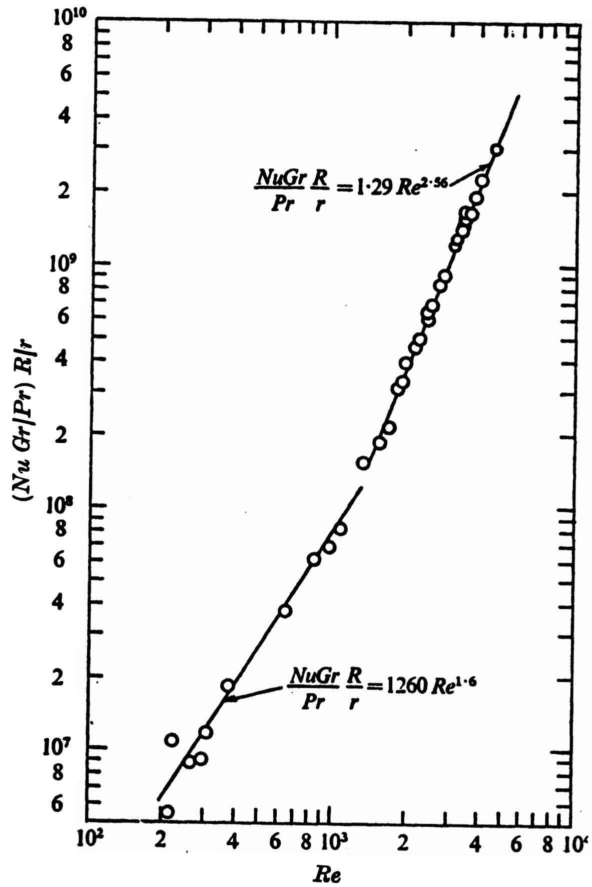


Figure 20. Effect of Reynolds number on heat transfer parameter (From Reference 149).

below a critical heat input where several flow patterns were possible. In the case of the three channel system, two metastable and one unstable arrangement were possible. Transition from laminar to turbulent flow was observed to occur at low Reynolds numbers, which agreed in order of magnitude with the results of other observers [32, 143, 160]. Also, partial reverse flows, and steady oscillations due to periodic boiling were observed in the apparatus. Two phase flow instabilities in forced and natural circulation systems are common and extensive reviews are available [161-163].

During the extended depressurization (hours 6-11) and the repressurization (hours 13-16) periods [5, 6] after the TMI-2 accident the core was partly uncovered and its partial cooling may have been accomplished with the system (e.g. core, primary vessel, piping, steam generator) acting as a thermosyphon. The principle of the two-phase thermosyphon is that heat can be transferred from one location to another by boiling, vapor flow, condensation and condensate return of a working fluid in a closed system. A very simple thermosyphon is a vertical sealed tube filled with a working fluid such as water. There has been considerable interest in thermosyphons during the last three decades because of their applications to many technologically important systems [164-168] and an extensive review of literature is available [169]. A number of studies have reported heat transfer data for steady operation of geometrically simple, two-phase thermosyphons [170-173], but these data are not likely to be relevant to the complex TMI-2 geometry and conditions. For example, results of an experimental study of boiling heat transfer in closed two-phase thermosyphons as functions of thermosyphon geometry, working fluid, internal pressure and percent filling with a working fluid have been reported by Bezrodnyy and Eleksyenko [172]. Experiments were performed with water, ethanol, methanol, Freon-11, Freon-113 and Freon-12. An empirical equation for predicting heat transfer coefficient for fully developed boiling in a thermosyphon is recommended.

3.3 Modeling of Natural Convection Circulation in Reactor Loops

A number of dynamic and steady-state models [e.g., 150, 151, 175-177] have been reported to determine the heat removal capability via natural convection circulation. The complexity of the thermal-hydraulic simulation of the primary reactor coolant loop and of the entire plant is recognized. The PWR plant, for example, may have two or more parallel loops coupled through the inlet and outlet plena of the reactor vessel. Some of the transients may be simulated by considering only a single loop provided that all loops are physically identical to each other and that the multi-dimensional effects in the common regions are negligible. All of the essential components, such as the reactor core, the plena, steam generators, pumps, pipings, valves, etc. need to be included in the simulation model. The degree of modeling sophistication required, however, can differ from one component to another. Detailed accounts of modeling efforts and results are given in the literature cited.

At this point it should be emphasized again that the basic information necessary for constructing realistic natural convection circulation models even in simple loops, not to mention the entire reactor loop, namely the natural convection fluid friction and heat transfer data for internal flow, are not available. These data are lacking for all internal geometries and rod bundles with laminar, transition and turbulent flow. The entrance length and range of critical Reynolds (Grashof) number have to yet be established even for simple geometries. Because of power skew, flow in some regions of the core (subassemblies) will be controlled by natural convection while others will behave as if a pressure difference (e.g. resulting from a net buoyancy force) has been imposed. This will result in a nonuniformity of flow rate and outlet temperature from one subassembly (region) to another across the core. Nonuniformities in heating and flow conditions may produce system instabilities. Problems of instability occurring in single and two phase natural circulation flow loops have been addressed [e.g. 159, 160, 178-181]. Experiments have shown that loop conditions, which give rise to inverted density profiles, can cause instabilities of the flow [181].

In order to develop realistic analyses of natural convection circulation in a nuclear reactor core under transient or accident conditions additional basic data are required. Some of these are identified below.

1. Friction Factor and Heat Transfer to Two-Phase Mixtures for Natural Convection

Boiling of water and condensation of steam has been studied extensively for forced but not under natural convection conditions. There are some experimental evidence [174] to indicate that the forced convection correlations for friction factor and heat transfer may not be directly applicable for two-phase flow natural circulation loops. It is necessary to verify if the correlations for forced flow are applicable for natural convection flow.

2. Sudden Expansion and Contraction Losses of Two-Phase Flow Mixture

The primary coolant (steam or steam-liquid mixture) has to pass through system components of differing cross sections. The available contraction and expansion loss coefficients for incompressible flow are unlikely to be valid since the fluid state and velocity variations could be greatly different.

3. Stratification

The coolant after condensation (e.g. in the steam generator in the TMI-2 accident) enters the bottom of the reactor core. The area ratio between the reactor core and the coolant pipe is sufficiently large to cause a sudden drop in coolant velocity and hence to reduce mixing under very low flow or natural circulation conditions. The temperature gradients existing in the reactor core due to power skew or core blockage may further distort the circulation pattern. It is necessary to understand the processes and to determine whether thermal stratification of the coolant can intensify and significantly influence the thermal-hydraulics performance under natural circulation conditions.

4. Blocked Core

Due to thermal disintegration of the fuel rods, the reactor core could be considerably blocked by the fuel debris. It is reported [6] that the central position of the TMI-2 core is practically fully blocked and circulation has to take place through the peripheral subassemblies. This flow blockage condition has to be viewed in addition to the radial temperature gradients that would be present in the core. Friction factor and heat transfer results for such geometric flow paths and temperature conditions are necessary.

5. Parallel Paths

There are several coolant paths connected to a reactor vessel. The buoyancy induced flow would distribute through different paths depending on their density differences and flow resistances. It is believed that once such path resistances are determined for the natural circulation conditions, the flow distribution in the different loops could be estimated.

6. Pumps

The stalled pump is likely to be the most severe source of flow resistance to buoyancy induced circulation. The resistances need to be determined for both single and two phase (e.g. water-inert gas mixture) flow conditions. Depending on the estimation of such flow resistance one would wonder if a provision for bypassing the high resistance paths should not be incorporated in the reactor system design. For example, if the stalled pump provides high resistance, the net buoyancy force may not be sufficiently large to continue the circulation. On the other hand, if the pump could be bypassed by a lower resistance path, the circulation would continue. Such bypasses could be brought into play manually or automatically when the core is cooled by natural circulation.

4. RECOMMENDATIONS FOR "BEST" AVAILABLE CORRELATIONS

After the literature review was completed, the question was raised as to which of the correlations are the most reliable and thus should be used for predicting convective heat transfer coefficients at very low forced flow rates and for natural convection conditions. In order to answer the question the available correlations were critically compared (taking into account the uncertainty of the data base) and the "best" correlations (i.e., those which appear to give the most reliable estimtates of convective heat transfer) are recommended in this section.

It should be emphasized that correlations for predicting local and/or average heat transfer coefficients in rod or tube bundles for longitudinal, very low flow conditions (with combined forced and natural as well as natural convection) where radial mixing is possible and rod spacers are present, could not be found in the literature. However, convective heat transfer correlations for turbulent flow in annuli, which could be considered as modeling longitudinal flow in rod bundles, are well documented [182]. Unfortunately, annular flow heat transfer correlations applicable for combined and natural convection conditions have not been reported. Therefore, a data base for longtidiual, very low flow rates in rod bundles must be developed and heat transfer correlations obtained. The equations for predicting convective heat transfer to be presented (for limiting but related geometries) are reasonably well established and could be used with some confidence. However, even for these geometrices, a more detailed and relevant data base needs to be developed and correlations updated as data becomes available.

4.1 Combined Forced and Natural Convection

Empirical convection heat transfer coefficient correlations for combined convection in tube or rod bundles with rod spacers in either upflow or downflow have not been found in the literature. Therefore, only related results for external flow over vertical plates and cylinders and for internal flow in vertical tubes are given here.

4.1.1 Combined Convection From Vertical Plates

For combined convection over vertical plates in laminar, aiding flow the local heat transfer can be predicted from empirical correlation presented by Churchill [58] which is of the form,

$$\left[\frac{Nu_x f_F(Pr)}{A_F Re_x^{1/2}\, Pr^{1/3}} \right]^3 = 1 + \left[\frac{A_N Ra_x^{1/4} f_F(Pr)}{A_F Re_x^{1/2} Pr^{1/3} f_N(Pr)} \right]^3 \qquad (47)$$

with

$$f_F(Pr) = [1 + (C_F/Pr)^{2/3}]^{1/4}$$

$$f_N(Pr) = [1 + (C_N/Pr)^{9/18}]^{4/9}$$

The values of constants A_F, A_N, C_F and C_N recommended are:

Uniform T	0.339	0.503	0.0468	0.492
Uniform q	0.464	0.563	0.0205	0.437

Equation (47) is based on a rather limited data base and is expected to be limited to approximately $Re_x < 10^4$ and $Ra_x < 10^9$, due to the onset of turbulent motion.

As an approximation, the correlation given by Eq. (47) can also be used for flow along vertical cylinders as long as curvature effects may be neglected (i.e., for cases where the boundary layer thickness is much smaller than the radius of the cylinder). The average heat transfer coefficient over a height L of a plate or cylinder can be obtained by averaging the local values.

Correlations for combined convection in turbulent, aiding flow along an isothermal plate apparently have not been reported. However, as an approximation, one might adopt the procedure suggested in the literature [183] for parallel flows, e.g.,

$$Nu_c^n = Nu_f^n + Nu_n^n \tag{48}$$

where the exponent n may have values $2 \leq n \leq 4$ and Nu_c, Nu_f and Nu_n are average Nusselt numbers for combined, forced and free convection, respectively. This type of procedure has been employed successfully for correla;ting data [58,84].

4.1.2 Combined Convection in Vertical Tubes

The empirical correlations of Petukhov and Strigin [184] and of Petukhov [185] for combined convection in vertical tubes have been critically compared. Based on this comparison, the more recent correlations of Petukhov [185] are recommended for use in the prediction of convective heat transfer coefficients for both upward and downward flow in vertical pipes. The correlations are as follows:

1. Upward Flow [$300 < Re < 3 \times 10^4$, $5 \times 10^3 < Ra < 8 \times 10^6$, $3 < Pr < 6$, $(x/D) > 40$]:

$$\frac{Nu}{Nu_T} = [1 + 720(Ra/Re^2)]^{-1} \quad \text{for } Ra/Re^2 \leq 10^{-4} \tag{49}$$

and

$$\frac{Nu}{Nu_T} = 3.97(Ra/Re^2)^{1/3} \quad \text{for } Ra/Re^2 > 10^{-4} \tag{50}$$

2. Downward Flow ($300 \leq Re \leq 2.5 \times 10^4$, $5 \times 10^3 < Ra < 13 \times 10^6$, $2 < Pr < 6$):

$$\frac{Nu}{Nu_T} = [1 + 0.031(Ra/Re)]^{1/3} - 0.15 \exp\{[-2[(Ra/Re)-8]^2\} \tag{51}$$

and if $(Ra/Re) > 16$

$$\frac{Nu}{Nu_T} = [1 + 0.031(Ra/Re)]^{1/3} \tag{52}$$

In Eqs. (49) through (52) the Nusselt number for turbulent forced convection, Nu_T, is given by

$$Nu_T = (f/8)(Re\ Pr/K) + 12.7\sqrt{f/8}\ (Pr^{2/3}-1) \tag{53}$$

where

$$f = (1.82 \log Re - 1.64)^{-2} \tag{54}$$

and

$$K = 1 + 900/Re \tag{55}$$

The experimental Nusselt number data are within about $\pm$ 10% of the values predicted by the correlations, and the fluid friction data are within $\pm$ 30% of values predicted by Eq. (54). Note that the same correlations cover the laminar, transition and turbulent flow regimes. This is due to the fact that in the presence of substantial buoyancy forces (e.g., sufficiently large Rayleigh numbers) the flow is expected to be turbulent [186, 187] for a Reynolds number even as low as about 2000.

4.2 Natural Convection

Neither emperical nor analytical correlations for predicting natural convection heat transfer in tube or rod bundles have been found in the literature. Therefore, only related results for external flow over vertical plates and cylinders and for internal flow in vertical channels will be given. It is emphasized that free convection over an external body (e.g. cylinder) immersed in a fluid of infinite extent is not expected to realistically model heat transfer from a rod in a closely spaced bundle because boundary layers formed on adjacent rods will interact and a mixing of flow between rods in a bundle is expected to take place.

4.2.1 Natural Convection from a Vertical Plate

For laminar flow along an isothermal vertical plate the average Nusselt number is correlated [183] by the equation,

$$\overline{Nu}_L = 0.68 + \frac{0.670\ Ra_L^{1/4}}{[1 + (0.492/Pr)^{9/16}]^{4/9}} \tag{56}$$

This equation provides a good representation for $Ra_L < 10^9$ and for fluids such as mercury, air, water, glycol, and other Newtonian-fluids, thus covering a wide range of Prandtl numbers.

For distances along an isothermal vertical plate upon which the flow is at first laminar and then becomes turbulent, the average Nusselt number can be best correlated by the empirical equation

$$\overline{Nu}_L = \{0.825 + \frac{0.387\ Ra_L^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}}\}^2 \tag{57}$$

The superiority of this equation for $Ra_L > 10^9$ over Eq. (56) is somewhat obscured by the lack of data for large Rayleigh numbers ($Ra_L > 10^{12}$).

For a uniformly heated wall, the equation

$$\overline{Nu}_L = \{0.825 + \frac{0.387\ Ra_L^{1/6}}{[1 + (0.437/Pr)^{9/16}]^{8/27}}\}^2 \tag{58}$$

correlates experimental data for all Prandtl and Rayleigh numbers which include both a laminar and turbulent flow regime. It should be noted that for the uniform heat flux correlation given above the Rayleigh number, Ra_L, is based on the temperature difference between the free stream and the midpoint of the plate, rather than on the heat flux at the wall (as is customarily done in the literature).

In the TRAC computer program [89] Eq. (30) is used to predict the natural convection heat transfer coefficient from a vertical plate. For Pr = 1 and $Ra_L = 10^{12}$, Eqns. (30) and (57) yield $\overline{Nu}_L = 1300$ and $\overline{Nu}_L = 1239$, respectively. This 6 percent difference is within the accuracy of experimental data and of the correlating equations.

4.2.2 Natural Convection From Vertical Cylinders

The following correlations, due to Martynenko et al. [92], are recommended for predicting the average natural convection heat transfer coefficient from a vertical, isothermal cylinder of diameter D and length L:

$$\overline{Nu}_L/Ra_L^{1/4} = 0.53\ Pr^{0.05} + 0.68\ (Ra_L^{-1/4} L/D) \text{ for } Ra_L^{-1/4}\ L/D < 2 \tag{59}$$

and

$$\overline{Nu}_L/Ra_L^{1/4} = 0.87\ (Ra_L^{-1/4} L/D)^{0.87} \text{ for } Ra_L^{-1/4}\ L/D > 2 \tag{60}$$

These equations are appropriate for constant property fluids which have Prandtl number in the range of 0.01 to 100.

For uniformly heated, vertical cylinders the correlations suggested by Nagendra et al. [96] for predicting the average heat transfer coefficient are recommended and are as follows:

$$Nu_D = 1.37\ (Ra_D \frac{D}{L})^{0.16} \quad \text{for long cylinders when } 0.05 < Ra_D \frac{D}{L} < 10^4 \qquad (61)$$

and

$$Nu_D = 0.6\ (Ra_D \frac{D}{L})^{0.25} \quad \text{for short cylinders when } Ra_D \frac{D}{L} > 10^4 \qquad (62)$$

The very limited experimental data available is in good agreement with the above correlations.

4.3 Combined Natural Convection and Radiation

Analytical and very limited experimental studies on combined natural convection and radiation suggest that at the temperature, pressure, and opacity conditions to be expected in water cooled nuclear reactors, the interaction between convection and radiation heat transfer may be justifiably neglected. That is, convection heat transfer can be assumed to be independent of that by radiation, and likewise, the radiation transfer can be considered to be independent of that by natural convection. Under such assumptions the total heat transfer at a surface may be estimated by summing the contributions due to the combined natural convection only model and the radiation only model.

5. SUMMARY AND RECOMMENDATIONS FOR ESTABLISHING LACKING DATA BASE

A review of the heat transfer and fluid friction data base at low flow and natural convection conditions has been conducted. Based on the survey areas of research needs have been identified, but no priorities are implied by the sequence of topics listed.

1. Heat transfer and fluid friction for combined convection in upward laminar flow in tubes and rod bundles simulating light water nuclear reactor transient and accident conditions should be thoroughly investigated. The velocity profile appears to be influenced more by buoyancy than the temperature profile. The combined effects of velocity and temperature profiles on the local bulk (mixed mean) temperature have not been carefully delineated. As a consequence, there appears to be some inconsistencies in the reported results, e.g. the wall heat flux is decreased but the heat transfer coefficient is increased.
2. Combined turbulent convection heat transfer in tubes, not to speak of rod bundles, is not completely understood. For example, the effect of buoyancy in turbulent downflow is to increase h but to decrease f. This is contradictory to Reynolds analogy, and possibly the analogy should not even be expected to hold for combined convection? Transition from laminar to turbulent flow has not been studied and the development length has not been established. The relevant scaling parameters need to be determined. The available results are contradictory, e.g. fluid friction data show an inflection point with Ra/Re^2 whereas the same is not reflected in heat transfer data.
3. Neither theoretical predictions nor experimental data are available

for internal natural convection (e.g. channels or rod bundles) at sufficiently large temperature (surface to fluid) differences when physical property variations must be accounted for. Both fluid friction and heat transfer results for laminar and turbulent flow are needed. The concept of hydraulic diameter for natural convection in internal flow does not appear to work. What should be the relevant characteristic length? Research needs to be performed to either validate the concept or establish appropriate procedures for correlating and presenting data.

4. Combined natural convection and radiation heat transfer to high pressure steam in a rod bundle needs to be modeled analytically and verified experimentally. Radiation heat exchange between rods and container wall, between rods and steam vapor-water droplet mixture have to be measured and theoretical models verified for rod bundles having different P/D ratios and thermal heating conditions.

5. There is no heat transfer or fluid friction data on partially blocked channels or rod bundles. Fluid friction is particularly important under natural convection conditions. Such data are essential if heat removal from a partially or severely damaged core is to be modeled.

6. Natural convection circulation in single and two-phase flow loops is not completely understood, and the heat transfer and fluid friction data base required for predicting the flow rate in the primary coolant loop and heat removal from the core is incomplete. Simple laboratory experiments rather than full scale tests are needed to obtain basic data. Friction factor data under natural convection conditions due to area reductions and expansions are required. The

losses are irreversible and for a two-phase mixture are going to change fluid flow, steam quality, void fraction and the velocity. Effects of blockage of core on stability of the natural circulation loop need to be determined and the secondary (local) circulation produced by blockage on the heat removal from the core needs to be studied.

7. The effects of inert gases on natural convection circulation in a loop have not been studied and are not understood. Continued accumulation of gases in the primary loop could partly block the circulation and reduce heat removal from the core. The transport and fate of inert gases need to be understood in order to develop procedures for dealing with their presence in the core and the primary loop under accident conditions.

8. At the liquid-vapor interface of a partly voided core (TMI-2) or during reflooding,heat transfer is by convection and radiation as well as the radial and axial conduction along the heat generating fuel rod. Because of the sharp temperature gradients along the fuel rods axial conduction may be important in datermining the position of the "void" or "reflooding" front. Both experimental and analytical studies are needed to understand the "conjugate" problem.

9. Experimental data are lacking for dry and wet steam at low flow rates and/or large surface to bulk temperature differences. The conditions (e.g., surface temperature, pressure, characteristic distance, water droplet concentration and size, etc.) under which radiation may be a significant mode of heat transfer need to be established experimentally and analytically. This is essential for developing understanding and predictive models to cope with nuclear reactor accidents where core is partly voided (TMI-2).

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