# Convective Flow Boiling In Coiled Tubes

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## ABSTRACT

Convective flow boiling of liquid nitrogen has been investigated using coiled-tube heat exchangers in order to determine the heat transfer characteristics. This project focused on the identification of the primary variables involved in establishing the flow boiling coefficient for nitrogen using a coiled-tube flow configuration.

The need for an accurate predictive correlation for this coefficient was recognized after an extensive testing program yielded results that did not correspond well with published information. A test method was developed which allowed for the accurate measurement of the boiling coefficient over a wide range of nitrogen flow rates, excess temperatures ($T_{wall}$ - $T_{sat}$), and tube coil geometries.

In evaluating nitrogen boiling rate data using a large number of flow conditions and coil geometries, it was found that vaporization rates are a function of three main variables. These variables are:

(1) Mass velocity of the liquid nitrogen inside the tubes;

(2) Excess temperature involved in the boiling process;

(3) Coiled-tube geometry, specifically the ratio of the inside tube diameter to the average coil diameter.

A correlation has been developed which accounts for these variables. This correlation has been shown to be accurate in predicting the flow boiling coefficient within 10% for all cases tested.

## INTRODUCTION

Graham Manufacturing makes coiled-tube heat transfer equipment (Heliflow heat exchangers) which is frequently utilized as cryogenic vaporizers, with 80% of this service dedicated to liquid nitrogen. This coiled-tube design is based on spiral coils held together between two flat surfaces. One of these surfaces is the base plate; the other is the end of the shell (or casing). Bolted together, the plate and shell confine a closed spiral-shaped fluid circuit outside the coil. Figure 1 illustrates a Heliflow configuration.

Previously, sizing of these Heliflows for liquid nitrogen use has been accomplished by utilizing nitrogen boiling data established using tube side boiling in a test Heliflow, with steam introduced on the shell side as the heating medium. It has generally been accepted that the vaporization rates were solely a function of the mass velocity of the nitrogen. The validity of this approach has been questioned for some time, but the procedure has remained in use due to lack of further information on the subject.

A literature search was conducted in an attempt to (1) verify the accuracy of the available heat transfer data, and (2) establish a reliable analytical approach for predicting boiling heat transfer coefficients. This search has yielded a number of interesting facts.

First, the heat transfer rates used to evaluate the Heliflow coiled-tube heat exchangers compared favorably with empirical data presented by others for forced convection film boiling in similar applications. Depending on the nitrogen flow rate, values of 20 to 150 Btu/hr-ft$^2$-°F have been reported in the published literature (References 1, 5, 6, and 7).

Second, in almost every case reported in the literature, an attempt was made to establish a suitable analytical correlation for the prediction of the boiling coefficients. In all of the literature examined, none of these correlations have been shown to have much reliability for convective flow boiling in coiled tubes. Often the author would try to establish an empirical approach by modifying the classic Dittus-Boelter relationship, or variations thereof. This approach has been reported to be in error by factors of as much as 4 to 26 (Reference 7). Initially our testing and evaluation for boiling rates followed the same general approach, and, like others, our attempts at predicting heat transfer coefficients using the Dittus-Boelter equation yielded inaccurate results. This approach was abandoned, and a new approach was established based upon the wide range of boiling data that has been collected during extensive testing.

Finally, indications have been reported in the literature that nitrogen vaporization rates might be a function of several variables other than the mass velocity of the liquid nitrogen. These variables could include such factors as the temperature of the system and the geometry of the coiled-tube heat exchanger.

## CONTROLLING FLOW INSTABILITIES

The phenomenon of unstable operation in cryogenic vaporizers is well documented. Numerous accounts of flow instabilities during vaporization are presented in reports published by cryogenic researchers (References 2, 3, and 4). The problem of surging when using the Heliflow coiled-tube heat exchanger as a cryogenic vaporizer has been prevalent since its inception. The surging experienced is generally a 10 to 50 PSI pressure fluctuation originating near the Heliflow liquid inlet point, and usually of a periodic nature with frequencies in the range of 0.2 to 1.0 cycles per second. It was essential to eliminate (or at least significantly reduce) this flow instability to maintain the high rate of heat transfer available with the Heliflow heat exchanger.

Cryogenic surging appears to be closely related to thermosiphon reboiler instability, which has been documented by Heat Transfer Research, Incorporated (HTRI). Their reports indicate periodic surging for steam vaporization near 0.33 cycles per second for tube side boiling (available from them by requesting Videotape HTRI-2, "Operational Characteristics of Vertical Thermosiphon Reboilers"). The mechanism behind this instability has been shown to be an enormously high heat flux causing unsteady, violent boiling. The surging that develops is thought to be due to feedback between flow rate, vapor volume, and pressure drop. At high excess temperature differences the velocity in the exit piping may reach the two-phase choked flow condition, causing the total pressure drop to exceed the available driving head. When this condition is reached, any additional vapor produced cannot exit the exchanger outlet, and the flow temporarily reverses. This causes a sudden decrease in pressure drop at the outlet, and flow surges back in the normal direction, picking up speed until choke velocity is again reached, and the cycle repeated.

Attempts to eliminate surging in the Heliflow focused on three main areas; heat transfer, hydraulics, and geometry. An extensive testing program ultimately resulted in solving the flow instability problem. Based on the results of this testing, it became apparent that the most effective method of controlling the flow instability in the Heliflow was to utilize an accumulator prior to entering the coiled tubes. This device is relatively inexpensive, has an insignificant pressure drop, and requires no alterations to existing designs. Figure 3 shows the effects that using an accumulator had on a typical flow instability situation.

## TEST APPARATUS

The equipment used for measuring the nitrogen vaporization rates is shown in Figure 2. For all cases run during this testing program, liquid nitrogen ($LN_2$) vaporization was accomplished using tube side boiling. Depending on the tube wall temperature desired, heating was achieved by using steam, water, ethlyene glycol, or methanol on the shell side.

$LN_2$ is stored in, and delivered from, a 1,600 gallon tank. The quality of the $LN_2$ entering the vaporizer is controlled through the use of a subcooler. This subcooler can deliver 5¶ of subcooling at 100 PPH, and up to 15°F of subcooling at 800 PPH. The subcooler is used in this system to ensure that the $LN_2$ is not being introduced in a mixed-phase condition. The intent is to assure that all of the surface area of the tubes is utilized in "boiling" the $LN_2$, not in increasing the sensible heat of vapor already present. An accumulator is introduced in the liquid nitrogen supply line just prior to the Heliflow inlet to act as a surge suppressor, allowing the vaporizer to operate under steady state (non-oscillatory) conditions.

Cryogenic temperature measurements are made with "T" type thermocouples, with all remaining temperature measurements taken using conventional RTDs. All pressure measurements are taken using absolute pressure transducers, backed up by direct reading pressure gauges. Pressure transducers in the cryogenic regime are insulated from the cold fluids by the use of stainless steel pig-tails. The data collected is sent to an electronic data acquisition system (DAS) and is used to calculate heat duties. flow rates. and other performance parameters.

Steam, ethylene glycol, and methanol shell side flow rates are measured using differential pressure across flow orifices. This pressure information is used by the DAS to calculate flow rates. Shell side water flow rates are indicated by high accuracy rotameters and manually input into the DAS. Nitrogen flow rates are determined at a location downstream from the primary vaporizer. After vaporization in the Heliflow, the nitrogen enters a second heat exchanger to superheat the vapor prior to flow measurement across a conventional critical orifice. Pressure and temperature before the orifice, and pressure after the orifice, are input to the DAS to calculate a flow rate.

An array of Heliflows permitted a systematic evaluation of boiling rates based upon the combined geometrical effects of tube diameter, tube length, and average tube bundle (helix) diameter. Each Heliflow could be fitted with a clear-ended casing when the test being conducted used water or ethylene glycol, which allowed for a visual evaluation of the shell side flow. Of prime importance in the shell side flow is the flow velocity distribution, and the subsequent formation of ice in low velocity areas. For all cases, if icing is present, it can be kept to a negligible amount by varying flow velocity and/or temperature to allow for the effective rating of each exchanger.

## TEST PROCEDURE

Measurement of the boiling rates was achieved using nitrogen mass flow rates ranging from 15 to 120 lbs/sec-ft$^2$. For all cases, shell side flow was established prior to introducing liquid nitrogen into the vaporizer. Nitrogen flow was gradually increased to allow for the system piping to cool down, and also to establish an insulating ice barrier on the outside of the piping and accumulator.

The flow rate of the nitrogen was limited to the region where boiling was guaranteed over the entire tube length of the specific model of Heliflow being tested. The lower limit of nitrogen flow was assumed to be at the point where the vapor exiting the exchanger was 20°F to 30°F higher than the saturation temperature for the given nitrogen exit pressure. The upper limit for the nitrogen flow rate was assumed to occur at the point where the shell side duty began to indicate less heat transfer than that which was required to vaporize the nitrogen.

In addition to the array of Heliflows tested, 14 different tube wall temperatures were examined during this investigation. The three coldest tube wall conditions were established by using methanol on the shell side at temperatures of -115°F, -90°F, and -60°F. Three intermediate tube wall conditions were established using ethlyene glycol on the shell side at temperatures of -30°F, 0°F, and 30°F, and four more using water on the shell side at temperatures of 60°F, 90°F, 140°F, and 190°F. Finally, four elevated tube wall conditions were established by condensing steam at atmospheric pressure (212°F), 20 PSIG (257°F), 50 PSIG (296°F), and 120 PSIG (350°F). In general, due to the relatively low boiling rate of liquid nitrogen, which is in the range of 20 to 100 Btu/hr-ft$^2$-°F compared to the shell side rate of 1000 to 2000 Btu/hr-ft$^2$-°F, the inside tube wall temperatures were always approximately 10°F colder than the shell side fluid temperature.

Several mass and energy balance checks were employed during the testing process to assure that the data taken was reasonable. The primary check was the energy balance between the tube side (liquid nitrogen) and the shell side (methanol, ethylene glycol, etc.). Whenever the imbalance reached 5%, the test was rejected, and then repeated. A secondary check was done to affirm that boiling occurred over the entire length of the Heliflow tubes. Again, in the event that this did not happen. the test was rejected, and then repeated. Finally, the clear-ended casing allowed visual inspection to assure that excessive ice was not forming on the outside tube surfaces.

## TEST RESULTS

The results of the extensive testing program that was carried out involved working with many Heliflow models operating under various conditions. Without a clear idea of the form that the data was going to take, our initial effort was to attempt to relate the measured heat transfer rates to the mass flow of the nitrogen passing through the Heliflow tubes. This endeavor followed the generally accepted form of the Dittus-Boelter equation, where the heat transfer coefficient “h” is a function of the flow characteristics through the tube, as well as the properties of the fluid. As has already been described, the actual data collected during this testing program did not correspond to the above approach. Figure 4 is a typical data curve which would be measured when testing any of the Heliflows. This particular Heliflow was fabricated with copper tubes, and had a tube wall temperature of 185°F.

The most obvious feature of this data curve is that it is a straight line, which means that the preferred form of the correlation should be:

$$N_{NU} = C1 \times N_{RE} + C2$$

Further testing with other Heliflow models at varying operating conditions revealed that the correlation was not quite so simple. Ultimately it was discovered that the liquid nitrogen boiling rates were a function of three main variables. These variables are:

(1) Mass velocity (Reynolds number) of the liquid nitrogen inside the tubes;

(2) Tube wall temperature;

(3) Heliflow geometry - specifically the ratio of the inside tube diameter to the average helix diameter.

The variables described above affect the heat transfer characteristics of the Heliflow in the following manner:

## MASS VELOCITY

Boiling heat transfer coefficients increase as the mass velocity of the liquid nitrogen through the tubes increases. This increase in the boiling coefficient is essentially a linear function of the mass velocity. Figure 4 illustrates this linear relationship.

## TUBE WALL TEMPERATURE

Boiling heat transfer coefficients decrease as the tube wall temperature increases. This decrease in the boiling coefficient is an exponentially decreasing function of the tube wall temperature. This is shown in Figure 5 for a Heliflow having a constant mass velocity of 80 lbs/sec-ft$^2$. One possible explanation for this phenomenon is that an insulating vapor blanket forms at the tube wall around a liquid core at the center. This vapor blanket would become thicker as the tube wall temperature increases. The result would be an increase in thermal resistance to heat transfer through the fluid.

## HELIFLOW GEOMETRY

Boiling heat transfer coefficients exponentially increase as the ratio of the inside tube diameter to the average helix diameter (A/R) increases. This phenomenon is shown in Figure 6 for several different coiled-tube configurations. One reason for this enhancement of heat transfer may be due to centrifuging of the liquid core toward the tube wall as the fluid continuously changes direction in the spiral bundle. When the liquid core approaches the tube wall, this increases the tendency for nucleate boiling to occur, which would greatly enhance the heat transfer rates. We would expect that increasing the A/R ratio would increase the centrifugal force experienced by the fluid for a given mass velocity, and correspondingly improve the heat transfer capabilities.

In order to account for the fact that the tube wall temperature and the Heliflow geometry affect the overall heat transfer characteristics, it was necessary to introduce modifications to our basic approach. Basically, the slope of the linear ratio between the Nusselt Number and the Reynolds Number changes as the temperature and geometry change.

The correlation described in this paper takes the form

$$N_{NU} = \text{“M”} \times N_{RE}$$

where the Reynolds Number is evaluated using vapor properties associated with the film temperature $\{(T_{sat} + T_{wall})/2\}$ and average pressure over the boiling region. The constant “M” can be obtained by using the graph in Figure 7. This graph requires knowledge of the coiled-tube A/R ratio and the tube wall temperature. This correlation has been shown to be accurate to within 10% for all cases tested at 60 PSIA. (Limited testing has been performed at an elevated pressure of 160 PSIA. At this higher pressure there is a decrease in the heat transfer coefficient of approximately 8%.)

Limitations in this study include a relatively fixed tube side boiling pressure (60 PSIA), and also tube wall boiling temperatures ranging between -130°F and + 320°F. Rating information for a tube wall temperature of -200°F is presented, but only as an extrapolation of known performance.

## CONCLUSIONS

An extensive testing program has been carried out with the purpose of establishing a method for rating coiled-tube heat exchangers for service as nitrogen vaporizers. Liquid nitrogen boiling coefficients have been measured for a number of Heliflow coiled-tube heat exchangers. As a result of this testing, an analytical formulation for predicting boiling heat transfer rates in coiled-tube heat exchangers has been developed. This technique accounts for the mass velocity of the nitrogen flowing in the tubes, the geometrical considerations of each style of coiled-tube heat exchanger, and the effect of the tube wall temperature. This correlation has been shown to be accurate in predicting the flow boiling coefficient within 10% for all cases tested.

## NOMENCLATURE

| | |
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| C1, C2, M | constants determined by measurement |
| $N_{RE}$ | Reynolds Number |
| $N_{NU}$ | Nusselt Number |
| Tsat | saturation temperature, °F |
| Twall | tube wall temperature, °F |

## REFERENCES


1. Flanigan, V.J., and Park, E.L, 1971, “A Study of Film Boiling of Corresponding-States Fluids Over a Wide Pressure Range with Cylindrical Heaters,” *Advances in Cryogenic Engineering*, Vol. 16, pp 402-411.
2. Friedly, J.C., Manganaro, J.L, and Kroeger, P.G., 1969, “Approximate Criterion for Prediction of Flow Oscillations in Supercritical Fluid Heat Exchangers,” *Advances in Cryogenic Engineering*, Vol. 14, pp 258-270.
3. Park, E.L., Colver, C.P., and Sliepcevich, C.M., 1966, “Nucleate and Film Boiling Heat Transfer to Nitrogen and Methane at Elevated Pressures and Large Temperature Differences,” *Advances in Cryogenic Engineering*, Vol. 11, pp 516-529.
4. Rogers, J.D., 1968, “Oscillations in Flowing and Subcritical Hydrogen,” *Advances in Cryogenic Engineering*, Vol. 13, pp 223-231.
5. Thurston, R.S., and Rogers, J.D., 1967, “Pressure Oscillations Induced by Forced Convection Heating of Dense Hydrogen,” *Advances in Cryogenic Engineering*, Vol. 12, pp 438-451.
6. Timmerhaus, K.D., and Schoenhals, R.J., 1974, “Design and Selection of Cryogenic Heat Exchangers,” *Advances in Cryogenic Engineering*, Vol. 19, pp 445-462.
7. Williamson, K.D., and Bartlit, J.R., 1965, “Experimental Study of $H_2O$-$LH_2$ and $H_2O$-$LN_2$ Heat Exchangers,” *Advances in Cryogenic Engineering*, Vol. 101 PP 375-381.


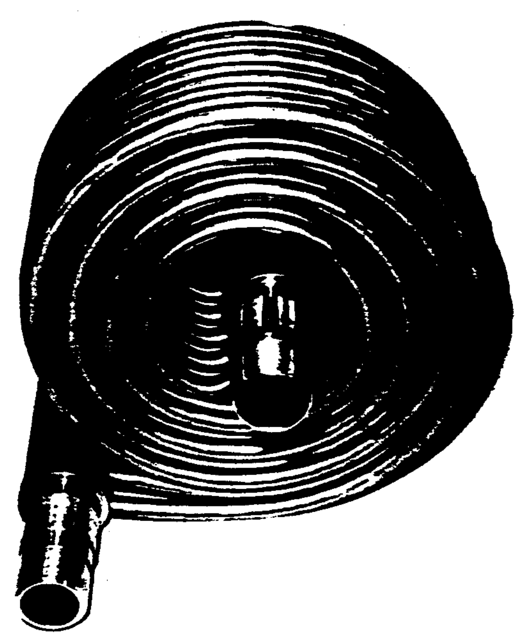

*Fig. 1 Heliflow heat exchanger*

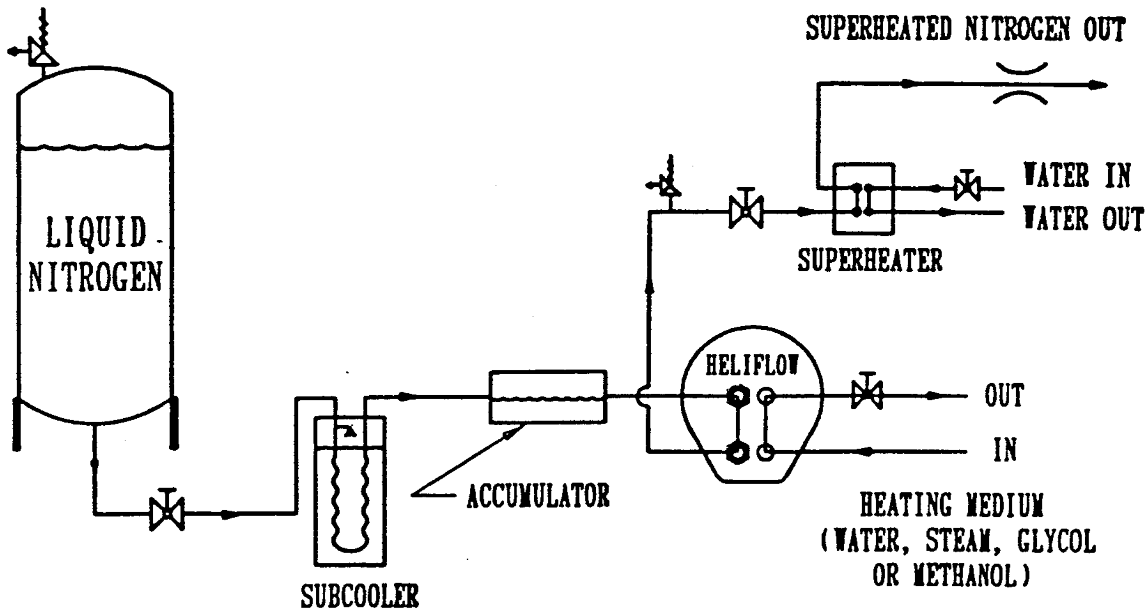


Fig. 2 Flow schematic of $LN_2$ vaporization test apparatus

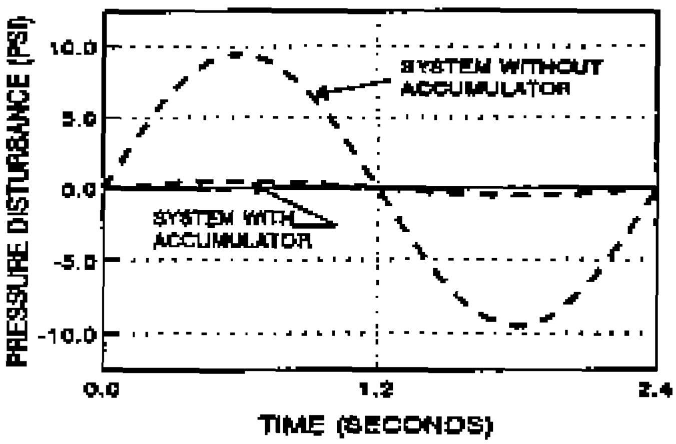


Fig. 3 Effect of accumulator on flow boiling instability

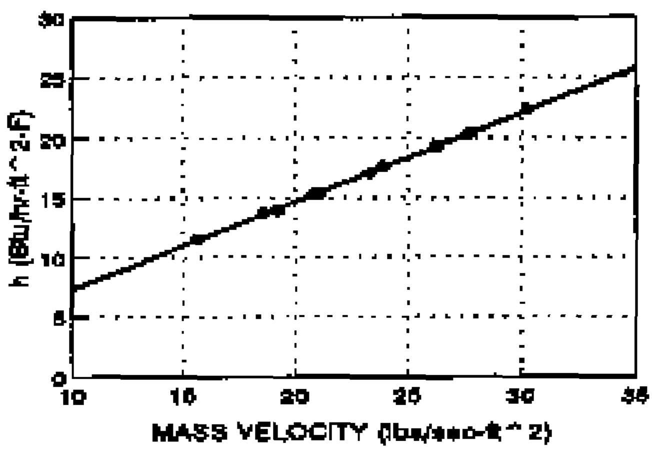


Fig. 4 Effect of mass velocity on the boiling coefficient

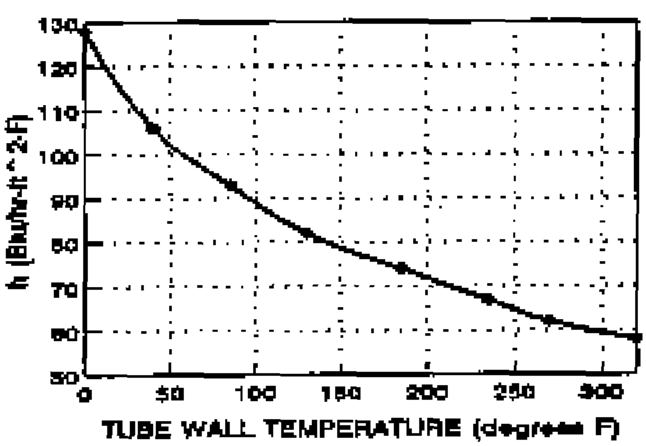


Fig. 5 Effect of tube wall temperature on the boiling coefficient

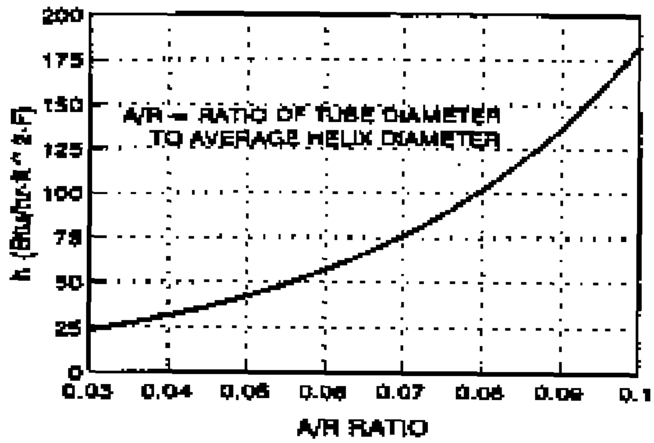


Fig. 6 Effect of coiled-tube geometry on the boiling coefficient

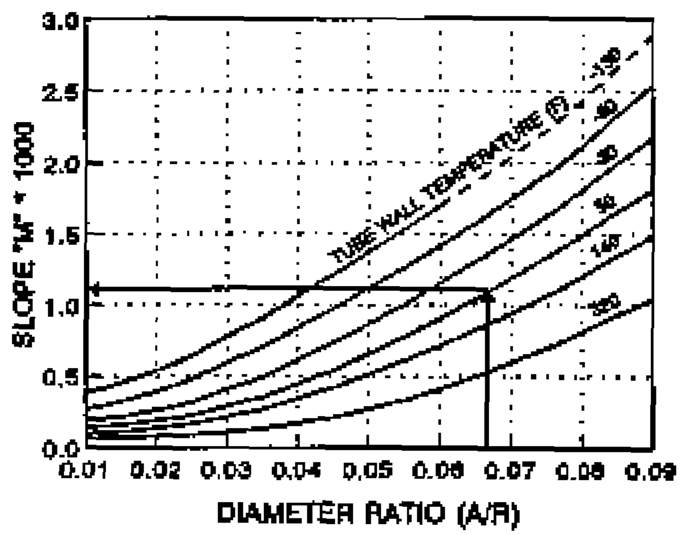


Fig. 7 Determination of Reynolds coefficient "M"