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BASELINE DATA ON FILM COEFFICIENT FOR HEATING ISOBUTANE INSIDE A TUBE AT 4.14 MPa (600 PSIA)

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ABSTRACT

Research designed to obtain heat transfer baseline data on working fluids in geothermal binary cycle systems is described. The experimental apparatus has the capability of providing data under clean conditions to determine inside and outside film coefficient, respectively, for heating and condensation of pure fluids or mixtures of pure fluids being considered for binary systems. The working fluid loop simulates the binary cycle with steam as the heating fluid and a throttling valve instead of the turbine.

In this paper, data on film coefficient for heating isobutane in the supercritical region (critical pressure 3.65 MPa) inside a tube at 4.14 MPa (600 psia) and at various temperatures and flow rates are presented. The isobutane was heated in a horizontal, type 316 stainless steel, instrumented tube by steam condensing on the outside. The tube was fitted with a total of fifteen thermocouples imbedded in the wall of the tube at 45 deg radially above a horizontal plane through the axis at five stations located equally along the length of the tube. The inside and outside wall temperature of the tube at each of the five stations was calculated from the location of the imbedded thermocouples and their temperatures. The heat rate was determined by measuring the rate of condensing steam on the outside of the tube under each of four sections by means of specially designed vapor-traced meters. The temperature of the isobutane inside the heater tube was measured by means of a traveling platinum resistance thermometer equipped with a mixing head at its tip and the temperature of the steam in the steam chest outside of the tube was measured by means of a platinum resistance thermometer.

Film coefficients for the heating of isobutane in the range of 0.6 to 2.5 kW/m^2C (115 to 400 $Btu/hr\ ft^2F$) were obtained for Reynolds number between 3×10^4 to 2×10^5. A plot of these data on log-log paper showed that they are correlated by the empirical equation

$$Nu = 0.022\ Re^{0.82}\ Pr^{0.4}$$

INTRODUCTION

In a geothermal power plant using the binary cycle, the cost of heat transfer equipment, principally to heat and condense the working fluid, accounts for approximately half the capital cost (1). Inexact estimate of the heat transfer coefficients for heating, cooling and condensing the working fluids can have serious consequences. If the estimated coefficients are too high, the plant may fail to meet its performance guarantee; if much too low, the plant will be overdesigned and wasteful.

In specifying heat transfer equipment for a process, the designer usually has information on the performance of equipment under similar or identical conditions. In specifying heat transfer equipment for a binary cycle geothermal power plant using isobutane as the working fluid, the engineer must obtain the needed information from literature. In the absence of data specifically applicable to the design at hand, engineers refer to heat transfer coefficient prediction methods. These general correlations are based on fluid transport properties and give the mean, or most probable value of a large amount of reliable experimental data. However, because of the influence sometimes exerted on heat transfer by minor and frequently unrecognized variations in conditions, a designer can only be reasonably sure that the performance of an individual heat exchanger will be within, plus or minus, 35 percent of the mean. Also, the correlation equations for the prediction of the heat transfer coefficients need accurate and reliable data on the transport properties of the working fluids at the operating conditions. Lack of data at these conditions makes the designer's task difficult at best. This lack of data is especially true for designs in which the working fluid is heated in the supercritical region, where computer studies (2) indicate that heating of the isobutane results in high cycle efficiency. Consequently, the most effective and reliable method for specifying heat transfer equipment for a geothermal plant is to determine experimentally the value of the heat transfer coefficient at the proposed operating conditions.

This experiment is a part of an overall program supported by the Division of Geothermal Energy of the U. S. Department of Energy to obtain data for the design of heat transfer equipment for geothermal plants using the binary cycle system. The objective of this experiment was to provide data under controlled conditions with clean surfaces to determine heat transfer film coefficients of various candidate working fluids for heating, boiling and condensation. Because of the conditions of the experiment, the coefficients deduced from the measurements can be considered as baseline numbers for the design of heat transfer equipment using the same fluid and operating at the same conditions. This paper presents data on the heating of isobutane inside an instrumented tube in the supercritical region, critical pressure 3.65 MPa (529 psia), at an absolute pressure of 4.14 MPa (600 psia) with heat flux ranging from 5.8 to 159 kW m^{-2} (1850 to 50,000 Btu h^{-1} ft^{-2}) in Reynold's number range of 2.5×10^4 to 2.3×10^5.

EXPERIMENTAL APPARATUS AND PROCEDURE

Figure 1 is a schematic flow diagram of the

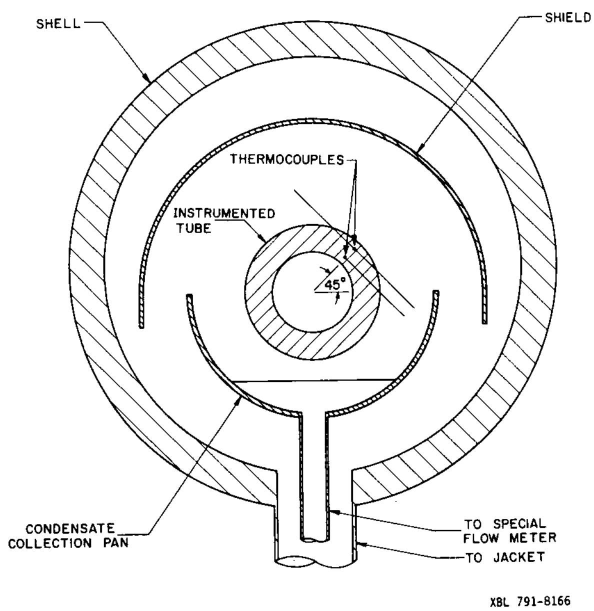

Fig.3 Heater cross section showing tube, condensate tray, drip shield and outer shell

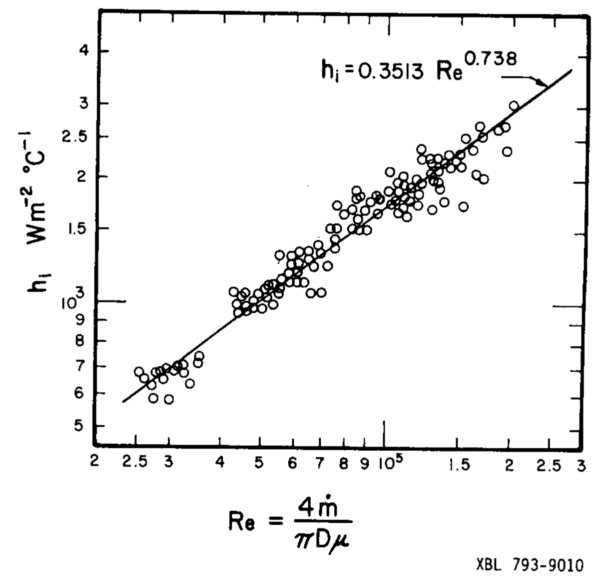

Fig.4 Inside heat transfer film coefficient for heating isobutane at 4.14 MPa Vs Reynolds number for data Set A away from the pseudocritical temperature

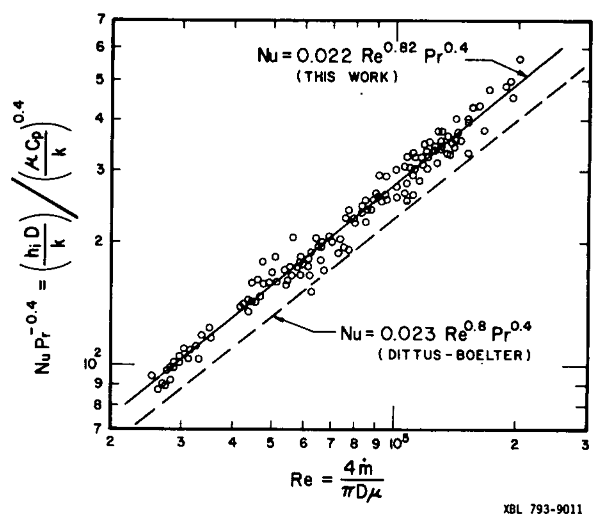

Fig.5 Dimensionless plot of data on heating isobutane at 4.14 MPa vs Reynolds number for data Set A away from the pseudocritical temperature

with water and its calibration was checked with an in-situ calibration system, not shown in Fig. 1, placed upstream from the high pressure pump.

Data were taken and recorded when conditions in the system approached steady state, that is, when cyclic variations in the isobutane flow meter readings, caused by fluttering of the expansion valve, were repeatable and of small amplitudes. The amplitudes were observed to be about one percent at high fluid-flow rates and as high as ten percent at low-flow rates. Time average values of the data were used in the computations. The bulk temperature of the fluid at different locations inside the heater tube was measured by the traveling probe after all other data were taken so that the flow profile of the fluid inside the tube, upstream from the probe, was not affected by the movement of the probe.

RESULTS

The results reported in this paper were obtained for the heating of isobutane, in the supercritical region, at an average pressure of 4.14 ± 0.14 MPa (600 ± 20 psia) and at temperatures ranging between 50 to 160 C (120 to 320 deg F) and for the positions of the imbedded thermocouples at 45 deg from vertical, as shown in Fig. 3. It was observed that as the temperature of the isobutane in the tube were close to the pseudocritical temperature, 142.5 C (228.5 deg F), a decrease occurred in the value of the heat transfer film coefficient. Consequently, in the presentation of the results, the data were divided into two sets: A and B; Set A away from the pseudocritical region and Set B close to the pseudocritical point. Figures 4 and 5 represent the results of Set A, while Fig. 6 represents the results from Set B.

Figure 4 shows a plot of the average inside heat transfer film coefficient, h_i, as a function of the Reynolds number, Re, where $\dot{m}$ is the weight flow rate of isobutane inside the tube, μ is the viscosity of the isobutane at the bulk fluid temperature and D is the inside diameter of the tube. The average film coefficient h_i, for each section was calculated from the equation

$$h_i = \frac{Q}{A\ \Delta T_m}$$

where Q is the heat rate for each section as determined from measurement of steam condensate rate, A is the inside surface area of the tube per section and ΔT_m is the logarithmic mean temperature difference between the inside wall temperature of the tube and the fluid bulk temperature at the inlet and outlet to the section. Figure 5 shows a dimensionless plot of

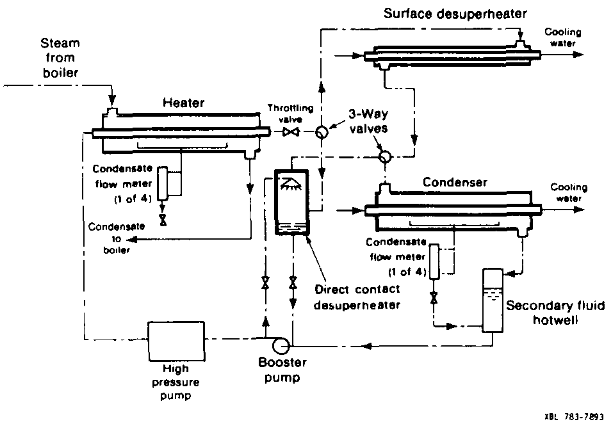

Fig.1 Binary fluid experiment, schematic flow diagram

of the apparatus. The experimental equipment consists of a stainless steel loop simulating a binary system with steam instead of brine as the heating fluid and a throttling valve instead of the turbine. The pressurized secondary fluid (isobutane) is heated inside a single instrumented tube with steam condensing on the outside of the tube. After heating, the fluid expands through a throttling valve and is introduced into a direct contact desuperheater or a surface desuperheater by means of two three-way valves. After desuperheating, the vapor enters the condenser, condenses on the outside of a single instrumented tube identical to that in the heater but containing cooling water inside. The condensed fluid is collected in the hotwell and enters the booster pump where it is slightly pressurized before entering the high pressure variable capacity diaphragm-type pump. After leaving the pump, the pressurized fluid passes through a turbine flow meter, and then into a steam preheater before entering the main heater to close the cycle. During operating with the direct contact desuperheater, a portion of the liquid is diverted to the direct contact desuperheater and the excess is returned to the suction side of the booster pump.

The tubes in each of the heater and condenser were made from a 31.8 mm (1.25 in.) O.D. and 19.1 mm (0.75 in.) I.D.-type 316 stainless steel tube that was cut into two pieces, about 2.7 m (106 in.) long, and honed to an inside diameter of 19.2 mm (0.756 in.). In order to obtain uniform wall thickness, the two pieces were machined and ground to an outside diameter of 30.2 mm (1.189 in.). The concentricity was then checked by ultrasonic measurement of the tube wall thickness at 101.6 mm (4.0 in.) intervals along the axis. At each location, the wall thickness was measured at four points 90 deg apart. These measurements were used to indicate the precise locations of thermocouples imbedded in the wall of each tube. Fifteen thermocouples were imbedded in the wall of the tube at five stations, 609.6 mm (24 in.) apart, with three thermocouples located at each station, as shown in Fig. 2. The inside and outside surface temperatures of the tube at each of the five stations were calculated from radial heat conduction through the tube wall by measuring the temperatures at the known locations of the thermocouples.

The bulk temperature of the fluid inside the heater tube was measured by a traveling probe which consisted of a 6.35 mm (0.25 in.) dia. x 2.54 m (100 in.) long stainless steel sheathed platinum resistance thermometer (RTD) equipped with a mixing head at its tip. Mechanisms and seals were provided

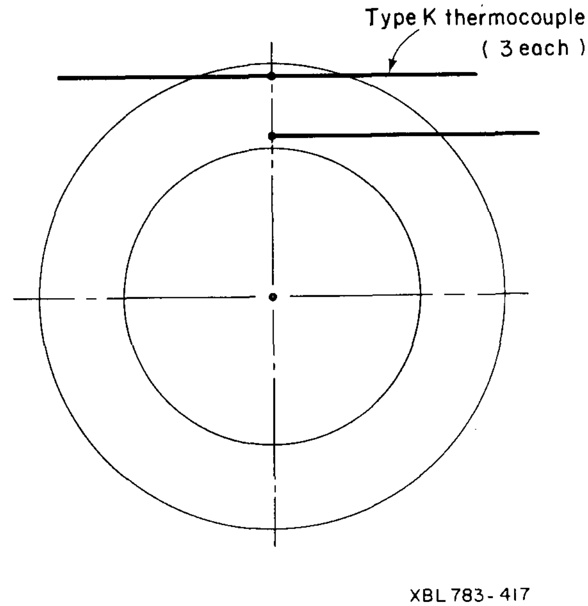

Fig.2 Tube cross section showing location of thermocouples

to move and locate the probe precisely inside the tube and prevent fluid leakage.

The temperature of the condensing vapor outside the tube in each of the heater and the condenser was measured by means of a calibrated RTD and three calibrated type K thermocouples located in the vapor space. The RTD was located at the opposite end from the vapor inlet, while one thermocouple was located at the vapor inlet, the second half-way along the length of the shell, and the third close to the RTD. These three thermocouples were used to check the reading of the RTD and also to indicate the presence of superheat in the vapor. The pressure in the vapor space was measured by means of calibrated pressure transducers and high accuracy Bourdon-type pressure gauges. The pressure measurement was used as a check on the saturated temperature of the vapor and also to detect the presence of noncondensable gases in the vapor space.

The rate of heat input to the isobutane heater and the rate of heat released by the condensing isobutane vapor in the condenser was determined by measuring the rate of condensing vapor on the outside of the tubes at the four sections. This was done by placing a four-section pan under each tube with the five ends of the sections located just underneath the five thermocouple stations. The condensate formed on the outside of the tube dripped into the separate sections of the pan and drained into four specially designed vapor-traced condensate flow meters. The meter consisted of a calibrated volume, a pneumatically operated valve, and a timer to measure the time required to fill the calibrated volume. This timer was operated by photocells to detect the rising condensate surface between two predetermined levels in the meter. Figure 3 shows a cross section of the heater. It shows the external shell, the instrumented tube with thermocouples at 45 deg from vertical, the pan and a hood placed above the tube to prevent any condensate forming on the inside surface of the shell from dripping into the pan.

The volumetric flow rate of the fluid was measured by a turbine flow meter placed downstream from the high pressure pump. The turbine flow meter was calibrated

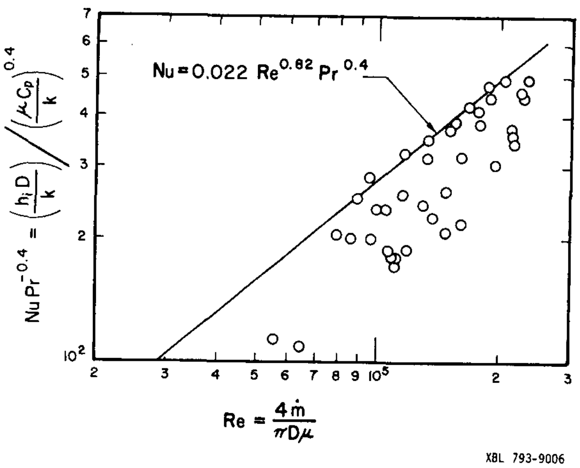

Fig.6 Dimensionless plot of data on heating isobutane at 4.14 MPa vs Reynolds number for data Set B close to the pseudocritical temperature

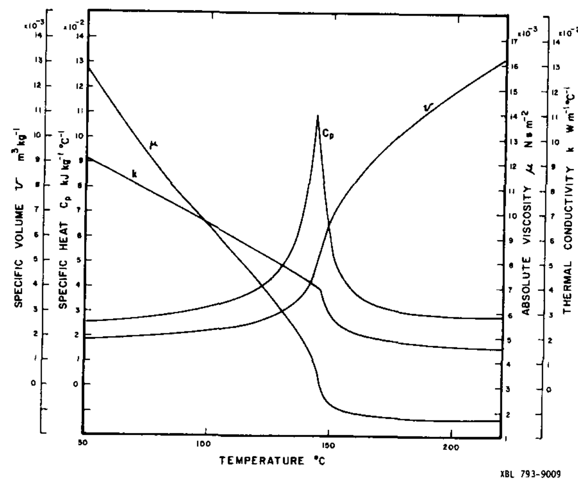

Fig.7 Selected properties of isobutane at 4.14 MPa as a function of temperature (ref. 4)

the data in Set A where k and c_p represent the thermal conductivity and specific heat of isobutane at the average bulk fluid temperature in the section. The dashed line shows the Dittus-Boelter correlation (3) with values of transport properties taken from Hanley (4). The solid line represents the best fit for the data using the least square method also with values of transport properties taken from Hanley.

Figure 6 shows a dimensionless plot of the data in Set B. The solid line represents the correlation for the data in Set A shown in Fig. 5. The scatter in the data is principally due to the heating of isobutane near the pseudocritical point.

DISCUSSION

Figure 7 shows a plot of some of the properties of isobutane at 4.14 MPa (600 psia) as a function of temperature. These properties were plotted from a computer program developed by Hanley (4). The pseudocritical temperature of a fluid at a given pressure is defined as the temperature at which the specific heat, c_p, reaches a maximum value. Figure 7 shows that for isobutane, the pseudocritical temperature at 4.14 MPa (600 psia) occurs at 142.5 C (288.5 deg F).

The results shown in Fig. 6 represent data for operation near the pseudocritical region, and show also that the film coefficient is lower than that predicted from the correlation of data Set A represented by the solid line. In previous work on heat transfer to fluids at supercritical pressure under constant heat flux (5,6), it was found that the film coefficient was a function of $q/\dot{m}$ near the pseudocritical point, where q is the heat flux and $\dot{m}$ is the weight rate of flow. In this work, $q/\dot{m}$ varied along the length of tube because of heating with saturated steam. Under the conditions of this work, it was observed that the film coefficient decreased near the pseudocritical point. The degree of decrease of film coefficient would appear to be related to the driving temperature difference between that of the wall and the bulk fluid and, also, how close the bulk fluid temperature is to the pseudocritical temperature, $t_{p.c.}$. Figure 8 is a replot of some of the results from Fig. 5 and all the results from Fig. 6, showing

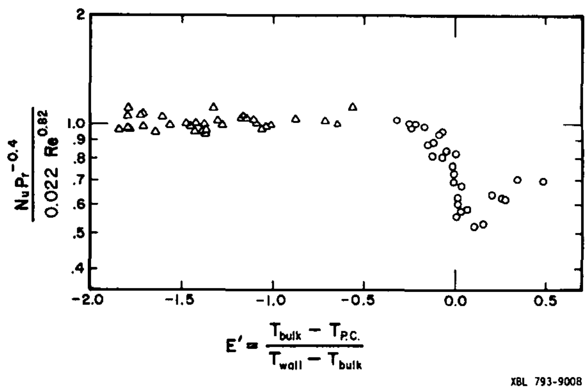

Fig.8 Plot of Nu $Pr^{-0.4}/0.022\ Re^{0.82}$ as a function of E for some results from data Set A, and all results from data Set B

the ratio $\beta = Nu\ Pr^{-0.4}/0.022\ Re^{0.82}$ as a function of the ratio $E' = \frac{t_b - t_{p.c.}}{t_w - t_b}$, a modified Eckert Number (5), in which t_w and t_b are the wall and bulk temperatures, respectively.

The triangles show the ratio β for some of the runs from Set A, while the circles show this ratio for the runs in Set B. It can be seen that this ratio has an average value of unity for the runs from Set A where E' is less than -0.5. At values of E' between -0.5 and -0.1, the data from Set B exhibit an average value of β equal to 1, however, at E' between -0.1 and 0.1 the ratio, β, decreases sharply reaching a value close to 0.5 and then increases gradually approaching unity as E' increases. It is interesting to note that the largest decrease of β occurs when $E' < |0.1|$. This can be interpreted to mean that the average isobutane bulk temperature, t_b, is close to the pseudocritical temperature when the heat flux is low, i.e., $t_w - t_b$ is relatively small, or that the average fluid bulk temperature was relatively far away from the pseudocritical temperature when the heat flux is high. It must be noted here that the calculated film coefficients are average values for 609.6 mm (24 in.) long sections of the tube and are not local values.

Two factors were considered in the placement of the imbedded thermocouples in the instrumented tubes. They were the effects of angular variation of condensation film thickness on the condensation film coefficient, and the effects of fluid buoyancy on the temperature profile inside the tube. Consequently, the imbedded thermocouples at three stations were located 90 deg apart from the other two stations.

The effects of buoyancy have been observed by Adebiyi and Hall (7) on heating carbon dioxide in the supercritical region and by Petukhov et al (8) on heating air with uniform heat flux in horizontal tubes. They observed that, depending on the distance from start of heating and flow regime, large variations of wall temperatures from top to bottom can occur causing enhancement of heat transfer at the bottom and reduction at the top.

The effects of variation of condensate film thickness on the outside of the tube can also contribute to the variation of wall temperature around the tube. Boelter et al (9) showed theoretically that the condensate film thickness on the outside of a horizontal tube decreases to a minimum between 10 to 15 degrees from the top and then increases until it reaches a maximum at the bottom. This distribution of condensate tends to enhance the heat transfer at the top and decrease it at the bottom partially counteracting the effects of buoyancy inside the tube.

Because the main objective of this experiment was to provide baseline coefficients for the design of heat transfer equipment using the same fluid at the same operating conditions, the data were presented as shown on Fig. 4 for direct use. Also because of this objective, the simplest form of correlation equation that can adequately present the experimental data was sought. However, regardless of the type of correlation equation used to display the results in a conventional manner, reliable data on the transport properties must be used. Comparison of transport properties of isobutane from three different sources (4, 11, 12) showed that some of the published values of the properties can differ by a factor of three to four. Consequently, comparison of the correlation of this work with other correlations (8, 13) must be used with the same source of information on transport properties (4) used in this work.

The present correlation from this work on the heating of isobutane, $Nu = 0.022\ Re^{0.82}\ Pr^{0.4}$, as shown by the solid line in Fig. 5, produces values of film coefficients consistently higher than those predicted from the Dittus-Boelter correlation, shown by the dashed line in Fig. 5, with values of fluid properties taken from the same sources. This is consistent with preliminary results found earlier in this experiment (10). Because of the conditions of the experiment, and the relatively clean conditions under which it was conducted, the film coefficient for heating isobutane calculated from the resulting correlation can be considered the maximum value that can be attained under actual plant conditions.

DEVIATION OF RESULTS FROM CORRELATION

The correlation equation for the results of data Set A represents the best fit using the least squares method. The degree of scatter from the correlation represents a deviation, the degree of which is important for designers of heat transfer equipment for this type of service. Figure 9 represents a statistical graph, plotted on arithmetic probability paper,

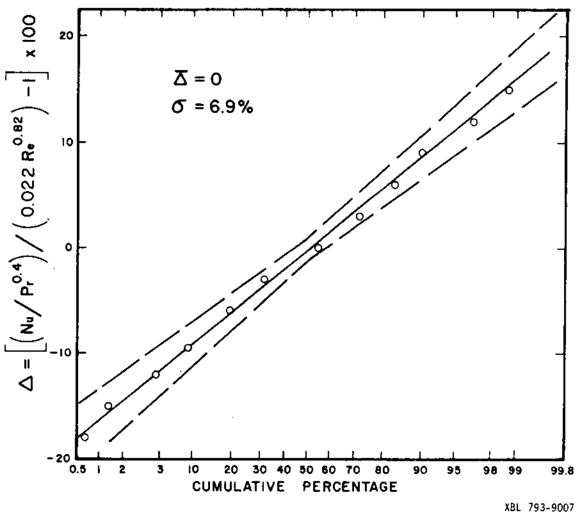

Fig.9 Plot of relative deviation, in percent, of results of data Set A from correlation as a function of cumulative percentage of data points. Outer lines show the upper and lower limit of 95 percent confidence envelope

of the relative deviation, Δ, of the experimental results from the correlation equation where Δ is given by:

$$\Delta = \frac{(Nu\ Pr^{-0.4})_{exp} - 0.022\ Re^{0.82}}{0.022\ R^{0.82}}$$

Figure 9 shows the normal distribution of $\underline{\Delta}$ for data Set A (148 data points). The mean value, $\bar{\Delta}$, is 0, and the standard deviation is 6.9 percent. The intercept of the distribution line and the value of Δ of 10 percent shows there is only 8 percent probability that Δ will be larger than 10 percent. This figure also shows the upper and lower levels of a 95 percent confidence envelope, indicating that the mean value of Δ can vary between ±1 percent. Furthermore, with 95 percent confidence, there is no more than 13 percent probability for Δ to be higher than 10 percent.

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REFERENCES

1 "10 MW Experimental Geothermal Power Plant Conceptual Design; PRE-TITLE 1 REPORT," prepared by Rogers Engineering Co., Inc., and Benham-Blair & Affiliates, San Francisco, Calif., for U. S. Atomic Energy Commission/Lawrence Berkeley Laboratory, Sept. 17, 1974.

2 Green, M. A. and H. S. Pines, "Calculation of Geothermal Power Plant Using Program GEOTHM," Paper No. VII-12, Second United Nations Geothermal Energy Symposium, San Francisco, Calif., May 1975.

3 McAdams, W. H., Heat Transmission, Third Edition, McGraw Hill Book Co., 1954, p. 338.

4 Hanley, H. J. M., "Prediction of the Viscosity and Thermal Conductivity Coefficients of Mixtures," Cryogenics, Nov. 1976, pp. 643-652; (private communication, L. F. Silvester, LBL, 1978).

5 Yamagata, K., et al, "Forced Convective Heat Transfer to Supercritical Water Flowing in Tubes," International Journal Heat Mass Transfer, Vol. 15, 1972, pp. 2575-2593.

6 Bourke, P. J., et al, "Forced Convective Heat Transfer to Turbulent CO_2 in the Supercritical Region," International Journal Heat Mass Transfer, Vol. 13, 1970, pp. 1339-1348.

7 Adebiyi, G. A. and W. B. Hall, "Experimental Investigation of Heat Transfer to Supercritical Pressure Carbon Dioxide in a Horizontal Pipe," International Journal Heat Mass Transfer, Vol. 19, Pergamon Press, 1976, pp. 715-720.

8 Petukhov, B. S., et al, "Turbulent Flow and Heat Transfer in Horizontal Tubes with Substantial Influence of Thermogravitational Forces," Paper No. NC 4.8, Proceedings of International Heat Transfer Conference, Tokyo, 1974, pp. 164-168.

9 Boelter, L. M. K., et al, "Heat Transfer Notes," Chap. XV, University of California Press, Berkeley, California, 1946.

10 Tleimat, B. W., et al, "Hydrocarbon Heat Transfer Coefficients - Preliminary Isobutane Results," Lawrence Berkeley Laboratory Report, LBL 8645, Feb. 1979.

11 Natural Gasoline Supply Men's Association, "Engineering Data Book," 1957.

12 American Petroleum Institute, "Technical Data Book -- Petroleum Refining," 2nd Edition, Chap. 12, 1970, pp. 5-23.

13 Sleicher, C. A. and M. W. Rouse, "A Convenient Correlation for Heat Transfer to Constant and Variable Property Fluids in Turbulent Flow," International Journal Heat Mass Transfer, Vol. 18, Pergamon Press, 1975, pp. 677-683.

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