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**SUPERCRITICAL BINARY GEOTHERMAL CYCLE
EXPERIMENTS WITH MIXED-HYDROCARBON
WORKING FLUIDS AND A NEAR-HORIZONTAL
IN-TUBE CONDENSER**

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ABSTRACT

The Heat Cycle Research Program, which is being conducted for the Department of Energy, has as its objective the development of the technology for effecting improved utilization of moderate temperature geothermal resources. Testing at the Heat Cycle Research Facility which was located at the DOE Geothermal Test Facility, East Mesa, California is presently being conducted to meet this objective. The testing effort discussed in this interim report involves a supercritical vaporization and counterflow in-tube condensing system with a near horizontal tube orientation. A previous report explored the supercritical heating, supersaturated turbine expansions and the condenser performance in the vertical orientation. This report presents a description of the test facility and results from a part of the program in which the condenser was oriented in a nearly horizontal orientation.

Results of the experiments for the in-tube condenser in a nearly horizontal orientation are given for both pure and mixed-hydrocarbon working fluids. Although most of the data is for a completely active condenser in countercurrent flow, some data is available for a configuration in which half of the tubes were plugged and some data for cocurrent (parallel) flow is analyzed. The horizontal-oriented condenser behavior predicted by the Heat Transfer Research Institute computer codes used for correlation of the data was not in agreement with experimental results at this orientation. Some reasons for this difference are discussed. A special series of tests, conducted with propane and up to approximately 40% isopentane concentration, indicated that a close approach to "integral" condensation has occurred as was the case with the horizontally oriented condenser (similar results were obtained for the vertical condenser).

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Those who have reviewed the report have each provided some additional insight into the work. At INEL, R. J. Kochan and T. W. Lawford gave a complete internal review of the document. The review of Dr. Duncan Chisholm and Dr. Raj Sardesai from Heat Transfer Research, Inc. gave the perspective of the heat exchanger industry as well as some basic insights into the condensation process. The view of the geothermal industry was given in the reviews of Mr. Richard Campbell of the Ben Holt Company and Michael Forsha of Barber Nichols Engineering. Dr. D. Y. Goswami of the Mechanical Engineering Department of North Carolina Agricultural and Technical State University presented an academic point of view.

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SUPERCRITICAL BINARY GEOTHERMAL CYCLE EXPERIMENTS WITH MIXED-HYDROCARBON WORKING FLUIDS AND A NEAR-HORIZONTAL IN-TUBE CONDENSER

SUMMARY

BACKGROUND

The overall objective of the Heat Cycle Research Program, which is being conducted for the Department of Energy (DOE), is to develop technology which will result in more effective utilization of moderate temperature geothermal resources; a major emphasis of the program has been directed toward binary cycle technology. Several binary cycle concepts, investigated analytically in earlier program efforts, have shown the potential for effecting significant performance gains for the production of electrical power in binary plants. Utilizing non-adjacent hydrocarbon mixtures for working fluids, which are vaporized at supercritical pressures, and a counterflow in-tube condenser to provide a close approach to integral condensation, are two concepts with the potential for significant performance gains. (Integral condensation refers to the maintaining of thermal equilibrium between phases during condensation, and minimizes condensing pressure for a given condensing temperature.) Additional performance gains were predicted through use of turbine exhaust recuperation, and through modification of turbine inlet state points to achieve supersaturated-vapor turbine-expansion processes. These advances, in total, were projected to increase present levels of net plant geofluid effectiveness (Wh/lbm geofluid) by as much as 28% using 360⁰F hydrothermal resources, and to double the utilization of moderate-temperature geothermal energy. Experiments for confirming the assumptions made in the performance projections, and for developing the technology needed to achieve counterflow integral condensation, are required to complete the technology development for utilization of these advanced-binary-plant concepts.

EXPERIMENTS CONDUCTED

To accomplish the objective of developing technology for advanced binary geothermal plants, a number of supercritical cycle experiments were conducted using nominal working fluids consisting of both pure and mixed hydrocarbons of the propane-isopentane (0, 5, 10% isopentane) and isobutane-hexane (0, 5, 10% hexane) families. In this interim report, only condensing of the pure and mixed-hydrocarbon vapors in an in-tube condenser in a near-horizontal orientation is discussed. The initial orientation of the condenser was vertical; these results were presented in an earlier report (Reference 1). The testing program initially consisted of testing the condenser in a number of orientations. It was initially thought that slip between the liquid and vapor phases could be controlled at some optimum angle and, thereby, create integral condensation. (This was found not to be a relevant issue after limited testing.) From an operational point-of-view, a horizontal unit is easier to perform maintenance on and might be preferable to a vertical or slanted unit. The slanted angles would correspond to condensing units in A-Frame type air-cooled condensers.

A series of tests was run with special propane-isopentane mixtures with isopentane weight fractions of up to 40% to investigate the departure from integral condensing exhibited by the condenser. Some special tests were conducted with the isobutane-hexane mixtures to determine condenser performance at other working fluid flow conditions than the actual design conditions. Testing, in the main, was done with countercurrent condenser flows, however, several series of tests were conducted with the flow parallel (cocurrent) instead of countercurrent, and also with about half of the tubes plugged.

The testing in the vertical orientation took place between February 1984 and August 1985. There was a 7 month period in that time period during which no data was taken because of well reworking. The condenser orientation was changed to 10 degrees from the horizontal and a hot well added to the system. Data acquisition in this configuration was begun in

May 1986 and concluded in February of 1987. The orientation of the condenser has since been changed to 30 degrees from the vertical and data is being taken in this third orientation.

The experiments were conducted in the Heat Cycle Research Facility (about 40 kW turbine power rating). This facility was formerly located at the Raft River test site; it was subsequently skid mounted and relocated at the DOE Geothermal Test Facility (GTF) in the Imperial Valley of Southern California.

In these investigations, the working fluid was heated and vaporized on the shell side of a pair of counterflow heat exchangers having externally finned tubing and connected together in series. Vaporizer and limited turbine performance results were reported in Reference 1, and are not repeated here because the configuration of these components has not changed. Heat was supplied by geofluid from GTF Well 6-2 at temperatures between 300 and 322⁰F. Condensing of the working fluid vapor was accomplished inside of internally-finned tubing in a counterflow, near-horizontally-oriented shell-and-tube condenser supplied with cooling water from the GTF wet cooling tower.

RESULTS

Approach to Integral Condensation - The results indicate that the condenser in the nearly horizontal orientation carried out the condensation of the mixed hydrocarbon working fluids with a minimum deviation from integral condensation. Even for mixtures of propane and isopentane of 60/40% (by mass) which had condensing ranges of around 60⁰F, no evidence of differential condensation was detected.

Comparison of Condenser Performance in Horizontal Orientation to that in Vertical Orientation - The overall heat transfer coefficient in the nearly horizontal orientation was 33 to 47% lower than the same condenser in the vertical orientation. This means that orientation of a condenser vertically would result in a decrease in size of 33 to 47% over orienting

the condenser horizontally. This difference, expressed in terms of an additional thermal resistance was approximately $0.0027 \text{ hr ft}^2 \text{ }^\circ\text{F}/\text{Btu}$ for pure fluids. For mixtures of 90% isobutane/ 10% hexane (by mass), this resistance increased to 0.0037. Expressed in terms of the inside heat transfer coefficient, the condensing coefficient, the difference ranged from 34 to 61% lower value for the horizontal orientation. It is apparent that in the horizontal orientation, the fins do not enhance the condensation as much as they do in the vertical orientation.

Analytical Predictions of the Nearly Horizontal Performance - The predictions of the performance of the condenser in its nearly horizontal orientation were performed using the Heat Transfer Research, Inc. (HTRI) shell and tube condenser computer program, CST-2 MOD 0.00-1.01. Because this program handles only internally plain tubes, input modeling was necessary to approximate the behavior of the internal fins in the condenser. Two models were used: one which substituted the equivalent diameter (hydraulic diameter) for the real diameter of the tube and one which assumed that the tube was a plain tube with the nominal inside diameter of the internally finned tube, but with the area enhancement of the fins added to the inside area by an artificial multiplier. Neither model adequately predicted the experimental results at this condenser orientation. Both methods overpredicted the average inside heat transfer coefficient (combined desuperheating and totally condensing) by between 50 and 150%. Either method would adequately predict the results with the condenser. The equivalent diameter method is preferred, however, on a conceptual basis because it correctly calculates the desuperheating and shear-controlled condensation.

Some interesting deviations were noted. For pure fluids, propane and isobutane, the difference between the calculated and experimental values expressed as a thermal resistance was generally between 0.003 and $0.004 \text{ hr ft}^2 \text{ }^\circ\text{F}/\text{Btu}$. This is more than can be explained by an incremental change in the fouling of the heat exchanger between the tests. It is felt that this represents a decrease in the efficiency of gravity-controlled condensation in the horizontal tube orientation. The longitudinal fins will block the natural drainage path around the sides of

the tube and hamper the film thinning at the top of the tubes. (No difference was noted for the vertical orientation in which the finned area simply added vertical surface, and perhaps some film thinning resulting from surface tension effects where the fin joins the tube wall.) The difference between the calculated and experimental thermal resistance also displayed a strong dependence on the condensing range of the working fluid. A look at the vertical comparison shows a similar trend. This might indicate that HTRI's method for calculating the mixture condensation heat transfer is not correctly accounting for diffusion effects or that the variable composition effects (increased thermal resistance) are enhanced by the finned geometry, that is, the fins inhibit the diffusion of material to and from the condensing surface.

SUMMARY OF CONCLUSIONS AND RECOMMENDATIONS

The results and conclusions of this work can be summarized as follows:

1. There is no evidence that the condensation in the nearly horizontal condenser deviated from integral condensation. There would be no thermodynamic penalty associated with orienting the condenser in a nearly horizontal position.
2. The heat transfer performance of the internally finned condenser in the nearly horizontal orientation is 33 to 47% worse than the same condenser in the vertical orientation. This means that a condenser in the vertical orientation could be 33 to 47% smaller than one in the horizontal orientation to perform the same duty, that is, produce the same turbine back pressure with a given cooling water inlet temperature and flow.
3. The method of predicting the performance of a condenser in the nearly horizontal orientation with internally finned tubes is not well established. The two models developed here do not give good results in their comparison. The design of this type of condenser in this orientation can now be approximated but the uncertainties are beyond

the practical limits desired. The flow in the vertical orientation is essentially one-dimensional while the flow in the horizontal orientation is two-dimensional.

The following additional actions are recommended in order to refine the design methods which will allow the supercritical technology to be put into practice:

1. At the end of the program, return the condenser to its vertical orientation and repeat some of the original tests. This will allow the amount of fouling during the testing period to be estimated with greater certainty and will allow the removal of some of the uncertainty from the conclusions presented in this report.
2. Develop a simple computer program which will allow exploration of the condensing process in detail. It is felt that the penalty of the fins in the horizontal orientation is only where gravity-controlled condensation takes place. It is impossible to analytically examine this hypothesis with the HTRI computer program. The question also arises concerning the method used to design with this type of system. (Is a combined mass transfer/heat transfer model needed?) This could be analytically explored with the present data if an appropriate computer program was available.

INTRODUCTION

The supercritical Rankine cycle experiments, discussed in this report, constitute the second phase of an advanced binary cycle experimental program in which the counterflow, in-tube condenser was oriented nearly horizontally (10° off the horizontal). The first experimental phase of this program in which the condenser was oriented in a vertical attitude was reported in Report EGG-EP-7076 (Reference 1). Those, and the present experiments, are parts of the Heat Cycle Research Program which is being conducted for the Department of Energy (DOE) to develop technology required to more fully utilize the moderate temperature geothermal resources for the production electrical energy. In this regard, a major concern of the program is directed toward advancing binary cycle technology for application with resources up to 400°F temperature.

The total Heat Cycle Research Program is summarized in some detail in Reference 2. Earlier results of the supercritical cycle experiments were presented at the Third and Sixth DOE Geothermal Technology Division Program Reviews of 1984 through 1988, and are included in References 3-6.

The work was supported by the U.S. Department of Energy, Geothermal Technology Division, under Contract No. DE-AC07-76ID01570. Mr. Raymond LaSala is the program manager at DOE Headquarters and Mr. K. J. Taylor provides DOE support at the Idaho Operations Office.

PREVIOUS ANALYSES OF ADVANCED PLANTS

Several advanced plant concepts have been investigated analytically, in earlier Heat Cycle Research Program efforts, for increasing the net plant geofluid effectiveness (Wh/lbm geofluid) of binary cycles utilizing a 360°F liquid dominated hydrothermal resource. These analyses have indicated that advanced binary plants could achieve performance improvements of up to 20% and cost of electricity improvements of as much as 13% relative to present state-of-the-art plants such as the Heber 45MW binary plant or the Raft River 5MW dual boiling plant, providing the

analysis methods and assumptions are valid. Plant modifications for these improvements would consist of use of non-adjacent hydrocarbon mixtures for working fluids, a counterflow condenser providing "integral" condensation; and, if the geofluid outlet temperature is limited to avoid silica precipitation, a turbine-exhaust recuperator. Further performance and cost-of-power improvements of up to 8 and 5.5%, respectively, were projected for utilization of modified turbine inlet state points which would result in metastable supersaturated-vapor turbine-expansion processes. (These improvements are summarized in Reference 7.) An independent market-penetration analysis (8-9), conducted by Technecon Analytical Research, Inc., indicates that these improvements are significant, and could result in an increased utilization of geothermal resources in the 350 to 400⁰F range of over 100% by the year 2000 if the required technologies can be developed.

EXPERIMENTAL APPROACH

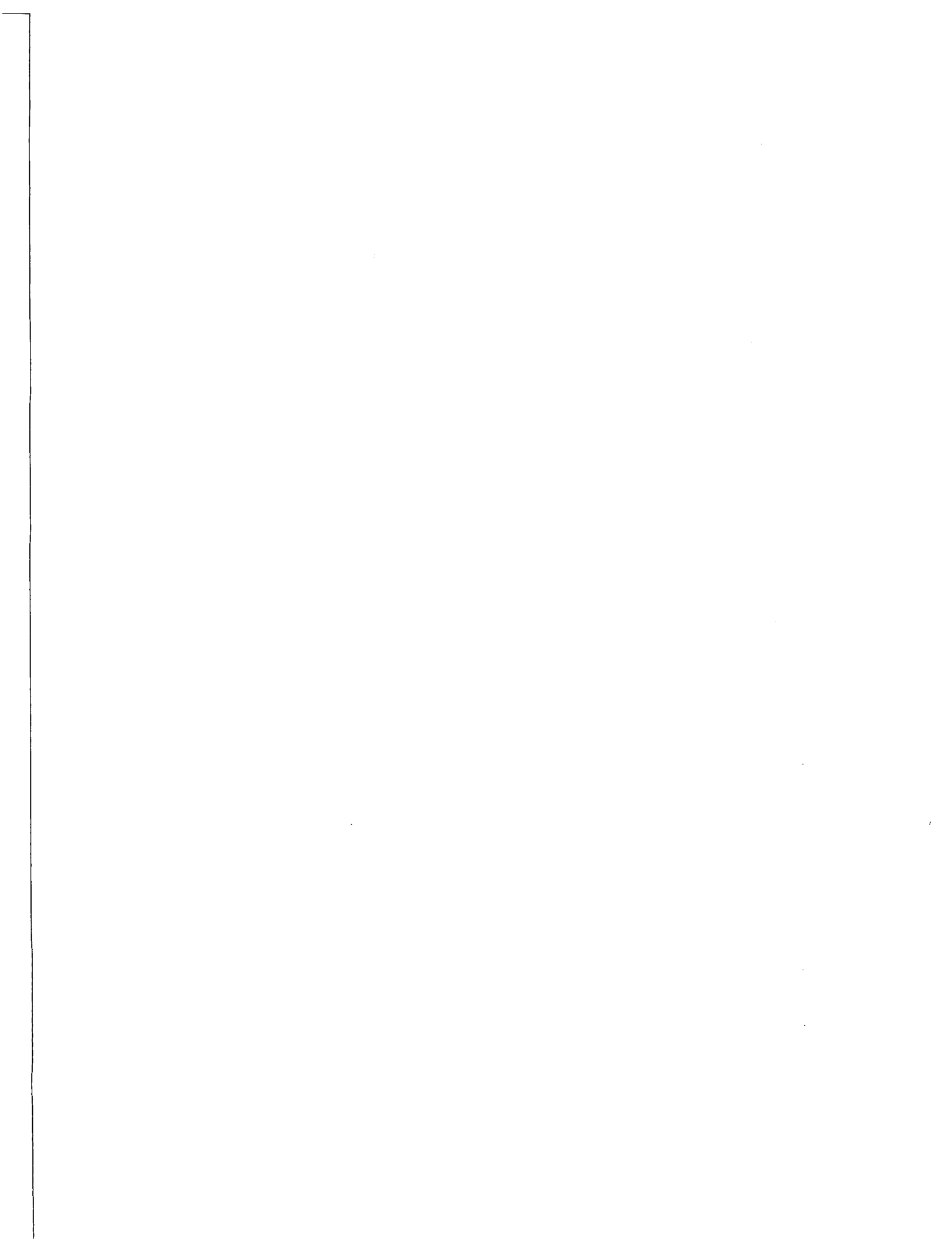
The approach taken in the present experimental program is to develop and/or validate the technology assumed in the plant improvement analyses previously conducted, utilizing the components assembled for this purpose in the Heat Cycle Research Facility (HCRF). The HCRF was located at the Department of Energy's geothermal test site in the Imperial Valley, California during this testing period. At this site, the geofluid at the inlet of the HCRF varied from 300 to 322⁰F. The more specific objective of the testing described in this report was to investigate the condensation of pure and mixed hydrocarbon vapors in a counterflow in-tube condenser (oriented in a near-horizontal attitude). The effort included: (1) the comparison of the experimental condensation processes with predictions made using state-of-the-technology heat-exchanger design computer codes, (2) the observation of the capability of the National Bureau of Standards (NBS) Code EXCST (developed using DOE funding) for predicting working fluid thermodynamic and transport properties (Reference 10), and (3) a comparison of the experimental condensation process in the vertical and horizontal attitudes.

The technology required to achieve the performance and resource-utilization advantages predicted for the advanced binary plants can be judged to have been developed adequately if our experiments show that:

1. State-point thermodynamic properties of the mixed hydrocarbon fluids can be predicted satisfactorily with the NBS properties code.
2. Counterflow integral condensation can be achieved within practical limits.
3. The mixed hydrocarbon condensation behavior can be predicted by state-of-the-technology condenser design codes.

SCOPE OF PRESENT EFFORT

The present report (considered as an interim report) presents results of a second phase of supercritical binary cycle experiments conducted with mixed hydrocarbon working fluids of the propane-isopentane and isobutane-hexane families with nominally 100, 95 and 90% by mass of the propane and isobutane components. During this phase of the program the condenser orientation was near-horizontal (10° from horizontal). (In a later phase, the inclination will be changed to 60° from the horizontal.) (Testing of the other components was reported with the vertical condenser results in Reference 1.) Only condenser test results are presented. A special series of propane-isopentane tests was run with isopentane concentrations up to 40% where the effects of any departure from integral condensation would be amplified. In addition, tests were conducted operating the condenser in cocurrent flow and with approximately half of the tubes plugged to investigate operating conditions with higher liquid loadings and larger approach temperature differences; code predictions of those tests are presented here for comparison with experimental values.



DESCRIPTION OF HEAT CYCLE RESEARCH FACILITY

The Heat Cycle Research facility (HCRF) is an experimental binary-cycle facility used to investigate different concepts and/or components for generating electrical power from a geothermal resource. In the binary power cycle, the energy from the geothermal fluid is transferred to a secondary working fluid, which is in turn expanded through a turbine driving an electrical generator. The facility, which was formerly located at the Raft River geothermal site in Idaho, was located at the DOE Geothermal Test Facility (GTF) in California's Imperial Valley when these tests were conducted. A photograph of this installation with the condenser oriented vertically is included as Figure 1. Figure 2 shows the facility with the condenser in its near-horizontal orientation (10 degrees from horizontal).

The HCRF is shown schematically in Figure 3. In this configuration the facility is operated as a supercritical cycle; that is, the working fluid vapor leaving the heaters is at a temperature and pressure higher than its critical point. As indicated in Figures 1 through 3, there are two supercritical heat exchangers, a preheater and a vapor generator. The energy from the geothermal fluid, which is flowing inside the tubes of the units, is used to heat a hydrocarbon working fluid flowing on the shell side. (The geothermal fluid was supplied from GTF Well 6-2, and entered the HCRF at a temperature between about 300 and 322°F.) The high-pressure working fluid vapor leaving the supercritical heaters can either be expanded through a turbine which drives an electrical generator (power loop operating mode), or be expanded through a turbine bypass valve (thermal loop operating mode). The low-pressure vapor leaving the turbine or bypass valve is discharged to the condenser where it is desuperheated and condensed. The liquid condensate is then pumped back to the heaters, and the cycle is repeated. In the condenser, which is a counterflow in-tube condensing unit, the heat rejected in condensing the working fluid vapor is transferred to cooling water on the shell-side of the unit. The orientation of the condenser can be changed. It was originally vertical (as shown in Figure 1), but has been lowered to an inclination of 10

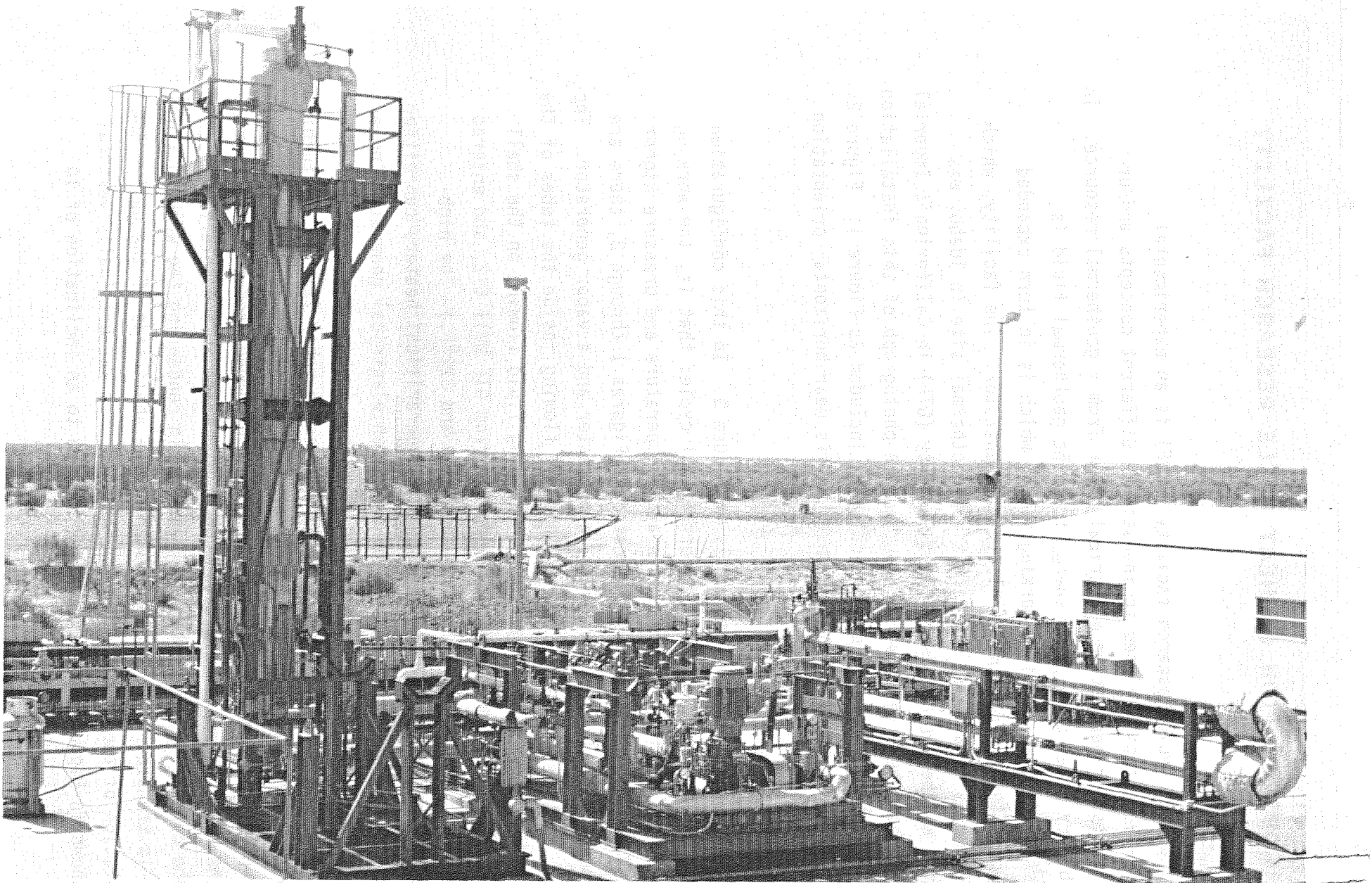


Figure 1. Heat Cycle Research Facility located at the DOE geothermal test site with the condenser oriented vertically.

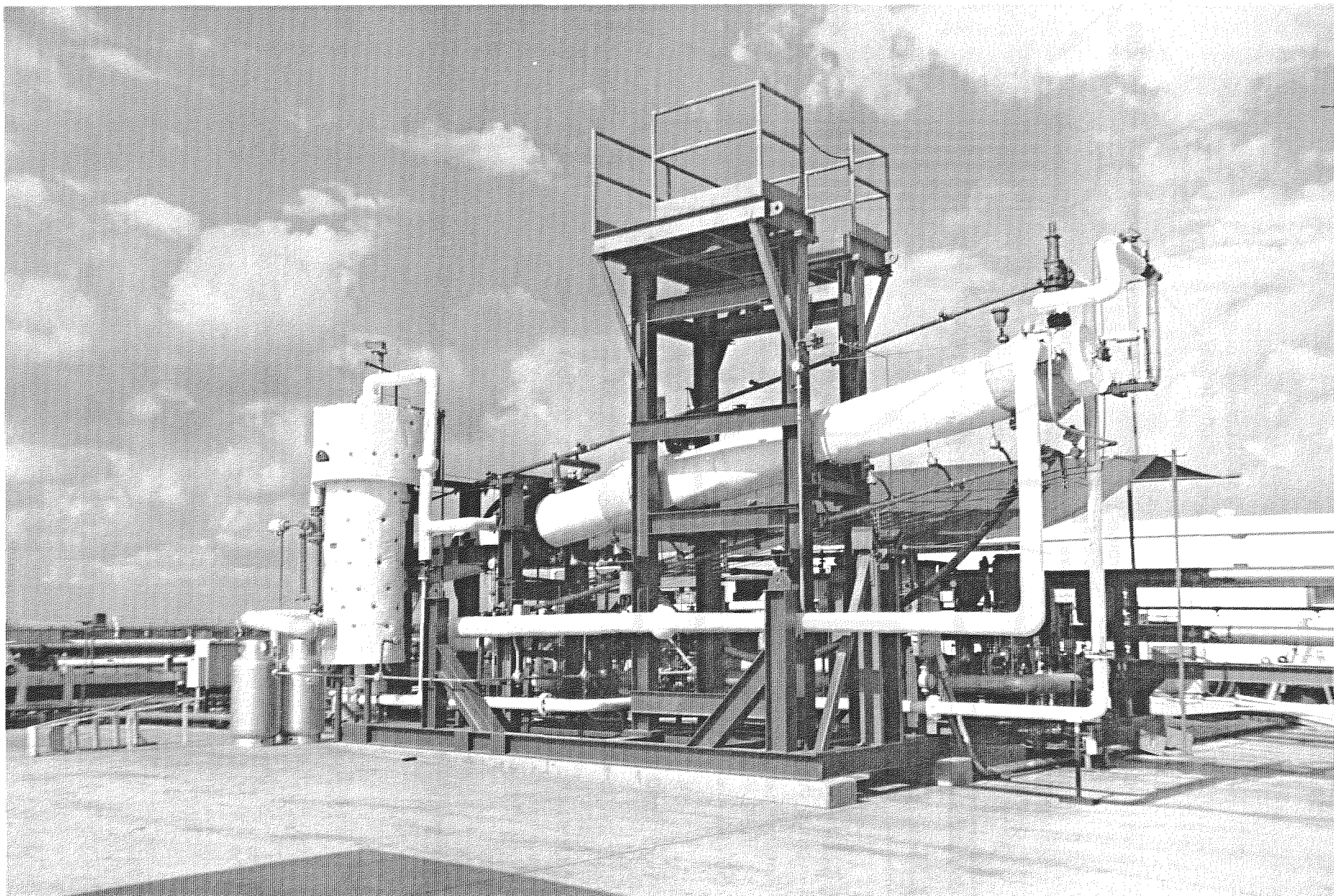
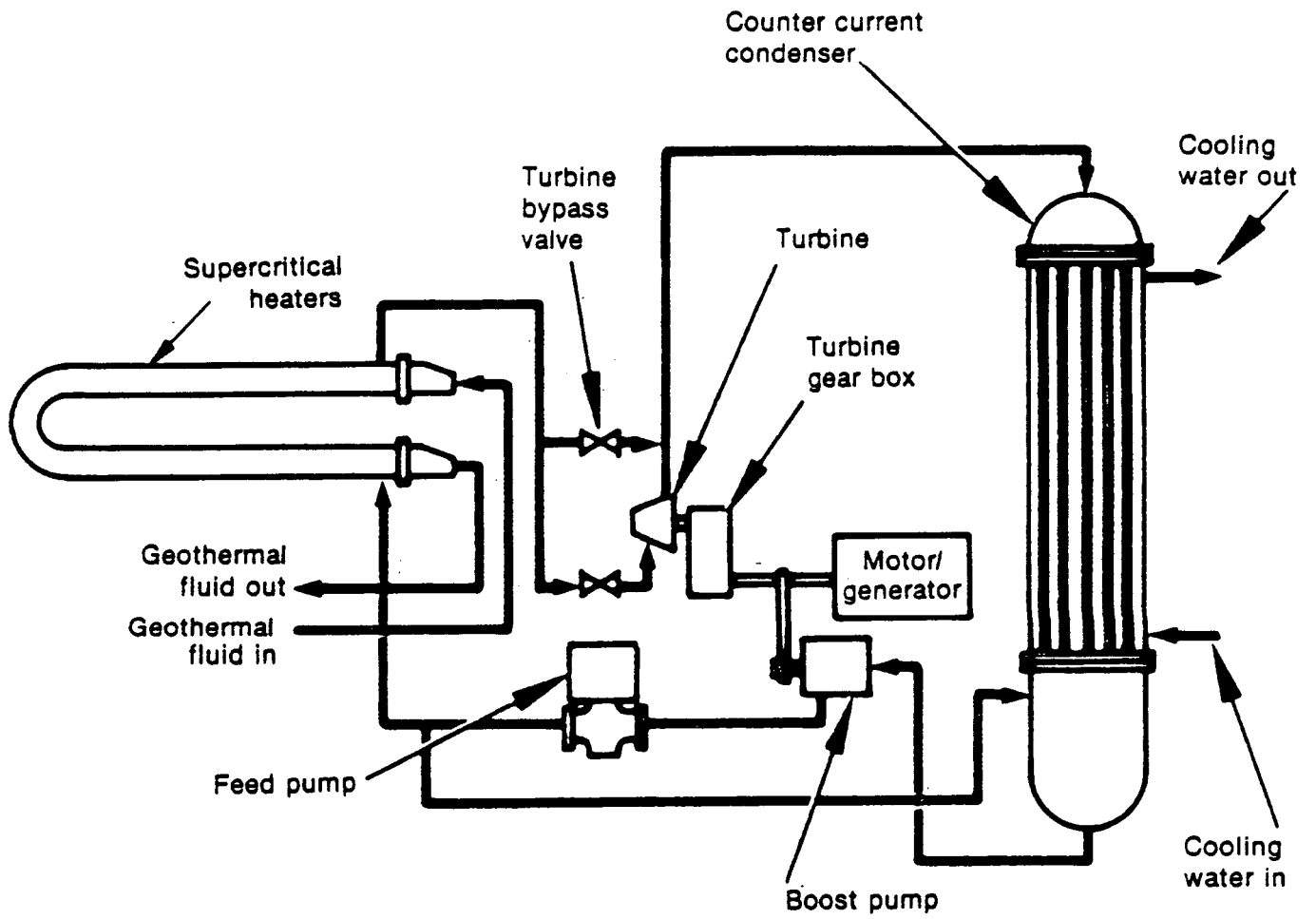


Figure 2. Heat Cycle Research Facility located at the DOE geothermal test site with the condenser oriented near-horizontally.



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Figure 3. Schematic of the Heat Cycle Research Facility.

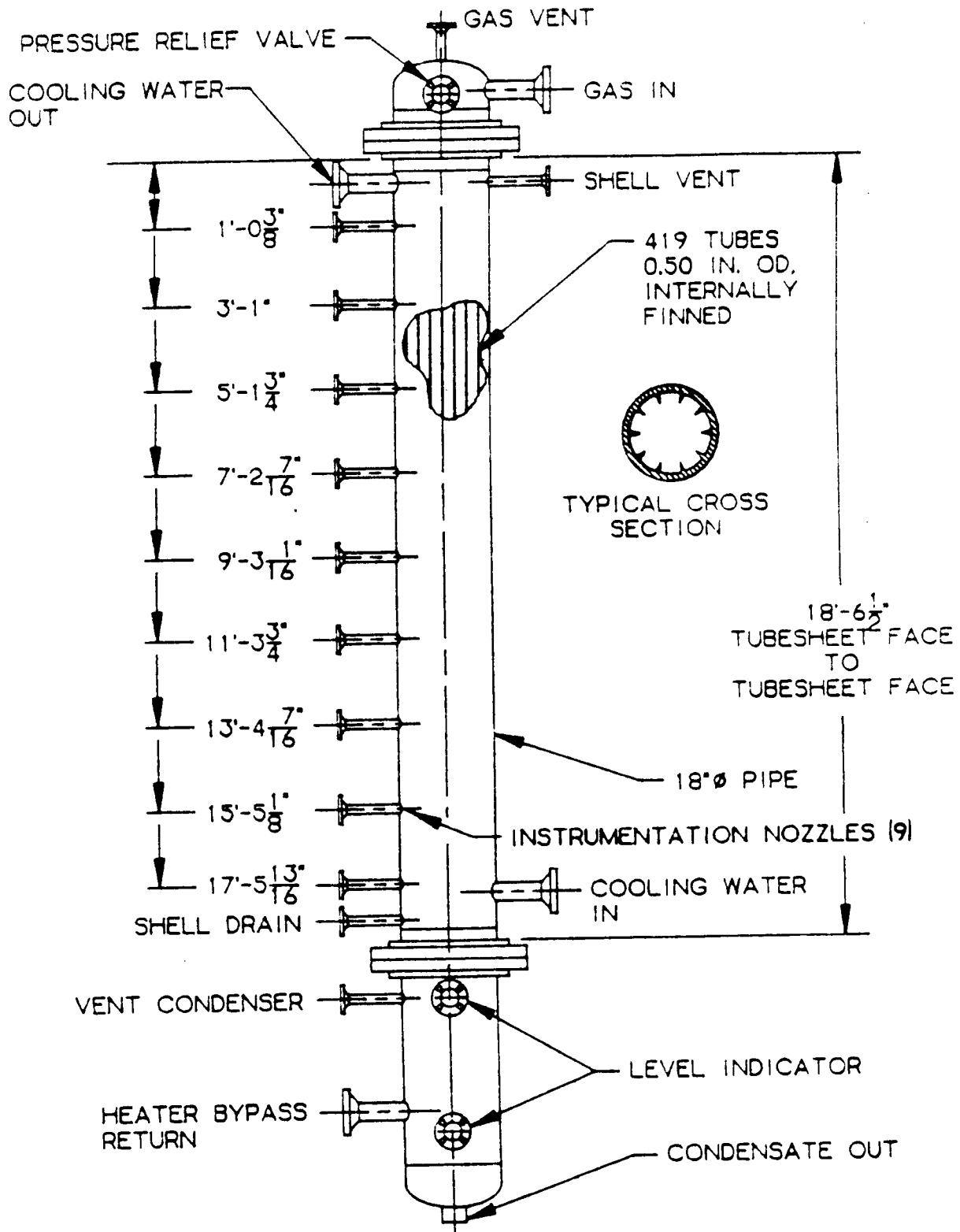
degrees to the horizontal (See Figure 2) and later will be run at an angle of 60 degrees from the horizontal. The cooling water is supplied from the GTF cooling-water system which includes a conventional wet cross-flow tower.

The various parts of the system are described in the previous report EGG-EP-7076 (Reference 1). The description of the condenser is repeated in the next section for the convenience of the reader in reviewing test results.

CONDENSER

The condenser, for the tests discussed in this report, was in an orientation with tubes inclined 10 degrees from the horizontal and having normally countercurrent flow paths. The condensation occurs on the inside of 1/2-inch OD, internally finned tubes made of 90/10 cupro-nickel (Noranda forge fin No. 6, with ten straight longitudinal fins inside each tube giving an inside-to-outside area ratio 1.3). See Reference 11. The vessel is 18 inches in diameter and contains 419 of the tubes which have a length of 18.54 feet (tubesheet face-to-face). The design temperature for the unit is 350°F with a tubeside design pressure of 350 psi and a shell-side design pressure of 175 psi. In its vertical orientation, the cooling water entered the shell-side just above the lower tubesheet and left vessel just below the upper tubesheet. The working fluid condensate collected in the lower portion of the vessel (below the lower tubesheet), which acted as a hot well. A sketch of the condenser is shown in Figure 4. Initially, the water side of the condenser was cleaned and the surface passivated with a phosphate.

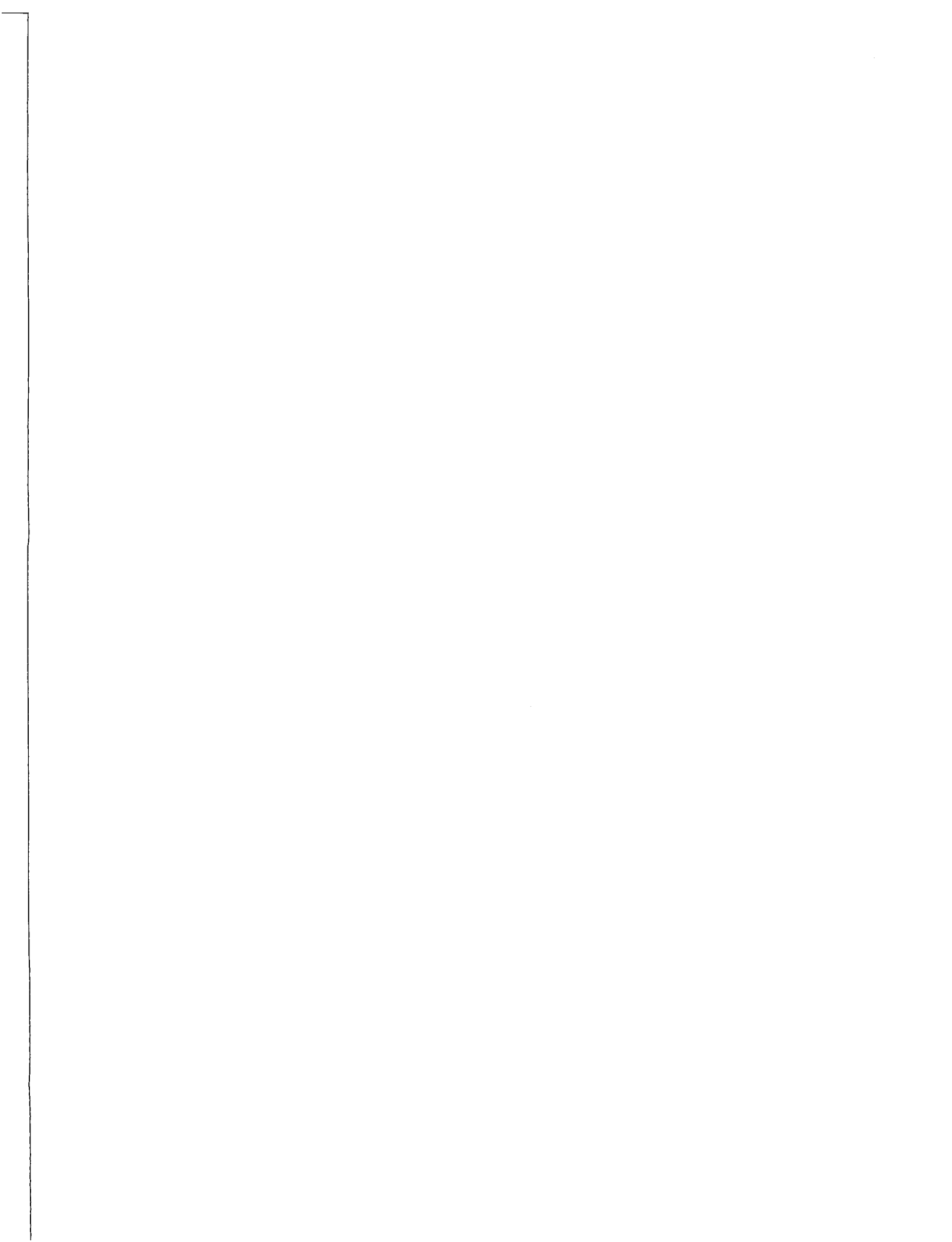
Three modifications to the condenser have been made for certain tests. First, an external vessel was added to make a hot well when the orientation was changed from the vertical. (See Figure 2.) Second, the flow path of the cooling water was reversed to achieve a parallel (cocurrent) flow in the heat exchanger for some tests. Third, for certain tests, approximately every other tube was temporarily plugged on the working fluid inlet side tube sheet to give 202 tubes through which the working fluid could flow instead of the original 419 active tubes.



CAD-468

Figure 4. Sketch of the counterflow, in-tube, condenser.

Working-fluid and cooling-water temperatures were measured entering and leaving the condenser were measured with platinum resistance temperature devices (RTDs). At nine intermediate locations within the condenser (shown in Figure 4) iron-constantan thermocouples were used. Working-fluid pressures were measured upstream and downstream of the condenser using electronic pressure transmitters, and cooling water pressures were monitored with mechanical gauges. Working-fluid flow through the condenser was determined during steady-flow conditions from a turbine flowmeter located at the preheater inlet (liquid flow at this point gave the most accurate measurement of working fluid flow). Cooling-water flow was measured using an orifice-plate flow meter located near the condenser outlet.



EXPERIMENTAL APPROACH

EXPERIMENTS CONDUCTED

The testing with the condenser in its vertical orientation, which was discussed in the previous report (1), was begun in the Fall of 1983 and ended in the Summer of 1985. Data was taken from February of 1984 through August of 1985. There was a 7 month period in which the system was not run because of the geothermal well being reworked. The pure propane tests were run followed by the propane/isopentane mixture tests (up to 10% isopentane). These were followed by the isobutane/hexane tests with composition of hexane increasing. Finally, the "integral condensation" tests were run with the propane/isopentane mixtures up to 40% isopentane.

There was a 9 month period in which the configuration of the plant was changed to accommodate the near horizontal condenser which included adding an external hot well. The first data for the near horizontal test series was taken on May 14, 1986. The order of testing was: propane/isopentane mixtures from 0 to 50% isopentane with no tubes plugged and countercurrent flow, isobutane/hexane mixtures from 0% to 10% hexane with no tubes plugged both countercurrent and cocurrent flow, and last, isobutane/hexane mixtures from 0 to 10% hexane with half the tubes plugged (countercurrent and cocurrent flow). The entire testing period lasted about 9 months ending in February of 1987. In the near-horizontal test series, testing with a given working fluid took between 1 and 2 weeks with 1 to 2 weeks between working fluids.

The emphasis during the current phase of testing has been to investigate the performance of the counterflow, internally-finned condenser in its near-horizontal orientation, particularly when mixed-hydrocarbon working fluids are used. First, baseline performance data was established with a single-component working fluid. Then mixtures were tested in which the primary component was the fluid used in the baseline tests with increasing amounts of a secondary fluid. Two families of nominal working fluids were tested; the isobutane/hexane family and the

propane/isopentane family (the primary constituent given first for each family). The order of testing for each family was single component (primary constituent), 95%/5%, and 90%/10%. For each fluid, i.e., 95% isobutane/5% hexane, data were taken at a number of different amounts of working fluid superheat entering the condenser, as well as varying working fluid and cooling water flow rates. A special series of tests was conducted using the propane-isopentane family of working fluids with isopentane concentrations ranging up to 40%, in order to further investigate the approach to integral condensation being achieved. At each test condition, the composition of the working fluid mixture was verified using a gas chromatograph analysis. The tests specifications were constructed to be like the countercurrent-condensation and integral condensation tests for the propane fluids in the vertical attitude. The actual test makeup is discussed in detail in Appendix A.

For the isobutane/hexane family of tests, in addition to the normal countercurrent flow tests repeated from the vertical operation, the exchanger was configured in three other ways: cocurrent with no tubes plugged, countercurrent with half the tubes plugged, and cocurrent with half the tubes plugged.

For the present testing with the condenser in the near-horizontal orientation a total of some 345 tests have been conducted. Of these, about 140 have been selected for detailed analyses to study the condenser behavior over the range of test conditions of interest. Appendix A outlines the test conducted, and presents data sheets for those selected for detailed analysis.

ASSESSMENT OF DATA AND THERMODYNAMIC PROPERTY CONSISTENCY.

Three comparisons were used to check the consistency of the experimental data recorded for the condenser. First, the approach to steady state was assessed by considering the change in pressure in the condenser hot well over the term of the test. If the change in pressure was more than 1% of the absolute pressure, the run was discarded. This eliminated two tests. In all of the remaining tests except for one, the maximum deviation was less than 0.5%. Second, an energy balance for the

condenser was made for each run. The consistency of the calculated heat transferred from the working fluid to that transferred to the cooling water was evaluated. Third, comparisons were made between the measured condenser pressure and the working fluid outlet temperature. These assessments of the experimental data, of necessity, involve the thermodynamic property relations. Working fluid enthalpies were used for the energy balances as calculated by the EXCST computer program (Reference 10). Condensing pressure and temperature relations are needed to test the consistency of working fluid pressure and outlet temperature measurements. The relationships were again taken from EXCST which assumes thermodynamic equilibrium between the liquid and vapor phases, that is integral condensation. A more complete discussion of the experimental evidence for integral condensation is given in the section on results. Good consistency was found in both of these comparisons when EXCST was used giving confidence to the measurements of working fluid composition, temperatures, pressures and flow rates and the properties generated by EXCST.

Comparing the heat transfer rate from the working fluid to the rate to the cooling water for the 138 runs evaluated indicated that the average difference was 3.2% (with the cooling water calculated rate being higher), with a standard deviation of 4.4%. Applying Chauvenet's criterion to the data, three runs had higher deviations than expected. When these three runs (all propane/isopentane mixtures with very low superheat) were removed, the average difference in the heat transfer rate was 2.8% (with the calculated heat transferred to cooling water remaining the greater) and the standard deviation was reduced to 3.4%. The distribution was approximately a normal one with 78% of the runs within one standard deviation of the average, 94% within two standard deviations, and 97% of the data within three (compared with 68.3, 95.4, and 99.7% respectively for a normal distribution). That is, 94% of the runs balanced within -3.9% to +9.5% (the plus indicating a larger value for the cooling water heat transfer). It was felt that this comparison showed very good consistency among the experimental quantities: working fluid composition, four temperatures, working fluid pressure and two flow rates along with the thermodynamic properties of the working fluid as predicted by the EXCST computer program.

The third consistency comparison was between the measured working fluid pressure, the working fluid outlet temperature and the thermodynamic properties for the phase change. Using the EXCST computer program and the measured working fluid composition, a bubble point pressure was determined for the experimentally measured outlet temperature. The bubble point is the point at which condensation is complete, and because EXCST assumes complete mixing of the liquid and vapor phases, the result applies for integral condensation.

Figure 5 shows the difference between the calculated bubble point temperature and the measured condenser working fluid outlet temperature for the both families of working fluids in all condenser configurations (counterflow/cocurrent, no tubes plugged/half tubes plugged) plotted with the working fluid condensing range, the difference between the dew point and bubble point temperatures for the measured composition and pressure. It was felt that the condensing range, rather than the percentage of the heavier constituent, was a better characterization of the working fluid because the heat transfer resistance is more closely related to this temperature difference than to the composition difference expressed by the composition variable. The different mixture compositions are noted by different symbols. The propane/isopentane mixtures are denoted by open symbols and the isobutane/hexane mixtures by shaded symbols. Note that if the instrumentation measures the correct values of condensing pressure, outlet temperature, and composition of the working fluid, the thermodynamic property relations are correct, the condensation path was integral, and there was no subcooling of the condensate in the condenser; the plotted temperature difference would be zero. The majority of the experimental points lie between temperature differences of 0 and 1. The average value is 0.50 °F. Many of the deviations from this relate to specific sets of data: The large positive differences are all for tests using 10% hexane in isobutane/hexane mixtures (the shaded diamonds). All of those values greater than 1 are for configurations in which half of the tubes were plugged (C240's and C260's). All but two of the 5% hexane isobutane/hexane mixtures (shaded triangles) which are negative are also plugged tube runs. If all of the plugged tube data is removed, the average value is 0.47 and the standard deviation is 0.52 °F.

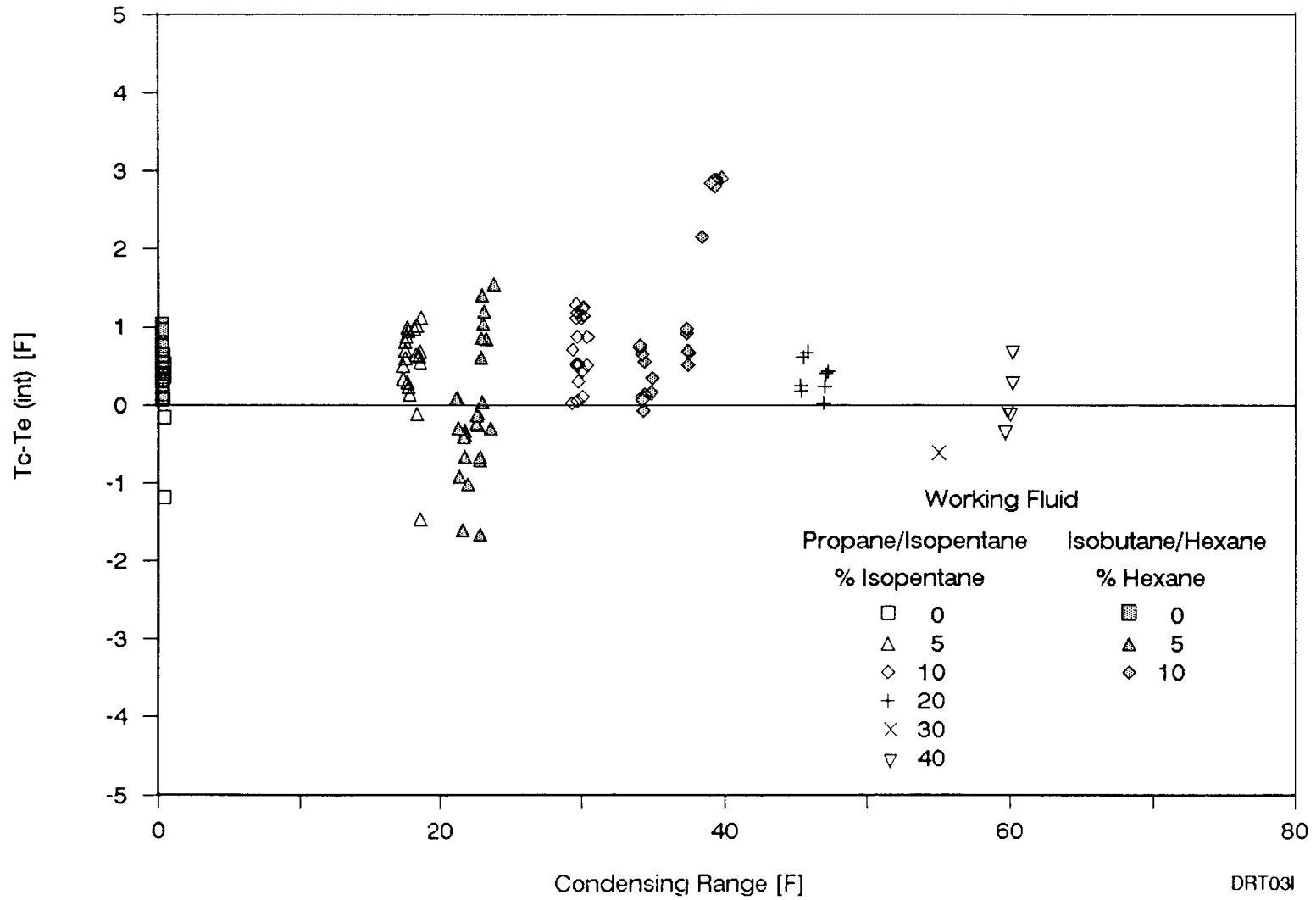


Figure 5. Temperature deviation under integral condensation assumption.

Approximately 75% of the data lies within plus and minus one standard deviation of the mean. This is approximately between 0 and 1. One possible explanation for this would be a slight subcooling of the condensed liquid taking place in the lower portion of the tubes. Subcooling of this order-of-magnitude would be expected in horizontal tubes as discussed by Mueller [12].

Figure 6 shows the same data presented in terms of the difference between the calculated pressure for a bubble point at the measured outlet temperature minus the measured condenser pressure. This figure is quite similar to the previous one with the difference. The spread of the data for each mixture is more nearly uniform for both the propane/isopentane and isobutane/hexane mixtures than in Figure 5 where the propane/isopentane data has substantially less scatter than the isobutane/hexane data. A possible explanation of this difference is related to the fact that the change in saturation temperature for a unit pressure change in propane is 42 to 43% that of isobutane at the temperatures in the condenser. Therefore, if there is a spread in pressure inherent in the pressure transducer, it would translate into a smaller spread in temperature for propane than isobutane.

The following statements summarize the findings in these comparisons:

1. The experimental data needed to evaluate the condenser performance are quite consistent among themselves.
2. The thermodynamic properties generated by the computer code, EXCST, are consistent with the data in enthalpies used in the energy balances and in prediction of the condensing pressure-temperature relationship.
3. The condensation appears to be integral with practically no subcooling in the working fluid leaving the condenser. (This third finding pertains also to condenser performance and is discussed further in the section on results).
4. Instruments appear to be quite accurate with the accuracy of the pressure measurement being within plus or minus 1 to 1.5 psi and the temperatures and compositions having little variation.

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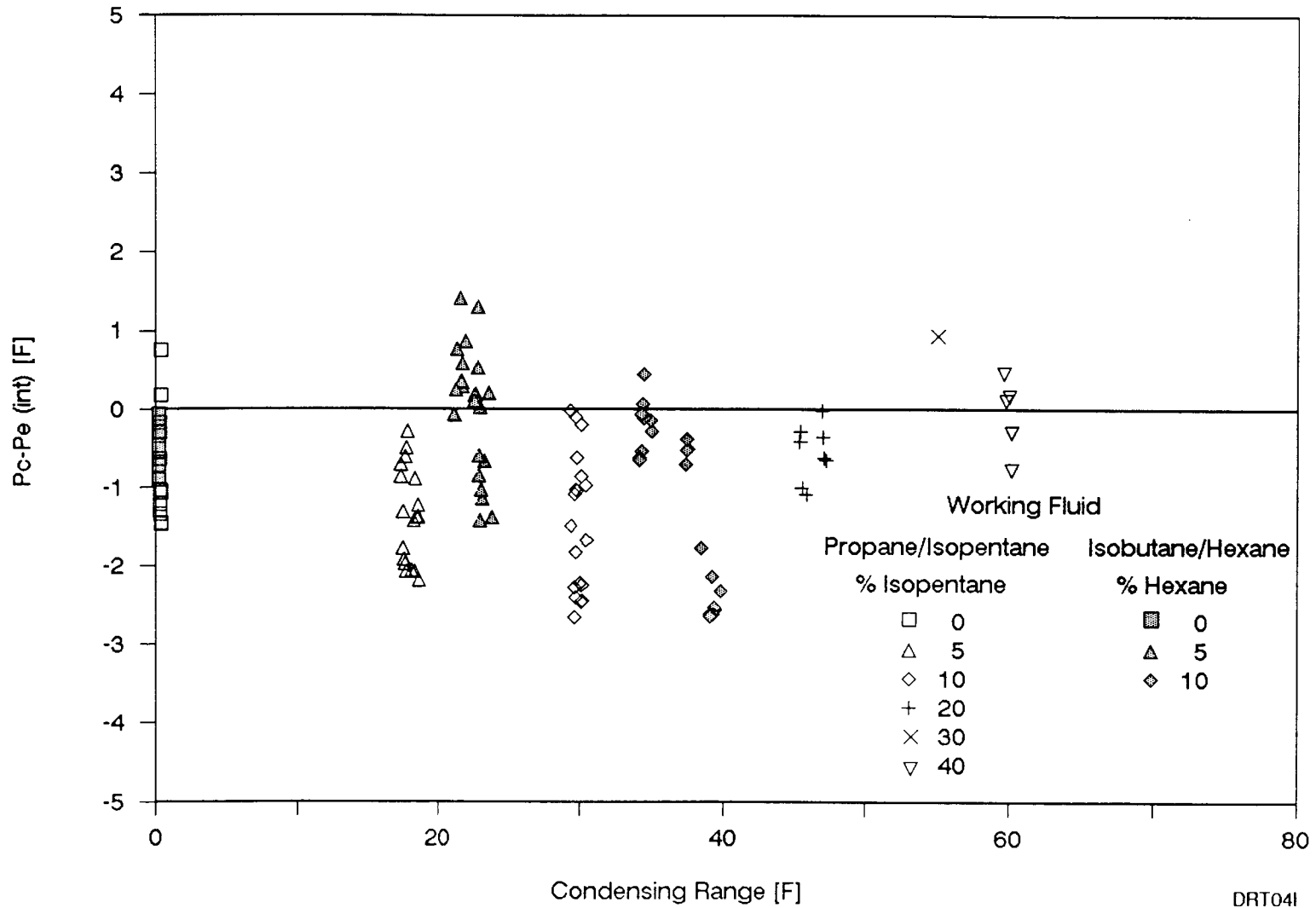
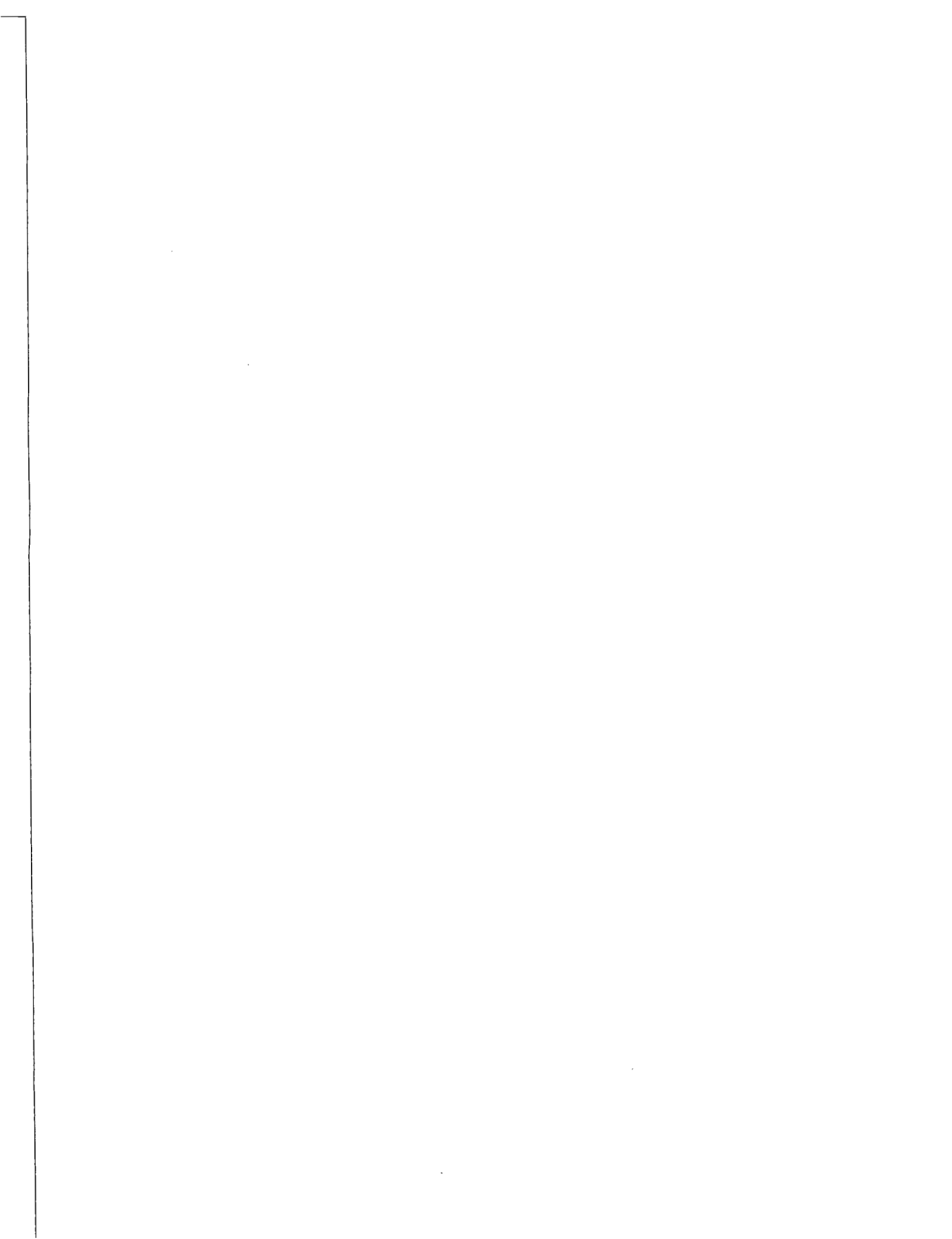


Figure 6. Pressure deviation under integral condensation assumption.



METHOD OF ANALYSIS

INTRODUCTION

The analysis of the condenser data from these experiments had a twofold purpose. First, data was obtained and verified for the phenomenon of the condensation of hydrocarbon mixtures inside finned tubes. Second, the data was used to determine how well a condenser similar to the one tested could be designed using standard techniques. To achieve these purposes, it was decided to use a computer program developed by Heat Transfer Research, Inc. (HTRI) to rate the condenser, because this code is commonly used for heat exchanger design, and a direct comparison between experiment and calculation will give a measure of how well the code serves as a design tool for this application.

In early 1988, HTRI introduced a new version of the shell-and-tube condenser program, CST-2 Mod 0.00-1.01. In the original work (Reference 1), CST-1 Mod 2.0 was used, modified in its application by utilizing the hydraulic diameter of the finned surface. This approach is described in detail in Reference 1. HTRI favored a simpler, less exact method, which "assumes" plain tubes of the actual internal diameter along with an area multiplier. At the Winter Annual Meeting of HTRI in February 1988, Ahmed Kassem of HTRI presented a variation of the method using options contained in the new version of CST which would handle condensation on augmented surfaces. Because of the existence of the new HTRI computer code, which includes this method for calculation of condensation on augmented surfaces, and which is available to industry; it was decided to switch to CST-2 MOD 0.00-1.01 and compare the results with the earlier analysis (as modified in Reference 1) for the vertical orientation. Unfortunately, it was found that the computer program would not use the input single phase correlations in the condensing calculations, so the original method (described in Reference 1) was used for this analysis. In addition, the simplified HTRI method was also used for comparison.

As discussed in Section 3.2, no significant deficiencies in the NBS properties were detected during the present experimental program in the calculation of properties of mixtures. Some uncertainty in bubble point properties, however, was indicated by the difference in bubble point temperatures at a given pressure shown by two saturation-line property options within EXCST (differences of about 1°F). On the recommendation of the code's author, the Peng-Robinson option was used.

DESCRIPTION OF THE CALCULATION METHODS

All calculations of condenser performance were carried out on the HTRI Condenser Computer Program CST2 MOD 0.00-1.01. Two models were used to approximate the behavior of condensation on the internally finned surfaces of the condenser tubes. The first method, the equivalent diameter model, was used extensively in the previous report which analyzed the case of the vertical condenser [1]. The second, the plain tube model, was originally recommended by HTRI. Both models result from the inability of CST to handle any geometry inside a tube other than a circular cross section.

Equivalent Diameter Method - In this method, the condenser tubes were approximated by plain tubes with an internal diameter equal to the hydraulic diameter of the finned tube as calculated in Reference 11. The wall thickness was assumed to be the nominal wall thickness of the finned tube. The number of tubes was determined to give the correct cross sectional (flow) area. This gives the correct inside surface area. The outside surface area is, however, in error. A multiplier was put on the outside convection resistance term ($1/h_o A_o$) to perform the necessary correction for the added fin area. The outside heat convection coefficient, h_o , was determined from computer runs with the correct outside geometry and entered as input to the program. h_o was found to be a function of the average cooling water temperature and the cooling water flow rate raised to the 0.6 power. This method should give the correct inside coefficients for desuperheating and the appropriate flow regimes. The condensing coefficients should be essentially correct for shear controlled flow regimes because the forced convection component is appropriately related to the Reynolds Number referenced to the hydraulic

diameter. The gravity controlled flow regimes should also be approximately correct in the vertical tube orientation because this method accounts for the liquid film thinning uniformly around the entire finned surface. For the horizontal orientation, this will not be correct.

Plain Tube Model - In this model, the tubes are approximated by plain tubes of the nominal inside and outside diameter of the finned tubes. Here the number of tubes is approximately correct because the finned cross section is a negligible part of the total flow area. The outside surface and flow conditions will be correct. The finned surface is accounted for by a multiplier on the inside convective resistance to account for the larger surface area. This multiplier might be expected to be the ratio of the actual finned surface area to the nominal surface area of the equivalent plain tube.

General Computational Methods - The condenser results were analyzed using CST2 MOD 0.0-1.01 and the thermodynamic properties (from the EXCST code) used in the analysis assumed completely mixed phases during the condensation (integral condensation). The condenser code treats variable working-fluid properties; the condenser is divided into a number of "constant-property" nodes. The model approximations described above were used to account for the presence of the internal fins on the working-fluid side of the tubes.

Because of the combination of very close approach temperature differences between working fluid and cooling water temperatures (as small as 1.5⁰F) in the condenser, and uncertainties in the condensing temperature as a function of measured condenser pressure, it was found that measured temperatures rather than measured condenser pressure, had to be used as code input quantities to best represent actual condenser conditions. The relationship between measured pressure and condensing temperature (bubble point temperature) contains some uncertainty in a number of items such as: pressure measurement accuracy, working fluid composition, accuracy of thermodynamic properties defining the saturation line, presence of noncondensibles, and the magnitude of condenser subcooling. The code was input assuming zero subcooling, zero pressure

drop in the tubing, and with the measured working fluid inlet and outlet state-point and flow conditions. Measured cooling-water-inlet and outlet temperatures were input, and the code was used to calculate a condensing temperature for which the required condenser heat-transfer area equalled the actual surface area. As will be discussed later, this calculated condensing temperature was correlated with the measured condenser outlet working-fluid temperature. Condenser pressure was determined from a pressure measurement in the working-fluid inlet piping corrected by a small calculated pressure drop (normally between 1 and 2 psi) from the pressure transmitter to the condenser inlet plenum, and correlated separately with the condensing temperature calculated by the HTRI computer code.

The simplifying assumptions of this method have, of course, introduced some potential deficiencies. The model does not treat some of the phenomena occurring in fin-augmented, gravity-controlled condensation. In the vertical orientation, one would expect that the finned surface was added vertical surface and, therefore, the condensate film would be spread uniformly around the entire surface at any point along the tube length. This effect, which tends to reduce the film thickness, is not included in the model and should result in underprediction of the heat transfer coefficients. (Additionally, there may be film thinning effects as a result of the curvature of the inside surface which is not included.) Further, the transition point from laminar to wavy-laminar to turbulent condensate film will not be totally correct because it depends on a Reynolds number based on film thickness.

The transition between shear controlled condensation and gravity controlled condensation may not be predicted quite correctly. If the hydraulic diameter were used in the "tubeside flow regime parameter" and the "condensation path parameter" because they come from force balances involving the friction factor, flow regime transitions might be more correctly modeled but at the expense of additional complication. This effect is examined in comparisons with the experimental data in Section 4.3.

The basic computer calculation is incremental and computes one desuperheating increment, one subcooling increment and seventeen condensing increments. This allows for correct interpretation of the varying heat transfer coefficient and stream-to-stream temperature difference. A more complete description of the working of the computer program is given in Reference 1.

METHODS OF COMPARISON OF EXPERIMENTAL AND ANALYTICAL RESULTS

Certain assumptions are necessary to carry out the design-type calculation using the HTRI method. The first assumption concerns the condensing curve for the working fluid. From the discussion in Section 4.2, it appears that integral condensation occurred with a maximum of 1⁰F of subcooling. As a first baseline assumption, the condensation was assumed to be integral with no subcooling. The second assumption concerned the effectiveness of the finned surface. Using the actual area ratio in the new HTRI method assumes that all of the surface was effective and at the same surface temperature. This implies a fin efficiency of 100%. There may be surface tension effects which thin films and create effectively higher heat transfer coefficients in gravity controlled flows when the orientation is near vertical. When the orientation is near horizontal, the fins may block the natural drainage of the condensate from the tops of the tubes. Both of these effects may be compensated for in the area multiplier (safety factor). These considerations suggest that the multiplier may need to be greater or less than the actual area ratio for accurate computations. As a second baseline assumption, the multiplier was taken to be the actual area ratio.

The first method of comparison uses both baseline assumptions. Here, it was assumed that there was integral condensation with no subcooling and that the fins were totally effective. For this comparison, the deviation between the calculation and the experimentally measured data is expressed as the ratio of the overall heat transfer coefficient calculated by the computer program to that experimentally measured. The comparison assumes that the deviation results from either experimental error or error in the calculation of the heat transfer coefficients. When this ratio is less

than one, it indicates that the calculational method is conservative in the design situation. That is, if the calculational method is used for design, it will predict that a larger surface area is required than was experienced in the experimental work. This method tests the overall calculation and gives a factor which could be used in conjunction with the computation for design.

The second method of comparison assumes that the calculated heat transfer coefficients are correct and that the fins are as effective as their area enhancement. The condensing pressure, as indicated by the bubble point temperature is changed to give agreement with the experimental data. This calculation is made assuming that the condensation is integral.

The third method assumes that the heat transfer coefficients are calculated correctly and that there is no subcooling of the working fluid after condensation. The effective area enhancement of the finned surface is varied to match the calculation with the experiment.

A fourth method of comparison of experiment to calculation exists in the incremental temperature measurements which have been taken for cooling water as it flows through the unit. This data may be able to discern deviations in different flow regimes, such as desuperheating, gravity-controlled condensation, shear-controlled condensation and subcooling. This will be discussed in greater detail in the next section.

RESULTS

Results are presented for tests of the supercritical binary cycle for which the counterflow in-tube condenser is oriented nearly horizontally. Nominal working fluids tested consist of the isobutane-hexane family with 0, 5, 10% hexane (by mass), and the propane-isopentane family with 0, 5, 10% isopentane. In the isobutane-hexane family, the flow direction was changed to allow for both parallel and counterflow test series, and also for a series of tests with half of the tubes blocked (both in counterflow and parallel flow). In addition, the mixture composition in the propane-isopentane family was changed in increments of 5% to as high as 40% isopentane. For each of these test series, condenser data were analyzed for from six to ten tests.

In the following discussion of the test results, the experimental evidence of integral condensation is considered first. Achieving integral condensation is a crucial assumption in the analysis of the test data. Then, a comparison is made of the measured performance of the condenser in its vertical orientation to the measured performance in the horizontal orientation. Small analytical corrections were applied to the vertical test results to correct for the slightly different operating conditions used for the tests being compared. (The vertical results were corrected instead of the horizontal results because of the excellent correlation between observed and predicted performance for the vertical orientation.) The last part of this section discusses comparison of the new calculational method and the experimental data for the near horizontal orientation. One of the prime objectives of this work is development of the ability to design the components in these advanced plants. By comparing the experimental results with predictions using the Heat Transfer Research, Inc. (HTRI) computer programs, one of the most universal design tools for heat exchangers is verified.

EXPERIMENTAL EVIDENCE OF INTEGRAL CONDENSATION

The comparisons of calculated and measured condenser pressure discussed in Section 3.2 (Figures 5 and 6, assuming integral condensation) indicate that the condensation was integral and that little subcooling resulted.

As a further verification that significant differential condensation does not occur; Figure 7, for the same data shown in Figures 5 and 6, shows the difference between the bubble point temperature calculated for differential condensation (with the working fluid leaving at the measured condensing pressure) and the measured outlet temperature. For this graph, the scale on temperature difference is approximately ten times that in Figures 5 and 6. For reference, the data in Figure 5 is replotted in Figure 8 to the same scale as Figure 7. In Figure 7, there is a deviation which is dependent on the composition of the working fluid, whereas, in Figure 8, there was no trend with composition. (Note that for a pure working fluid (condensing range of 0), there is no difference between integral and differential condensation.) The magnitude of the deviation in Figure 8 is approximately 1 °F change in bubble point temperature and for most data, whereas in Figure 7 the magnitude of the change is much larger for compositions far from pure fluids. The greater the deviation from the composition of a pure substance, the greater the deviation between results in Figures 7 and 8. This comparison indicates that differential condensation did not occur to an appreciable degree in these tests.

PERFORMANCE OF CONDENSER IN HORIZONTAL ORIENTATION COMPARED TO THE VERTICAL ORIENTATION

Similar test conditions were used in the testing with the condenser in the vertical orientation and the horizontal orientation. Flow rates of working fluid and cooling water were matched along with the amount of superheat on the working fluid entering the condenser. Therefore, experimental comparisons of performance in the two orientations are possible.

Table 1 shows the matching of runs with the condenser vertical to those with the condenser near horizontal. The ratio of the mass flow rate in the vertical orientation to that in the horizontal orientation is given for both the working fluid and the cooling water. In addition, the difference in the amount of superheat in the working fluid entering the

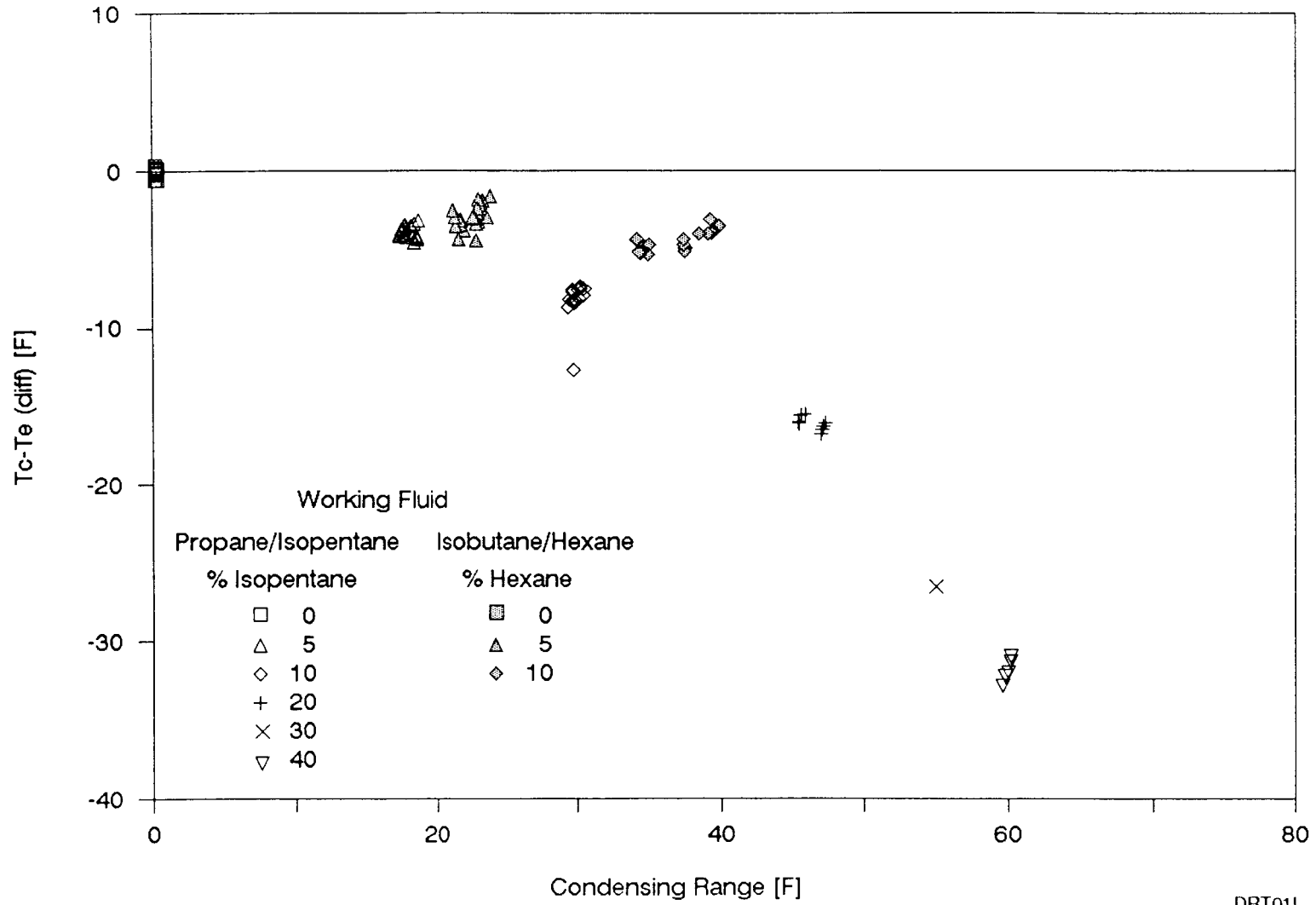


Figure 7. Temperature deviation under differential condensation assumption.

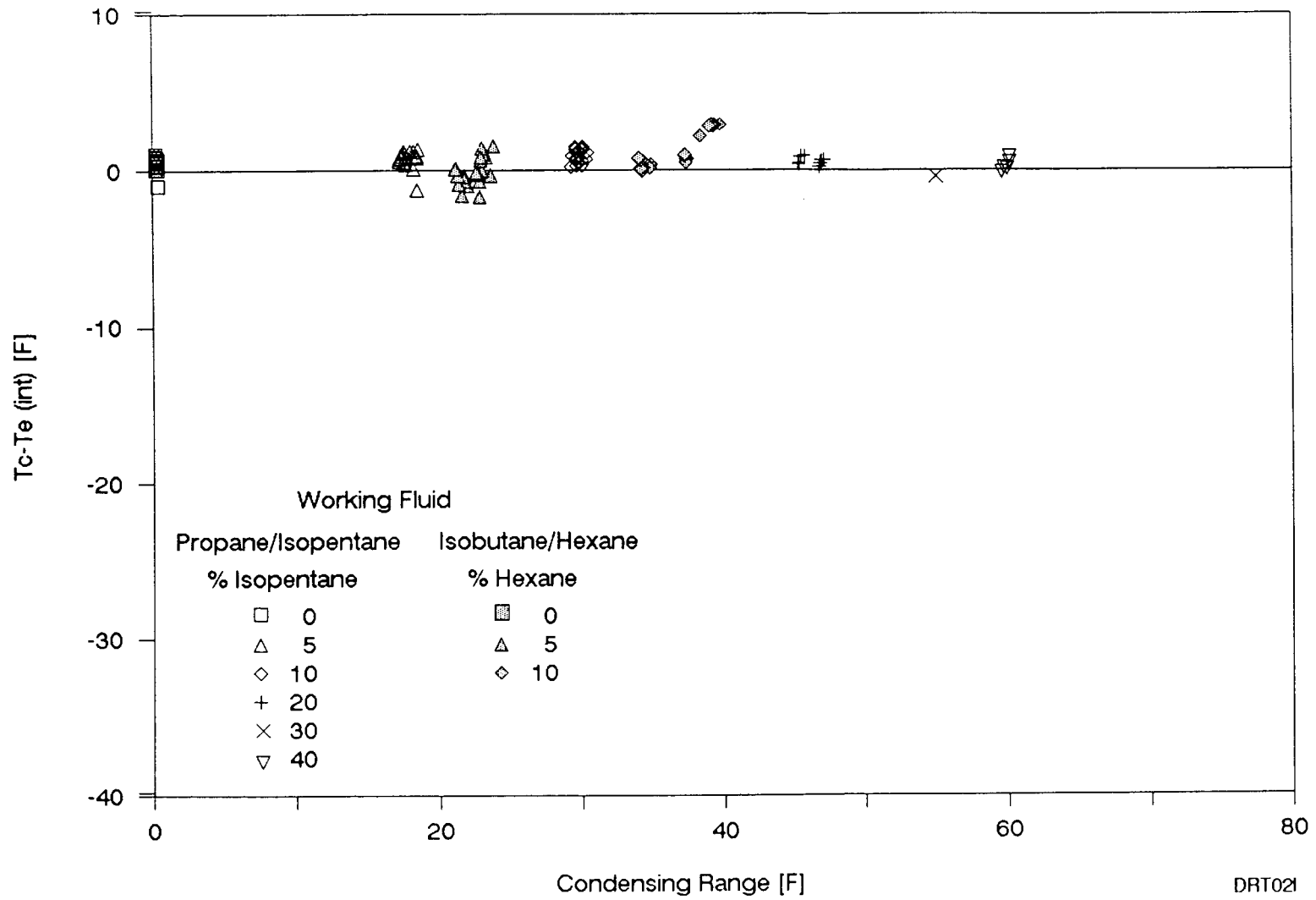


Figure 8. Temperature deviation under integral condensation assumption.

Table 1. Matching of vertical to horizontal tests.

VERTICAL RUN	HORIZONTAL RUN	RATIO OF TO VERTICAL HORIZONTAL		DIFFERENCE BETWEEN VERT. AND HORIZ.	
		WF FLOW RATE	CW FLOW RATE	SUPERHEAT [F]	CONDENS. RANGE [F]
A028A	A200	0.8270	0.9971	0.0	1.00
A028C	A202A	1.0066	1.0121	4.4	0.94
A053	A205	0.7211	1.0014	-4.4	0.90
A061	A206	1.0035	0.9996	-3.1	0.85
A066	A207	0.9935	1.0017	-9.8	0.86
A071	A208	1.0086	1.0238	-2.1	0.88
B028A	B200	0.7983	0.9992	3.7	-1.20
B055	B205A	0.9686	1.0027	-4.5	0.72
B061	B206A	0.9806	1.0039	-3.3	0.81
B066	B207A	0.9787	1.0059	-0.8	0.84
B071	B208A	0.9467	1.0048	3.4	0.05
B076	B209A	0.9337	1.0065	0.0	-2.75
C100	C200	0.9617	1.0016	-51.2	-1.93
C102	C208A	0.9521	1.0010	2.6	0.75
C103	C208A	1.0486	1.0004	1.6	0.75
D028X	D202A	1.0495	1.0148	3.6	0.00
D055	D205	1.0469	1.0085	14.3	0.03
D062	D206	0.9974	0.9999	-0.4	0.01
D065	D207	1.0143	1.0104	7.4	0.02
D071	D208	1.0227	1.0098	2.0	0.01
E028C	E202	0.9647	0.9573	9.9	0.19
E061	E206	1.0100	0.9843	1.5	0.43
E066	E207	1.0129	0.9893	1.5	0.45
E071	E208	1.0079	0.9869	-0.8	0.48
E076	E209	1.0202	0.9873	0.0	0.35
F028A	F200	0.8003	1.0112	15.4	-0.07
F028C	F202	0.9521	1.0118	18.7	0.12
F055	F205	1.0018	0.9340	19.8	-0.14
F061	F206	1.0124	0.9937	4.4	0.14
F066	F207	1.0071	0.9934	5.0	0.20

condenser is shown. Note that 90% of the working fluid flow rates are within 10% of matched cases and 95% within 20%. All of the cooling water flow rates are within 5%. The differences in the amount of superheat are generally less than 10°F with 5 exceptions. In most runs with large deviations, the amount of superheat varies from around 100°F to 50°F. The corrections in the extreme cases were only in the order of 5% of the experimental value. The condensing ranges of the working fluids was generally within 1 °F with the maximum deviation being 2.75 °F.

The only problem was that the absolute condensing temperatures and pressures could not be compared directly because the cooling water temperature changed with the time of day and year. Corrections for this condition were made by adjusting the vertical data to conditions for the corresponding horizontal test by an increment calculated from the computer program assuming a vertical orientation. Because of the accuracy of the calculation in prediction of the vertical experimental data as shown in Appendix B, this approach was felt to be justified.

The corrections to the measured overall heat transfer coefficient using the computer results for the different process conditions averaged about a 4% change in the measured value with the largest correction being less than 10%. Similar variation was noted in the local condensing coefficient correction with an average correction of 5% and a maximum correction slightly greater than 10%. The correction was small when compared to the measured values.

Figure 9 compares the performance of the vertical and horizontal condenser orientations using the first method of comparison, the overall heat transfer coefficient. The data for the matched runs is plotted against the fraction of the total heat load which was desuperheating. In desuperheating, the orientation of the condenser should have negligible effect on the heat transfer and the ratio of heat transfer coefficients should approach unity. The ratio of overall heat transfer coefficients does appear to approach unity as the desuperheating fraction increases. Scatter in the data is increased because of the use of two experimental points for each point on the plot. There appears to be no bias associated

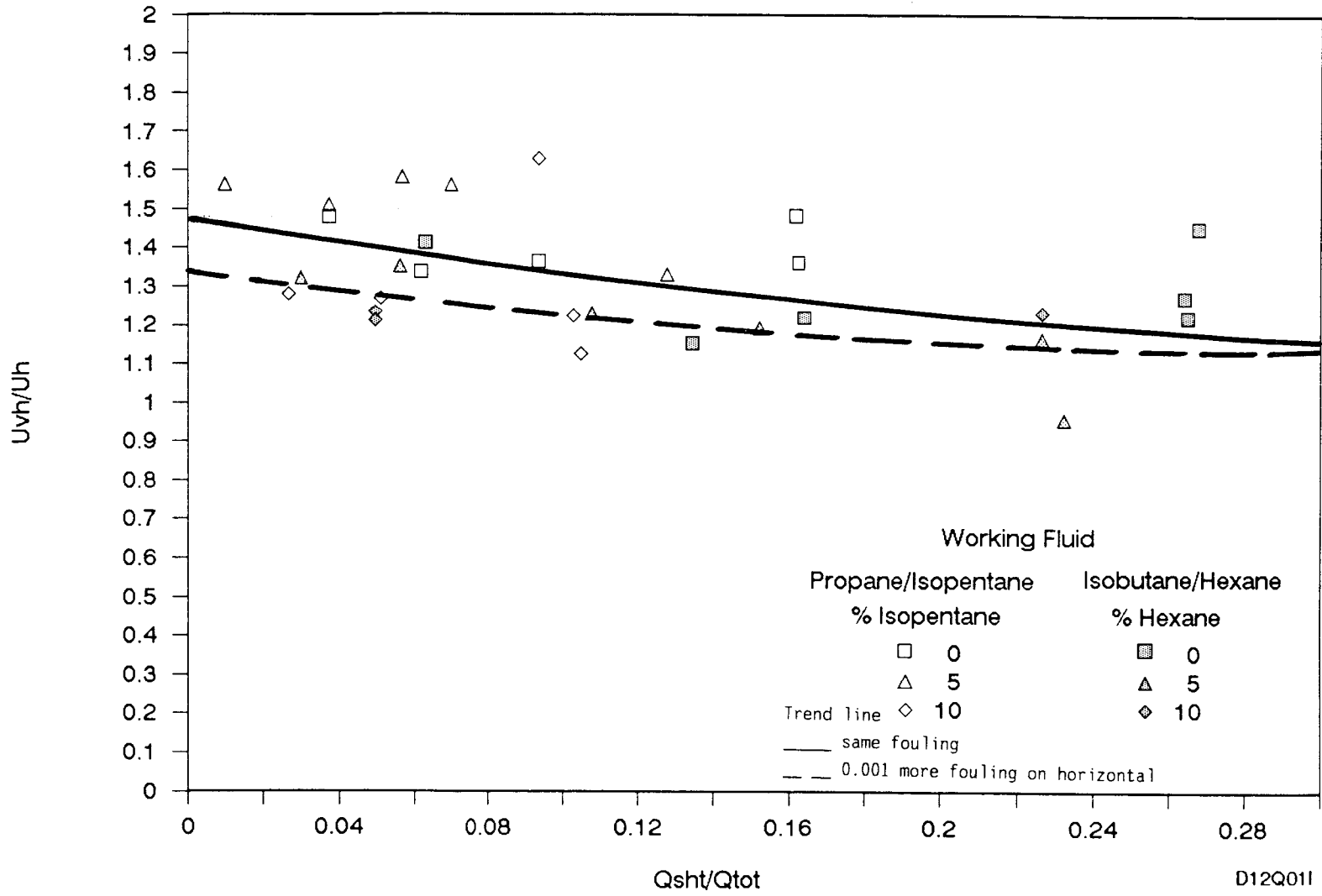


Figure 9. Overall heat transfer coefficient ratio vertical to horizontal comparison.

with the composition of the working fluid, however. The solid line on the plot is a curve-fit of a cubic equation which approaches U_v/U_h of one as the system approaches a desuperheating condition throughout (the desuperheating fraction of the total heat load approaches one). Actually, the shear-controlled portion of the condensation should also be the same for the two geometries. Extrapolation of the data to a purely condensing load (desuperheating fraction of zero) indicates that the overall heat transfer coefficient for the vertical orientation is approximately 47% greater than that in the corresponding horizontal orientation. As the fraction of desuperheating increases, the differences decrease. This would indicate that to perform the same duty in the horizontal orientation would require, on the average, an exchanger 47% larger than one in the vertical orientation.

The horizontal testing period was one to two years after the vertical testing period. The above comparison assumed that the fouling resistance was the same for both cases. If one assumed that the incremental change in fouling resistance between the two tests was $0.001 \text{ hr ft}^2 \text{ }^\circ\text{F}/\text{Btu}$, the curve-fit average would have been the dashed line. This is an extreme increment because the asymptotic fouling on the cooling water side is expected to be less than this value. However, this would have changed the ratio for pure condensation to 33% from the 47% mentioned above and similarly a reduction in the required area to perform the same service to 33% greater in the horizontal orientation.

Figure 10 expresses the difference in calculated and measured values to a more absolute base. Here, the thermal resistance which would be needed to make the calculated overall heat transfer coefficient (calculated with no fouling resistance) and the experimental coefficient equal to one another. The plot shows the relationship between this thermal resistance and the condensing range. The maximum difference in condensing range between vertical and horizontal runs was approximately $2 \text{ }^\circ\text{F}$ with the average well below $1 \text{ }^\circ\text{F}$. The line represents a least-squares straight line through all of the data. This indicates that for a pure fluid (zero condensing range) the incremental thermal

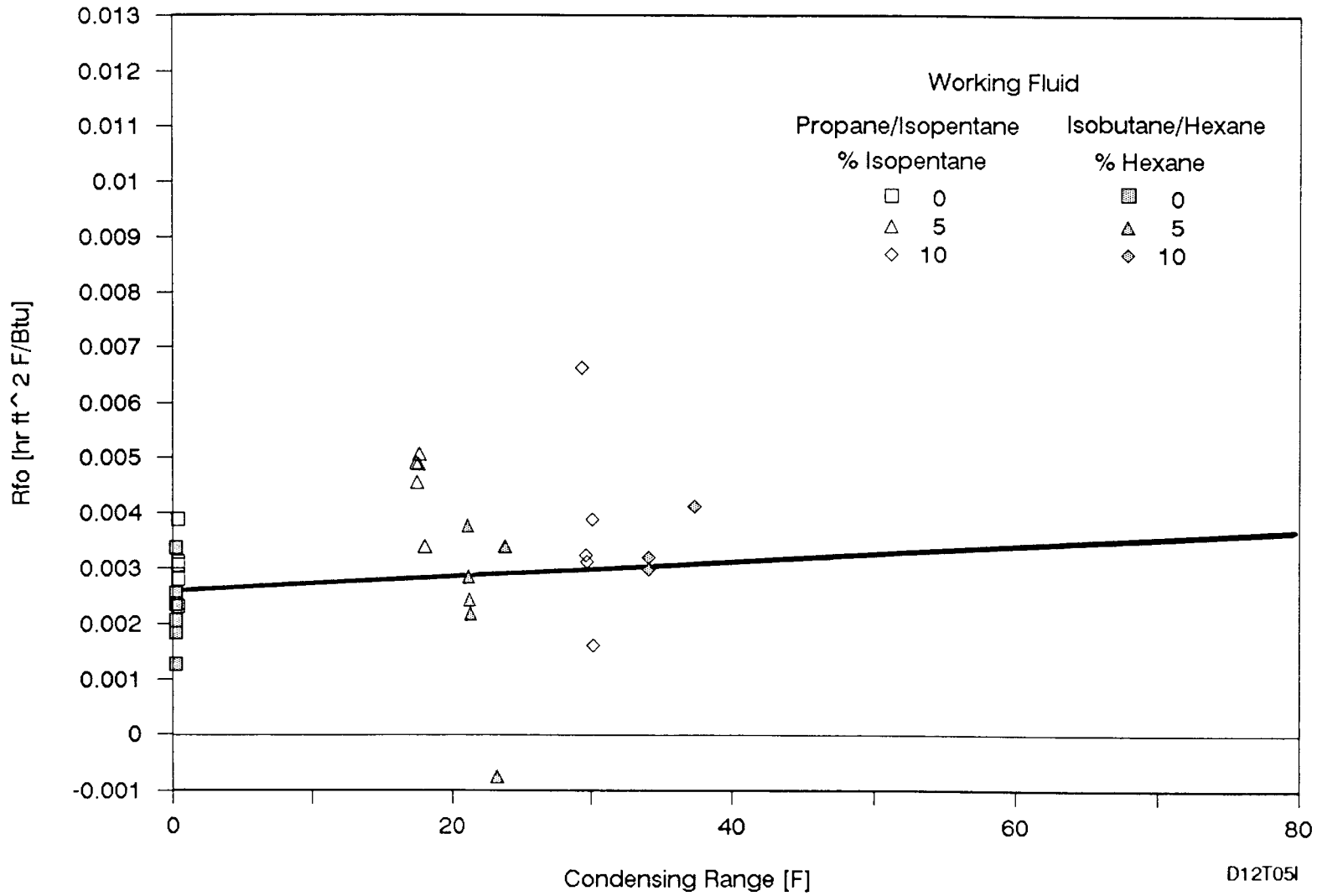


Figure 10. Overall heat transfer coefficient incremental thermal resistance vertical to horizontal comparison.

resistance would be $0.0027 \text{ hr ft}^2 \text{ }^\circ\text{F}/\text{Btu}$. This is higher than the "design" fouling which would be expected to be between 0.001 and 0.002. It is not felt that this is related to fouling of the heat exchanger, but to a difference in heat transfer in the two orientations. There is some dependence on the condensing range of the working fluid. The curve-fit showed that incremental resistance would double in going from a pure fluid to a fluid with a condensing range of $109 \text{ }^\circ\text{F}$. Condensing range appears to be an appropriate method of correlation of the working fluids because the isobutane mixtures (solid symbols) and the propane mixtures (open symbols) follow the same trend.

Figure 11 expresses the comparison in slightly different terms. Here the difference in working fluid outlet temperature is plotted against desuperheating fraction. This comparison shows that since there is little or no subcooling (See the section on the assessment of data and thermodynamic consistency), the outlet temperature of the condensing fluid in the vertical orientation is 1 to $3 \text{ }^\circ\text{F}$ lower than in a horizontally oriented condenser of identical geometry with the same inlet conditions. The behavior is somewhat dependent on the fluid with the pure fluids showing less difference in temperature than the mixtures. The close approach temperature differences (pinch points) with pure fluids results in greater changes in heat transfer with smaller changes in condensing temperature than for the mixtures. This method is not easily extrapolated to other situations and is shown to give a general idea of the effect. This behavior would result in a decrease in power generated in the turbine as a result of higher back pressure for the horizontal condenser. As with the previous comparison, as the desuperheating fraction increases, the deviation between the orientations decreases. As with the overall heat transfer coefficient results, the initial assumption was that the vertical and horizontal systems had the same amount of fouling. This is possibly not the case.

Figure 12 shows a comparison of the inside condensing coefficients of the vertical and horizontal condensers. This comparison assumes that the outside coefficient in both measured cases was identical; this should be a good assumption because of the correction applied to the vertical

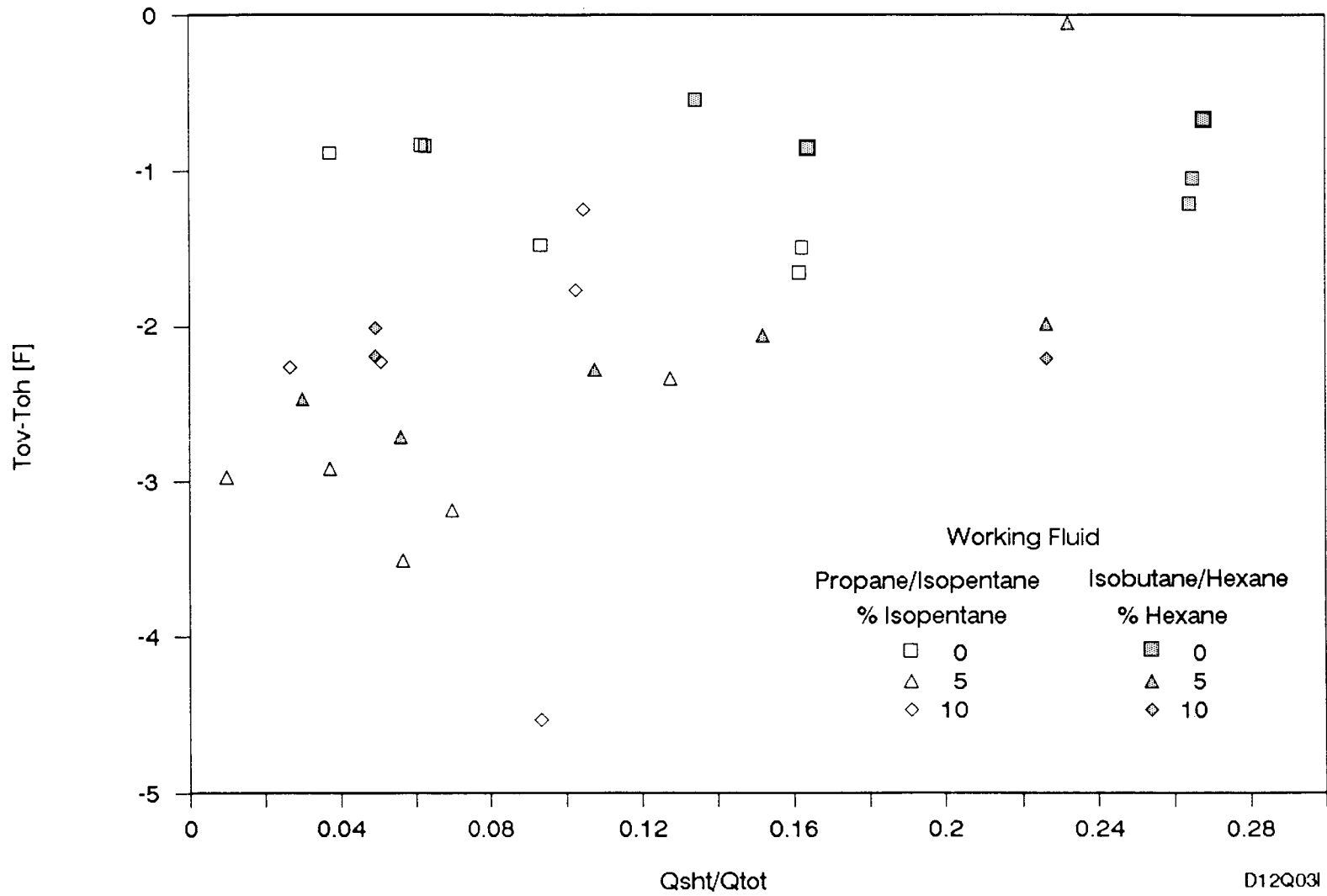


Figure 11. Difference in working fluid outlet temperature vertical to horizontal comparison.

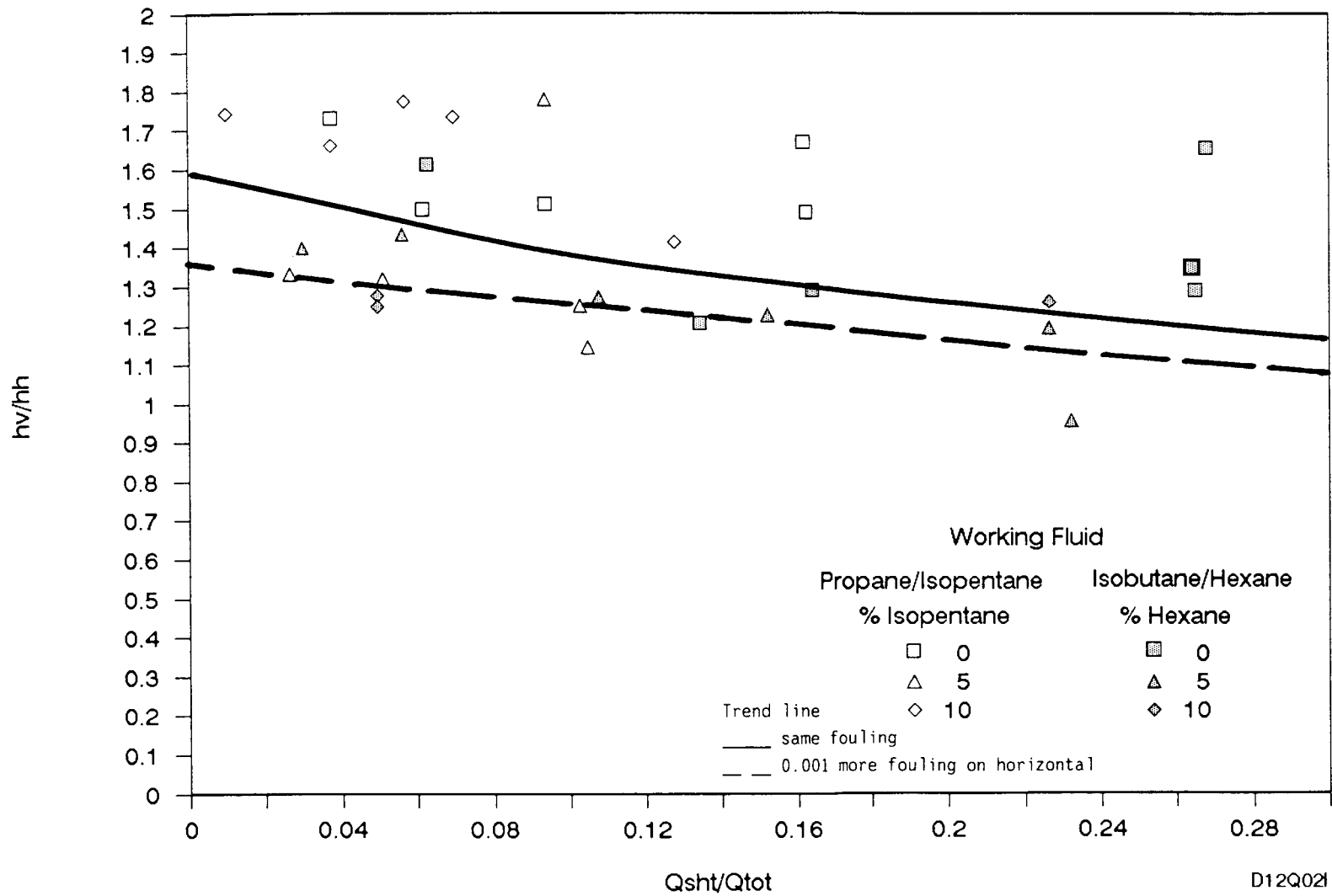


Figure 12. Condensing heat transfer coefficient vertical to horizontal comparison.

measurement which should correct for any difference in outside (water-side) coefficient. The solid line is a curve-fit as discussed for Figure 9. Difference in the condensing coefficient of between 40 and 80% are noted. For a purely condensing load, the average value is 61%. Again, these results assume the same fouling in the vertical and horizontal cases. If the horizontal case has a fouling $0.001 \text{ hr ft}^2 \text{ }^\circ\text{F/ Btu}$ greater than the vertical, the dashed line represents the average trend. Note here the difference in condensing coefficients for a purely condensing load is 34%. Because this calculation is made on the basis of the finned area, the difference is probably a result of the fins not operating as effectively in the horizontal orientation as in the vertical orientation. In the vertical orientation, the fins give extra vertical surface for condensation to occur upon. In the horizontal orientation, the fins may be interrupting the natural drainage of film around the circumference of the tube.

To summarize, in condensing:

1. The overall heat transfer coefficient appears to be 33 to 47% higher, on the average, in the vertical orientation than in the horizontal orientation depending on the change in fouling.
2. This could result in a decreased size for a vertical condenser over a horizontal condenser of 33 to 47%.
3. In terms of thermal resistance, the difference in orientation gives an additional thermal resistance of between 0.0027 and $0.0037 \text{ hr ft}^2 \text{ }^\circ\text{F/ Btu}$ for the mixtures up to 10% of the heavier component. (Although the vertical and horizontal data was taken at different times and some fouling may have taken place between the tests, it is doubtful that it could have been of this order.)
4. The condensing heat transfer coefficient would be between 34 and 61% higher in the vertical orientation than in the horizontal orientation, depending on the amount of fouling increase between the two tests. (The lower value is expected to be closer to the actual difference because there was some fouling during the vertical tests and the value of 0.001 is large relative to the asymptotic value of cooling tower water fouling factor.

5. It appears that the fins are totally effective in the vertical orientation (possible with an additional film thinning due to surface tension), while the fins appear to be ineffective in the horizontal orientation and indeed may cause a reduction in performance.

ANALYTICAL RESULTS FOR THE CONDENSER IN THE HORIZONTAL ORIENTATION--COMPARISON OF HTRI CODE WITH EXPERIMENT

Design of the heaters and condensers can be done using computer codes like those of Heat Transfer Research, Inc. (HTRI). These codes are generally known to the A & E firms designing systems similar to binary geothermal plants and to the heat exchanger manufacturers.

The HTRI code ST5 was developed to design heat exchangers without phase change, such as the supercritical heaters in a binary plant. The main problem with this code is that it assumes constant thermophysical properties and linear temperature distributions with enthalpy within the exchangers. In the design of the Heber plant, Fluor Engineers, Inc. was able to use ST4, the forerunner to ST5, because there were six different units in series and the variation in each was small (13).

At the Heat Cycle Research Facility, the heaters have finned tubes and the overall length is significantly reduced. To model the heater, each unit was divided into increments and the code was used on each increment. Here end corrections automatically added in the code had to be removed to give the correct result. Details of this procedure are given in Reference 1 along with the experimental verification of this modeling technique.

The condenser at the Heat Cycle Research Facility (and the configuration recommended for further application) has the working fluid condensing inside the tubes on an internally finned surface. As was mentioned in the previous section, the HTRI condenser code CST2 does not explicitly handle such surfaces. This section discusses the comparison of the experimental results to the two computational models mentioned previously, the equivalent diameter model and the plain tube model.

The equivalent diameter method - Figure 13 shows the ratio of calculated to measured overall heat transfer coefficient plotted against the fraction of the heat load which is in desuperheating. The results plotted are for the tests in which the heat exchanger flow was countercurrent and no tubes were plugged. If it is assumed that the single phase heat transfer can be calculated correctly, the ratio of heat transfer coefficients will approach unity as the entire load becomes desuperheating. The solid line is a curve-fit using this assumption with a cubic equation. This indicates that if the deviation results from the condensing section of the exchanger, the overall heat transfer calculation is 2.135 times as large as the experimentally measured value. The calculation was carried out assuming that there was no fouling in the exchanger. If a fouling factor of $0.001 \text{ hr ft}^2 \text{ }^\circ\text{F} / \text{Btu}$ is introduced into the calculation, the curve-fit average would be displaced to the dashed curve. This is of the order which the asymptotic fouling might be in this unit. (Preliminary calculations using the HTRI cooling tower water fouling model indicate that the asymptotic coolant fouling should be in the range 0.0005 to $.0007 \text{ hr ft}^2 \text{ }^\circ\text{F}/\text{Btu}$.) This brings the calculated and measured values to closer agreement, but with pure condensation, fraction of desuperheating equal to zero, the calculated value is still 85.8% greater than the measured value. Another potential cause of disagreement between the calculated and measured values is the uncertainty in the actual fluid mixture composition. Calculations were carried out with compositions of the lesser component (the heavy component) 10% greater and less than the measured value. (That is for a 90%/10% isobutane hexane mixture, calculated values were determined for 91/9 and 89/11% mixtures.) The deviations when plotted on Figure 13 were found to be of the order of the width of a symbol and have, therefore, not been shown. Note that this change not only changes the calculated overall heat transfer coefficient, but also changes the desuperheating fraction and the condensing range. The desuperheating fraction is changed between 0.005 and 0.006 and the condensing range is changed between 2.2 and 2.7 $^\circ\text{F}$.

Figure 14 shows the same data plotted against the condensing range of the working fluid. If mixture effects are the cause of the disagreement, they should show up in this type of plot. Although the agreement is

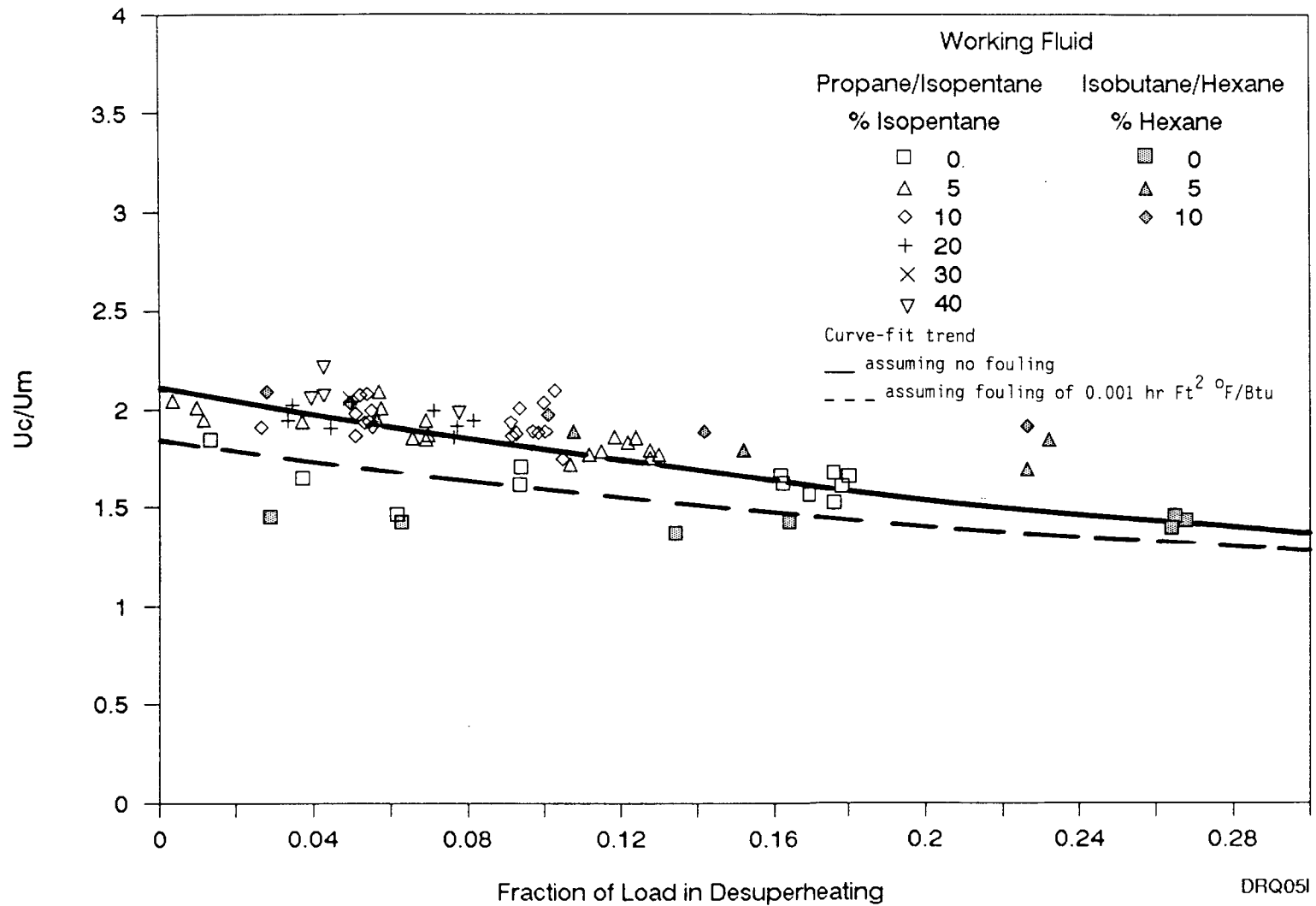


Figure 13. Comparison of equivalent diameter method calculation to experiment overall heat transfer coefficient.

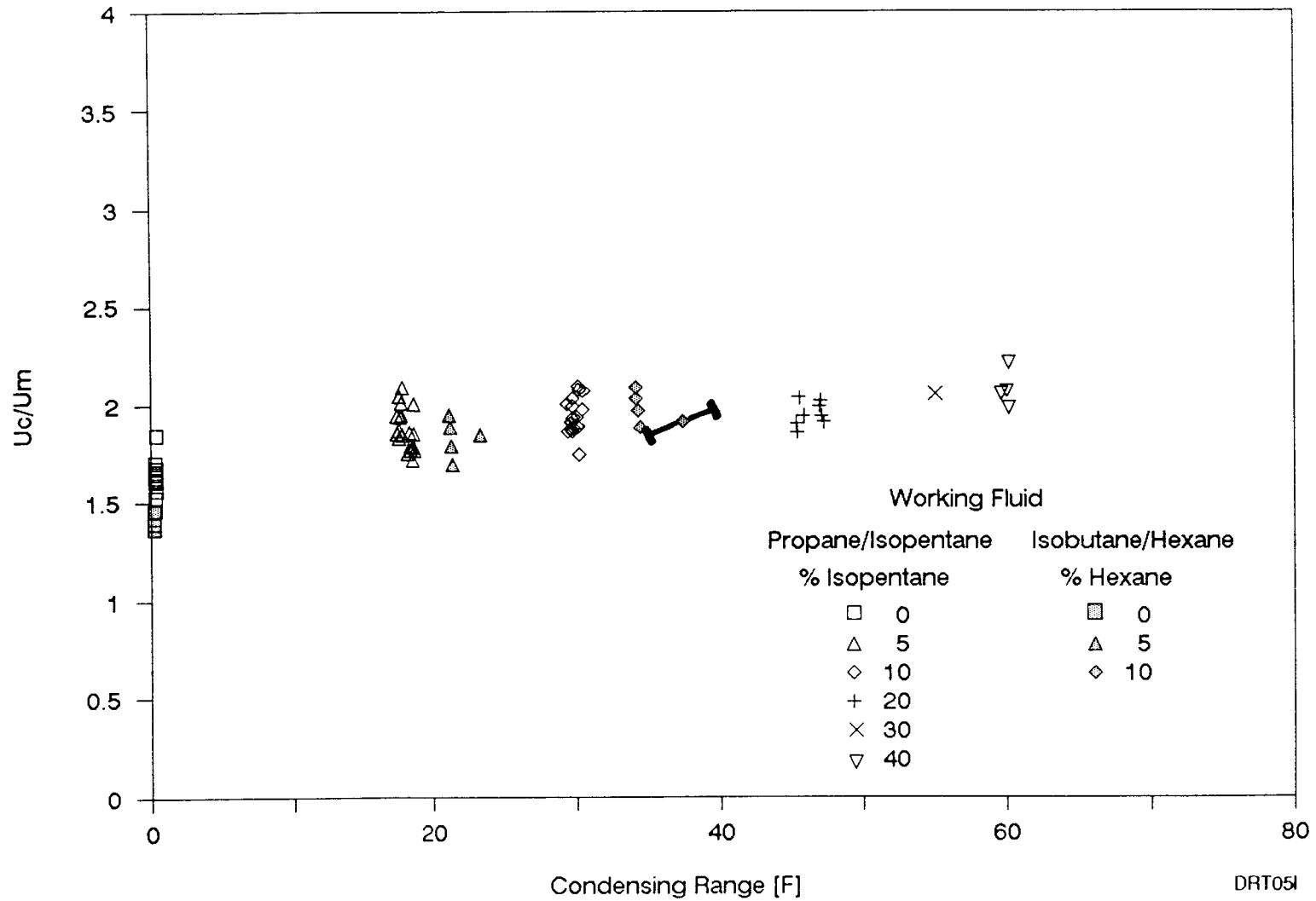


Figure 14. Comparison of equivalent diameter method calculation to experiment overall heat transfer coefficient.

better for the near pure substances, condensing range of zero, there is no clear trend of the data. Error bands for the composition changes described in the discussion of Figure 13 are shown here. Note that the error bands are marked on the data point at 37.3 °F. However, if this data is replotted in terms of the absolute difference in thermal resistance between the measured and calculated values, a definite trend develops. Figure 15 shows this parameter plotted with the condensing range (mixture composition). Figure 15 contains all of the data for runs in which the condenser had no tubes plugged. This included some data with the coolant flowing cocurrently with the condensing working fluid. Exclusion of the cocurrent data did not change the overall trend line as shown on the graph. The line represents a least squares fit of the data. The movement of the results with changing composition is somewhat parallel to the trend line. It is expected that errors in composition will not appreciably effect the results presented here.

An interesting comparison can be made between this figure for the near horizontal orientation and Figure B-3, the corresponding figure for the vertical orientation. For the pure fluids, the difference in thermal resistance between the horizontal and vertical results (Figure 15 and B-3) is about $0.006 \text{ hr ft}^2 \text{ }^\circ\text{F} / \text{Btu}$. This is approximately ten times the expected asymptotic fouling for the cooling water. This difference is postulated to be a combination of fouling (a minor part) and the detrimental effect which the fins produce in the horizontal tube. In the vertical tube, the fins add vertical surface for condensation, thereby thinning the condensing film and enhancing the heat transfer. Surface effects may give some additional film thinning as a result of the change in direction between the fin and the tube surface. In the horizontal tube, however, the fins may be an impediment to the natural drainage of the liquid film from the top and sides of the tube to the bottom thus creating lower condensation coefficients than in a plain tube.

The slope of the resistance curve with condensing temperature range (difference between bubble point and dew point temperatures) is quite large. The slope is comparable in magnitude to the slope for the vertical configuration resistances with isobutane working fluids. The propane

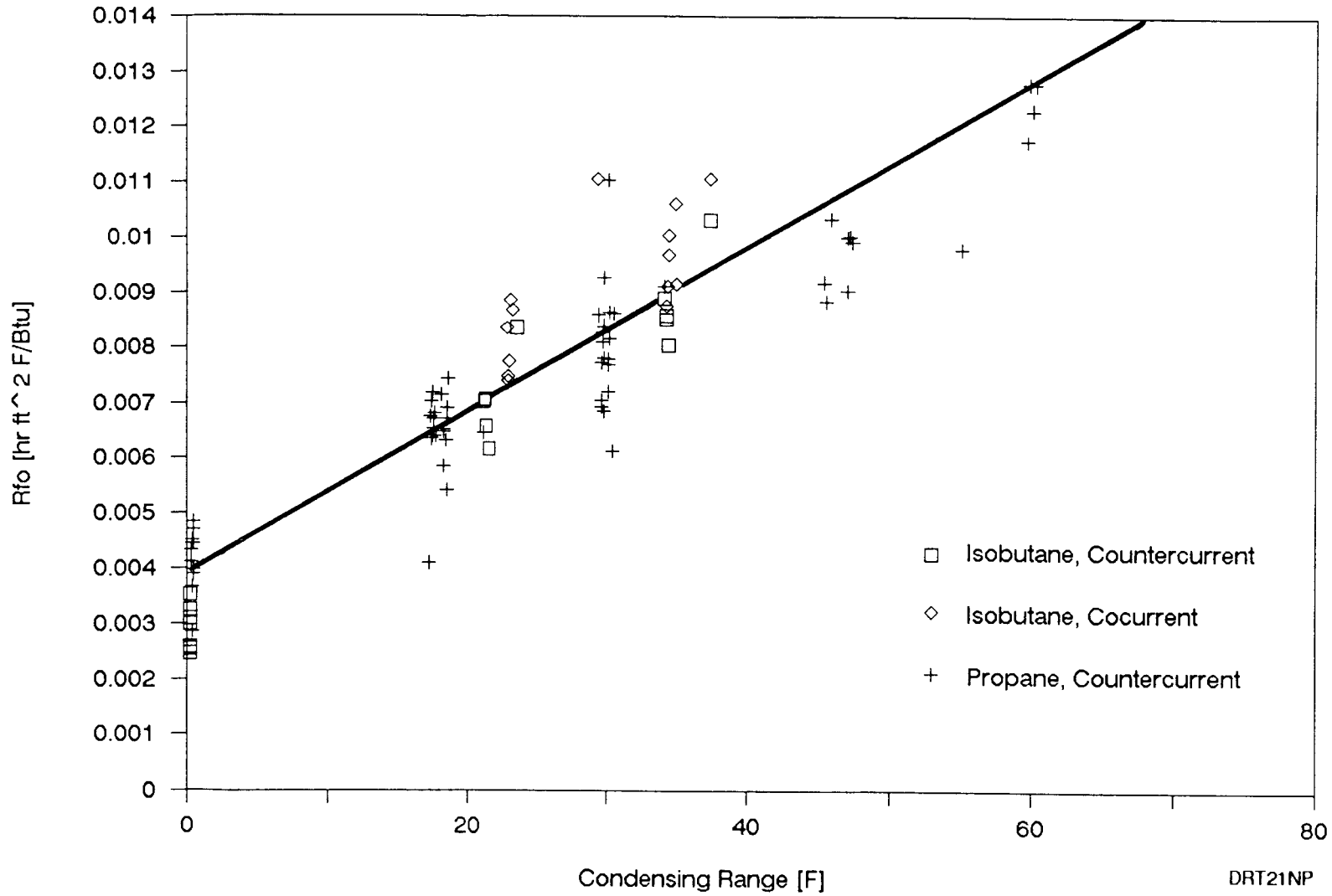


Figure 15. Comparison of equivalent diameter method calculation to experiment thermal resistance (no tubes plugged).

series appears to have a lower slope for the vertical orientation. At this time, the reason for this is unknown. This dependence of deviation between calculated and experimental values on condensing range indicates that some aspects of the mixture condensation process are not being calculated correctly in the HTRI program. The reason for this deviation is possibly magnified by the particular process conditions for this experiment. The liquid loadings are low and the effect of the fins is to decrease the effective diameter of the tube. Each of these effects tends to decrease the Reynolds Number. The regions near the wall between the fins will be regions of low velocity and perhaps will have significantly less macroscopic (turbulent) mixing than in a normal plain tube. For these reasons, molecular diffusion may play a more important role than for a plain tube. It is possible, therefore, that it will be necessary to use a combined heat and mass transfer model such as Colburn and Drew [14] and Krishna and Standart [15] rather than an approximate method based on Silver [16] and Bell and Ghaly [17] with mass transfer corrections. (A computational methodology for the Colburn-Drew method was developed by Price and Bell [18].) Figure 16 shows the same type of data for the cases in which half the tubes were plugged. Here, the trend with increasing condensing range is less pronounced than in Figure 15. It might be expected that the higher flow rate per tube in the plugged tube case would result in better mixing and less adverse concentration gradients and temperature gradients. Indeed, the HTRI program, CST, uses different correlations for the condensing coefficient at higher liquid loadings. It is estimated that different flow regimes are encountered. The data, however, does not correlate with working fluid flow rate as is indicated in Figure 17. There may be some effect, but it is not the predominant effect.

This same data can be presented in a number of other ways. Figure 18 and 19 show the inside (condensing and desuperheating) convection coefficient plotted as the ratio of calculated (by the computer program) to the measured value. This comparison assumes that the entire inner surface is active (the fin efficiency is 100%.) Figures 20 and 21 show this data under the assumption that the area is only partially active and

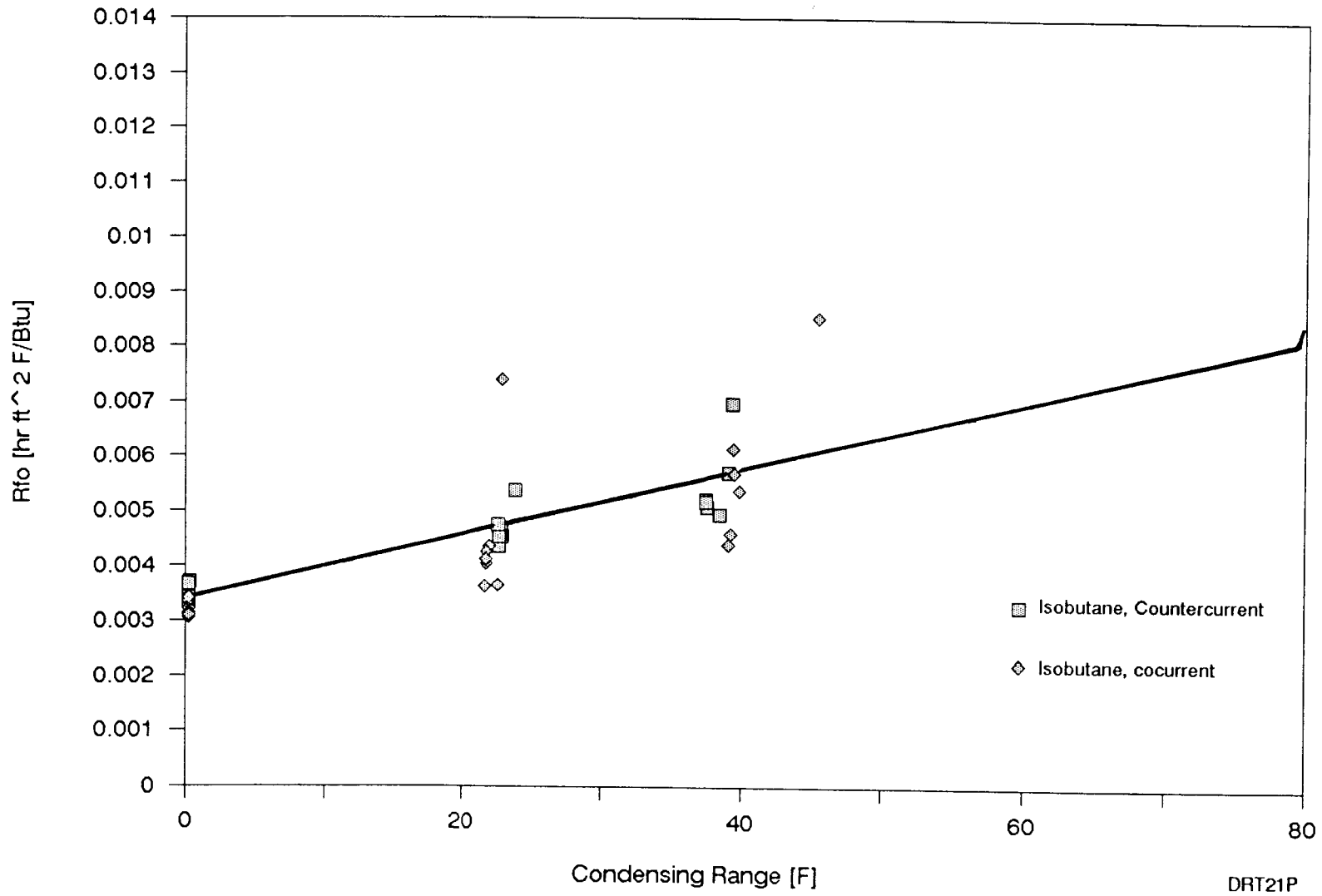


Figure 16. Comparison of equivalent diameter method calculation to experiment thermal resistance (half tubes plugged).

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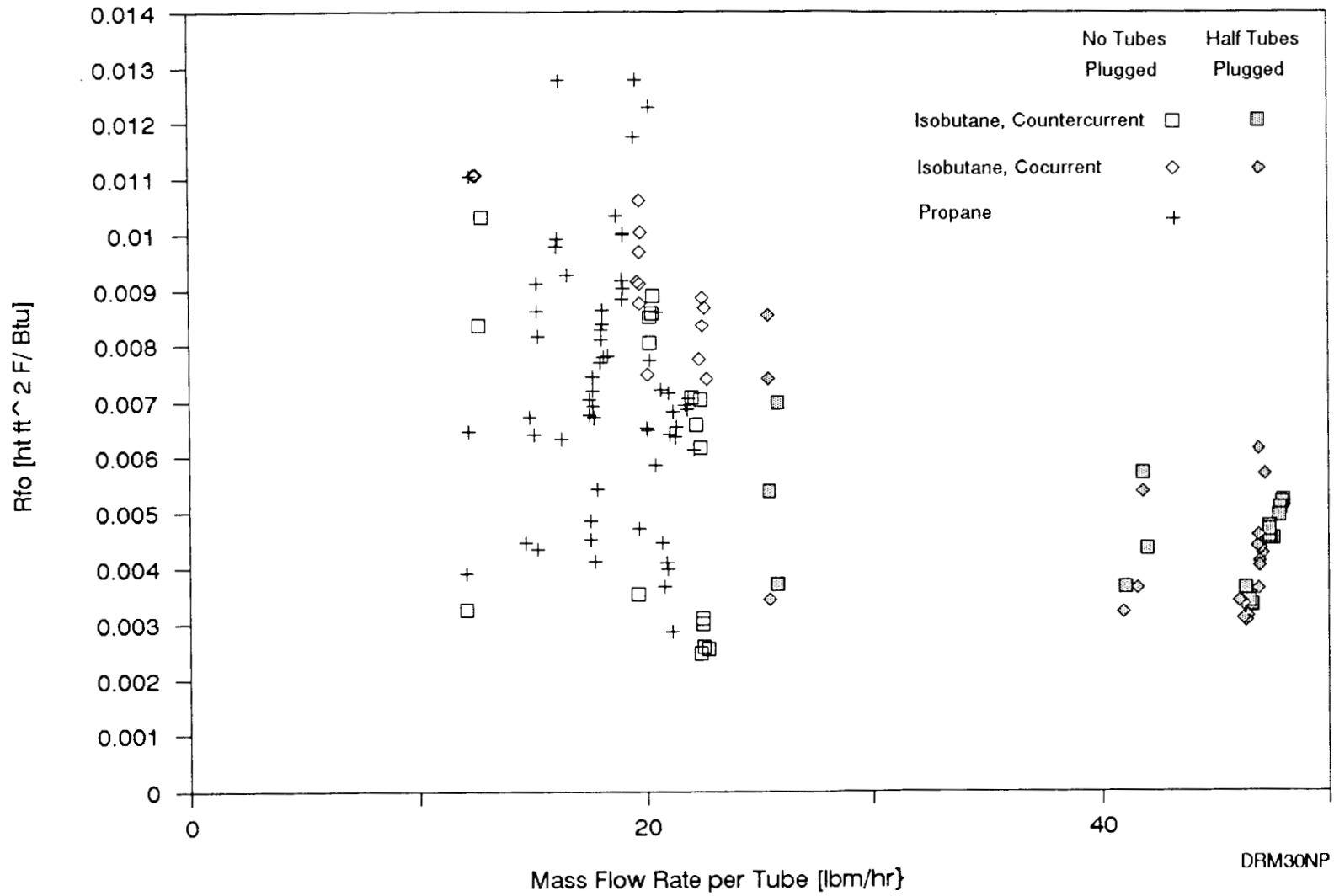


Figure 17. Comparison of equivalent diameter method calculation to experiment thermal resistance dependence on mass flow rate.

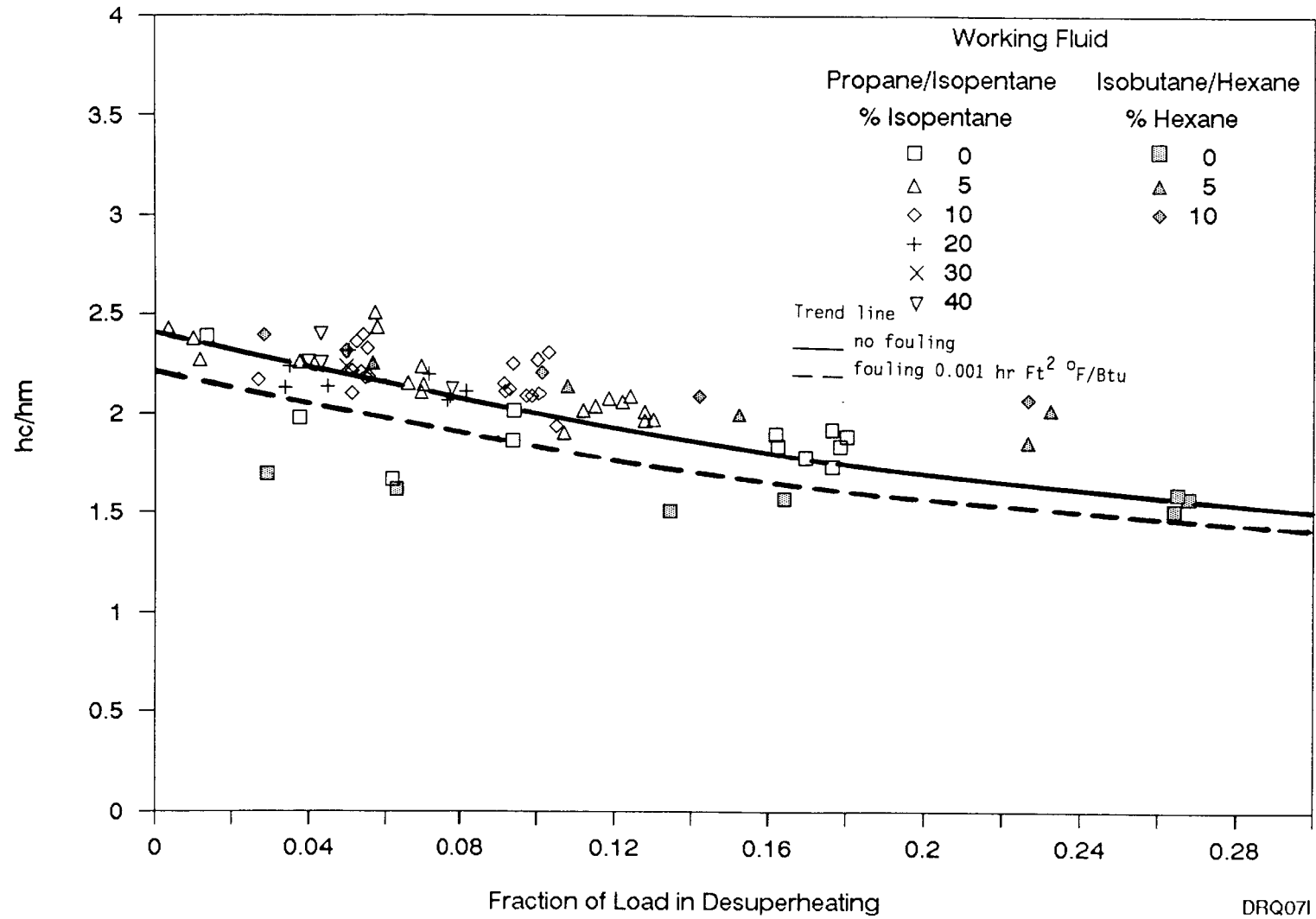


Figure 18. Comparison of equivalent diameter method calculation to experiment convection coefficient ratio.

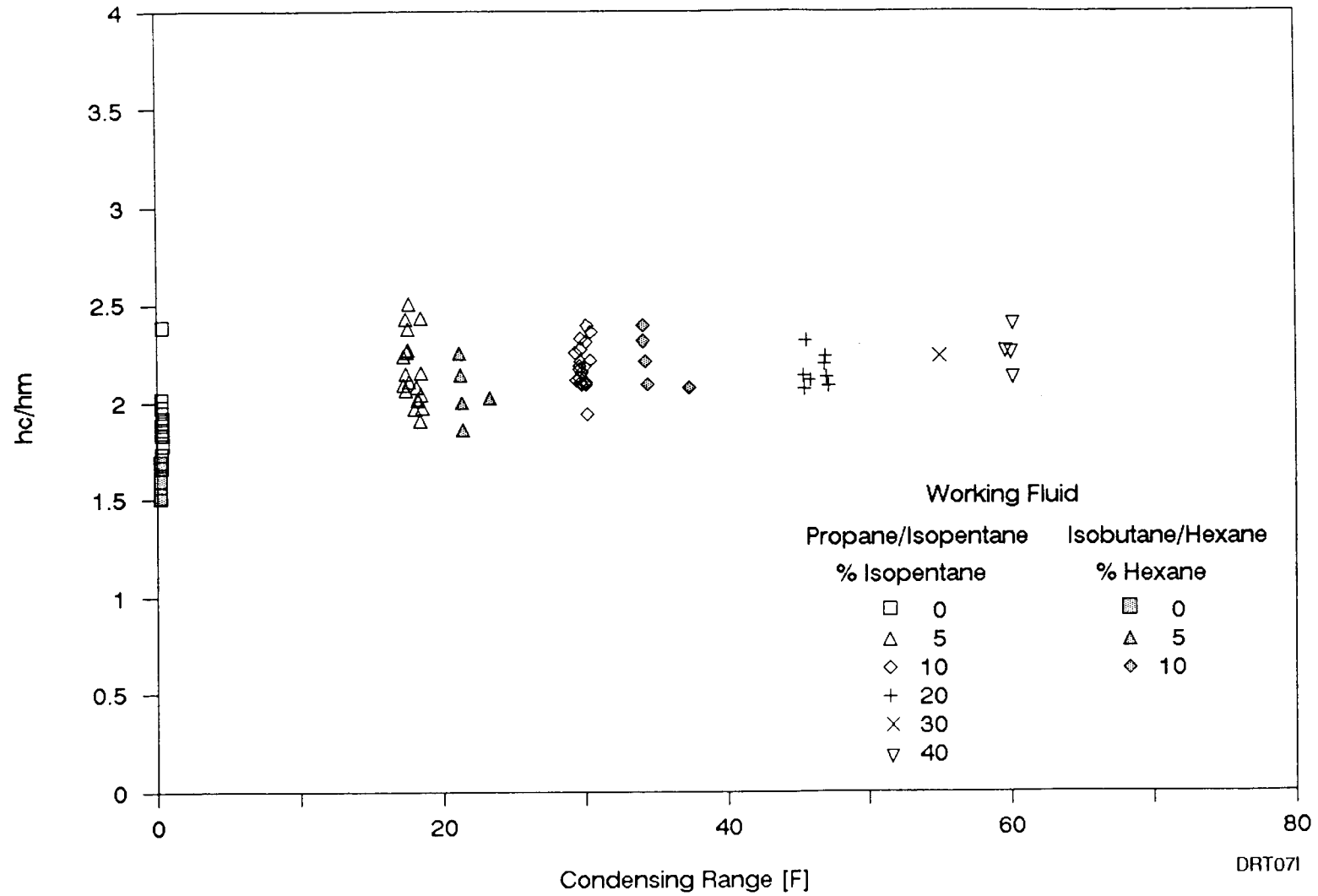


Figure 19. Comparison of equivalent diameter method calculation to experiment convection coefficient ratio.

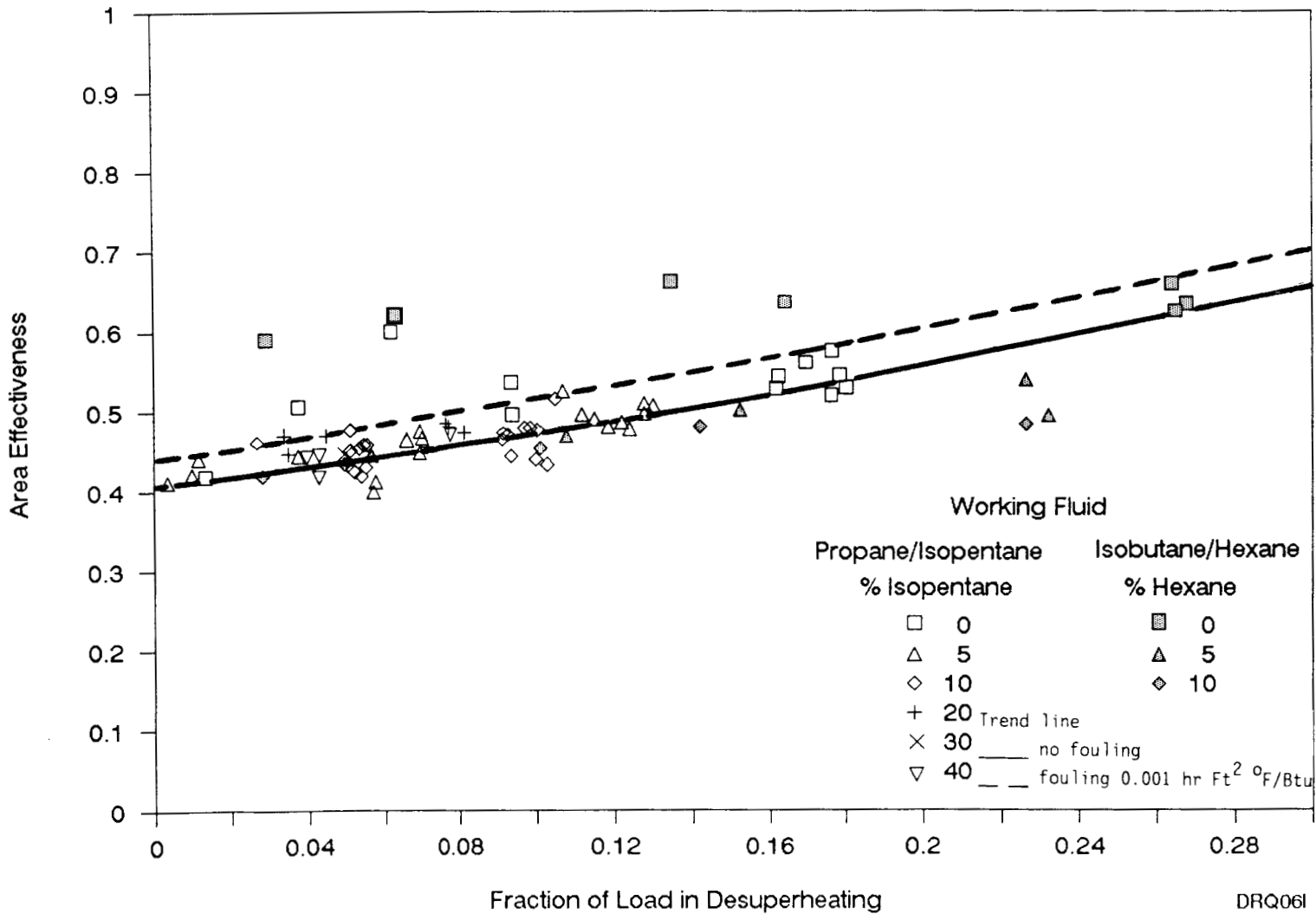


Figure 20. Comparison of equivalent diameter method calculation to experiment area effectiveness.

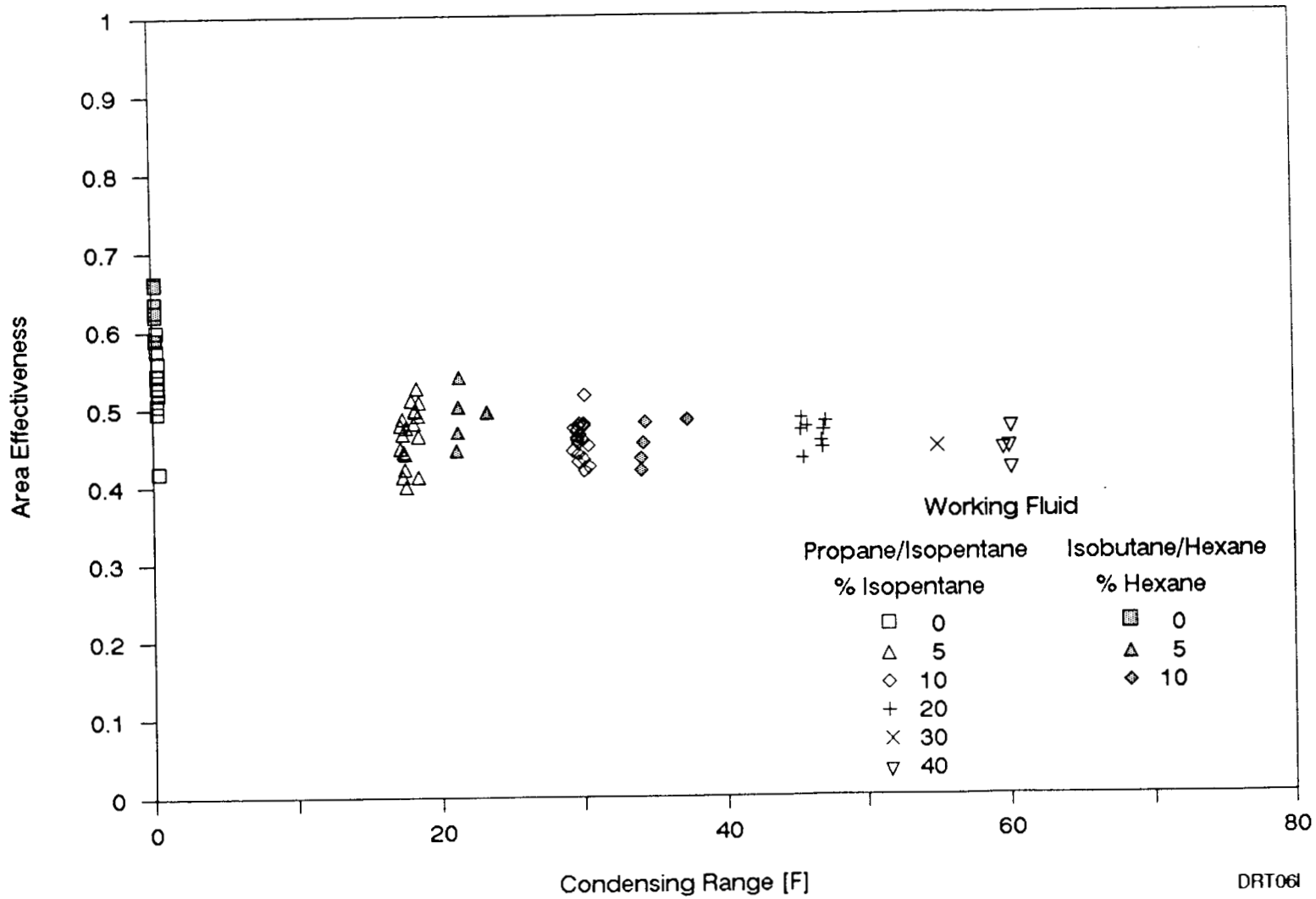


Figure 21. Comparison of equivalent diameter method calculation to experiment area effectiveness.

the heat transfer coefficient is calculated correctly. As with the overall coefficient, no trend is evident with condensing range. The curve fit of the data is similar to that in Figure 13. The dashed line indicates the trend with a fouling of $0.001 \text{ hr ft}^2 \text{ }^\circ\text{F} / \text{Btu}$.

Figure 22 is a plot similar to one in Reference 1. This shows the change in outlet temperature (bubble point temperature) necessary to make the calculated and experimental data coincide. A recalculation of the vertical data can be seen in Figure B-7. In the vertical case, the spread of the results was between plus and minus $2 \text{ }^\circ\text{F}$. For the horizontal data the deviations are much greater and there is a trend to higher values at mixtures with larger condensing ranges. Little generalization can be made from this plot.

Figures 23 and 24 compare the results for this computational model for the different flow directions and individual tube loading (plugged/not plugged tubes). Both in the case of the overall heat transfer coefficient and the condensing (inside) coefficient, the trends are the same. With no tubes plugged, lower flow per tube, the countercurrent flow data deviated more than the cocurrent data with the pure isobutane showing the least deviation. For the plugged tube data, both flow configurations and all three mixtures appeared to follow the same trend,

The results of investigating the equivalent diameter model may be summarized as follows:

1. This model overpredicts the heat transfer coefficient associated with the desuperheating and condensing the working fluids by 50 to 100%. This is a much larger discrepancy than can be explained by changes in system fouling.
2. There is a trend toward larger disagreement between calculated and experimental results when expressed as a thermal resistance which is nearly linear with the condensing range of the working fluid. This trend is also evident in the data which compares the vertical

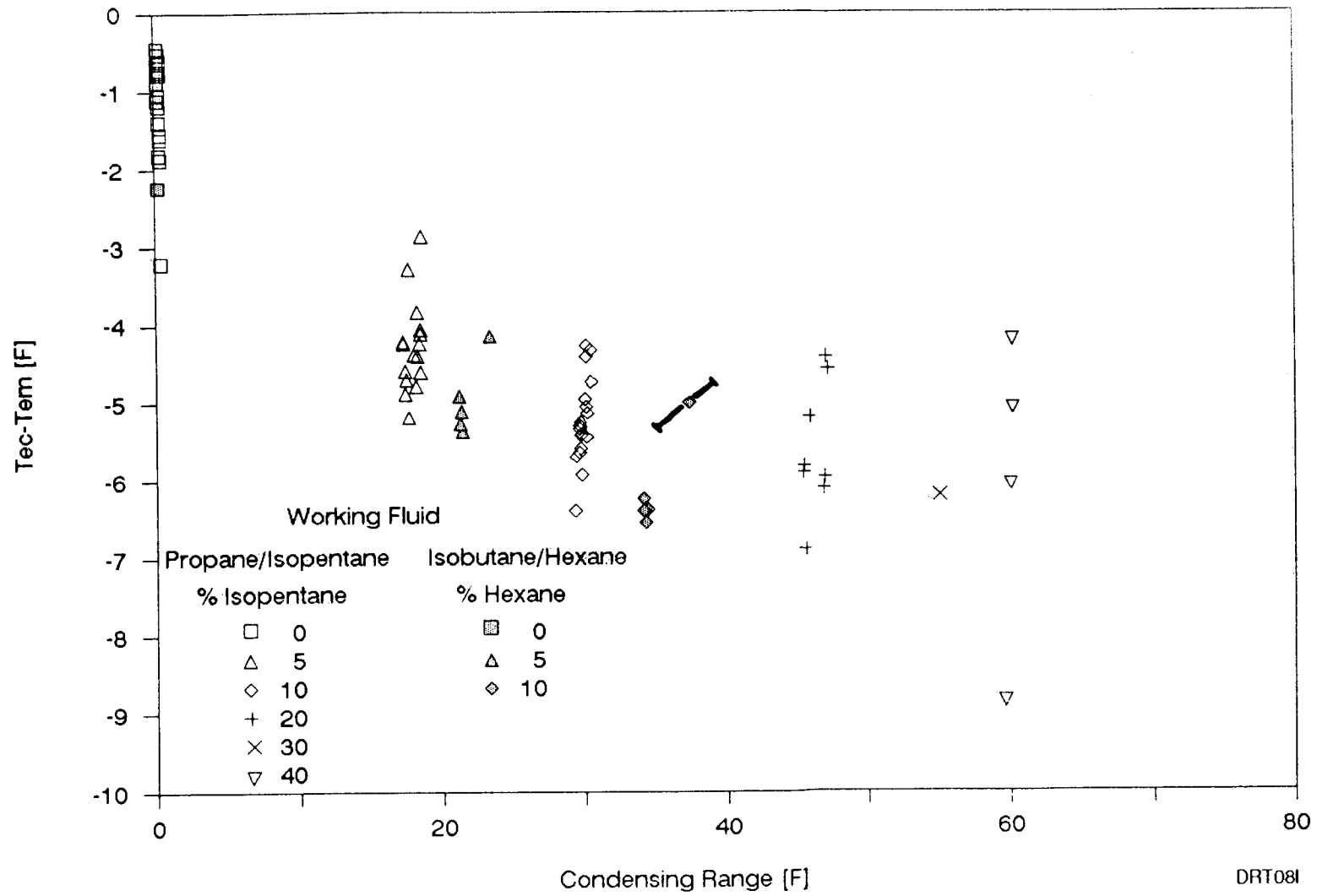
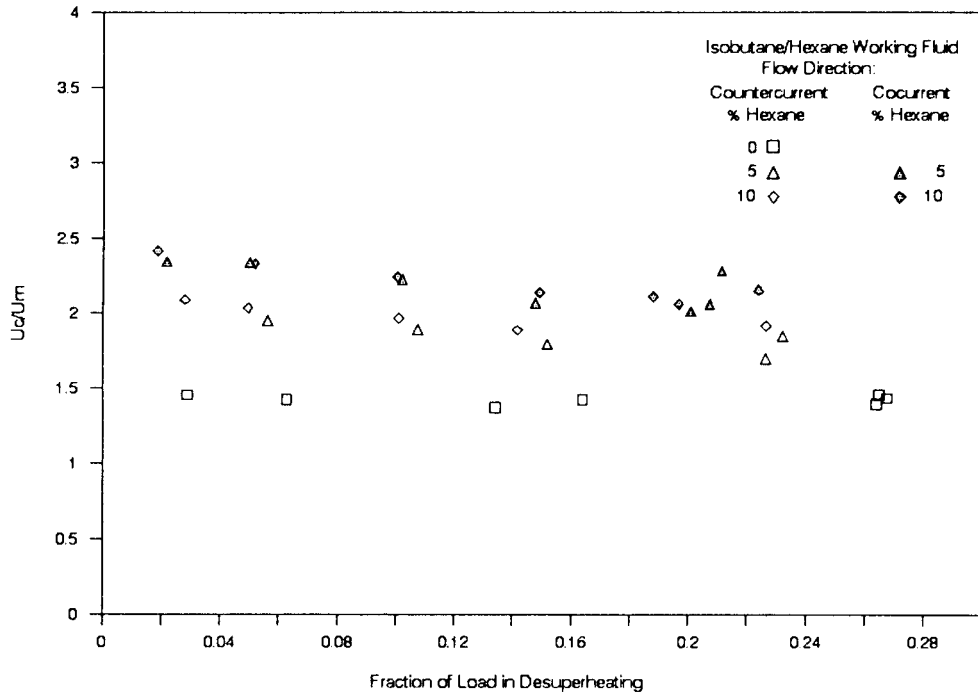
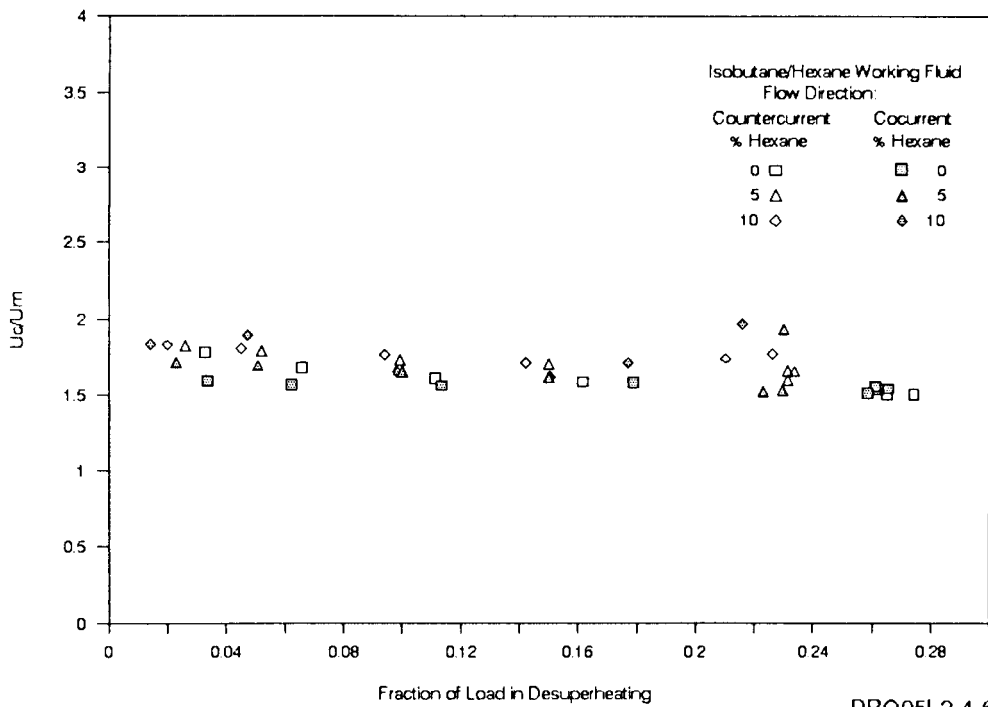


Figure 22. Comparison of equivalent diameter method calculation to experiment outlet temperature.



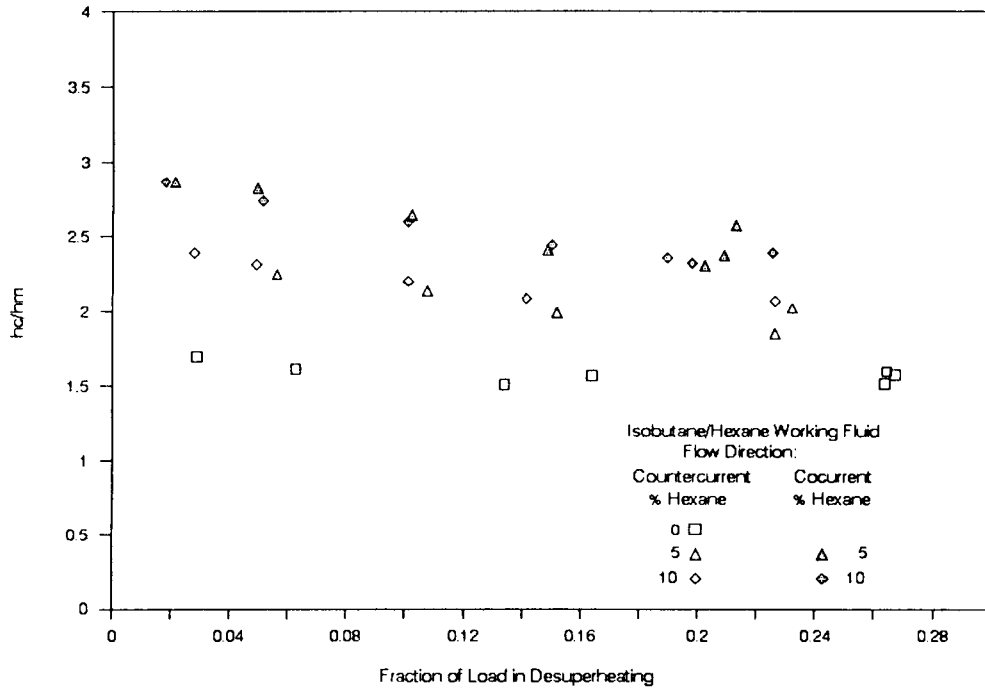
a) No Tubes Plugged



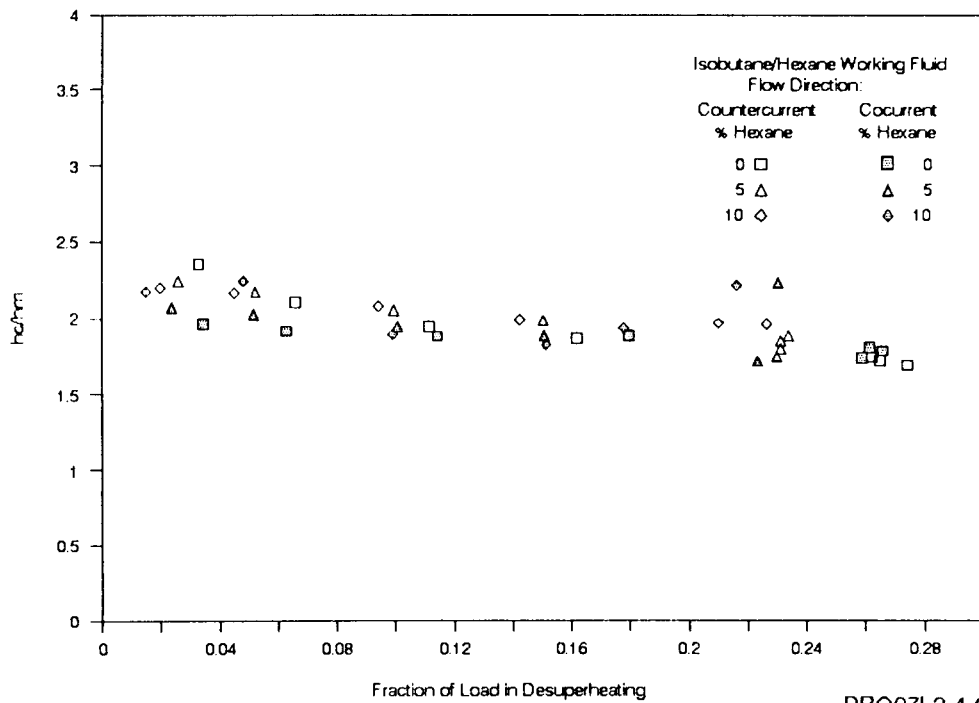
b) Half Tubes Plugged

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Figure 23. Comparison of equivalent diameter method calculation to experiment cocurrent/countercurrent plugged/not plugged.



a) No Tubes Plugged



b) Half Tubes Plugged

DRQ071-2-4-6

Figure 24. Comparison of equivalent diameter method calculation to experiment cocurrent/counter-current plugged/not plugged.

calculation to the vertical data. The slope is similar for both comparisons. This indicates that the computer program model for calculating mixture condensation is not correctly handling all of the mixture effects. It may be necessary to use a more complete mass transfer/heat transfer model even for these mixtures for which the approximate method (Silver/Bell-Ghaly) has been thought to give good agreement.

3. For pure fluids the difference between the calculated and experimental results (expressed as a thermal resistance) is much higher than that for the similar comparison for the vertical orientation. This indicates that there must be a detrimental effect of the internal fins over a plain internal surface in the horizontal orientation. It is felt that this is a result of the fins blocking the normal draining mode of film flow in horizontal plain tubes. In the vertical orientation, this is not a problem because the fins offer additional vertical surface to thin the film.

Plain Tube Model - For the second computational model, the results have generally the same trends as for the equivalent diameter model. Figures 25 through 30 show similar results for the plain tube model to Figures 13 through 24 where the equivalent diameter model was used. There is an apparent discrepancy among some of these results. The overall heat transfer coefficient results appear to give close agreement between the model and the experimental results, while the convective heat transfer (condensing) results show as much disagreement as do the equivalent diameter comparisons. The reason for this is that the area enhancement factor resulting from the internal fins is not accounted for in the overall heat transfer calculations, but is in the inside convection coefficient data reduction. That is, good agreement is obtained if the fins are not accounted for. When this model was used with the vertical orientation data, the agreement was similar to the equivalent diameter model.

In summary, this method appears to offer no advantages in a data comparison.

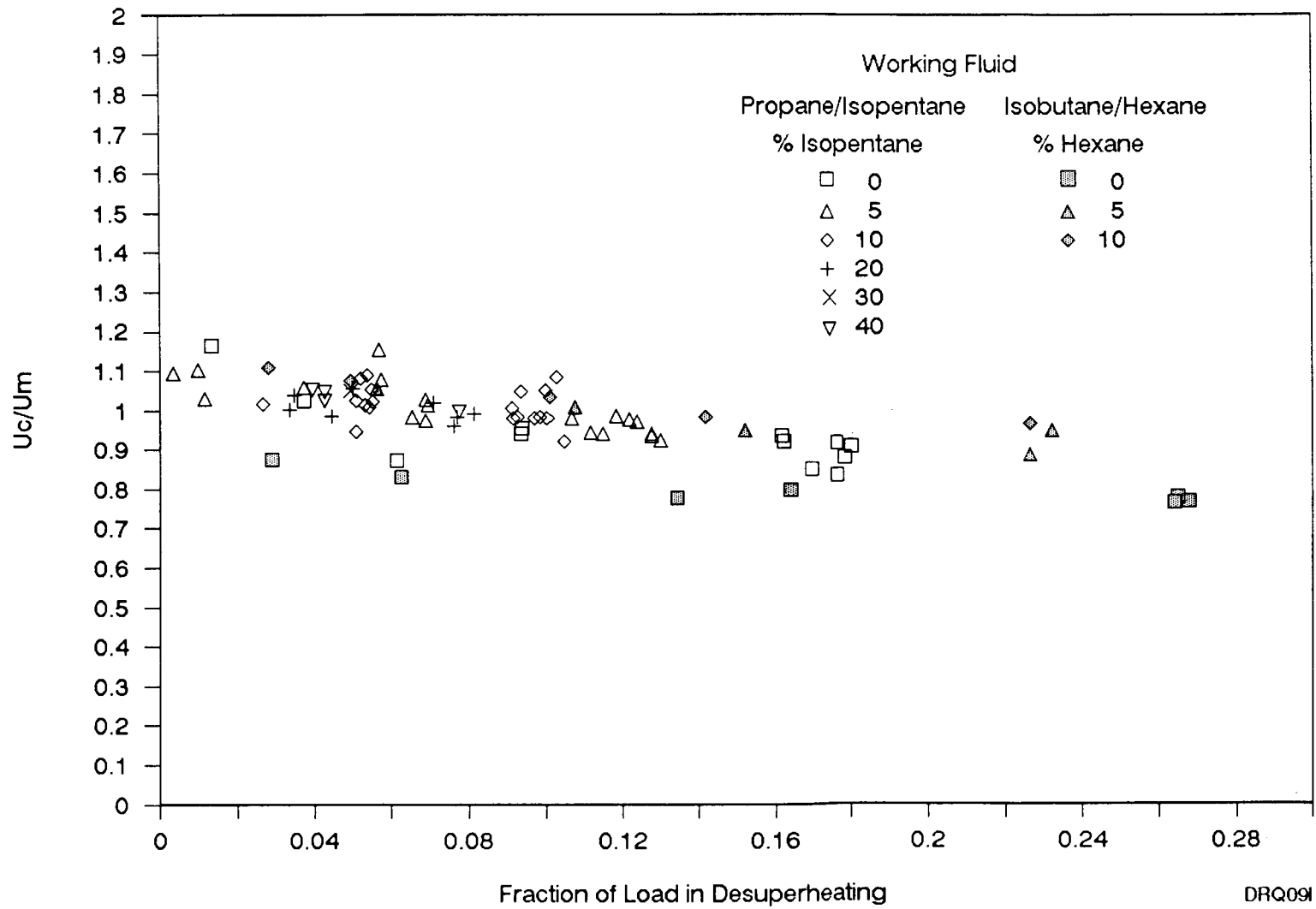


Figure 25. Comparison of equivalent diameter method calculation to experiment overall heat transfer coefficient.

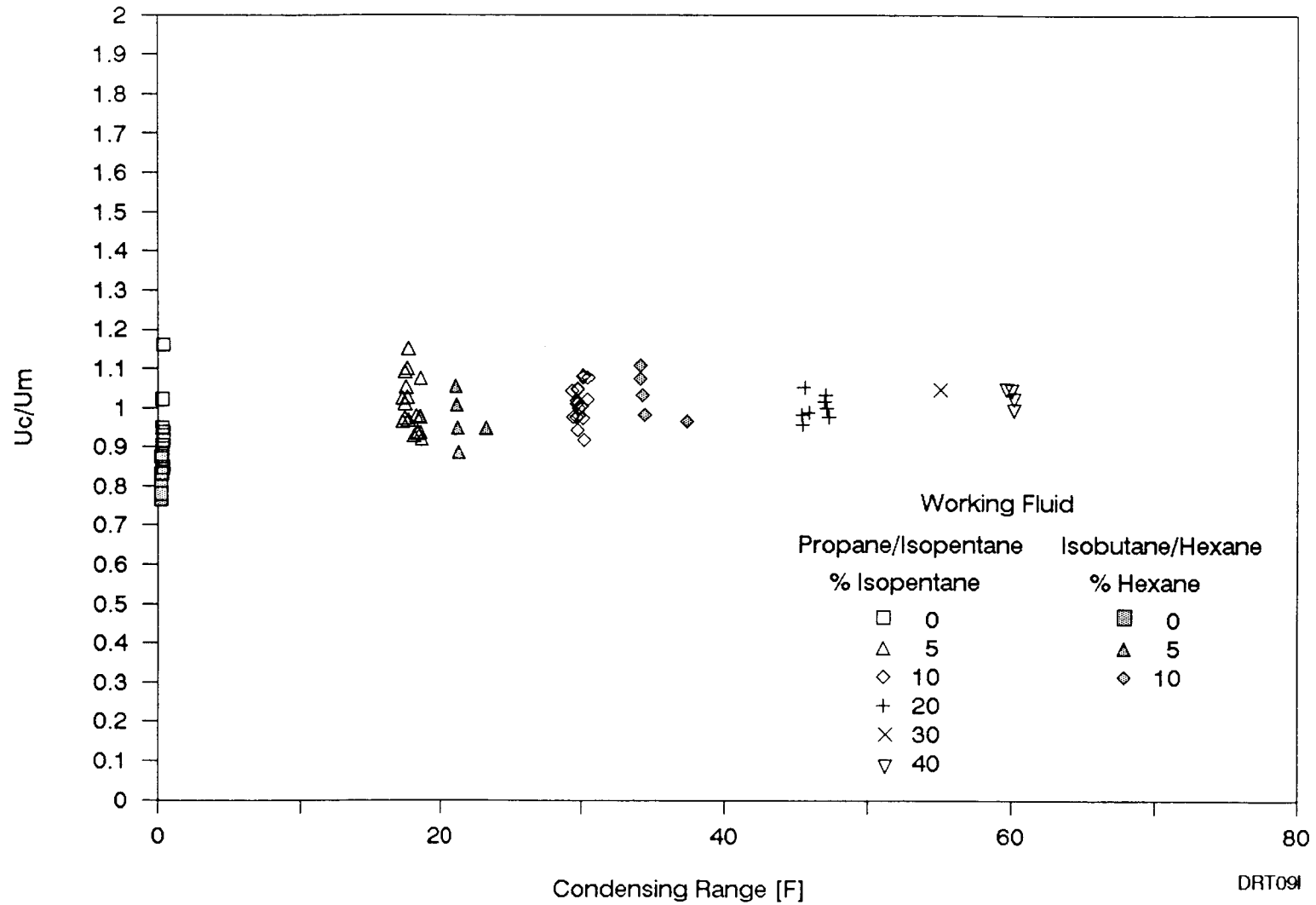


Figure 26. Comparison of plain tube method calculation to experiment overall heat transfer coefficient.

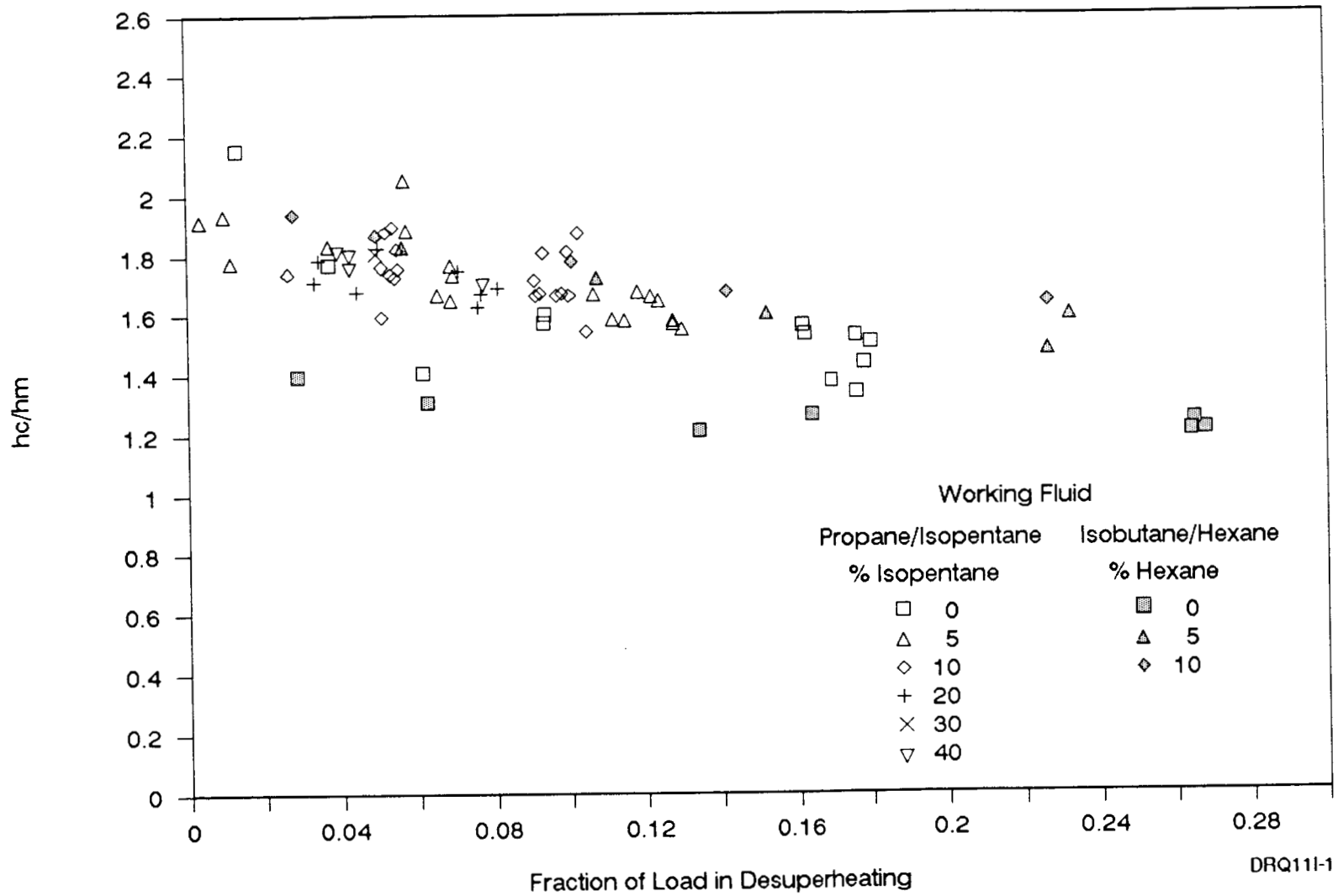


Figure 27. Comparison of plain tube method calculation to experiment convective heat transfer coefficient (inside).

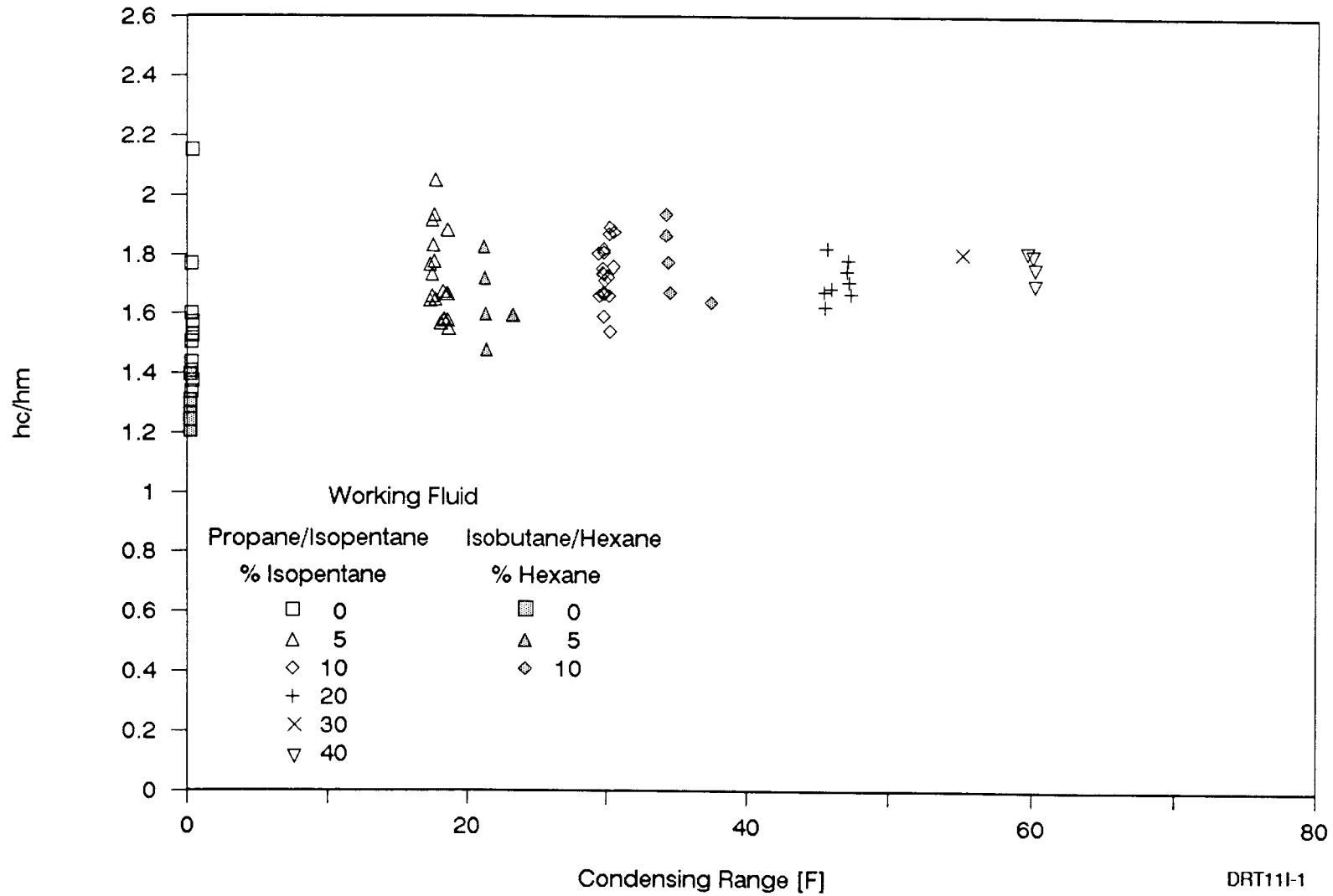
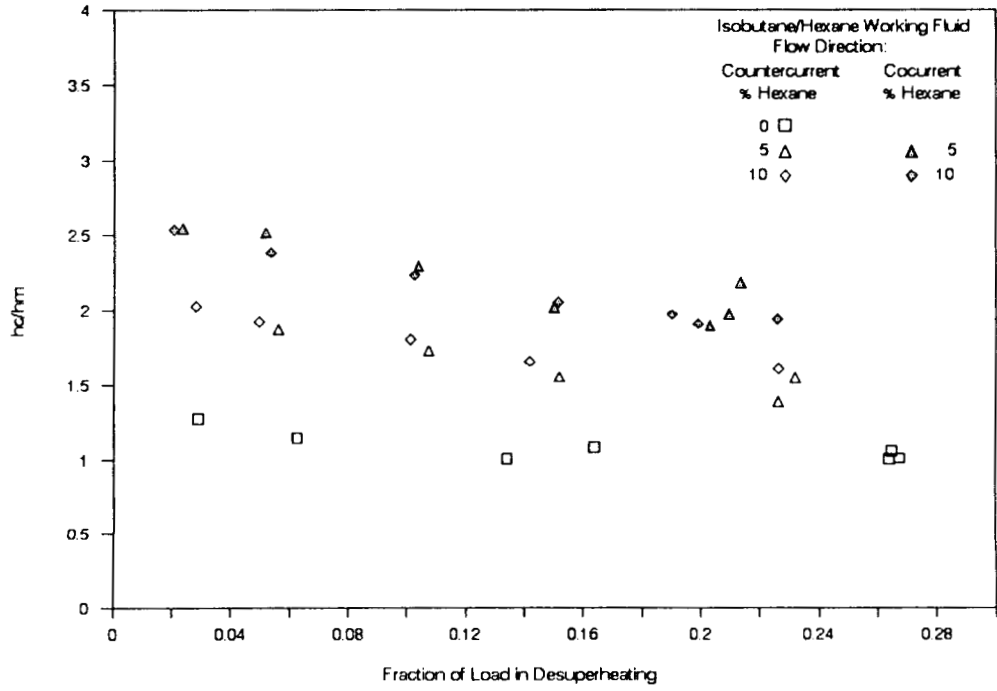
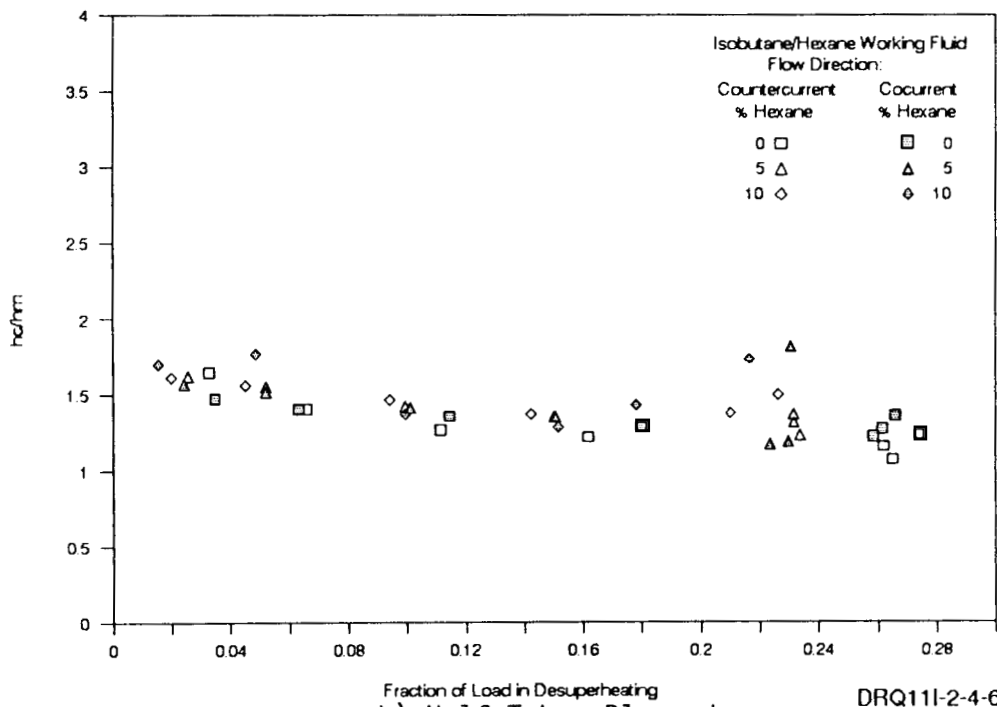


Figure 28. Comparison of plain tube method calculation to experiment convective heat transfer coefficient (inside).

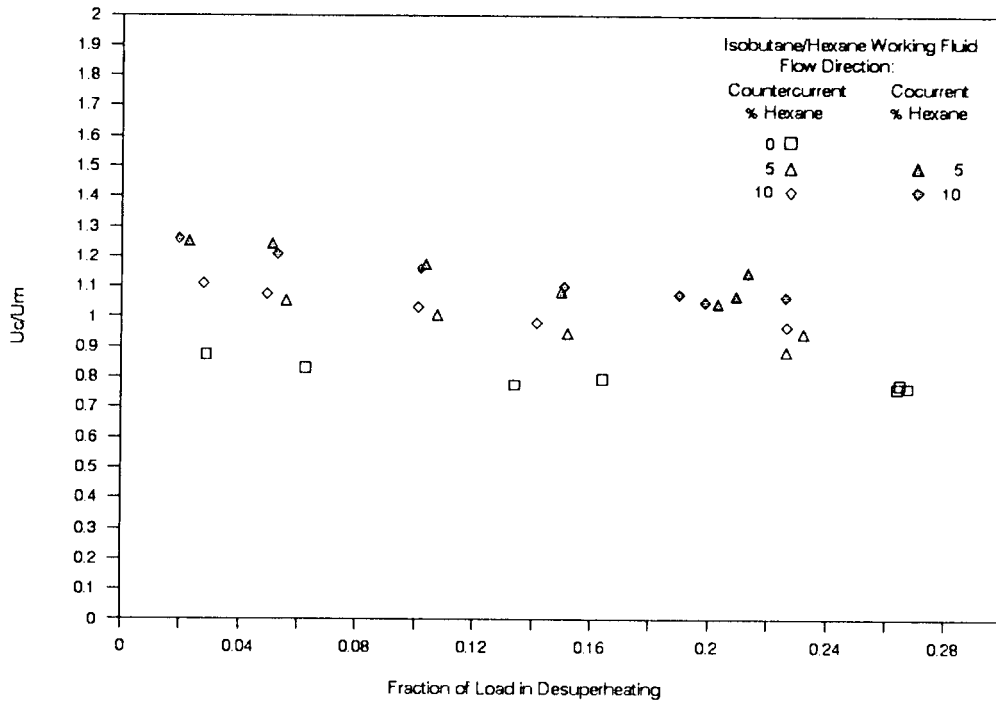


a) No Tubes Plugged

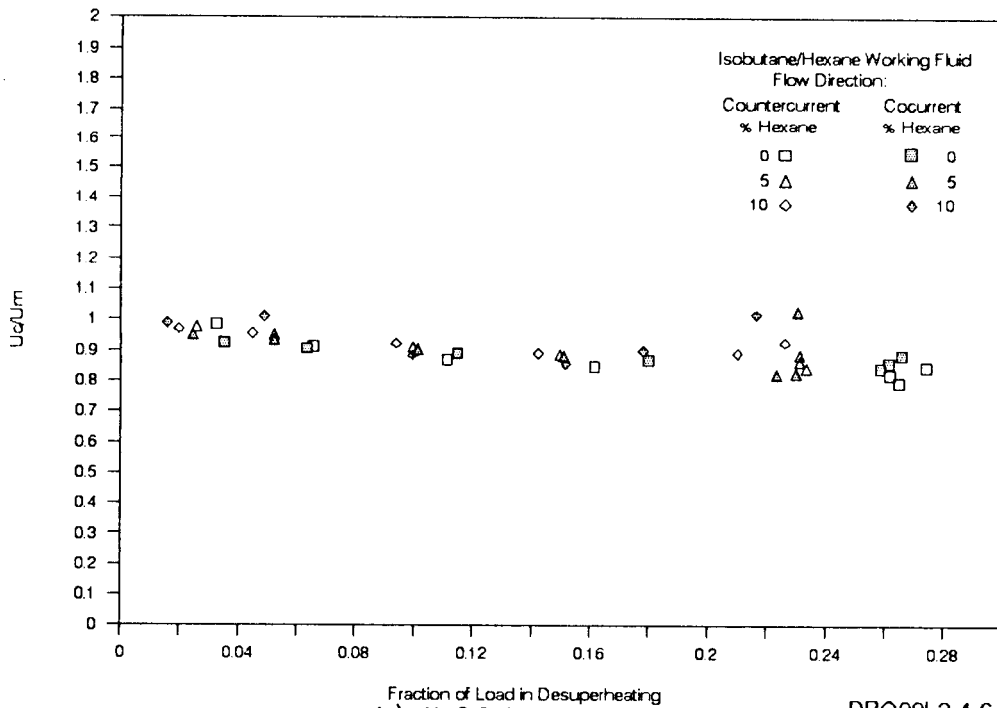


b) Half Tubes Plugged DRQ111-2-4-6

Figure 29. Comparison of plain tube method calculation to experiment cocurrent/counter-current plugged/not plugged.



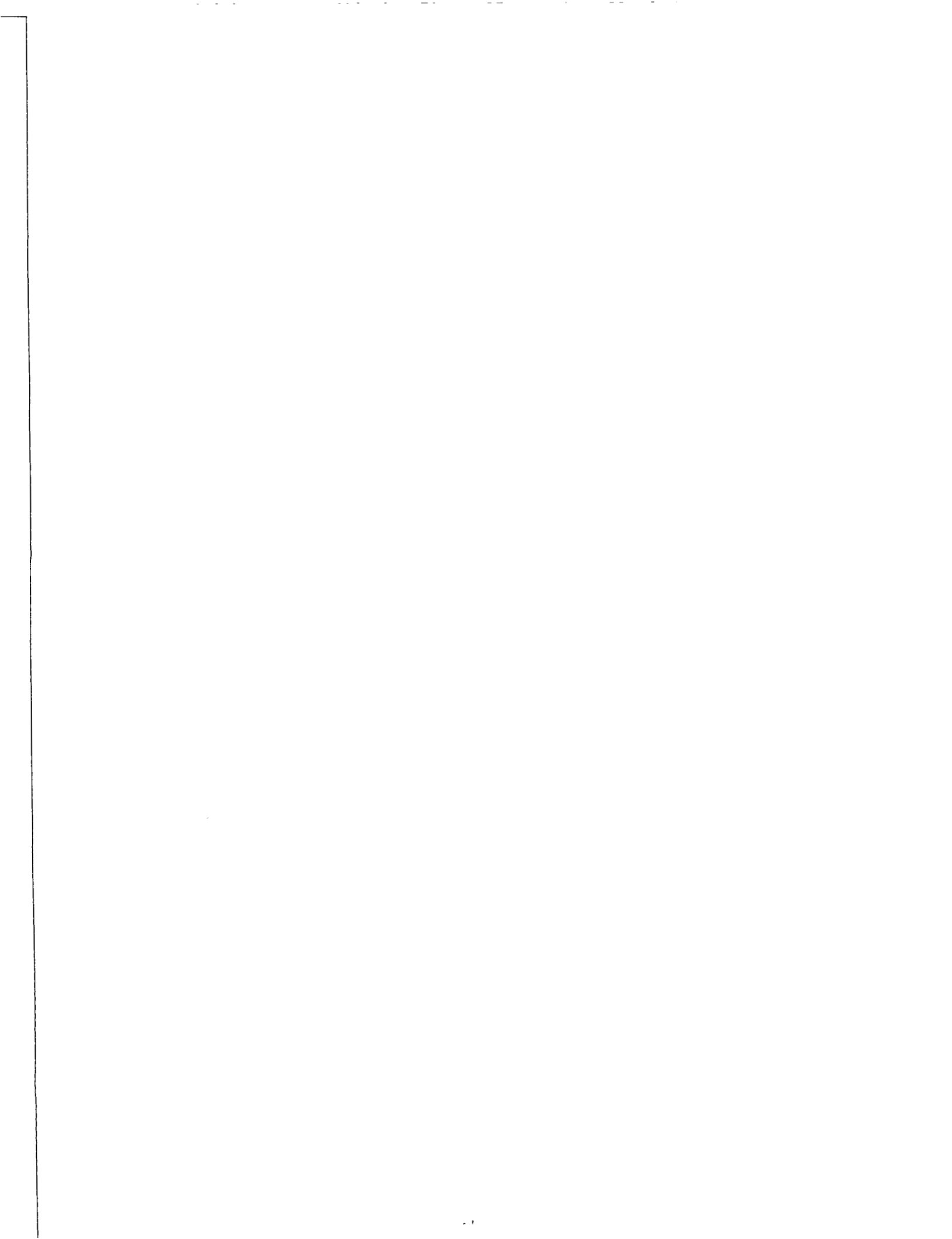
a) No Tubes Plugged



b) Half Tubes Plugged

DRQ091-2-4-6

Figure 30. Comparison of plain tube method calculation to experiment cocurrent/counter-current plugged/not plugged.



CONCLUSIONS AND RECOMMENDATIONS

The results and conclusions of this work can be summarized as follows:

1. There is no evidence that the condensation in the nearly horizontal condenser deviated from integral condensation. This means that there would be no thermodynamic penalty associated with orienting the condenser in a nearly horizontal position.
2. The heat transfer performance of the internally finned condenser in the nearly horizontal orientation is 33 to 47% worse than the same condenser in the vertical orientation. This means that a condenser in the vertical orientation could be 33 to 47% smaller than one in the horizontal orientation to perform the same duty, that is, produce the same turbine back pressure with a given cooling water temperature and flow.
3. The method of predicting the performance of a condenser in the nearly horizontal orientation with internally finned tubes is not well established. The two models developed here do not give good results in their comparison. They both greatly overpredict the observed performance of the heat exchanger. The design of this type of condenser in this orientation can now be approximated but the uncertainties are beyond the practical limits desired. It is expected that these uncertainties would not change the conclusions above.
4. Based on results to date, internal filled tubes would not be recommended for near horizontal applications because of the apparent inefficiency of the fins.

The following additional actions are recommended to meet the goals of being able to put this type of system into practice:

1. At the end of the program, return the condenser to its vertical orientation and repeat some of the original tests. This will allow the amount of fouling during the testing period to be estimated with

greater certainty and will allow the removal of some of the uncertainty from the conclusions presented in this report.

2. Develop a simple computer program which will allow exploration of the condensing process in detail. It is felt that the penalty of the fins in the horizontal orientation is only where gravity-controlled condensation takes place. It is impossible to analytically examine this hypothesis with the HTRI computer program. The question also arises concerning the method used to design with this type of system. (Is a combined mass transfer/heat transfer model needed?) This could be explored with the data already taken analytically if a computer program was available.

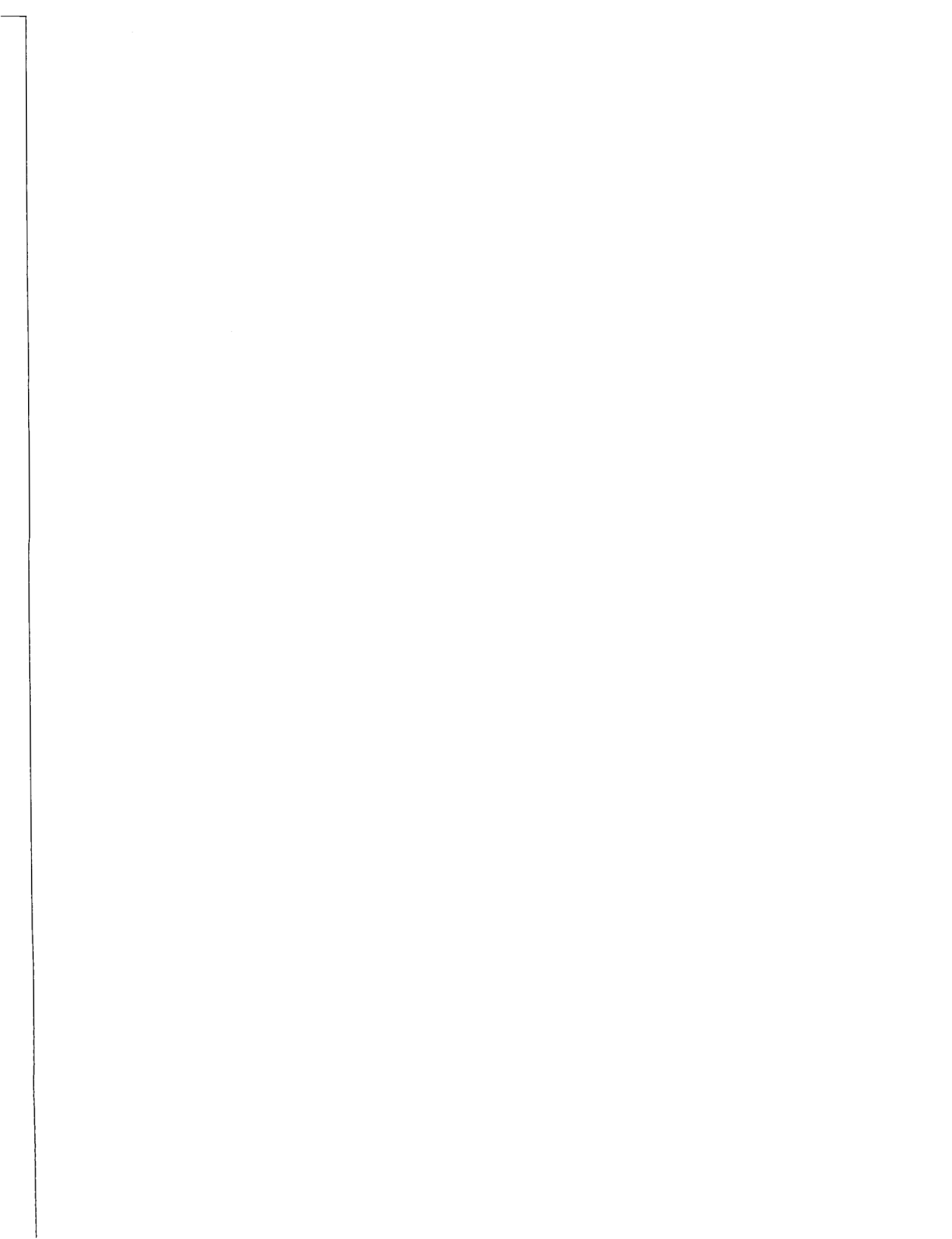
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APPENDIX A

EXPERIMENTS CONDUCTED AND SELECTED DATA



The following describes the parameters tested during the phase of testing being reported. Specific test conditions are not given for each of the individual tests; rather the different parameters that were varied are identified. Unless otherwise stated, the conditions listed are for the working fluid heater outlet vapor stream or the working fluid condenser vapor inlet stream. Those conditions which were limited by the temperature of the geothermal resource are so indicated with and "*".

COUNTERCURRENT CONDENSER TESTS

Isobutane Working Fluid Family:

Fluids: isobutane
 95% isobutane, 5% hexane
 90% isobutane, 10% hexane

Cooling Water

Flow Rate: 50000 lb/hr
 67000 lb/hr
 75000 lb/hr

For each of the working fluids, tests were conducted at three cooling water flow rates; 50000, 67000, and 75000 lb/hr. The nominal flow rate for most of the tests was 67000 lb/hr. For selected conditions, tests were conducted with all the parameters held constant except the cooling water flow which was varied.

Working Fluid

Flow Rate: 3700-11000 lb/hr (isobutane)
 3700-9500* lb/hr (95% isobutane 5 % hexane)
 3700-7000* lb/hr (95% isobutane 10 % hexane)

For each of the working fluids, the working fluid flow rate was varied +/- 25% from the predicted flow rate throughout the HCRF turbine at that heater pressure. For selected test run data was also collected at 50, 75, and 100% of the specified flow rate. Data for both the

condenser and heater were taken during these tests. In addition, specific tests unique to the condenser were run with each fluid. These condenser tests were run from 6600 to 9800 lb/hr. Because of the brine temperature limitations at the time the particular test series were being conducted, it was not possible to obtain data on all of the desired conditions, particularly with the 90% isobutane, 10% hexane fluid.

Inlet

Superheat: 80°F*
60°F*
40°F*
20°F*
10°F*

The condenser test data taken in conjunction with the heater testing did not attempt to maintain the level of the superheat entering the condenser. The tests unique to the condenser however did control the level of superheat in the working fluid entering the condenser to the values indicated for each of the fluids tested (with the exception of the 90% isobutane, 10% hexane for which testing was limited due to the brine temperature).

COUNTERCURRENT CONDENSER TESTS

Propane Working Fluid Family:

Fluids: propane
95% propane, 5% isopentane
90% propane, 10% isopentane

Cooling Water

Flow Rate: 50000 lb/hr
67000 lb/hr
75000 lb/hr

For each of the working fluids, tests were conducted at three cooling water flow rates; 50000, 67000, and 75000 lb/hr. The nominal flow rate for most of the tests was 67000 lb/hr. For selected conditions, tests were conducted with all the parameters held constant except the cooling water flow which was varied.

Working Fluid

Flow Rate: 3700-11600 lb/hr (propane)
3700-11400 lb/hr (95% propane 5 % isopentane)
3700-11000 lb/hr (95% propane 10 % isopentane)

For each of the working fluids, the working fluid flow rate was varied +/- 25% from the predicted flow rate throughout the HCRF turbine at that heater pressure. For selected test run data was also collected at 50, 75, and 100% of the specified flow rate. Data for both the condenser and heater were taken during these tests. In addition, specific tests unique to the condenser were run with each fluid. These condenser tests were run from 6000 to 9000 lb/hr.

Inlet

Superheat: 40°F*
30°F*
20°F*
10°F*
5°F*

The condenser test data taken in conjunction with the heater testing did not attempt to maintain the level of the superheat entering the condenser. The tests unique to the condenser however did control the level of superheat in the working fluid entering the condenser to the values indicated for each of the fluids tested

Integral Condensation Tests:

Fluids: propane
90% propane, 10% isopentane
80% propane, 20% isopentane
75% propane, 25% isopentane
70% propane, 30% isopentane
65% propane, 35% isopentane
60% propane, 40% isopentane

Fluids used for this test series ranged in composition from a pure (technical grade) propane fluid to a mixture of 60% propane, 40% isopentane for the purpose of attempting to identify the deviation from the assumption of integral condensation in the condenser performance model. Testing to examine the performance of fluids with higher levels of isopentane was limited due to the low brine temperatures.

Working Fluid

Flow Rate: 6400 lb/hr
7700 lb/hr
9000 lb/hr

Performance data was collected for each of the fluids tested at working fluid flow rates of 6400 to 9000 lb/hr.

Cooling Water

Flow Rate: 50000 lb/hr
67000 lb/hr
75000 lb/hr

At a nominal working fluid flow rate (7700 lb/hr) for each fluid, the cooling water flow rate was varied from 50000 to 75000 lb/hr.

Inlet

Superheat: 30⁰F*
10⁰F*

The amount of superheat in the working fluid vapor entering the condenser was varied from 30 to 10⁰F at each of the working fluid and cooling water flow rates tested for all of the working fluids used in this test series.

The following tables identify the fluids tested (nominal chemistry) and the test conditions that were initially evaluated. Note the alphanumeric designation given to each of the test conditions; the alpha designation identifies the nominal chemistry of the fluid being tested. The second table (giving test conditions) lists the approximate values of the controlled parameters for the heater and condenser tests initially evaluated. The test data sheets and working fluid chemistry for each run have been compiled in a separate data report.

Table A-1. A series tests with no tubes plugged

CONDENSER ORIENTATION: 10 degrees off horizontal

FLUID: isobutane

TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
A200	550 psia	290 F	TBD	25 gpm	130 gpm	countercurrent
A201	" "	TBD	40 F	" "	" "	" "
A202	" "	TBD	20 F	" "	" "	" "
A203	550 psia	290 F	TBD	30 gpm	130 gpm	countercurrent
A204	" "	TBD	40 F	" "	" "	" "
A205	" "	TBD	20 F	" "	" "	" "
A206	550 psia	290 F	TBD	30 gpm	100 gpm	contercurrent
A207	" "	TBD	40 F	" "	" "	" "
A208	" "	TBD	20 F	" "	" "	" "
A209	550 psia	290 F	TBD	30 gpm	150 gpm	countercurrent
A210	" "	TBD	40 F	" "	" "	" "
A211	" "	TBD	20 F	" "	" "	" "
A212	550 psia	290 F	TBD	35 gpm	150 gpm	contercurrent
A213	" "	TBD	40 F	" "	" "	" "
A214	" "	TBD	20 F	" "	" "	" "

Table A-2. A series tests with plugged tubes

CONDENSER ORIENTATION: 10 degrees off horizontal

FLUID: isobutane

TUBES PLUGGED: approx. half

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
A240	550 psia	290 F	TBD	25 gpm	130 gpm	countercurrent
A241	" "	TBD	40 F	" "	" "	" "
A242	" "	TBD	20 F	" "	" "	" "
A243	550 psia	290 F	TBD	30 gpm	130 gpm	countercurrent
A244	" "	TBD	40 F	" "	" "	" "
A245	" "	TBD	20 F	" "	" "	" "
A246	550 psia	290 F	TBD	30 gpm	100 gpm	countercurrent
A247	" "	TBD	40 F	" "	" "	" "
A248	" "	TBD	20 F	" "	" "	" "
A249	550 psia	290 F	TBD	30 gpm	150 gpm	countercurrent
A250	" "	TBD	40 F	" "	" "	" "
A251	" "	TBD	20 F	" "	" "	" "
A252	550 psia	290 F	TBD	35 gpm	150 gpm	countercurrent
A253	" "	TBD	40 F	" "	" "	" "
A254	" "	TBD	20 F	" "	" "	" "
A260	550 psia	290 F	TBD	25 gpm	130 gpm	cocurrent
A261	" "	TBD	40 F	" "	" "	" "
A262	" "	TBD	20 F	" "	" "	" "
A263	550 psia	290 F	TBD	30 gpm	130 gpm	cocurrent
A264	" "	TBD	40 F	" "	" "	" "
A265	" "	TBD	20 F	" "	" "	" "
A266	550 psia	290 F	TBD	30 gpm	100 gpm	cocurrent
A267	" "	TBD	40 F	" "	" "	" "
A268	" "	TBD	20 F	" "	" "	" "
A269	550 psia	290 F	TBD	30 gpm	150 gpm	cocurrent
A270	" "	TBD	40 F	" "	" "	" "
A271	" "	TBD	20 F	" "	" "	" "
A272	550 psia	290 F	TBD	35 gpm	150 gpm	cocurrent
A273	" "	TBD	40 F	" "	" "	" "
A274	" "	TBD	20 F	" "	" "	" "

Table A-3. B series tests with no tubes plugged

CONDENSER ORIENTATION: 10 degrees off horizontal
 FLUID: 95% isobutane, 5% hexane
 TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
B200	550 psia	298 F	TBD	25 gpm	130 gpm	countercurrent
B201	" "	TBD	40 F	" "	" "	" "
B202	" "	TBD	20 F	" "	" "	" "
B203	550 psia	298 F	TBD	30 gpm	130 gpm	countercurrent
B204	" "	TBD	40 F	" "	" "	" "
B205	" "	TBD	20 F	" "	" "	" "
B206	550 psia	298 F	TBD	30 gpm	100 gpm	countercurrent
B207	" "	TBD	40 F	" "	" "	" "
B208	" "	TBD	20 F	" "	" "	" "
B209	550 psia	298 F	TBD	30 gpm	150 gpm	countercurrent
B210	" "	TBD	40 F	" "	" "	" "
B211	" "	TBD	20 F	" "	" "	" "
B212	550 psia	298 F	TBD	35 gpm	150 gpm	countercurrent
B213	" "	TBD	40 F	" "	" "	" "
B214	" "	TBD	20 F	" "	" "	" "
B220	550 psia	298 F	TBD	25 gpm	130 gpm	cocurrent
B221	" "	TBD	40 F	" "	" "	" "
B222	" "	TBD	20 F	" "	" "	" "
B223	550 psia	298 F	TBD	30 gpm	130 gpm	cocurrent
B224	" "	TBD	40 F	" "	" "	" "
B225	" "	TBD	20 F	" "	" "	" "
B226	550 psia	298 F	TBD	30 gpm	100 gpm	cocurrent
B227	" "	TBD	40 F	" "	" "	" "
B228	" "	TBD	20 F	" "	" "	" "
B229	550 psia	298 F	TBD	30 gpm	150 gpm	cocurrent
B230	" "	TBD	40 F	" "	" "	" "
B231	" "	TBD	20 F	" "	" "	" "
B232	550 psia	298 F	TBD	35 gpm	150 gpm	cocurrent
B233	" "	TBD	40 F	" "	" "	" "
B234	" "	TBD	20 F	" "	" "	" "

Table A-4. B series tests with plugged tubes

CONDENSER ORIENTATION: 10 degrees off horizontal
 FLUID: 95% isobutane, 5% hexane
 TUBES PLUGGED: approx. half

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
B240	550 psia	298 F	TBD	25 gpm	130 gpm	countercurrent
B241	" "	TBD	40 F	" "	" "	" "
B242	" "	TBD	20 F	" "	" "	" "
B243	550 psia	298 F	TBD	30 gpm	130 gpm	countercurrent
B244	" "	TBD	40 F	" "	" "	" "
B245	" "	TBD	20 F	" "	" "	" "
B246	550 psia	298 F	TBD	30 gpm	100 gpm	countercurrent
B247	" "	TBD	40 F	" "	" "	" "
B248	" "	TBD	20 F	" "	" "	" "
B249	550 psia	298 F	TBD	30 gpm	150 gpm	countercurrent
B250	" "	TBD	40 F	" "	" "	" "
B251	" "	TBD	20 F	" "	" "	" "
B252	550 psia	298 F	TBD	35 gpm	150 gpm	countercurrent
B253	" "	TBD	40 F	" "	" "	" "
B254	" "	TBD	20 F	" "	" "	" "
B260	550 psia	298 F	TBD	25 gpm	130 gpm	cocurrent
B261	" "	TBD	40 F	" "	" "	" "
B262	" "	TBD	20 F	" "	" "	" "
B263	550 psia	298 F	TBD	30 gpm	130 gpm	cocurrent
B264	" "	TBD	40 F	" "	" "	" "
B265	" "	TBD	20 F	" "	" "	" "
B266	550 psia	298 F	TBD	30 gpm	100 gpm	cocurrent
B267	" "	TBD	40 F	" "	" "	" "
B268	" "	TBD	20 F	" "	" "	" "
B269	550 psia	298 F	TBD	30 gpm	150 gpm	cocurrent
B270	" "	TBD	40 F	" "	" "	" "
B271	" "	TBD	20 F	" "	" "	" "
B272	550 psia	298 F	TBD	35 gpm	150 gpm	cocurrent
B273	" "	TBD	40 F	" "	" "	" "
B274	" "	TBD	20 F	" "	" "	" "

Table A-5. C series tests with no tubes plugged

CONDENSER ORIENTATION: 10 degrees off horizontal
 FLUID: 90% isobutane, 10% hexane
 TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
C200	550 psia	308 F	TBD	25 gpm	130 gpm	countercurrent
C201	" "	TBD	40 F	" "	" "	" "
C202	" "	TBD	20 F	" "	" "	" "
C203	550 psia	308 F	TBD	30 gpm	130 gpm	countercurrent
C204	" "	TBD	40 F	" "	" "	" "
C205	" "	TBD	20 F	" "	" "	" "
C206	550 psia	308 F	TBD	30 gpm	100 gpm	countercurrent
C207	" "	TBD	40 F	" "	" "	" "
C208	" "	TBD	20 F	" "	" "	" "
C209	550 psia	308 F	TBD	30 gpm	150 gpm	countercurrent
C210	" "	TBD	40 F	" "	" "	" "
C211	" "	TBD	20 F	" "	" "	" "
C212	550 psia	308 F	TBD	35 gpm	150 gpm	contercurrent
C213	" "	TBD	40 F	" "	" "	" "
C214	" "	TBD	20 F	" "	" "	" "
C220	550 psia	308 F	TBD	25 gpm	130 gpm	cocurrent
C221	" "	TBD	40 F	" "	" "	" "
C222	" "	TBD	20 F	" "	" "	" "
C223	550 psia	308 F	TBD	30 gpm	130 gpm	cocurrent
C224	" "	TBD	40 F	" "	" "	" "
C225	" "	TBD	20 F	" "	" "	" "
C226	550 psia	308 F	TBD	30 gpm	100 gpm	cocurrent
C227	" "	TBD	40 F	" "	" "	" "
C228	" "	TBD	20 F	" "	" "	" "
C229	550 psia	308 F	TBD	30 gpm	150 gpm	cocurrent
C230	" "	TBD	40 F	" "	" "	" "
C231	" "	TBD	20 F	" "	" "	" "
C232	550 psia	308 F	TBD	35 gpm	150 gpm	cocurrent
C233	" "	TBD	40 F	" "	" "	" "
C234	" "	TBD	20 F	" "	" "	" "

Table A-6. C series tests with plugged tubes

CONDENSER ORIENTATION: 10 degrees off horizontal
 FLUID: 90% isobutane, 10% hexane
 TUBES PLUGGED: approx. half

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
C240	550 psia	308 F	TBD	25 gpm	130 gpm	countercurrent
C241	" "	TBD	40 F	" "	" "	" "
C242	" "	TBD	20 F	" "	" "	" "
C243	550 psia	308 F	TBD	30 gpm	130 gpm	countercurrent
C244	" "	TBD	40 F	" "	" "	" "
C245	" "	TBD	20 F	" "	" "	" "
C246	550 psia	308 F	TBD	30 gpm	100 gpm	countercurrent
C247	" "	TBD	40 F	" "	" "	" "
C248	" "	TBD	20 F	" "	" "	" "
C249	550 psia	308 F	TBD	30 gpm	150 gpm	countercurrent
C250	" "	TBD	40 F	" "	" "	" "
C251	" "	TBD	20 F	" "	" "	" "
C252	550 psia	308 F	TBD	35 gpm	150 gpm	countercurrent
C253	" "	TBD	40 F	" "	" "	" "
C254	" "	TBD	20 F	" "	" "	" "
C260	550 psia	308 F	TBD	25 gpm	130 gpm	cocurrent
C261	" "	TBD	40 F	" "	" "	" "
C262	" "	TBD	20 F	" "	" "	" "
C263	550 psia	308 F	TBD	30 gpm	130 gpm	cocurrent
C264	" "	TBD	40 F	" "	" "	" "
C265	" "	TBD	20 F	" "	" "	" "
C266	550 psia	308 F	TBD	30 gpm	100 gpm	cocurrent
C267	" "	TBD	40 F	" "	" "	" "
C268	" "	TBD	20 F	" "	" "	" "
C269	550 psia	308 F	TBD	30 gpm	150 gpm	cocurrent
C270	" "	TBD	40 F	" "	" "	" "
C271	" "	TBD	20 F	" "	" "	" "
C272	550 psia	308 F	TBD	35 gpm	150 gpm	cocurrent
C273	" "	TBD	40 F	" "	" "	" "
C274	" "	TBD	20 F	" "	" "	" "

Table A-7. D series tests

CONDENSER ORIENTATION: 10 degrees off horizontal

FLUID: propane

TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
D200	650 psia	230 F	TBD	20 gpm	132 gpm	countercurrent
D201	" "	" "	" "	27 gpm	" "	" "
D202	" "	" "	" "	34 gpm	" "	" "
D203	650 psia	230 F	TBD	" "	150 gpm	countercurrent
D204	" "	" "	" "	" "	100 gpm	" "
D205	" "	" "	" "	36 gpm	132 gpm	" "
D206	650 psia	TBD	30 F	" "	" "	countercurrent
D207	" "	" "	20 F	" "	" "	" "
D208	" "	" "	10 F	" "	" "	" "
D209	650 psia	TBD	5 F	" "	150 gpm	countercurrent
D210	650 psia	232 F	TBD	25 gpm	130 gpm	" "
D211	" "	TBD	30 F	" "	" "	" "
D212	" "	TBD	10 F	" "	" "	contercurrent
D213	650 psia	232 F	TBD	30 gpm	" "	" "
D214	" "	TBD	30 F	" "	" "	" "
D215	" "	TBD	10 F	" "	" "	countercurrent
D216	650 psia	232 F	TBD	" "	100 gpm	" "
D217	" "	TBD	30 F	" "	" "	" "
D218	" "	TBD	10 F	" "	" "	countercurrent
D219	650 psia	232 F	TBD	" "	150 gpm	" "
D220	" "	TBD	30 F	" "	" "	" "
D221	" "	TBD	10 F	" "	" "	countercurrent
D222	650 psia	232 F	TBD	" "	130 gpm	" "
D223	" "	TBD	30 F	35 gpm	" "	" "
D224	" "	TBD	10 F	" "	" "	" "

Table A-8. E series tests

CONDENSER ORIENTATION: 10 degrees off horizontal

FLUID: 95% propane, 5%isopentane

TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
E200	650 psia	241 F	TBD	20 gpm	132 gpm	countercurrent
E201	" "	" "	" "	27 gpm	" "	" "
E202	" "	" "	" "	34 gpm	" "	" "
E203	650 psia	241 F	TBD	" "	150 gpm	countercurrent
E204	" "	" "	" "	" "	100 gpm	" "
E205	" "	" "	" "	36 gpm	132 gpm	" "
E206	650 psia	TBD	30 F	" "	" "	countercurrent
E207	" "	" "	20 F	" "	" "	" "
E208	" "	" "	10 F	" "	" "	" "
E209	650 psia	TBD	5 F	" "	150 gpm	countercurrent
E210	650 psia	241 F	TBD	25 gpm	130 gpm	" "
E211	" "	TBD	30 F	" "	" "	" "
E212	" "	TBD	10 F	" "	" "	contercurrent
E213	650 psia	241 F	TBD	30 gpm	" "	" "
E214	" "	TBD	30 F	" "	" "	" "
E215	" "	TBD	10 F	" "	" "	countercurrent
E216	650 psia	241 F	TBD	" "	100 gpm	" "
E217	" "	TBD	30 F	" "	" "	" "
E218	" "	TBD	10 F	" "	" "	countercurrent
E219	650 psia	241 F	TBD	" "	150 gpm	" "
E220	" "	TBD	30 F	" "	" "	" "
E221	" "	TBD	10 F	" "	" "	countercurrent
E222	650 psia	241 F	TBD	" "	130 gpm	" "
E223	" "	TBD	30 F	35 gpm	" "	" "
E224	" "	TBD	10 F	" "	" "	" "

Table A-9. F series tests

CONDENSER ORIENTATION: 10 degrees off horizontal
 FLUID: 90% propane, 10% isopentane
 TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
F200	650 psia	250 F	TBD	20 gpm	132 gpm	countercurrent
F201	" "	" "	" "	27 gpm	" "	" "
F202	" "	" "	" "	34 gpm	" "	" "
F203	650 psia	250 F	TBD	" "	150 gpm	countercurrent
F204	" "	" "	" "	" "	100 gpm	" "
F205	" "	" "	" "	36 gpm	132 gpm	" "
F206	650 psia	TBD	30 F	" "	" "	countercurrent
F207	" "	" "	20 F	" "	" "	" "
F208	" "	" "	10 F	" "	" "	" "
F209	650 psia	TBD	5 F	" "	150 gpm	countercurrent
F210	650 psia	250 F	TBD	25 gpm	130 gpm	" "
F211	" "	TBD	30 F	" "	" "	" "
F212	" "	TBD	10 F	" "	" "	countercurrent
F213	650 psia	250 F	TBD	30 gpm	" "	" "
F214	" "	TBD	30 F	" "	" "	" "
F215	" "	TBD	10 F	" "	" "	countercurrent
F216	650 psia	250 F	TBD	" "	100 gpm	" "
F217	" "	TBD	30 F	" "	" "	" "
F218	" "	TBD	10 F	" "	" "	countercurrent
F219	650 psia	250 F	TBD	" "	150 gpm	" "
F220	" "	TBD	30 F	" "	" "	" "
F221	" "	TBD	10 F	" "	" "	countercurrent
F222	650 psia	250 F	TBD	" "	130 gpm	" "
F223	" "	TBD	30 F	35 gpm	" "	" "
F224	" "	TBD	10 F	" "	" "	" "

Table A-10. H series tests

CONDENSER ORIENTATION: 10 degrees off horizontal
 FLUID: 80% propane, 20% isopentane
 TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
H210	650 psia	266 F	TBD	25 gpm	130 gpm	" "
H211	" "	TBD	30 F	" "	" "	" "
H212	" "	TBD	10 F	" "	" "	countercurrent
H213	650 psia	266 F	TBD	30 gpm	" "	" "
H214	" "	TBD	30 F	" "	" "	" "
H215	" "	TBD	10 F	" "	" "	countercurrent
H216	650 psia	266 F	TBD	" "	100 gpm	" "
H217	" "	TBD	30 F	" "	" "	" "
H218	" "	TBD	10 F	" "	" "	countercurrent
H219	650 psia	266 F	TBD	" "	150 gpm	" "
H220	" "	TBD	30 F	" "	" "	" "
H221	" "	TBD	10 F	" "	" "	countercurrent
H222	650 psia	266 F	TBD	" "	130 gpm	" "
H223	" "	TBD	30 F	35 gpm	" "	" "
H224	" "	TBD	10 F	" "	" "	" "

Table A-11. I series tests

CONDENSER ORIENTATION: 10 degrees off horizontal
 FLUID: 75% propane, 25% isopentane
 TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
1210	650 psia	273 F	TBD	25 gpm	130 gpm	" "
1211	" "	TBD	30 F	" "	" "	" "
1212	" "	TBD	10 F	" "	" "	countercurrent
1213	650 psia	273 F	TBD	30 gpm	" "	" "
1214	" "	TBD	30 F	" "	" "	" "
1215	" "	TBD	10 F	" "	" "	countercurrent
1216	650 psia	273 F	TBD	" "	100 gpm	" "
1217	" "	TBD	30 F	" "	" "	" "
1218	" "	TBD	10 F	" "	" "	countercurrent
1219	650 psia	273 F	TBD	" "	150 gpm	" "
1220	" "	TBD	30 F	" "	" "	" "
1221	" "	TBD	10 F	" "	" "	countercurrent
1222	650 psia	273 F	TBD	" "	130 gpm	" "
1223	" "	TBD	30 F	35 gpm	" "	" "
1224	" "	TBD	10 F	" "	" "	" "

Table A-12. J series tests

CONDENSER ORIENTATION: 10 degrees off horizontal

FLUID: 70% propane, 30% isopentane

TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
J210	650 psia	280 F	TBD	25 gpm	130 gpm	" "
J211	" "	TBD	30 F	" "	" "	" "
J212	" "	TBD	10 F	" "	" "	countercurrent
J213	650 psia	280 F	TBD	30 gpm	" "	" "
J214	" "	TBD	30 F	" "	" "	" "
J215	" "	TBD	10 F	" "	" "	countercurrent
J216	650 psia	280 F	TBD	" "	100 gpm	" "
J217	" "	TBD	30 F	" "	" "	" "
J218	" "	TBD	10 F	" "	" "	countercurrent
J219	650 psia	280 F	TBD	" "	150 gpm	" "
J220	" "	TBD	30 F	" "	" "	" "
J221	" "	TBD	10 F	" "	" "	countercurrent
J222	650 psia	280 F	TBD	" "	130 gpm	" "
J223	" "	TBD	30 F	35 gpm	" "	" "
J224	" "	TBD	10 F	" "	" "	" "

Table A-14. K series tests

CONDENSER ORIENTATION: 10 degrees off horizontal
 FLUID: 65% propane, 35% isopentane
 TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
K210	650 psia	287 F	TBD	25 gpm	130 gpm	" "
K211	" "	TBD	30 F	" "	" "	" "
K212	" "	TBD	10 F	" "	" "	countercurrent
K213	650 psia	287 F	TBD	30 gpm	" "	" "
K214	" "	TBD	30 F	" "	" "	" "
K215	" "	TBD	10 F	" "	" "	countercurrent
K216	650 psia	287 F	TBD	" "	100 gpm	" "
K217	" "	TBD	30 F	" "	" "	" "
K218	" "	TBD	10 F	" "	" "	countercurrent
K219	650 psia	287 F	TBD	" "	150 gpm	" "
K220	" "	TBD	30 F	" "	" "	" "
K221	" "	TBD	10 F	" "	" "	countercurrent
K222	650 psia	287 F	TBD	" "	130 gpm	" "
K223	" "	TBD	30 F	35 gpm	" "	" "
K224	" "	TBD	10 F	" "	" "	" "

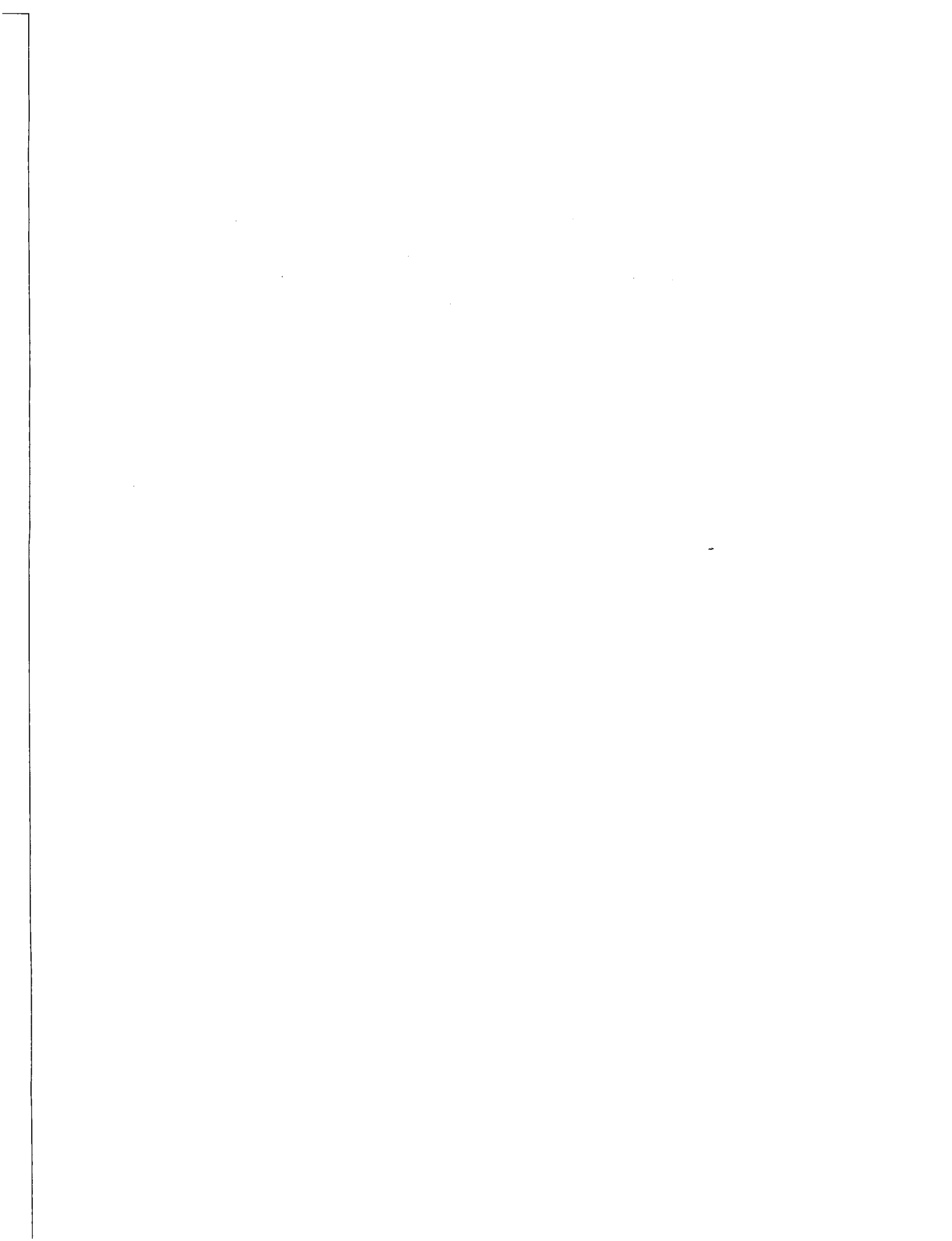
Table A-15. L series tests

CONDENSER ORIENTATION: 10 degrees off horizontal
 FLUID: 60% propane, 40% isopentane
 TUBES PLUGGED: none

TEST NO.	HEATER PRESSURE	HEATER OUTLET TEMP.	CONDENSER SUPERHEAT	WF FLOW RATE	CW FLOW RATE	CW FLOW DIRECTION
L210	650 psia	295 F	TBD	25 gpm	130 gpm	" "
L211	" "	TBD	30 F	" "	" "	" "
L212	" "	TBD	10 F	" "	" "	countercurrent
L213	650 psia	295 F	TBD	30 gpm	" "	" "
L214	" "	TBD	30 F	" "	" "	" "
L215	" "	TBD	10 F	" "	" "	countercurrent
L216	650 psia	295 F	TBD	" "	100 gpm	" "
L217	" "	TBD	30 F	" "	" "	" "
L218	" "	TBD	10 F	" "	" "	countercurrent
L219	650 psia	295 F	TBD	" "	150 gpm	" "
L220	" "	TBD	30 F	" "	" "	" "
L221	" "	TBD	10 F	" "	" "	countercurrent
L222	650 psia	295 F	TBD	" "	130 gpm	" "
L223	" "	TBD	30 F	35 gpm	" "	" "
L224	" "	TBD	10 F	" "	" "	" "

APPENDIX B

COMPARISON OF CALCULATIONAL METHODS FOR VERTICAL CONDENSER



The results of the component and cycle performance found with the supercritical cycle testing with the condenser in the vertical position are described in detail in a previous work [1] and in the section describing the analytical models. (Equivalent Diameter Model). The HTRI code using the original method predicted the condenser performance quite well. If this code was used for design in conjunction with the NBS property code and the method described previously, the resulting condenser would produce a condensing (or bubble point) temperature which would be within 1°F of that predicted by the code [1]. Because the version of the computer program CST had changed, it was felt that the data should be recalculated using the latest version, CST2 MOD 0.0-1.01.

The results of this recalculation are summarized in Figures B1 through B7. Figures B1 and B2 show the ratio of calculated to measured overall heat transfer coefficient. In Figure B1 it is plotted against the fraction of the heat duty in desuperheating. 84% of the data lies within 20% of unity (where the two values are equal), and 93% is within 30%. There is no trend with desuperheating fraction. The calculation assumed no fouling resistance. This assumption appears to be good because there is no bias to the results. Figure B2 shows the same data plotted against the condensing range. Here different mixture compositions are sorted out. Here there seem to be a trend of $U_c < U_m$ at low values of condensing range (nearer to pure fluids) and to $U_c > U_m$ for higher values.

The trend with working fluid is more evident if the difference between the measured and calculated coefficients is stated in terms of an incremental thermal resistance as was done in the text. Figure B3 is similar to a plot comparing the near horizontal data in the text. Here, if all of the data is considered, the effect of mixture composition is less than for the horizontal case. A least-squares fit of a straight line is shown by the solid line. (For a pure fluid, the difference is -0.00158 indicating that the calculated heat transfer coefficient is lower than the measured value, but with a 40 °F condensing range, slightly greater than

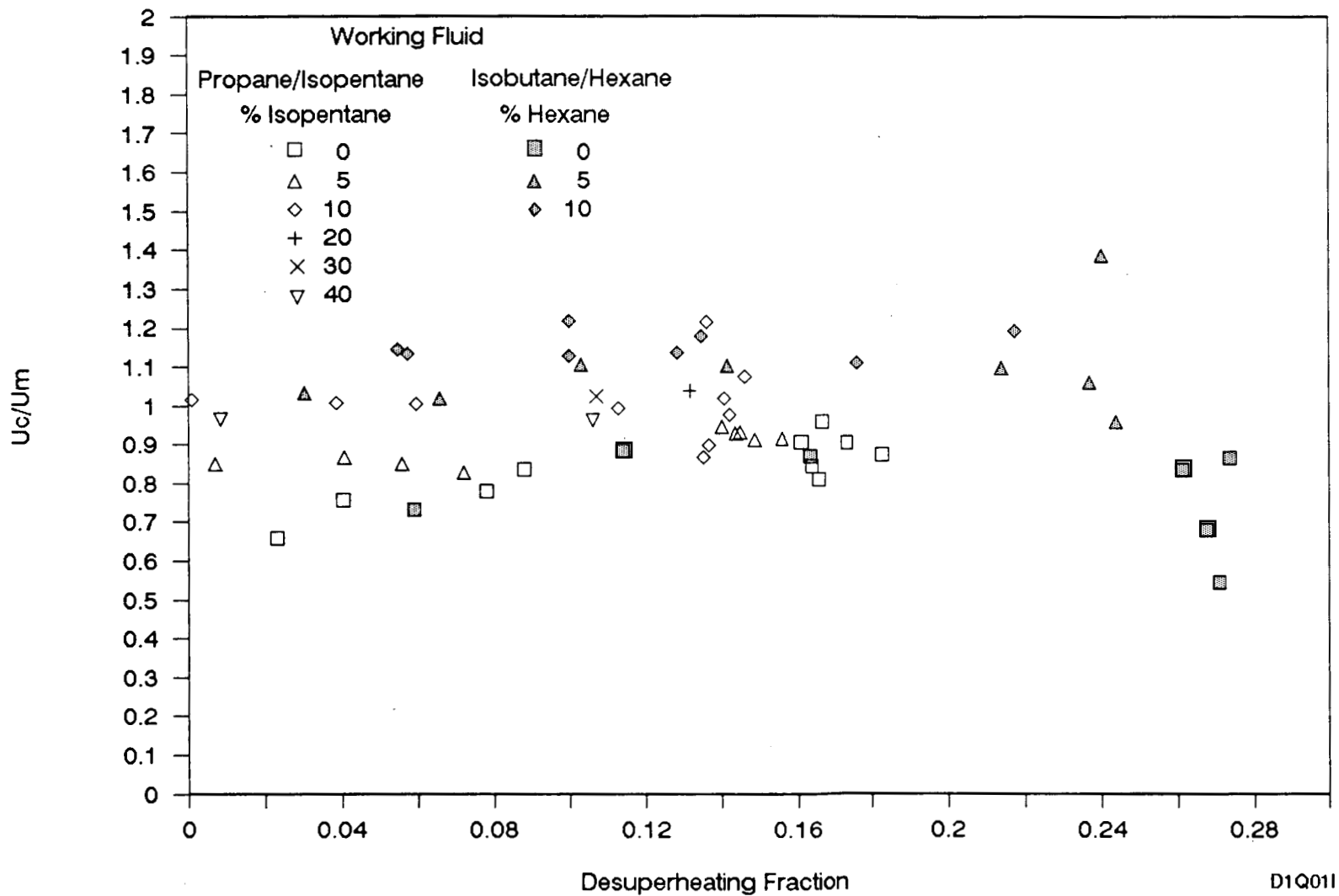


Figure B1. Condenser performance and the original predictive method in the vertical orientation (overall heat transfer coefficient).

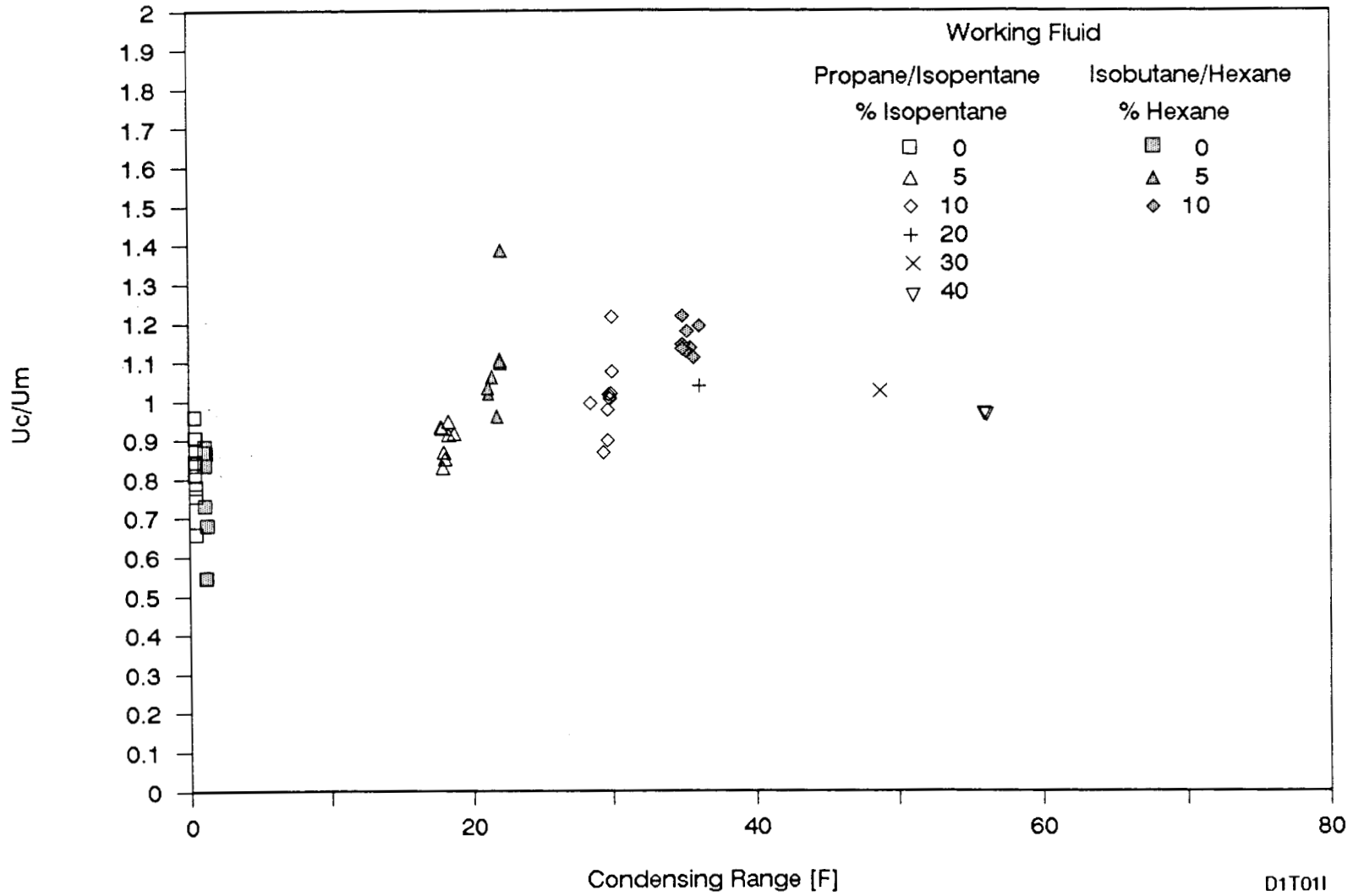


Figure B2. Condenser performance and the original predictive method in the vertical orientation (overall heat transfer coefficient).

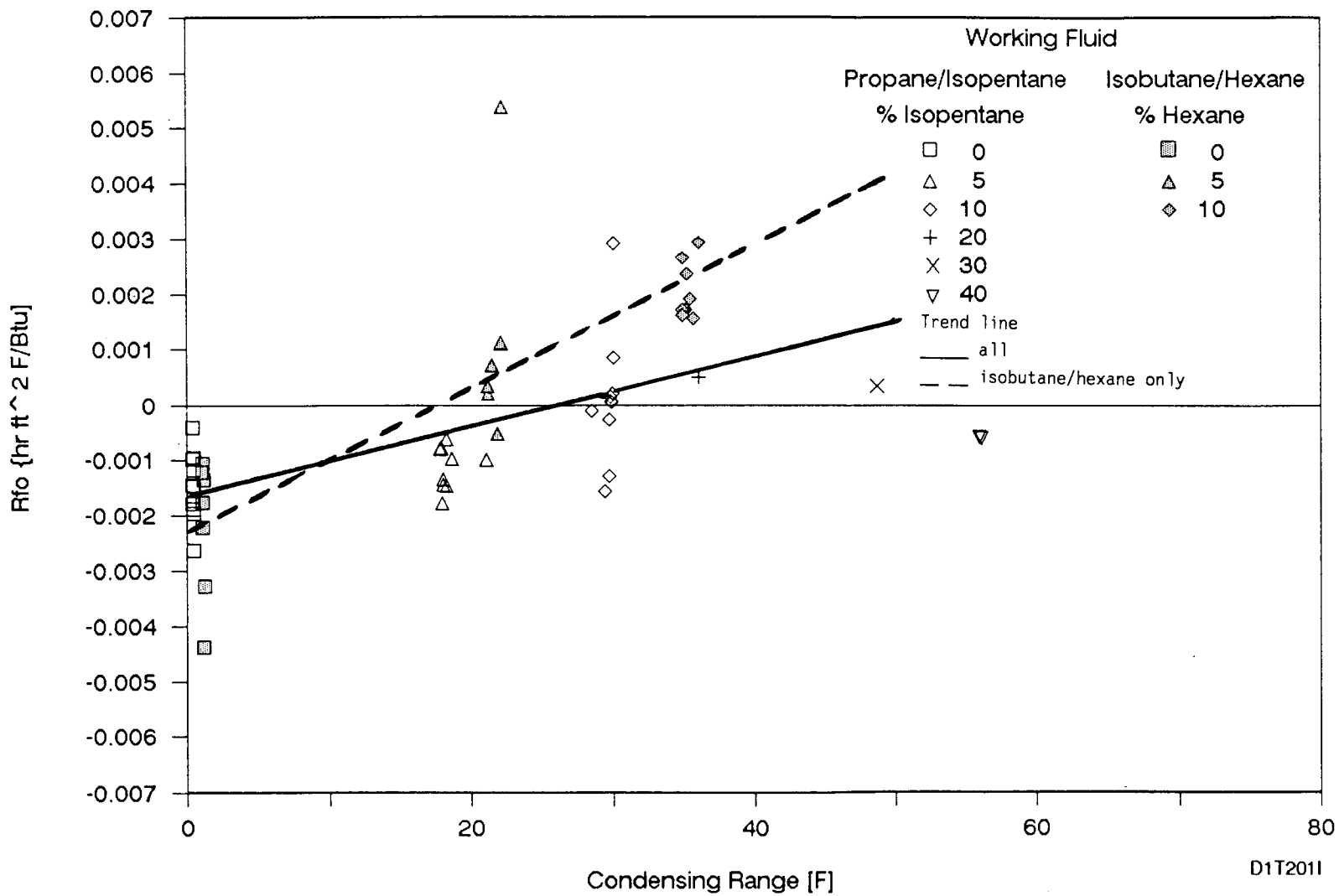


Figure B3. Condenser performance and the original predictive method in the vertical orientation (thermal resistance).

10% of the heavier component, the value is +0.00106). If only the isobutane mixture points (the solid symbols) are considered, the slope is more like that for the horizontal orientation with a much larger dependence on mixture composition. The dashed line shows this trend. There is a trend with increasing condensing range, but with the small amount of data reduced at large condensing range and the discrepancy between the isobutane and propane mixtures, the problem cannot be resolved at this time. More data must be reduced at large condensing range. To clarify any dependence on actual fouling, additional vertical tests should be performed at the end of the test series.

A more basic variable is the condensing heat transfer coefficient (the inside coefficient. Assuming that the water-side coefficient (shell-side) is correct, the condensing coefficient can be determined. Figures B4 and B5 show the ratio of calculated to actual coefficient plotted against desuperheating ratio and condensing range respectively. The results look quite similar to those for overall coefficient (Figures B1 and B2). Here, 74% of the data is within 20% of the line which represents $U_c = U_m$, and 92% is within 30%. There is no trend with desuperheating ratio, but with condensing range, a similar trend is shown as with the overall coefficient.

For completeness as presented in reference 1, the difference between the calculated condensing temperature (assuming that the coefficients were correctly calculated) minus the measured outlet temperature is plotted in Figures B6 and B7 against the same variables as in the previous sets. Here, the data shows a bias with 49% of the data between 0 and 1 °F and 32% between -1 and 0. There appears to be a dependence on condensing range with the pure fluids attaining greater positive values. This is most evident for the isobutane series (the solid symbols).

In summary, the calculation does quite well in predicting the heat transfer with the condenser in the vertical orientation. The bulk of the data was within 30% of the measured value of condensing heat transfer coefficient and also the overall coefficient. The temperature difference

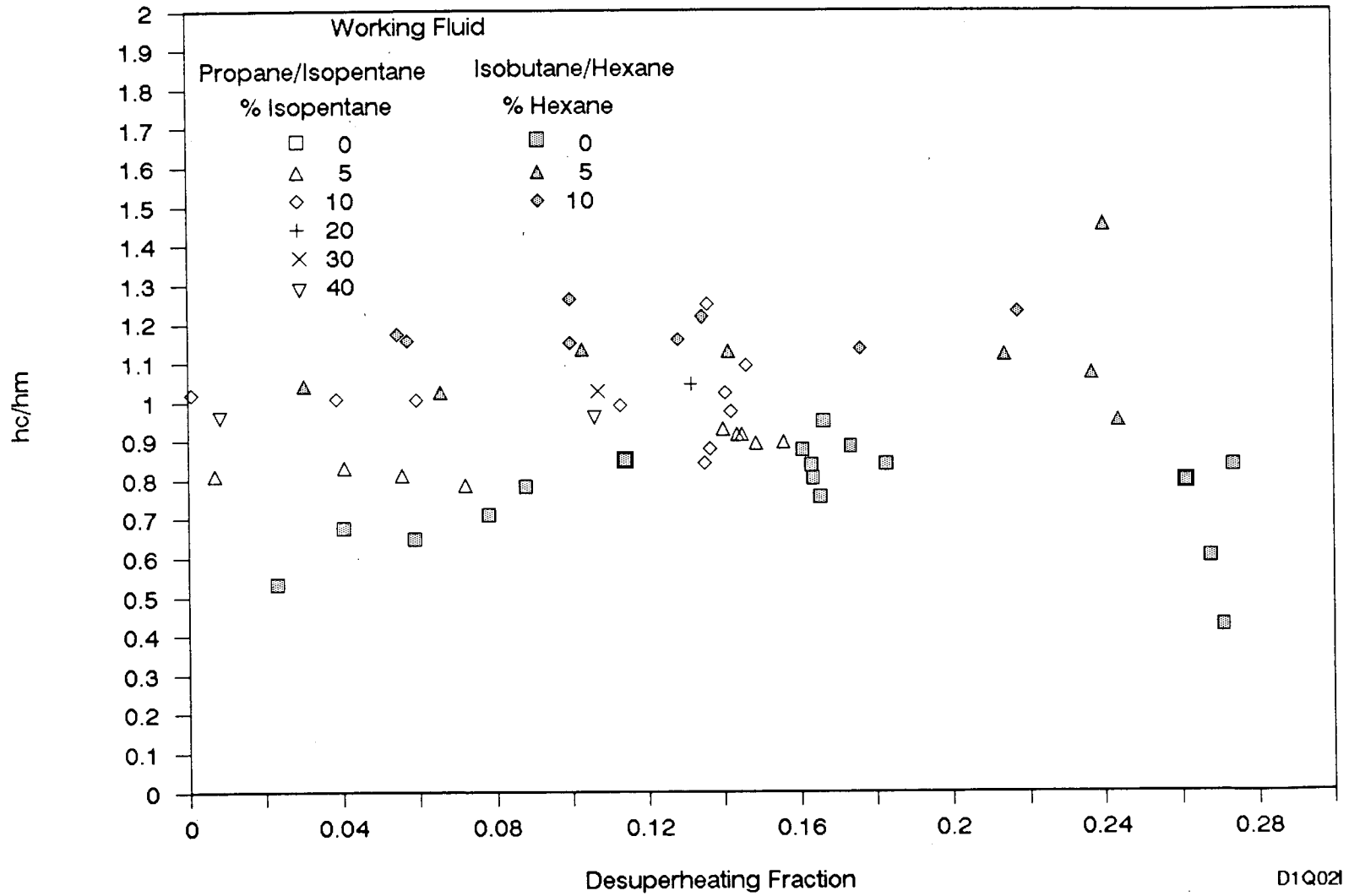


Figure B4. Condenser performance and the original predictive method in the vertical orientation (condensing heat transfer coefficient).

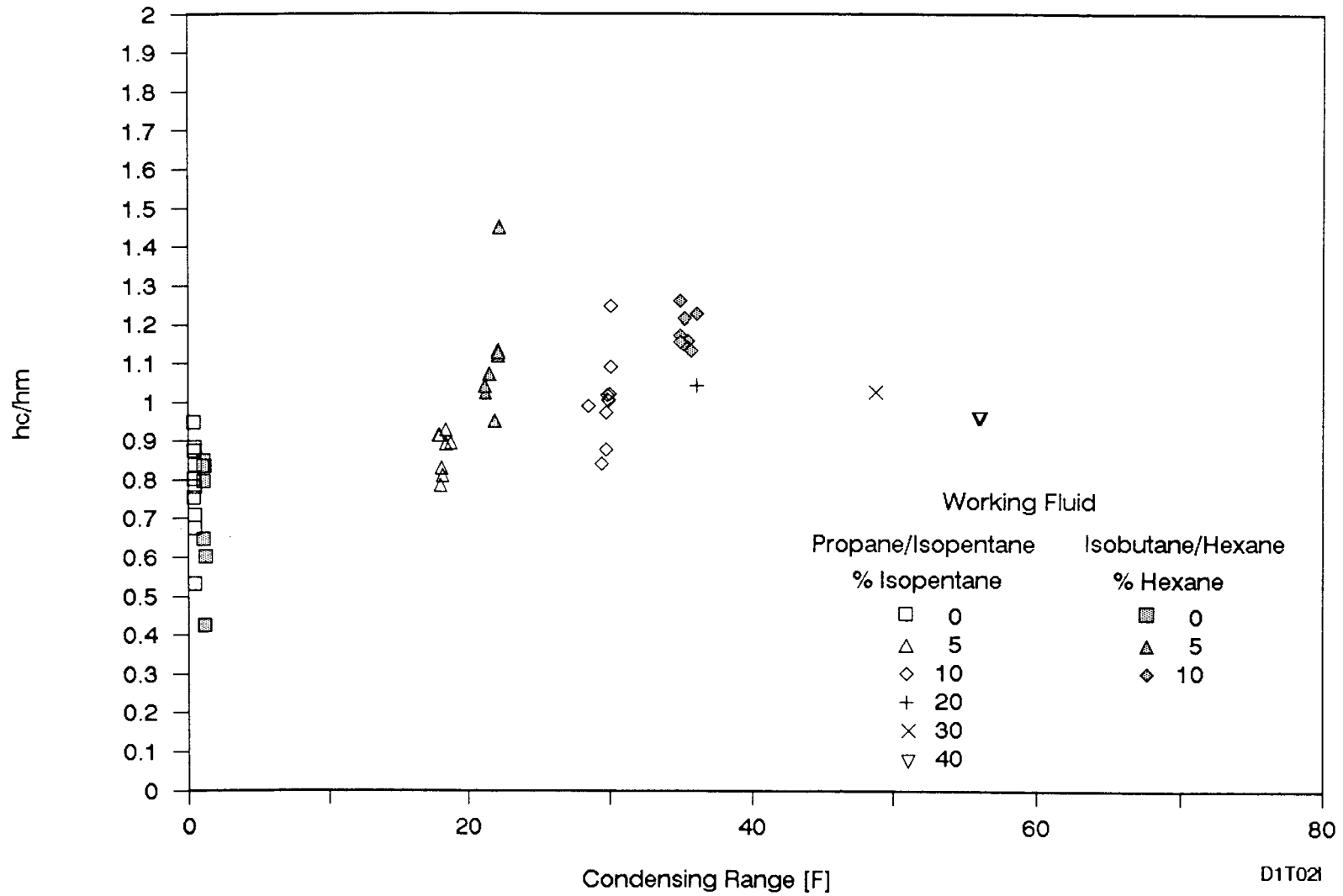


Figure B5. Condenser performance and the original predictive method in the vertical orientation (condensing heat transfer coefficient).

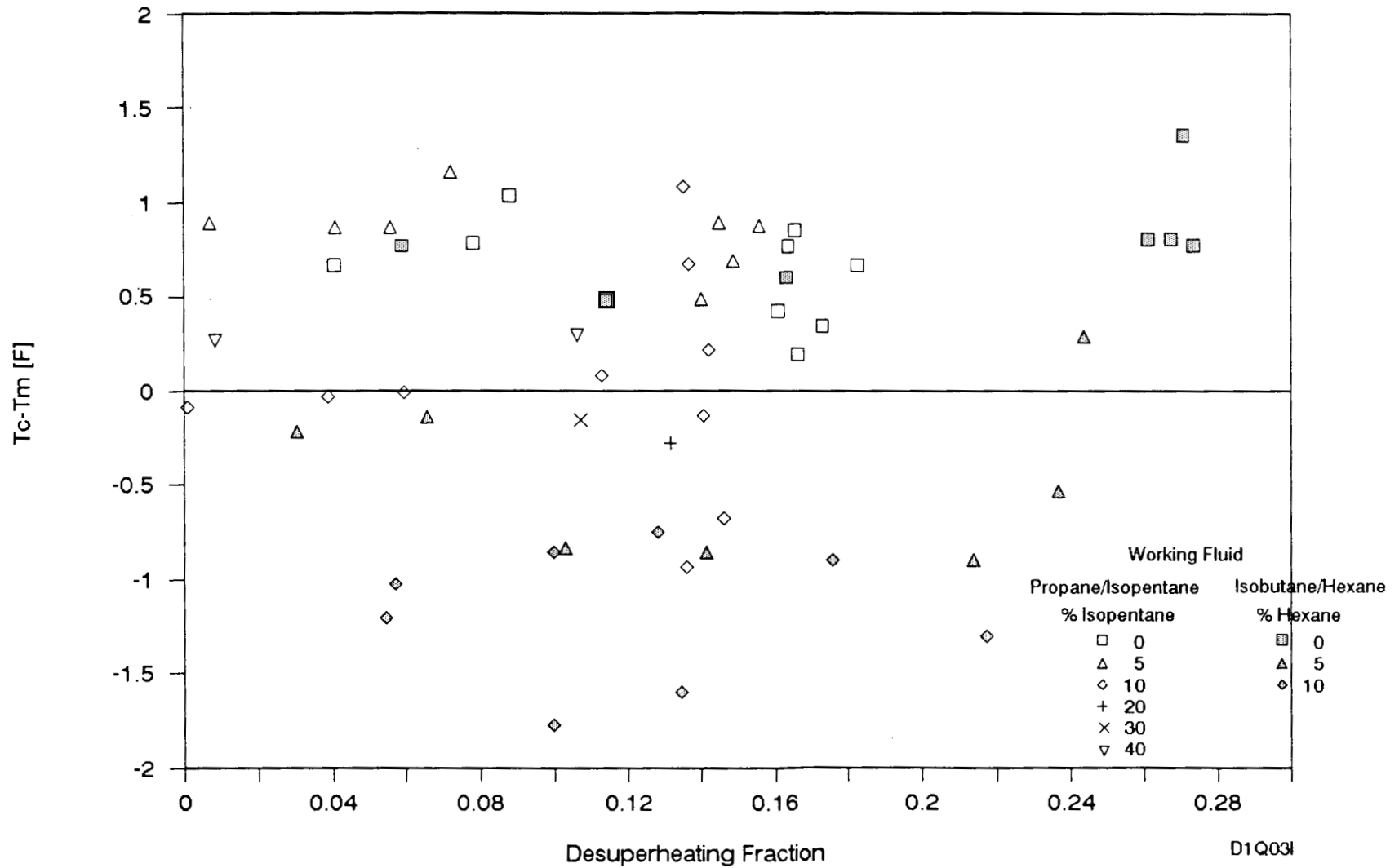


Figure B6. Condenser performance and the original predictive method in the vertical orientation (outlet temperature difference).

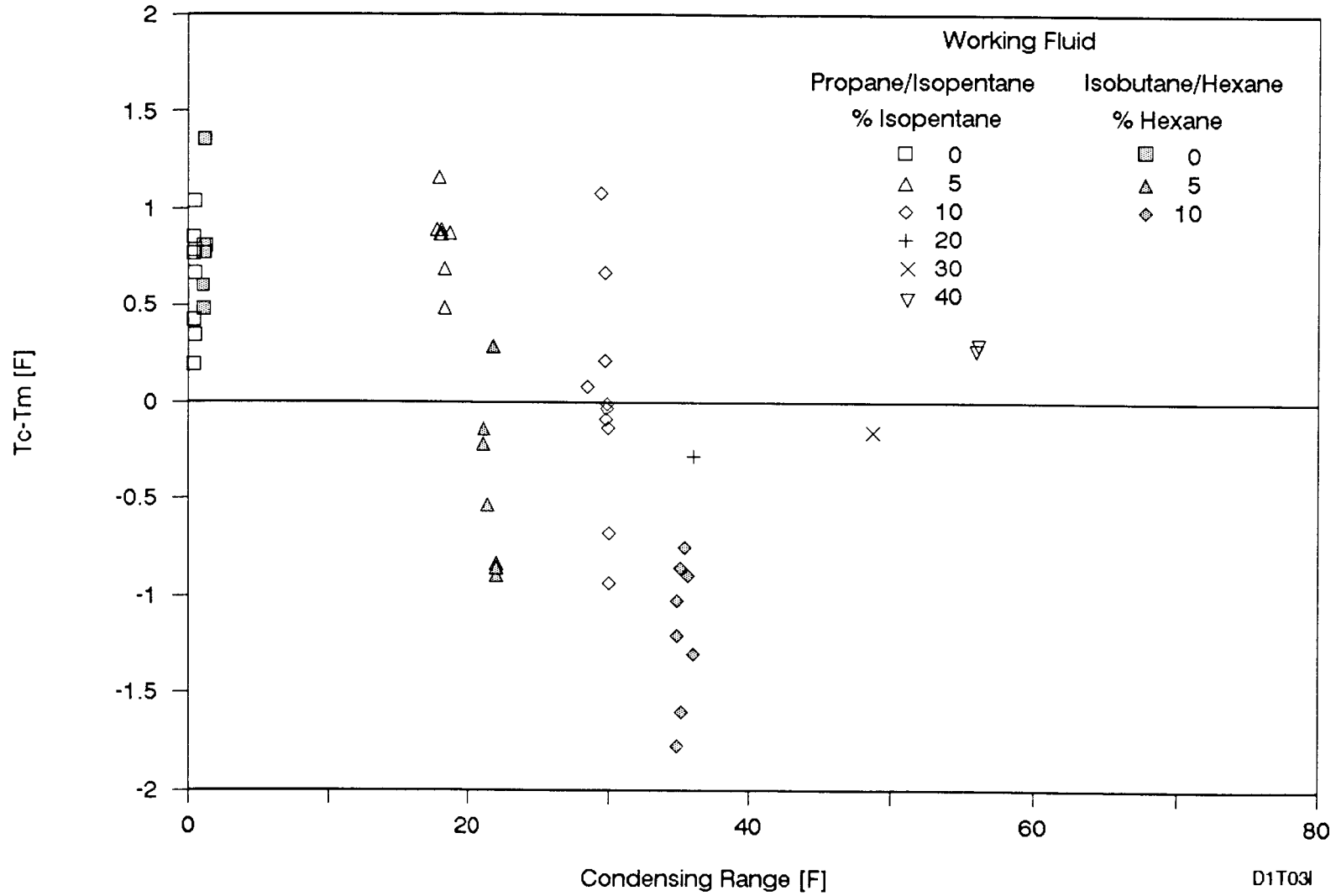


Figure B7. Condenser performance and the original predictive method in the vertical orientation (outlet temperature difference).

showed greater divergence and more dependence on mixture. This might be expected because of the changes in pinch point (minimum approach temperature difference) with composition changes.

Additional data is needed to resolve the dependence of the heat transfer on mixture composition. Some final data with the condenser returned to the vertical orientation would answer the question about the amount of the deviation resulting from fouling. Additional data taken at the higher heavy component concentrations should be reduced to determine if there is a strong dependence on mixture composition or not.