HEAT TRANSFER FOR BOILING INSIDE

HORIZONTAL TUBES

By

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1934

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1936

Submitted in Partial Fulfillment of the

Requirements for the Degree of

DOCTOR OF SCIENCE

From the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

1940

Signature of Author.......................................

Department of Chemical Engineering, May 9, 1940

Signature of Professor
in Charge of Research.

Signature of Chairman of Department
Committee on Graduate Students..........................

1 Parker Street
Cambridge, Mass.
May 9, 1940

Professor George W. Swett
Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge, Massachusetts

Dear Sir:

Attached herewith is a copy of my thesis entitled "Heat Transfer for Boiling Inside Horizontal Tubes", submitted in partial fulfillment of the requirements for the degree of Doctor of Science in the field of Chemical Engineering.

Respectfully yours,

Wallace Kelly Woods

ACKNOWLEDGEMENT

The author is deeply indebted to Professor W. H. McAdams, under whom this work was carried out. The success of the thesis is due in very large extent to the interest in the program, cooperation in interpreting the data, and the very helpful advice received from Professor McAdams.

The author wishes to express his appreciation for the donations of finished oil from Dr. H. D. Wilde, Jr., of the Humble Oil and Refining Company, and from Mr. J. K. Roberts of the Standard Oil Company (Indiana). The high-speed photographs of the glass return bends were taken by Professor H. E. Edgerton to whom thanks are due. Professor W. G. Whitman allotted funds which helped to defray the costs of construction of the apparatus. The brazing done by H. F. Gray comprised an essential step in the construction of the apparatus.

The author is grateful for the cooperation of R. L. Bryan and L. C. Heroman in obtaining data and performing many tedious calculations; of D. J. Bringardner and W. Wilson in obtaining basic data on the benzene-oil mixtures; and of his wife, Lydia H. Woods, in assisting with the runs on water.

TABLE OF CONTENTS

TABLE OF CONTENTS (cont'd)

LIST OF FIGURES

<u>Appendix</u>

LIST OF TABLES

A B S T R A C T

HEAT TRANSFER FOR BOILING INSIDE HORIZONTAL TUBES

ABSTRACT

The advent of the solvent-treating of lubricating oils has raised a considerable problem in the most economical recovery of the solvent from the treated oil. It is known that the rate of heat transfer to mixtures which are rich in viscous oil is very low; if a mixture of solvent and oil is fed into a steam-heated tank from which a heavy oil mixture is to be continuously withdrawn, whatever vaporization is done must occur under conditions quite unsatisfactory for heat transfer since mixing in the tank causes the entire pool to have the composition of the withdrawn product. It has been suggested that higher average rates of heat transfer might be obtained by feeding a lean mixture into one end of a long steam-heated horizontal tube and withdrawing a liquid rich in oil from the other end. At the feed end the concentration of oil is low and the rate of heat transfer would be high; near the discharge end the rate of heat transfer would be low but only a small amount of solvent would remain to be evaporated from the rich oil mixture. Moreover, the vapor generated within the tube might well cause such turbulence near the discharge end of the pipe that unusually high rates of heat transfer would occur despite the large concentration of oil in the liquid phase. No data are available in the literature for predicting the magnitude of heat transfer coefficients under such conditions although the problem was suggested by and is of considerable interest to industry.

A semi-commercial Yaryan-type evaporator was constructed in order to obtain coefficients of heat transfer to mixtures which are progressively evaporated as they move through the inside of a horizontal pipe. The evaporator consisted of four twelve-foot lengths of copper pipe, one-inch standard pipe size, connected in series by glass return bends. Each copper pipe carried three separate steam jackets, three feet two inches long, and condensate was collected from each individual steam jacket in order that changes in the rate of heat transfer along the pipe could be followed. The vapor-liquid mixture leaving the evaporator was separated, the vapor condensed and the liquid cooled, and the two streams continuously mixed and returned to the evaporator through a pump. An overall heat balance could be obtained from the rate of condensation of steam and the rates of flow and temperature rises of the cooling water in the condenser and cooler. The feed rate was determined by heat and material balances and checked by means of an orifice. Pressure drops across each of the four passes were obtained, and fluid temperatures were measured at the entrance to each pass. For each jacket two thermocouples were imbedded in the wall of the copper pipe. Drop-wise condensation of the steam was promoted by the use of octyl thiocyanate.

Commercial benzene was chosen as a solvent because of its convenient boiling point. A series of runs were made on pure benzene, followed by a series of runs on mixtures of benzene and oil. The oil used was a finished propane-treated bright stock of viscosity S.A.E. 50, containing no light ends. Finally a series of runs was made on distilled water. High speed photographs of the glass return bends were taken by stroboscopic light during some of the runs on water.

The results of the runs on both pure water and pure benzene indicated that the overall heat transfer coefficients increased as the cumulative weight per cent of the vapor increased, and subsequently decreased sharply at high values of the cumulative weight per cent vapor. During early stages of the evaporation the liquid tended to flow along the bottom of the pipe with a vapor layer above it except for some intermittent surging of slugs of liquid, whereas further along a dense mixture of vapor and spray filled the tube. The increase in the heat transfer coefficient is possibly due to the increase in wetted area around the circumference of the pipe, as well as to the increased turbulence. After large cumulative per cents of the feed had been vaporized it was noted that the liquid phase was carried as a spray down the center of the pipe, the pipe walls apparently becoming relatively dry and "starved" for liquid, causing a large decrease in the heat transfer coefficient.

In the runs on water the overall heat transfer coefficients rose from 1000 B.t.u./(hr.)(sq.ft.)(°F.) at initial boiling to 2000 at a cumulative weight per cent vapor of 40%. (Figure 30) At 70% vaporized the coefficient had decreased to 1000, and when evaporation was nearly complete coefficients in the range of 50 to 100 were obtained. A moderate increase in steam pressure increased the heat transfer coefficient when the cumulative weight per cent of vapor was below 40%. These results were relatively independent of a variation in feed rate from 300 to

1000 lbs./hr. The maximum coefficient of 2000 was obtained with vapor velocities ranging from 100 to 400 ft./sec. (Figure 32)

For benzene at low steam pressures (around 2 lbs./sq.in.gage) the initial heat transfer coefficients at the beginning of boiling were around 300, independent of feed rate. (Figure 36) The maximum heat transfer coefficient and the corresponding cumulative per cent vapor, however, depended on the feed rate, being 940 and 45%, 700 and 55%, and 540 and 60% when the feed rates were 1000, 700, and 400 lbs./hr., respectively. At high steam pressures the heat transfer coefficients were reduced to well below 100 (Figure 37), as occurs when boiling outside of tubes at very high temperature differences. Hence, it is seen that in forced circulation evaporators "vapor binding" can result both from excessive temperature differences and also from too high per cent vaporization. Moreover, it appears from the benzene runs (Figure 37b) that higher temperature differences increase the tendency to become vapor bound at high cumulative per cent vaporization. A limited amount of data indicates that the critical temperature difference for maximum coefficient at low cumulative per cent vaporization is about 22° to 26°C. Critical temperature differences of this same magnitude have been obtained when boiling benzene outside of horizontal pipes. At vaporizations greater than 95% the heat transfer coefficients obtained were substantially equal to predicted heat transfer coefficients for superheating benzene vapor.

A striking result of the runs on benzene and on water was the abnormal behavior of the first jacket after a return bend after large per cents of the feed had been vaporized. If the walls of the jacket just prior to the bend are dry and the liquid is being carried as a spray down the center of the tube, the effect of the return bend is to whirl part of this spray against the walls of the first jacket after the return bend, resulting in increased heat transfer rates in that jacket. No such beneficial effect of a return bend was noted until after the heat transfer coefficient had begun to decrease because of the high cumulative per cent vapor. When the cumulative weight per cent vapor was in the range of 60% to 90% the heat transfer coefficient in the first jacket after the U-bend was usually 200% to 300% greater than was obtained in the second and third jackets after the U-bend. The use of a twisted ribbon inside the pipe is indicated, thereby breaking up the core of spray and whirling it against the pipe wall.

In the runs on mixtures of benzene and oil the feed composition varied from 90 to 12 weight per cent benzene. With the available steam pressure the heaviest liquid product contained 4% benzene. The feed rate was held substantially constant at 1000 lbs./hr.; a difficult problem in analysis of the data was also considerably simplified when no consistent effect of temperature difference could be detected. After boiling had commenced a substantial portion of the heat transferred was used in raising the temperature of the vapor and the residual liquid because of the increased boiling point as the benzene evaporated. It appears that the effect of temperature difference was considerably minimized by the decrease in relative amount of heat used for vaporization. For a given constant composition of the residual liquid, the heat transfer coefficient after 40% of the feed had been vaporized was approximately double that during the early stages of boiling. (Figure 42) Lines of constant liquid composition on a plot of heat transfer coefficient versus cumulative per cent vapor were similar in shape to the curve for water; the curve for 10% benzene in the residual liquid showed a heat transfer coefficient of about 140 B.t.u./(hr.)(sq.ft.)(°F.) at a vaporization of 40%.

For the benzene-oil runs, a beneficial effect of the return bends was noted even at low percentage vaporization. As the liquid-vapor mixture flows through the pipe it is believed that the liquid near the walls becomes depleted in benzene. The effect of a return bend is to destroy this concentration gradient so that the first jacket after a return bend contains a benzene concentration at the wall as high as that which prevails in the main body of the liquid. For a given feed, of which a given cumulative per cent has been vaporized before entering the jacket, a low coefficient will be obtained if the jacket is just prior to a return bend, while a higher coefficient would have been obtained had the jacket immediately followed a return bend. (Figure 40)

Fanning friction factors were calculated for all runs based upon an assumption that the vapor and liquid were moving at the same velocity. When the cumulative weight per cent vapor was less than 20% abnormally low friction factors were obtained. If the liquid is actually creeping along the bottom of the tube and the vapor is speeding over the top of the liquid, the assumption that the liquid is moving as fast as the vapor ascribes a larger change in kinetic energy to the stream than actually exists. This erroneously large change in kinetic energy accounts for so much of the pressure drop that a relatively small proportion remains to be accounted for by friction.

At vaporizations in excess of 20% both the benzene runs and the water runs yielded friction factors in the vicinity of 0.006. For those benzene runs in which the tube was vapor bound by excessive temperature difference the friction factors were lower, -- around 0.004. For the benzene-oil mixtures the friction factors, as expected due to the higher viscosity, ran higher, averaging about 0.012.

Because the apparatus is so well suited for such studies, its use is recommended in a long-term intensive investigation of point coefficients of heat transfer for boiling inside of horizontal tubes. Of special interest will be the effects resulting from the insertion of a twisted metal ribbon inside of the pipe.

In summary, the accomplishments of this thesis are: (1) determination of heat transfer coefficients for three different liquids boiling inside of a one-inch horizontal pipe, and the determination of the effect of cumulative weight per cent vapor upon the coefficients; (2) determination of friction factors; (3) discovery of two types of vapor binding for vaporization inside tubes, one being due to excessive cumulative per cent vaporization and the other to excessive temperature difference; (4) with pure liquids, detection of a beneficial effect of return bends at high cumulative per cent vaporization only; and (5) with benzene-oil solutions, detection of a beneficial effect of return bends even with small cumulative per cent vaporization.

I N T R O D U C T I O N

INTRODUCTION

The object of this thesis is to investigate the rate of heat transfer to liquids boiling inside of horizontal heated tubes, paying specific attention to changes in the rate of heat transfer as large fractions of the liquid are vaporized.

In industrial practice, liquids are frequently boiled inside of pipes, in contrast with passing heated pipes through a pool of liquid. The primary cause of this practice lies in the comparative ease of removing scale deposits from the inside of tubular heating surfaces. Two disadvantages may result from the practice: (a) the pressure drop required to force the fluid through the pipe requires an extra expenditure for power and fixed charges on pumps, and (b) since the liquid is heated to a temperature higher than its boiling point at discharge pressure the available difference in temperature between the hot wall and the boiling liquid is reduced. These disadvantages inherent to <u>forced circulation</u> are minimized when boiling inside vertical tubes by using <u>natural circulation</u> (natural <u>convection</u>). Natural convection acts on the same principle as is utilized in air lifts for pumping corrosive nitric acid: vapors generated in vertical pipes may "pump" the feed through the pipes. Obviously, the rate of feed to the pipe depends upon the rate of heat transfer in natural circulation.

Forced circulation is also used frequently in the belief that the added turbulence resulting from high velocities through the inside of the pipes should improve the rate of heat transfer. If the vaporization is accompanied by chemical decomposition, as occurs in the thermal cracking of oil, forced convection is imperative in order to limit the time that the liquid is in contact with the heating surface and thus to control the extent of the reaction.

Industrially, liquids are frequently boiled which are (practically) completely volatile. For instance, in making liquid air some of the liquid product is boiled by the absorption of heat from warm air, and the dry vapors may then be sent to expansion turbines. The production of distilled water from natural water is another example. But many times liquids which are boiled industrially are mixtures of volatile and relatively non-volatile substances. Aqueous solutions of caustic soda and of glycerine are concentrated by evaporation of water from the mixture. Another very important case has recently arisen as a result of the solvent treating of petroleum products.

In the propane de-waxing of lubricating oils the oil is diluted with liquid propane and chilled. At low temperatures the paraffin wax precipitates out and is filtered from the thinned mixture. It is then necessary to remove and recover the propane from the relatively non-volatile oil. Similarly, under different operating conditions, oil may also be propane de-asphaltized. The addition of liquid propane results in the formation of two liquid layers, an oil layer and an asphalt layer, from which the propane is recovered after separation. It is to be noted that such solvent-oil mixtures may contain more than 80% solvent, by weight. A wide variety of such solvents exists, including sulfur dioxide, furfural, benzene, and various mixtures.

Special problems arise in connection with the boiling of such mixtures in consequence of the changing liquid composition. As the volatile component is boiled off, the boiling point of the solution rises and the available temperature difference between the heating medium and the liquid decreases. In fact, it may be necessary to resort to steam stripping in order to remove the last traces of solvent. Moreover, as the amount of non-volatile increases the viscosity of the solution increases (despite the increasing temperature). This results in decreasing the rate of heat transfer at a given temperature difference. Solely from a heat transfer standpoint, the ideal way of conducting such a vaporization would be by batch operation. Solvent would be removed from the thin solution under conditions favorable for heat transfer, and as the heat transfer conditions become less favorable smaller and smaller amounts of solvent would remain to be removed. The inherent disadvantages of batch operation in large scale work almost force the use of continuous operation.

If the mixture is continuously fed into a heated tank and non-volatile product is continuously withdrawn from the tank, the mixing which results from the boiling action provides that the liquid in the tank will have substantially the same composition as that of the product. Hence, although the feed may contain but small amounts of non-volatile, whatever solvent is evaporated will be removed under conditions very unfavorable for heat transfer. If several tanks in series are used this unsatisfactory condition is modified, and in the limit an infinite number of tanks with a differential change in composition between the tanks would be equivalent to heat transfer conditions encountered in batch concentration. A practical method of obtaining this objective may be to feed into one end of a heated pipe and to withdraw product from the other end. This suggestion presupposes that steady flow through the pipe will eliminate objectionable mixing of thin and concentrated liquids, and is of course subject to the two disadvantages of forced convection discussed on page 1.

An important factor in the above suggestion is the possibility that the large volume of vapor generated may create such turbulence near the discharge end of the pipe that substantial improvement in the heat transfer conditions at this end (where it is badly needed) may be noted. This imposes a long pipe, or a series of pipes, with separation of vapor from contact with the liquid only at the discharge end. Because of the long pipe length probably necessary for the vaporization of large per cents of a feed fed at a reasonable rate, vertical pipes appear inadvisable. Such a horizontal-tube evaporator, with boiling occurring inside the tubes, is known as a Yaryan-type evaporator.

A very limited amount of data exists in the literature on Yaryan-type evaporators. Badger (3) reports the results of tests on boiling water inside of a 2 1/2 inch standard steel pipe, steam-jacketed, mounted horizontally, 50 feet long, and containing no bends. Distilled water entered at 196°F. and discharged at atmospheric pressure. The steam pressure was about 10 lbs./sq.in.ga. and the feed rate was varied from 1000 to 6000 lbs./hr. The maximum overall heat transfer coefficient obtained was only 370 B.t.u./(hr.)(sq.ft.)(°F.), —

a value so low that it must be concluded that very poor condensing steam side conditions were seriously reducing the overall coefficient. However, Badger did observe that the average heat transfer coefficient for the apparatus increased as the feed rate was increased from 1000 to 3600 lbs./hr., although there was a subsequent small decrease at higher feed rates.

Under the author's supervision, Slade (12) carried out work on the boiling of water inside of a 31-inch length of horizontal stainless steel tubing, 0.200-inch inside diameter, heated by direct passage of electric current through the metal walls of the tube. An unfortunate drop in water pressure resulted in overheating and burning out the test section after only a very limited amount of data had been collected.

Considerable work has been done on boiling water inside of vertical tubes. In most of this work (9,11) only small per cents of the feed were vaporized and the heat transfer coefficients obtained indicated that the mechanism of boiling was largely one of superheating the feed with subsequent flashing of vapor. Oliver (10), working under the author's supervision, showed a definite effect of temperature difference by using high-pressure, well-promoted steam. Oliver used entering feed velocities as high as 16 feet per second, and because of a short heated length vaporized relatively small per cents of the feed despite the large temperature differences encountered. Stroebe (13) used a long vertical steam heated tube and evaporated, at times, almost all of the feed.

Under the author's supervision heat transfer studies were made in small apparatus using mixtures of benzene and oil. Wilson (15) boiled benzene-oil mixtures in the short vertical tube used subsequently by Oliver (10); he did not attempt to promote dropwise condensation of the steam, and his thermocouple calibration was unsatisfactory to such an extent that no trustworthy heat transfer coefficients could be obtained from his data. Bringardner (4) boiled benzene-oil mixtures in natural convection outside of horizontal tubes. By the use of high steam pressures he was able to vapor-bind the surface of the tube when mixtures containing 80% to 100% benzene were being boiled. With increasing amounts of oil not only did the boiling point increase, but also the temperature difference required to obtain the maximum heat flux increased. Brief tests were made on mixtures containing only 6% benzene by weight, but the heat transfer was so small that errors in determining heat loss from the apparatus seriously affected the precision of the data.

P R O C E D U R E

PROCEDURE

A semi-commercial Yaryan type evaporator was used to obtain the results of this thesis. The heating section consisted of four twelve-foot sections of copper pipe, 1-inch standard pipe size. On each of these pipes were placed three steam jackets, constructed of 2-inch standard steel pipe and 3 feet 2 inches long.

Steam from a 185 lbs./sq.in. steam main was dried by passing it through a centrifugal (cyclone) separator. The steam pressure was measured just before the steam entered a steam header from which it was distributed to the individual jackets. Small holes were drilled in each of the jackets to prevent the accumulation of non-condensable gases. In most runs the steam was promoted with octyl thiocyanate[16] to assist in obtaining dropwise condensation of steam on the outside of the copper pipe. Steam condensate from each of the twelve jackets flowed through a steam trap and a drain line and was collected in receivers over a measured period of time.

The feed was pumped through an orifice and through the four sections of copper pipe in series, -- each section of pipe serving as one pass through the heating section. The copper pipes were connected by glass return bends or U-bends. The fluid leaving the last pass entered a vapor-liquid separator from which the vapors were sent to a water-cooled condenser. The liquid leaving the separator was mixed with the condensed vapors and returned to the pump. During the runs on benzene-oil mixtures it was necessary to pass the hot liquid oil from the separator through a cooler before the oil could be mixed with the condensed benzene. The frictional resistance of the cooler created a pressure of less than atmospheric on the intake side of the pump, resulting in some air leaking into the oil through leaks in the pipe connections and the soldered joints of the cooler. Accordingly, a large drum, called the settler, was installed between the pump and the feed orifice in order to allow this air to settle out and be vented off before the feed entered the heating section.

Copper-constantan thermocouples were installed for the purpose of measuring pipe-wall temperatures. Pressure connections were made to each pipe at the end of the pass and the pressure drop across the pass (and the preceding U-bend) was measured with mercury-filled manometers. Glass thermometers measured the temperature of the fluid as it entered each pass. The rise in temperature of the water flowing through the condenser and through the cooler was measured by fractional thermometers and the rates of each of these water streams was measured by orifices.

The oil cooler and the settler drum were thermally insulated with wool felt. During all water runs, benzene runs B6 and B7, and benzene-oil runs BO-12 to BO-24 the steam condensate drain lines were insulated with an approximately half inch thickness of a mixture of rock wool and magnesia. The steam condensate drain lines were left bare for the rest of the runs. The rest of the apparatus was insulated with at least 1-inch thickness of either magnesia or a mixture of rock wool and magnesia (Johns-Manville 450). The average thickness of insulation on the steam jackets was close to two inches.

A sketch of the apparatus is given in Fig. I. Photographs of the apparatus in various stages of construction are included in Illustrations A to J. Details of construction are given in the Appendix (pp. 151 to 161).

Operation

A series of fourteen runs on pure benzene were taken by R.L. Bryan (5) and the author. Five more runs on benzene were taken by L. C. Heroman (6) and the author, followed by twenty-four runs on mixtures of benzene and oil and by two more benzene runs. The author concluded the date with fifteen runs on pure water and a special water run for the purpose of obtaining high speed photographs of the U-bends during a run.

In starting up a run on a pure component (benzene or water) the condenser water was first turned on, the pump started, and the feed rate and steam pressure gradually adjusted to their desired condition. Valves in the lines between the pump and the condenser and separator were opened wide. The system was then either partially drained or make-up liquid was added until liquid levels were visible in sight glasses attached to the condenser and separator.

A similar procedure was used in starting up a run on a benzene-oil mixture, except that the liquid from the separator was passed through the cooler instead of mixing directly with the condensed benzene. It was necessary to drain all water from the cooler when starting up, only turning the cooling water on gradually after hot oil was flowing through the cooler. Otherwise the oil would become so cold and viscous in the cooler that an insufficient rate of flow could be obtained. Feed composition was controlled approximately by partially closing the valve in the line leading from the condenser to the pump, thereby reducing the rate of recirculation of benzene. All liquid levels were adjusted after steady conditions had been obtained.

The first run of the day was allowed to continue for at least half an hour in order to allow the apparatus and lagging to become warm.

In subsequent runs the operating conditions were changed and the apparatus usually attained steady conditions during the time required to measure the condensate collected in the preceding run. Especially in the benzene-oil runs it was difficult to obtain specified operating conditions, for a slight drop in steam pressure, say, would affect the pressure drop through the apparatus and hence the rate of flow, and would accordingly affect the feed composition and temperature.

Shortly before taking data the cooling water rates were adjusted to insure substantially equal precision in reading the rate of flow and the temperature rise. The following readings were then taken for all runs:

Condenser water manometer.

Condenser water temperatures in and out.

Fluid temperatures at the entrance to each pass, leaving the separator, and leaving the condenser.

Feed orifice manometer.

Manometers for pressure drop across each pass and for static pressure at the end of the fourth pass.

Steam pressure gage.

Volume of condensate collected from each of twelve steam jackets and time of collection. Also temperature of condensate as measured.

In addition the following measurements were taken during the benzene-oil runs:

Cooler water orifice manometer.

Cooler water temperature in and out.

Oil temperature leaving the cooler.

Analysis of sample of feed and of liquid in separator.

For all water runs and the first six benzene runs:

Reading of 23 thermocouples, by potentiometer.

Cold junction thermometer (held at $0^{o}C.$)

Finally, observations were made of the following glasses:

Description of flow in the U-bends.

Glass window in the separator.

Sight glasses on the condenser and separator.

Air vent on the condenser.

Air vent on the settler (when used).

In addition to the centrifugal separator in the steam line, mentioned above, an auxiliary half-inch line was allowed to exhaust to the roof, increasing the load on the line from the powerhouse, and thereby minimizing heat losses per pound of steam and helping to reduce the possibility of wet steam.

It was found desireable to reduce the time of a run to a minimum in order to avoid fluctuations in operation due to failure of the apparatue to maintain a steady state. After the first ~~few~~ runs on benzene the system of simultaneous collection of condensate from all jackets was inaugurated: each receiving bottle was put "on stream" five seconds after the preceding bottle, and the steam condensate collection was discontinued at five second intervals in the same order. Hence, condensate from the last jacket was measured over a period of time that started 55 seconds after the corresponding period of time for the first jacket. The thermodouple readings were time consuming and were dispensed with for the runs on benzene and benzene-oil after the first six runs had indicated that the resistance to heat transfer of the condensing steam film was minor. With two operators, one man could collect all other data on the apparatus while the second was collecting the steam condensate. The steam pressure and feed rate were always noted at both the beginning and end of the run. Thermocouples were used for all the water runs. While one operator measured thermocouple readings the other operator collected all data on the apparatus, collected steam condensate, and returned for check readings on the temperatures and pressures. Averages of the two readings were used in the calculations.

To insure clean dry benzene during the runs on benzene, the apparatus was operated with substantially total vaporization for a half an hour each day just before runs, and sometimes between runs. The small amount of liquid which appeared in the separator contained a concentration of any contaminants and was drained out of the separator until it was water white. These contaminants consisted largely of brass filings from the pump, the benzene washing out all lubrication.

Samples of about 300 cc. were withdrawn from the settler and from the separator during the runs on benzene-oil mixtures. Half of each sample was withdrawn at the start of the run and the other half a few minutes later at the end of the run. The samples were well mixed before analyzing 100 cc. of the sample for benzene content by simple distillation.

Treatment of Data

All steam condensate measurements were corrected for the density at the temperature of measurement and reported as grams of condensate per minute. Measurements of temperature and steam pressure were corrected in accordance with calibration of the thermometers and pressure gages. The fractional thermometers used to measure cooling water temperatures were graduated in tenths of a degree Centigrade and were not calibrated. Cooler and condenser orifices were calibrated in terms of manometer reading. The feed orifice reading was reported as measured. All other pressure drop readings were converted to and expressed as centimeters of mercury. Details of the subsequent calculations are given in the Appendix, pp. 313 to 328.

A heat balance on the apparatus was the first calculation to be made. This heat balance compared the indicated heat transfer based on steam condensate readings with the indicated heat transfer based upon rise in temperature of condenser and cooler water. For the water runs the heat flux per jacket was based directly on the steam condensate measurements. For the benzene-oil runs and the benzene runs, Series A, the heat flux per jacket was based upon the steam condensate measurements corrected by the ratio of the heat transfer as determined by the cooling water to the heat transfer as determined by the steam condensate. For the benzene runs, with the exception of Series A, the correction factor used was the ratio of the average of the two heat transfer determinations to the heat transfer as determined by steam condensate measurements. If the heat balance failed to check within 10$^o/_o$ ~~of~~ the run was discarded.

For those runs where the steam condensate drain lines were not insulated the heat transfer indicated by the steam condensate readings assumed that any steam formed by the flashing of the steam condensate to atmospheric pressure would be subsequently re-condensed in the drain lines. For the runs after the drain lines had been insulated the measured condensate was first reduced by the amount of condensate equivalent to the heat losses from the steam jackets <u>and</u> drain lines, and the net amount of condensate was then multiplied by a flashing factor to convert from the net amount of condensate collected to the amount of condensate formed at the higher pressure.

Where possible, the rate of feed was determined independently of the feed orifice reading, the rate indicated by the orifice being used merely as a check reading. Thus, if the benzene were completely vaporized and superheated, the known rate of heat transfer indicated the feed rate. Similarly, for the benzene-oil mixtures, the known rate of heat transfer plus the known inlet and outlet temperatures and compositions indicated the rate of flow. In other cases the rate of flow was determined by the feed orifice.reading.

For all cases in which pure benzene or pure water was partially vaporized the temperature was determined with greatest precision by using the saturation temperature at the known pressure. For superheated vapor and for benzene-oil mixtures the observed temperatures were used.

Knowing the feed rate, the rate of heat transfer, and the pressures and temperatures throughout the apparatus, the cumulative weight per cent of the feed vaporized, and the composition of the residual liquid phase for benzene-oil mixtures, could be calculated for any point between jackets.

In mixing standard samples of benzene and oil it was found that the volumes were substantially additive. The difference in density between the oil and the benzene is so slight that analysis of liquid on a volume basis is substantially the same as analysis of liquid on a weight basis. Benzene distilled out of the heavy oil was perfectly water white.

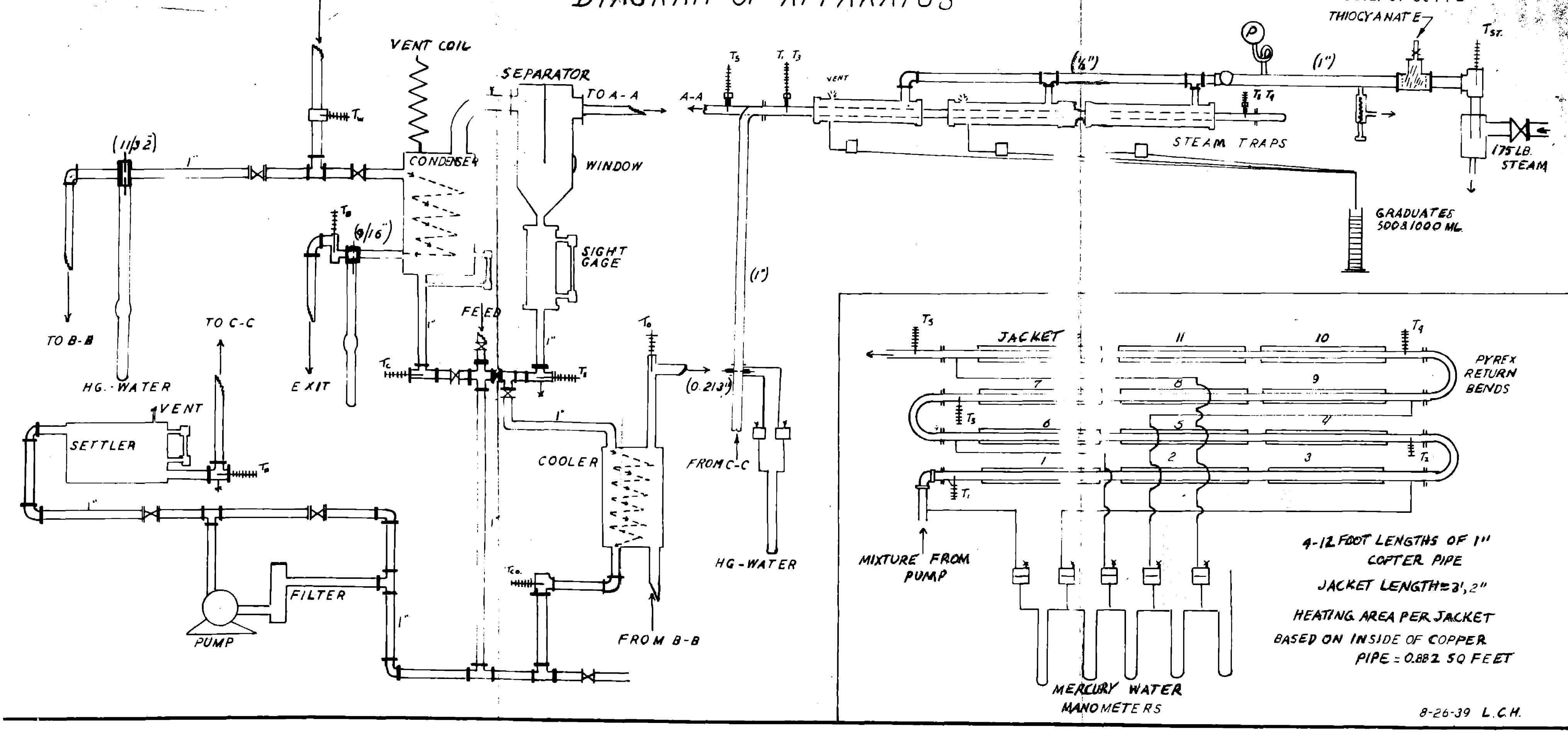


FIGURE I
DIAGRAM OF APPARATUS

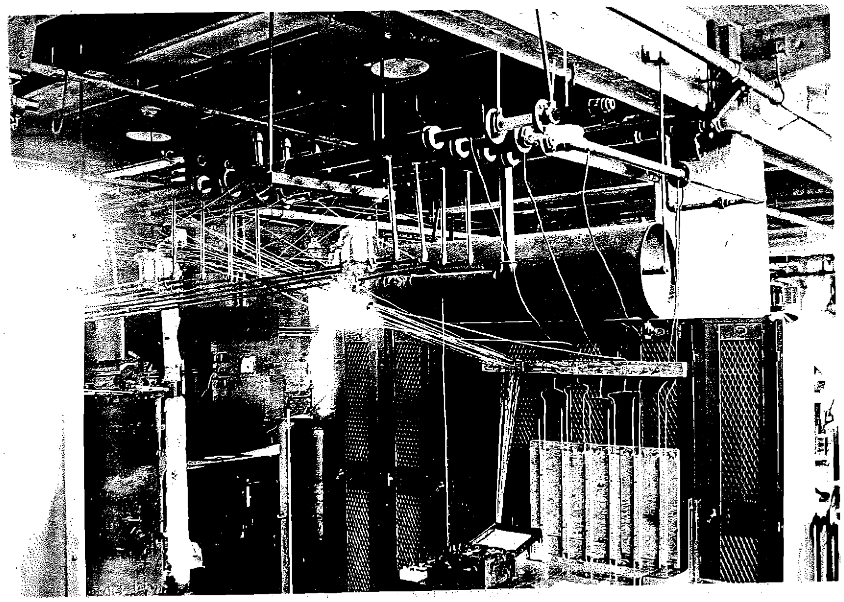


Illustration A. View of heater before insulating and before construction of separator and condenser. Jackets 12, 7, 6, and 1 are visible in the center of the picture (reading from left to right). The straight glass nipple at the end of the fourth pass was removed for mechanical reasons when the separator was installed.

Picture taken by W. K. Woods.

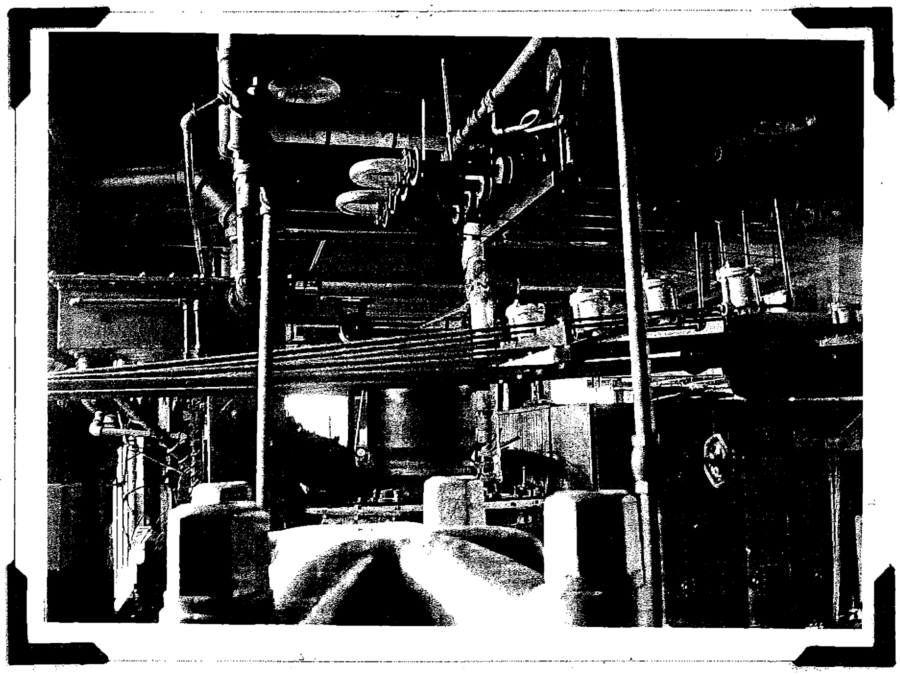


Illustration B. View of heater before insulating. Steam traps for Jackets 3, 4, 9, and 10 visible. Steam pressure gage and glass U - bends at end of first and third passes visible.

Picture taken by W. K. Woods.

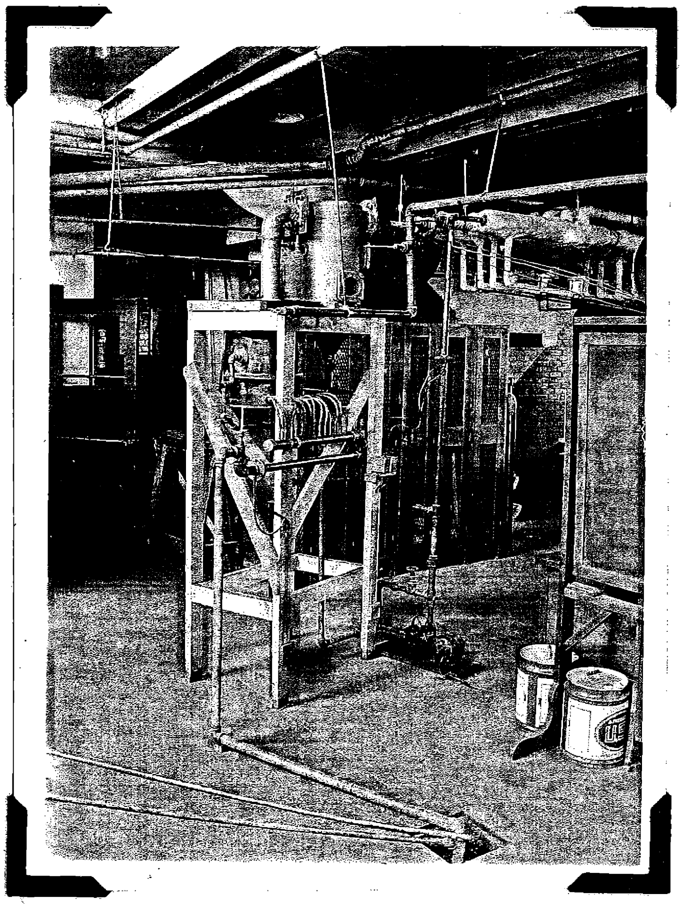


Illustration C. View of separator and condenser before insulating. No cooler installation. Insulated heater appears in upper right-hand corner.

Picture by R. L. Bryan.

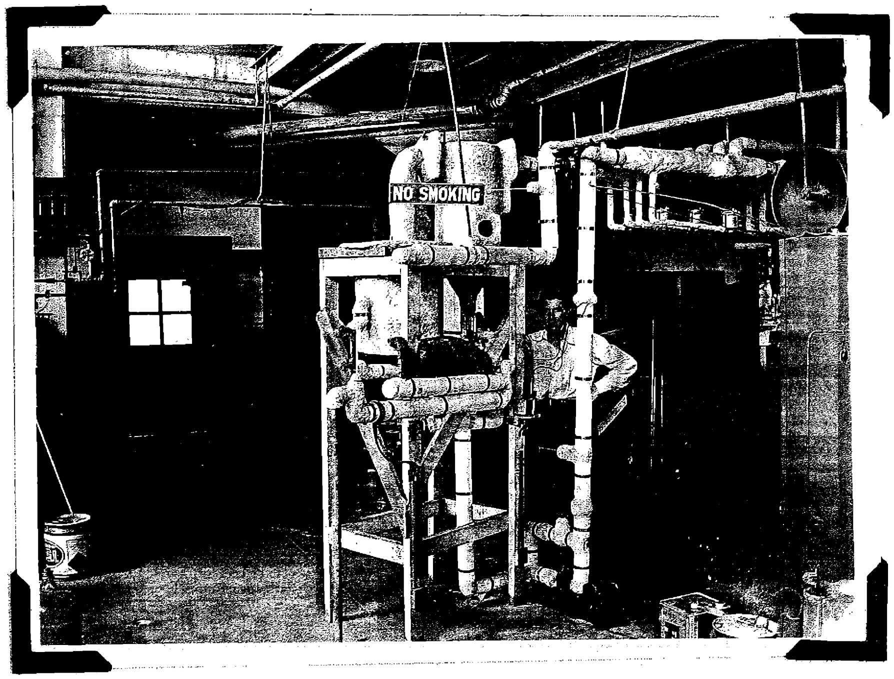


Illustration D. View of separator and condenser after insulating. (Refer to Illustration C for contrasting view).

Picture by R. L. Bryan.

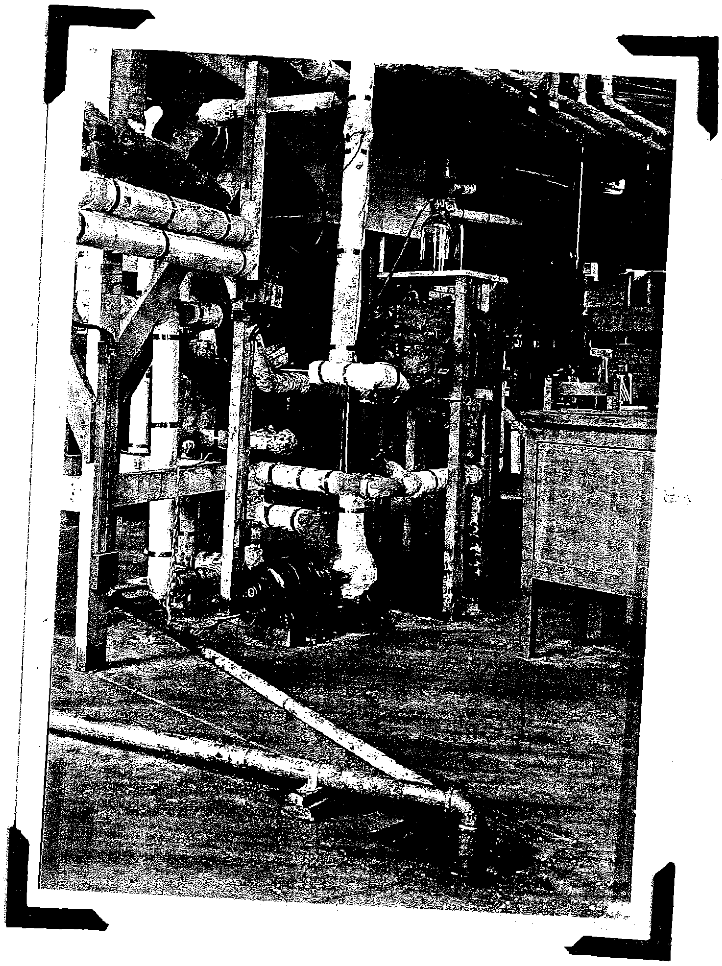


Illustration E. View of pump and settler drum. Condensate drain lines are visible in the upper right-hand corner. Manometers for the condenser water orifice and the feed orifice are visible. The bottle on top of the settler drum catches liquid blown over with the vented air.

Picture taken by L. C. Heroman

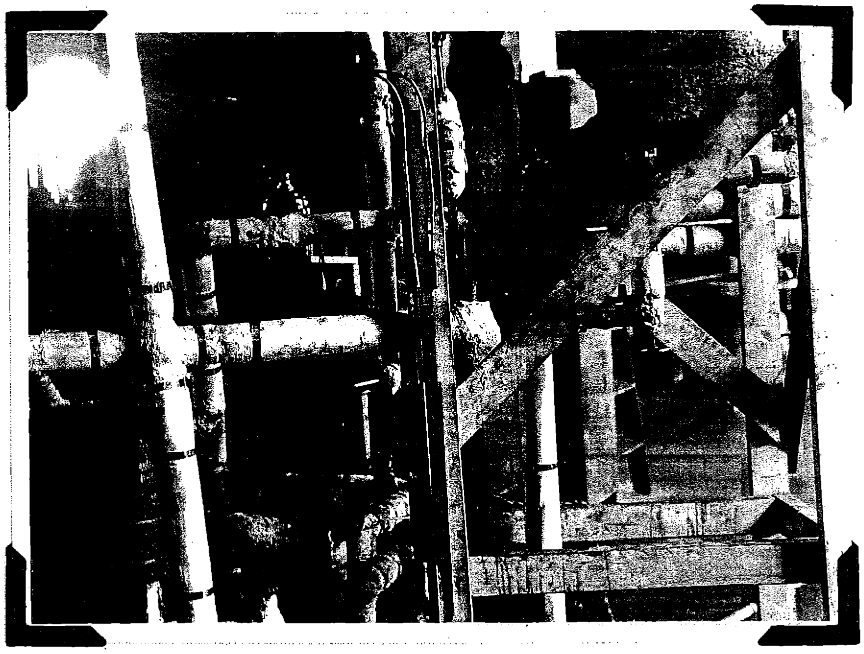


Illustration F. Opposite view to Illustration E. Cooler orifice manometer in center. Cooler in lower left hand corner. Bottom of condenser in upper right hand corner.

Picture taken by L. C. Heroman

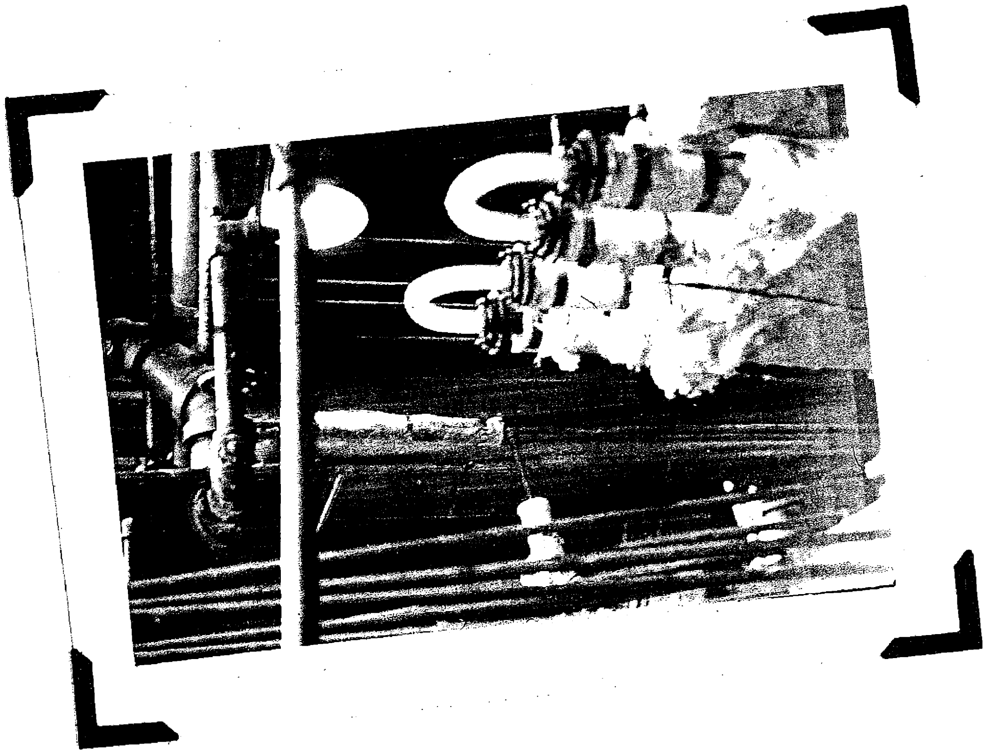


Illustration G. View of glass return bends (U-bends) at end of first and third passes. Picture taken during operation of the evaporator. Non-insulated condensate drain lines appear along the bottom of the picture.

Picture by R. L. Bryan.

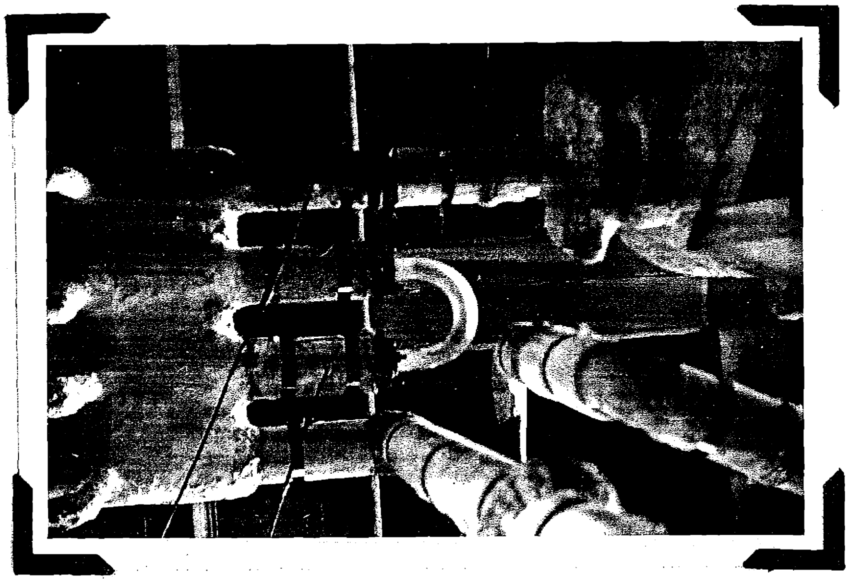


Illustration H. View of glass U - bend at end of second pass. The camera is facing straight up. The horizontal line at the top of the picture leads from the last pass into the separator. Note the insulation on the heater.

Picture by R. L. Bryan.

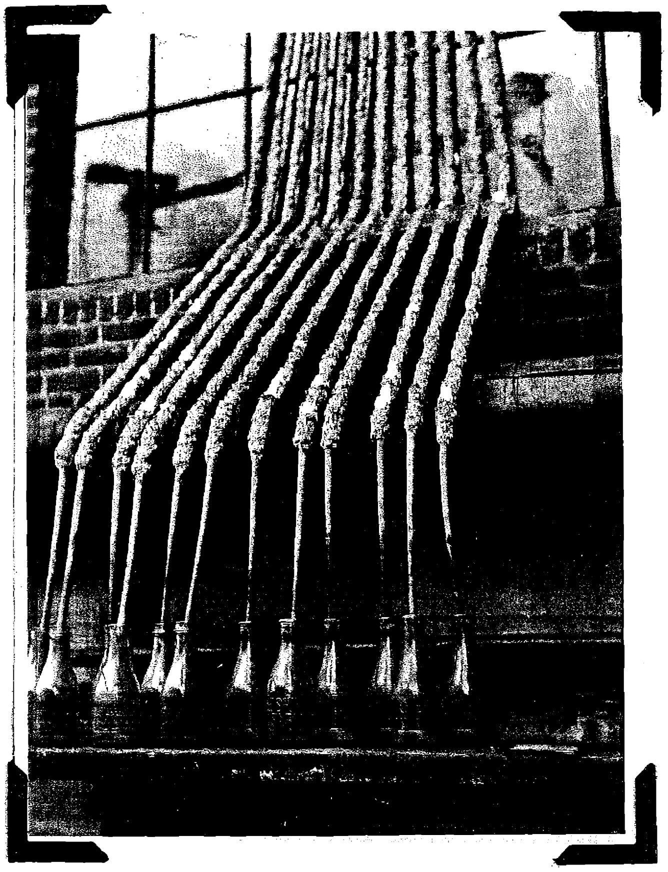
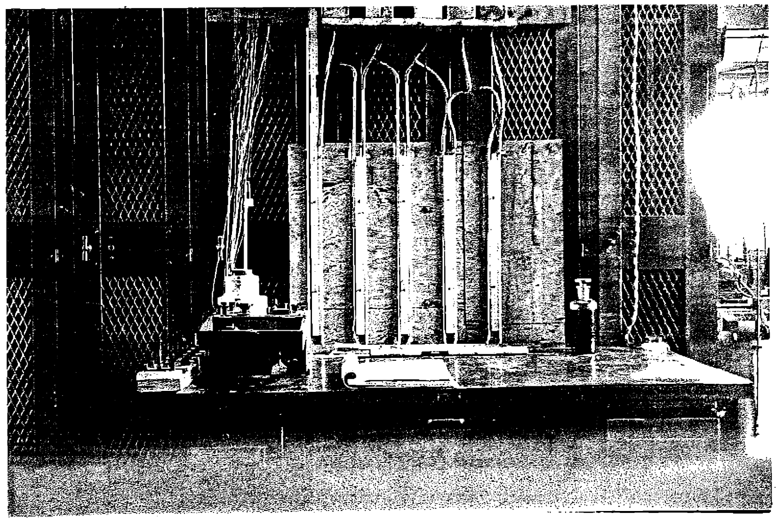


Illustration I. Method of collecting steam condensate. Condensate drain lines insulated.

Picture taken by L. C. Heroman



<u>Illustration J.</u> View of the manometers for measuring the pressure drop across each pass. The cubical boxes overhead are seal-cups, providing a benzene-water interface at substantially constant elevation. These seal cups were dispensed with during the water runs. Thermocouple switches, potentiometer, and cold junction are visible on the left. The bottle contains octyl thiocyanate.

Picture by R. L. Bryan.

RESULTS AND DISCUSSION

Section I. - Heat Transfer to Boiling Water

Results. Section I. - <u>Heat Transfer to Boiling Water</u>

The operating conditions for the runs on water are summarized in Table I. The symbols listed in Table I are used in all the subsequent correlations of the data on boiling water - specifically, Figures 29-32, 43, and 46.

The results of the runs on water are presented graphically in Figures 1 - 14, as plots of static fluid pressure (P), cm. Hg. ga.; condensing steam temperature (T_{st}), °C.; fluid temperature (T_{fl}), °C.; heat flux (q/A), B.t.u./(hr.)(sq.ft.); overall heat transfer coefficient (U), B.t.u./(hr.)(sq.ft.)(°F.); and the cumulative weight per cent of the feed vaporized (p); versus tube length, expressed in terms of jacket numbers.

The data for the boiling section of Figures 1 - 14 are replotted in Figures 15 - 28, showing the fluid velocity (V), ft./sec.; overall heat transfer coefficient (U), B.t.u./(hr.)(sq.ft.)(°F.); heat flux (q/A), B.t.u./(hr.)(sq.ft.); overall temperature difference (Δt), °C.; and cumulative pressure drop from the feed inlet (ΔP), cm. Hg.; plotted on a logarithmic scale as a function of the cumulative weight per cent of the feed vaporized (p). The abscissa scale in Figures 15 - 28 varies from run to run.

Figure 29 is a logarithmic plot of the overall heat transfer coefficient (U) versus the overall temperature difference (Δt), °C. Straight lines are drawn approximately through points corresponding to data taken at substantially equal steam pressures.

Figure 30 is a plot of the lograithm of the heat transfer coefficient (U) versus the cumulative weight per cent of the feed vaporized (p). The line drawn through the points divides into two branches at low values of p, corresponding to data taken at different steam pressures. The dotted lines in Figure 30 are transposed from Figure 30 (a), which is a semilogarithmic plot of film heat transfer coefficients (h) versus cumulative weight per cent of the feed vaporized (p), as obtained inside of an horizontal electrically-heated stainless steel tube of small diameter [12] but operated with entering velocities comparable to those used in the semi-commercial apparatus.

The data of Figure 30 are replotted in Figure 31 on logarithmic paper.

Figure 32 is a logarithmic plot of the overall heat transfer coefficient (U) versus the fluid velocity (V). At low velocities the curve through the points divides into two branches according to the steam pressures used in obtaining the data; at high velocities the curve divides into two branches which represent lines of constant feed rate (W), lbs./hr.

Figure 33 is a plot of the average overall coefficient $(U)_{av}$ for the entire boiling section of the water runs, based upon and plotted against the logarithmic mean of the overall temperature differences existing at the beginning and end of the boiling section. The broken lines in Figure 33 are transposed from Figure 33 (a), which is a plot of the apparent overall heat transfer coefficient (U) versus the apparent overall temperature difference (Δt), as obtained in a short vertical steam-heated nickel tube of 0.495-in. I.D.[10]

The pictures following Figure 33 are high speed photographs*

*Taken by Prof. H.E.Edgerton, using stroboscopic light. Exposure = 1/100,000 seconds.

of the U - bends at one end of the apparatus taken during a special run on water at the conclusion of the data runs. The left-hand U - bend is located at the end of the first pass through the apparatus, while the right-hand U - bend is located at the end of the third pass. Illustrations K and L were taken a few seconds apart at a time when the feed rate was 530 lbs./hr. and the calculated cumulative weight per cent of the feed vaporized was 8% and 47% at the end of the first and third passes, respectively. The steam valve was then opened and Illustration O was taken after the steam pressure had risen from 20 to 80 lbs./sq.in.ga., the feed rate decreasing somewhat because of the increased pressure drop through the apparatus.

Table I

Runs on Boiling Water

(A = 0.88 sq.ft./jacket)

Symbol	Run No.	Feed rate lbs./hr.	Feed Water	Steam Pressure lbs./sq.in.ga.	Number Heated Jackets	Initial Feed temp.($^{\circ}$C.)	Heat Transfer (B.t.u./hr.)*	Wt. % of feed vaporized
◇	W 5	984	Rusty	23.4	12	88.3	423,000	41
△	W 7	1022	Clean	21.7	12	81.5	426,000	39
□	W 9	1110	Rusty	21.2	9	87.0	376,000	32
▽	W 10	935	Clean	22.1	6	85.9	308,000	31
◆	W 3	1037	Rusty	11.4	12	88.2	227,000	20
×	W 15	626	Clean	(70)	6	82.8	(616,000)	(98)
+	W 13	696	Clean	(64)	6	74.4	552,000	77
◐	W 11	672	Clean	35.1	6	79.3	465,000	67
▼	W 2	690	Rusty	10.5	12	80.8	244,000	33
▲	W 6	685	Clean	10.3	12	78.5	245,000	33
■	W 8	674	Rusty	10.1	9	82.4	200,000	27
✱	W 12	348	Clean	(79)	6	63.1	358,000	99
○	W 4	425	Rusty	20.8	12	66.7	417,000	95
●	W 1	372	Rusty	9.7	12	65.5	273,000	69

* Based on steam condensate measurements

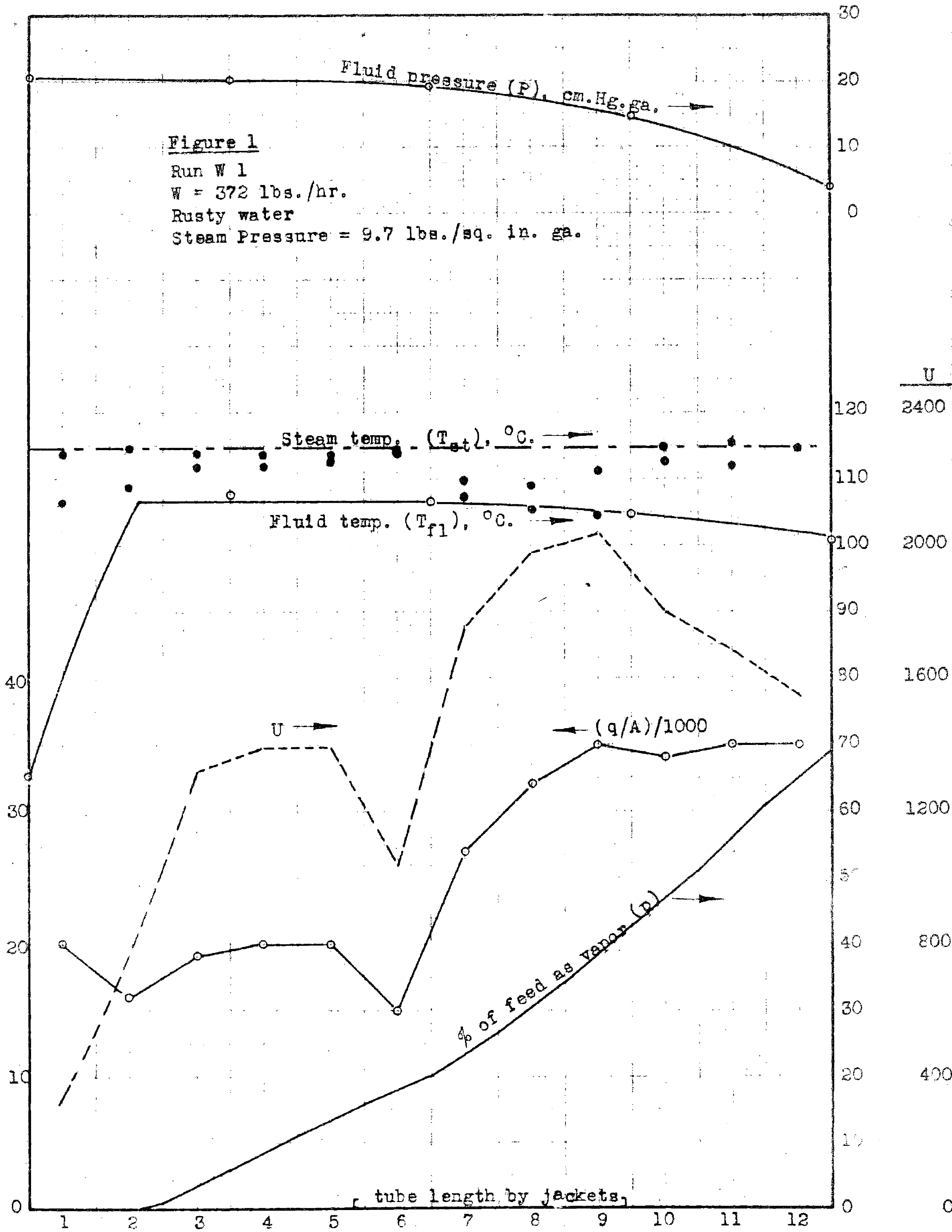


Figure 1
Run W 1
W = 372 lbs./hr.
Rusty water
Steam Pressure = 9.7 lbs./sq. in. ga.
Fluid pressure (P), cm.Hg.ga.
Steam temp. (T_{st}), °C.
Fluid temp. (T_{fl}), °C.
U
(q/A)/1000
% of feed as vapor (p)
tube length by jackets
U
0
10
20
30
0
10
20
30
40
50
60
70
80
90
100
110
120
0
400
800
1200
1600
2000
2400
1
2
3
4
5
6
7
8
9
10
11
12

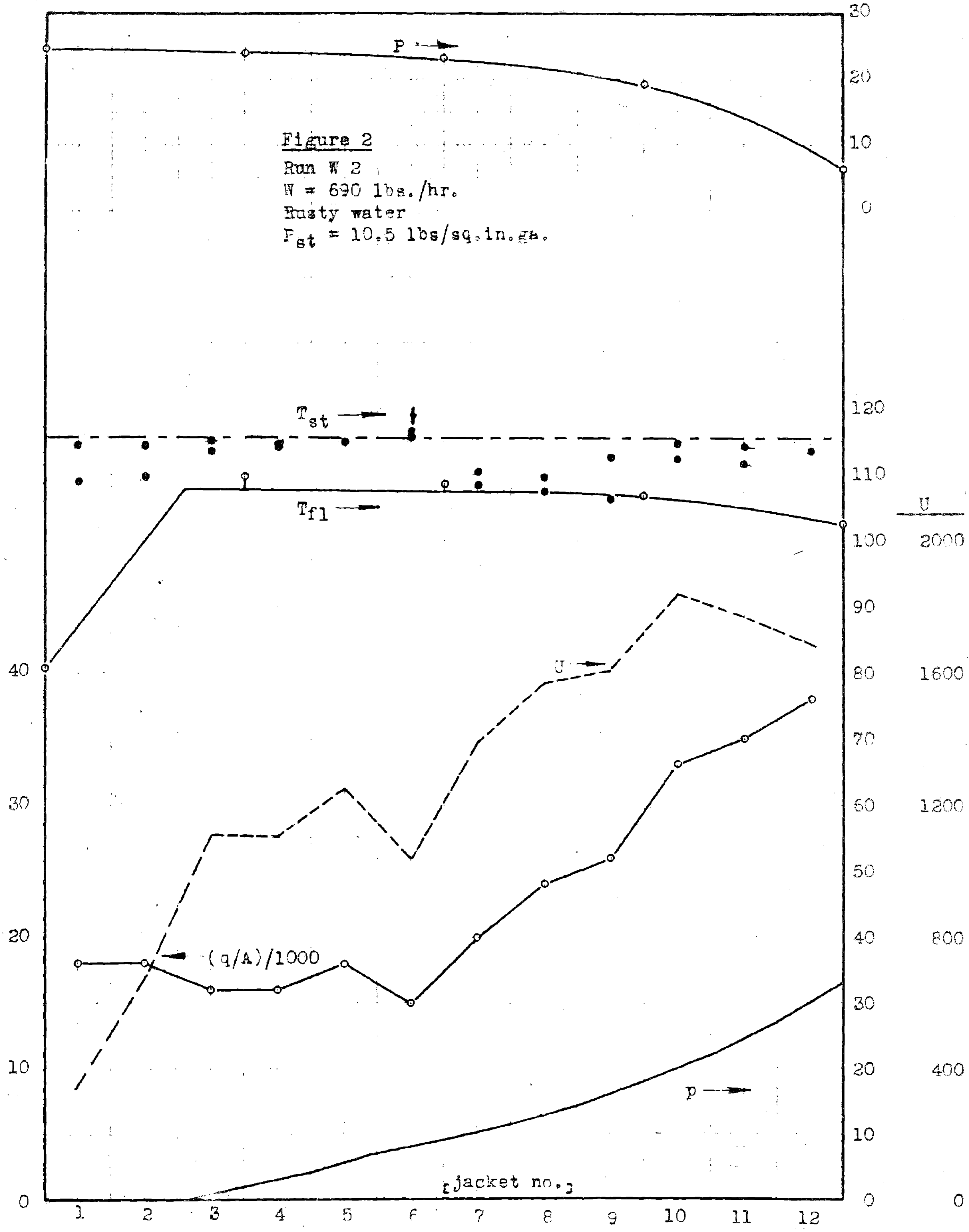


Figure 2
Run W 2
W = 690 lbs./hr.
Rusty water
P_st = 10.5 lbs/sq.in.ga.
p
T_st
T_fl
U
(q/A)/1000
p
jacket no.
1 2 3 4 5 6 7 8 9 10 11 12
0 10 20 30 40
0 10 20 30 40 50 60 70 80 90 100 110 120
0 10 20 30
U
0 400 800 1200 1600 2000

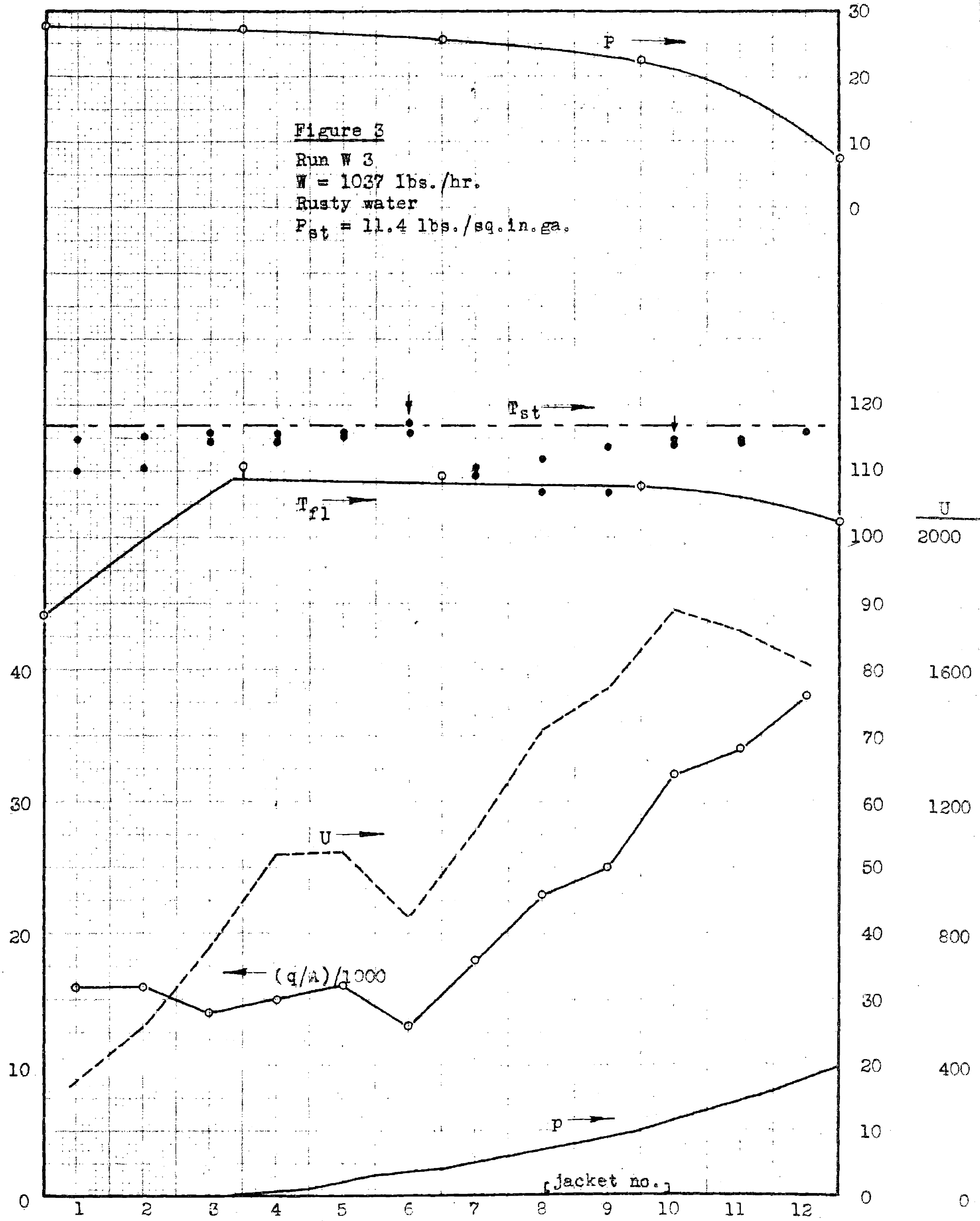


Figure 3
Run W 3
W = 1037 lbs./hr.
Rusty water
P st = 11.4 lbs./sq.in.ga.
P
T st
T fl
U
(q/A)/1000
p
jacket no.
U
2000
1600
1200
800
400
0
30
20
10
0
120
110
100
90
80
70
60
50
40
30
20
10
0
40
30
20
10
0
1
2
3
4
5
6
7
8
9
10
11
12

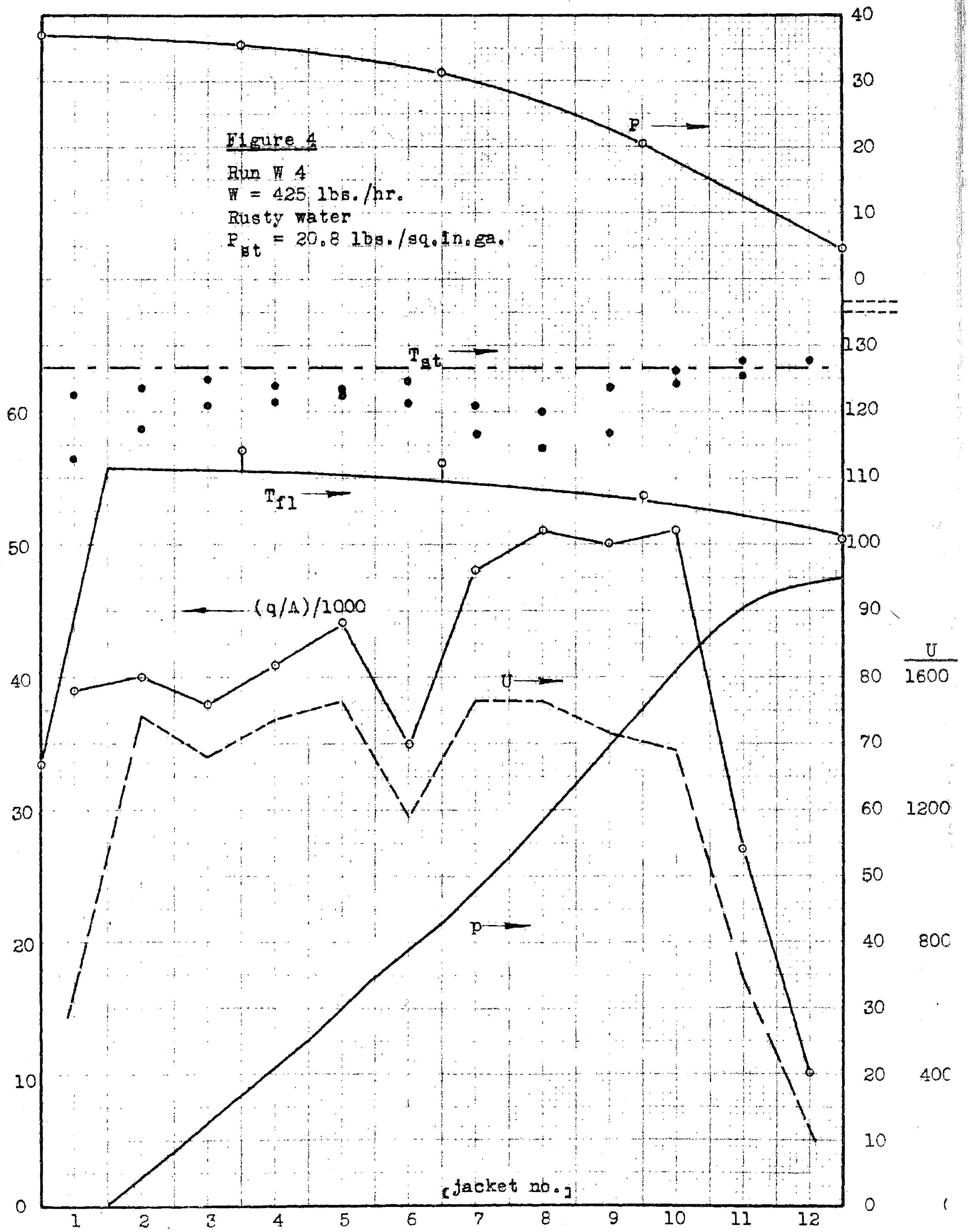


Figure 4
Run W 4
W = 425 lbs./hr.
Rusty water
P st = 20.8 lbs./sq.in.ga.
P
T st
T fl
(q/A)/1000
U
p
jacket no.
1 2 3 4 5 6 7 8 9 10 11 12
0 10 20 30 40 50 60
40 30 20 10 0
130 120 110 100 90 80 70 60 50 40 30 20 10 0
U
1600 1200 800 400

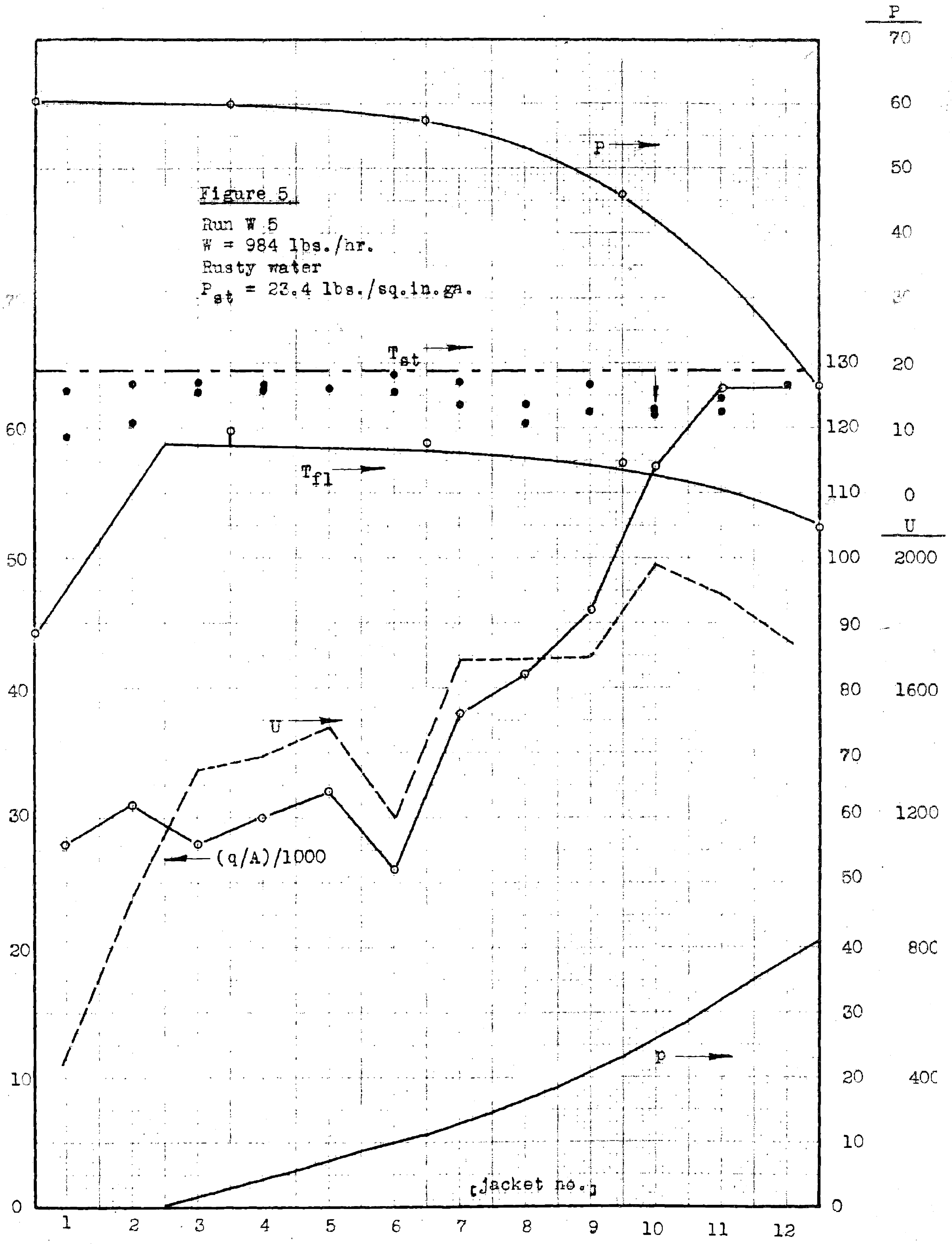


Figure 5
Run W 5
W = 984 lbs./hr.
Rusty water
P_{st} = 23.4 lbs./sq.in.ga.
P
T_{st}
T_{fl}
U
(q/A)/1000
p
Jacket no.
0
10
20
30
40
50
60
1
2
3
4
5
6
7
8
9
10
11
12
100
110
120
130
70
2000
1600
1200
800
400

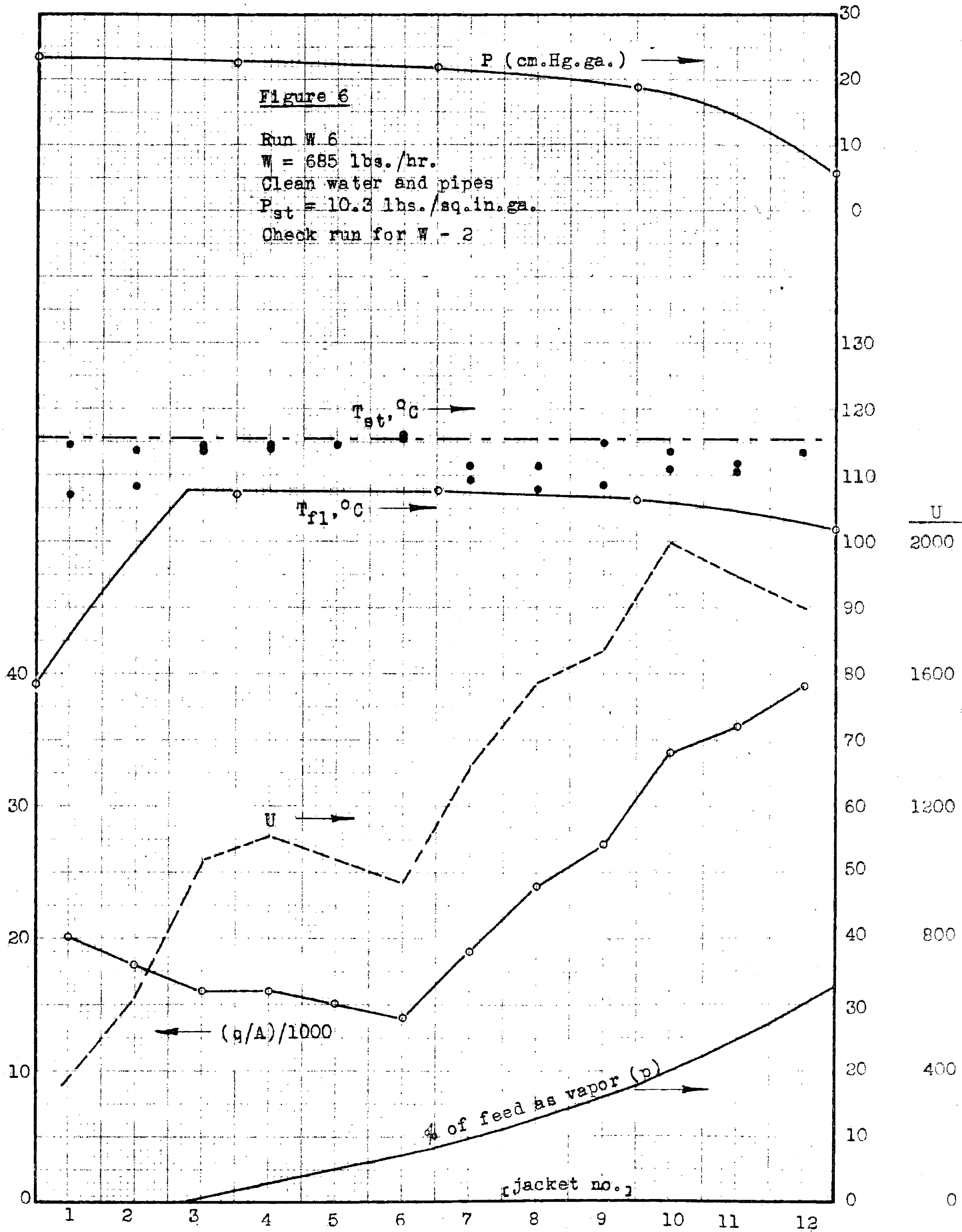


Figure 6
Run W 6
W = 685 lbs./hr.
Clean water and pipes
P_st = 10.3 lbs./sq.in.ga.
Check run for W - 2
P (cm.Hg.ga.)
T_st, °C
T_fl, °C
U
(q/A)/1000
% of feed as vapor (p)
[Jacket no.]
30
20
10
0
130
120
110
100
90
80
70
60
50
40
30
20
10
0
2000
1600
1200
800
400
0
40
30
20
10
0
1
2
3
4
5
6
7
8
9
10
11
12

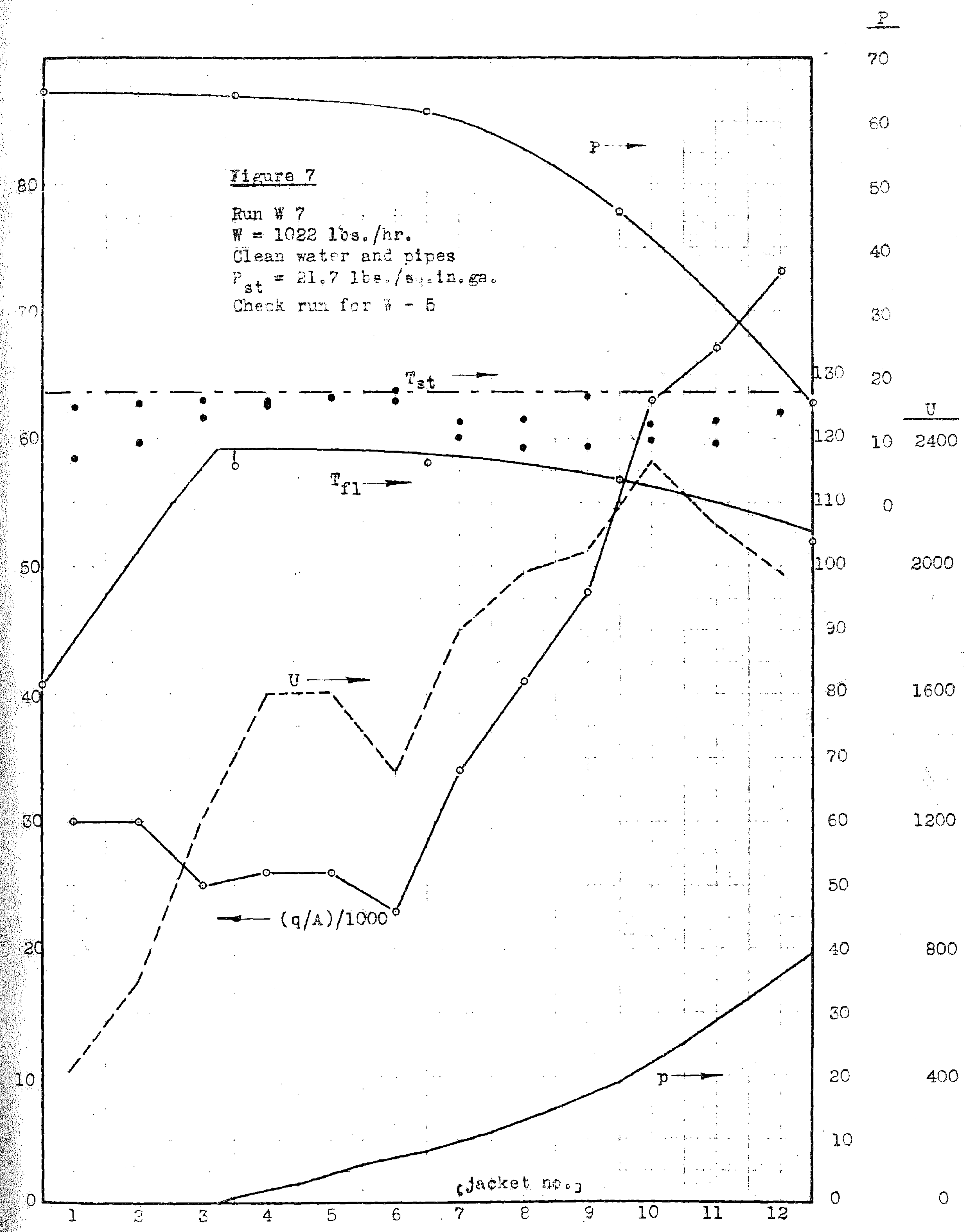


Figure 7
Run W 7
W = 1022 lbs./hr.
Clean water and pipes
P_{st} = 21.7 lbs./sq.in.ga.
Check run for W - 5
P
T_{st}
T_{fl}
U
(q/A)/1000
p
Jacket no.
P
U
0
10
20
30
40
50
60
70
80
1
2
3
4
5
6
7
8
9
10
11
12
100
110
120
130
0
400
800
1200
1600
2000
2400

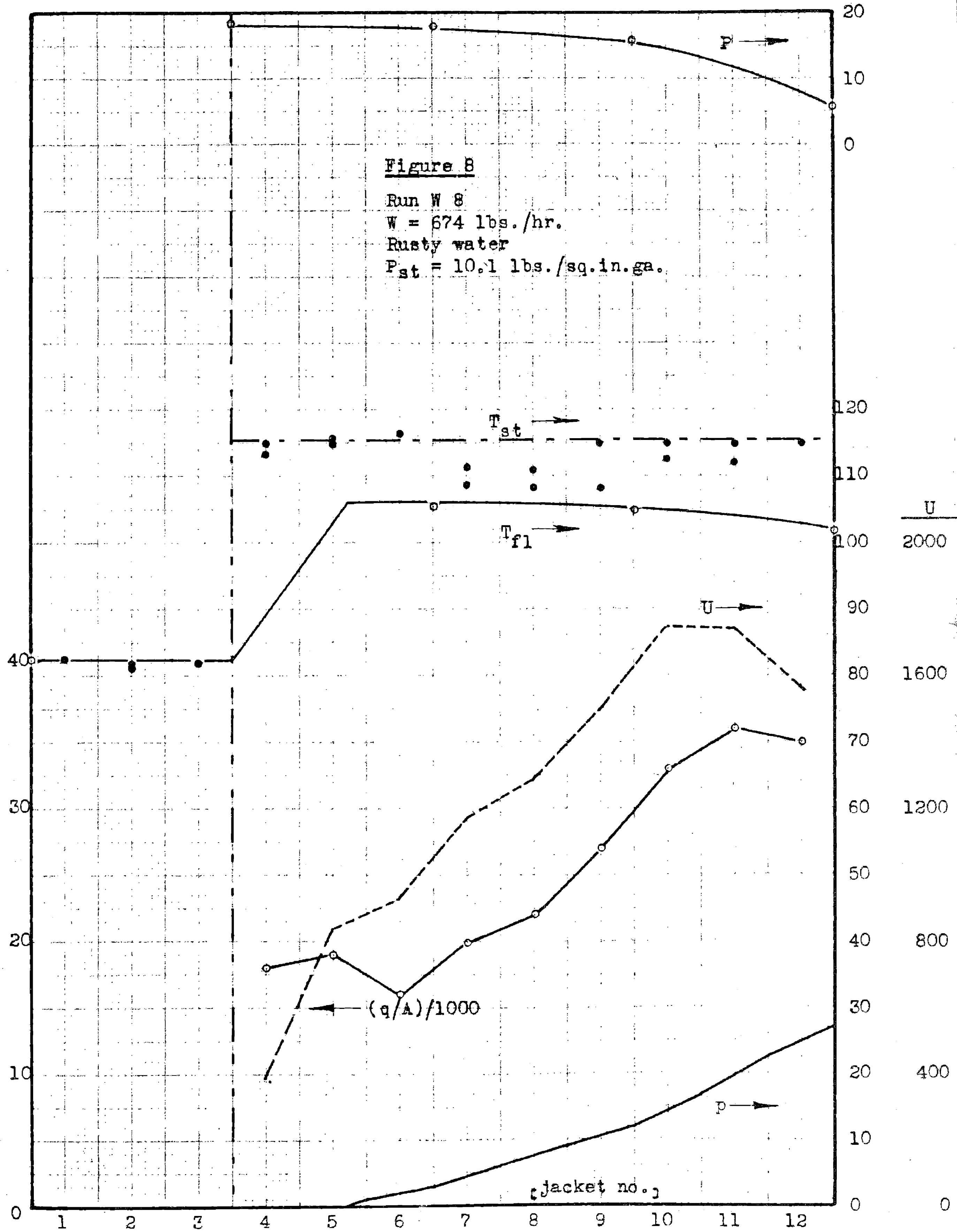


Figure 8
Run W 8
W = 674 lbs./hr.
Rusty water
Pst = 10.1 lbs./sq.in.ga.
P
Tst
Tfl
U
(q/A)/1000
p
jacket no.
U
1 2 3 4 5 6 7 8 9 10 11 12
0 10 20 30 40
0 10 20 30 40 50 60 70 80 90 100 110 120
0 10 20
0 400 800 1200 1600 2000

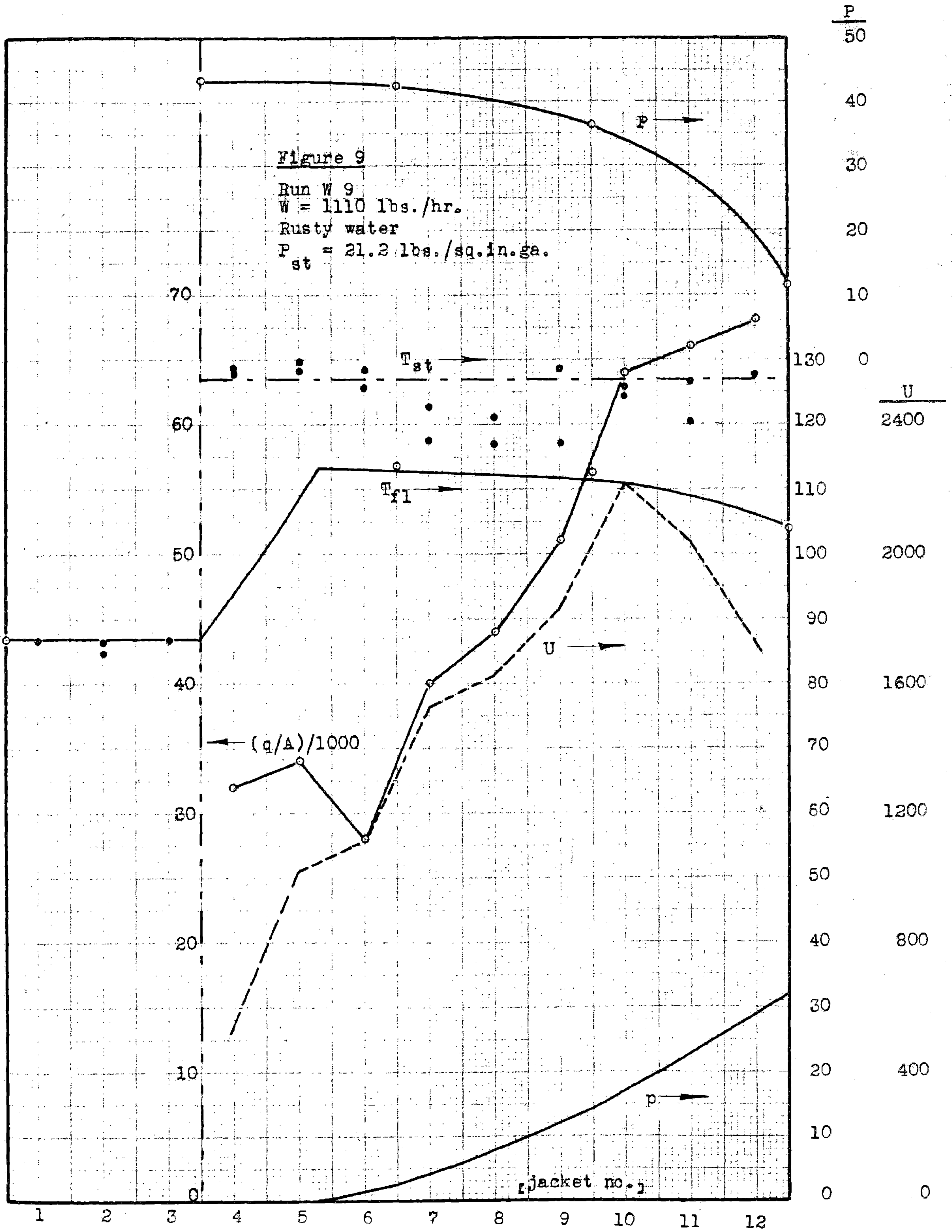


Figure 9
Run W 9
W = 1110 lbs./hr.
Rusty water
P_{st} = 21.2 lbs./sq.in.ga.
P
T_{st}
T_{fl}
U
(q/A)/1000
p
[jacket no.]
P/50
U/2400
0 10 20 30 40 50 60 70
0 10 20 30 40 50 60 70 80 90 100 110 120 130
0 10 20 30 40 50
0 400 800 1200 1600 2000 2400
1 2 3 4 5 6 7 8 9 10 11 12

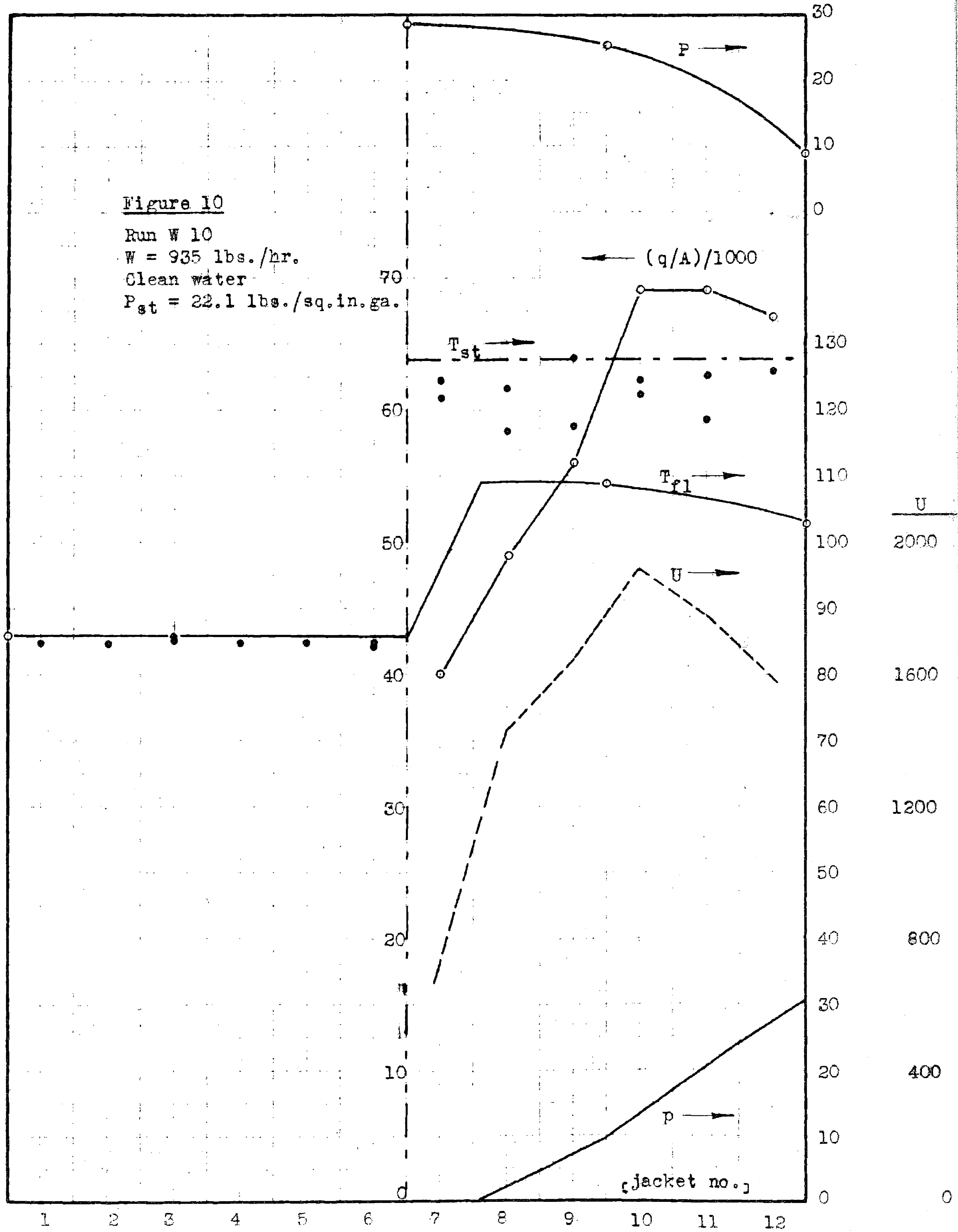


Figure 10
Run W 10
W = 935 lbs./hr.
Clean water
P$_{st}$ = 22.1 lbs./sq.in.ga.
P
(q/A)/1000
T$_{st}$
T$_{fl}$
U
p
U
jacket no.
1 2 3 4 5 6 7 8 9 10 11 12
0 10 20 30
70 60 50 40 30 20 10 0
130 120 110 100 90 80 70 60 50 40 30 20 10 0
2000 1600 1200 800 400 0

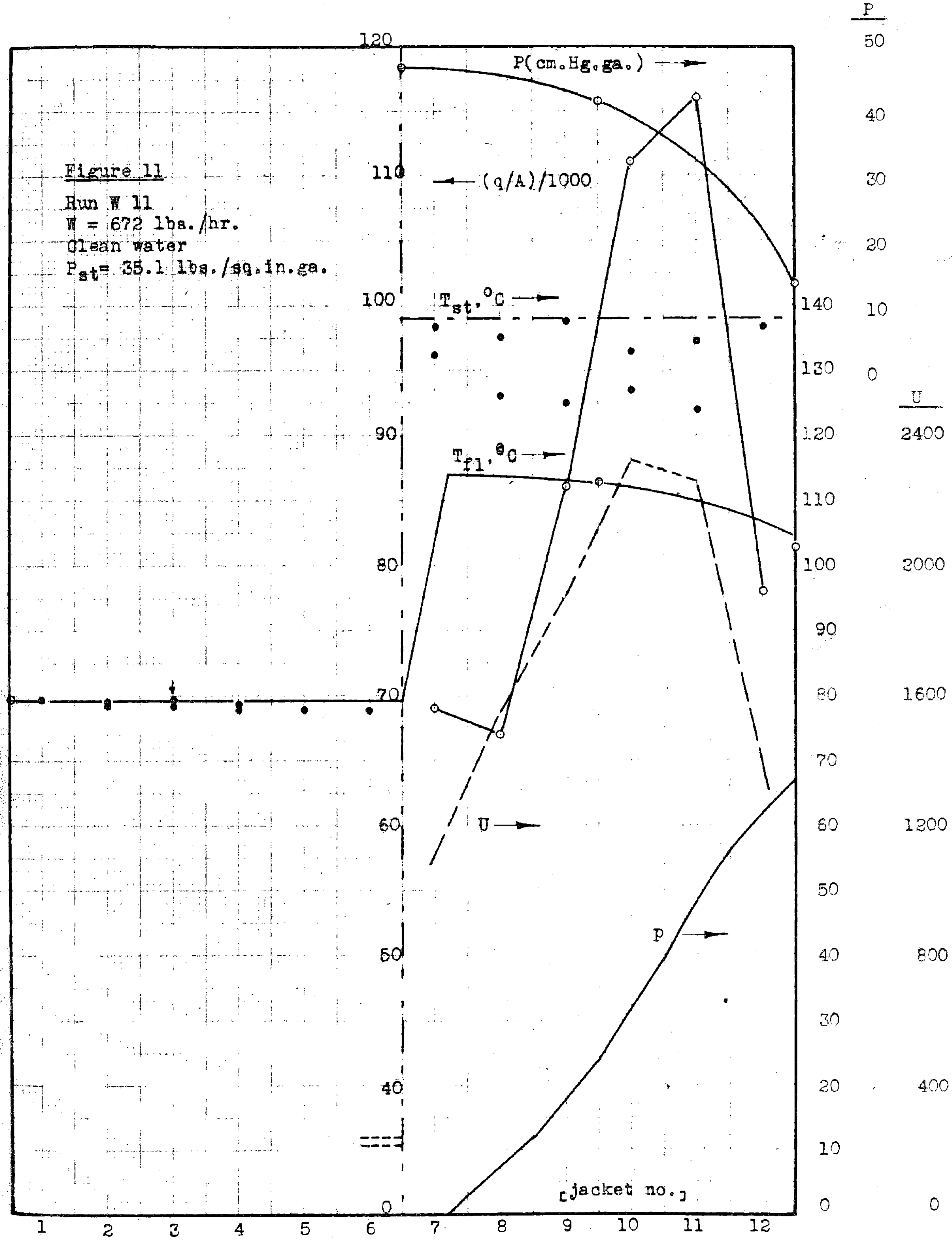


Figure 11
Run W 11
W = 672 lbs./hr.
Clean water
P st= 35.1 lbs./sq.in.ga.
P(cm.Hg.ga.)
(q/A)/1000
T st, °C
T fl, °C
U
P
[jacket no.]
P
U
120
110
100
90
80
70
60
50
40
0
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
50
40
30
20
10
0
2400
2000
1600
1200
800
400
0
1
2
3
4
5
6
7
8
9
10
11
12

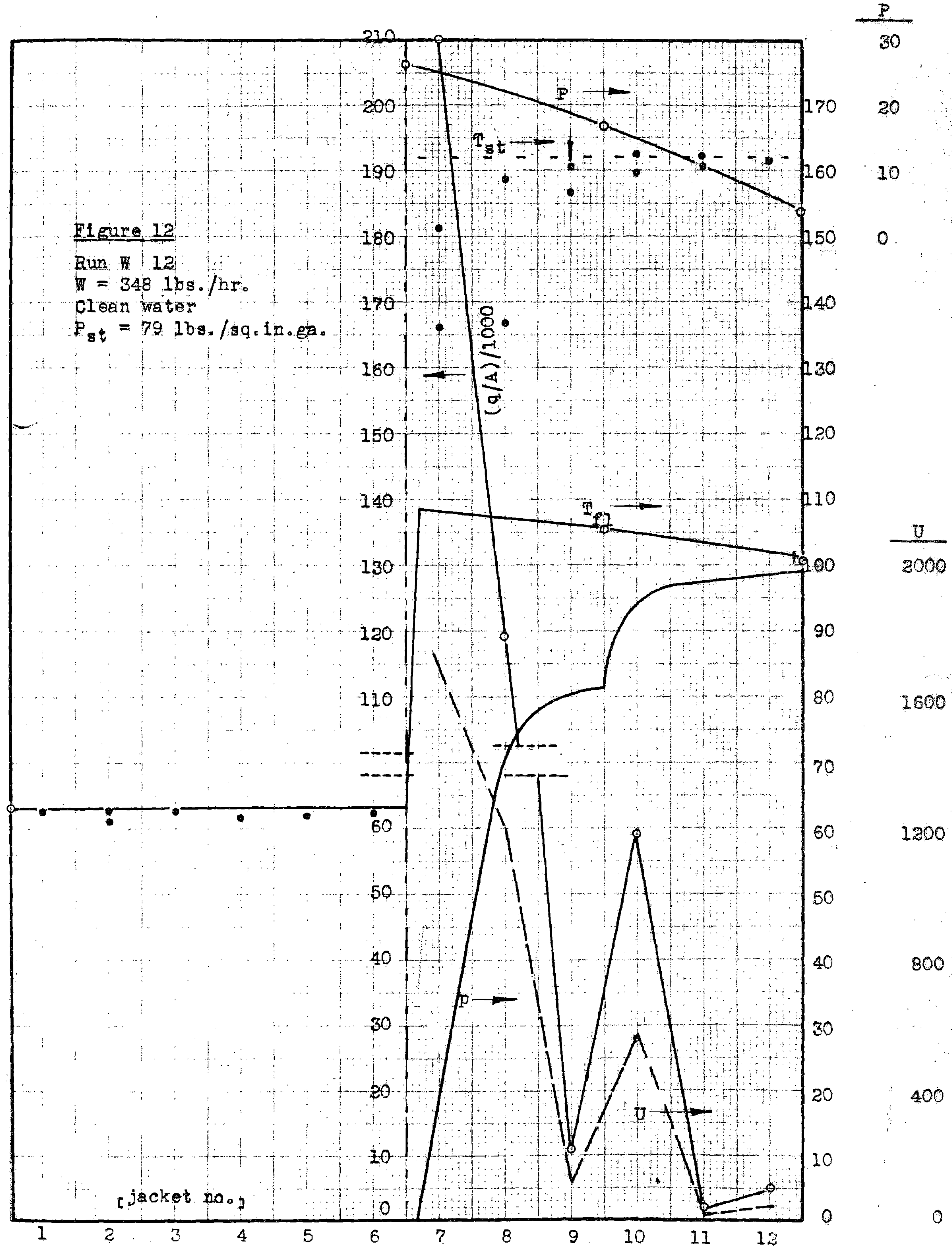


Figure 12
Run W 12
W = 348 lbs./hr.
Clean water
P_{st} = 79 lbs./sq.in.ga.
P
T_{st}
(q/A)/1000
T_f
p
U
jacket no.
1
2
3
4
5
6
7
8
9
10
11
12
210
200
190
180
170
160
150
140
130
120
110
60
50
40
30
20
10
0
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
P
30
20
10
0
U
2000
1600
1200
800
400
0

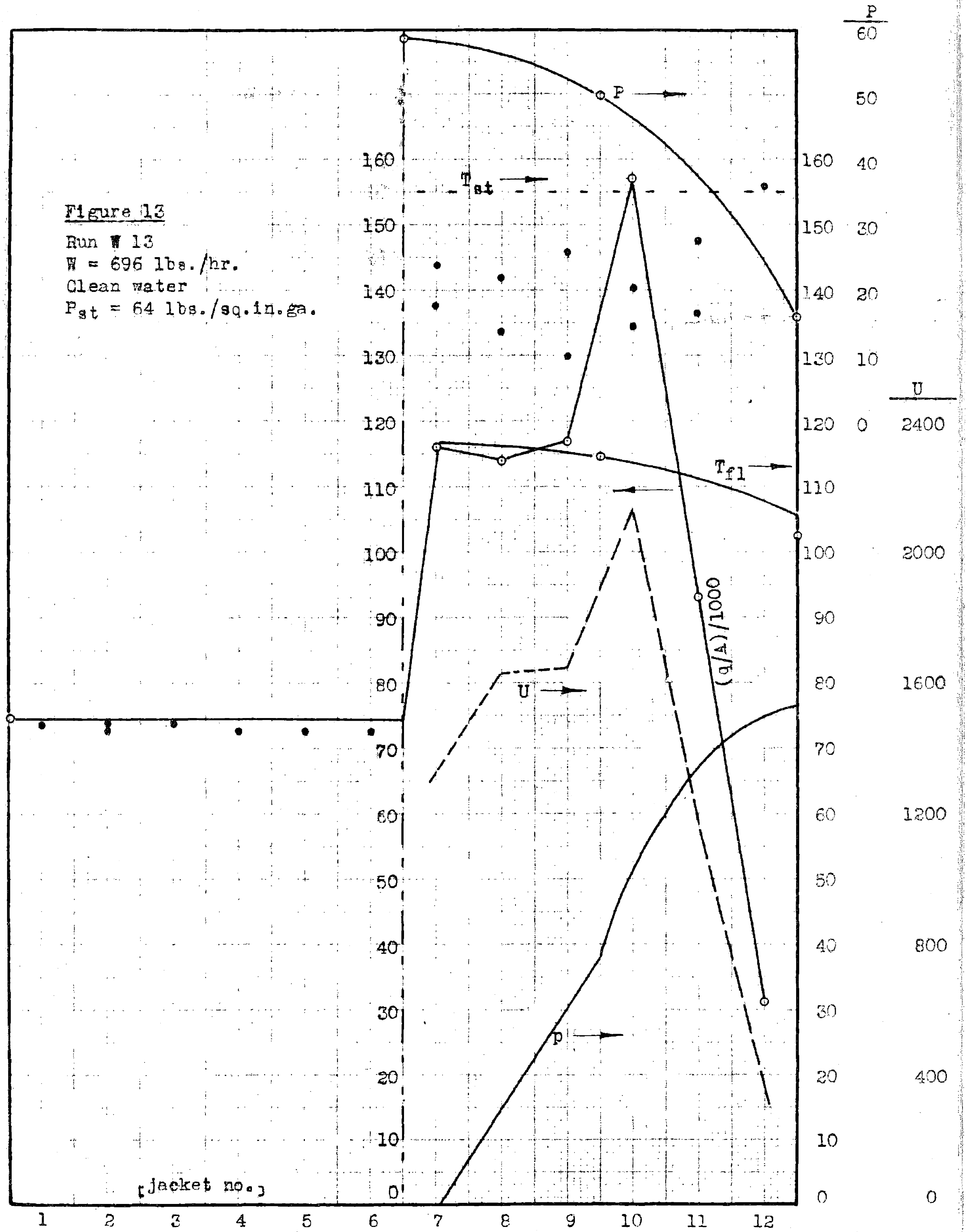


Figure 13
Run W 13
W = 696 lbs./hr.
Clean water
P_{st} = 64 lbs./sq.in.ga.
P
T_{st}
T_{fl}
U
p
(q/A)/1000
P
60
50
40
30
20
10
0
U
2400
2000
1600
1200
800
400
0
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
[jacket no.]
1
2
3
4
5
6
7
8
9
10
11
12

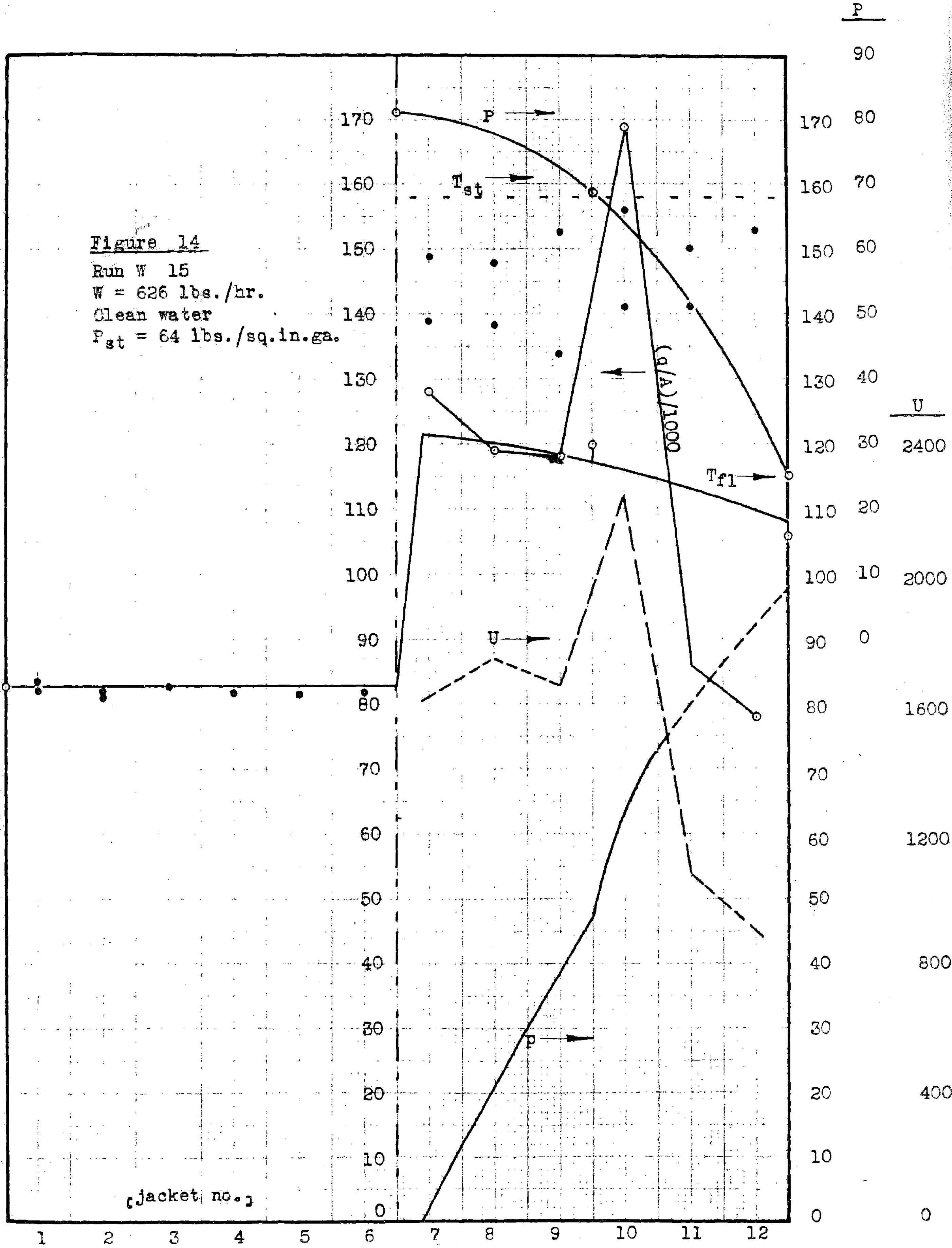


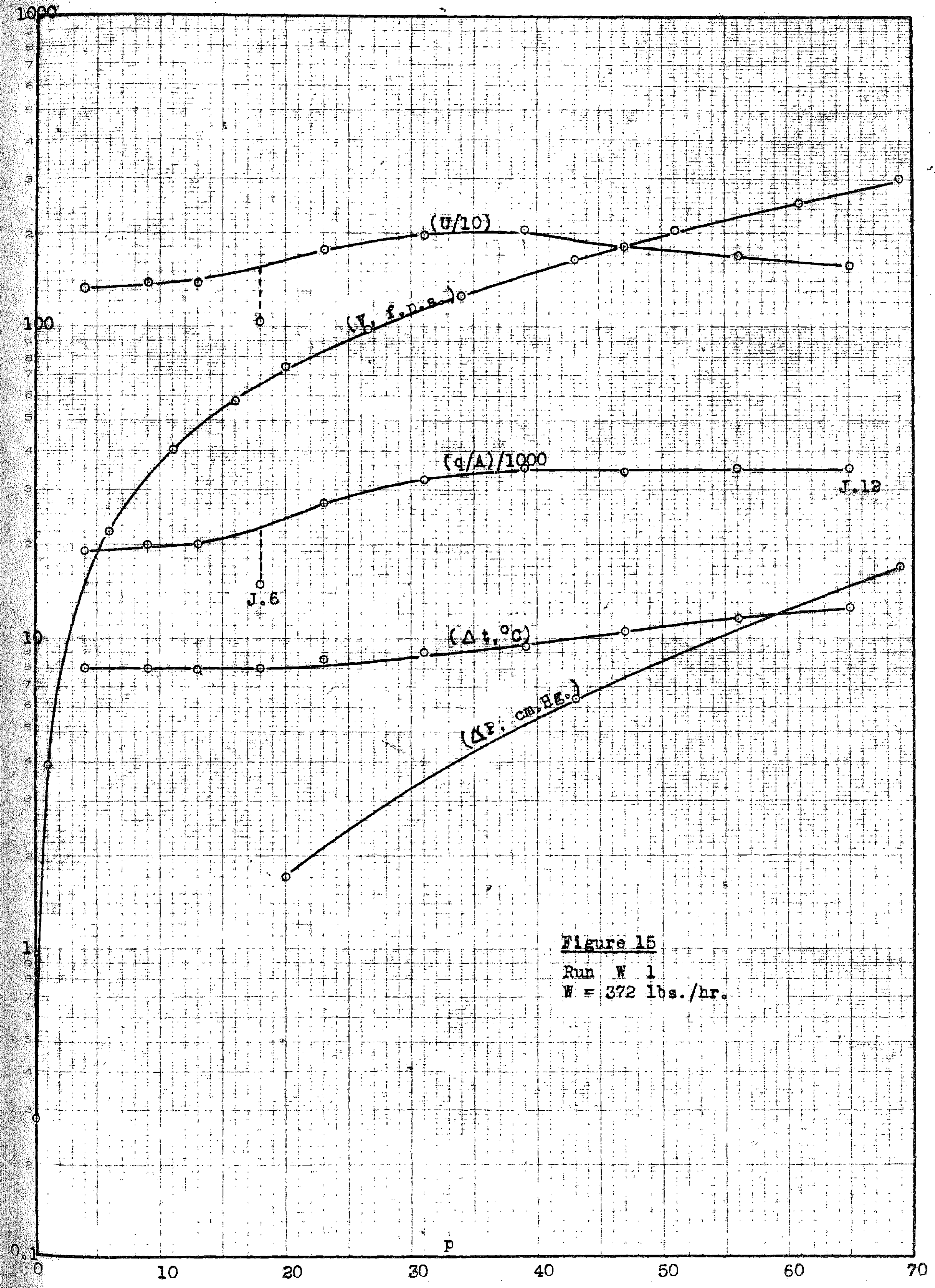
Figure 14

Run W 15

W = 626 lbs./hr.

Clean water

P_{st} = 64 lbs./sq.in.ga.



1000
100
10
1
0.1
0
10
20
30
40
50
60
70
P
(U/10)
(V, f.p.s.)
(q/A)/1000
J.6
J.12
(Δ t, °C)
(ΔP, cm.Hg.)
Figure 15
Run W 1
W = 372 lbs./hr.

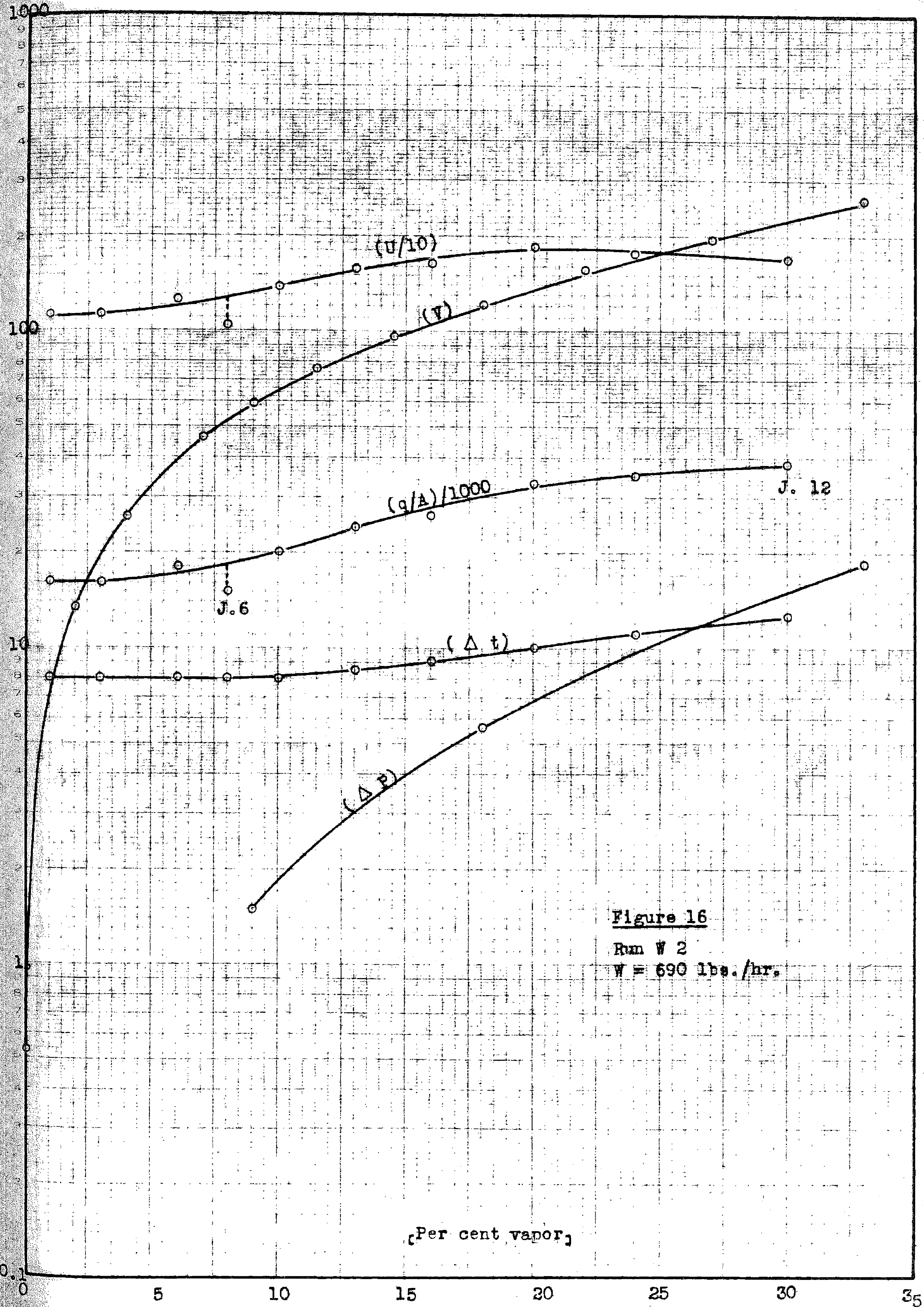


Figure 16

Run W 2

W = 690 lbs./hr.

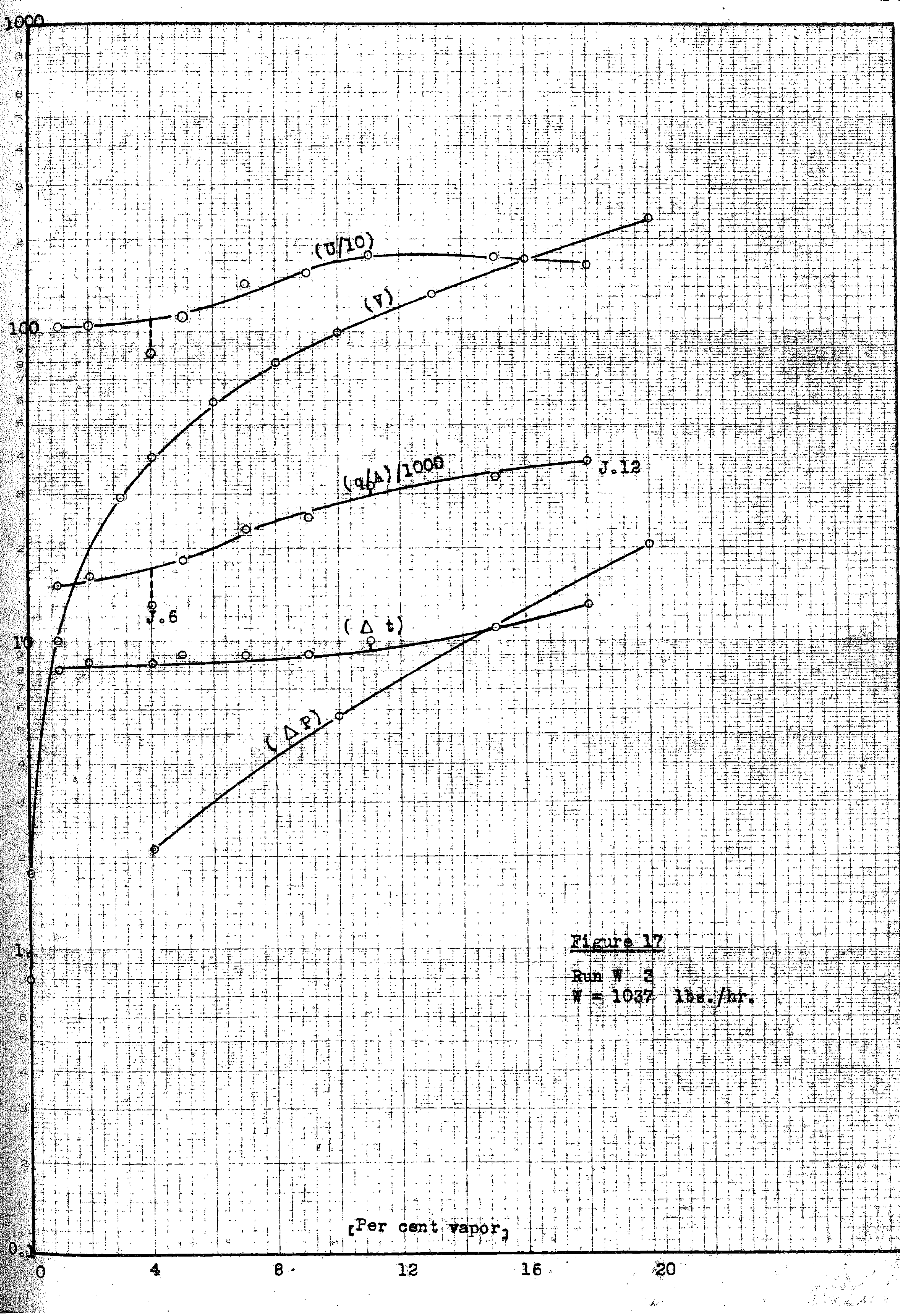


Figure 17

Run W 3

W = 1037 lbs./hr.

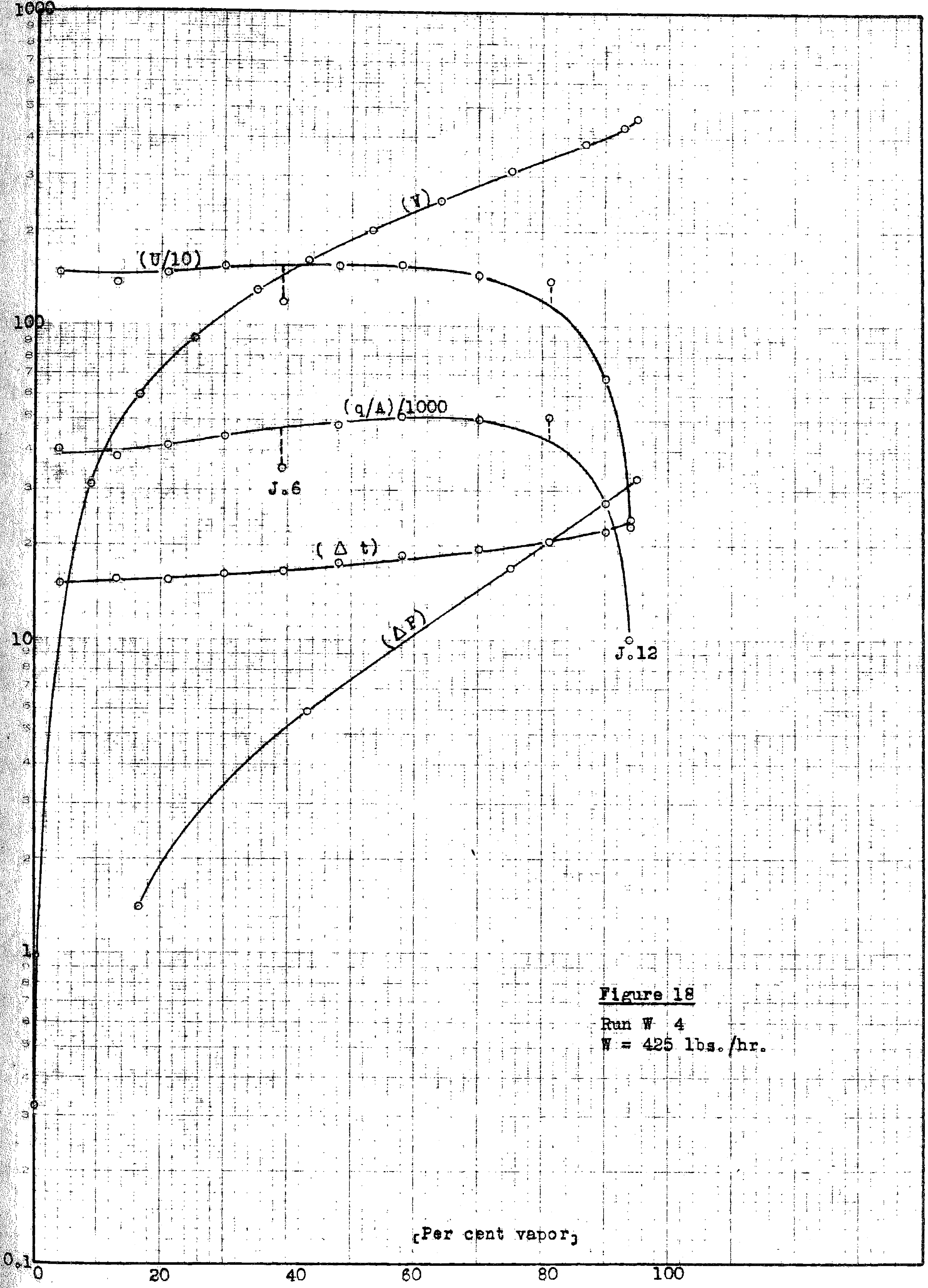


Figure 18

Run W 4
W = 425 lbs./hr.

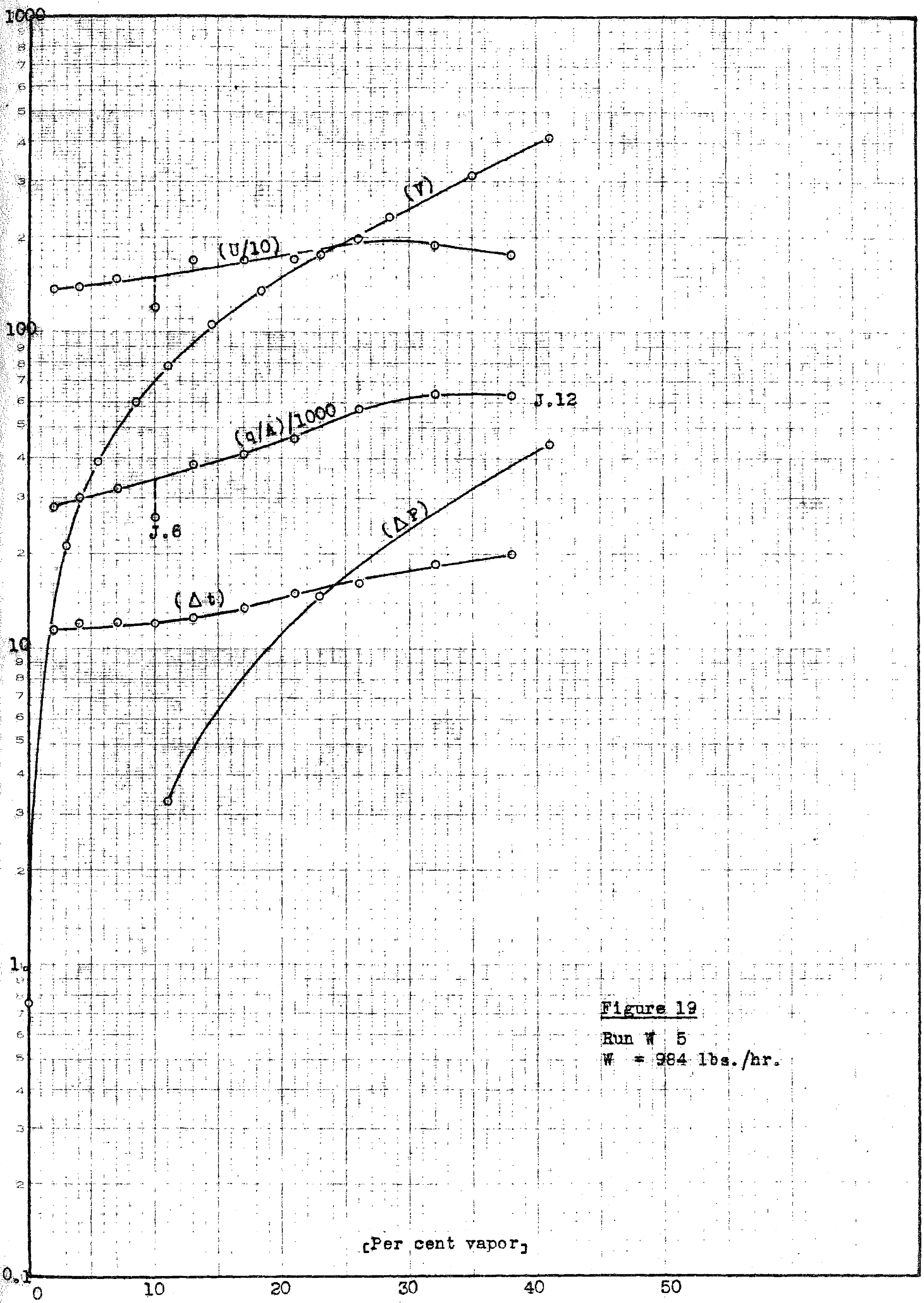


Figure 19

Run W 5
W = 984 lbs./hr.

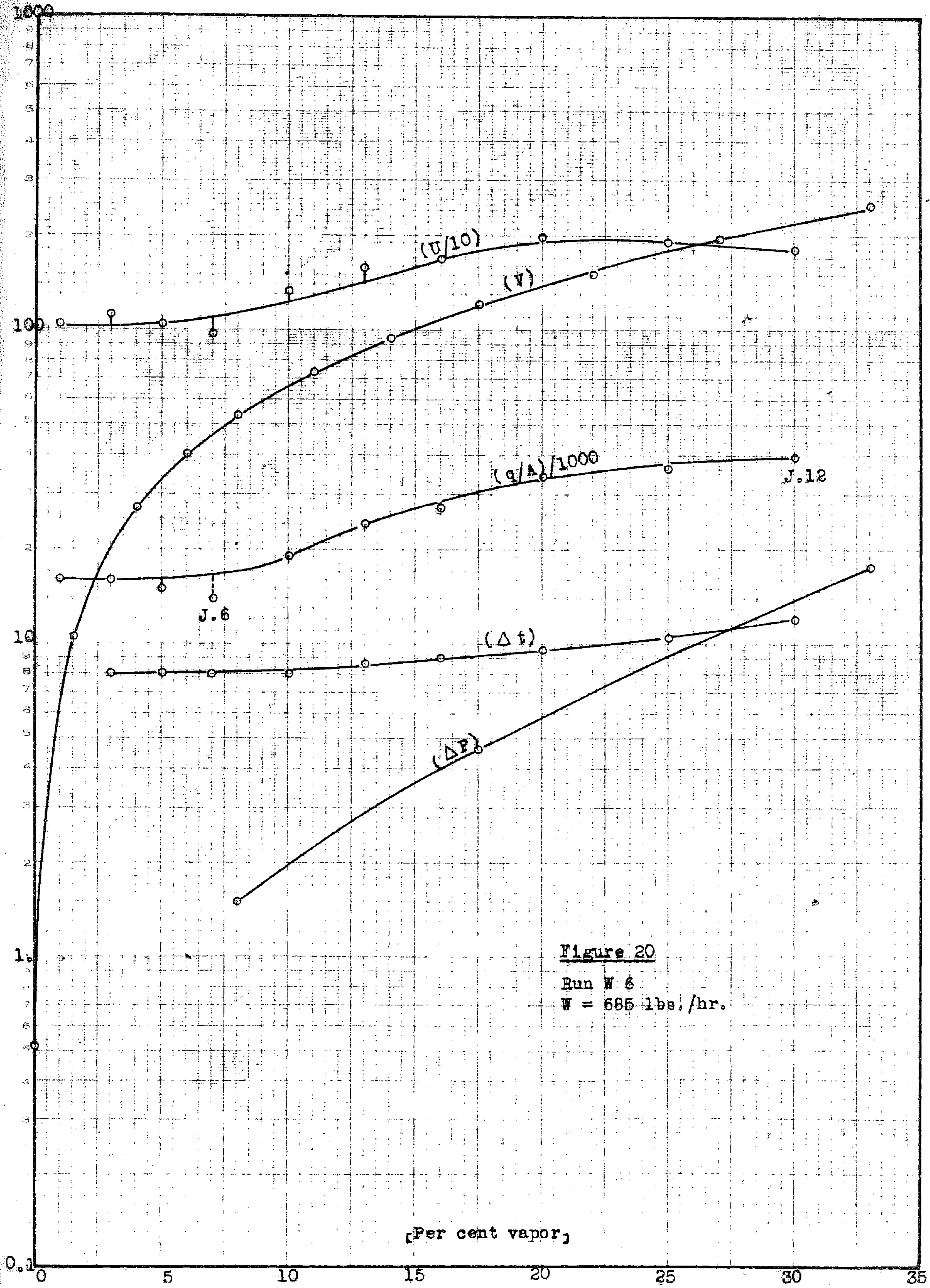


Figure 20

Run W 6
W = 685 lbs./hr.

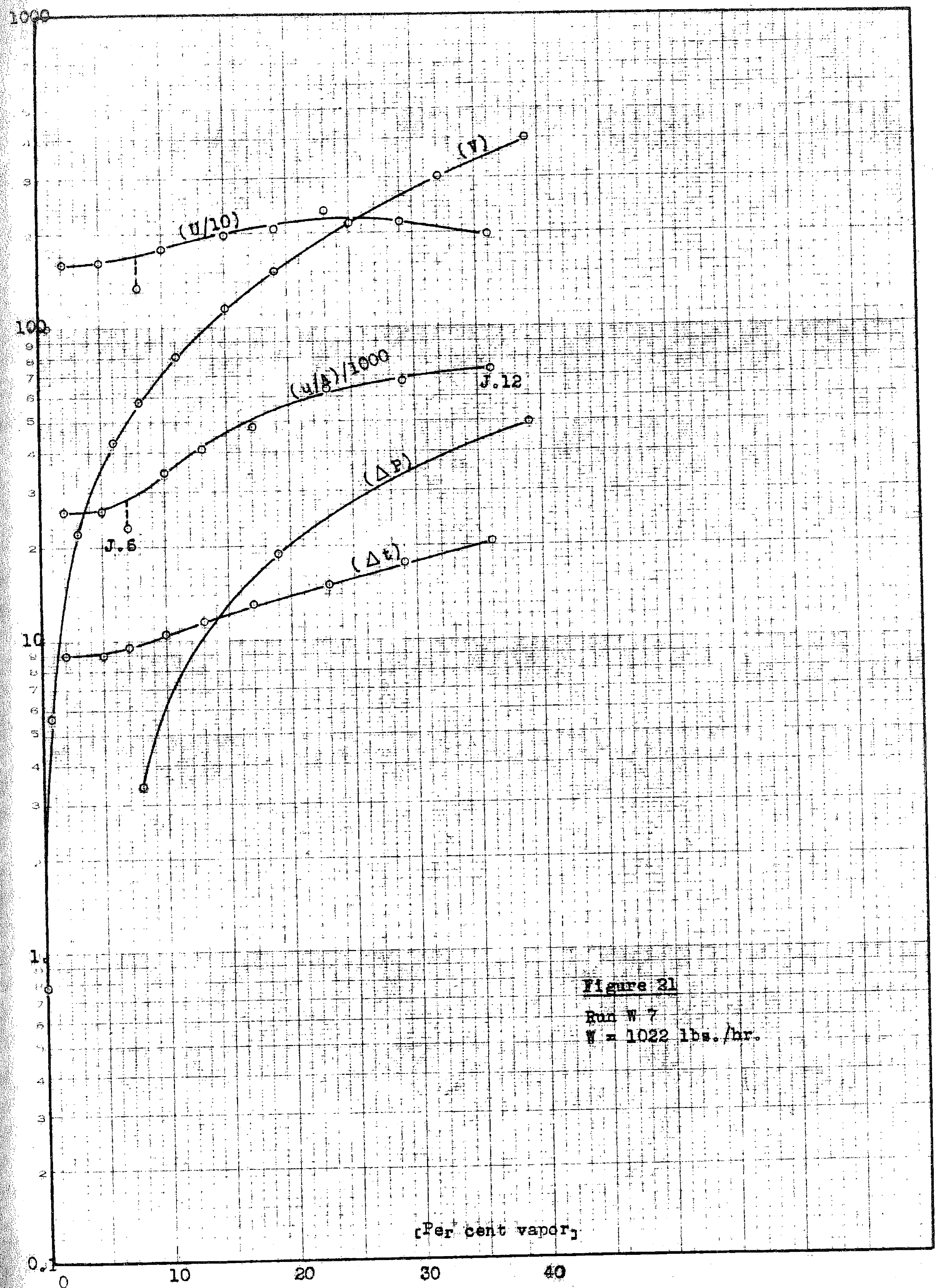


Figure 21

Run W 7

W = 1022 lbs./hr.

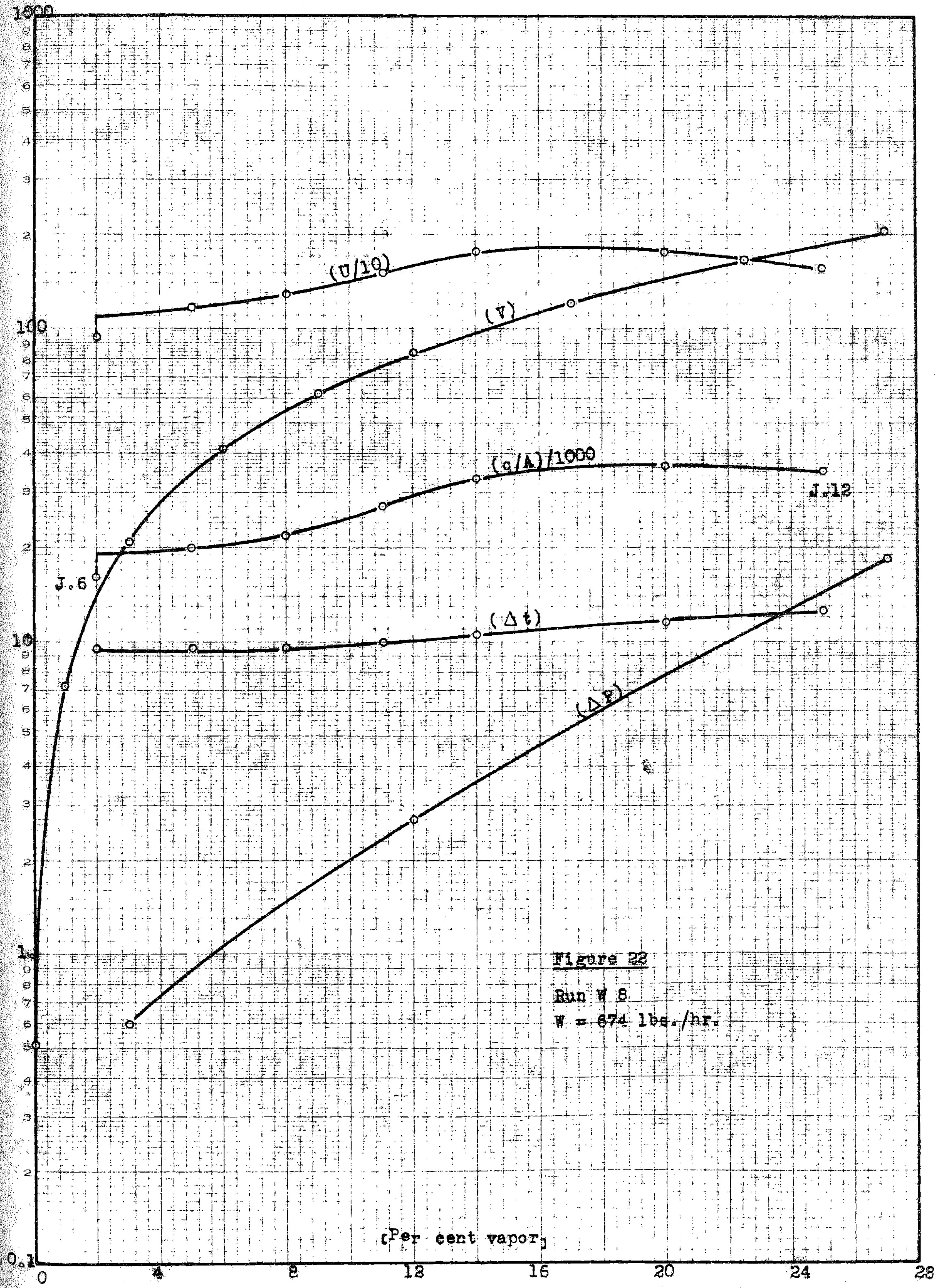


Figure 22

Run W 8

W = 674 lbs./hr.

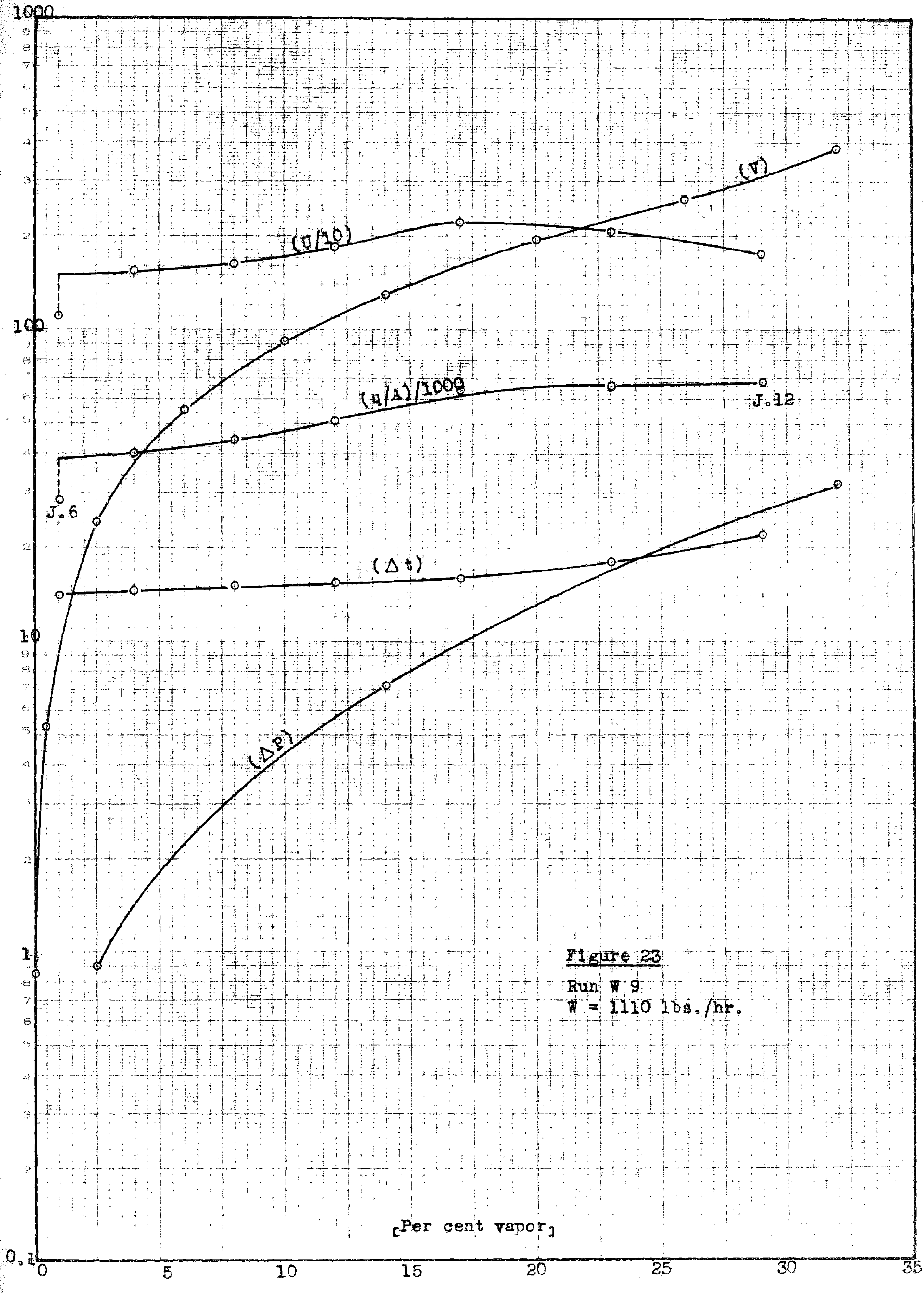


Figure 23

Run W 9
W = 1110 lbs./hr.

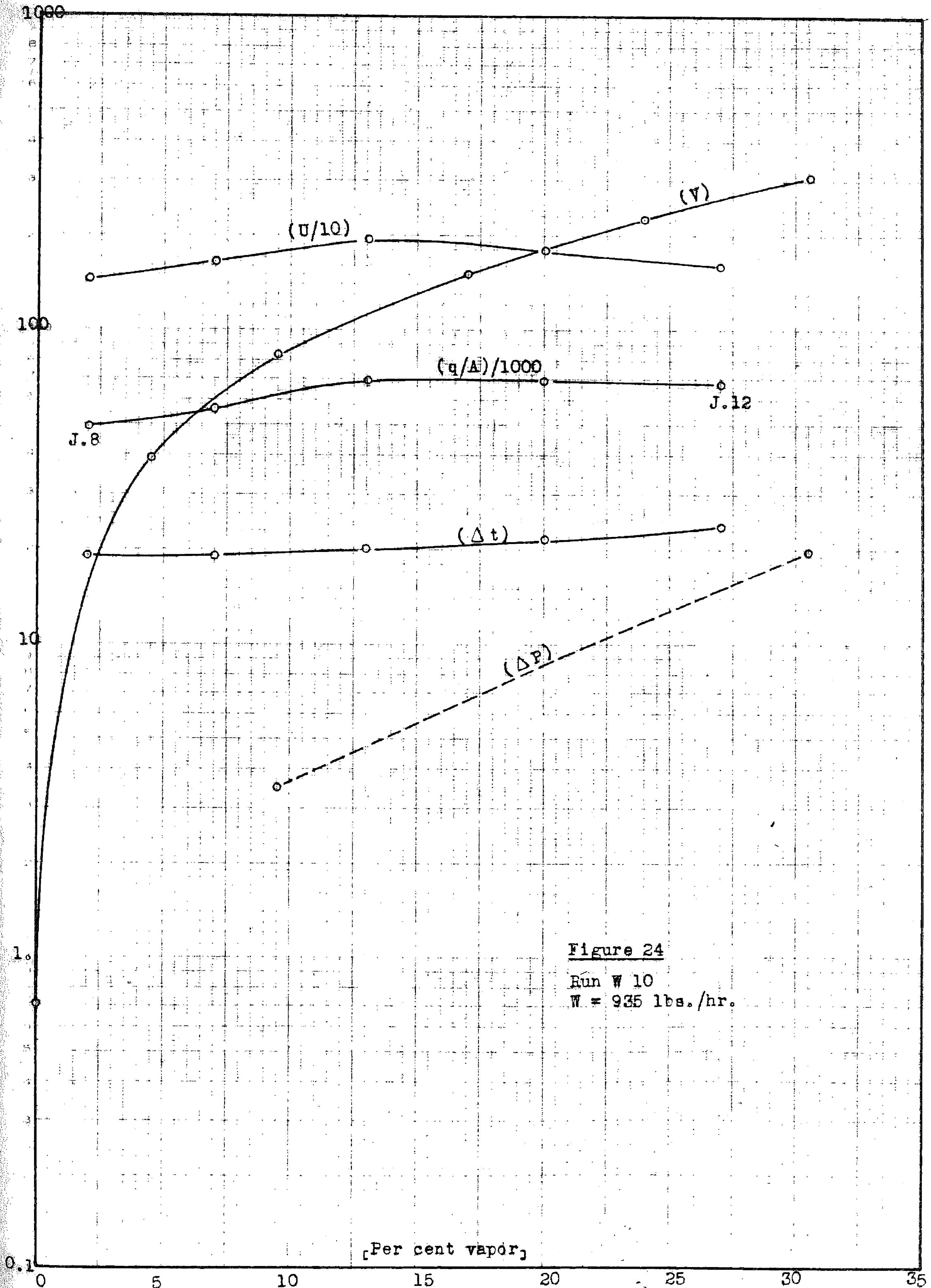


Figure 24

Run W 10

W = 935 lbs./hr.

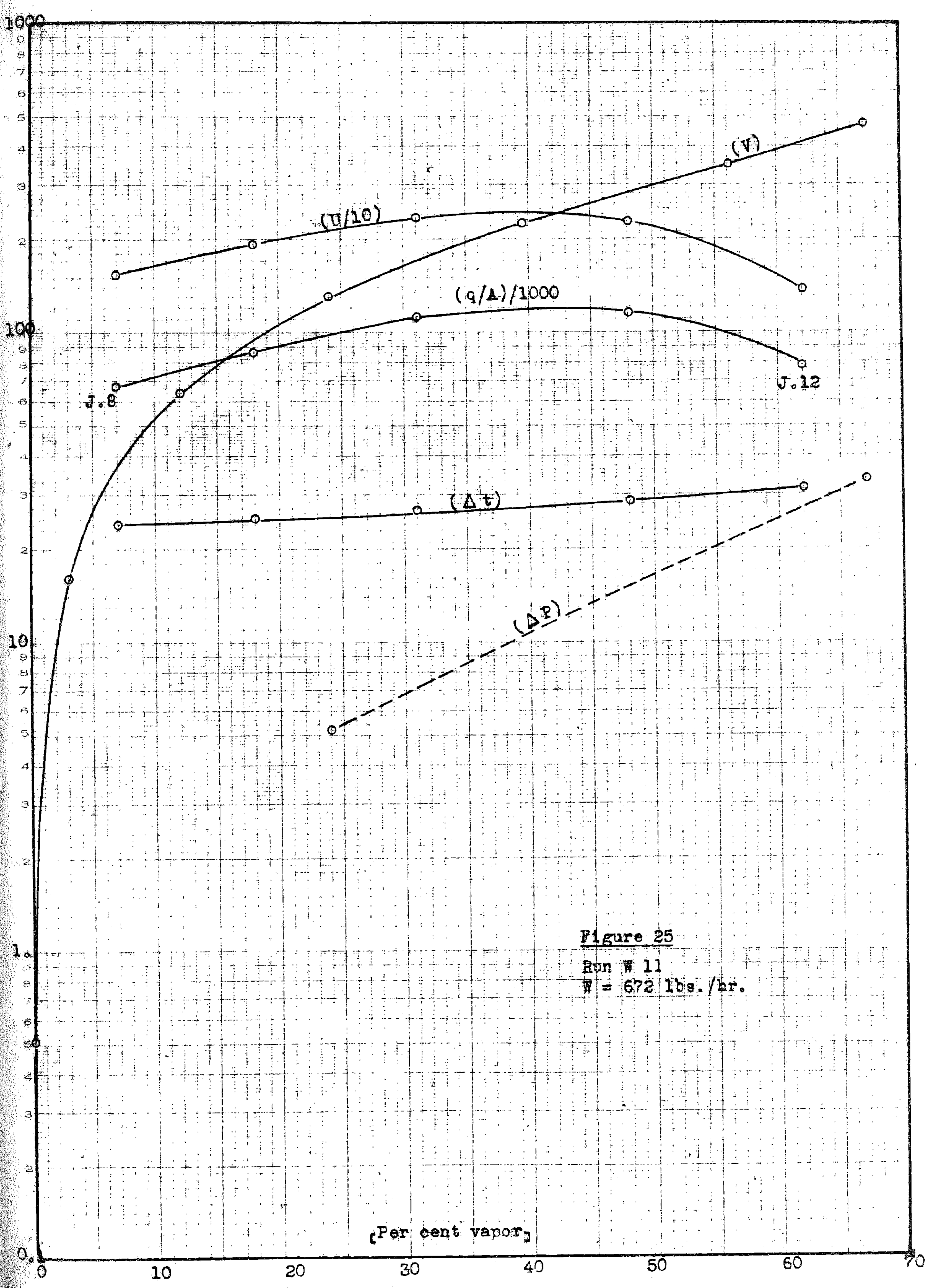


Figure 25

Run W 11

W = 672 lbs./hr.

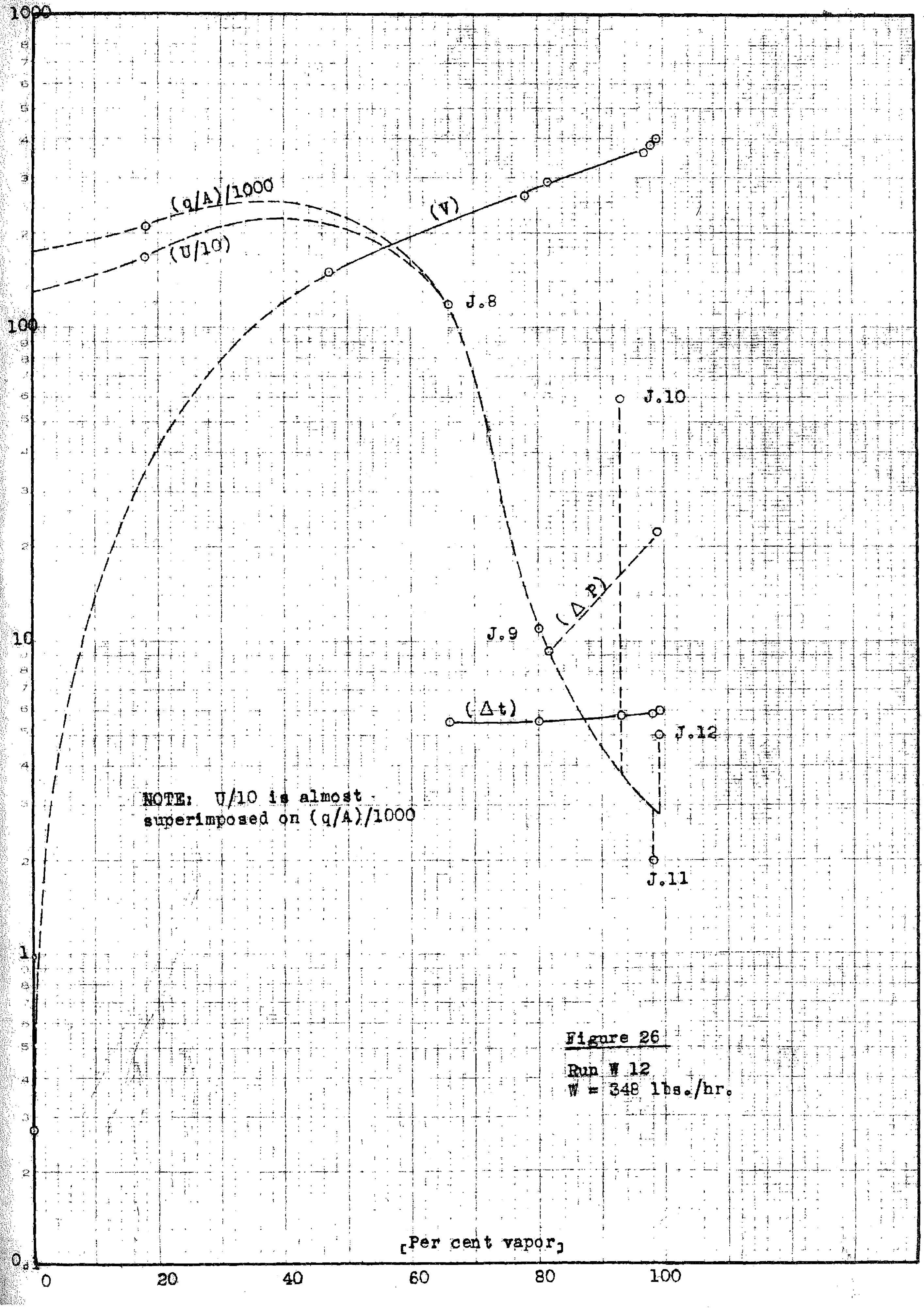


Figure 26

Run W 12
W = 348 lbs./hr.

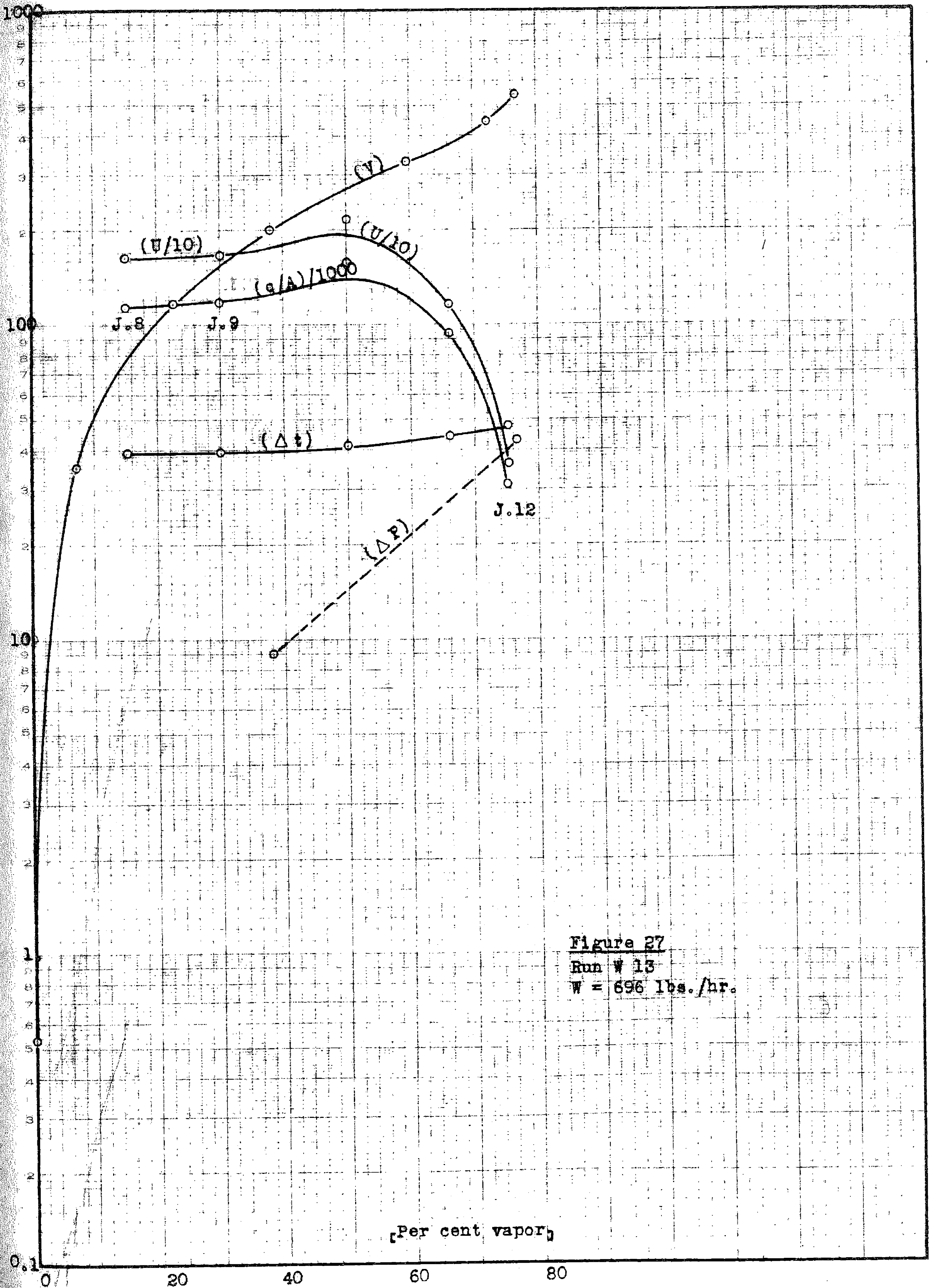


Figure 27
Run W 13
W = 696 lbs./hr.

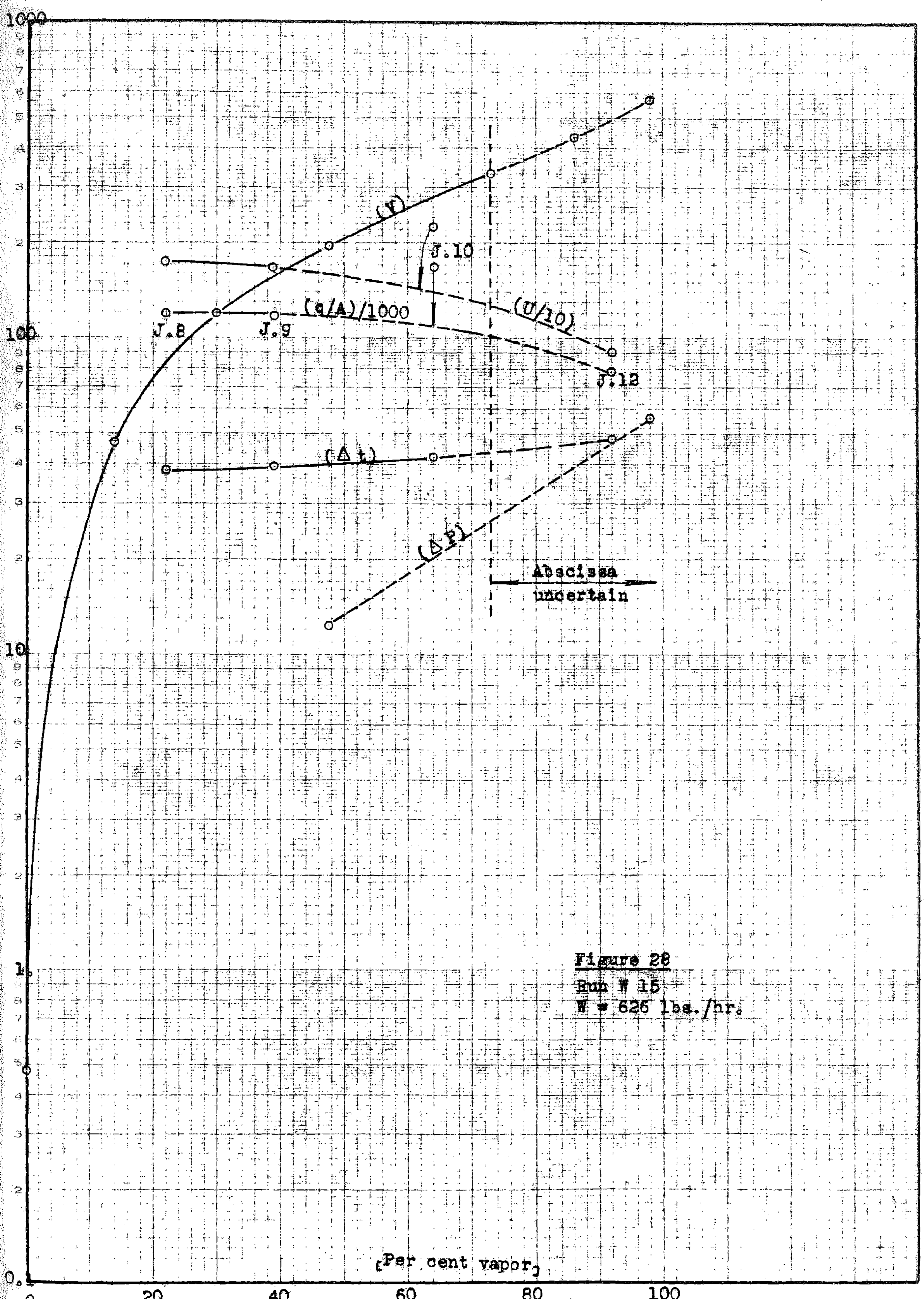


Figure 28
Run W 15
W = 626 lbs./hr.

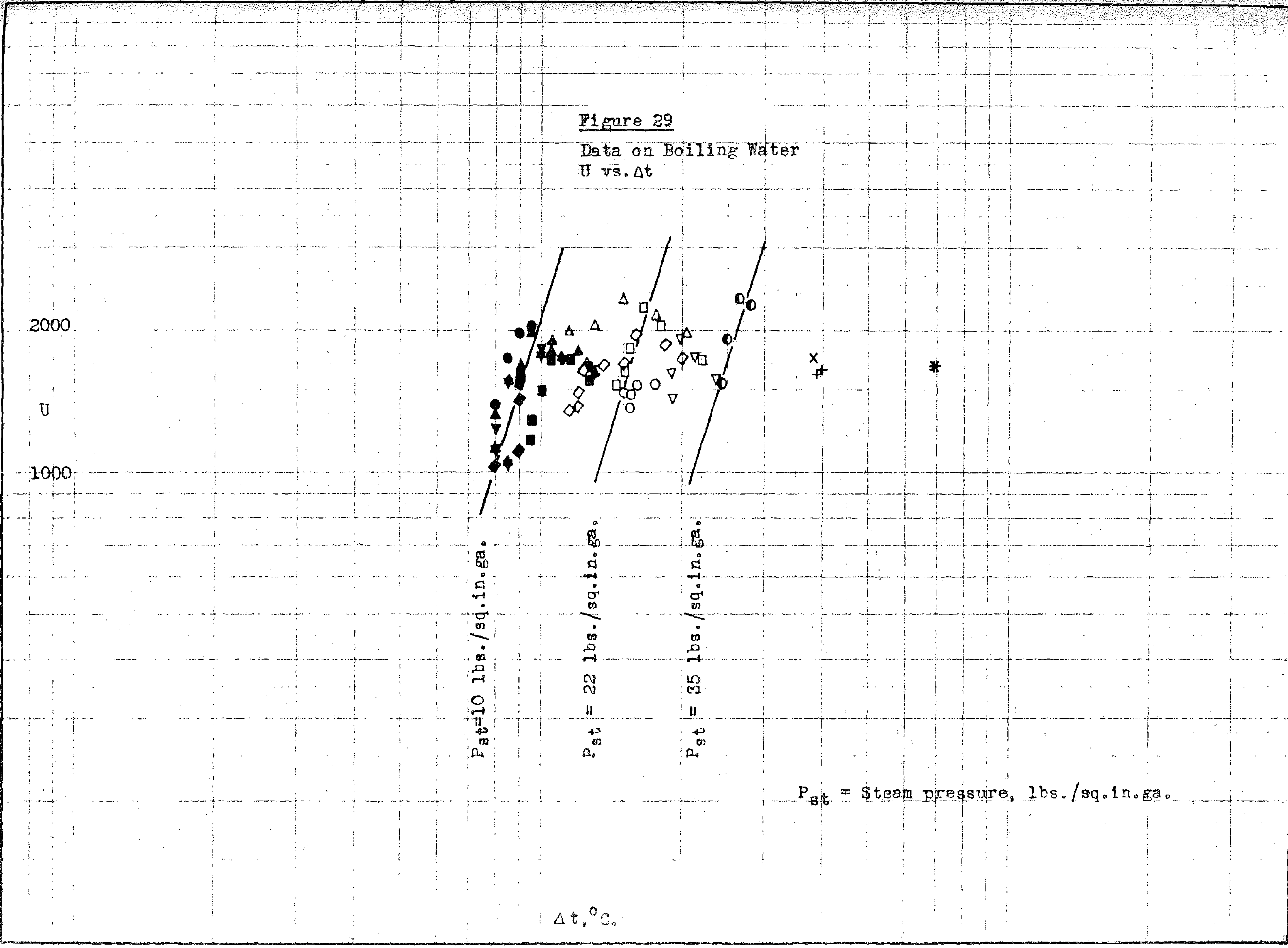


Figure 29

Data on Boiling Water

U vs. Δt

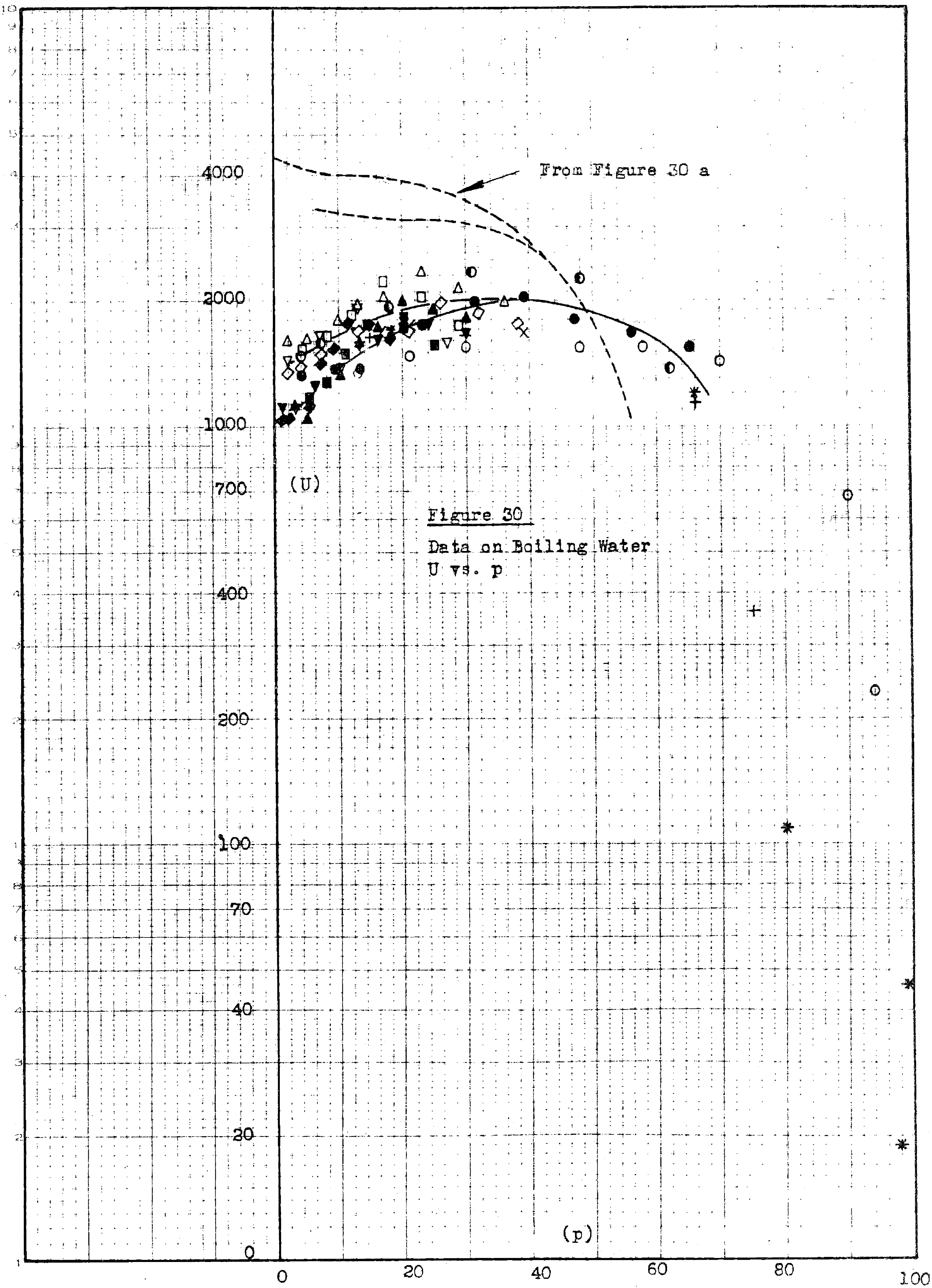


Figure 30

Data on Boiling Water
U vs. p

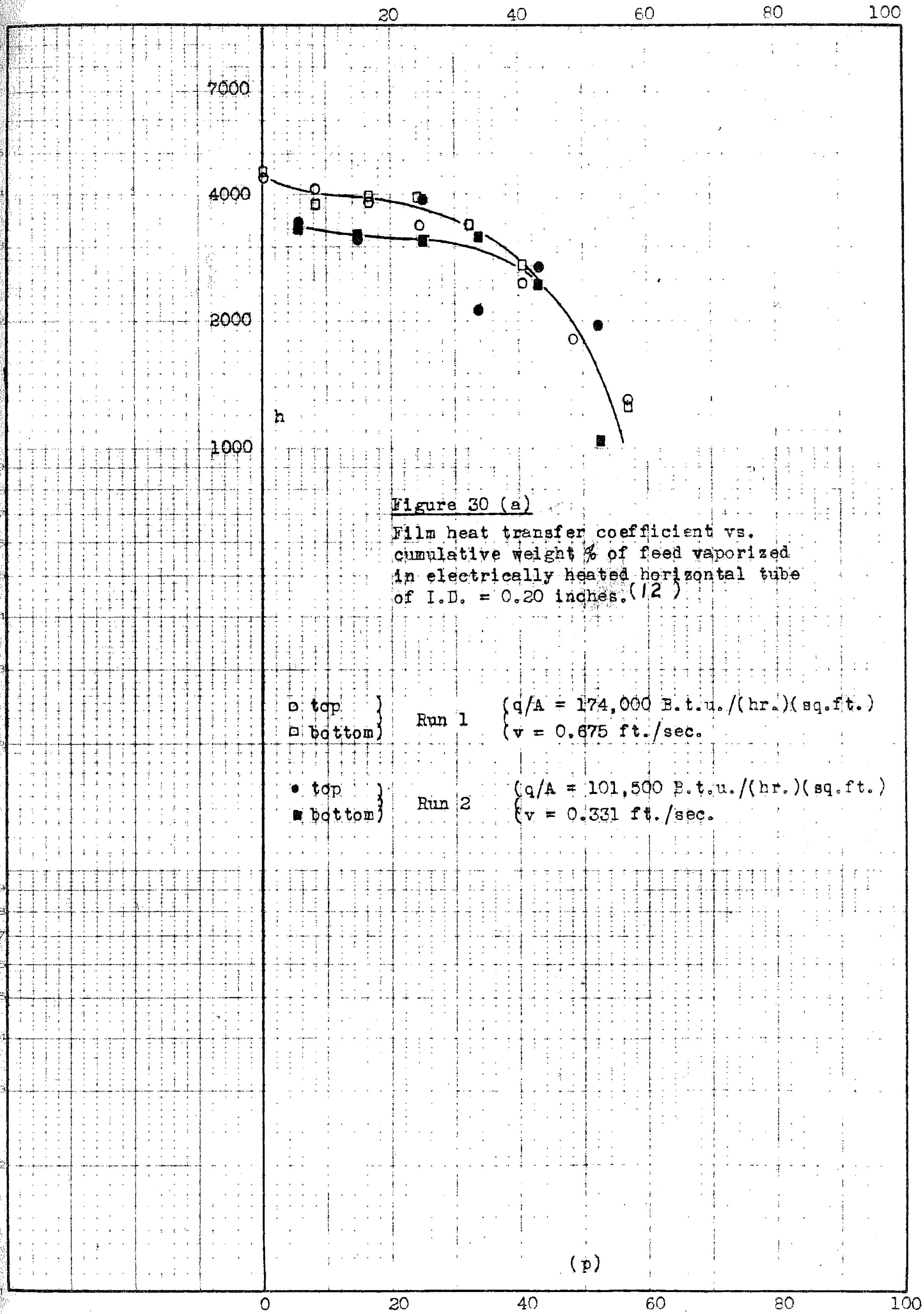


Figure 30 (a)

Film heat transfer coefficient vs. cumulative weight % of feed vaporized in electrically heated horizontal tube of I.D. = 0.20 inches.(12)

○ top, □ bottom	Run 1	q/A = 174,000 B.t.u./(hr.)(sq.ft.); v = 0.675 ft./sec.
● top, ■ bottom	Run 2	q/A = 101,500 B.t.u./(hr.)(sq.ft.); v = 0.331 ft./sec.

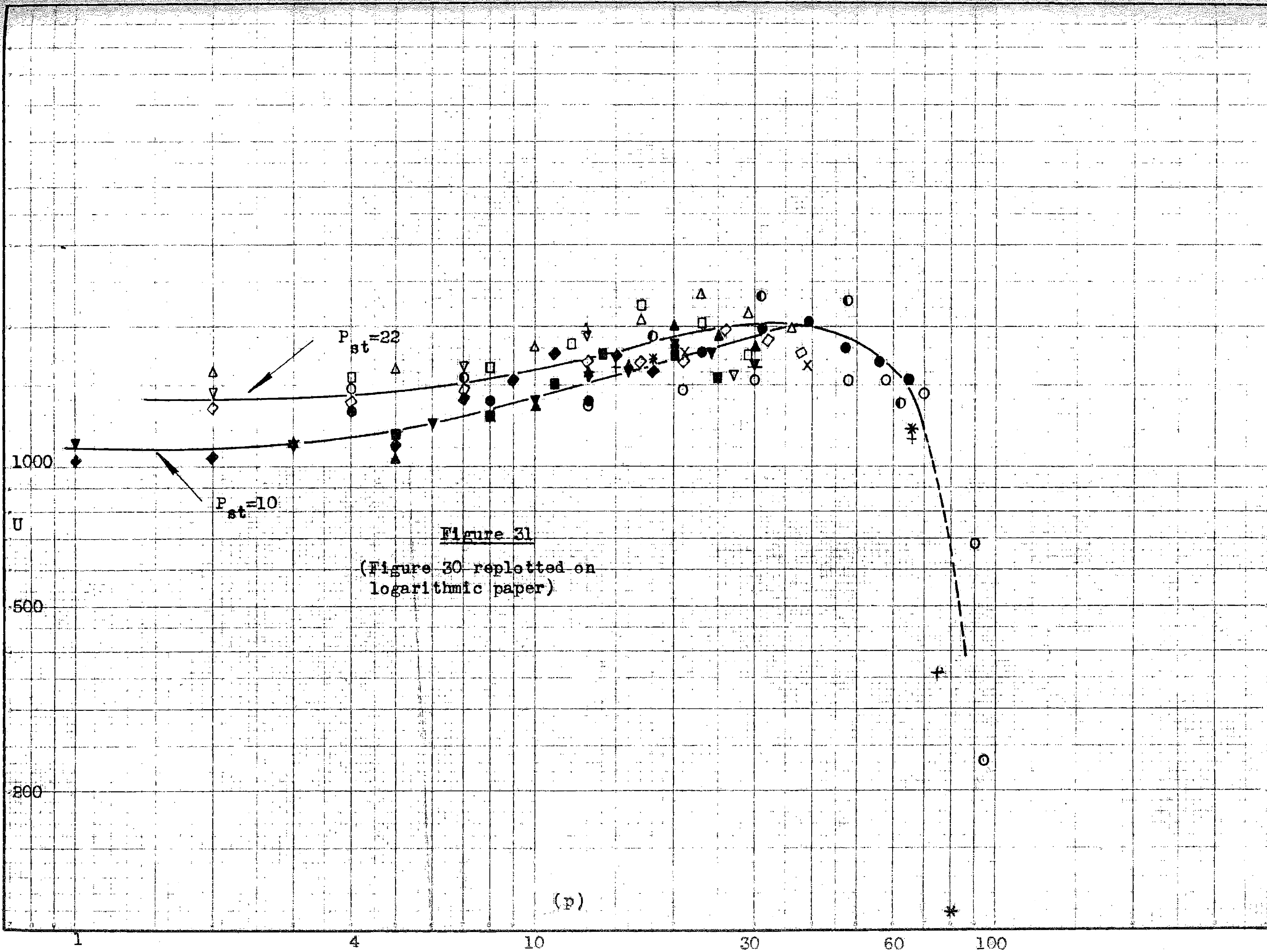


Figure 31

(Figure 30 replotted on logarithmic paper)

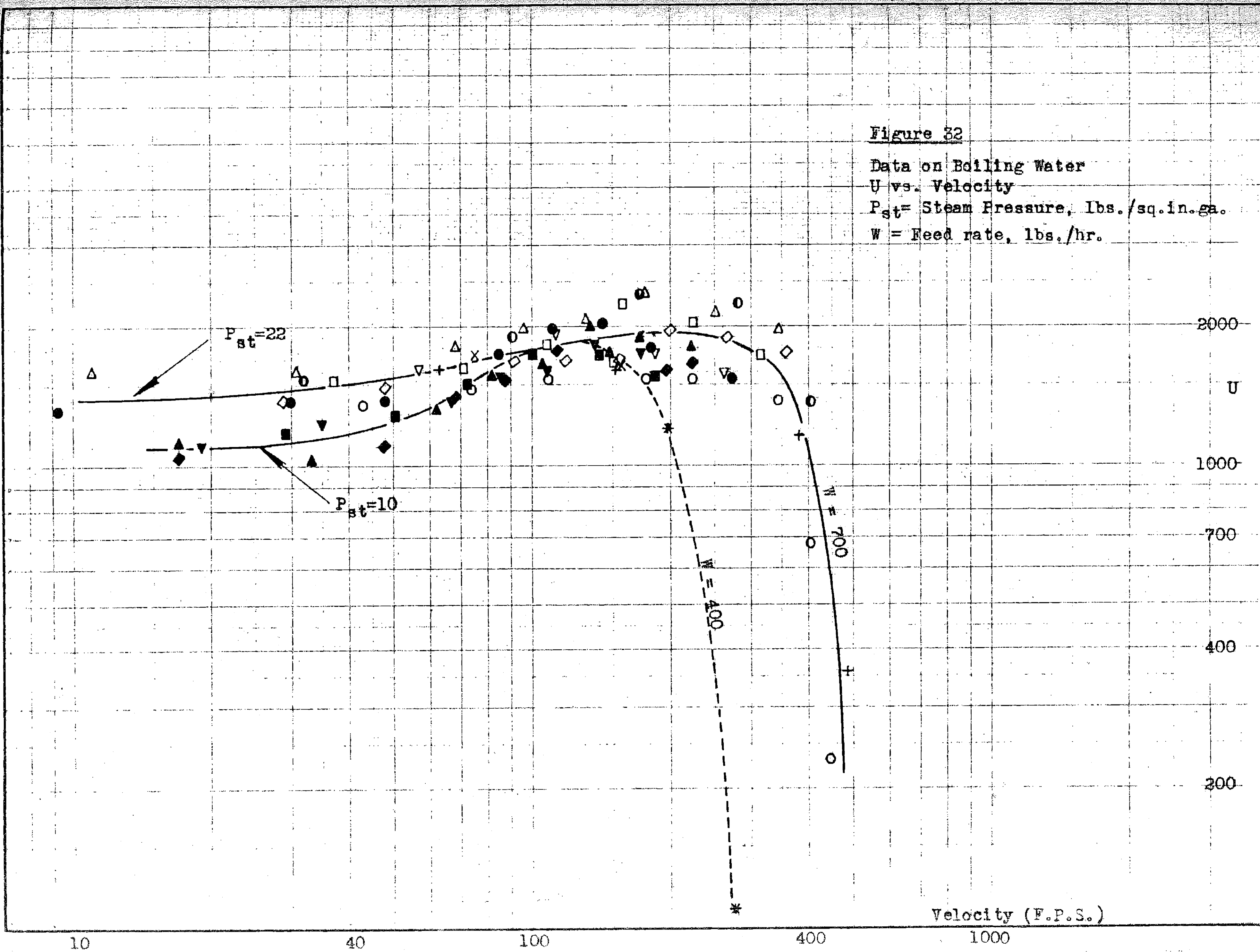


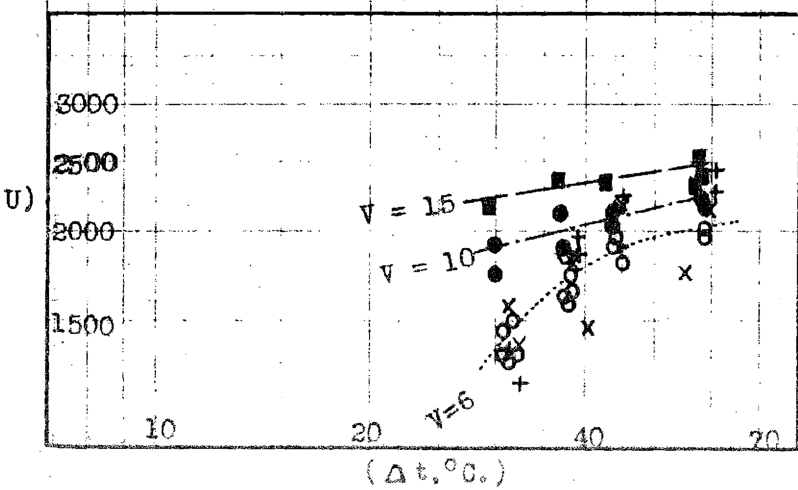
Figure 32

Data on Boiling Water
U vs. Velocity
P_{st}= Steam Pressure, lbs./sq.in.ga.
W = Feed rate, lbs./hr.

Figure 33 a

Apparent overall heat transfer coefficients vs. apparent overall Δt in steam-heated vertical tube. (10)

- ■ Feed velocity=136-15.4 F.P.S.
- ● Feed velocity= 9.4-11.7 F.P.S.
- o Feed velocity= 5.7-8.4 F.P.S.
- x Feed velocity= 3.2-3.6 F.P.S.
- + Feed velocity = 2.8 F.P.S.



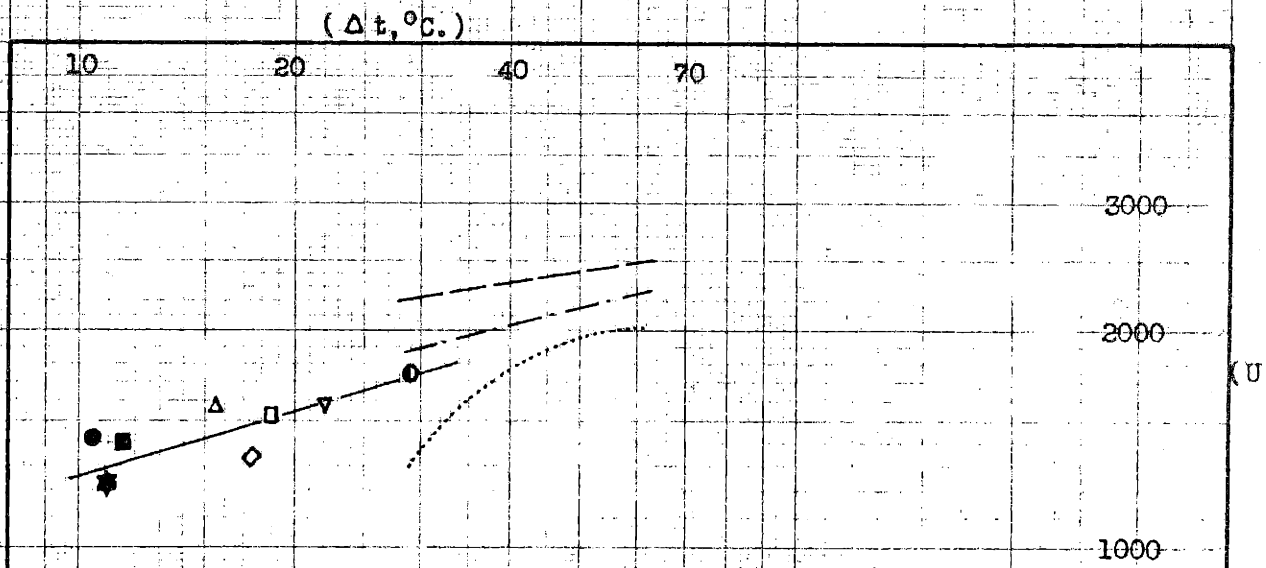


Figure 33

Data on boiling water. Average U for boiling section of runs on water vs. average terminal overall temperature differences.

— — — V = 15 F.P.S.
—·—·— V = 10 F.P.S. } From Figure 33 a
········ V = 6 F.P.S.

Symbols described in Table

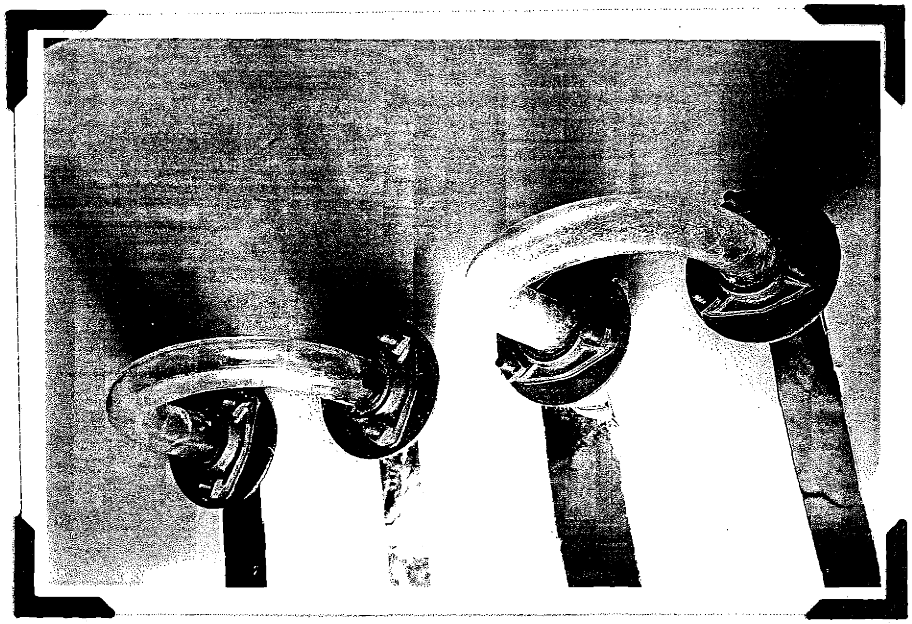


Illustration K. High-speed photograph of U-bends at end of 1st and third passes. Taken between surges at a feed rate of 530 lbs. of water per hour and a steam pressure of 20 lbs./sq.in.ga. Calculated vaporization equals 8% of the feed (by weight) in the first U-bend and 47% in the third. Note the quiet liquid layer in the bottom of the first U-bend. A cardboard background was placed behind the U-bends.

Picture taken by Prof. H. E. Edgerton

Exposure = 1,100,000 sec.

Illustration L. High-speed photograph of U-bends at end of first and third passes. Picture taken a few seconds after Illustration K , during a surge in the first U - bend.

Picture taken by Prof. H. E. Edgerton.

<u>Illustration M</u>. High-speed photograph of U-bends at end of first and third passes. Picture taken a few minutes after Illustration L , at a somewhat higher steam pressure.

Picture taken by Prof. H. E. Edgerton

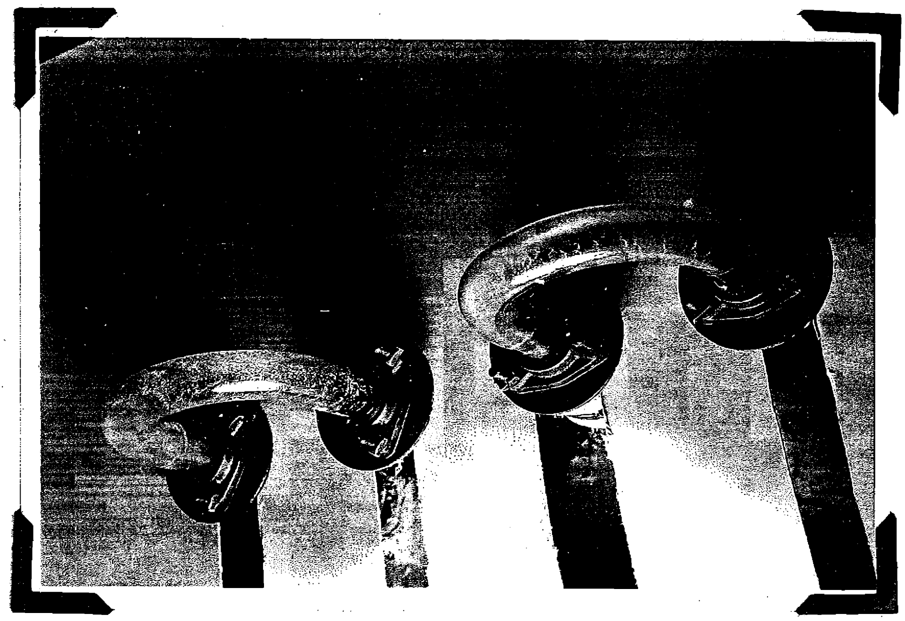


Illustration N. High-speed photograph of U-bends at end of first and third passes. Picture taken a few minutes after Illustration M , at a still higher steam pressure.

Picture taken by Prof. H. E. Edgerton.

Illustration O. High-speed photograph of U-bends at end of first and third passes. Picture taken a few minutes after Illustration N, after the steam pressure had climbed to 80 lbs./sq.in.ga. The feed rate was somewhat less than 530 lbs./hr. because of the increased pressure drop through the apparatus with no corresponding change in feed valve settings.

Picture taken by Prof. H. E. Edgerton

DISCUSSION OF RESULTS. Section I. - Heat Transfer to Boiling Water

Chronologically, the data on water were collected last. However, the water data are presented and discussed first because these results help to answer some questions which arise in connection with the earlier data.

Although the heating surface was of copper, most of the auxiliary piping was of iron. At the conclusion of the runs on benzene-oil mixtures the apparatus was repeatedly flushed with benzene. After the flushed benzene was water-white, the system was successively flushed with tap water until no further benzene odor could be detected. Finally, the system was flushed three times with distilled water, and the water was heated to its boiling point in order to vaporize any last traces of benzene. This water was then left in the apparatus 48 hours, so that when the initial water runs were made the water was turbid and red with rust.

This rusty water was used for the first five runs. The system was then flushed with distilled water until all color had disappeared. The glass U-bends were removed and the pipes were swabbed with pieces of cloth wrapped around a wire brush. This swabbing was continued until the cloths came out clean, although a slight rusty discoloration was still left on the pipe surface. The system was refilled with distilled water and Runs W6 and W7 were taken the same day. The operating conditions of these two runs duplicate those of two earlier runs with rusty water, Runs W2 and W5.

Runs W8 and W9 were taken with the same water used in Runs W6 and W7 but which had stood in the apparatus overnight and had become rusty. These runs had substantially the same water rates and steam pressures as Runs W2 and W5, but differed in that no steam was supplied to the first three jackets (the first pass).

Prior to taking the remaining runs the system was thoroughly flushed with distilled water but the metal pipes were not swabbed. Steam was supplied to only the last six of the twelve jackets. In anticipation of higher steam pressures it was planned to replace the low-range calibrated pressure gage by the high-range calibrated pressure gage. However, in the process the high-range gage was dropped, possibly affecting its performance, so another gage, previously calibrated by Oliver [10], was installed.

It was thought that steam at 59 lbs./sq.in.ga. was being used in Run W14. However, the steam condensate was flashing so violently that no readings of condensate rate could be taken and on lowering the pressure it was found that the borrowed steam gage had become jammed, despite the gentle tapping that had previously sufficed for the regular high-range gage. Run W15 was then taken at presumably 54 lbs./sq.in.gage pressure for comparison with Run W13. Subsequent inspection of the thermocouple readings indicated that Runs W12, W13, and W15 were all taken at pressures higher than indicated by the faulty gage. For these last three runs the reported steam pressure is based upon a condensing steam temperature estimated from the thermocouple readings.

The most striking feature of the water runs was the apparent independence of the results with regard to rusty or clean water. It was anticipated that the rusty water would give a lower heat flux for a given steam pressure than would the clean water and freshly swabbed pipes. The data of Runs W2 and W6 refute this prediction. With practically identical water rates and steam pressures in Runs W2 and W6, heat fluxes were obtained in the individual jackets for each run which (with the exception of Jackets 1 and 5) checked to within 1000 B.t.u./(hr.)(sq.ft.), --well within the precision of the data. (cf. page 163) Run W5 gave substantially the same results as Run W7.

Inspection of Figures 1 to 9 shows that Jacket 6 yielded heat flux and overall coefficients which were consistently lower than obtained in the adjacent jackets. That this phenomenon is typical of Jacket 6 is illustrated by its continued presence when Jackets 1, 2, and 3 are not heated (Runs W8 and W9). The same phenomenon is clearly shown in the curves for heat flux and coefficients in Figures 15 to 23, where the points for Jacket 6 are lower than for adjacent jackets. The abnormally low flux in Jacket 6 could be due to (a) inadequate venting of non-condensable gas, due to plugging of the air bleeder hole in the jacket, (b) partial plugging of the condensate drain line, resulting in blanketing a portion of the tube in condensate, or (c) fouling of the steam side of the heating surface. No data based directly on Jacket 6 were used in subsequent correlations.

In Run W15, the condensate sample from Jacket 11 was lost before measurement, although a fair estimate could be made of its volume. Accordingly, no data from Jackets 11 or 12 of Run W15 are used in the correlations.

For each run the temperature of the fluid could be measured at the point of entry to each of the four passes and at the point of discharge from the last pass. The measured temperatures (T_{f1}) of the fluid are indicated in Figures 1 to 14. In the boiling section the curve of T_{f1} is drawn through the saturation temperature of water corresponding to the measured pressure. The average deviation between the observed and saturation temperatures was less than 1°C., with a maximum deviation of less than 3°C. No general trend could be observed, other than that the effect was small. It is felt that the difference between the two temperatures is typical of the precision of measurement. Although both the measured temperature and pressure fluctuated with time, the variation of the saturation temperature corresponding to the pressure varied less than the measured temperature (page 168). Temperature differences, used in calculating the heat transfer coefficients, were based upon the curve of T_{f1} corresponding to the pressure.

It is noted that in Runs W12, W13, and W15 the heat flux in Jacket 10 was abnormally high compared with that in adjacent jackets. In Run W4 the heat flux in Jacket 10, although not out of line, could be much lower without destroying the continuity of the curve. The operating characteristic which is peculiar to these four runs is as follows: In Runs W4, W12, W13, and W15 at least 60 weight per cent of the feed has been vaporized by the time Jacket 10 has been passed. It is therefore concluded that the heat flux in Jacket 10 is abnormally high (relative to the adjacent jackets) when large per cents of the feed have been vaporized. The explanation for this phenomenon lies in the fact that Jacket 10 is located immediately after a U-bend and is the first jacket in a pass. This description applies equally well to Jackets 4 and 7 but in no run was as much as 60% of the feed vaporized by the time Jacket 4 or 7 was reached. When over 60% of the feed has been vaporized the residual liquid is carried as a spray down the center of the pipe. When the mixture swirls around a U-bend part of this liquid is thrown to the outside by centrifugal force, resulting in increased vaporization immediately after the bend.

Analogous to the above is the sharp decrease in rate of heat transfer after large per cents of the feed have been vaporized, -- about 60%. This is illustrated in the runs cited above, and also in Jacket 12 of Run W11. This phenomenon is one of vapor binding, caused not by abnormally high temperature differences but by the mere presence of a small amount of liquid entrained in a large amount of insulating vapor.

With the exception of this data taken after large weight per cents of the feed have been vaporized, the general trend of the data is to yield increasingly larger heat fluxes with increasing heated length. Undoubtedly this effect is due at least in part to the increasing temperature difference with increasing heated length. As the pressure on the liquid decreases because of the pressure drop through the apparatus, the boiling temperature decreases and the temperature between the steam and the boiling liquid increases. It would be possible to have a separate steam control on each jacket and to hold the temperature difference approximately constant all the way but the apparatus did not possess these controls. The heat transfer coefficients also increase as the heated length increases but this can be due in part to the increasing temperature difference, since the coefficient need not be independent of the temperature difference.

Figure 29 shows a plot of the overall coefficient, U, vs. the temperature difference. All points involving heat transfer after 60% or more of the feed had been vaporized are deleted from the plot, as well as all data taken from Jacket 6. The correlation is quite unsatisfactory. For runs at a given steam pressure it is noted that the coefficient tends to rise sharply with an increase in temperature difference, but that separate "lines" are obtained for different steam pressures. This lack of correlation of the data taken at the various steam pressures is probably the result of some additional variable, such as the per cent of the feed which has been vaporized up to any point in the apparatus.

The symbol (p) is used to indicate the cumulative weight per cent of the feed which is vaporized up to any point in the apparatus. Occasionally, as above, one refers to the p entering or leaving a jacket, meaning the per cent of the feed which has been vaporized up to the entrance or exit from a jacket. However, when the heat transfer coefficient for a jacket is compared with the cumulative vaporization (p) for the jacket, the value of p for the jacket is read from Figures 1 to 14 as the value of p at the point where the curve for per cent vaporization passes the midpoint of the jacket. In most cases the p for the jacket is substantially the arithmetic average of the p's entering and leaving the jacket. Occasionally, when boiling starts, say, 20% of the distance through the jacket, the p for the jacket is considerably less than half of the p leaving the jacket.

Figure 30 is a plot of U, the overall heat transfer coefficient for the jacket, vs. p for the jacket. Boiling data for all runs are shown, excluding the data from Jacket 6 for all runs and the data from Jacket 10 for Runs W4, W12, W13, and W15. Figure 31 is a re-plot of the data of Figure 30, using a logarithmic abscissa scale instead of a linear scale. Figure 31 was drawn primarily for comparison with Figure 32 (discussed below). In so far as the value of p is affected by errors in determining the feed rate by means of the feed orifice, Figure 31 is a more satisfactory method of plotting than Figure 30.

In Figures 30 and 31 it is noted that at vaporizations of less than 10% the higher steam pressures (about 22 lbs./sq.in.) give higher coefficients than the lower steam pressures (about 10 lbs./sq.in.), but this effect fades out as increasing amounts of feed are vaporized. The curve drawn through the points is accordingly divided into two branches at low values of p, giving an approximate measure of the effect of temperature difference. After 40% of the feed has been vaporized the coefficients fall off, although the data are not consistent as to the path: Runs W12 and W13 indicate a different path than Run W4. No data were taken for runs having a feed rate of 1000 lbs./hr.

and vaporization (p) greater than 40%.

In analyzing Figure 30 several facts must be born in mind. In Run W12 47% of the feed was vaporized in the first heated jacket; in Run 3 the total vaporization in all 12 jackets was only 20%. Since the data of Figure 30 includes data taken under both of these conditions the abscissa involves a considerable degree of uncertainty as to the proper method of determining the effective value of p for the jacket. Each point represents data on a single jacket, although smoother curves might have been obtained by averaging the results, say, of three jackets in a given pass. The coefficients are based upon overall temperature differences, and variation in condensing steam side conditions (due, say, to non-uniform promotion of dropwise condensation) introduces an added uncertainty. Finally, all data represent averages of fluctuating readings, and even the amount of fluctuation varied in intensity from run to run. In light of these factors the correlation as shown in Figure 30 is quite satisfactory.

The dotted lines on Figure 30 are transposed from Figure 30a. Figure 30a shows data of Slade [12] as a plot of the film heat transfer coefficient (h) plotted on semi-logarithmic paper versus the cumulative weight per cent of the feed vaporized. The data were taken in two runs in a 31 inch length of stainless steel horizontal tubing, 0.200 inch inside diameter, heated by direct passage of electricity through the tube walls. Tube wall temperatures were measured at intervals along the tube by thermocouples installed on the top and bottom of the tube. Liquid temperatures were measured at inlet and outlet and were estimated at intermediate points by estimates of the pressure gradient through the tube, using the saturation temperature at the assumed pressure. Temperature differences are of questionable accuracy, as indicated by the erratic difference between recorded coefficients at the top and bottom of the tube. The data were taken in two runs, using a constant heat flux of 174,000 and 101,500 B.t.u./(hr.)(sq.ft.) and an entering water velocity of 0.675 and 0.331 ft./sec., respectively. Since a feed rate of 1000 lbs./hr. in the steam heated apparatus corresponds to an entering water velocity of 0.745 ft./sec., the data are comparable on a linear velocity or mass velocity (lbs.)/(hr.)(sq.ft. of cross section) basis, and differences must be ascribed primarily to the much smaller tube diameter.

The data of Slade is are qualitatively similar to the data taken on the steam heated apparatus. A two-fold change in feed rate had minor effect on the heat transfer coefficient. At high flux (and high temperature differences) larger heat transfer coefficients were obtained than at low heat flux, provided the per cent of the feed vaporized was small. At high percentage vaporization the coefficients dropped rapidly, although the "critical" value of p, above which vapor binding occurred, was smaller than obtained in the steam heated apparatus, being about 30% instead of 40%. At high percentage vaporization the coefficients were relatively independent of heat flux. No appreciable increase in h with increasing p was noticed. Coefficients for warming liquids in turbulent flow are known to vary as the inverse fifth root of the pipe diameter; with a diameter ratio of about 53:1 it would be expected that warming coefficients would vary as about 2.2:1; it is noted that the coefficients obtained by Slade at low values of p are about twice as large as those obtained in the steam-heated apparatus at low values of p.

Comparing the thesis results with the other data cited in the Introduction on horizontal pipes, it is noted that the overall heat transfer coefficients obtained are far greater in the individual jackets than the overall heat transfer coefficients obtained by Badger (3) in a horizontal steam-heated tube. The maximum vaporization obtained by Badger was 20%, so his low coefficients scarcely could have been due to vapor binding. Undoubtedly, the low overall coefficients obtained by Badger were the result of poor steam-side conditions. The results of the water runs do not confirm the effect of feed rate which Badger obtained.

As mentioned, examination of Figure 30 indicates that the heat transfer is independent of the feed rate. It is not unreasonable that when only small per cents of the feed have been vaporized the turbulence set up by the surging action may completely dominate variations in turbulence resulting from varying the feed rate. However, it would seem that when large amounts of the feed had been vaporized, establishing very high velocities, doubling the feed rate should approximately double these very high velocities and create an additional turbulence which should dominate variations in turbulence resulting from the surging action.

The higher static pressures encountered at high feed rates tend to minimize velocity variations but are insufficient to completely eliminate such variations. As a check on the conclusion that the coefficients are independent of rate of flow, a plot (Figure 32) was made of the overall coefficients vs. the geometric mean of the entering and leaving velocities. It is noted that the coefficients fall off at low velocities when the feed rate is low, but persist to higher velocities when the feed rate is higher, confirming the results of Figure 30. (Some of the data at small values of p in Figure 31 yielded average velocities too small for inclusion in Figure 32).

In those runs where the total vaporization does not reach as high as 60% it is noted nevertheless that the coefficient usually undergoes a gradual decrease in the three jackets of the last pass. It is possible that a minor effect of the third U-bend causes this, with a residual effect carrying on into the middle jacket, so that only the third jacket of the pass is behaving normally, unaffected by the U-bend.

This marked effect of the U-bends after 60% of the feed has been vaporized suggests the possibility of modifying the apparatus when large percent vaporization is needed. Thus, one could use a series of rather short pipes connected by many U-bends. More practical would be the use of a steam-jacketed spiral coil, but this design makes difficult cleaning of the inside heating surface. Probably the best suggestion is the use of a helical surface inside the tube, -- a twisted metal strip slipped into the pipe and extending the length of the pass. This method should set up a whirling flow which would continually throw a vapor spray back onto the walls, -- in addition to possibly increasing the heated area as a result of heat conduction by the metal strip. The strip could be pulled out prior to brushing or swabbing the inside of the pipe. Use of such twisted metal strips (known as "whirlers") in condenser tubes has proven unsatisfactory, for the increase in heat transfer rate was insufficient to justify the large increase in pressure drop. It would appear that the poor heat transfer in the case of vapor binding inside pipes as a result of excessive vaporization is far more adaptable to improvement by the use of whirlers.

No use has been made of the thermocouple data other than to estimate the steam pressure in Runs W12, W13, and W15. The actual treatment of the thermocouple data is presented in the Appendix, page 178 . It is felt that the thermocouple data is very unreliable, probably as a result of unsatisfactory methods of providing dry junctions. The constantan-copper thermocouples were made up with a common copper lead, and the constantan wire leaving the metal surface was insulated for about an inch with a glass capillary. The changing calibration of some of the couples indicates that these capillaries may have broken as a result of thermal expansion and contraction, allowing steam condensate to wet the junction and establishing an electrochemical cell between the copper and the constantan.

The results of the water runs have been compared with other data on boiling inside of horizontal tubes. It remains to compare the data with other data on boiling inside of <u>vertical</u>

tubes. Of special interest is the work of Stroebe (13), who boiled water inside of a 20 -foot vertical tube, using such rates of flow and such steam pressures that extremely high percentages of the feed were vaporized. Stroebe reports film coefficients of heat transfer, the tube wall temperature being measured by thermocouples and the liquid temperature being measured by a travelling thermocouple which was moved up and down inside of the tube. Under all operating conditions the film coefficients of heat transfer obtained by Stroebe in a 1.74 -inch I.D vertical tube were between 1000 and 2000 B.t.u./(hr.)(sq.ft.)(^{o}F.). With good promotion of dropwise condensation of steam, the overall heat transfer coefficients obtained in the apparatus of this thesis should be but slightly smaller than the film coefficients of heat transfer; the data of Figures 30 to 32 show overall heat transfer coefficients in the range of 1000 to 2000 B.t.u./(hr.)(sq.ft.)(^{o}F.).

Oliver (10) obtained heat transfer coefficients for boiling water inside of a 0.495 -inch inside diameter vertical steam-heated nickel tube, 1.72 -feet long. He promoted his steam with oleic acid. Because of the high entering water velocities (up to 16 feet per second) and the short heated length, Oliver frequently did little more than superheat the feed, boiling only relatively small per cents of the feed. Figure 33 is presented because of the fortuitous agreement with the overall coefficients obtained by Oliver, although no especial significance is attached to the plot. The overall coefficients obtained by Oliver are plotted in Figure 33(a) and the lines from Figure 33 (a) transposed to Figure 33. Incidentally, Oliver obtained film coefficients of heat transfer as high as 9,000 B.t.u./(hr.)(sq.ft.)(°F.) at high steam pressures. Oliver's overall coefficients were apparent coefficients, based upon the difference in temperature between the condensing steam and the average of the fluid temperatures in and out of the apparatus.

The heat transferred in the boiling section of all the runs on water except those in which bad vapor binding occurred (Runs W4, W12, W13, and W15) was divided in succession by the area of the boiling section and the logarithmic average of the overall temperature differences at the beginning and end of the boiling section. The resultant average coefficient for the apparatus was plotted in Figure 33 vs. the temperature difference used in calculating the average coefficient.

R E S U L T S   A N D   D I S C U S S I O N

Section II. - Heat Transfer to Boiling Benzene

RESULTS. Section II. - <u>Heat Transfer to Boiling Benzene</u>

The operating conditions for the runs on boiling benzene are summarized in Table II. The symbols listed in Table II are used in all the subsequent correlations of the data on boiling benzene, --specifically, Figures 36-38, 44, and 47.

The results of two typical runs on benzene are presented graphically in Figures 34 and 35. Analagous to Figures 1-14 for water, Figures 34 and 35 are plots of the data of benzene runs B 5 and B 9, showing curves for static fluid pressure (P), cm.Hg.ga.; condensing steam temperature (T_{st}), °C.; fluid temperature (T_{fl}), °C.; heat flux (q/A), B.t.u./(hr.)(sq.ft.); overall heat transfer coefficient (U), B.t.u./(hr.)(sq.ft.)(°F.); and cumulative weight per cent of the feed vaporized (p); plotted versus tube length, expressed in terms of jacket numbers.

Figure 36 is a plot of the overall heat transfer coefficient (U) versus the cumulative weight per cent of the feed vaporized (p), for all data taken at steam pressures close to 2 lbs./sq.in.ga. Separate plots are made for each feed rate, and the curves from each of these plots are superimposed on one another in Figure 36 (a).

Figures 37 (a), (b), and (c) are plots of the overall coefficient (U) versus the cumulative weight per cent of the feed vaporized (p) for data taken at higher steam pressures. The figures are for feed rates (W) of 400, 700 and 1000 lbs./hr., respectively, and the suitable curve for low steam pressures is transposed from Figure 36.

Figure 38 is a plot of the overall heat transfer coefficient (U) versus the fluid velocity (V), ft./sec., for all runs on benzene taken at steam pressures of close to 2 lbs./sq.in.ga.

Table II

Runs on Boiling Benzene

Symbol	Run No.	Feed Rate lbs./hr.	Steam Pressure lbs./sq.in.	Initial Feed Temp.°C.	Heat* Transfer B.t.u./hr.	Weight % of Feed Vaporized
†	B 6	1010	114	54.5	136,000	66
⸸	B 3	990	108	52.5	161,000	82
‡	B 2	906	12.7	45.0	183,000	100
+	B 4	1080	1.9	55.8	149,000	66
×	B 7 A	1030	2.1	58.5	146,000	73
▬	B 11	818	64	50.5	141,000	87
▮	B 10	689	34	44.5	152,000	100
/	B 9	710	28	45.0	157,000	100
\	B 8	750	13.4	45.7	157,000	100
✱	B 1	615	1.5	42.5	125,000	100
●	B 5	650	1.5	47.0	122,000	96
■	B 7	700	1.2	45.5	127,000	91
◆	B 14	713	1.1	44.0	124,000	85
▲	B 1 A	704	1.6	47.5	129,000	91
▼	B 4 A	674	1.7	51.5	127,000	100
▶	B 6 A	720	2.1	48.0	131,000	92
△	B 12	448	74	45.5	87,000	98
○	B 2 A	434	1.6	41.0	91,000	100
□	B 5 A	453	1.8	46.0	92,000	100
◑	B 13	289	70	37.5	60,000	100
◐	B 3 A	282	1.8	36.0	60,000	100

* Based on steam condensate measurements.

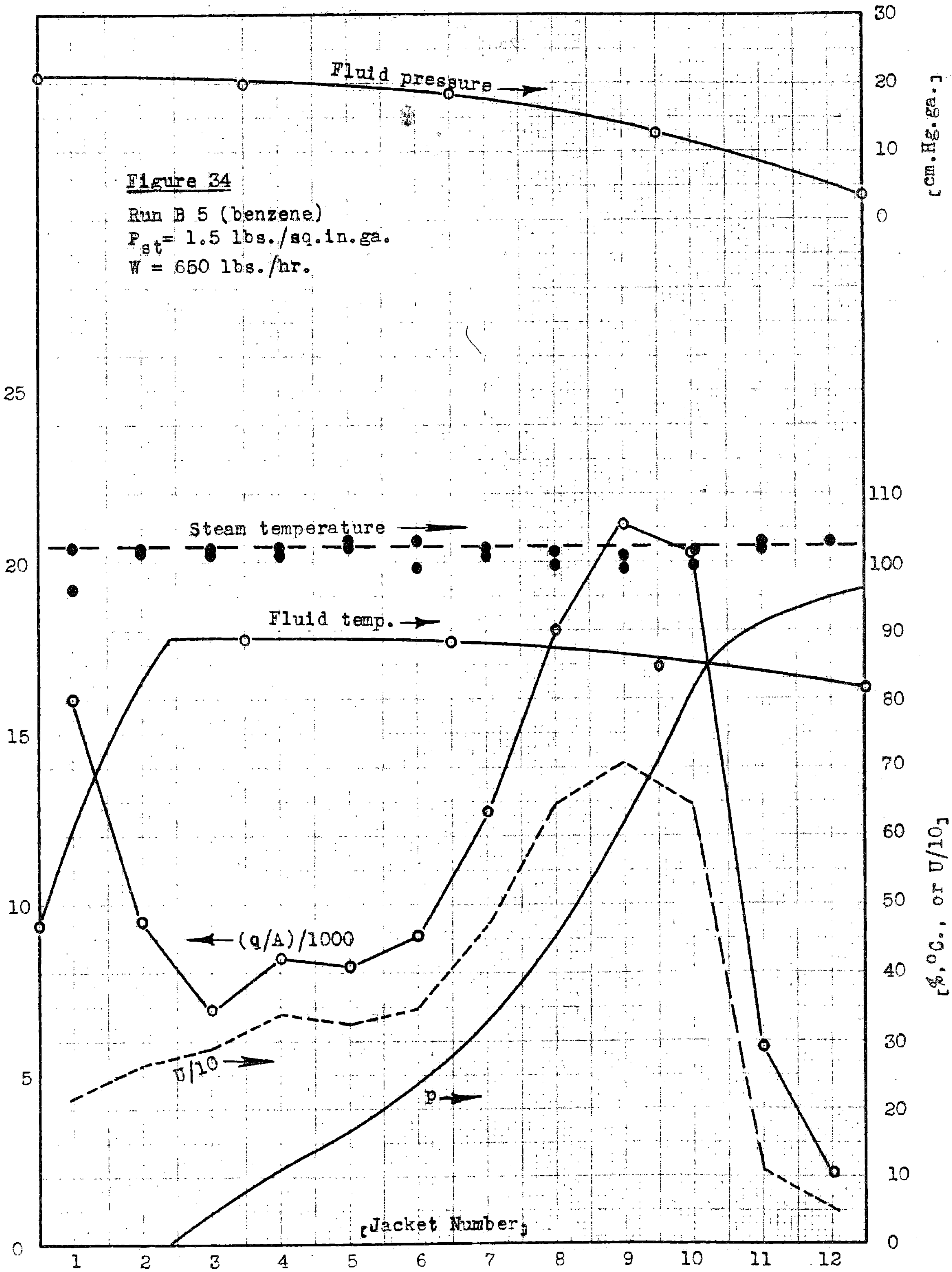


Figure 34
Run B 5 (benzene)
P_{st}= 1.5 lbs./sq.in.ga.
W = 650 lbs./hr.
Fluid pressure
Steam temperature
Fluid temp.
(q/A)/1000
U/10
p
[Jacket Number]
[cm.Hg.ga.]
[%, °C., or U/10]
0
5
10
15
20
25
1
2
3
4
5
6
7
8
9
10
11
12
30
20
10
0
110
100
90
80
70
60
50
40
30
20
10
0

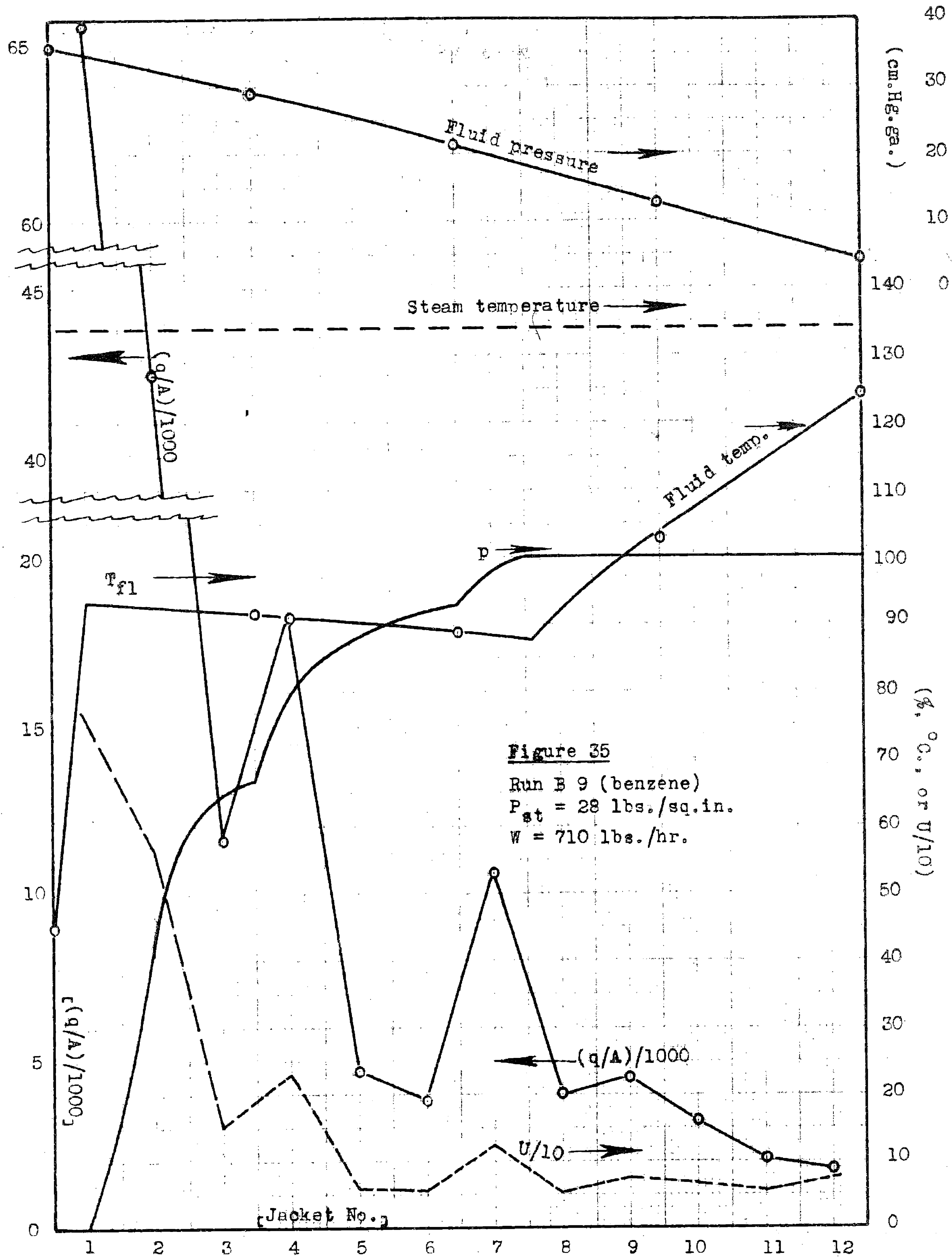


Figure 35

Run B 9 (benzene)

P_{st} = 28 lbs./sq.in.

W = 710 lbs./hr.

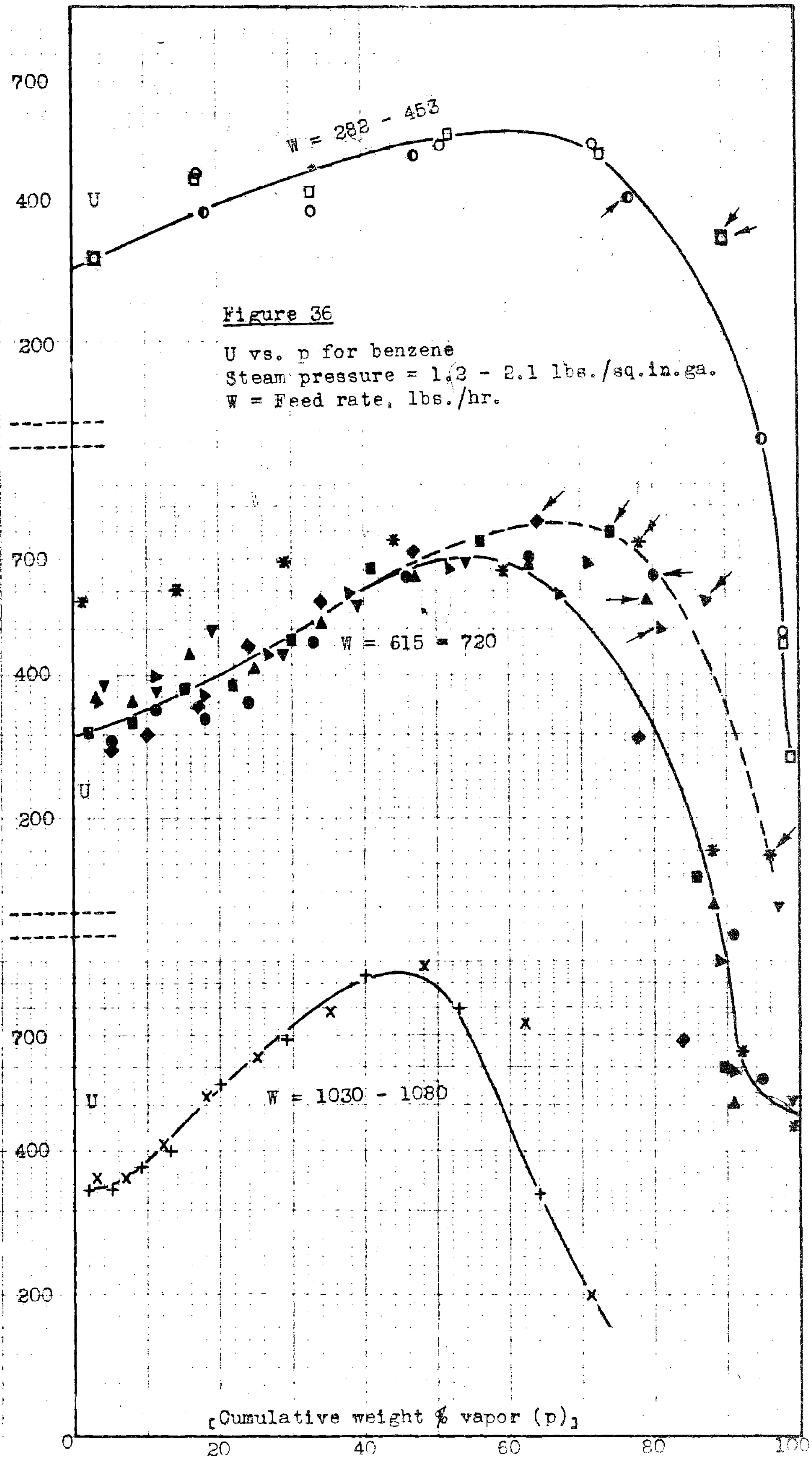


Figure 36
U vs. p for benzene
Steam pressure = 1.2 - 2.1 lbs./sq.in.ga.
W = Feed rate, lbs./hr.
W = 282 - 453
W = 615 - 720
W = 1030 - 1080
U
700
400
200
Cumulative weight % vapor (p)
0
20
40
60
80
100

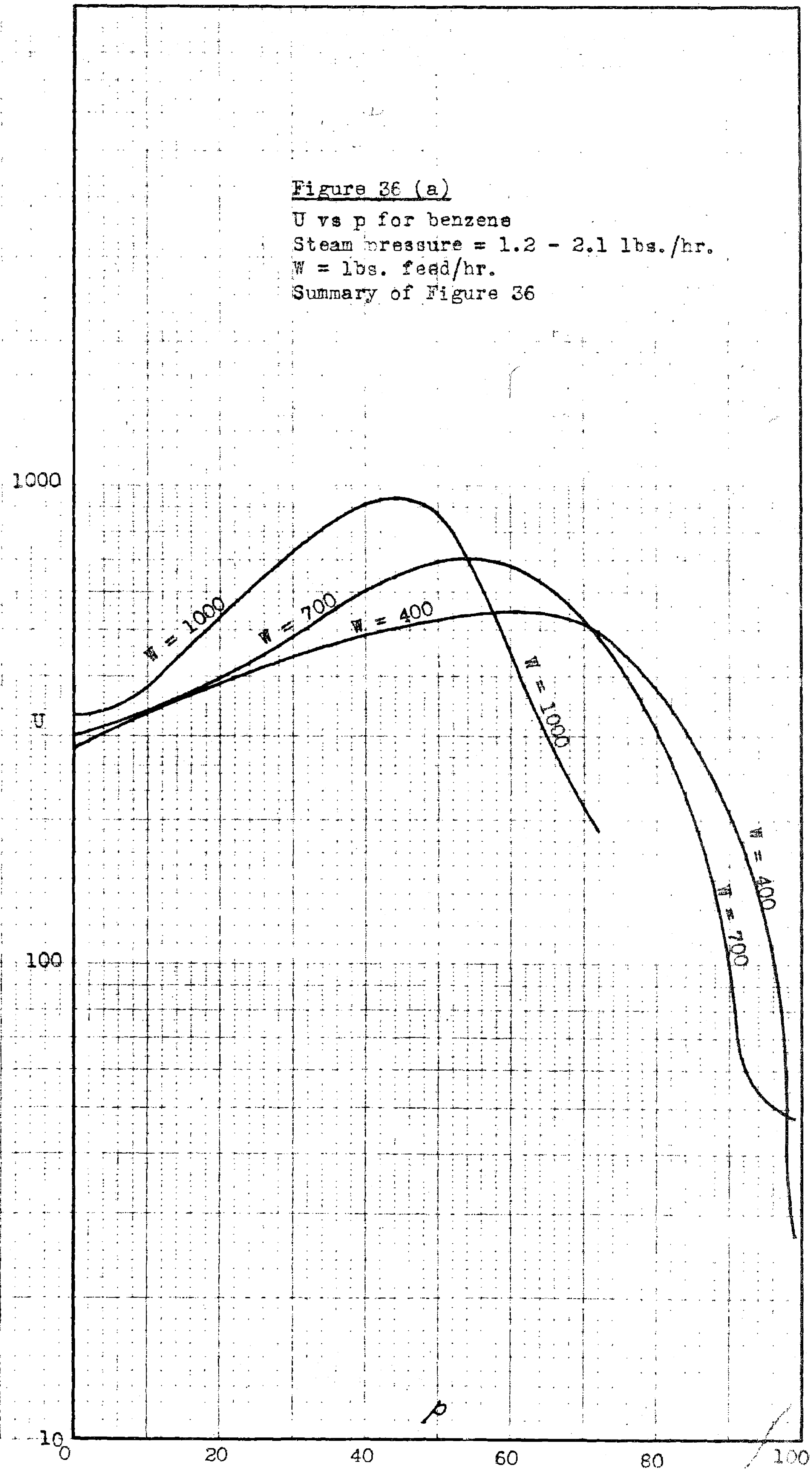


Figure 36 (a)
U vs p for benzene
Steam pressure = 1.2 - 2.1 lbs./hr.
W = lbs. feed/hr.
Summary of Figure 36
1000
100
10
U
p
0
20
40
60
80
100
W = 1000
W = 700
W = 400
W = 1000
W = 400
W = 700

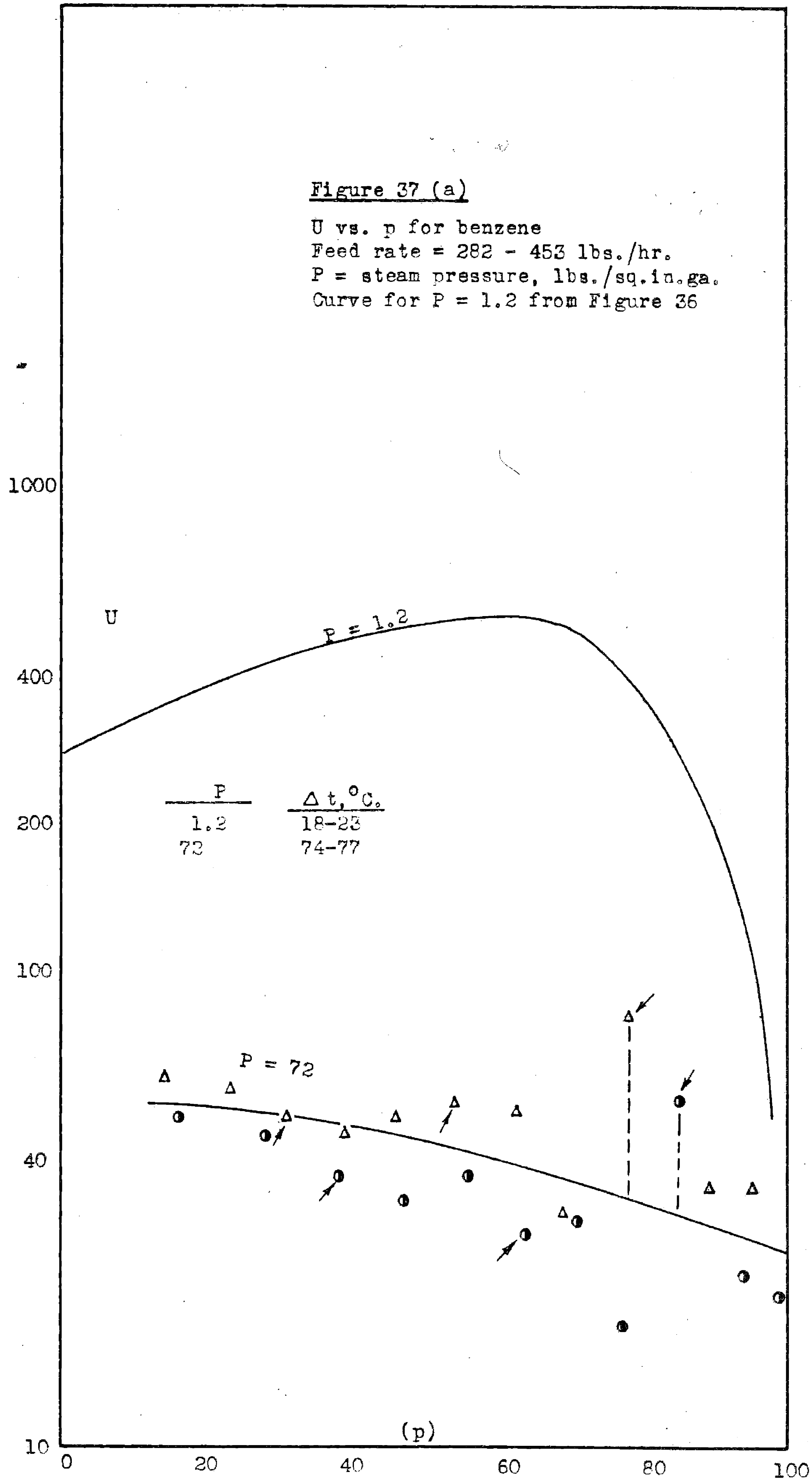


Figure 37 (a)
U vs. p for benzene
Feed rate = 282 - 453 lbs./hr.
P = steam pressure, lbs./sq.in.ga.
Curve for P = 1.2 from Figure 36
U
P = 1.2
P = 72
P
1.2
72
Δ t, °C.
18-23
74-77
1000
400
200
100
40
10
0
20
40
60
80
100
(p)

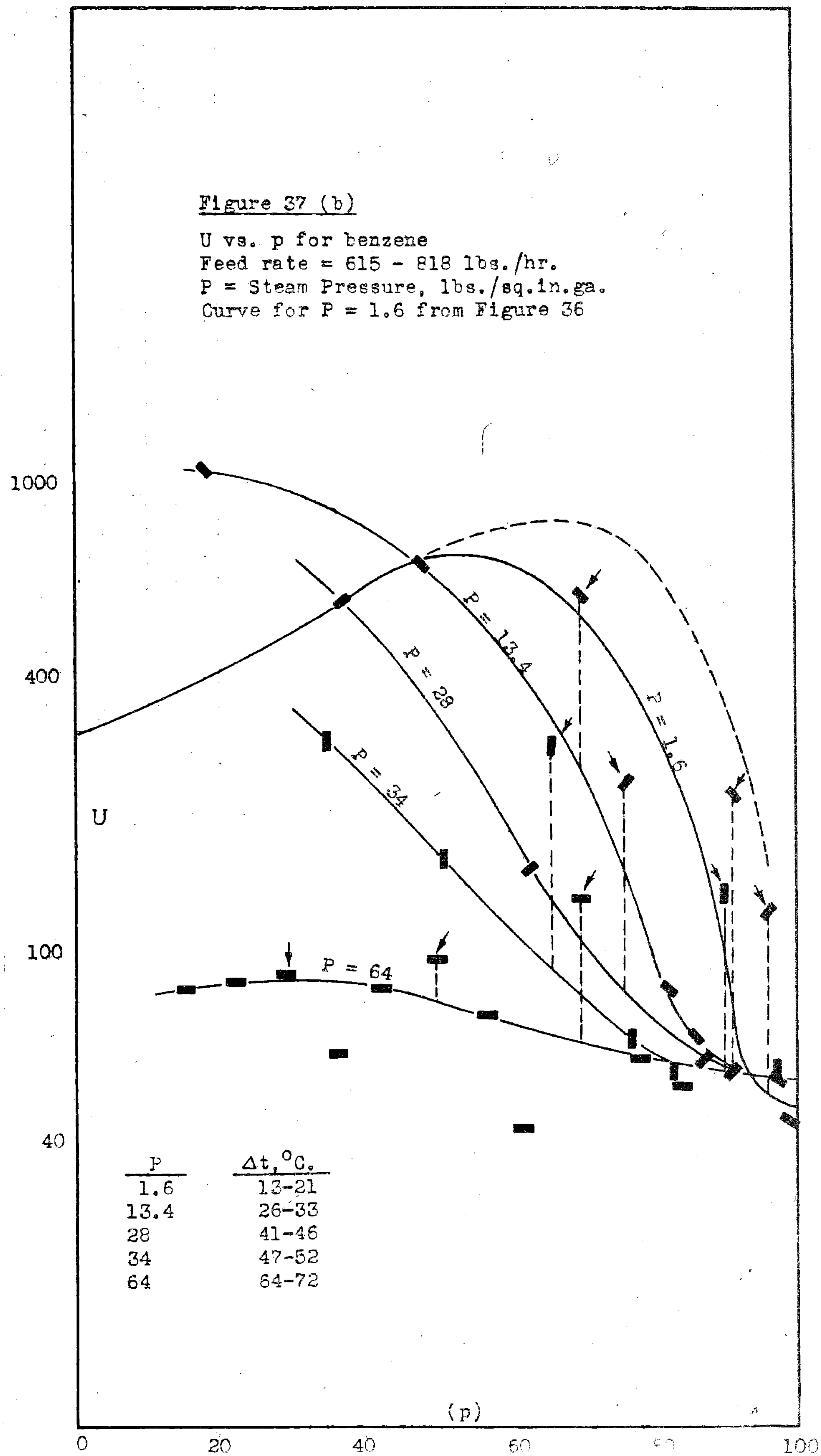


Figure 37 (b)
U vs. p for benzene
Feed rate = 615 - 818 lbs./hr.
P = Steam Pressure, lbs./sq.in.ga.
Curve for P = 1.6 from Figure 36
1000
400
100
40
U
P = 13.4
P = 28
P = 34
P = 1.6
P = 64
P
1.6
13.4
28
34
64
Δt, °C.
13-21
26-33
41-46
47-52
64-72
(p)
0
20
40
60
100

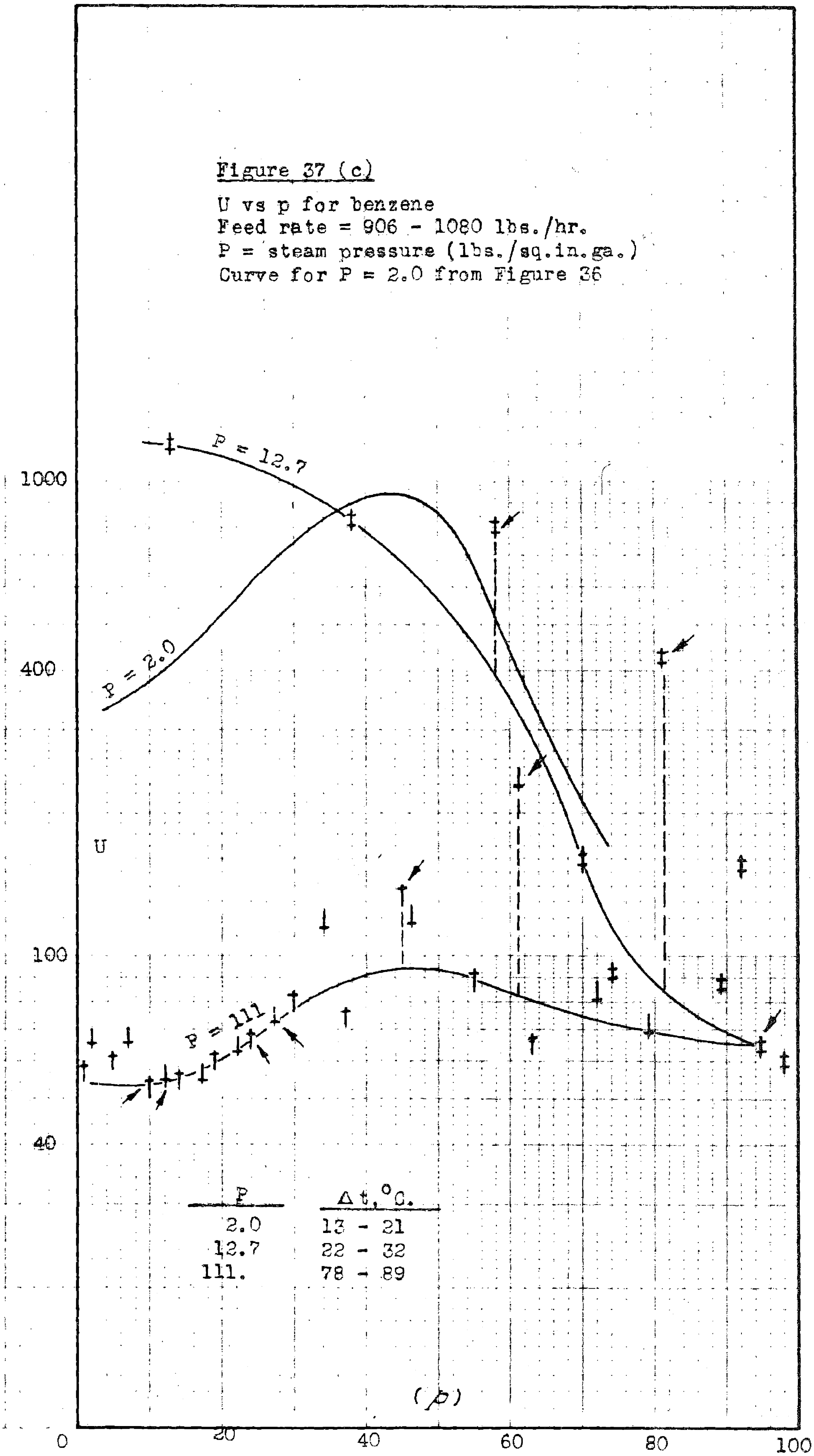
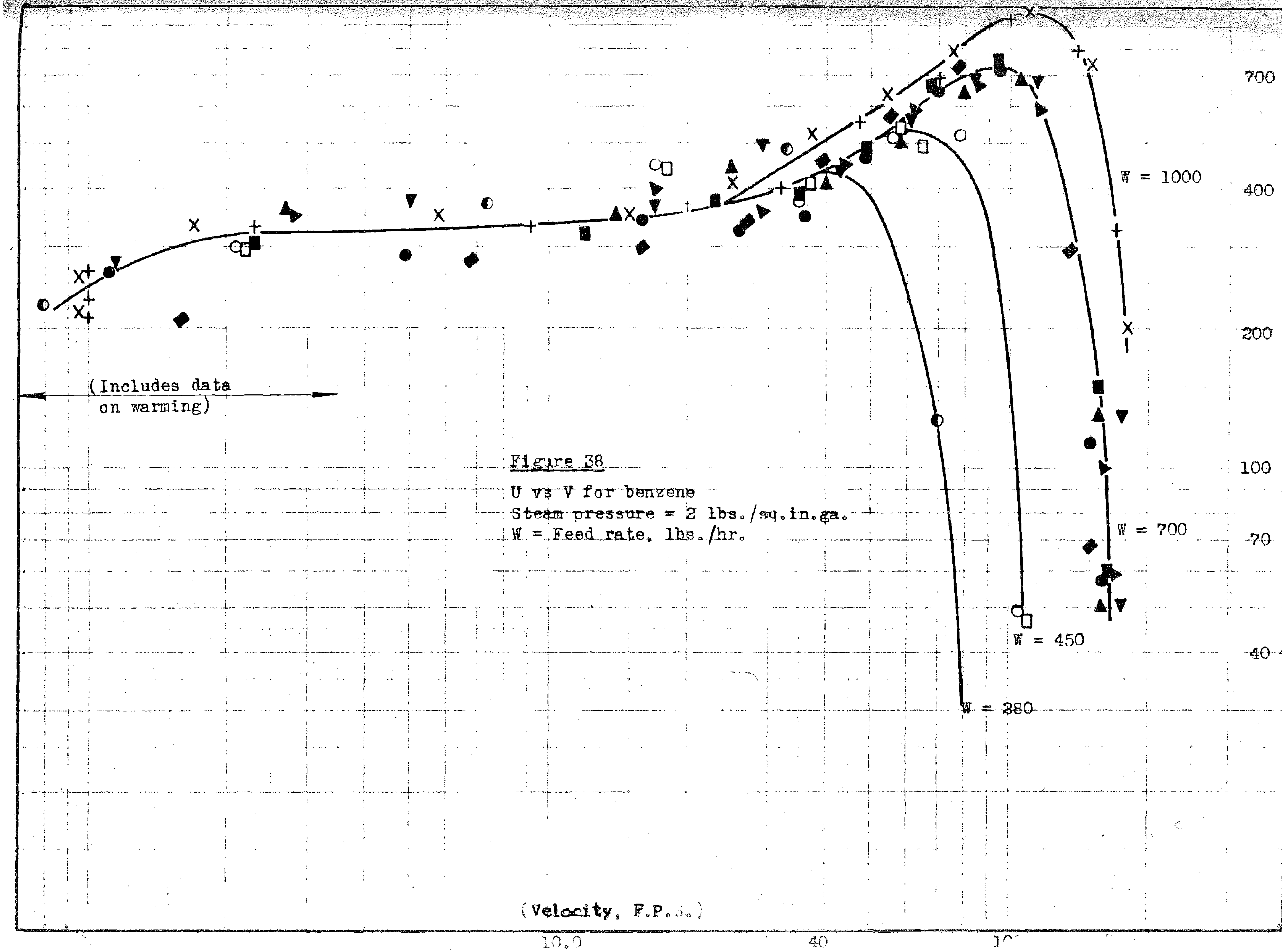


Figure 37 (c)
U vs p for benzene
Feed rate = 906 - 1080 lbs./hr.
P = steam pressure (lbs./sq.in.ga.)
Curve for P = 2.0 from Figure 36
P = 12.7
P = 2.0
P = 111
U
1000
400
100
40
P
2.0
12.7
111.
Δ t, °C.
13 - 21
22 - 32
78 - 89
(p)
0
20
40
60
80
100


(Includes data
on warming)
Figure 38
U vs V for benzene
Steam pressure = 2 lbs./sq.in.ga.
W = Feed rate, lbs./hr.
W = 1000
W = 700
W = 450
W = 380
700
400
200
100
70
40
U
10.0
40
(Velocity, F.P.S.)

DISCUSSION OF RESULTS. Section II. - Benzene Runs

The benzene runs were the first runs taken in the apparatus. Thermocouple readings were taken for the first six benzene runs. These readings indicated that the temperature drop from the steam to the tube wall was quite small, --that the overall coefficient of heat transfer was substantially the same as the film coefficient of heat transfer. Since the thermocouple reading was time-consuming, --resulting in longer test periods and increasing the possibility of changes in operating conditions during the test period, -- the thermocouple readings were discontinued in all subsequent runs. Although the thermocouples had been installed almost a year before the first data runs were taken, the thermocouples operated far more satisfactorily during the benzene runs than they did a few months later during the water runs: the intermittent heating and cooling of the steam jackets probably resulted in breaking the glass capillaries around the thermocouple leads during the time between the benzene runs and the water runs. The results of a typical run (B 5) during which thermocouple measurements were taken are presented in Figure 34.

Considerable time was consumed during Run B1 in taking condensate ~~runs.~~ readings. Subsequently, the system of taking all condensate readings simultaneously (cf. page 21) was developed. During the first five runs condensate measurements were taken twice: the check between the two readings was sufficiently good that in all subsequent runs only one set of readings was taken.

At the end of the first four runs Run B5 was taken as a check run for Run B1. The results were very discouraging, for in Run B5 the heat transfer in the early jackets was far less than prevailed during Run B1. Accordingly, Run B6 was taken as a check run for Run B3, at a steam pressure of about 110 lbs./sq.in. The check was fairly satisfactory. Then Run B7 was taken at the same steam pressure as Run B1 and B5. Run B7 checked Run B5 very satisfactorily. Subsequently, Runs B14, B1A, B4A, and B6A were taken at these same operating conditions, all runs confirming Run B5. It is accordingly concluded that in Run B1, --the first data run in the apparatus, --abnormal surface conditions were encountered which could not be duplicated in subsequent runs.

After constructing the heater, but prior to the installation of the condenser, city water from the Cambridge mains was heated in the apparatus for test purposes. Effluent was sent to the sewer. This procedure, using a continuous supply of fresh city water, resulted in forming a thick layer of scale on the inside of the copper pipes. This scale was laboriously removed by means of a 1-inch pipe brush. Consequently, the first run on benzene was made with unique liquid-side surface conditions. It is very unlikely that steam-side conditions affected the discrepancy between Run B1 and Run B5, for the steam-side surface conditions resulting from the initial test runs remained unchanged for the benzene runs.

All of the runs in Series A were taken by the same operators, and were all taken at steam pressures of around 2 lbs./sq.in.ga. However, Runs B6A and B7A were taken at the conclusion of the runs on mixtures of benzene and oil. The check between Runs B6A and B5 indicated that surface conditions remained unchanged during the benzene-oil runs. An auxiliary reason for taking the runs of Series A was to investigate the effect of feed rate at low steam pressure. Runs B2A and B5A are check runs.

During the water runs, high steam pressures were not used with high feed rates because of the abnormally large pressure drops developed in the apparatus. Benzene vapor, however, has a specific volume so much smaller than that of water vapor that far greater rates of flow of vapor could be tolerated. Accordingly, in many of the benzene runs the feed was completely vaporized and the discharged vapor was highly super-heated. The results of a typical run (B9) which displayed this condition are presented in Figure 35.

Figures similar to Figures 34 and 35 have been presented for all the benzene runs in the theses of Bryan [5] and Heroman [6]. It is sufficient for this discussion, however, merely to illustrate typical runs. Figures 34 and Figures 35 differ from the original drawings in that the actual cumulative weight per cent of the feed vaporized has been calculated and plotted, whereas the original drawings plotted the per cent of vapor which would have been present had the mixture been flashed to atmospheric pressure.

A glance at Table II reveals that many runs were taken at very low steam pressures, that there are considerable runs at very high steam pressures, but that there is a great scarcity of data on the intermediate steam pressures (10 to 50 lbs./sq.in.ga.). The work on benzene was intended primarily to high-light the field, and completely satisfactory correlation of all runs has not been obtained.

Figure 36 shows a plot of the overall heat transfer coefficient obtained in each jacket of the boiling section for all runs taken at steam pressures of around 2 lbs./sq.in.ga., --plotted against the cumulative weight per cent of the feed vaporized. Separate ordinates are used for each feed rate, and the curves are summarized in Figure 36 (a). Arrows are used to designate data taken in the first jacket after a U-bend when the per cent vapor is greater than 60%. The dotted line on the plot for feed rate of 700 lbs./hr. is drawn through these special points, and indicates the abnormally high heat flux obtained in the first jacket after the U-bend when the per cent vaporization is high. The coefficients of Run B1 are seen to check the coefficients of Runs B5, B7, etc. at high vaporization, but to be abnormally high during the early jackets. (The data for Run B1 are indicated by an asterisk, in accordance with Table II.)

In Figure 36(a) it is observed that high feed rates give high heat transfer coefficients, but that the coefficients start decreasing at smaller values of **p**, the cumulative weight per cent vapor.

Figures 37(a), (b), and (c) are plots of the rest of the benzene data. Separate plots are made for the data at each feed rate, and the curves for low steam pressure are transposed from Figure 36. Arrows are used to designate <u>all</u> data taken from Jackets 4, 7, and 10 (immediately following a U-bend).

When boiling liquids under natural convection on the outside of steam-heated pipes it is known that a moderate increase in temperature difference causes an increase in the coefficient, but that further increase causes the coefficient to go through a maximum and subsequently to decrease to very low values. When only very small coefficients are obtained despite the large temperature difference the tube is described as "vapor bound" since it can be seen that a film of vapor insulates the tube from the liquid. Similarly, it is noted in Figure 37 that when steam pressures greater than 60 lbs./sq.in.ga. are used the heat transfer coefficients are reduced to well below 100. So far as is known, this is the first time that vapor binding has been reported for boiling liquids inside pipes at high temperature differences, although such an effect might have been predicted. Figure 37(b) also indicates that increasing steam pressures increase the tendency of the pipe wall to become vapor bound at high percentage vaporization.

Although the data is limited in amount, the highest heat transfer coefficients during the early stages of boiling were obtained when a steam pressure of about 13 lbs./sq.in.ga. was used, corresponding to a temperature difference of from 22° to 26°C. Bringardner[4], boiling benzene outside of a steam-heated pipe, obtained maximum values of the heat transfer coefficient at a temperature difference of 28°C. At 28°C., Bringardner obtained coefficients as high as 1900 B.t.u./(hr.)(sq.ft.)(°F.), whereas the maximum coefficient for boiling benzene as obtained in this investigation was only 1200 B.t.u./(hr.)(sq.ft.)(°F.). Figure 37(b) indicates that when temperature differences equal to or greater than the critical temperature difference are being used the effect of increasing the cumulative per cent vapor is not beneficial. An inconsistency with this observation is found in Figure 37(c) for the points at 111 lbs./sq.in.ga. steam pressure, where a pronounced "hump" in the curve is noted.

A wealth of data testifies to the fact that the U-bend effect (discussed with the results of the water runs) exists at high cumulative per cent vapor, regardless of the steam pressure. In Figure 35 it is seen that no U-bend effect was obtained in Jacket 10 when the benzene vapor was dry and superheated.

The data on water and benzene agree on the salient points: (1) at low temperature differences the heat transfer coefficients increase to a certain maximum and then decrease as the cumulative per cent vaporization increases; (2) a moderate increase in steam pressure (or temperature difference) causes higher heat transfer coefficients during the early stages of boiling; (3) a pronounced and beneficial U-bend effect is consistently noted when the cumulative vaporization is high and the tube is therefore vapor bound. Quantitatively, discrepancies exist between the results of the benzene and the water runs which are not easily explained. The results of the water runs showed no effect of feed rate; the benzene runs at low steam pressure indicated a definite effect of feed rate. The water runs indicated that temperature difference had minor effect on the heat transfer coefficients at high cumulative vaporization; the benzene runs indicated that an increase in steam pressure increased the vapor binding at high cumulative vaporization. Undoubtedly, a certain amount of these discrepancies may be due to differences in such physical properties as interfacial (surface) tension of the liquid, specific volume of the vapor, etc. The possibility exists that the absence of data on water at simultaneously high steam pressures and rates of flow has resulted in conclusions on water that are not completely justified. Most likely, however, the differences may be largely due to the higher boiling point of water (causing smaller temperature differences when a given steam pressure is used) and the much larger critical temperature difference at which maximum heat transfer coefficients for boiling water would be obtained. Figure 33 shows that Oliver (10) went to overall temperature differences as high as 60°C. without becoming vapor bound when boiling water; in only the last jacket of Run W12 was a temperature difference as high as 60°C. reached during the water runs. If data were obtained on a pure liquid boiling at a higher temperature than benzene heat transfer coefficients could be obtained at small temperature differences well below the critical temperature difference, and the results should be more strictly comparable with the results obtained in the water runs. Thus, toluene has an atmospheric boiling point of 111°C., and a considerable range of steam pressures could be covered without the development of vapor binding due to excessive temperature differences.

Figure 38 shows the same coefficients that appear in Figure 36, --plotted against the average velocity in the jacket instead of the average cumulative weight per cent vapor. Some additional data is included for low velocities where vaporization had not yet begun. The correlation is an improvement over the family of curves in Figure 36(a). Attention is called to the comparison of Figure 38 with Figure 32, which is a similar plot for the data obtained in the water runs. In each plot all the coefficients for a particular (low) steam pressure fall on one general curve with the exception of the vapor binding that occurs at various velocities for the various feed rates.

Treatment of Warming and Preheating Data:

No general correlation is presented of the heat transfer coefficients for pre-heating the feed. The warming of liquids in tubes where the pipe-wall is maintained at a temperature above the boiling point of the feed results in vaporization of some of the feed in the film next to the wall, followed by subsequent re-condensation of the vapor in the cold main body of the liquid. The literature is not fully developed on this phenomena and it is felt that this phase of the thesis must be reserved for further study. Suffice it to state that the heat transfer coefficients obtained for preheating the benzene feed ran from two to four times as large as those predicted by the recognized formulas for the warming of liquids in turbulent flow inside of horizontal pipes. These abnormally high coefficients may be due (1) to increased turbulence resulting from vaporization in the superheated films adjacent to the hot wall, and (2) to the turbulence resulting from the surging flow inside the tubes, superimposed upon the turbulence which would normally exist in steady flow.

The benzene runs offer the only opportunity to study data on the superheating of vapor. The only difficulty here is that these coefficients are relatively small, accompanied by small temperature differences and small heat flux. At small heat flux the observed rate of heat transfer is most inaccurate due both to the intermittent discharge of the steam traps and to the relatively large percentage error made in estimating the heat losses. It is, then, sufficient to compare the observed superheating coefficients with the theoretical coefficients predicted by the approximate equation for warming common gases:(14)

$$h = 0.0144\ C_p G^{0.8}/D^{0.2}$$

The summary of this comparison is given in Table II (a). Considering the low heat fluxes encountered, the comparison is quite satisfactory. Note the absence of any effect of the U-bend preceding Jacket 10.

TABLE II (a)

Overall Heat Transfer Coefficients for Superheating Dry Benzene Vapor Compared with Coefficients Predicted by Equation 126, Page 113. P.C.E.(14)

Run	W(lbs./hr.)	Jacket	$(\Delta t)_{av}$ °C.	$U_{obs.}$	$U_{pred.}$	$\frac{U_{obs.}}{U_{pred.}}$
8	750	11	20	65	82	0.79
		12	17 1/2	58	82	0.71
9	710	9	34	76	78	0.97
		10	27	66	78	0.85
		11	20	57	78	0.73
		12	13	75	78	0.96
10	689	10	37	71	76	0.93
		11	26 1/2	68	76	0.89
		12	19	107	76	1.41
5A	453	10	16	63	55	1.15
		11	9	111	55	2.02
		12	6	28	55	0.51
2A	434	10	14 1/2	69	53	1.30
		11	9 1/2	41	53	0.77
		12	4	69	53	1.30
3A	282	7	14 1/2	27	38	0.71
		8	11 1/2	15	38	0.40
		9	9 1/2	18	38	0.48
		10	7	48	38	1.26
		11	4	42	38	1.11
		12	2 1/2	22	38	0.58

R E S U L T S   A N D   D I S C U S S I O N

Section III. - Heat Transfer to Boiling Benzene - Oil Mixtures

Results. Section III. - <u>Heat Transfer to Boiling Benzene-Oil Mixtures.</u>

The operating conditions for the runs on benzene-oil mixtures are summarized in Table III. The symbols listed in Table III are used in all subsequent correlations of the data on boiling mixtures of benzene and oil, - specifically, Figures 40, 41, 45, and 48.

The results of a typical run on a benzene-oil mixture are presented graphically in Figure 39. Analagous to Figures 1 - 14 for water, Figure 39 is a plot of the data of benzene-oil run BO - 22, showing curves for static fluid pressure (P), cm.Hg.ga.; condensing steam temperature (T_{st}), ^{o}C.; fluid temperature (T_{fl}), ^{o}C.; heat flux (q/A), B.t.u./(hr.)(sq.ft.); overall heat transfer coefficient (U), B.t.u./(hr.)(sq.ft.)(^{o}F.); cumulative weight per cent of the feed vaporized (p); and composition of the residual liquid phase (x), weight per cent benzene; plotted versus tube length, expressed in terms of jacket numbers.

Figure 40 is a plot of the overall heat transfer coefficient (U) versus the cumulative weight per cent of the feed vaporized (p) for all data on boiling benzene oil mixtures at a feed rate of 1000 lbs./hr. The data is presented in the form of three separate plots, A, B, and C, representing data based on the first, second, and third jackets after the U - bend, respectively. Lines of approximately constant feed composition (f) are drawn through the points on each of these plots. The lines from Figures 40 (B) and 40 (C) are transposed to Figures 40 (D) and 40 (E), respectively, and dotted lines of constant residual liquid phase composition are sketched in, based upon the stoichiometric relationships between feed composition, per cent vaporization, and residual liquid composition.

Figure 41 is a plot of the apparent overall heat transfer coefficient for the entire boiling section of the runs on benzene-oil mixtures, based upon and plotted against the final temperature difference obtained during the run. The broken lines in Figure 41 represent plots at two different temperature differences (30°C. and 50°C.) of the film heat transfer coefficient (h) versus the liquid composition in the still, as obtained in a natural convection steam-heated apparatus, boiling occurring on the outside of a horizontal nickel-plated copper tube of 0.54 inch O.D.[4]

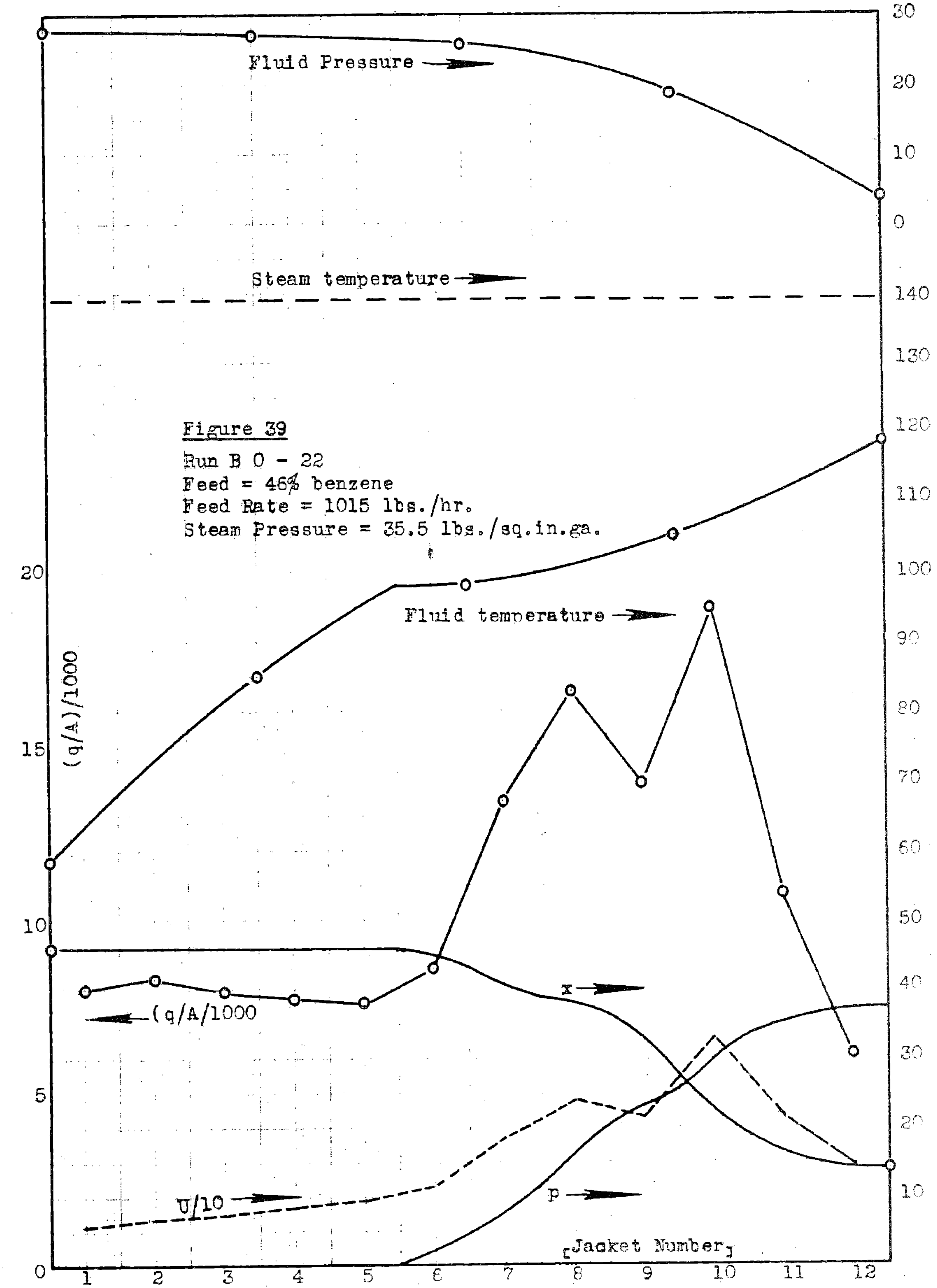
Figure 42 is a summary plot of the overall heat transfer coefficient (U) versus the cumulative weight per cent of the feed vaporized (p) as obtained in the previous sections for a feed rate (W) equal to 1000 lbs./hr. The curve for water is transposed from Figure 32; for benzene, from Figure 37 (c); for constant liquid phase composition of benzene-oil mixtures, from the dotted lines of Figure 40 (E).

Table III

Runs on Benzene - Oil Mixtures

Symbol	Run No.	Feed Rate lbs./hr.	Wt. of Feed	benzene Product	Steam Pressure lbs./sq.in.	Initial Feed Temp.°C.	Heat* Transfer B.t.u./hr.	Wt. % of Feed Vaporized
+	BO-1	702	90.5	60.5	1.8	53.0	105,000	76
×	BO-2	775	93.5	75.5	1.9	55.5	115,00	73.5
✳	BO-6	498	88.5	10.5	26.0	46.0	102,000	89
⊗	BO-5	545	70.5	23.0	9.1	38.5	83,000	71
⊗	BO-24	884	76.0	7.0	74	56	169,000	74
◑	BO-23	996	62.0	13.0	37	52	145,000	56
◐	BO-21	990	56.5	4.0	120	66	170,000	55
○	BO-22	1015	46.0	14.0	35.5	58.5	111,000	37
□	BO-17	1020	45.0	8.0	53.3	65.5	123,000	40
△	BO-16	1011	41.0	6.0	78.3	70	123,000	37
▽	BO-20	1050	43	4.0	120	70	143,000	40.5
●	BO-7	984	33.5	18.5	22	59	69,000	18.5
■	BO-13	1080	25.7	15.4	36	68	62,000	12
◆	BO-14	1093	31.3	10.7	52	68	86,000	23
▲	BO-15	970	30.0	7.5	80	74	101,000	24
▼	BO-19	1050	29.0	4.0	119	79	113,000	26
▬	BO-8	1008	21.0	17.0	38.5	68.5	47,000	5
▮	BO-11	1078	16.0	9.0	70	78.5	59,000	8
/	BO-18	1025	13.0	6.0	119	85	75,000	7.5
\	BO-12	1065	12	6.0	120	87.5	65,000	6.5

* Based on steam condensate measurements.



Fluid Pressure
Steam temperature
Figure 39
Run B 0 - 22
Feed = 46% benzene
Feed Rate = 1015 lbs./hr.
Steam Pressure = 35.5 lbs./sq.in.ga.
Fluid temperature
(q/A)/1000
(q/A/1000
x
U/10
p
Jacket Number
[cm.Hg.ga.]
[%, °C., or U/10]
0
5
10
15
20
1
2
3
4
5
6
7
8
9
10
11
12
30
20
10
0
140
130
120
110
100
90
80
70
60
50
40
30
20
10

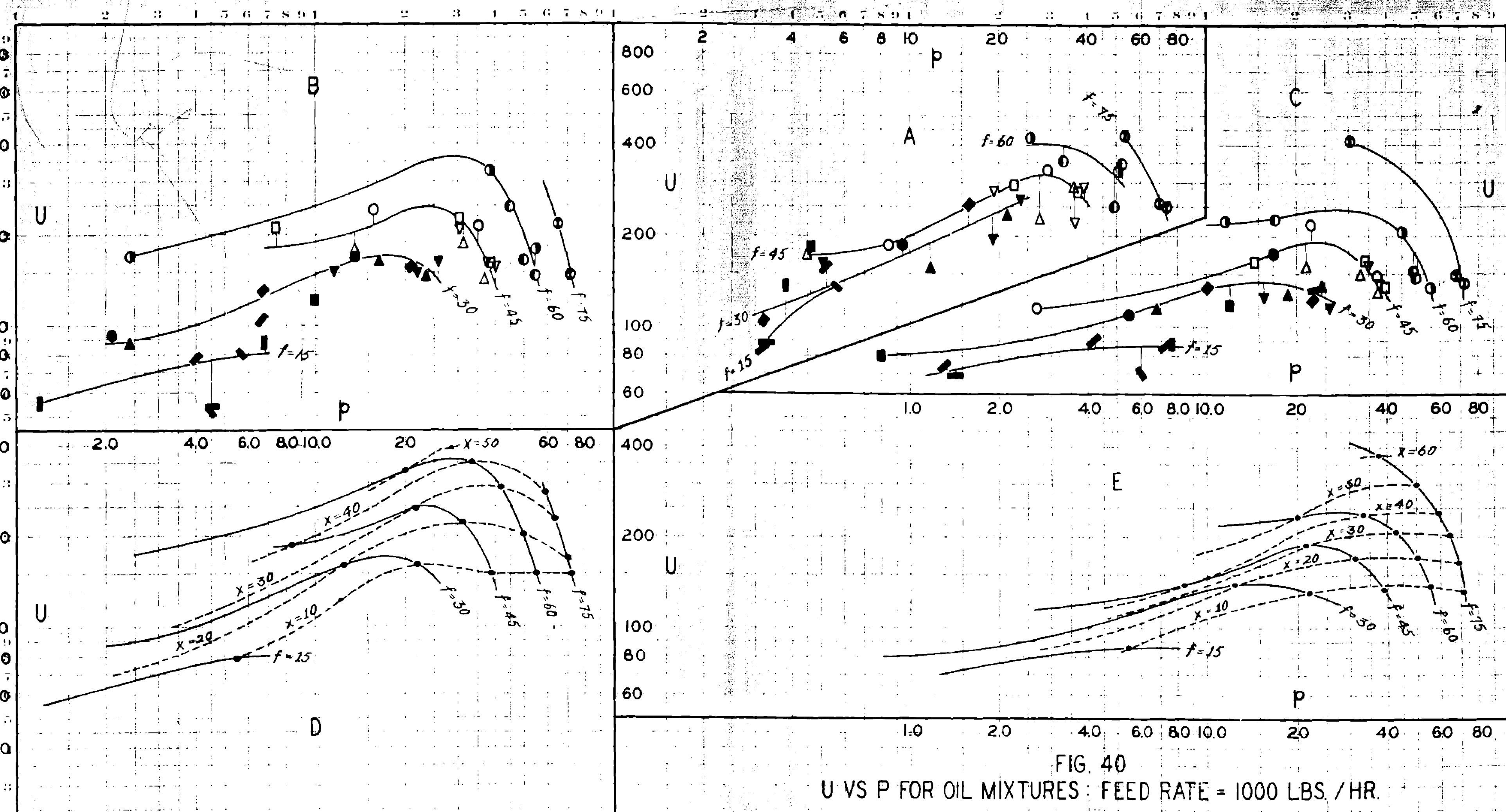


FIG. 40
U VS P FOR OIL MIXTURES : FEED RATE = 1000 LBS. / HR.

A = 1 ST JACKET AFTER BEND
B AND D = 2 ND " " "
C AND E = 3 RD " " "

f = % BENZENE IN FEED
x = % BENZENE IN LIQUID PHASE

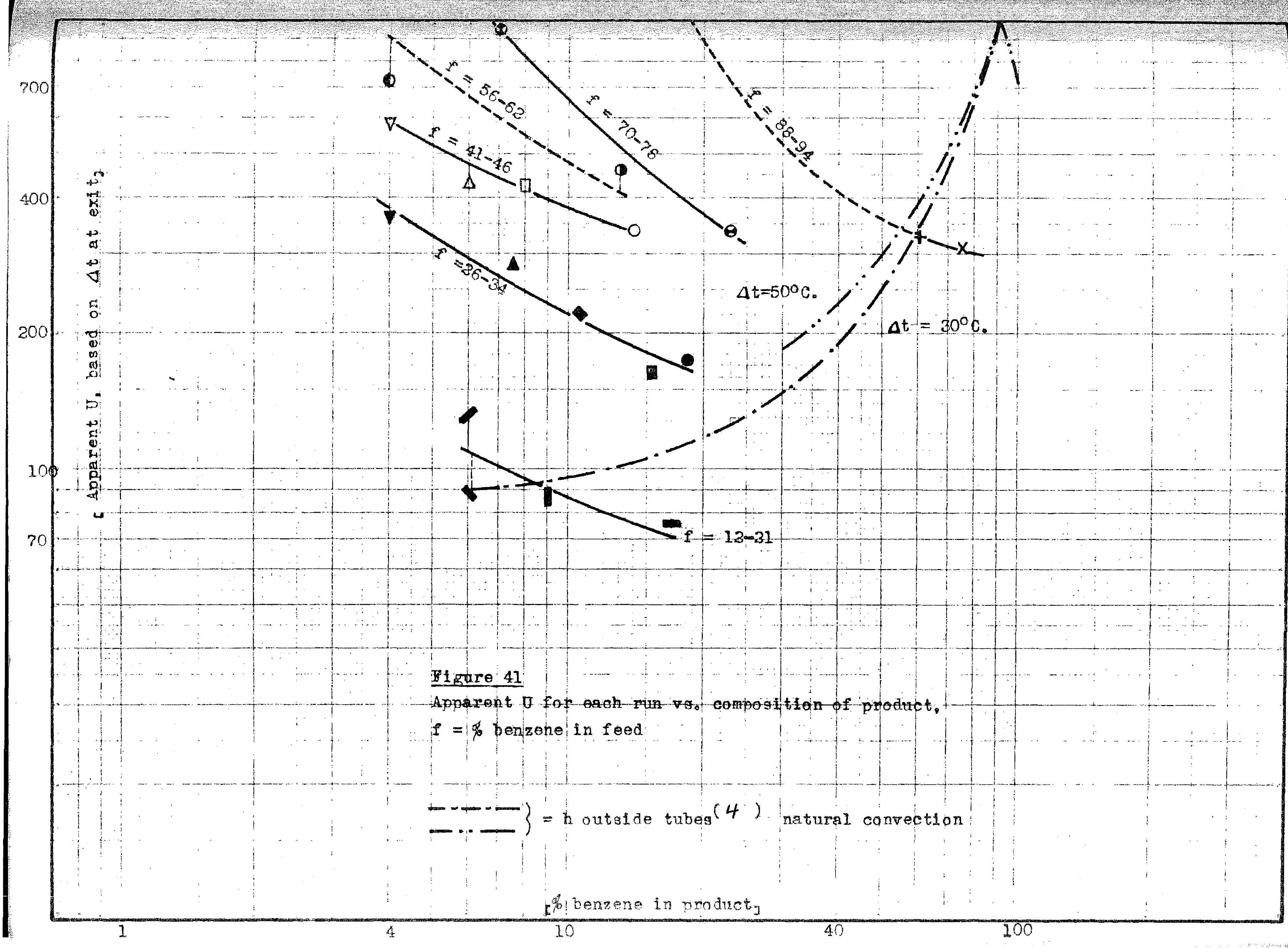


Figure 41

Apparent U for each run vs. composition of product,

f = % benzene in feed

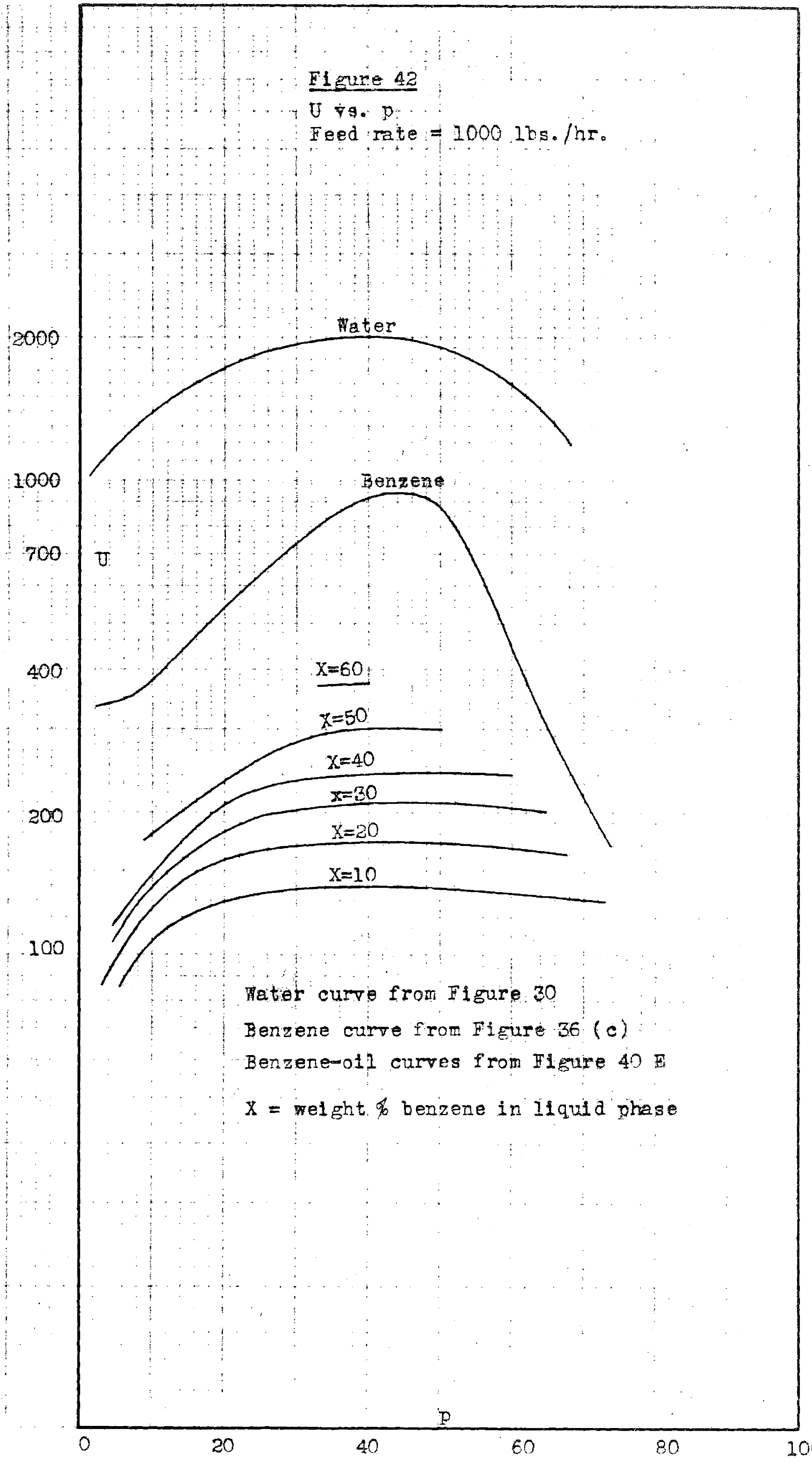


Figure 42
U vs. p
Feed rate = 1000 lbs./hr.
Water
Benzene
X=60
X=50
X=40
x=30
X=20
X=10
2000
1000
700
400
200
100
U
p
0
20
40
60
80
100
Water curve from Figure 30
Benzene curve from Figure 36 (c)
Benzene-oil curves from Figure 40 E
X = weight % benzene in liquid phase

40 E

DISCUSSION OF RESULTS. Section III. - Benzene-oil Runs

The runs taken on mixtures of benzene and oil were characterized by difficulty in obtaining satisfactory heat balances at the relatively low heat flux encountered. Out of a series of twenty-four runs, four were discarded solely because of unsatisfactory heat balances. A series of runs on pure oil were taken but the results were not presented. For these runs on pure oil the heat flux was so extremely small that heat losses from the apparatus ruined the precision of determining rates of heat transfer.

Subsequent to Run BO-6 all runs were taken at a feed rate of close to 1000 lbs./hr. The presence of the oil composition as a variable necessitated keeping some other variable constant if a correlation was to be attempted.

The results of a typical run on a mixture of benzene and oil are presented graphically in Figure 39. The oil used in all runs was a finished propane-treated bright stock, of viscosity S.A.E. 50, from the Baytown Refinery of the Humble Oil and Refining Company, Houston, Texas.

In the runs on pure benzene and pure water it was noted that the heat transfer coefficients frequently were higher in the later boiling jackets than in the jackets where boiling was just commencing. Correlating on a temperature difference basis proved unsatisfactory and it was concluded that this increase in coefficient, while partially due perchance to the increasing temperature difference, was also created by the increasing amount of vapor formed. The outstanding contribution of the benzene-oil runs is the verification of this hypothesis. While the decreasing static pressure through the apparatus tends to lower the boiling point of the fluid, the boiling out of the benzene leaves behind a liquid containing larger percentages of oil than the feed and having a higher boiling point. The result is that the temperature drop between the condensing steam and the boiling liquid continually decreases. Yet not only the heat transfer coefficient but even the heat flux is larger in the later jackets than in the jackets where boiling first starts. The conclusion is inescapable that this increase in rate of heat transfer must be a result of the changing amounts of vapor present.

Figure 40 is a plot of the overall heat transfer coefficient obtained in each jacket in the boiling section of all runs having a feed rate of about 1000 lbs./hr., plotted against the cumulative weight per cent of the feed vaporized. It was not only necessary to segregate the first jacket after the U-bends, but it was also noted that the second jacket after the U-bends gave coefficients somewhat larger than did the third jackets after a U-bend, for a given feed and a given cumulative vaporization. Hence, three separate plots (A, B, and C) are made, representing the data on the first, second, and third jackets after the U-bends. Lines of constant feed composition are drawn through the points. It should be borne in mind that a line representing constant feed composition involves not only a smaller temperature difference as the per cent vaporization increases, but also an increasing amount of oil in the residual liquid phase.

Analysis of Figures 40 A, B, and C is instructive. The correlation of data taken in the first jacket after the U-bend is relatively poor, but indicates a smaller effect of feed composition than is true of the other jackets. Especially for feeds containing small amounts of benzene, the data are quite scattered. It is not necessary to have 60% or more of the feed vaporized in order to get increased vaporization in the first jacket after the U-bends. Since a feed containing 30% benzene cannot be more than 30% vaporized, the curves stop at various abscissa values depending upon the feed strength. No discrimination has been made on the basis of temperature difference, although the results on pure liquids indicated an appreciable effect of temperature difference. The initial temperature difference varied from 18° to 64°C.

The minor effect of temperature difference in these runs on mixtures of benzene and oil is probably due to the fact that the heat transferred not only goes into vaporizing some of the benzene but also into heating the residual liquid and the vapor. Since sensible heat transfer is less dependent on temperature difference it is rational to believe that if large percentages of the heat transferred are used merely to warm up the fluid the effect of temperature difference on the rate of heat transfer should be minimized. Thus, the heat transfer in this apparatus is rather different in mechanism from the heat transferred by natural convection to liquids boiling at constant temperature outside of horizontal tubes.

The large rates of heat transfer obtained in the first jacket after the U-bend when boiling pure liquids was explained by the hypothesis that the walls of the tube became starved for liquid, the liquid phase being carried as a spray down the center of the tube until the U-bend whirled the spray against the wall. This effect was only encountered with pure liquids when large percentages of the feed had been vaporized and also was only encountered in the first jacket after the U-bend. With the benzene-oil mixtures a similar effect is encountered at low percentage vaporization and a residual effect carries over into the second jacket after the U-bend. The situation in the benzene-oil runs is somewhat different. As the mixture moves down the tube the benzene is boiled out of the liquid near the wall, leaving behind a liquid phase which is richer in oil than the main body of the stream moving through the center of the pipe. The U-bend upsets this condition at the end of the pass by whirling the mixture around, mixing it again, and allowing a representative sample of the stream to contact the walls of the first jacket after the U-bend. Whereas the jacket just before the U-bend was starved for benzene because most of the benzene was in the center of the tube, the first jacket after the U-bend has the benzene evenly distributed across the cross-section of the tube. With pure liquids, the spray thrown against the wall immediately after a U-bend was quickly evaporated; with benzene-oil mixtures the uniform distribution of benzene right after the U-bend changes gradually to the non-uniform distribution existing at the end of the pass, leaving an higher coefficient (other things being equal) in the second jacket after the U-bend than in the third jacket after the U-bend.

It is noted in Figures 40 A, B, and C that the feed composition is an unsatisfactory correlating variable in a plot of U vs. p because the maximum abscissaes for each curve is limited by the feed composition. However, if a certain per cent is vaporized of a feed containing a certain amount of benzene, one can stoichiometrically calculate the per cent benzene in the residual liquid. The curves of Figures 40B and 40C are transposed to Figures 40D and 40E, respectively, and dotted lines of constant composition of residual liquid phase are placed on the diagram as tie lines between the solid transposed lines. These dotted lines indicate the true effect of per cent feed vaporized, for they represent lines of constant liquid composition exposed to varying amounts of benzene vapor.

In an evaporator of the submerged-tube (Swenson) type the temperature of the pool is at the boiling temperature of the withdrawn product; as the lean feed enters the pool it is rapidly mixed with the contents of the pool as a result of the turbulence set up by the boiling action. The big advantage of boiling inside tubes is the avoidance of this mixing action. When boiling outside of tubes the heat transfer coefficient is based upon the difference between the steam temperature and the boiling point of the product; for comparison with natural circulation evaporators the heat transfer coefficients for boiling inside of tubes should also be based upon the difference between the steam temperature and the temperature of the product. In Figure 41 are shown the apparent heat transfer coefficients for all the runs on benzene-oil mixtures, based upon the final temperature difference and plotted against the composition of the product. Lines of constant feed composition are drawn through the points. These apparent coefficients were determined by taking the total heat transferred after boiling had commenced and dividing in succession by the area of the boiling section and the final temperature difference. The point for Run BO-6 (feed composition equal 88.5% benzene) had so large a coefficient that it could not be included on the plot, but the curve for very lean feed was drawn with this point in mind. On the same plot are shown (by broken lines) the data of Bringardner (4) obtained by boiling these same benzene-oil mixtures outside of a submerged steam-heated nickel-plated horizontal tube. Figure 41 indicates that if a product containing 10% benzene is to be obtained from a feed containing 12% to 21% benzene substantially the same heating area will be needed whether boiling occurs inside or outside of the tube; but that if the feed contains 41% to 46% benzene, boiling inside of the tube will require only about one-quarter as much heated area. Of course, the use of a series of pools of progressively varying composition will greatly reduce this advantage enjoyed by the forced circulation apparatus.

In order to compare the results obtained in the benzene-oil runs with the results obtained in the runs on water and on benzene, typical curves from each group of runs were transposed to Figure 42. Figure 42 shows: for water, the curve common to all the water runs at a steam pressure of 10 lbs./sq.in.ga., and therefore valid for a feed rate of 1000 lbs./hr.; for benzene, the curve obtained at a steam pressure of 2 lbs./sq.in. and a feed rate of 1000 lbs./hr.; for benzene-oil mixtures, the curves obtained for constant composition of residual liquid at a feed rate of 1000 lbs./hr., based upon data from the third jacket after the U-bend. These curves are conservative except for the case of extreme steam pressures: the water and the benzene coefficients can be increased at low values of cumulative vaporization by moderate increases in steam pressure, and at high values of cumulative vaporization by considering the first jacket after a return bend; the benzene-oil coefficients can be increased throughout their range by considering the first or the second jacket after a return bend. The data are not compared at the same temperature differences.

It must be borne in mind that the results are based upon flow through a pipe of one-inch inside diameter. While tube diameter is not important when boiling liquids outside of horizontal pipes, it may be very critical when boiling liquids inside of pipes because of its possible effect on the character of the flow. Thus, in a two-inch diameter pipe stratification of vapor and liquid might persist to far greater cumulative vaporization, whereas in a half-inch pipe appreciable stratification might never occur.

But little success has been obtained in predicting the coefficients of heat transfer to liquids boiling in natural convection outside of pipes. It is not to be expected that data on water and benzene-oil mixtures could be used to predict the actual magnitude of the coefficients which would be obtained if other fluids were used in the apparatus. Using the apparatus of this thesis, however, a long range investigation (say, of five or six years) should enable one to predict quite accurately the heat transfer coefficients to be obtained in a one-inch copper pipe under any conditions of flow of a fluid for which the heat transfer coefficients in a simple natural convection "tea-kettle" had been obtained.

R E S U L T S   A N D   D I S C U S S I O N

Sect. IV. -- Studies on Pressure Drop

RESULTS. Sect. IV --- Studies on Pressure Drop.

The Symbols used in the plots of this section are the same symbols used in Tables I, II, and III of the earlier sections. Figs. 43 and 46 use the symbols from Table I, p.39; Figs. 44 and 47 use the symbols from Table II, p.95; and Figs. 45 and 48 use the symbols from Table III, p.117.

Figs. 43, 44, and 45 represent plots of the average overall heat transfer coefficient in each pass plotted versus the observed pressure drop (cm. Hg.) in the pass, for the runs on water, benzene, and benzene-oil mixtures, respectively. The length of one pass is about 13 feet, including the U-bend. Two lines are drawn on Fig.44, corresponding to data taken at low steam pressures (about 2 lbs./sq.in. ga.) and high steam pressures (greater than 60 lbs./sq.in.ga.). These lines are transposed as dotted lines in Fig. 45. Two solid lines are drawn on Fig. 45 corresponding approximately to data based on lean and rich benzene-oil feed strengths.

Figs. 46 to 48 are plots of the indicated Fanning friction factor (f) plotted against the cumulative weight per cent vapor at the <u>entrance</u> to the pass $(p)_{in}$, for the runs on water, benzene, and benzene-oil mixtures, respectively. Two lines are drawn on Fig. 47 corresponding to data taken at high and at low steam pressures.

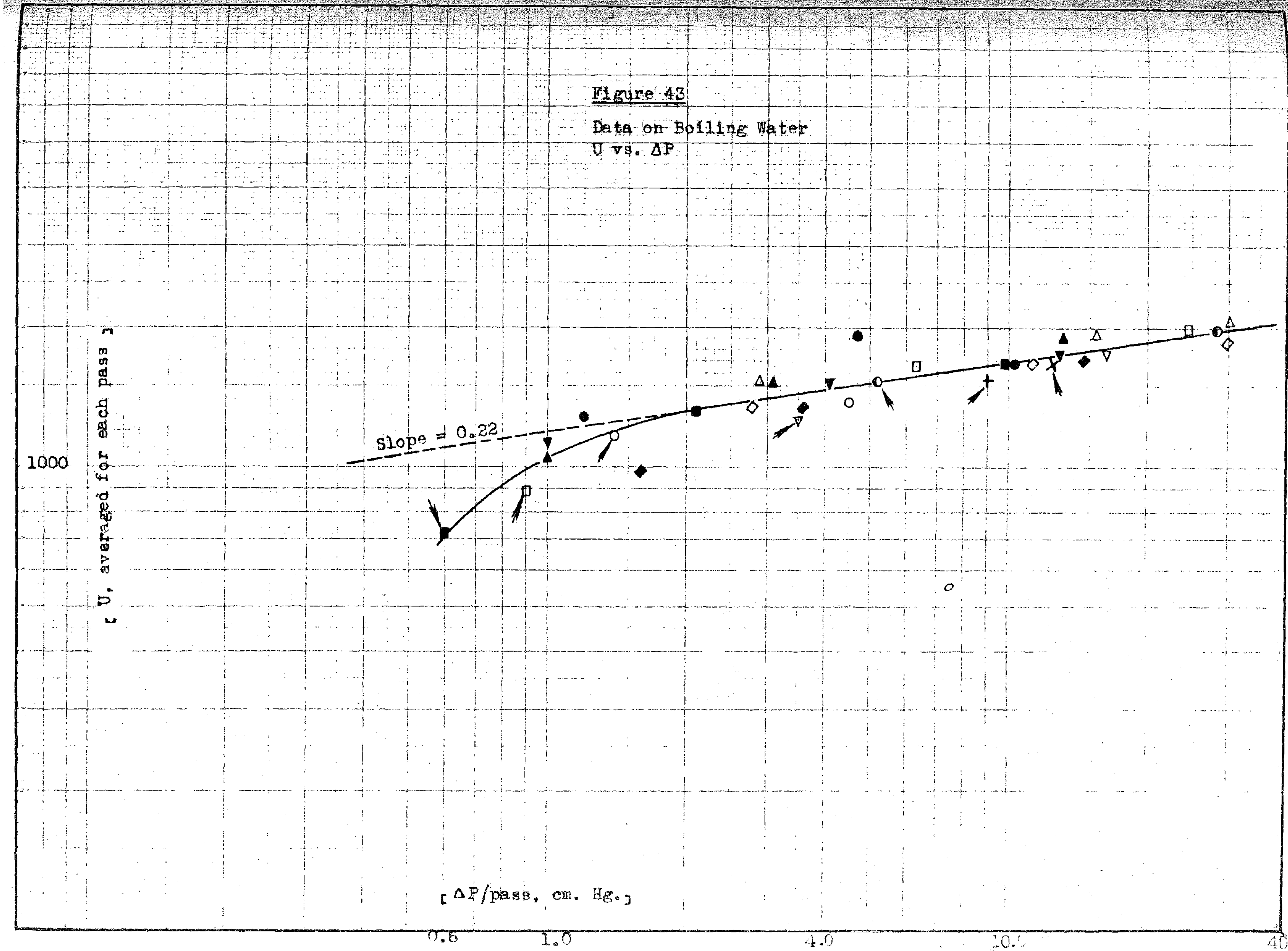


Figure 43
Data on Boiling Water
U vs. ΔP
Slope = 0.22
1000
[U, averaged for each pass]
[ΔP/pass, cm. Hg.]
0.6
1.0
4.0
10.0
40

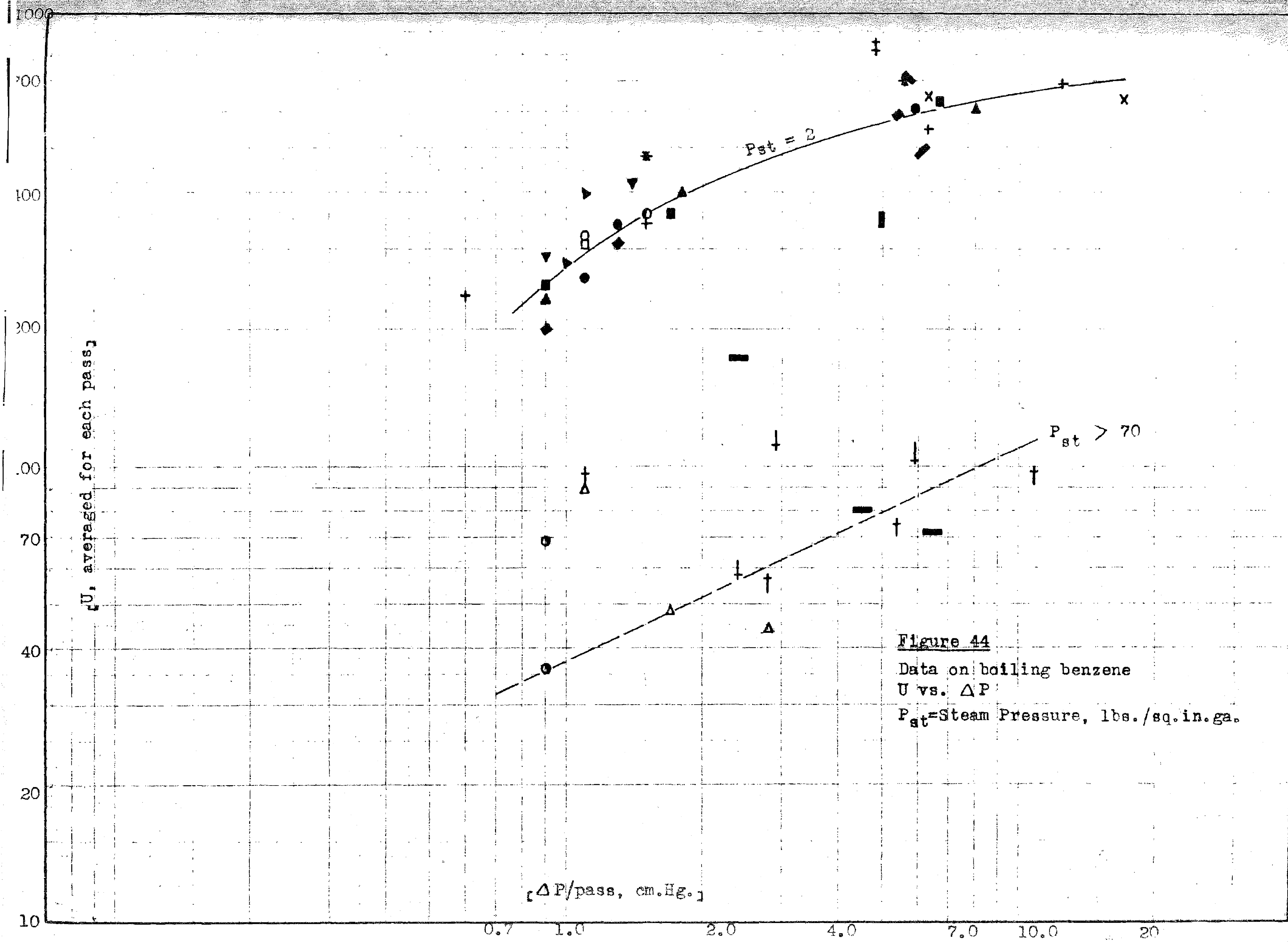


Figure 44

Data on boiling benzene
U vs. ΔP
P_{st}=Steam Pressure, lbs./sq.in.ga.

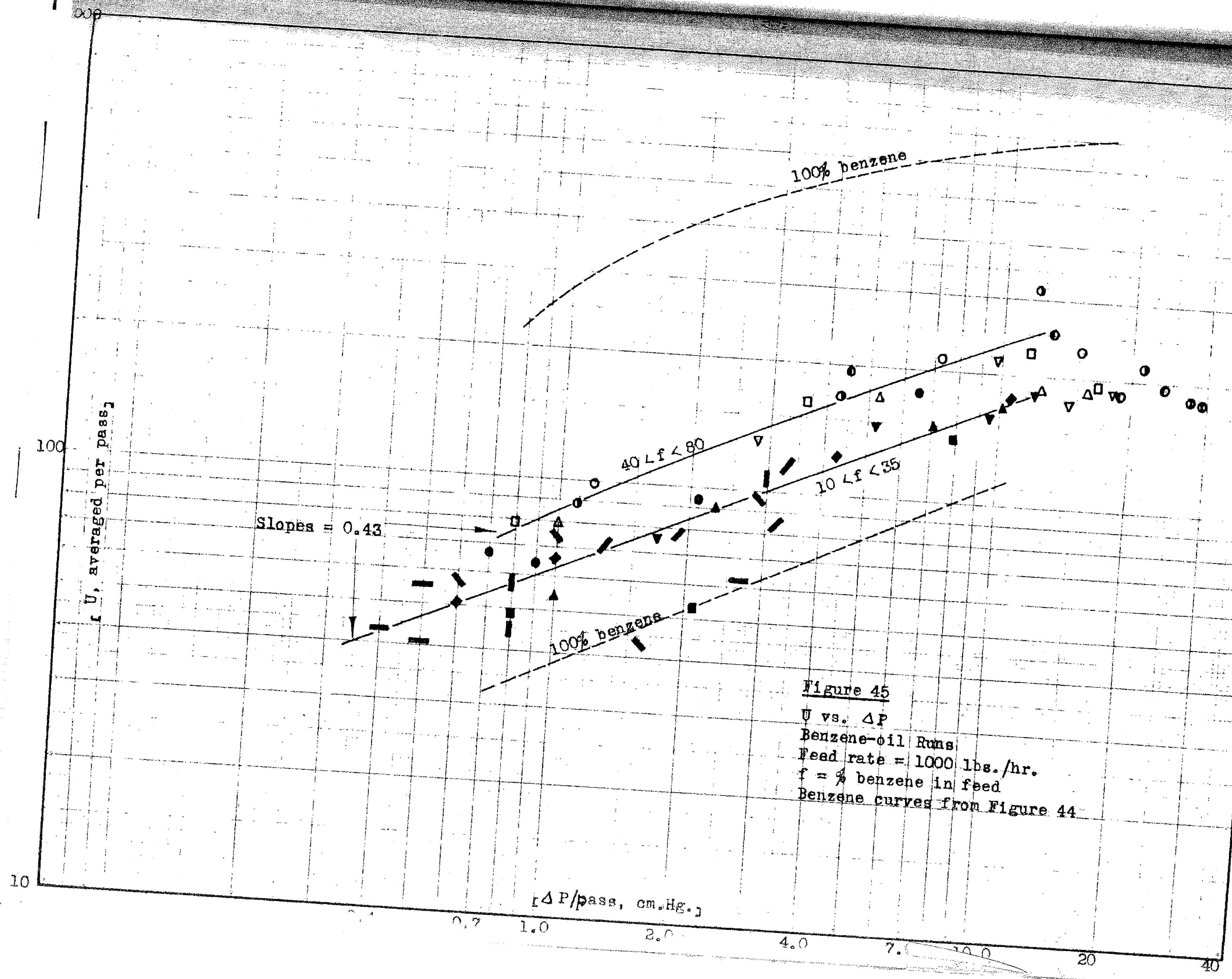


Figure 45

U vs. ΔP
Benzene-oil Runs
Feed rate = 1000 lbs./hr.
f = % benzene in feed
Benzene curves from Figure 44

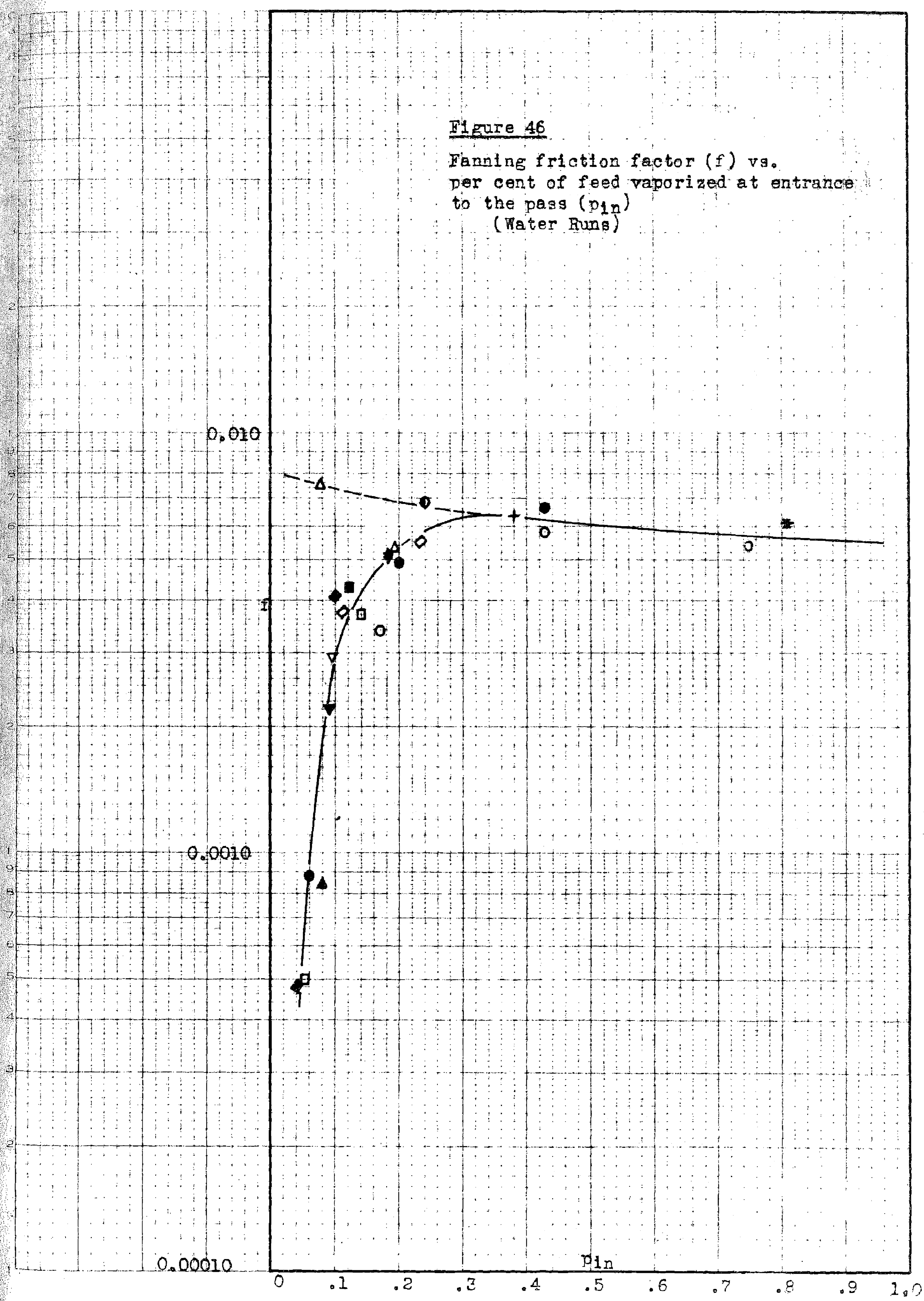


Figure 46
Fanning friction factor (f) vs.
per cent of feed vaporized at entrance
to the pass (p_{in})
(Water Runs)
0.010
0.0010
0.00010
f
p_{in}
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0

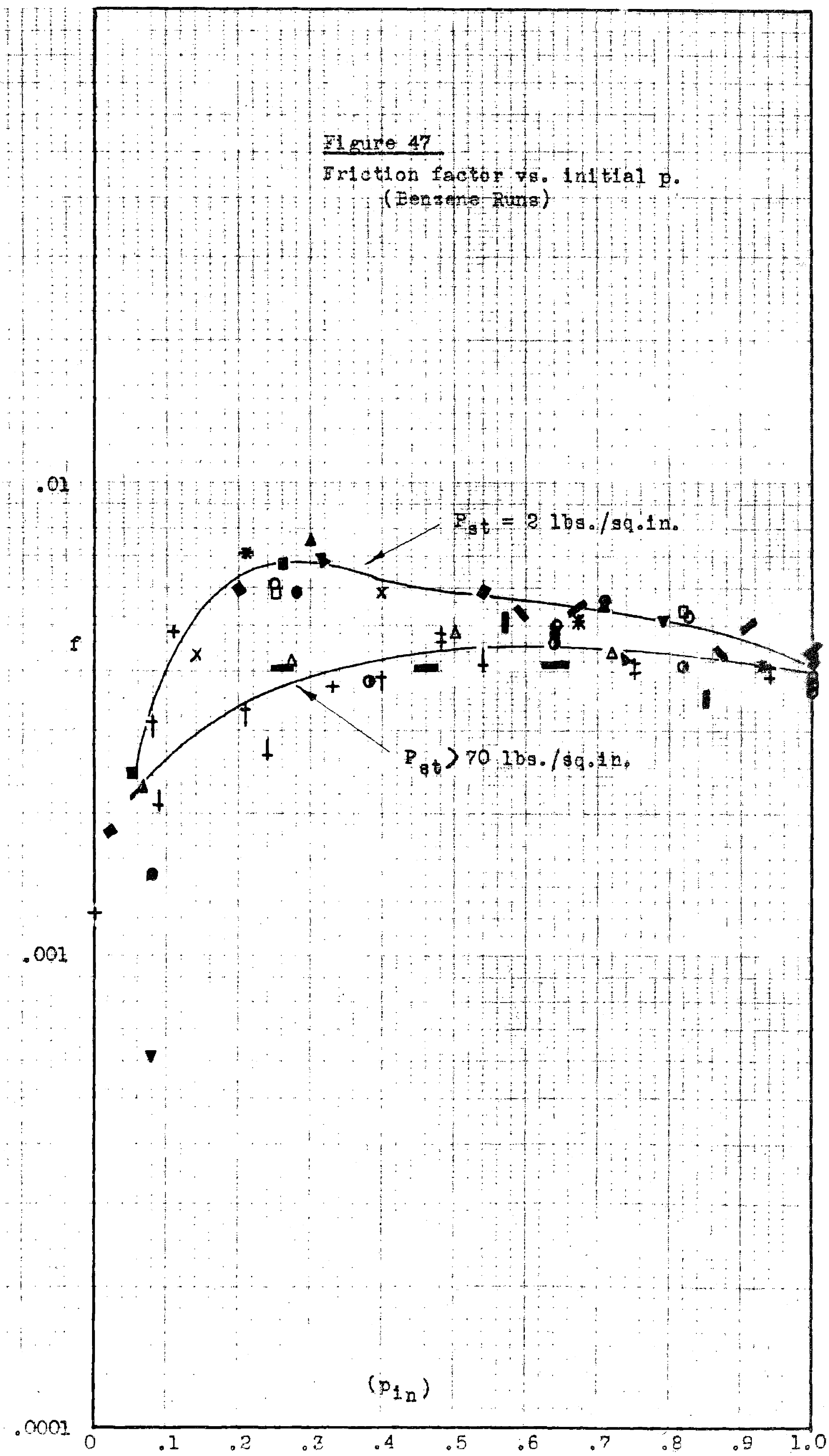


Figure 47

Friction factor vs. initial p.
(Benzene Runs)

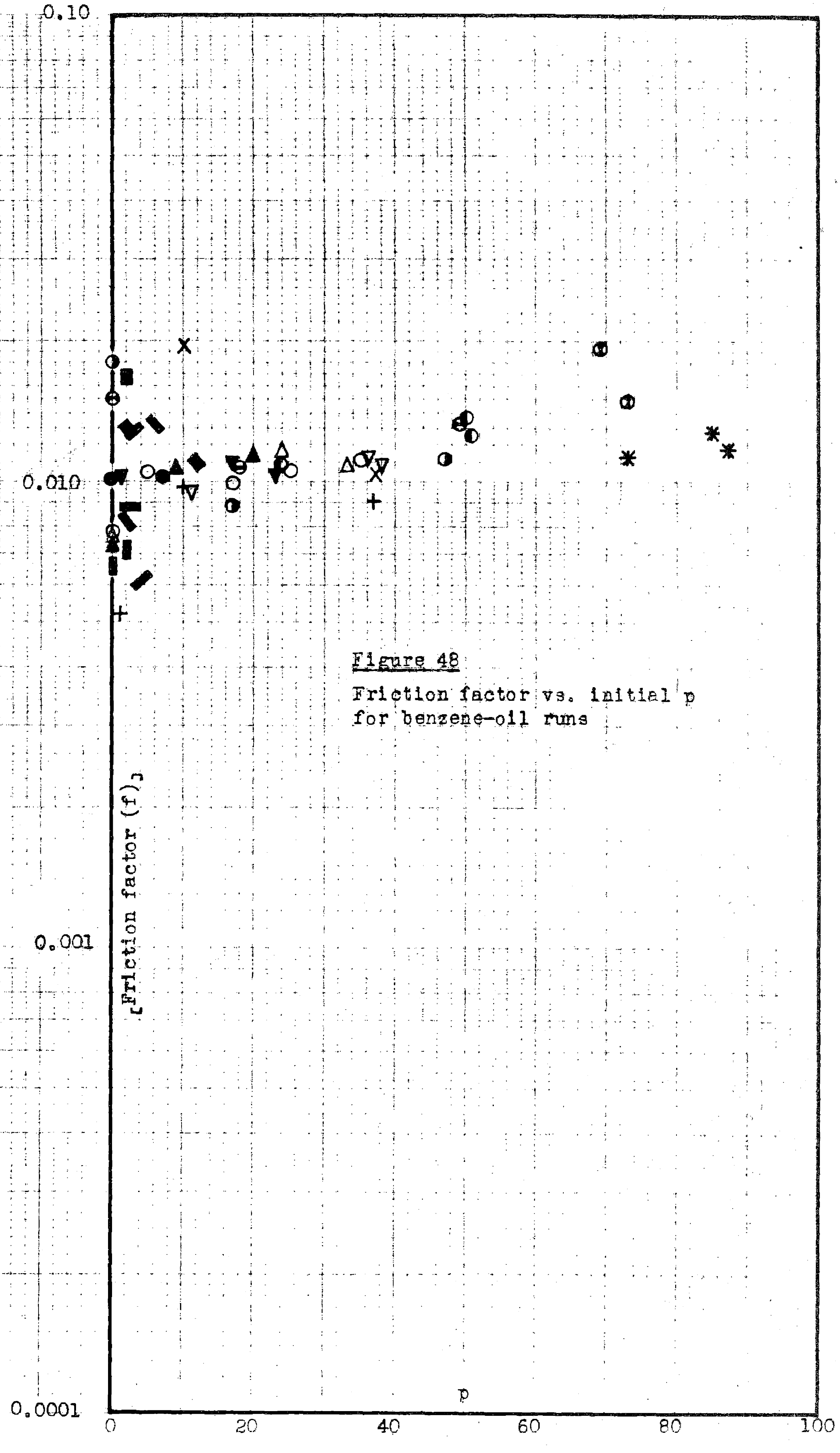


Figure 48

Friction factor vs. initial p
for benzene-oil runs

DISCUSSION OF RESULTS. Section IV. - Pressure Drop Studies

A glance at the results of the individual runs discloses that in most cases the heat transfer is relatively low in the early jackets and considerably higher in the subsequent jackets, except where vapor binding occurs as a result of excessive cumulative vaporization. Similarly, the pressure drop in the early jackets was quite small, but increased rapidly in the subsequent jackets. Purely as an empirical study a plot of the heat transfer coefficient versus the pressure drop was indicated. Since the pressure drops were determined across groups of three jackets, the heat transfer coefficients in each of these three jackets were averaged arithmetically and the average coefficient thus obtained was used as the ordinate in Figures 43, 44, and 45. To avoid the confusion resulting from vapor binding at excessive cumulative vaporization, data was deleted from Figures 43 and 44 if the cumulative vaporization at the end of the pass was equal or greater than 74%. In Figure 43 (water runs) it is noted that the data at 10 lbs./sq.in.ga. steam pressure fell on the same curve as data at higher steam pressures, independent of feed rate. In Figure 44 (benzene runs) one line was drawn through the data taken at very low steam pressure, another through data taken at very high steam pressure, and data at intermediate steam pressures were left scattered in between. While the correlation of Figure 44 is not all that could be desired, no general trend as a function of feed rate was discernible. In Figure 45 the points could be divided approximately into two groups of lean and rich feed composition. It is believed that this division is not particularly significant, but it was felt that the plot did not warrant further study, e.g., attempting to draw lines of constant composition of the residual liquid phase, etc.

In Figure 43 arrows are used to designate those points in which boiling had not commenced at the entrance to the jacket. Since, however, points were deleted if the pressure drop across the pass was equal or less than 0.5 cm.Hg., these points which are represented by arrows still represent a considerable amount of vaporization in the pass. These points, indicated by arrows, do not deviate far from the curve through the rest of the points, indicating that this method of correlating the data is quite satisfactory even when some of the heat transfer is used only for warming the feed.

Heat transfer coefficients for boiling inside of horizontal tubes are increased by a moderate increase in the cumulative per cent vaporization. That this increase may be rather expensive, however, is indicated by the flat slope of the lines in Figures 43 to 45. Thus, in Figure 43 it is noted that only a two-fold increase in the heat transfer coefficient (from 1000 to 2000) results from a thirty-three fold increase in the pressure gradient (from 0.9 to 30 cm.Hg./pass). In addition to the disadvantage of vapor binding at high temperature differences because such vapor binding requires increased heat transfer area, Figure 44 shows that vapor binding of the metal wall due to high steam pressures will result in increased pumping charges for a given total heat transfer.

A fundamental pressure drop analysis, however, must be made on the basis of a mechanical energy balance. The details of the derivation of the mechanical energy balance are given on pages 171 to 173 . In order to simplify the integration of the equation two assumptions were made: (1) it was assumed that the volume occupied by the residual liquid phase was negligible, and (2) it was assumed that the velocity of the liquid phase was euqal to the velocity of the vapor phase. The calculations were simplified by the assumption that the volume of the vapor evolved could be calculated by the perfect gas laws: this assumption, though unnecessary, was quite good. The first basic assumption is only as good as the second: if the two phases are moving with the same velocity after ten per cent of the feed has been vaporized, then the ratio of vapor to liquid at that particular point in the apparatus is 1:10, by weight, but is 186:1, by volume, for water at atmospheric pressure; on the other hand the liquid might be moving so slowly that although ten per cent of the feed has been vaporized the liquid phase may fill half of the volume of the tube. Another bad effect of the second assumption (of equal velocities of the two phases) shows up as follows: if the liquid is really merely creeping along the bottom of the tube its change in kinetic energy is negligible, yet the assumption implies that the change in kinetic energy of the liquid per unit weight is as great as the change in kinetic energy per unit weight of vapor. The second assumption may be quite good after large weight fractions of the feed have been vaporized and the liquid phase is either intimately mixed with the vapor phase or is carried as a spray suspended in the vapor phase.

In Figures 46 and 47, the Fanning friction factors calculated for the runs on water and on benzene are plotted against the cumulative per cent vapor as the fluid enters each pass. Data on the first heated pass in the apparatus have been deleted, since it was known that the assumptions could not be valid before vaporization had commenced. The abnormally low friction factors encountered when less than 20% of the feed had been vaporized are due to the assumption of an erroneously large change in kinetic energy, leaving but a small proportion of the total pressure drop to be accounted for by friction. Above 20% vaporization the assumptions seem to be valid, and friction factors in the reasonable vicinity of 0.006 are obtained.

If the assumption is made that the Reynolds number is a satisfactory criterion for determining friction factors for flow of mixed phases in pipes, the only problem involved in the calculation of the Reynolds number is the decision as to what viscosity to use. For a feed rate of 700 lbs./hr. of water flowing at atmospheric pressure at its boiling point inside of a 1.048-inch pipe, the Reynolds number is 14,700 if the viscosity of the liquid is used, and 3,200,000 if the viscosity of the vapor is used. Corresponding friction factors are 0.007 and 0.0024, respectively. It is concluded that the velocity of the liquid is of primary importance in determining the effective Reynolds number, but the data are not sufficiently precise to formulate general rules for calculating the effective Reynolds number at this time.

Figure 48 is a plot of the Fanning friction factors versus the entering value of the cumulative vaporization as obtained for the runs on benzene-oil mixtures. It would be advisable to draw lines of constant average composition of residual liquid; however, the liquid phase composition frequently varied over such extreme limits during its passage through one pass that a satisfactory method of averaging could not be decided on.

Figure 48 differs from Figures 46 and 47 in two respects: (1) the random variation of the friction factors at low values of entering cumulative vaporization, and (2) the higher average friction factors at high vaporizations. Since large friction factors were obtained after less than 5% of the feed had been vaporized, it is suggested that the liquid in these runs is truly being carried along at a velocity approximately the same as that of the vapor even at small vaporization. At high vaporization the friction factors average around 0.012. Friction factors in the region of 0.012 correspond to the critical values of Reynolds numbers where the flow may be either turbulent or streamline, -- a region referred to as the "dip" region. The erratic nature of the friction factors appearing in Figure 48 may in part be due to a range of transition from streamline to turbulent flow, although such an hypothesis is open to the very violent objections which must be made to a suggestion that any kind of an effective streamline flow can be maintained after any vaporization has occurred.

The mere fact that considerably larger friction factors were obtained during the runs on the benzene-oil mixtures indicates again the importance of the viscosity of the liquid phase even though considerable portions of the feed have been vaporized.

C O N C L U S I O N S

CONCLUSIONS

A. For Pure Liquids

1. A moderate increase in cumulative weight per cent vapor increases the heat transfer coefficient for liquids boiling inside of horizontal tubes at moderate temperature differences.
2. When high steam pressures are used the resulting large temperature differences will cause vapor binding to occur when liquids are boiling inside of horizontal tubes.
3. When liquids are boiling inside of horizontal tubes at a temperature difference equal or somewhat greater than that required to produce the maximum heat transfer coefficient, an increase in cumulative weight per cent vapor is not beneficial and may be detrimental.
4. At moderate temperature differences, the tube may become vapor-bound by large cumulative weight per cent vapor.
5. When the tube is vapor-bound by excessive cumulative per cent vapor, a return bend has a very beneficial effect, increasing the heat transfer in the jacket immediately following the bend.
6. No appreciable beneficial effect of a return bend occurs unless the tube is vapor bound by excessive cumulative per cent vapor.
7. Friction factors may be calculated conservatively by using the viscosity of the liquid phase in determining the Reynolds number.

B. For Benzene-Oil Mixtures

8. In an actual run the coefficient eventually decreases not because of vapor binding due to excessive cumulative per cent vapor, but because of the increasing per cent of oil in the residual liquid phase.

9. For a constant composition of residual liquid phase, increasing cumulative per cent vapor increases the heat transfer coefficient. This effect is sufficiently great to cause a temporary increase in the heat transfer coefficient obtained from a given feed, despite the increasing per cent of oil in the residual liquid phase and despite the decreasing temperature difference as the cumulative per cent vapor increases.

10. A beneficial effect of a return bend occurs even with small cumulative per cent vapor. A residual effect is carried over into the second jacket after the return bend.

11. For a given cumulative per cent vapor, the heat transfer coefficient decreases as the amount of benzene in the residual liquid phase decreases.

RECOMMENDATIONS

RECOMMENDATIONS

1. It is recommended that an intensive study of heat transfer to boiling liquids be initiated, using the apparatus from which the results of this thesis were obtained. Such a program of study would presumably require five or six years of work.

2. It is recommended that runs on boiling toluene be included in one phase of the above study.

3. It is recommended that the effects resulting from the insertion of twisted metal ribbons inside of the horizontal tubes be studied as another phase of the above program.

A P P E N D I X

D E S C R I P T I O N   O F   A P P A R A T U S

DESCRIPTION OF APPARATUS

Cooling Water System

The condenser water entered a manifold through a valve in a 1-1/4 inch standard steel pipe. Inlet temperature was measured by a fractional thermometer in a well prior to the valve. Rubber pressure tubing connected the manifold to the thirteen coils of the condenser. The coils were made of 3/8-inch copper tubing, 15 feet long, so that a cooling surface of 19 sq. ft. was exposed. Rubber tubing connected the discharge (bottom) end of the coils to another manifold. The discharge water passed through three feet of 1-1/4 inch standard steel pipe to an orifice. This orifice had a diameter of 63/64 inches during the benzene runs B1 to B14. During the rest of the runs an orifice plate of 11/16 inches diameter was used. Flange taps connected the orifice to a mercury filler manometer. The discharge temperature was measured by a fractional thermometer immersed in the stream below the orifice(no thermometer well).

During the benzene-oil runs an oil cooler was used. The water for the cooler came from a tee in the condenser water line located downstream from the inlet thermometer well and upstream from the dendenser water valve. A separate valve was located in the cooler water line just below this tee. A 1-inch standard steel pipe was used for the cooler water. The water passed through a 2-foot section of pipe to an orifice of 9/32 inch diameter. The orifice was connected with flange taps to a mercury-filled manometer.

The cooler was made of a 20 gal. steel drum. The cooler water entered the bottom of the drum and flowed upwards on the outside of the coils and out the top. The outlet temperature was measured by a fractional thermometer immersed directly in the stream, no thermometer well being used.

Steam System

The steam was obtained from a 185 lbs./sq. in. ga. steam line, but the pressure allowed in the steam jackets was limited to 125 lbs./sq.in.ga. both beaause the jackets were made of standard steel pipe and because there was danger of melting the soft solder used with the thermocouples. The steam from the high pressure line passed through about 40 feet of extra heavy 1-inch pipe to a reducing valve. This line contained eight elbows and a tee. A separate 1/2 inch line was connected to the high pressure steam line and was exhausted to the atmosphere to increase the rate of flow through the high pressure main and lower the quality of the steam. This line was also used occasionally to effect minor adjustments in steam pressure. Below the reducing valve the steam system was constructed entirely of standard steel pipe. The steam passed through a centrifugal separator constructed of a 3-inch tee. The steam enter the tee tangentially, The uptake pipe extended down near the bottom of the tee. At the top of the separator was located a thermometer well, but any readings were meaningless because of the pressure drop between this well and the steam gage. The steam left the thermometer well and passed through a 1-1/4 inch tee filled with glass wool, intended to serve as a filter. Since the tee was located horizontally it was subsequently noticed that all the glass wool packed down at the bottom and the filtering action was probably highly ineffective. Since any rusty water which accumulated in the line overnight was drained off in the separator, however, the absence of the filter may not have been too serious. The octyl thiocyante promoter of dropwise condensation of steam on copper was added at this tee. The steam then passed through a tee to which was fastened a steam pressure regulator, home made. The regulator consisted of an orifice partially plugged by a brass ball which was forced into the orifice by a spring. The tension on the spring was subject to adjustment. However, the operation of the pressure regulator was not very satisfactory and its use could probably have been dispensed with. The steam pressure gage was located in a tee connected to the steam header by a close nipple. A siphon was used to prevent live steam inside of the pressure gage. Two calibrated gages were used in various ranges, gage no.25 (0 - 30 lbs./sq.in.ga.) and gage number 205 (5 - 180 lbs./sq.in.). The header consisted of three 1-inch tees and one 1-inch elbow.

The steam jackets were numbered in the order in which the boiling fluid flowed through them. A 1/2 inch line left the elbow of the steam header and passed through two tees and a 1/2 inch elbow. The first tee led into Jacket No. 3, the second into Jacket No. 2, and the remainder of the steam which left the header by this line continued to Jacket No. 1. The three other lines from the header led successively to Jackets 4, 5, and 6, and Jackets 9, 8, and 7, and to Jackets 10, 11, and 12.

The steam jackets were made of standard 2-inch steel pipe, each section of pipe being 3 ft. 2-inches long. A packing gland was constructed for each end of the jacket. The jackets were arranged symmetrically on the 12-foot copper pipe, being separated from one another by 6 inches and extending to within 9 inches of the end of the pipe. The steam entered on the top side of the jacket 7 inches from the end nearest the steam header. The steam condensate left through a 1/4 inch standard steel pipe located on the bottom side of the jacket 6 inches from the end farthest removed from the steam header. This 1/4-inch pipe projected into the steam jacket, but the end was notched to prevent the accumulation of stagnant condensate. A small hole (0.002 -inch diameter) was drilled in the top of each jacket near the discharge end as a vent to prevent the accumulation of non-condensable gases in the jacket. After drilling, these holes were peened partially closed by tapping with a hammer. The connections to the jacket were first threaded into holes in the jacket and then brazed in place.

Before installing the steam jackets the outside of the copper pipe was scoured with "Old New Hampshire Household Cleaner". It was then washed with a hot Na_2CO_3 solution, dried, and polished with a Ritz "Silvena" polishing cloth.

The condensate drained out through twelve separate No. 70 Strong steam traps. The drain lines were located with a downward slope and the condensate was collected in quart milk bottles over a time interval measured by stop watch, and was measured in a liter graduate. The temperature of the condensate at the time of measuring was taken by a 400° F. thermometer because of the long stem. When the condensate lines were not lagged the condensate lines were continually checked to note whether appreciable amounts of live steam were blowing out the end. It proved more satisfactory to lag the condensate drain lines and check to insure that there was always at least some live steam blowing out the end, changing the calculations accordingly. The condensate lines were lagged after Run BO-8.

Fluid system.

The original pump consisted of an eight-vane bronze positive displacement pump of rated capacity equal to 3.5 cu.ft./min. against a 50 foot head. The rate of flow was controlled by a valve in a by-pass line around the pump. This pump was used on all of the benzene runs except Runs B6A and B7A. Prior to Run BO - 7 a Micro Westco, Betlendorf No. 1a, pump was installed. This pump had a rated capacity of 6.8 gal./min. against a 20 foot head and 1.0 gal./min. against a 150 ft. head. The original pump was so scored that it failed to approach its rated capacity and gave an insufficient feed rate after the oil cooler (with its accompanying frictional resistance) was installed. The new pump had the advantage of drawing in less air on the suction side of the pump, of quieter operation, and of satisfactory lubrication even when operating on pure benzene. The feed rate with the new pump was controlled by a by-pass valve and also a valve in the main feed line between the pump and the feed orifice.

Prior to installation of the settling drum before Run BO-13 the feed flowed through a vertical 1-inch standard steel pipe and entered the first pass of the heating section through an elbow. An orifice was installed in this line 2-1/2 feet from the top end. The orifice plate was 0.206 inches in diameter and the orifice was connected with flange taps to a mercury filled manometer.

The settler drum was constructed of a 5-gal. ether can and was installed in the line between the pump and the orifice. The drum was placed with its axis horizontal, the feed entering near the top of one end of the drum and the air-free mixture leaving near the bottom of the other end. Bulkheads were placed at each end of the drum and connected with tie-rods to serve as reinforcement. A sight-glass was installed on one end of the drum but bulging of the ends threw the glass out of alignment and the glass leaked so badly that it was not used. A small cock on the top of the drum at the end farthest removed from the end where the feed entered served as an air vent. Before starting a run the vent was opened and all the air was vented out. This valve was then left slightly opened during the rest of the run. A certain amount of air was not completely separated and left through the vent along with considerable liquid, so it was necessary to have the vented air discharge its liquid content into a collecting bottle.

Before the settler drum was installed a thermometer well was installed in the line between the pump and the orifice. After the settler drum was installed this well was installed in the 1-inch line leaving the settler. The temperature indicated by a thermometer in this well was taken as the feed temperature. A sampling cock was inserted in the line between the pump and the settler drum. While the settler drum was being installed the feed orifice plate was enlarged to 0.213 inches diameter. An unobstructed length of two feet of pipe was left between the thermometer and the orifice. The heating section consisted of four 12-ft. lenghts of 1-inch standard pipe size copper pipe, purchased from the Chase Brass and Copper Co. The pipes were steam jacketed as described above. The pipes were connected in series by 1-inch Pyrex glass U-bends, purchased from the Corning Glass Co. The gasket material between the U-bends and the pipes was originally made of paraffin-impregnated asbestos, but the use of Vellumoid gasket material was later found more advisable. Vellumoid is manufactured by the Vellumoid Co., Worcester, Mass., and was used on all runs after BO-2. Great care was taken that the gasket in no way obstructed the flow of fluid. The U-bends were bolted to the copper pipes through rail flanges.

The entire heater was supported on brackets suspended from the ceiling. Three brackets were used, passing under the center of each group of four steam jackets. The heater was suspended about eight feet from the ground and was carefully levelled, although the slanting condensate drain lines created an optical illusion that indicated otherwise. Nevertheless, on draining the apparatus it was found that the glass U-bend at the end of the second pass drained clean whereas almost half an inch of liquid was left in the other two U-bends. This indicated a pitch of about 1/2-inch out of 13 feet, or a little over 2^{o}. The steam jackets were carefully spaced and then tied to the brackets by metal strips (pipe clips), thus relieving strain on the glass U-bends.

A small hole was drilled and tapped 4 inches from the entering end of each copper pipe, into which was screwed a short 1/8-inch nipple. Through this nipple a thermometer was inserted into the stream. The nipple was drilled to a sufficiently large diameter that the thermometer could be inserted. The nipple usually projected about an eighth of an inch beyond and inside of the inside pipe wall. A bushing and cap (with a hole drilled in the cap) tightly held a cork around the thermometer to prevent leakage of fluid. The thermometers were wired in place.

At the other end of the pipe, the discharge end, a small hole of 1/8-inch diameter was drilled completely through the copper pipe on the bottom side 4 inches from the end of the pipe. This hole was then countersunk with a 3/16-inch drill and the manometer lead soldered into the countersunk hole. A similar pressure tap was located two feet downstream from the feed orifice and 6 inches before the elbow which brought the feed into the first pass.

The vapor-liquid separator consisted of a cast steel cylinder with a vertical baffle plate and two observation windows. The fourth copper pipe was connected by union to a 1-inch standard steel nipple which in turn was screwed into a 1/8-inch steel plate and bolted onto the entrance to the separator. The liquid-vapor mixture impinged against the baffle plate: the liquid fell to the conical bottom of the separator, while the vapor whirled beneath the baffle plate and left on the other side of the plate near the top of the separator. When running on mixtures of benzene and oil no discoloration could be detected in the benzene condensate, indicating that the separation was quite complete.

The condenser was made from a 15 gal. steel drum, 13 inches in diameter and 22 inches in length. The condenser was built and originally used by Abbot and Comley (1). The drum was mounted with its axis vertical and a length of stove-pipe, with an elbow, carried the vapors from the separator to the top side of the condenser. All seams in the stove pipe were soldered to prevent leaks. The vapor condensed on the outside of the copper coils through which the condenser water flowed. A vent was located on the top of the condenser to permit the escape of air when starting up the apparatus. During the benzene runs B1 to B14 it was noted that the "breathing action" of the apparatus caused the exhaling of a certain amount of benzene vapor, followed by an inhaling of air. To remedy this situation 10 feet of 3/8 inch copper tubing in the form of a coil was fastened to the top of the vent. This coil served a two-fold purpose: (1) the partial condensation of benzene vapor blown into the coil and (2) the inhaling of benzene vapor rather than air. During the last few water runs leaks in the condenser restored the same condition existing with the vent without the copper coil.

A sight glass was attached to the condenser, indicating the liquid level inside the condenser. This prevented blanketing the cooling surface with accumulated condensate, and also facilitated the preservation of a constant head of liquid on the suction side of the pump. The line leading from the condenser to the pump contained a valve and a tee. The liquid from the separator entered by the tee and was well mixed with the condensate as the mixture passed through the pump. A thermometer well was also located in the benzene line.

The bottom of the separator led into a 1-1/4 inch nipple, ten inches long, to which was attached a sight glass. This sight glass was not used during the runs on pure liquids. During the benzene-oil runs it was desireable to maintain a constant head in the separator (as well as in the condenser) to assure steadiness of operation. Surges in the amount of liquid removed in the separator were so great that the level in the separator sight glass rythmically vanished out of sight at both ends of the glass. So the 1-1/4 inch nipple was replaced after run BO-2 with a 4-inch standard steel nipple (8 inches long), to which the sight glass was re-affixed.

At the bottom end of the vertical nipple was a tee. A thermometer well entered one end of the tee and the liquid flowed out the other. On the bottom of the tee a hole was drilled and tapped for the insertion of a sampling cock. Standard 3/4 inch pipe was used for the rest of the piping until the liquid again mixed with the condensate.

Close to the above-mentioned tee was a cross. One arm of the cross was used for the admission of liquid to the system. Another arm led through a valve directly to the condensate line and thence to the orifice. The third arm led through a valve to the cooler.

The cooler, like the settler, was constructed of a 5-gal. steel ether can, 11 inches in diameter and 11 inches long, mounted with the axis horizontal. The oil from the separator flowed into a manifold to which was affixed six copper nipples, made of 1/4-inch pipe. These nipples were soldered to six copper coils inside the drum, each coil being 16 feet in length and constructed of 1/2-inch., No. 20 BWG tubing. The coils discharged into a second header. The oil flowed down-hill through the coils, countercurrent to the cooler water. From the discharge header a line led back to the condensate line and thence to the pump. A thermometer well was installed in this line.

A drain valve was located at the bottom of the system on the suction side of the pump.

Manometer system

All manometers were made of 5 mm. glass tubing and filled with water and mercury. The leads which carried benzene (all leads from the heating section and the two leads from the feed orifice) were made from 3/16-inch copper tubing. These leads were soldered into the top of a 3-inch cubical box, referred to as a "seal cup". A short length of copper was soldered to the bottom of this box and fastened to the manometer by rubber pressure tubing. A cock was soldered into the top of the seal cup. Through this cock water was added until the cup was half full. Benzene was then allowed to flow into the cup through the manometer lead, air into the cup escaping through the cock. There was thus provided a large benzene-water interface at substantially constant elevation, and the glass-to-metal connections could be made with rubber tubing, unexposed to benzene. The cups were made of galvanized iron.

The seal cups were removed during the water runs.

Thermocouple system.

The thermocouples were installed two years before the thesis was completed and accordingly are not in agreement with what is today recognized as recommended installation. Two single-wire thermocouples were installed on the top and bottom of the copper pipe in the center of each jacket. (The wire from the bottom of the twelfth jacket broke inside the jacket shortly after installation, and no data were taken on this thermocouple.) Holes (0.035-inch diameter) were drilled tangentially into the walls of the pipe, by the method used by Akin (2). Two holes (for each jacket) were drilled on opposite sides of the copper pipe (top and bottom), separated by three inches to facilitate the subsequent soldering of each thermocouple,individually.These holes were drilled to such a depth that the bottoms of them were on line with the center of the pipe and were midway in the pipe wall. A constantan wire (Advanced Ideal, No. 30 BWG) was sheathed in a two-inch length of drawn capillary glass tubing, fused at the ends after insertion on the wire. One-eighth of an inch of bare wire projected beyond one end of the glass tubing. This end was tinned with soft solder and immersed inthe above-mentioned hole, which had been filled with molten solder. The wire was then brought out through 1/8-inch close nipples in the wall of the jacket. The close nipples were lined with rubber tubing and filled with porcelain cement.and capped, a hole being drilled inthe cap to allow the passage of the constantan wire. This wire (cloth covered) lead to a switch box from which it could be electrically connected with the constantan of the cold junction. Although the actual switches were of copper, the current flowed through only a two-inch length of copper strip along which temperature variations should be minor. A common copper wire was soldered to each of the four copper pipes and led to the potentiometer. The other (copper) lead from the potentiometer went to the copper side of the cold junction. The cold junction was immersed in an ice bath, and the temperature of $0^{\circ}C.$ was regularly checked by a thermometer. Precaution had to be taken that only one of the twenty-three knife switches was closed at a time, else the readings were some sort of an average value.

The potentiometer used in the first six benzene runs was a Leeds and Northrup Precision Model No. 98141, M.I.T. No. 361. During the water runs the potentiometer was a Leeds and Northrup Industrial Model No. 362.

The chronological order of the runs is:

B1 to B14

B1A to B5A

BO-1 to BO-24

B6A to B7A

W1 to W15

E X P A N S I O N O F D I S C U S S I O N O F R E S U L T S

Precision Study

Analysis of Pressure Drops

Estimate of Pressure Drop through Steam Lines

Treatment of Thermocouple Data

PRECISION STUDY

The rate of condensate flow, as measured, may be in error by as much as 5 gms./min. for normal rates of flow. A similar error should be affixed to the heat losses, leaving a maximum error in the rate of flow of condensate of ± 10 gms./(min.)(jacket), which is equivalent to an error in q/A of about 1300 B.t.u./(hr.)(sq.ft.). Imposed on this is an additional possible error due to the intermittent operation of the traps. In general, the flow of condensate from the discharge lines was quite steady, the long pipes tending to iron out fluctuations in flow. At high rates the fluctuations were very rapid, if they occurred at all, and the effect of steam trap discharge could be serious only at low rates of flow. On the other hand, at high rates of flow several measurements of volume were needed. It is accordingly recommended that a possible error of about ± 2000 B.t.u./(hr.)(sq.ft.) be considered for each jacket. That this error is not consistently high or low is attested to by the good overall heat balances obtained. Prior to lagging the steam condensate drain lines the possibility existed of live steam leaking through the steam trap, condensing because of heat loss from the drain lines, and resulting in high condensate readings; after the condensate drain lines were insulated and the heat loss through this insulation considered in the calculations it is felt that this possibility of error was eliminated.

The fluctuations in the operation of the apparatus (due to surging of the feed through the heated pipes) affected all the rest of the readings except for the condenser and cooler water rates. Manometers and thermometer readings fluctuated, and the reading reported was the result of a visual average over a relatively short space of time. The accuracy of such averages is questionable; e.g., averages of Pitot tube readings in a fluctuating stream do not give an accurate measure of the average stream velocity.

In measuring the heat picked up by the condenser an attempt was made to keep the condenser water rise, in °C., almost numerically equal to the condenser water orifice reading, in cm.Hg. This technique could not be used with the cooler; with the cooler it was necessary to adjust the water rate so that the oil leaving the cooler was cool enough to be mixed with the benzene condensate but was not so cold that it would "freeze" in the cooler or cause extreme pressure drops in the cooler. In both pieces of equipment the water rise was measured with fractional thermometers which read to the nearest tenth of a degree Centigrade, could be estimated to the nearest twentieth, and were probably accurate to at least 0.2 °C. (as indicated by an earlier calibration of different fractional thermometers which were not used at the time when these data were collected). Thus, the difference in readings was accurate to at least 0.4°C. A further inaccuracy was introduced by the fluctuating readings, causing a further error of ± 0.5 °C. With an average rise of about 20°C., an error of about 5% may therefore be expected in determining the rise in temperature of the water. The condenser and cooler water rates were dependent only on the inlet water pressure and independent of the behavior of the apparatus. Calibration indicated reporducibility to within 5%, although the small readings obtained from the large condenser water orifice during benzene Runs B1 to B14 increased the possible error in measuring the condenser water rate for these runs. Hence, the heat picked up by the condenser and cooler water is accurate to at least 10%.

Comparing the overall heat balances, as corrected for estimated heat losses, all of the water runs checked within 8%. The heat transfer as measured by steam condensate averaged 3% lower than the heat transfer as determined by the condenser readings. For the benzene-oil runs, the heat balances checked to within 7% at all times, with an arithmetic average error of very close to zero, although the average deviation was ± 2%. For the benzene runs the heat balance was less satisfactory. In Run B13 an extreme deviation was obtained when the heat transfer as indicated by steam condensate was 19% higher than the heat transfer indicated by the condenser water. For all the benzene runs the heat transfer by steam condensate averaged 3-1/2% higher than the heat transfer by condenser water.

In the benzene-oil runs the heat picked up by the cooler varied from 5% (Run BO-6) to 78% (Run BO-8) of the total heat transfer indicated by the cooler and condenser waters.

Using the low range steam pressure gage the steam pressure could be estimated to the nearest tenth of a lb./sq.in. However, at any one time the steam pressure fluctuated up and down over a range of from 0.5 to greater than 1 lb./sq.in. In addition, during the course of a run it was possible for the average steam pressure to rise or fall within the same limits. The limit of accuracy of all reported steam pressures of less than 30 lbs./sq.in. is probably ± 1.0 lb./sq.in. At 10 lbs./sq.in.ga. pressure an error of 1 lb./sq.in. changes the apparent condensing steam temperature by about 1.2°C., and this change in temperature per unit change in pressure decreases at higher pressures, amounting to less than 0.5°C. at a gage pressure of 70 lbs./sq.in. In addition, an error of as much as 0.5°C. may be introduced in estimating the condensing steam temperature in the water runs (cf. pp. 174 to 177), as a result of pressure drop in the steam line between the gage and the jacket. Neglect of this pressure drop yields conservative coefficients.

In water Runs W12, W13, and W15 the condensing steam temperature was estimated by thermocouple readings. This estimate may well be in error by as much as 5°C., causing an error of about 10% in the observed temperature difference, and an error of about 10 lbs./sq.in. in the indicated steam pressure.

A peculiar phenomenon was noted as a result of observation of the pressure drop in the first heated pass. At low feed rates a small positive reading would be interrupted by a sudden and momentary negative reading, say of about 5 cm.Hg. This can best be explained by the assumption that a slug of liquid moving down the pipe becomes superheated, suddenly explodes into vapor, and builds up a temporary high pressure at the downstream end of the pipe. This explanation is verified by the observation of the first U-bend during those runs where the first pass was not heated; the forward flow of the liquid would be occasionally interrupted by a brief jerk backwards.

With no liquid flowing, it was noted that if one of the manometers was set into swinging motion by squeezing on a piece of the rubber tubing this motion was transmitted through the seal cups to the other manometers. It is not known whether this effect tended to magnify or to dampen the surging of the manometer during an actual run.

The fluctuations in manometer readings usually were comparatively small for the early passes but increased in intensity for the later passes and the static pressure at the end of the fourth pass. Exceptions to this generalized statement were erratic and unexplainable. Run 11 was remarkably steady, the pressure drop across the last heated pass, and the pressure leaving the fourth pass fluctuating over a range of only 2 cm.Hg., and possessing a rythm so regular that these pressure drop readings could be taken as accurate to within 1/2 cm.Hg. although the observed average readings were 28 and 13 cm. Hg., respectively.

In general, the pressure drop across the first heated pass was characterized by a fluctuation in one arm of the manometer over a range of about 2 cm. Hg., or a total fluctuation in reading of 4 cm. Hg., although a sudden water hammer or extra violent surge might cause occasional variation in reading of twice this amount. The normal swing of the reading for the last heated pass and in the static pressure manometer was about 8 or 10 cm. Hg. The reported pressure drop is probably no more accurate than 10% of this normal swing, resulting in an error in reading of $\pm$ 0.5 cm. at early pressure drops and $\pm$ 1 cm. Hg. for later pressure drops. Accordingly, the absolute pressure at the entrance to the apparatus could be off by as much as 4 cm. Hg., resulting in possible error in the estimated initial boiling point of about 1°C.

Fluid temperatures were measured by thermometer readings. Only the inlet thermometer reading was used in the calculation of results, the rest of the readings being used merely as a check on the temperature taken as the saturation temperature at the pressure in question, except in the benzene-oil runs. This check was usually within 1°C. The temperatures fluctuated up and down over a range of as much as 3°C. (or more).

The following paragraph refers only to the benzene runs and the water runs. When the feed was completely vaporized the feed rate was substantially as accurate as the heat balance. In other cases, the feed rate was estimated solely from the feed orifice reading. Errors in the feed rate of course make corresponding errors in the calculation of the cumulative per cent of feed vaporized. Since the heat balances were good, the per cent of feed vaporized times the feed rate reported is a good measure of the amount of vapor passing through the tube at any point. Hence, the calculated vapor velocity probably has somewhat more precision than the calculated per cent vaporized. The feed orifice usually fluctuated over about 10% of the reading, with occasional surges amounting to 50% of the reading. A time average of the square root of the reading would probably be most accurate if it were not hampered by the momentum of the column of mercury as it swung back and forth in the manometer. Arithmetic averages of the limits of the average swing were used in calculating the reported head of Hg. across the orifice. Because of the occasional surges, the reading is probably no more accurate than the average fluctuation, say 10%. Accordingly, the reported feed rate is subject to a probable error of about 5%.

For the benzene-oil runs full reliance was placed upon the reported fluid temperatures. The vapor pressure data at hand was not considered sufficiently reliable to justify drawing any conclusions as to whether or not the fluid in the pipes was at equilibrium temperature with the benzene vapor. The feed rates reported were only as reliable as (1) the heat balance, and (2) the analyses of the feed and product. The analysis of a given sample could be checked to within 0.5%; whether or not the sample was representative is a different question. The product sample was taken from the separator at a slightly lower pressure than the pressure at the end of the twelfth jacket, so a small amount of benzene may have flashed, causing the reported amount of benzene in the product to be a little low. The sample from the separator, however, was drawn from a fairly large pool, whose composition should have remained fairly constant. The sample of the feed was drawn from the line leaving the pump; since a constant head of both benzene and oil was maintained on the pump the composition of the feed should not have fluctuated seriously. Half of the samples was withdrawn at the start of the run and the other half being withdrawn at the end, the two halves being mixed before analyzing. Tests on known samples indicated that about 0.5 cc. of benzene was held up in the condenser and flask so this amount of benzene was arbitrarily added to the reported value of benzene distilled out of the 100 cc. sample.

Two further factors should be mentioned in connection with the per cent vaporization. Per cent vaporization specifies the weight fraction of the feed which passes a given point as vapor. At this same point the pipe might well be quite full of slowly moving liquid. Hence, although the per cent vaporized at a given point is 50%, this does not mean that the instantaneous vapor: liquid weight ratio at that point is one to one.

Figures 1 to 14 (and others) indicate that at a certain point boiling commences. Actually a surge of liquid may proceed a jacket or so beyond this point before starting to boil, while in the ebb boiling may take place a jacket or so ahead of this indicated point. The percent vaporized, as reported, is a time average. The actual variations in instantaneous rate of flow and percent vaporized cover extremely wide limits.

Finally it should be noted that the relatively high coefficients obtained in the runs on water make these runs more sensitive to varying steam side conditions than is true of the runs on benzene and benzene-oil mixtures.

Analysis of Pressure Drops

Basis: 1 pound of vapor-liquid mixture passing any section.

p = weight fraction of feed as vapor.

v = specific volume of mixture

$$v = p(v_v) + (1 - p)(v_f) \tag{1}$$

Where v_v = specific volume of vapor

v_f = specific volume of liquid

$$\text{Also } dv = pd(v_v) + (v_v)\,dp - (v_f)\,dp \tag{1A}$$

$V^2/2g$ = kinetic energy of mixture

$$V^2/2g = p\,(V_v)^2/2g + (1-p)(V_f)^2/2g \tag{2}$$

Where V_v = velocity of vapor

V_f = velocity of liquid

$$\text{Also } d(V)^2 = pd(V_v)^2 + (V_v)^2dp + (1-p)d(V_f)^2 - (V_f)^2\,dp \tag{2A}$$

Mechanical energy balance:

$$-vdP = d(V)^2/2g + d(F) \tag{3}$$

Where P = static fluid pressure

$$d(F) = \text{friction loss} = (4f)(V^2/2g)d(N)/D \tag{3A}$$

f = friction factor

D = pipe diameter

N = pipe length

Substituting of Equation (1), (2A), and (3A) in Equation 3:

$$-p(v_v)dP - (1-p)(v_f)dP = [pd(V_v)^2+(V_v)^2dp + (1-p)d(V_f)^2-(V_f)^2dp]/2g$$
$$+ (4f)[p(V_v)^2 + (1-p)(V_f)^2]\,d(N)/2gD \tag{4}$$

$$\text{Let } M = (V_f/V_v)^2;\ (V_f)^2 = M(V_v)^2;\ \ d(V_f)^2 = Md(V_v)^2 + (V_v)^2dM \tag{5}$$

Substitution Equation (5) in Equation (4), eliminating (V_f):

$$-p(v_v)dP - (1-p)(v_f)dP = (V_v dV_v/g)(p+M-pM) + (V_v^2/2g)\,[(1-M)dp + (1-p)dM +$$
$$(4f)(p+M-pM)dN/D] \tag{6}$$

Assume perfect gas laws for vapor:

$$v_v = RT/P \tag{7}$$

Where R = gas law constant

T = absolute temperature

$$V_v = pWv_v/S = pWRT/PS \tag{8}$$

Where W = rate of flow of mixture

S = cross section

$$d(V_v) = (WRT_{av}/P_{av}S)dp \tag{8A}$$

Substituting Equation (7),(8) and (8A) into Equation (6), and simplifying:

$$-\left[RT/P + \frac{(1-p)}{\rho} v_f \right] dP = \left[\frac{W\ R\ T_{av}}{P_{av}S\sqrt{2g}}\right]^2 \left[2(p+M-pM)dp + \right.$$

$$\left. p\left[(1-M)dp+(1-p)dM+(4f)(p+M-pM)dN/D \right]\right] \tag{9}$$

If it can be assumed that M = 1, and that the volume occupied by the liquid is negligible:

$$(R\ T\ d\ P)/P = \left[(W\ R\ T_{av})/P_{av}\ S\ \sqrt{2g}\right]^2 \left[2dp + 4fpdN/D \right] \tag{10}$$

$$(P_{av}\ S^2\ g)/(W^2\ R\ T_{av})\ \Delta P = \Delta p + 2f\ p_{av}\ N/D \tag{10A}$$

$$f = D\ /\ (2p_{av}\ N)\ \left[(P_{av}\ S^2\ g)/(W^2\ R\ T_{av})\Delta P - \Delta p \right] \tag{10B}$$

R = 1544/18 = 85.8 for water

= 1544/78 = 19.8 for benzene

$S = \pi D^2/4$

$g = 32.2 \times (3600)^2$

P (lbs./sq.ft.) = (14.7 x 144/76) P (cm.Hg.abs.)

N = 13 ft. per pass, including U - bend. (Neglect first pass because of assumption).

$T\ (^{\circ}R) = (1.8)(T, ^{\circ}C.) + 492$

$D = 1.062/12$

Whence: $$f = \frac{\alpha\,(P_{av})\quad(\Delta P)}{W^2\ T_{av}\ p_{av}} - \frac{\Delta p}{293\ p_{av}} = A - B \qquad (11)$$

Where P = cm.Hg.abs.

p = % benzene (not fraction)

W = lbs./hr.

T = $^{\circ}$R.

α = 48,800 for water

α = 211,000 for benzene

Estimate of Pressure Drop through Steam Lines

The pressure gage is located on a one inch pipe. Immediately downstream from the gage is the header, which distributes the steam to the various passes. All the steam for Jackets 1,2, and 3 enter a 1/2 inch line (I.D.= 0.622 inches). After 10 inches of pipe the steam for Jacket 3 leaves by a tee. The steam for Jackets 1 and 2 continues on for 3 feet 8 inches, at which point the steam for Jacket 2 enters its jacket. The remaining steam flows through an additional 3 feet 8 inches of 1/2 inch pipe into Jacket 1. The above discussion is applicable to the other passes by referring to Jackets 6,5, and 4, or 7,8, and 9, or 12,11, and 10, instead of Jackets 1,2, and 3.

The following calculations are made in order to obtain the order of magnitude of this pressure drop, indicating the magnitude of the error introduced by assuming that the pressure indicated by the steam gage prevails in the Jacket. Neglect of this pressure drop makes the reported coefficients conservative. The error is far more serious in the water runs, with their higher heat fluxes.

Assuming that the pressure drop and velocity changes are small in magnitude, one may write:

$$144 \Delta P = \frac{2 f V^2 N}{g \; Dv} \qquad \text{(P.C.E.}^{(14)}\text{, p.90)}$$

Where ΔP = pressure drop, lbs./sq. in.

V = velocity, ft./sec.

N = pipe length, ft.

g = 32.2 ft./(sec.)^2

D = pipe diameter = 0.622/12

Estimate of Pressure Drop through Steam Lines (cont'd):

v = specific volume, lbs./cu.ft.

f = dimensionless function of Re (P.C.E., p.78)

Re = $4W/\pi Dz$

W = lbs. of steam/hr.

z = viscosity of steam, lbs./(ft.)(hr.)

z is taken as $2.42 \times .012 = .029$ at an average steam temperature of 130°C. Variations in z and Re have only minor effect on f (P.C.E., Figure 225, page 687)

$$\Delta P = \frac{2 f u^2 N \times 12}{144 \times 32.2 \times 0.622 \times v} = f u^2 N/v \times 120$$

$$Re = (4W \times 12)/(\pi \times 0.622 \times .029) = 850\ W$$

Sample Calculation No.1

Based on Run W-15, 3rd pass (Jackets 7,8, and 9). This selection was made because of the abnormally high steam comsumption in this pass.

Indicated steam pressure = 70 lbs./sq.in.ga. = 85 lbs./sq.in.abs.

Specific volume (v) = 5.17 cu.ft./lb. (Steam Tables) (8)

Gross condensate collected from J.7 = 870 gms./min.)
" " " " J.8 = 812 gms./min.) Cf. p. 196
" " " " J.9 = 800 gms./min.)

W from J. 8 to J. 7 = 800 x 60/454 = 106 lbs./hr.

W from J. 9 to J. 8 = (800 + 812) 60 / 454 = 214 lbs./hr.

W from header to J. 9 = (870 + 812 + 800) 60/454 = 328 lbs./hr.

From J. 8 to J. 7: Re = 850 x 106 = 90,000; f = 0.0055

N = 3' 8" = 3.67 ft.

$$V = (106/3600)(5.17)/(\pi (0.622/24)^2) = 72 \text{ ft./sec.}$$

Sample Calculation No. 1 (cont'd)

$\Delta P = 0.0055 \times (72)^2 \times (3.67) / (5.17)(120) = 0.17$ lbs./sq. in.

From J. 9 to J. 8: $Re = 850 \times 214 = 182{,}000$; $f = 0.005$

$N = 3.67$ ft.

$V = 72\ (214/106) = 145$ ft./sec.

$\Delta P = 0.005 \times (145)^2 \times 3.67/5.17 \times 120 = 0.62$ lbs./sq.in.

From Header to J. 9: $Re = 850 \times 328 = 280{,}000$; $f = 0.0045$

$N = 10'' = 0.83$ ft.

$V = 72\ (328/106) = 222$ ft./sec.

$\Delta P = (0.0045)\ (222)^2\ (0.83)/(5.17)(120) = 0.30$ lbs./sq.in.

Total ΔP = 0.17 + 0.62 + 0.30 = 1.09 lbs./sq.in.

This is equivalent to an error of about 0.50°C. in estimating the temperature of the steam condensing in J. 7, and represents an extreme condition.

Sample Calculation No. 2.

Based on Run W-1, 4th pass (Jackets 12,11, and 10). This selection was made because of the abnormally high steam consumption in this pass at such a low steam pressure.

Indicated steam pressure = 9.7 lbs./sq. in. ga. = 24.4 lbs./sq.in.abs.

Spec. volume (v) = 16.7 cu.ft./lb.

Gross condensate from J. 12 = 249 cc./min.) cf. p. 183
" " " J. 11 = 250 cc./min.)
" " " J. 10 = 241 cc./min.)

W from J. 11 to J. 12 = 33 lbs./hr.

W from J. 10 to J. 11 = 66 lbs./hr.

W from Header to J. 10 = 98 lbs./hr.

Sample Calculation No. 2 (cont'd)

From J. 11 to J. 12: Re = 850 x 33 = 28,000; $\mathfrak{f}$ = 0.007

$N = 3.67$ ft.

$V = (33/3600)(16.7)/(\pi\,(.622/24)^2) = 73$ ft./sec.

$\Delta P = (0.007)(73)^2(3.67)/(16.7)(120) = 0.07$ lbs./sq.in.

From J. 10 to J. 11: Re = 850 x 66 = 56,000; $\mathfrak{f}$ = 0.006

$N = 3.67$ ft.

$V = 146$ ft./sec.

$\Delta P = (0.006)(146)^2(3.67)/(16.7)(120) = 0.23$ lbs./sq.in.

From Header to J. 10: Re = 850 x 98 = 83,000; $\mathfrak{f}$ = 0.0055

$N = 0.83$ ft.

$V = 217$ ft./sec.

$\Delta P = (0.0055)(217)^2\,(0.83)/(16.7)(120) = 0.11$ lbs./sq.in.

Total ΔP = 0.07 + 0.23 + 0.11 = 0.41 lbs./sq. in.

This is also equivalent to an error of about 0.5°C. in J. 12 at this low pressure. The runs selected represent extreme conditions, - for all benzene and benzene-oil runs the effect will be far smaller.

TREATMENT OF THERMOCOUPLE DATA

Bringardner (4) carefully calibrated copper-constantan thermocouples in an oil bath over a wide range of temperatures. The readings were confirmed by recording data both when the oil bath was being heated and when it was being cooled. He found a definite but constant deviation from the standard electromotive forces reported in the I.C.T.(7) tables, which was ascribed to the Advanced Ideal wire having slightly different properties than had the constantan wire used in collecting data for the I.C.T. tables. Since the constantan wire used in constructing the thermocouples of the semi-commercial apparatus came from the same spool that Bringardner's constantan wire did, Bringardner's calibration curve was used to interpret thermocouple readings for the first six benzene runs. This calibration curve was verified approximately by taking thermocouple readings at a definite steam pressure when the inside of the copper pipes was dry.

Just prior to the runs on water the thermocouples were again calibrated by taking readings at a known steam pressure when the pipes were dry. The results of Table IV were obtained. In Table IV those readings which are low compared to the rest of the readings are underlined. The readings which are not underlined are in fairly satisfactory agreement with the Bringardner calibration. In this range a one millivolt change in thermocouple reading corresponds to about a 22°C. change in temperature; hence, the underlined thermocouple readings indicated a condensing steam temperature about 8° or 9°C. lower than did the rest of the thermocouple readings. Hence, a separate calibration curve was made for these faulty thermocouples. The top thermocouple in Jacket 9 was inconsistent.

During the runs on water, both of the thermocouples in Jackets 1, 3, 4, 5, 10, 11, and 12 were read from the Bringardner calibration curve, in accordance with Table IV. Both of the thermocouples in Jackets 7 and 8 were read from the special calibration curve, as indicated by Table IV. Although the calibration of Table IV indicated that the bottom thermocouple in Jacket 6 was normal, this thermocouple consistently read lower than the other thermocouple in the jacket and the two thermocouples in Jacket 5, indicating that its calibration had changed. So the top thermocouple of Jacket 6 was read from the Bringardner calibration, while the bottom thermocouple was read from the special calibration curve. On the other hand, the bottom thermocouple in Jacket 2 (and also the top thermocouple in Jacket 9) behaved perfectly normal, reading substantially the same as the top thermocouple in Jacket 2. Disregarding Table IV, both thermocouples in Jacket 2 and the top thermocouple in Jacket 9 were read from the Bringardner calibration curve.

This erratic change in the calibration of the thermocouples made the thermocouple readings highly unsatisfactory, and no weight was given to them except when it was necessary to estimate the steam pressure in the last three water runs. It should be noted that the thermocouple readings were quite satisfactory in the unheated jackets, complications arising only when steam was condensing in the jackets. It should also be noted that had the Bringardner calibration curve been used for the thermocouples in Jackets 7 and 8, the indicated wall temperature would have been in many cases well below the fluid temperature.

TABLE IV

Thermocouple Calibration Data (September 11, 1939)

(NOTE: the reported steam pressures are in lbs./sq.in.ga. and have been corrected in accordance with the steam gage calibration. The cold junction was maintained at $0^{\circ}C.$ at all times.)

Thermocouple	P_{st}	milli-volts	P_{st}	milli-volts	P_{st}	milli-volts
1 top	7.8	4.48	24.8	5.28	73	6.60
1 bottom	7.9	4.52	24.7	5.30	73	6.65
2 top	8.0	4.48	25.4	5.30	73	6.60
2 bottom	8.3	4.30	25.5	5.05	73	6.28
3 top	8.8	4.58	25.5	5.32	72	6.60
3 bottom	9.1	4.58	25.5	5.30	85	6.85
4 top	9.2	4.58	25.6	5.30	84	6.82
4 bottom	9.2	4.58	25.6	5.32	84	6.85
5 top	8.9	4.55	25.6	5.32	84	6.82
5 bottom	8.7	4.58	25.7	5.32	84	6.85
6 top	8.7	4.58	25.7	5.38	84	6.90
6 bottom	8.6	4.58	25.7	5.32	84	6.85
7 top	8.6	4.28	25.7	5.08	84	6.55
7 bottom	8.5	4.28	25.7	5.02	84	6.48
8 top	8.5	4.30	25.7	5.08	84	6.55
8 bottom	8.5	4.22	25.6	5.02	84	6.48
9 top	8.4	4.48	25.6	5.05	84	6.70
9 bottom	8.3	4.22	25.5	5.00	84	6.48
10 top	8.2	4.55	25.5	5.38	84	6.92
10 bottom	8.0	4.50	25.4	5.32	84	6.85
11 top	7.9	4.52	25.4	5.35	84	6.90
11 bottom	7.7	4.48	25.4	5.30	84	6.82
12 top	7.6	4.52	25.3	5.35	84	6.90

O R I G I N A L   D A T A

Section I. Water Runs

ORIGINAL DATA

Section I. Water Runs

Run W - 1. Sept. 12, 1939 - 1:08 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	150	4.57	4.24
2	122	4.59	4.34
3	143	4.57	4.49
4	150	4.55	4.49
5	152	4.54	4.52
6	122	4.59	4.35
7	196	4.18	4.08
8	230	4.15	4.00
9	251	4.45	3.97
10	241	4.61	4.51
11	250	4.63	4.49
12	249	4.59	--

Steam Pressure = 9.7 lbs./sq.in.ga.
Condenser orifice = 16.4 cm.Hg.
Condenser water rise = 18.1° C.
Condensate temp.=48°C.
Feed orifice = 4.9 cm.Hg.

Notes:
Rusty water
Smooth operation
Steam side promoted

	Pressure drop (cm. Hg.)	Liq.temp. (° C.)	Description of U - bends
Entrance		65.5	
1st Pass	0.5		
End 1st Pass		107.8	Surging liquid layer fills 1/3 of tube
2nd Pass	1.2		
End 2nd Pass		106.7	Dense vapor-liquid mixture
3rd Pass	4.7		
End 3rd Pass		104.9	Spray (fast)

Run W - 2. Sept. 12, 1939 - 1:59 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	138	4.59	4.35
2	132	4.59	4.39
3	122	4.62	4.57
4	122	4.60	4.58
5	134	4.62	4.62
6	108	4.65	4.44
7	151	4.20	4.12
8	173	4.17	4.07
9	192	4.52	4.04
10	238	4.60	4.50
11	252	4.58	4.47
12	270	4.55	--

Steam Pressure =
10.5 lbs./sq.in.ga.
Condenser Orifice =
20.5 cm.Hg.
Condenser water rise =
15.85° C.
Condensate temp.=48°C.
Feed orifice = 16.9 cm.Hg.

Notes:
Same water as W -2
Some water hammer

	Pressure drop (cm. Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		80.8	(Same as W - 1)
1st Pass	0.5		
End 1st Pass		109.7	
2nd Pass	1.0		
End 2nd Pass		108.5	
3rd Pass	4.1		
End 3rd Pass		106.9	
4th Pass	12.9		
End 4th Pass	6.1	102.3	

Run W - 3. Sept. 12, 1939 - 2:45 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	119	4.60	4.39
2	119	4.63	4.41
3	106	4.65	4.58
4	109	4.65	4.60
5	116	4.64	4.63
6	98	4.64	4.48
7	137	4.20	4.16
8	164	4.27	4.06
9	182	4.55	4.05
10	228	4.57	4.60
11	240	4.61	4.59
12	266	4.67	--

Steam Pressure =
11.4 lbs./sq.in.ga.
Condenser orifice =
18.6 cm.Hg.
Condenser water rise =
15.8° C.
Condensate temp.= 49°C.
Feed orifice = 38.1 cm.Hg.

Notes:
Same water as W-2
Smooth operation

	Pressure drop (cm. Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		88.2	
1st Pass	0.5		1/2 full of liquid. Surges
End of 1st Pass		110.7	of liquid fill U - bend.
2nd Pass	1.6		
End of 2nd Pass		109.3	1/4 full of liquid and spray
3rd Pass	3.6		
End of 3rd Pass		107.8	Dense spray
4th Pass	14.7		
End of 4th Pass	7.5	102.3	

Run W - 4. Sept. 12, 1939 - 4:42 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	273	4.95	4.53
2	282	4.99	4.72
3	264	5.06	4.88
4	288	4.99	4.89
5	308	4.98	4.94
6	252	5.04	4.64
7	340	4.63	4.45
8	357	4.60	4.38
9	354	5.00	4.46
10	354	5.11	5.02
11	196	5.19	5.07
12	83	5.18	--

Steam Pressure = 20.8 lbs./sq.in.ga.
Condenser orifice = 31.4 cm. Hg.
Condenser water rise = 21.85° C.
Condensate temp.=63.5°C.
Feed orifice=6.4 cm. Hg.

Notes:
Same water as W-3
Bad surges and water hammer.

	Pressure drop (cm. Hg.)	Liq.temp. (° C.)	Description of U - bends
Entrance		66.7	
1st Pass	1.4		
End of 1st Pass		114.0	Dense mixture. Liquid fills 1/3 of tube.
2nd Pass	4.5		
End of 2nd Pass		112.1	Spray
3rd Pass	11.0		
End of 3rd Pass		107.2	Faint fog
4th Pass	15.6		
End of 4th Pass	4.6	100.9	Baffle in separator dripping

Run W - 5

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	206	5.09	4.78
2	218	5.14	4.88
3	205	5.15	5.08
4	210	5.12	5.09
5	230	5.11	5.11
6	185	5.20	4.83
7	269	4.90	4.75
8	292	4.75	4.62
9	322	5.15	4.70
10	398	4.93	4.96
11	435	5.04	4.95
12	430	5.12	--

Steam Pressure =
23.4 lbs./sq.in.ga.
Condenser orifice =
30.7 cm.Hg.
Condenser water rise =
22.0° C.
Condensate temp. =
68° C.
Feed orifice = 34.3 cm.Hg.

Notes:
Same water as W - 4
No water hammer

	Pressure drop (cm. Hg.)	Liq. temp. ($^{\circ}$ C.)	Description of U - bends
Entrance		88.3	
1st Pass	0.5		
End of 1st Pass		119.2	1/4 full - surges
2nd Pass	2.8		
End of 2nd Pass		117.6	Dense mixture
3rd Pass	11.3		
End of 3rd Pass		114.7	Fast spray
4th Pass	29.7		
End of 4th Pass	16.2	104.7	

Run W - 6. Sept. 13, 1939 - 6:35 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	150	4.59	4.26
2	138	5.57	4.32
3	120	4.60	4.56
4	122	4.58	4.60
5	110	4.60	4.60
6	107	4.67	4.43
7	140	4.24	4.16
8	171	4.24	4.10
9	195	4.62	4.13
10	241	4.56	4.44
11	260	4.49	4.42
12	276	4.55	--

Steam Pressure = 10.3 lbs./sq.in.ga.
Condenser orifice = 17.2 cm.Hg.
Condenser water rise = 16.75^{o} C.
Condensate temp.=38^{o}C.
Feed Orifice=16.6 cm.Hg.

Notes:
Pipes swabbed, system flushed and refilled with distilled water.
Slight rusty discoloration still left on pipes.
No water hammer.
Promoted 5:45 p.m. Odor present in steam condensate.

	Pressure drop (cm. Hg.)	Liq. temp. (^{o} C.)	Description of U - bends
Entrance		78.5	
1st Pass	0.5		
End of 1st Pass		107.1*	Pipe 1/3 full of liquid. Surges of liquid
2nd Pass	1.0		
End of 2nd Pass		107.8	Dense mixture, mostly water.
3rd Pass	3.1		
End of 3rd Pass		106.2	Fast dense spray
4th Pass	13.1		
End of 4th Pass	5.6	101.8	

* Uncalibrated thermometer

Run W - 7. Sept. 13, 1939 - 7:24 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	212	5.05	4.69
2	214	5.08	4.81
3	178	5.10	4.98
4	190	5.09	5.07
5	185	5.13	5.13
6	166	5.18	4.85
7	246	4.71	4.60
8	290	4.72	4.54
9	336	5.14	4.55
10	434	4.93	4.84
11	463	4.97	4.81
12	502	5.02	--

Steam Pressure = 21.7 lbs./sq.in.ga.
Condenser orifice = 22.4 cm. Hg.
Condenser water rise = 25.3° C.
Condensate temp. = 54° C.
Feed orifice = 37.1 cm. Hg.

Notes:
Same water as W - 6.
At end of run water had faint rusty color

	Pressure drop (cm. Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		81.5	
1st Pass	0.5		
End of 1st Pass		115.6*	1/3 full. Occasional liquid surges fill the tube.
2nd Pass	2.9		
End of 2nd Pass		116.1	Dense mixture, 1/4 full of liquid
3rd Pass	15.5		
End of 3rd Pass		113.6	Fast spray
4th Pass	30.2		
End of 4th Pass	15.5	103.8	

* Uncalibrated thermometer

Run W - 8. Sept. 14, 1939 - 3:04 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	--	3.22	3.22
2	--	3.19	3.17
3	--	3.20	3.20
4	138	4.62	4.55
5	145	4.64	4.62
6	118	4.69	4.45
7	155	4.24	4.13
8	159	4.23	4.07
9	196	4.62	4.11
10	234	4.61	4.52
11	258	4.61	4.49
12	260	4.62	--

Steam Pressure =
10.1 lbs./sq.in.ga.
Condenser orifice =
13.5 cm. Hg.
Condenser water rise =
15.9° C.
Condensate Temp. = 43° C.
Feed orifice= 16.1 cm.Hg.

Notes:
Same water as W - 8
Water a little muddy.
Condenser leaked steam.
Promoted - strong ordor in steam condensate.

	Pressure drop (cm. Hg.)	Liq.temp. (° C.)	Description of U - bends
Entrance		82.4	
1st Pass	--		
End of 1st Pass		83 - 87° C.	Full of liquid *
2nd Pass	0.6		
End of 2nd Pass		105.4	1/4 full of liquid Surges of mixture
3rd Pass	2.3		
End of 3rd Pass		104.9	Dense mixture
4th Pass	10.6		
End of 4th Pass	6.2	101.8	

* Vapor bubble from Jacket 4 sometimes backed up part way into this U - bend.

Run W - 9. Sept. 14, 1939 - 4:37 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Thermocouples (M.V.) Bottom
1	—	3.40	3.40
2	—	3.39	3.32
3	—	3.40	3.40
4	226	5.23	5.19
5	246	5.27	5.21
6	205	5.22	4.84
7	283	4.71	4.50
8	3155	4.63	4.48
9	362	5.23	4.48
10	446	5.09	5.03
11	461	5.12	4.85
12	474	5.17	—

Steam Pressure = 21.2 lbs./sq.in.ga.
Condenser orifice = 26.0 cm. Hg.
Condenser water rise = 21.15° C.
Condensate temp.=63° C.
Feed orifice = 43.6 cm.Hg.

Notes:
Same water as W - 8

	Pressure drop (cm. Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		87.0	
1st Pass	—		
End of 1st Pass			Full, no bubble as in W-8
2nd Pass	0.9		
End of 2nd Pass		113.7	1/4 full - frequent surges Almost a dense mixture
3rd Pass	6.3		
End of 3rd Pass		112.4	Fast mixture - dense spray
4th Pass	24.6		
End of 4th Pass	11.4	103.8	

Run W - 10. Sept. 16, 1939 - 2:06 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	--	3.32	3.32
2	--	3.31	3.31
3	--	3.32	3.34
4	--	3.32	3.32
5	--	3.32	3.32
6	--	3.32	3.31
7	284	4.78	4.68
8	346	4.73	4.46
9	394	5.19	4.50
10	479	5.04	4.93
11	481	5.07	4.79
12	465	5.10	--

Steam Pressure = 22.1 lbs./sq.in.ga.
Condenser orifice = 20.6 cm. Hg.
Condenser water rise = 19.3° C.
Condensate temp.=59° C.
Feed orifice=31.1 cm.Hg.

Notes:
Settler drum removed.
Seal cups on manometers removed and manometers refilled with clean mercury.
System flushed and refilled with distilled water. Pipes were not swabbed.
Promotor odor in steam condensate.

	Pressure drop (cm. Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		85.9	
1st Pass	—		
End of 1st Pass		—	Full
2nd Pass	—		
End of 2nd Pass		86	Full
3rd Pass	3.8		
End of 3rd Pass		109.0	Fast mixture
4th Pass	17.6		
End of 4th Pass	9.3	102.8	

Run W - 11. Sept. 16, 1939 - 2:45 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Thermocouples (M.V.) Bottom
1	--	3.08	3.08
2	--	3.07	3.04
3	--	3.05	3.08
4	--	3.05	3.02
5	--	3.02	3.02
6	--	3.02	3.02
7	479	5.30	5.12
8	563	5.24	4.85
9	595	5.62	4.81
10	760	5.41	5.15
11	787	5.48	5.00
12	538	5.58	--

Steam Pressure = 35.1 lbs./sq.in.ga.
Condenser orifice = 26.2 cm. Hg.
Condenser water rise = 27.05° C.
Condensate temp. = 71° C.
Feed orifice = 16.0cm.Hg.

Notes:
Same water as W - 10
Unusually steady operation.
Faint promotor odor in steam condensate

	Pressure drop (cm. Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		79.3	
1st Pass	--		
End of 1st Pass		--	Full
2nd Pass	--		
End of 2nd Pass		79.5	Full
3rd Pass	5.6		
End of 3rd Pass		112.6	Fast spray
4th Pass	30.5		
End of 4th Pass	13.8	102.8	

Run W - 12. Sept. 16, 1939 - 3:20 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	--	2.39	2.39
2	--	2.39	2.33
3	--	2.39	2.39
4	--	2.35	2.35
5	--	2.37	2.37
6	--	2.38	2.35
7	1419	5.94	5.29
8	808	6.27	5.32
9	91	6.52	6.34
10	408	6.78	6.65
11	32	6.77	6.64
12	46	6.74	--

Steam Pressure = ** 46.7 lbs./sq.in.ga.
Condenser orifice = 26.1 cm. Hg.
Condenser water rise = 20.5° C.
Condensate temp.= 63° C.
Feed orifice= 4.3 cm.Hg.

Notes:
Same water as W - 11

	Pressure drop (cm. Hg.)	Liq.temp. (° C.)	Description of U - bends
Entrance		63.1	
1st Pass	--		
End of 1st Pass		--	Full
2nd Pass	--		
End of 2nd Pass		70± 1°C.	Full*
3rd Pass	10.0		
End of 3rd Pass		105.4	Faint mist
4th Pass	13.9		
End of 4th Pass	4.2	100.8	Baffle in separator dripping

* Occasional vapor bubble from J. 7 surges back into this U - bend.

** Erroneous. Thermocouple readings indicate steam pressure of about 79 lbs./sq.in.

Run W - 13. Sept. 16, 1939 - 4:05 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	--	2.85	2.85
2	--	2.85	2.80
3	--	2.85	2.85
4	--	2.80	2.80
5	--	2.80	2.80
6	--	2.80	2.80
7	789	5.61	5.35
8	773	5.54	5.17
9	797	6.00	5.02
10	1059	5.75	5.48
11	635	6.08	5.57
12	217	6.14	--

Steam Pressure = *
43.7 lbs./sq.in.ga.
Condenser orifice =
30.6 cm. Hg.
Condenser water rise =
29.3° C.
Condensate temp. = 68°C.
Feed orifice =17.2 cm.Hg.

Notes:
Same water as in W-12.

	Pressure drop (cm. Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		74.4	
1st Pass	—		
End of 1st Pass		--	Full
2nd Pass	—		
End of 2nd Pass		74.5° C.	Full (stationary bubble in top)
3rd Pass	9.7		
End of 3rd Pass		114.7	Fast light spray
4th Pass	36.5		
End of 4th Pass	16.6	102.8	Liquid flowing in separator.

* Erroneous. Thermocouple readings indicate steam pressure of about 64 lbs./sq.in.ga.

Run W - 15. Sept. 16, 1939 - 5:33 p.m.

Jacket Number	Steam Condensate gms./min.	Thermocouples (M.V.) Top	Bottom
1	--	3.20	3.22
2	--	3.20	3.15
3	--	3.22	3.22
4	--	3.18	3.18
5	--	3.18	3.18
6	--	3.20	3.20
7	870	5.83	5.40
8	812	5.78	5.38
9	800	6.32	5.19
10	1142	6.14	5.79
11		6.20	5.79
12	536	6.32	--

Steam Pressure =* 54.3 lbs./sq./in./ga.
Condenser orifice = 35.6 cm. Hg.
Condenser water rise = 30.50°C.
Condensate temp.= 75° C.
Feed orifice= 13.9 cm.Hg.

Notes:
Same water as in W-13.
As drained out after the run, water was rather rusty, although it looked quite clear in the U-bends.
Condensate collected from Jacket 11 was spoiled for measurement.

	Pressure drop (cm. Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		82.8	
1st Pass	—		
End of 1st Pass		--	Full
2nd Pass	—		
End of 2nd Pass		83.2° C.	Full
3rd Pass	13.4		
End of 3rd Pass		119.9	Thin mixture
4th Pass	47.2		
End of 4th Pass	26.0	105.9	Liquid flowing in separator.

* Erroneous. Thermocouple readings indicate steam pressure of about 70 lbs./sq.in.ga.

O R I G I N A L   D A T A

Section II - Benzene Runs

ORIGINAL DATA

Section II - Benzene Runs

Run B - 1. May 21, 1939, 3:35 to 4:35 p.m.

Jacket Number	Steam condensate gms./ 4 min.	Thermocouples (M.V.) Top	Bottom
1	545	4.06	3.75
2	460	4.03	3.80
3	425	4.05	4.00
4	490	4.05	3.90
5	560	4.05	4.00
6	510	4.05	3.75
7	605	4.00	3.83
8	160	4.00	3.78
9	70	4.00	3.98
10	170	4.15	4.13
11	60	4.15	4.13
12	40	4.13	—

Steam Pressure=1.5 lbs./ sq.in.ga.
Condenser orifice=4.8cm.
Condenser water rise = 6.5°C.
Condensate temp.=43.4°C.
Feed orifice =13.4 cm.

Notes: Steam side promoted prior to run.

	Pressure drop (cm.Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		42.7	
1st Pass	1.5		
End of 1st Pass		89	Liquid fills 1/4 of tube. Surges of dense spray.
2nd Pass	5.6		
End of 2nd Pass		86	
3rd Pass	7.0		
End of 3rd Pass		84.1	Fine spray moving fast. Became almost dry before end of run.
4th Pass	6.1		
End of 4th Pass	2.5	83.5	Separator dry.

Run B - 2. May 27, 1939, 5:25 to 6:50 p.m.

Jacket Number	Steam Condensate gms./3 min.	Thermocouples (M.V.) Top	Bottom
1	830	4.67	4.12
2	1040	4.57	4.33
3	740	4.72	4.42
4	705	4.80	4.30
5	160	4.83	4.83
6	90	4.88	4.77
7	430	4.70	4.45
8	100	4.72	4.67
9	185	4.72	4.70
10	85	4.82	4.80
11	85	4.82	4.80
12	80	4.82	—

Steam Pressure =12.7 lbs./sq.in.ga.
Condenser orifice = 5.2 cm.
Condenser water rise = 9.1°C.
Condensate temp.=45.7°C.
Feed orifice = 30.0 cm.

Note: Steam side promoted at 5 p.m. Condenser vent working hard. Steam pressure varied from 12.1 to 15.7 lbs./sq.in.ga.

	Pressure drop (cm. Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		45	
1st Pass	4.8		
End of 1st Pass		95.0	Dense mixture
2nd Pass	9.8		
End of 2nd Pass		91.0	Fine spray moving fast
3rd Pass	10.9		
End of 3rd Pass		87.5	Wisps of fog
4th Pass	11.9		
End of 4th Pass	6.2	88.0	Separator dry

Run B - 3. May 28, 1939, 2:10 to 2:40 p.m.

Jacket Number	Steam Condensate gms./ 5 min.	Thermocouples (M.V.) Top	Bottom
1	1395	7.11	6.17
2	375	7.00	6.75
3	375	7.12	7.07
4	315	7.20	7.20
5	325	7.23	7.25
6	370	7.22	7.05
7	435	7.02	6.95
8	670	7.05	6.85
9	690	6.97	6.77
10	1370	7.12	6.93
11	500	7.17	7.10
12	435	7.15	—

Steam Pressure = 108 lbs./sq.in.ga.
Condenser orifice = 6.6 cm.
Condenser water rise = 7.2 °C.
Condensate temperature = 48.7 °C.
Feed orifice = 36.4 cm.

Notes: Promoter added. Large, slow surges in operation.

	Pressure drop (cm.Hg.)	Liq. Temp. (°C.)	Description of U - bends
Entrance		52.5	
1st Pass	2.8		
End of 1st Pass		91.5	Mostly liquid
2nd Pass	2.4		
End of 2nd Pass		88.0	Fast dense mixture
3rd Pass	5.9		
End of 3rd Pass		87.7	Fast mist
4th Pass	13.3		
End of 4th Pass	8.5	81.6	Separator flowing

Run B - 4. June 1, 1939, 10:55 a.m. to 12:05 p.m.

Jacket Number	Steam Condensate gms./6 min.	Thermocouples (M.V.) Top	Bottom
1	675	4.07	3.73
2	515	4.05	3.85
3	380	4.10	4.05
4	350	4.12	4.07
5	350	4.10	4.12
6	405	4.15	3.98
7	465	4.00	3.95
8	640	4.00	3.93
9	830	3.97	3.88
10	1200	4.05	4.00
11	1120	4.05	3.95
12	505	4.15	—

Steam Pressure = 1.9 lbs./ sq.in.ga.
Condenser orifice = 6.0 cm.
Condenser water rise = 6.7°C.
Condensate temp.=45.7°C.
Feed orifice =44.2 cm.

Notes: A strong odor of promoter in condensate. Condenser vent working hard.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		55.8	
1st Pass	0.6		
End of 1st Pass		91.0	Half full of liquid. Surges of spray about every 2 seconds.
2nd Pass	1.5		
End of 2nd Pass		91.6	Mixture
3rd Pass	6.3		
End of 3rd Pass		88.7	Fast dense mist
4th Pass	12.4		
End of 4th Pass	6.2	82.1	Separator flowing

Run B - 5. June 1, 1939 - 4:30 to 5:10 p.m.

Jacket Number	Steam Condensate gms./ 7 min.	Thermocouples (M.V.) Top	Thermocouples (M.V.) Bottom
1	815	4.08	3.80
2	490	4.05	3.85
3	370	4.10	4.05
4	440	4.10	4.05
5	425	4.12	4.10
6	470	4.15	3.95
7	650	4.04	3.95
8	915	4.00	3.88
9	1070	3.97	3.85
10	1030	4.06	3.95
11	310	4.10	4.03
12	130	4.09	—

Steam Pressure = 1.5 lbs./sq.in.ga.
Condenser orifice = 4.6 cm.
Condenser water rise = 6.2 °C.
Condensate temp. = 45.0°C.
Feed orifice = 15.4 cm.

Notes: Steam condensate had strong promoter odor.

	Pressure drop (cm. Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		47.0	
1st Pass	1.1	[illegible]	Liquid fills 1/4 of tube.
End of 1st Pass		88.5	Erratic surges of spray.
2nd Pass	1.3		
End of 2nd Pass		88.1	Dense mixture
3rd Pass	5.9		
End of 3rd Pass		84.7	Fast dense mist
4th Pass	9.1		
End of 4th Pass	3.5	81.6	Separator flowing

Run B - 6. June 4, 1939: 11:30 a.m. to 12:05 p.m.

Jacket Number	Steam Condensate gms./ 6 min.	Thermocouples (M.V.) Top	Bottom
1	1510	7.23	6.40
2	415	7.17	6.83
3	450	7.25	7.20
4	415	7.35	7.32
5	425	7.37	7.37
6	460	7.40	7.18
7	495	7.20	7.15
8	600	7.23	7.13
9	580	7.18	7.05
10	1010	7.22	7.15
11	690	7.22	7.15
12	520	7.22	——

Steam Pressure = 114 lbs./sq.in.ga.
Condenser orifice = 6.4 cm.
Condenser water rise = 6.0°C.
Condensate temp.=42.5°C.
Feed orifice = 38.0 cm.

Notes: Jacket 1 filled with water and blew condensate out of vent-ten minutes after collecting condensate. Trap cleaned before next run.

	Pressure drop (cm.Hg.)	Liquid Temp. (°C.)	Description of U - bends
Entrance		54.5	
1st Pass	1.1		
End of 1st Pass		89.5	Half full of liquid
2nd Pass	2.8		
End of 2nd Pass		88.6	Dense mixture
3rd Pass	5.4		
End of 3rd Pass		86.7	Dense spray
4th Pass	10.8		
End of 4th Pass	5.4	81.5	Separator flowing

Run B - 7. June 4, 1939: 4:15 to 4:30 p.m.

Steam Pressure = 1.2 lbs./sq.in.ga.

Condenser orifice = 4.8 cm.

Condenser water rise = 6.5°C.

Condensate temperature = 44.0°C.

Feed orifice = 17.8 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./ 7 min)	890	500	390	405	475	490	630	865	1100	1200	270	130

	Pressure drop (cm.Hg.)	Liq. Temp. (°C.)	Description of U - bends
Entrance		45.5	
1st Pass	0.9		
End of 1st Pass		88.9	Erratic variation from almost full to almost empty.
2nd Pass	1.7		
End of 2nd Pass		88.1	Dense mixture
3rd Pass	6.7		
End of 3rd Pass		84.7	Dense mist
4th Pass	9.1		
End of 4th Pass	2.9	81.7	Separator flowing

Notes: Condenser vent working easy. Steam side promoted prior to run and strong odor in condensate.

Run B - 8. June 4, 1939: 4:55 to 5:10 p.m.

Steam pressure = 13.4 lbs./sq.in.ga.

Condenser orifice = 4.8 cm.

Condenser water rise = 7.95 ^{o}C.

Condensate temperature = 45.3^{o}C.

Feed orifice = 20.4 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./4 min.30 sec.)	1100	1660	1070	965	160	130	420	120	100	145	90	80

	Pressure drop (cm.Hg.)	Liq. temp. (^{o}C.)	Description of U - bends
Entrance		45.7	
1st Pass	5.7		
End of 1st Pass		92.1	Thin mixture
2nd Pass	8.9		
End of 2nd Pass		88.6	Dense mixture
3rd Pass	9.3		
End of 3rd Pass		92.1	Little wisps of mists
4th Pass	9.3		
End of 4th Pass	4.4	102	Separator bone dry

Notes: Condenser vent working easy. Slight odor of promoter in steam condensate.

Run B - 9. June 4, 1939 - 5:30 p.m.

Steam pressure = 28 lbs./sq. in. ga.

Condenser orifice = 4.8 cm.

Condenser water rise = 8.0°C.

Condensate temperature - 45.0°C.

Feed orifice = 18.4 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./4 min.)	1940	1260	360	695	155	130	330	135	150	110	80	70

	Pressure drop (cm.Hg.)	Liquid temp. (°C.)	Description of U - bends
Entrance		45.0	
1st Pass	6.1		
End of 1st Pass		91.5	Mist
2nd Pass	8.5		
End of 2nd Pass		88.9	Fine mist
3rd Pass	8.7		
End of 3rd Pass		102.8	Very faint mist
4th Pass	8.7		
End of 4th Pass	4.0	124	Separator dry

Notes: Condenser vent working easy. Steam side promoted prior to run and odor present in condensate.

Run B - 10. June 4, 1939 - 6:25 p.m.

Steam pressure = 34 lbs./sq. in. ga.

Condenser orifice = 4.8 cm.

Condenser water rise = 7.6 °C.

Condensate temperature = 44.5°C.

Feed orifice = 17.2 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./ 5 min.)	2460	935	540	935	245	210	485	230	200	205	140	160

	Pressure drop (cm.Hg.)	Liquid temp. (°C.)	Description of U - bends
Entrance		44.5	
1st Pass	5.0		
End of 1st Pass		90.5	Dense mist
2nd Pass	7.4		
End of 2nd Pass		88.1	Mist
3rd Pass	6.3		
End of 3rd Pass		95.3	Wisps of mist
4th Pass	8.0		
End of 4th Pass	3.9	122.5	Separator dry

Notes: Condenser vent working easy. Steam side promoted at 6:10 p.m. and odor present in condensate.

Run B - 11. June 4, 1939 - 7:00 p.m.

Steam pressure = 64 lbs./sq.in.ga.

Condenser orifice = 4.8 cm.

Condenser water rise = 7.1°C.

Condenser temperature = 47.0°C.

Feed orifice = 25.0 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./5 min.)	1910	410	450	445	340	450	475	380	265	675	355	330

	Pressure drop (cm.Hg.)	Liquid temp. (°C.)	Description of U - bends
Entrance		50.5	
1st Pass	2.4		
End of 1st Pass		90.4	Mixture
2nd Pass	4.5		
End of 2nd Pass		87.9	Dense spray
3rd Pass	6.5		
End of 3rd Pass		85	Dense mist
4th Pass	9.6		
End of 4th Pass	3.9	81.3	Separator flowing

Notes: Condenser vent barely operating because of smooth operation. Steam side promoted and odor present in condensate.

Run B - 12. June 4, 1939 - 7:20 p.m.

Steam pressure = 74 lbs./sq.in.ga.

Condenser orifice = 4.8 cm.

Condenser water rise = 4.15^{o}C.

Condensate temperature = 38.0^{o}C.

Feed orifice = 7.4 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./7 min.)	1395	485	460	420	385	420	445	430	280	675	340	325

	Pressure drop (cm.Hg.)	Liquid temp. (^{o}C.)	Description of U - bends
Entrance		45.5	
1st Pass	1.1		
End of 1st Pass		84.1	Mixture
2nd Pass	1.7		
End of 2nd Pass		83.3	Dense spray
3rd Pass	2.8		
End of 3rd Pass		82.4	Dense mist
4th Pass	3.7		
End of 4th Pass	0.9	81.3	Separator flowing

Notes: Odor of promoter in steam condensate.

Run B - 13. June 4, 1939 - 7:40 p.m.

Steam pressure = 70 lbs./sq.in.ga.

Condenser orifice = 4.4 cm.

Condenser water rise = 2.9°C.

Condensate temperature = 32.5°C.

Feed orifice = 2.0 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./ 9 min.)	1330	535	495	415	380	415	325	345	230	535	290	265

	Pressure drop (cm.Hg.)	Liquid temp. (°C.)	Description of U - bends
Entrance		37.5	
1st Pass	0.9		
End of 1st Pass		82.7	Mixture
2nd Pass	0.9		
End of 2nd Pass		81.7	Mixture
3rd Pass	1.3		
End of 3rd Pass		81.6	Mixture
4th Pass	1.5		
End of 4th Pass	0.2	86.8	Separator baffle dripping

Notes: Odor of promoter in steam condensate.

Run B - 14. June 4, 1939 - 8:05 p.m.

Steam pressure = 1.1 lbs./sq.in.ga.

Condenser orifice = 4.3 cm.

Condenser water rise = 6.9°C.

Condensate temperature = 41.0°C.

Feed orifice = 18.4 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./6 min.)	740	390	250	305	340	385	510	660	860	1085	415	120

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		44	
1st Pass	0.9		
End of 1st Pass		88.5	Surges of liquid
2nd Pass	1.3		
End of 2nd Pass		87.6	Liquid mixture
3rd Pass	5.4		
End of 3rd Pass		85.2	Dense spray
4th Pass	10.4		
End of 4th Pass	3.3	81.6	Separator flowing

Notes: Strong odor of promoter in steam condensate.

Run B - 1 A. June 13, 1939 - 2:35 p.m.

Steam pressure = 1.6 lbs./sq.in.ga.

Condenser orifice = 25.4 cm.

Condenser water rise = 7.15°C.

Condensate temperature = 46.0°C.

Feed orifice = 18.3 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./ 7 min.)	840	505	459	450	568	559	706	937	1095	1000	258	128

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		47.5	
1st Pass	0.9		
End of 1st Pass		89.3	Mostly liquid
2nd Pass	1.8		
End of 2nd Pass		88.4	Mixture
3rd Pass	8.1		
End of 3rd Pass		84.5	Dense spray
4th Pass	10.1		
End of 4th Pass	2.4	82.2	Separator flowing

Run B - 2 A. June 13, 1939 - 3:25 p.m.

Steam pressure = 1.6 lbs./sq.in.ga.

Condenser Orifice = 11.8 cm.

Condenser water rise = 7.35°C.

Condensate temperature = 41.0°C.

Feed orifice = 6.3 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./7 min.)	755	494	725	611	855	890	595	117	85	119	70	60

	Pressure drop (cm.Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		42.2	
1st Pass	1.1		
End of 1st Pass		85.2	1/4 full. Surges of mixture
2nd Pass	3.4		
End of 2nd Pass		84.2	Spray
3rd Pass	4.0		
End of 3rd Pass		86.0	Wisps
4th Pass	3.2		
End of 4th Pass	1.2	96.1	Separator dry

Run B - 3 A. June 13, 1939 - 3:55 p.m.

Steam pressure = 1.8 lbs./sq.in.ga.

Condenser orifice = 9.0 cm.

Condenser water rise = 5.5 °C.

Condensate temperature = 34.5°C.

Feed orifice = 2.5 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./7 min.)	735	690	894	736	258	85	70	50	48	65	50	40

	Pressure drop (cm.Hg.)	Liq. Temp. (°C.)	Description of U - bends
Entrance		36.0	
1st Pass	1.5		
End of 1st Pass		83.2	Lean mixture
2nd Pass	1.8		
End of 2nd Pass		88.9	Moist
3rd Pass	1.2		
End of 3rd Pass		96.5	Dry
4th Pass	1.2		
End of 4th Pass	0.2	101.3	Separator dry

Run B - 4 A. June 13, 1939 - 4:25 p.m.

Steam pressure = 1.7 lbs./sq.in.ga.

Condenser orifice = 8.8 cm.

Condenser water rise = 11.95°C.

Condensate temperature = 52.0°C.

Feed orifice = 17.5 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./7 min.)	730	465	469	465	607	558	726	954	1035	939	233	127

	Pressure Drop (cm.Hg.)	Liq. temp. (° C.)	Description of U - bends
Entrance		52.8	
1st Pass	0.9		
End of 1st Pass		89.3	Dense mixture
2nd Pass	1.4		
End of 2nd Pass		88.4	Spray
3rd Pass	7.9		
End of 3rd Pass		84.5	Dry
4th Pass	9.4		
End of 4th Pass	2.7	82.2	Separator dry

Run B - 5 A. June 13, 1939. 4:45 p.m.

Steam pressure = 1.8 lbs./sq.in.ga.

Condenser orifice = 9.0 cm.

Condenser water rise = 8.55 ^{o}C.

Condensate temperature = 45.0^{o}C.

Feed orifice = 6.7 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./7 min.)	690	504	738	688	910	861	585	114	84	117	117	47

	Pressure drop (cm.Hg.)	Liq. temp. (^{o}C.)	Description of U - bends
Entrance		46.8	
1st Pass	1.1		
End of 1st Pass		85.7	1/4 full. Surges of mixture
2nd Pass	3.6		
End of 2nd Pass		84.2	Spray
3rd Pass	4.4		
End of 3rd Pass		89.6	Wisps
4th Pass	3.2		
End of 4th Pass	1.4	98.2	Separator dry

Run B - 6 A. August 29, 1939 - 11:05 a.m.

Steam pressure = 2.1 lbs./sq.in.ga.

Condenser orifice = 10.2 cm.

Condenser water rise = 11.3 ^{o}C.

Condensate temperature = 50.0^{o}C.

Feed orifice = 19.1 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./5 min.)	605	390	340	384	355	448	610	740	680	612	150	105

	Pressure drop (cm.Hg.)	Liquid temp. (^{O}C.)	Description of U - bends
Entrance		48.0	
1st Pass	1.0		
End of 1st Pass		89.3	1/4 full. "Wine drops".
2nd Pass	1.1		
End of 2nd Pass		88.3	Spray
3rd Pass	8.7		
End of 3rd Pass		85.5	Dense spray
4th Pass	8.3		
End of 4th Pass	3.5	82.2	

Notes: Odor of promoter present in steam condensate.

Run B - 7 A. August 29, 1939 - 11:20 a.m.

Steam pressure = 2.1 lbs./sq.in.ga.

Condenser orifice = 10.2 cm.

Condenser water rise = 12.6°C.

Condensate temperature = 53.5°C.

Feed orifice = 39.7 cm.

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./4 min.)	423	325	230	245	240	275	350	441	582	805	665	230

	Pressure drop (cm.Hg.)	Liquid temp. (°C.)	Description of U - bends
Entrance		58.5	
1st Pass	—		
End of 1st Pass		90.3	1/2 full. Surging
2nd Pass	—		
End of 2nd Pass		91.4	Spray. "Wine drops"
3rd Pass	6.3		
End of 3rd Pass		89.0	Dense spray
4th Pass	17.0		
End of 4th Pass	6.4	81.2	

Notes: Odor of promoter present in steam condensate. Leaking seal cup spoiled pressure drop readings in first two passes.

O R I G I N A L   D A T A

Section III. - Benzene - Oil Runs

Original Data

Section III. Benzene-Oil Runs

Run B O - 1. June 14, 1939 - 4:55 p.m.

Steam pressure = 1.8 lbs./sq.in.ga.

Condenser orifice = 11.8 cm.

Condenser water rise = 8.68 °C.

Condensate temperature = 43.0°C.

Cooler orifice = ——

Cooling water rise = ——

Temperature leaving cooler = ——

Feed orifice = 20.5 cm.

Feed = 90.5% benzene

Product =60.5% benzene

Notes: No oil cooler installation.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		53.0	
1st Pass	0		Steady liquid stream,
End of 1st Pass		86.3	darker than in U - 2.
2nd Pass	1.0		
End of 2nd Pass		87.8	Light colored dense mixture. Surges.
3rd Pass	4.8		
End of 3rd Pass		87.0	Fast spray
4th Pass	12.1		
End of 4th Pass	3.1	83.3	Separator flowing

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./10 min.)	724	561	388	402	377	407	541	746	1160	1531	1195	740

Run B O - 2. June 15, 1939

Steam pressure = 1.9 lbs./sq.in.ga.

Condenser orifice = 13 cm.

Condenser water rise = 8.7°C.

Condensate temperature = 51°C.

Cooler orifice = —

Cooling water rise = —

Temperature leaving cooler = —

Feed orifice = 19.4 cm.

Feed = 93.5 % benzene

Product = 75.5% benzene

Notes: No oil cooler installation.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U-bends
Entrance		55.5	
1st Pass	0.1		
End of 1st Pass		87.3	Half-full of clear, brown liquid. Surges.
2nd Pass	0.6		
End of 2nd Pass		88.4	Mixture. Liquid surges fill tube.
3rd Pass	9.5		
End of 3rd Pass		87.5	Pale green spray
4th Pass	16.1		
End of 4th Pass	2.9	82.7	Separator flowing

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./7 min.)	595	377	313	289	289	328	446	642	960	1220	885	466

Run B 0 - 5. July 7, 1939 - 4:15 p.m.

Steam pressure = 9.1 lbs./sq.in.ga.

Condenser orifice = 9.6 cm.

Condenser water rise = 6.5 °C.

Condensate temperature = 44°C.

Cooler orifice = 8 cm.

Cooling water rise = 6.6°C.

Temperature leaving cooler = 26.5°C.

Feed orifice = 12.6 cm.

Feed = 70.5% benzene

Product = 23.0% benzene

Notes: Oil cooler installed for this and following runs. Promoted, and odor present in condensate.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		38.5	
1st Pass	1.6		
End of 1st Pass		85.0	Half-full of liquid. Surges.
2nd Pass	1.8		
End of 2nd Pass		89.0	Dense mixture
3rd Pass	4.7		
End of 3rd Pass		93.0	Fast spray
4th Pass	9.3		
End of 4th Pass	0.5	101.0	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./10 min.)	731	588	554	495	479	573	770	945	768	681	616	293

Run B 0 - 6. July 7, 1939 - 5:20 p.m.

Steam pressure = 26.0 lbs./sq.in.ga.
Condenser orifice = 9.3 cm.
Condenser water rise = 8.8°C.
Condensate temperature = 48.5°C.
Cooler orifice = 2.9 cm.
Cooling water rise = 5.0°C.
Temperature leaving cooler = 25.5°C.
Feed orifice = 8.2 cm.

Feed = 88.5 % benzene
Product = 10.5% benzene

	Pressure drop (cm.Hg.)	Liq.temp. (°C.)	Description of U - bends
Entrance		46.0	
1st Pass	5.6		
End of 1st Pass		98.7	Fast, dense spray
2nd Pass	7.6		
End of 2nd Pass		120.4	Fast, dense spray
3rd Pass	9.9		
End of 3rd Pass		127.1	Fast dense spray
4th Pass	10.5		
End of 4th Pass	1.5	130.8	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./8 min.)	2035	2390	1100	659	300	205	200	135	115	110	85	70

Run B 0 - 7. July 7, 1939 - 6:10 p.m.

Steam pressure = 22 lbs./sq.in.ga.

Condenser orifice = 4.1 cm.

Condenser water rise = 5.3°C.

Condensate temperature = 37.5°C.

Cooler orifice = 6.9 cm.

Cooling water rise = 18°C.

Temperature leaving cooler = 65°C.

Feed orifice = 38.2 cm.

Feed = 33.5% benzene

Product = 18.5% benzene

	Pressure drop (cm.Hg.)	Liq.temp. (°C.)	Description of U - bends
Entrance		59.0	
1st Pass	0.9		
End of 1st Pass		79.1	Surges of liquid fill the tube.
2nd Pass	0.7		
End of 2nd Pass		93.5	Dense mixture
3rd Pass	2.1		
End of 3rd Pass		98.2	Fast, dense spray
4th Pass	6.5		
End of 4th Pass	0.5	104.9	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./9 min.)	534	524	464	435	376	361	376	420	474	710	605	573

Run B O - 8. July 13, 1939 - 10:10 a.m.

Steam pressure = 38.5 lbs./sq.in.ga.

Condenser orifice = 3.2 cm.

Condenser water rise = 1.1°C.

Condensate temperature = 29°C.

Cooler orifice = 24.5 cm.

Cooling water rise = 12.05°C.

Temperature leaving cooler = 71.5°C.

Feed orifice = 45.2 cm.

Feed = 21.0% benzene

Product = 17.0% benzene

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		68.5	
1st Pass	0.4		
End of 1st Pass		83.7	Full of dark oil. Surges
2nd Pass	0.5		
End of 2nd Pass		94.5	Full of dark oil. Surges
3rd Pass	0.5		
End of 3rd Pass		103.3	Full, but occasional surges of spray.
4th Pass	2.7		
End of 4th Pass	1.5	110.6	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./9 min.)	447	428	420	371	326	346	361	306	405	484	317	332

Run B O - 11. July 15, 1939 - 5:35 p.m.

Steam pressure = 70 lbs./sq.in.ga.

Condenser orifice = 2.5 cm.

Condenser water rise = 2.7°C.

Condensate temperature = 32.0°C.

Cooler orifice = 22.6 cm.

Cooling water rise = 15.3°C.

Temperature leaving cooler = 83.5°C.

Feed orifice = 46.8 cm.

Feed = 16.0% benzene

Product = 9.0% benzene

Notes: Condensate drain lines lagged for this and following runs. No steam side promotion.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		78.5	
1st Pass	0		
End of 1st Pass		97.5	Full of black liquid
2nd Pass	0.8		
End of 2nd Pass		110.6	Full of black liquid
3rd Pass	0.8		
End of 3rd Pass		119.3	Full of black liquid. occasional surges of dense spray.
4th Pass	3.0		
End of 4th Pass	0.6	125.8	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./8 min.)	595	476	466	413	378	384	374	355	226	613	399	409

Run B O - 12. July 17, 1939 - 11:35 a.m.

Steam pressure = 120 lbs./sq.in.ga.

Condenser orifice = 2.9 cm.

Condenser water rise = 2.4°C.

Condensate temperature = 29°C.

Cooler orifice = 16.1 cm.

Cooling water rise = 20.85°C.

Temperature leaving cooler = 93.5°C.

Feed orifice = 40 cm.

Feed = 12.0% benzene

Product = 6.0% benzene

Notes: Steam side promoted, and odor in condensate. Jackets 6 and 7 flashed so badly that some condensate was blown out of receiver.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		87.5	
1st Pass	1.0		
End of 1st Pass		116.5	Full of oil
2nd Pass	0.6		
End of 2nd Pass		128.0	Very dense mixture
3rd Pass	1.6		
End of 3rd		137.4	Dense mixture, surges.
4th Pass	2.9		
End of 4th Pass	0.4	145.3	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./8 min.)	786	621	555	471	429	(390)	(390)	386	291	645	400	368

Run B O - 13. August 3, 1939 - 12:20 p.m.

Steam pressure = 36 lbs./sq.in.ga.

Condenser orifice = 5.0 cm.

Condenser water rise = 3.65°C.

Condensate temperature = 35.0°C.

Cooler orifice = 2.2 cm.

Cooling water rise = 29.9°C.

Temperature leaving cooler = 73.0°C.

Feed orifice = 39.8 cm.

Feed = 25.7% benzene

Product = 15.4% benzene

Notes: Settler drum installation for this and following runs. Feed orifice enlarged to 0.213 in. (No. 3 drill). Steam side promoted at 10:00 a.m. and odor present in condensate.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		68.0	
1st Pass	0.8		
End of 1st Pass		84.2	Full of liquid. Surges of spray.
2nd Pass	2.1		
End of 2nd Pass		96.6	Full of spray. Surges of liquid.
3rd Pass	0.2		
End of 3rd Pass		103.5	Dense mixture. Surges. "Wine drops".
4th Pass	7.9		
End of 4th Pass	0.8	110.6	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./9 min.)	429	439	445	419	394	372	372	412	444	815	570	522

Run B O - 14. August 3, 1939 - 1:45 p.m.

Steam pressure = 52 lbs./sq.in.ga.

Condenser orifice = 5.0 cm.

Condenser water rise = 6.45°C.

Condensate temperature = 45.5°C.

Cooler orifice = 4.4 cm.

Cooling water rise = 25.4°C.

Temperature leaving cooler = 78°C.

Feed orifice = 39.2 cm.

Feed = 31.3 % benzene

Product = 10.7% benzene

Notes: Steam side promoted at 12:45 p.m.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		68.0	
1st Pass	0.6		
End of 1st Pass		88.3	1/2 full. Surges of liquid fill the tube.
2nd Pass	1.0		
End of 2nd Pass		101.2	1/2 full. Surges of spray. "Wine drops".
3rd Pass	4.3		
End of 3rd Pass		111.0	Steady, dense spray.
4th Pass	10.6		
End of 4th Pass	0	123.3	

Jacker Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./8 min.)	455	465	467	456	445	465	601	689	656	1000	612	468

Run B O - 15. August 5, 1939 - 11:15 a.m.

Steam pressure = 80 lbs./sq.in.ga.

Condenser orifice = 6.2 cm.

Condenser water rise = 6.25°C.

Condensate temperature - 47°C.

Cooler orifice = 6.3 cm.

Cooling water rise = 24.55°C.

Temperature leaving cooler = 87°C.

Feed orifice = 38.3 cm.

Feed = 30.0% benzene

Product = 7.5% benzene

Notes: Promoted at 10:40 and odor present in condensate.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		74.0	
1st Pass	1.0		
End of 1st Pass		96.5	Half-full. Surges of liquid.
2nd Pass	2.3		
End of 2nd Pass		109.5	1/4 full. "Wine drops".
3rd Pass	7.1		
End of 3rd Pass		126.0	Steady stream of dense spray.
4th Pass	10.3		
End of 4th Pass	1.3	142.3	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./7 min.)	470	470	474	489	536	643	810	750	547	749	427	370

Run B O - 16. August 5, 1939 - 12:15 p.m.

Steam pressure = 78.3 lbs./sq.in.ga.

Condenser orifice = 6.3 cm.

Condenser water rise = 9.85°C.

Condensate temperature = 54°C.

Cooler orifice = 8.5 cm.

Cooling water rise = 19.75°C.

Temperature leaving cooler = 85°C.

Feed orifice = 39.3 cm.

Feed = 41.0% benzene

Product = 6.0% benzene

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		70	
1st Pass	1.0		
End of 1st Pass		100.0	Half full. Surges of liquid.
2nd Pass	5.3		
End of 2nd Pass		111.0	Steady stream of spray.
3rd Pass	12.5		
End of 3rd Pass		133.7	Steady stream of dense spray.
4th Pass	15.9		
End of 4th Pass	3.0	145.3	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./sec.)	1.40	1.33	1.44	2.25	2.20	1.85	2.28	1.66	1.18	1.61	0.80	0.72

Run B O - 17. August 5, 1939 - 1:40 p.m.

Steam pressure = 53.3 lbs./sq.in.ga.

Condenser orifice = 6.2 cm.

Condenser water rise = 10.4 °C.

Condensate temperature - 54.5°C.

Cooler orifice = 7.9 cm.

Cooling water rise = 17.8°C.

Temperature leaving cooler = 78.5°C.

Feed orifice = 39.7 cm.

Feed = 45.0% benzene

Product = 8.0% benzene

Notes: Odor of promoter has disappeared from condensate.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		65.5	
1st Pass	0.8		
End of 1st Pass		95.0	Half-full. Surges of liquid.
2nd Pass	3.6		
End of 2nd Pass		105.4	One-fourth full.
3rd Pass	11.6		
End of 3rd Pass		121.0	Steady stream of dense spray.
4th Pass	16.8		
End of 4th Pass	3.7	133.0	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./7 min.)	589	540	541	628	945	734	1150	815	554	700	380	322

Run B 0 - 18. August 6, 1939 - 3:50 p.m.

Steam pressure = 119 lbs./sq.in.ga.

Condenser orifice = 2.5 cm.

Condenser water rise = 3.8°C.

Condensate temperature = 26.0°C.

Cooler orifice = 18.2 cm.

Cooling water rise = 19.7°C.

Temperature leaving cooler = 95.0°C.

Feed orifice = 42.9 cm.

Feed = 13.0% benzene

Product = 6.0% benzene

Notes: Steam side promoted at 2:20 p.m. Very slight odor in condensate.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		85.0	
1st Pass	1.9		
End of 1st Pass		116.0	Full of liquid. Ebbs to half-full.
2nd Pass	1.3		
End of 2nd Pass		128.7	Full of liquid. Surges of spray.
3rd Pass	3.2		
End of 3rd Pass		141.1	Spray
4th Pass	3.3		
End 4th Pass	0.5	150.5	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./6 min.)	615	465	435	390	350	360	390	346	351	459	312	282

Run B 0 - 19. August 6, 1939 - 4:30 p.m.

Steam pressure = 119 lbs./sq.in.ga.

Condenser orifice = 6.3 cm.

Condenser water rise = 7.65 °C.

Condensate temperature = 32.5°C.

Cooler orifice = 36 cm.

Cooling water rise = 13.25°C.

Temperature leaving cooler = 96.0°C.

Feed orifice = 41.7 cm.

Feed = 29.0% benzene

Product = 4.0% benzene

Notes. Steam side not promoted.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		79.0	
1st Pass	1.7		
End 1st Pass		107.3	Liquid surges
2nd Pass	5.2		
End of 2nd Pass		125.6	Dense spray
3rd Pass	9.5		
End 3rd Pass		146.2	Dense spray
4th Pass	12.0		
End 4th Pass	2.2	159.0	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./6 min.)	503	514	562	814	714	548	732	515	403	560	332	254

Run B O - 20.   Autust 6, 1939 - 4:55 p.m.

Steam pressure = 120 lbs./sq.in.ga.

Condenser orifice = 10.7 cm.

Condenser water rise = 9.2°C.

Condensate temperature = 42.5°C.

Cooler orifice - 32.6 cm.

Cooling water rise = 12.1°C.

Temperature leaving cooler = 93.0°C.

Feed orifice = 40.7 cm.

Feed = 43.0% benzene

Product = 4.0% benzene

Notes: Steam side not promoted.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		70	
1st Pass	2.8		
End of 1st Pass		108.0	Light mixture
2nd Pass	9.8		
End of 2nd Pass		130.8	Dense spray
3rd Pass	14.5		
End of 3rd Pass		153	Dense spray
4th Pass	18.1		
End of 4th Pass	4.0	163	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./sec.)	1.81	2.13	2.88	3.64	2.53	1.74	1.94	1.25	0.96	1.36	0.75	0.61

Run B O - 21. August 6, 1939 - 5:40 p.m.

Steam pressure - 120 lbs./sq.in.ga.

Condenser orifice = 12.7 cm.

Condenser water rise = 10.4°C.

Condensate temperature = 54.5°C.

Cooler orifice = 31.1 cm.

Cooling water rise = 10.4°C.

Temperature leaving cooler = 94.5°C.

Feed orifice = 42.1 cm.

Feed = 56.5% benzene

Product = 4.0% benzene

Notes: Steam side not promoted.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		66.0	
1st Pass	4.5		
End of 1st Pass		110.8	Dense spray
2nd Pass	13.1		
End of 2nd Pass		136.0	Dense spray
3rd Pass	18.8		
End of 3rd Pass		157	Dense spray
4th Pass	24.0		
End of 4th Pass	6.5	165	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./6 min.)	880	1440	1205	1620	1000	560	668	390	312	474	234	191

Run B O - 22. August 8, 1939 - 10:55 a.m.

Steam pressure = 35.5 lbs./sq.in.ga.

Condenser orifice = 6.1 cm.

Condenser water rise = 9.45°C.

Condensate temperature = 49.0°C.

Cooler orifice = 21 cm.

Cooling water rise = 9.35°C.

Temperature leaving cooler = 69.5°C.

Feed orifice = 41.0 cm.

Feed = 46.0% benzene

Product = 14.0% benzene

Notes: Steam side promoted at 10:00 a.m. Slight odor in condensate.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		58.5	
1st Pass	0.7		
End of 1st Pass		85.0	1/2 full. Liquid surges
2nd Pass	1.2		
End of 2nd Pass		98.1	Dense mixture
3rd Pass	7.3		
End of 3rd Pass		105.0	Dense spray
4th Pass	15.2		
End 4th Pass	3.5	118.4	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms.7 min.)	472	475	456	447	444	488	725	865	747	970	583	382

Run B O - 23. August 8, 1939 - 1:10 p.m.

Steam pressure = 37 lbs./sq.in.ga.

Condenser orifice = 11.2 cm.

Condenser water rise = 10.3°C.

Condensate temperature 52.5°C.

Cooler orifice = 19 cm.

Cooling water rise = 7.95 °C.

Temperature leaving cooler = 64.0°C.

Feed orifice = 37.7 cm.

Feed = 62.0% benzene

Product = 13.0% benzene

Notes: Steam side promoted 12:35 and odor present in condensate.

	Pressure drop (cm.Hg.)	Liq.temp. (°C.)	Description of U - bends
Entrance		52.0	
1st Pass	1.1		
End 1st Pass		89.0	Half full of liquid. Surges back and forth.
2nd Pass	4.3		
End of 2nd Pass		98.7	Dense spray
3rd Pass	11.9		
End of 3rd Pass		109.5	Dense spray
4th Pass	21.2		
End of 4th Pass	5.8	121.5	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./7 min.)	620	590	567	567	694	874	1532	1113	675	869	450	329

Run B 0 - 24. August 8, 1939 - 1:40 p.m.

Steam pressure = 74 lbs./sq.in.ga.

Condenser orifice = 10.8 cm.

Condenser water rise = 12.9°C.

Condensate temperature = 61.0°C.

Cooler orifice = 19 cm.

Cooling water rise = 25.95°C.

Temperature leaving cooler = 58.0 °C.

Feed orifice = 36.7 cm.

Feed = 76.0% benzene

Product = 7.0% benzene

Notes: Pressure drop across 1st two passes doubtful because of leaking seal cup.

	Pressure drop (cm.Hg.)	Liq. temp. (°C.)	Description of U - bends
Entrance		56.0	
1st Pass	(8.5)		
End 1st Pass		95.5	Dense spray
2nd Pass	(11.7)		
End 2nd Pass		127.5	Dense spray
3rd Pass	27.7		
End 3rd Pass		144	Dense spray
4th Pass	29.0		
End 4th Pass	7.0	151.6	

Jacket Number	1	2	3	4	5	6	7	8	9	10	11	12
Steam Condensate (gms./5 min.)	800	1310	1380	1315	635	383	495	266	221	266	167	143

C A L C U L A T E D   D A T A

Section I. Water Runs

Run W - 1

Condensing steam temperature = 114.8 °C.

Feed rate (W) = 372 lbs./hr.

Total heat transfer = 273,000 B.t.u./hr. (by steam side)

= 263,000 B.t.u./hr. (by condenser)

Initial boiling point = 70% of J. 2

Jacket	q/A	Δt, °C	U	% vapor leaving	Average % vapor	Exit velocity (F. P. S.)	Average velocity (geom. mean)
1	20,000	$(34.1)_{lm}$	325	t=92.4°C.	% —	.28	—
2	16,000	$(11.5)_{lm}$	770	1	—	3.9	—
3	19,000	8	1320	6	4	22	9.3
4	20,000	8	1390	11	9	40	30
5	20,000	8	1390	16	13	58	48
6	15,000	8	1040	20	—	73	65
7	27,000	8 1/2	1760	26 1/2	23	98	85
8	32,000	9	1980	34	31	125	111
9	35,000	9 1/2	2040	43	39	165	144
10	34,000	10 1/2	1800	51	47	200	182
11	35,000	11 1/2	1690	61	56	250	224
12	35,000	12 1/2	1550	69	65	300	274

	Pressure cm.Hg.ga.	Sat. temp. ° C.	Cumulative Pressure drop (cm. Hg.)
Entrance	20.8	106.8	—
End 1st Pass	20.3	106.7	.5
End 2nd Pass	19.1	106.4	1.7
End 3rd Pass	14.4	104.8	6.4
End 4th Pass	4.0	101.4	16.8

Run W - 1

Condensing steam temperature = 115.8°C.

Feed rate (W) = 690 lbs./hr.

Total heat transfer = 244,000 B.t.u./Hr.(by steam side)
= 257,000 B.t.u./hr. (by condenser)
Initial boiling point = 10% of J.3

Jacket	q/A	Δt, °C.	U	% vapor leaving	Average % vapor	Exit Velocity (F.P.S.)	Average velocity (geom.mean)
1	18,000	$(28.1)_{1m}$	356	t_2=93.6	—	.53	—
2	18,000	$(14.8)_{1m}$	675	t =106.6	—	.53	—
3	16,000	8	1110	2	1	13.5	2.7
4	16,000	8	1110	4	3	26	19
5	18,000	8	1250	7	6	46	35
6	15,000	8	1040	9	—	59	52
7	20,000	8	1390	11 1/2	10	76	67
8	24,000	8 1/2	1570	14 1/2	13	96	85
9	26,000	9	1610	18	16	120	107
10	33,000	10	1840	22	20	155	136
11	35,000	11	1770	27	24	195	174
12	38,000	12 1/2	1690	33	30	260	225

	Pressure Cm.Hg.ga.	Sat. temp. (° C.)	Cumulative pressure drop (cm. Hg.)
Entrance	24.6	108	—
End of 1st Pass	24.1	107.9	.5
End of 2nd Pass	23.1	107.6	1.5
End of 3rd Pass	19.0	106.3	5.6
End of 4th Pass	6.1	102.2	18.5

Run W - 3

Condensing steam temperature = 116.9°C.

Feed rate (W) = 1037 lbs./hr.

Total heat transfer = 227,000 B.t.u./hr.(by steam side)

= 245,000 B.t.u./hr.(by condenser)

Initial boiling point = 90% of J. 3

Jacket	q/A	Δt, °C.	U	% vapor leaving	Average % vapor	Exit Velocity (F.P.S.)	Average Velocity (geom.mean)
1	16,000	$(24.9)_{lm}$	357	t_2= 95.7	—	.79	—
2	16,000	$(17.2)_{lm}$	517	t =103.2	—	.79	—
3	14,000	$(10.2)_{av}$	760	0.1	—	1.75	—
4	15,000	8	1040	1	1	10	4.2
5	16,000	8 1/2	1050	3	2	29	17
6	13,000	8 1/2	850	4	—	39	34
7	18,000	9	1110	6	5	59	48
8	23,000	9	1420	8	7	79	68
9	25,000	9	1540	10	9	99	88
10	32,000	10	1780	13	11	130	113
11	34,000	11	1720	16	15	170	149
12	38,000	13	1620	20	18	230	198

	Pressure cm.Hg.ga.	Sat. temp. (° C)	Cumulative pressure drop (cm. Hg.)
Entrance	27.9	108.9	—
End of 1st Pass	27.4	108.8	.5
End of 2nd Pass	25.8	108.3	2.1
End of 3rd Pass	22.2	107.3	5.7
End of 4th Pass	7.5	102.6	20.4

Run W - 4

Condensing steam temperature = 126.6 °C.

Feed rate of (W) = 425 lbs./hr.

Total heat transfer = 417,000 B.t.u./hr (by steam side)

= 435,000 B.t.u./hr. (by condenser)

Initial boiling point = Start of J. 2

Jacket	q/A	Δt, °C.	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom.mean)
1	39,000	$(32.8)_{l.m.}$	660	0	—	0.32	0.32
2	40,000	15	1480	8 1/2	4	31	3.2
3	38,000	15 1/2	1360	16 1/2	13	60	43
4	41,000	15 1/2	1470	25	21	91	74
5	44,000	16	1530	35	30	130	109
6	35,000	16 1/2	1180	43	—	160	144
7	48,000	17 1/2	1530	53	48	200	179
8	51,000	18 1/2	1530	64	58	250	224
9	50,000	19 1/2	1430	75	70	310	278
10	51,000	20 1/2	1380	86 1/2	81	380	344
11	27,000	22	680	93	90	430	405
12	10,000	24	232	95	94	460	445

	Pressure cm.Hg.ga.	Sat. temp. (° C.)	Cumulative pressure drop (cm. Hg.)
Entrance	37.1	111.6	—
End of 1st Pass	35.7	111.2	1.4
End of 2nd Pass	31.2	109.9	5.9
End of 3rd Pass	20.2	106.6	16.9
End of 4th Pass	4.6	101.6	32.5

Run W - 5

Condensing steam temperature = 128.9° C.

Feed rate (W) = 984 lbs./hr.

Total heat transfer = 423,000 B.t.u./hr. (by steam side)

= 433,000 B.t.u./hr. (by condenser)

Initial boiling point = start of J. 3

Jacket	q/A	Δt,°C	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom.mean)
1	28,000	(33.0) l.m.	470	t=102.4	—	.75	—
2	31,000	(17.9) l.m.	960	0	—	.75	—
3	28,000	11 1/2	1350	3	2	21	4.0
4	30,000	12	1390	5 1/2	4	39	29
5	32,000	12	1480	8 1/2	7	60	48
6	26,000	12	1200	11	—	78	68
7	38,000	12 1/2	1690	14 1/2	13	105	91
8	41,000	13 1/2	1690	18 1/2	17	135	119
9	46,000	15	1700	23	21	175	154
10	57,000	16	1980	28 1/2	26	230	200
11	63,000	18 1/2	1890	35	32	310	267
12	63,000	20	1750	41	38	410	357

	Pressure cm.Hg.ga.	Sat. temp. (°C.)	Cumulative pressure drop (cm. Hg.)
Entrance	60.5	117.3	—
End of 1st Pass	60.0	117.2	.5
End of 2nd Pass	57.2	116.6	3.3
End of 3rd Pass	45.9	113.8	14.6
End of 4th Pass	16.2	105.5	44.3

Run W - 6

Condensing steam temperature = 115.6°C.

Feed rate (W) = 685 lbs./hr.

Total heat transfer = 245,000 B.t.u./hr. (by steam side)

= 249,000 B.t.u./hr. (by condenser)

Initial boiling point = 10 % of J. 3

Jacket	q/A	Δt.°C.	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom.mean)
1	20,000	(29.4) l.m.	380	t_2=92.8	—	.52	—
2	18,000	(16.0) l.m.	625	t =105.0	—	.52	—
3	16,000	(8 1/2) av	1040	1 1/2	1	10.5	2.3
4	16,000	8	1110	4	3	27	17
5	15,000	8	1040	6	5	40	33
6	14,000	8	970	8	—	53	46
7	19,000	8	1320	11	10	73	62
8	24,000	8 1/2	1570	14	13	93	82
9	27,000	9	1670	17 1/2	16	120	106
10	34,000	9 1/2	1990	22	20	150	134
11	36,000	10 1/2	1900	27	25	195	171
12	39,000	12	1800	33	30	250	221

	Pressure cm.Hg.ga.	Sat. temp. (°C.)	Cumulative pressure drop (cm.Hg.)
Entrance	23.3	107.7	—
End of 1st Pass	22.8	107.5	.5
End of 2nd Pass	21.8	107.2	1.5
End of 3rd Pass	18.7	106.2	4.6
End of 4th Pass	5.6	101.9	17.7

Run W - 7

Condensing steam temperature = 127.3°C.

Feed rate (W) = 1022 lbs./hr.

Total heat transfer = 426,000 B.t.u./hr. (by steam side)

= 428 B.t.u./hr. (by condenser)

Initial boiling point = 70 % J.3

Jacket	q/A	Δt, °C.	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom.mean)
1	30,000	$(38.4)_{l.m.}$	434	t=95.7	—	.78	—
2	30,000	$(24.0)_{l.m.}$	695	t=109.7	—	.78	—
3	25,000	$(11\ 1/2)_{av}$	1210	0.7	—	5.7	—
4	26,000	9	1600	3	2	22	11
5	26,000	9	1610	6	5	43	31
6	23,000	9 1/2	1350	8	—	58	50
7	34,000	10 1/2	1800	11	10	80	68
8	41,000	11 1/2	1980	15	13	115	96
9	48,000	13	2050	19	17	150	131
10	63,000	15	2330	25	23	210	175
11	67,000	17 1/2	2130	32	29	300	251
12	73,000	20 1/2	1980	39	36	400	346

	Pressure cm.Hg.ga.	Sat. temp. (° C.)	Cumulative pressure drop (cm.Hg.)
Entrance	64.6	118.3	—
End of 1st Pass	64.1	118.2	.5
End of 2nd Pass	61.2	117.5	3.4
End of 3rd Pass	45.7	113.7	18.9
End of 4th Pass	15.5	105.2	49.1

Run W - 8

Condensing steam temperature = 115.3°C.

Feed rate (W) = 674 lbs./hr.

Total heat transfer = 200,000 B.t.u./hr. (by steam side)

= 211,000 B.t.u./hr. (by condenser)

Initial boiling point = 80 % J. 5

Jacket	q/A	Δt,°C.	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom.mean)
4	18,000	(25.6) l.m.	390	t=95.8	—	.51	.51
5	19,000	(12.6) av.	840	1	—	7.2	1.9
6	16,000	9 1/2	935	3	—	21	12.3
7	20,000	9 1/2	1170	6	5	41	29
8	22,000	9 1/2	1290	9	8	62	50
9	27,000	10	1500	12	11	83	72
10	33,000	10 1/2	1750	17	14	120	100
11	36,000	11 1/2	1740	22 1/2	20	165	141
12	35,000	12 1/2	1560	27	25	205	184

	Pressure cm.Hg.ga.	Sat. temp. (° C.)	Cumulative pressure drop (cm.Hg.)
Entrance	18.4	106.2	—
End of 2nd Pass	17.8	105.9	.6
End of 3rd Pass	15.7	105.3	2.7
End of 4th Pass	5.9	102.0	12.5

Run W - 9

Condensing steam temperature = 127°C.

Feed rate (W) = 1,110 lbs./hr.

Total heat transfer = 376,000 B.t.u./hr. (by steam side)

= 386,000 B.t.u./hr. (by condenser)

Initial boiling point = 80% of J. 5

Jacket	q/A	Δt, °C.	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom.mean)
4	32,000	(32.6)	545	t=101	—	.85	0.85
5	34,000	(18 1/2) l.m. av.	1020	1/2	—	5.3	2.1
6	28,000	14	1110	2 1/2	—	24	11.3
7	40,000	14 1/2	1530	6	4	55	37
8	44,000	15	1630	10	8	92	71
9	51,000	15 1/2	1830	14	12	130	109
10	64,000	16 1/2	2220	20	17	195	159
11	66,000	18	2040	26	23	260	225
12	68,000	22	1720	32	29	380	315

	Pressure cm. Hg.	Sat. temp. (°C.)	Cumulative pressure drop (cm. Hg.)
Entrance	43.2	113.2	—
End of 2nd Pass	42.3	112.9	.9
End of 3rd Pass	36.0	111.2	7.2
End of 4th Pass	11.4	103.9	31.8

Run W - 10

Condensing steam temperature = 127.8°C.

Feed rate (W) = 935 lbs./hr.

Total heat transfer = 308,000 B.t.u./hr. (by steam side)

= 309,000 B.t.u./Hr. (by condenser)

Initial boiling point = 10% of J. 8

Jacket	q/A	Δt, °C.	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom. mean)
7	40,000	$(30.5)_{l.m.}$	730	t=106.6	—	.71	.71
8	49,000	19	1430	4 1/2	2	39	5.3
9	56,000	19	1640	9 1/2	7	83	57
10	69,000	20	1920	17	13	150	112
11	69,000	21 1/2	1780	24	20	225	184
12	67,000	23 1/2	1590	30 1/2	27	310	264

	Pressure (cm.Hg.)	Sat. temp. (°C.)	Cumulative pressure drop (cm. Hg.)
Entrance	28.8	109.2	—
End of 3rd Pass	25.3	108.3	3.5
End of 4th Pass	9.0	103.1	19.8

Run W - 11

Condensing steam temperature = 138.1°C.

Feed rate (W) = 672 lbs./hr.

Total heat transfer = 465,000 B.t.u./hr. (by steam side)

= 494,000 B.t.u./hr. (by condenser)

Initial boiling point = 70 % of J.7.

Jacket	q/A	Δt, °C.	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom.mean)
*						(.51)	
7	69,000	(34 1/2) av.	1110	3	—	16	2.9
8	67,000	24	1550	12	7	64	32
9	86,000	25	1910	24	18	130	91
10	111,000	26 1/2	2330	39 1/2	31	225	171
11	116,000	28	2260	56	48	350	280
12	78,000	31 1/2	1380	67	62	470	405

	Pressure (cm. Hg.)	Sat. temp. (°C.)	Cumulative pressure drop (cm. Hg.)
Entrance	46.7	114	—
End of 3rd Pass	41.5	112.7	5.2
End of 4th Pass	13.3	104.6	33.4

Run W - 12

Condensing steam temperature = 162°C. (estimated from T C readings)

Feed rate (W)= 348 lbs./hr.

Total heat transfer = 358,000 B.t.u./hr. (by steam side)

= 375,000 B.t.u./hr. (by condenser)

Initial boiling point = 20% of J. 7

Jacket	q/A	Δt, °C.	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom. mean)
~~6~~						(.27)	
7	210,000	(69) av.	1690	47	18	150	6.4
8	119,000	55	1200	78	66	260	198
9	11,000	56	109	81 1/2	80	290	275
10	59,000	57 1/2	570	97	93	360	323
11	2,000	58 1/2	19	98	98	380	370
12	5,000	60	46	99	99	400	390

	Pressure (cm.Hg.)	Sat. temp. (°C.)	Cumulative pressure drop (cm.Hg.)
Entrance	26.2	108.5	—
End of 3rd Pass	16.9	105.6	9.3
End of 4th Pass	4.0	101.4	22.2

Run W - 13

Condensing steam temperature = 155°C. (estimated from T.C. reading)

Feed rate (W) = 696 lbs./hr.

Total heat transfer = 552,000 B.t.u./hr. (by steam side)

= 572,000 B.t.u./hr. (by condenser)

Initial boiling point = 50% of J. 7

Jacket	q/A	Δt^o, C.	U	% vapor leaving	Average % vapor	Exit veloc. (F.P.S.)	Average veloc. (geom.mean)
7	116,000	(48 1/2)av.	1330	7	—	(.53) 35	4.3
8	114,000	39	1630	22 1/2	15	115	63
9	117,000	39 1/2	1650	38	30	200	152
10	157,000	41	2130	60	50	330	257
11	93,000	44	1170	72	66	440	381
12	31,000	47 1/2	360	76 1/2	75	540	486

	Pressure (cm.Hg.)	Sat. temp. (°C.)	Cumulative pressure drop (cm. Hg.)
Entrance	58.8	116.8	—
End of 3rd Pass	49.8	114.7	9.0
End of 4th Pass	16.0	105.4	42.8

Run W - 15

Condensing steam temperature = 158°C. (estimated from T.C. reading)

Feed rate (W) = 626 lbs./hr.

Total heat transfer = 616,000* B.t.u./hr. (by steam side)

= 635,000 B.t.u./hr. (by condenser)

Initial boiling point = 40% of J. 7

Jacket	q/A	Δt, °C.	U	% vapor leaving	Average % vapor	Exit velocity (F.P.S.)	Average velocity (geom. mean)
						(.48)	
7	128,000	(44) av.	1620	12	—	47	4.8
8	119,000	38	1740	30	21	120	75
9	118,000	39 1/2	1660	47 1/2	39	195	153
10	169,000	42	2240	73	64	330	254
11	86,000*	44 1/2	—	(86)	—	430	—
12	78,000	48	900	(98)	—	570	—

	Pressure (cm.Hg.)	Sat. temp. (°C.)	Cumulative pressure drop (cm. Hg.)
Entrance	81.1	121.7	—
End of 3rd Pass	68.7	117.2	12.4
End of 4th Pass	25.0	108.1	56.1

* Heat transfer in Jacket 11 very uncertain.

C A L C U L A T E D   D A T A

Section II - Benzene Runs

Run B - 1

Condensing steam temperature = 102.5°C.

Feed rate = 615 lbs./hr.

Total heat transfer = 125,000 B.t.u./hr. (by steam side)

= 124,000 B.t.u./hr. (by condenser)

Initial boiling point = 40% of J. 2

Final boiling point = 50% of J. 12

Jacket	q/A	Δt. °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	19,300	39	275	t=80°C.	—	0.57	.57
2	16,400	16	570	8	1	11	2.6
3	15,100	14	600	21	14	29	18
4	17,400	14	690	36	29	50	38
5	19,800	14 1/2	760	52	44	73	60
6	18,200	15 1/2	655	67	59	96	84
7	21,600	16	750	86	78	125	110
8	5,300	17	170	90	88	140	130
9	2,200	18 1/2	65	93	92	140	140
10	5,700	19 1/2	165	98	96	160	150
11	1,700	21	45	99.6	99	165	160
12	1,000	20 1/2	25	vapor	—	170	165

	Pressure (cm.Hg.ga.)	Saturation (Temp., °C.)
Entering 1st Pass	22.7	89.8
End 1st Pass	21.2	89.4
End 2nd Pass	15.6	87.2
End 3rd Pass	8.6	84.0
End 4th Pass	2.5	81.2

Run B - 2

Condensing steam temperature = 117.5°C.

Feed rate = 906 lbs./hr.

Total heat transfer = 183,000 B.t.u./hr. (by steam side)

= 180,000 B.t.u./hr. (by condenser)

Initial boiling point = 10% of J. 2

Final boiling point = 50% of J. 12

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	38,700	44 1/2	480	t=91.5	—	0.84	0.84
2	48,500	22 1/2	1200	27	13	47	6.3
3	34,500	23	830	48	38	85	63
4	33,900	23 1/2	800	68	58	120	100
5	6,900	24 1/2	160	72	70	130	125
6	4,200	25 1/2	92	75	74	140	135
7	20,300	26 1/2	425	87	81	165	150
8	4,200	27 1/2	87	90	89	180	170
9	8,200	29	155	94	92	190	185
10	3,500	30 1/2	65	97	95	210	200
11	3,400	31 1/2	60	99.2	98	220	215
12	3,200	30 1/2	58	vapor	—	240	230

	Pressure (cm.Hg.ga.)	Saturation (Temp., °C.)
Entering 1st Pass	43.6	95.8
End 1st Pass	38.8	94.4
End 2nd Pass	29.0	91.4
End 3rd Pass	18.1	87.4
End 4th Pass	6.2	82.8

Run B - 3

Condensing steam temperature = 170°C.

Feed rate = 990 lbs./hr.

Total heat transfer = 161,000 B.t.u./hr. (by steam side)

= 162,000 B.t.u./hr. (by condenser)

Initial boiling point = 10% of J. 2

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	35,500	98	200	t=92.1	—	.93	.93
2	9,200	78	66	5	2	11	3.2
3	9,200	78	66	9	7	19	14
4	7,700	78 1/2	55	14	12	31	24
5	7,900	79	55	19	17	40	35
6	9,100	79 1/2	63	24	22	52	46
7	10,800	80 1/2	73	30	27	67	59
8	16,900	81 1/2	115	39	34	89	77
9	17,400	82 1/2	118	54	46	125	105
10	34,800	83 1/2	230	68	61	160	140
11	12,400	85 1/2	81	76	72	190	175
12	10,800	87 1/2	69	82	79	220	205

	Pressure (cm.Hg.ga.)	Saturation (temp.°C.)
Entering 1st Pass	32.9	93.4
End 1st Pass	30.1	92.4
End 2nd Pass	27.7	91.6
End 3rd Pass	21.8	89.4
End 4th Pass	8.5	84.0

Run B - 4

Condensing steam temperature = 103.4°C.

Feed rate = 1080 lbs./hr.

Total heat transfer = 153,000 B.t.u./hr (by steam side)

= 144,000 B.t.u./hr. (by condenser)

Initial boiling point = 90% of J. 3

Jacket	q/A	Δt,°C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	15,400	40 1/2	211	t=72°C.	—	1.0	1.0
2	11,600	28	230	t=82°C.	—	1.0	1.0
3	8,400	17 1/2	264	.4	—	1.0	1.0
4	7,700	13	330	3	2	5.5	2.3
5	7,800	13	330	6	5	15	9.1
6	8,900	13 1/2	370	11	9	27	20
7	10,000	14	400	16	13	38	32
8	14,500	14 1/2	555	24	20	59	47
9	19,100	15 1/2	685	33	29	82	70
10	27,800	16 1/2	935	46	40	120	100
11	25,800	18	795	61	53	160	140
12	11,400	19 1/2	325	66	64	185	170

	Pressure (cm.Hg.ga.)	Saturation (Temp.°C.)
Entering 1st Pass	27.0	90.7
End 1st Pass	26.4	90.5
End 2nd Pass	24.9	90.0
End 3rd Pass	18.6	87.8
End 4th Pass	6.2	82.8

Run B - 5

Condensing steam temperature = 102.5°C.

Feed rate = 650 lbs./hr.

Total heat transfer = 125,000 B.t.u./hr. (by steam side)

= 117,000 B.t.u./hr. (by condenser)

Initial boiling point = 95% of J. 2

Jacket	q/A	Δt,°C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	16,000	41	218	t=73°C	—	.60	.60
2	9,500	20	265	0.4	—	2.0	1.1
3	6,900	13	290	8	5	12	4.9
4	8,400	13 1/2	340	14	11	21	16
5	8,200	14	325	21	18	32	26
6	9,100	14 1/2	350	28	24	41	36
7	12,700	15	470	39	33	59	49
8	18,000	15 1/2	645	54	46	83	70
9	21,100	16 1/2	710	71	63	110	95
10	20,300	17 1/2	645	88	80	140	125
11	5,800	19	114	93	91	155	150
12	2,100	20	57	96	95	165	160

	Pressure (cm.Hg.ga.)	Saturation (Temp.°C.)
Entering 1st Pass	20.9	89.0
End 1st Pass	19.8	88.5
End 2nd Pass	18.5	88.2
End 3rd Pass	12.6	85.8
End 4th Pass	3.5	81.8

Run B - 6

Condensing steam temperature = 172°C.

Feed rate = 1010 lbs./hr.

Total heat transfer = 138,000 B.t.u./hr. (by steam side)

= 133,000 B.t.u./hr. (by condenser)

Initial boiling point = 20% of J. 2

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	31,900	103	172	t=84°C.	—	0.94	0.94
2	8,200	83	58	3	1	7.3	2.6
3	9,000	82	61	8	5	18	11.5
4	8,200	82	54	12	10	27	22
5	8,400	82 1/2	56	16	14	36	31
6	9,200	83	62	21	19	48	42
7	10,000	83 1/2	68	27	24	62	55
8	12,300	84	82	33	30	76	69
9	11,700	84 1/2	76	40	37	94	85
10	21,000	85 1/2	137	51	45	120	105
11	14,200	87	91	59	55	150	135
12	10,500	88 1/2	66	66	63	175	160

	Pressure (cm.Hg.ga.)	Saturation (Temp.°C.)
Entering 1st Pass	25.5	90.2
End 1st Pass	24.4	89.8
End 2nd Pass	21.6	88.8
End 3rd Pass	16.2	87.2
End 4th Pass	5.4	82.4

Run B - 7

Condensing steam temperature = 102°C.

Feed rate = 700 lbs./hr.

Total heat transfer = 129,000 B.t.u./hr. (by steam side)

= 125,000 B.t.u./hr. (by condenser)

Initial boiling point = 10% lf J. 3

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	17,600	44	222	t = 68°C.	—	.65	.65
2	9,700	24	224	t = 86°C.	—	.65	.65
3	7,400	13 1/2	305	5	2	8.4	2.3
4	7,800	13 1/2	320	11	8	18	12
5	9,200	13 1/2	375	18	15	29	23
6	9,500	14	380	26	22	42	35
7	12,400	14 1/2	475	35	30	57	49
8	18,100	15	670	48	41	80	68
9	21,900	16	760	64	56	110	94
10	25,000	17 1/2	790	84	74	150	130
11	5,000	18 1/2	150	89	86	160	155
12	2,200	19 1/2	60	91	90	170	165

	Pressure (cm.Hg.ga.)	Saturation (Temp. °C.)
Entering 1st Pass	21.3	88.8
End 1st Pass	20.4	88.4
End 2nd Pass	18.7	87.8
End 3rd Pass	12.0	85.2
End 4th Pass	2.9	81.4

Run B - 8

Condensing steam temperature = 119.5°C.

Feed rate = 750 lbs./hr.

Total heat transfer = 162,000 B.t.u./hr. (by steam side)

= 151,000 B.t.u./hr. (by condenser)

Initial boiling point = beginning of J. 2

Final boiling point = 50% of J. 10

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	33,000	45 1/2	403	0	—	.69	.69
2	50,100	26	1070	35	18	53	6.0
3	32,000	26 1/2	670	59	48	90	69
4	28,800	27 1/2	580	80	70	125	105
5	4,400	28 1/2	85	84	82	135	130
6	3,500	29 1/2	68	87	86	140	135
7	12,400	31	220	95	91	160	150
8	3,200	32 1/2	55	97	96	170	165
9	2,600	32	45	99	98	180	175
10	4,000	24 1/2	90	vapor	—	190	185
11	2,300	20	65	vapor	vapor	200	195
12	1,800	17 1/2	58	vapor	vapor	205	200

	Pressure (cm.Hg.ga.)	Saturation (Temp. °C.)
Entering 1st Pass	37.6	94.0
End 1st Pass	31.9	92.2
End 2nd Pass	23.0	89.2
End 3rd Pass	13.7	85.8
End 4th Pass	4.4	82.0

Run B - 9

Condensing steam temperature = 134°C.

Feed rate = 710 lbs./hr.

Total heat transfer = 160,000 B.t.u./hr. (by steam side)

= 153,000 B.t.u./hr. (by condenser)

Initial boiling point = 50% of J. 1

Final boiling point = 10% of J. 8

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit* velocity (F.P.S.)	Average velocity
1	65,800	48 1/2	755	17	—	25	4.1
2	42,400	41 1/2	565	59	37	85	46
3	11,600	42	152	67	63	100	92
4	18,200	43	230	85	76	130	115
5	4,700	44	60	90	87	140	135
6	3,800	45	57	93	91	150	145
7	10,600	46	123	99.6	96	160	155
8	4,000	42	53	vapor	vapor	165	160
9	4,500	34	76	vapor	vapor	175	170
10	3,200	27	66	vapor	vapor	190	180
11	2,100	20	57	vapor	vapor	195	190
12	1,800	13	75	vapor	vapor	210	200

	Pressure (cm.Hg.ga.)	Saturation (Temp. °C.)
Entering 1st Pass	36.0	93.5
End 1st Pass	29.0	91.6
End 2nd Pass	21.4	88.8
End 3rd Pass	12.7	85.4
End 4th Pass	4.0	81.8

* Entering velocity = 0.66 ft./sec.

Run B - 10

Condensing steam temperature = 138°C.

Feed rate = 689 lbs./hr.

Total heat transfer = 158,000 B.t.u./hr. (by steam side)

= 145,000 B.t.u./Hr.(by condenser)

Initial boiling point = 50% of J. 1

Final boiling point = 60% of J. 9

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit* velocity (F.P.S.)	Average velocity
1	64,800	58 1/2	610	25	—	36	4.8
2	24,200	47	285	46	35	67	49
3	13,800	47 1/2	160	57	51	83	75
4	24,100	48	280	74	66	110	95
5	6,000	49	67	80	77	120	115
6	4,900	50	57	85	83	135	125
7	12,400	51	133	95	90	150	140
8	5,500	52	58	98	97	165	155
9	4,800	47 1/2	55	vapor	—	170	165
10	4,800	37	71	vapor	vapor	180	175
11	3,200	26 1/2	68	vapor	vapor	190	185
12	3,700	19	107	vapor	vapor	200	195

	Pressure (cm.Hg.ga.)	Saturation (Temp. °C.)
Entering 1st Pass	30.6	91.8
End 1st Pass	25.6	90.2
End 2nd Pass	18.2	97.6
End 3rd Pass	11.9	85.0
End 4th Pass	3.9	81.8

* Entering velocity = 0.64 ft./sec.

Run B - 11

Condensing steam temperature = 154°C.

Feed rate = 818 lbs./hr.

Total heat transfer = 146,000 B.t.u./hr. (by steam side)

= 136,000 B.t.u./hr. (by condenser)

Initial boiling point = 60% of J. 1

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit * velocity (F.P.S.)	Average velocity
1	48,900	78	350	12	—	22	4.1
2	10,000	64	85	19	15	34	27
3	10,100	64	88	26	22	46	40
4	10,800	64 1/2	92	33	29	59	52
5	7,200	65	62	39	36	71	65
6	10,100	65 1/2	85	46	42	86	78
7	11,600	66	98	54	50	100	93
8	9,200	67	75	60	57	115	105
9	5,100	68	43	64	62	125	120
10	16,800	69	132	75	70	150	135
11	7,500	70	60	81	78	170	160
12	6,800	71 1/2	53	87	84	190	180

	Pressure (cm.Hg.ga.)	Saturation (Temp. °C.)
Entering 1st Pass	26.9	90.6
End 1st Pass	24.5	89.8
End 2nd Pass	20.0	88.2
End 3rd Pass	13.5	85.8
End 4th Pass	3.9	81.8

* Entering velocity = .76 ft./sec.

Run B - 12

Condensing steam temperature = 158°C.

Feed rate = 448 lbs./hr.

Total heat transfer = 93,900 B.t.u./hr. (by steam side)

= 80,400 B.t.u./hr. (by condenser)

Initial boiling point = 70% of J. 1

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit* velocity (F.P.S.)	Average velocity
1	23,800	87	152	9	—	11	2.2
2	7,900	74	59	19	14	22	16
3	7,400	74	56	27	23	31	26
4	6,600	74	49	35	31	40	35
5	6,100	74	45	42	39	47	43
6	6,700	74 1/2	49	50	46	57	52
7	7,200	75	52	59	54	68	62
8	6,800	75	50	67	63	78	73
9	4,200	75 1/2	31	72	69	85	81
10	11,000	76	80	85	78	100	92
11	5,100	76 1/2	35	92	89	110	105
12	5,000	77	35	98	95	120	115

	Pressure (cm.Hg.ga.)	Saturation (Temp.°C.)
Entering 1st Pass	10.2	84.4
End 1st Pass	9.1	84.0
End 2nd Pass	7.4	83.4
End 3rd Pass	4.6	82.2
End 4th Pass	0.9	80.6

* Entering velocity = .42 ft./sec.

Run B - 13

Condensing steam temperature = 156.5°C.

Feed rate = 289 lbs./hr.

Total heat transfer = 65,100 B.t.u./hr. (by steam side)

= 54,900 B.t.u./hr. (by condenser)

Initial boiling point = 70% of J. 1

Final boiling point = 70% of J. 12

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit* velocity (F.P.S.)	Average velocity
1	17,600	86 1/2	113	10	—	11	1.7
2	6,600	74 1/2	49	22	16	21	15
3	6,000	74 1/2	45	33	28	29	25
4	5,000	74 1/2	37	42	38	36	32
5	4,400	74 1/2	33	51	47	42	39
6	5,000	74 1/2	37	60	56	49	45
7	3,800	75	28	67	64	55	52
8	4,000	75	30	74	71	60	57
9	2,400	75 1/2	18	79	77	64	62
10	7,200	76	53	91	85	74	69
11	3,200	76 1/2	23	97	94	78	76
12	2,800	73	21	vapor	99	80	79

	Pressure (cm.Hg.ga.)	Saturation (Temp.°C.)
Entering 1st Pass	4.8	82.4
End 1st Pass	3.9	82.0
End 2nd Pass	3.0	81.4
End 3rd Pass	1.7	80.9
End 4th Pass	0.2	80.3

* Entering velocity = 0.27 ft./sec.

Run B - 14

Condensing steam temperature = 102°C.

Feed rate = 713 lbs./hr.

Total heat transfer = 124,000 B.t.u./hr. (by steam side)

= 126,000 B.t.u./hr. (by condenser)

Initial boiling point = 60% of J. 3

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	17,500	46	211	t =67°C.	—	0.66	0.66
2	8,800	27	181	t = 82°C.	—	0.66	0.66
3	5,500	14 1/2	210	2	—	3.8	1.6
4	6,900	13 1/2	280	7	5	12	6.8
5	7,600	14	300	13	10	22	16
6	8,700	14	340	20	17	33	27
7	11,900	14 1/2	460	28	24	47	39
8	15,500	15	575	39	34	65	55
9	20,300	15 1/2	730	54	47	91	77
10	25,800	17	840	75	64	130	110
11	9,600	18	295	82	78	145	135
12	2,400	19 1/2	68	85	84	160	150

	Pressure (cm.Hg.ga.)	Saturation (Temp. °C.)
Entering 1st Pass	21.3	88.8
End 1st Pass	20.4	88.2
End 2nd Pass	19.1	87.8
End 3rd Pass	13.7	85.8
End 4th Pass	3.3	81.8

Run B - 1 A

Condensing steam temperature = 103°C.

Feed rate = 704 lbs./hr.

Total heat transfer = 129,000 B.t.u./hr. (by steam side)

= 128,000 B.t.u./hr. (by condenser)

Initial boiling point = 10% of J. 3

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	16,500	41	220	t=73.1	—	0.65	0.65
2	9,700	21 1/2	250	t=88.1	—	0.65	0.65
3	8,800	13 1/2	360	7	3	11	2.7
4	8,500	13 1/2	350	12	8	19	14
5	11,000	14	440	21	16	33	25
6	10,750	14 1/2	410	30	25	48	40
7	13,800	15	510	41	34	67	57
8	18,500	16	640	55	47	92	79
9	21,800	18	680	71	63	120	105
10	19,800	19 1/2	570	84	79	150	135
11	4,600	20	130	90	88	160	155
12	1,900	20 1/2	50	91	91	165	160

	Pressure (cm.Hg.ga.)	Saturation (Temp.°C.)
Entering 1st Pass	23.3	89.5
End 1st Pass	22.4	89.1
End 2nd Pass	20.6	88.5
End 3rd Pass	12.5	85.5
End 4th Pass	2.4	81.4

Run B - 2 A

Condensing steam temperature = 103°C.

Feed rate = 434 lbs./hr.

Total heat transfer = 90,800 B.t.u./hr. (by steam side)

=90,900 B.t.u./hr. (by condenser)

Initial boiling point = 20% of J. 2

Final boiling point = 40% of J. 9

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	14,900	40	210	t=78.8	—	0.40	0.40
2	9,550	18	300	10	3	11	2.1
3	14,300	18	450	25	17	27	17
4	12,000	18	375	43	33	46	35
5	17,000	18 1/2	515	62	51	68	56
6	17,800	19	520	83	72	90	78
7	11,600	19 1/2	330	96	90	105	95
8	1,700	19 1/2	49	99.5	98	110	105
9	1,000	19 1/2	28	vapor	—	115	110
10	1,800	14 1/2	69	vapor	vapor	120	115
11	700	9 1/2	41	vapor	vapor	120	120
12	500	4	69	vapor	vapor	120	120

	Pressure (cm.Hg.ga.)	Saturation (Temp.°C.)
Entering 1st Pass	12.9	85.5
End 1st Pass	11.8	85.0
End 2nd Pass	8.4	83.8
End 3rd Pass	4.4	82.2
End 4th Pass	1.2	80.8

Run B - 3 A

Condensing steam temperature = 103.5°C.

Feed rate = 282 lbs./hr.

Total heat transfer = 60,500 B.t.u./hr. (by steam side)

= 60,800 B.t.u./hr. (by condenser)

Initial boiling point = 80% of J. 1

Final boiling point = 10% of J. 6

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit* velocity (F.P.S.)	Average velocity
1	14,600	36	225	5	—	2.4	0.79
2	13,600	20 1/2	370	32	18	23	7.4
3	17,900	20 1/2	485	64	47	47	33
4	14,600	20 1/2	400	90	77	66	56
5	4,700	20 1/2	127	99.8	95	74	70
6	1,100	19 1/2	31	vapor	vapor	78	76
7	700	14 1/2	27	vapor	vapor	78	78
8	300	11 1/2	15	vapor	vapor	80	79
9	300	9 1/2	18	vapor	vapor	80	80
10	600	7	48	vapor	vapor	80	80
11	300	4	42	vapor	vapor	81	80
12	100	2 1/2	22	vapor	vapor	81	81

	Pressure (cm.Hg.ga.)	Saturation (Temp. °C.)
Entering 1st Pass	5.9	82.8
End 1st Pass	4.4	82.2
End 2nd Pass	2.6	81.4
End 3rd Pass	1.4	81.0
End 4th Pass	0.2	80.4

* Entering velocity = 0.26 ft./sec.

Run B - 4 A

Condensing steam temperature = 103°C.

Feed rate = 674 lbs./hr.

Total heat transfer = 126,700 B.t.u./hr. (by steam side)

= 133,500 B.t.u./hr. (by condenser)

Initial boiling point = 80% of J. 2

Final boiling point = 90% of J. 12

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	15,200	38	220	t=76.2	—	0.62	0.62
2	9,400	18 1/2	280	1	—	2.1	1.15
3	9,600	14	380	8	5	12	5.0
4	9,400	14	370	14	11	23	17
5	12,550	14	500	24	19	36	29
6	11,500	14 1/2	440	33	29	51	43
7	15,150	15	560	45	39	72	61
8	20,100	16	690	62	54	97	84
9	21,900	17 1/2	680	79	71	135	115
10	19,800	19 1/2	570	95	87	175	155
11	4,950	21	130	98.5	97	175	175
12	2,000	21 1/2	51	vapor	99	180	175

	Pressure (cm.Hg.ga.)	Saturation (Temp. °C.)
Entering 1st Pass	22.3	89.0
End 1st Pass	21.4	88.7
End 2nd Pass	20.0	88.2
End 3rd Pass	12.1	85.5
End 4th Pass	2.7	81.4

Run B - 5 A

Condensing steam temperature = 103 1/2 °C.

Feed rate = 453 lbs./hr.

Total heat transfer = 92,300 B.t.u./hr. (by steam side)

= 92,700 B.t.u./hr. (by condenser)

Initial boiling point = 30% of J. 2

Final boiling point = 70% of J. 9

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	13,600	38 1/2	196	t=79.0	—	0.42	0.42
2	9,800	18 1/2	295	10	3	11.5	2.2
3	14,650	18 1/2	440	25	17	28	18
4	13,600	18 1/2	410	43	33	48	37
5	18,300	19	540	62	52	71	58
6	17,300	19 1/2	495	82	73	92	65
7	11,500	19 1/2	330	96	90	110	100
8	1,700	20	47	99	98	115	110
9	1,000	20 1/2	27	vapor	99	120	115
10	1,800	16	63	vapor	vapor	125	120
11	1,800	9	111	vapor	vapor	130	125
12	300	6	28	vapor	vapor	130	130

	Pressure (cm.Hg.ga.)	Saturation (Temp.°C.)
Entering 1st Pass	13.7	86.0
End 1st Pass	12.6	85.5
End 2nd Pass	9.0	84.1
End 3rd Pass	4.6	82.2
End 4th Pass	1.4	80.9

Run B-6 A

Condensing steam temperature = 104°C.

Feed rate = 720 lbs./hr.

Total heat transfer = 131,000 B.t.u./hr. (by steam side)

= 130,000 B.t.u./hr. (by condenser)

Initial boiling point = beginning of J. 3

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	16,900	42	220	t=73.5	—	0.67	0.67
2	10,600	21 1/2	270	0	—	0.67	0.67
3	9,150	14 1/2	350	7	3	12	2.8
4	10,400	14 1/2	400	14	11	23	17
5	9,600	15	360	22	18	36	29
6	12,300	15 1/2	440	32	27	52	43
7	17,000	16	590	45	38	74	62
8	20,700	17	670	59	52	100	86
9	19,000	18	590	74	67	130	115
10	17,000	19	500	85	81	155	140
11	3,600	20	100	90	89	165	160
12	2,300	21 1/2	59	92	91	175	170

	Pressure (cm.Hg.ga.)	Saturation (Temp. °C.)
Entering 1st Pass	22.6	89.2
End 1st Pass	21.6	88.8
End 2nd Pass	20.5	88.4
End 3rd Pass	11.8	85.2
End 4th Pass	3.5	81.7

Run B - 7 A

Condensing steam temperature = 104°C.

Feed rate = 1030 lbs./hr.

Total heat transfer = 146,000 B.t.u./hr. (by steam side)

= 144,000 B.t.u./hr. (by condenser)

Initial boiling point = 70% of J. 3

Jacket	q/A	Δt, °C.	U	% vapor leaving	% vapor average	Exit velocity (F.P.S.)	Average velocity
1	14,600	37 1/2	216	t=73.9	—	0.95	0.95
2	11,100	24	257	t=85.6	—	0.95	0.95
3	7,600	12 1/2	330	1	—	3.0	1.7
4	8,100	12 1/2	350	5	3	11	5.8
5	7,950	12 1/2	350	9	7	20	15
6	9,400	12 1/2	410	14	12	30	25
7	11,900	13	520	21	18	46	37
8	15,200	13 1/2	630	29	25	63	54
9	20,400	14 1/2	785	40	35	88	75
10	28,300	16	975	56	48	130	110
11	23,700	18	740	69	62	170	150
12	7,600	21	200	73	71	190	180

	Pressure (cm.Hg.g.a)	Saturation (Temp.°C.)
Entering 1st Pass	—	—
End 1st Pass	—	—
End 2nd Pass	29.7	91.6
End 3rd Pass	23.4	89.6
End 4th Pass	6.4	83.0

CALCULATED DATA

Section III. - Benzene - Oil Runs

Run B O - 1

Condensing steam temperature = 102°C.

Feed rate = 702 lbs./hr. (by material balance)

= 704 lbs./hr. (by orifice)

Total heat transfer = 104,800 B.t.u./hr. (by steam side)

107,000 B.t.u./hr. (by condenser)

Initial boiling point = 80% of J. 3

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	$\bar{p}_{out}$	X_{out}	$\bar{p}_{av}$	X_{av}
1	10,000	82	122	65	—	—	—	—
2	7,600	61	125	77	—	—	—	—
3	5,100	42	122	86	1.0	90.4	—	—
4	5,300	33	160	87.5	4.5	90.1	2.1	90.3
5	4,900	32	153	87.5	6.9	89.8	5.0	90.0
6	5,300	32	166	87.5	10.4	89.4	8.7	89.6
7	7,300	32	228	87.5	15.2	88.8	12.8	89.1
8	10,400	32	325	87.5	22.8	87.7	18.8	88.3
9	16,500	32	515	87.0	36.7	85.0	30.2	86.4
10	22,100	33	670	86.0	53.5	79.6	46.0	82.4
11	17,000	35	485	85.0	67.0	71.2	61.4	75.4
12	10,200	37	275	83.5	76.0	60.5	72.1	65.9

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	210	21.0	20.0	15.2	3.1

Run B O - 2

Condensing steam temperature = 103.5°C.

Feed rate = 775 lbs./hr. (by material balance)

= 684 lbs./hr. (by orifice)

Total heat transfer = 115,400 B.t.u./hr. (by steam side)

= 112,900 B.t.u./hr. (by condenser)

Initial boiling point = 60% of J. 3

Jacket	q/A	Δt, °F.	U	Exit Temp. (°C.)	P_{out}	X_{out}	P_{av}	X_{av}
1	11,300	78	145	67	—	—	—	—
2	6,900	59	117	78	—	—	—	—
3	5,700	41	139	87	1.5	93.4	—	—
4	5,100	32	160	88	4.4	93.1	3.0	93.3
5	5,100	31	165	88.5	7.2	93.0	5.8	93.1
6	5,900	31	190	88.5	9.7	92.8	9.7	92.8
7	8,300	31	268	88.3	16.7	92.2	14.5	92.4
8	11,200	31	360	88.0	24.4	91.4	20.8	91.8
9	18,700	32	585	87.5	36.9	89.7	31.6	90.5
10	24,000	33	728	86.5	53.5	86.0	46.7	87.8
11	17,100	36	475	85	66.3	80.7	60.9	83.4
12	8,700	39	224	83	73.5	75.5	70.3	78.1

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	29.2	29.1	28.5	19.0	2.9

Run B O - 5

Condensing steam temperature = 114°C.

Feed rate = 545 lbs./hr. (by material balance)

= 555 lbs./hr. (by orifice)

Total heat transfer = 82,400 B.t.u./hr. (by steam side)

= 83,600 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 11,100 B.t.u./hr.

Initial boiling point = 20% of J. 4

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	P_{out}	X_{out}	P_{av}	X_{av}
1	9,700	125	78	54	—	—	—	—
2	7,600	96	80	70	—	—	—	—
3	7,200	68	103	85	—	—	—	—
4	6,300	52	122	89	4.8	69.0	2.3	69.9
5	6,100	48	127	89	10.9	66.9	7.8	68.0
6	7,400	48	154	89	18.3	63.9	14.7	65.4
7	10,300	47	220	89.5	27.5	59.3	23.0	61.7
8	12,800	46	278	91	39.6	51.2	34.0	55.3
9	10,200	42	243	93	48.6	42.6	44.4	47.0
10	9,000	39	240	95	56.4	32.4	52.9	37.5
11	5,100	35	146	97.5	60.2	25.9	58.4	29.2
12	3,400	29	117	101	61.7	23.0	61.0	24.4

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	17.9	16.3	14.5	9.8	0.5

Run B O - 6

Condensing steam temperature = 132°C.

Feed rate = 498 lbs./hr. (by material balance)

= 439 lbs./hr. (by orifice)

Total heat transfer = 101,600 B.t.u./hr. (by steam side)

= 102,200 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 5,200 B.t.u./hr.

Initial boiling point = 60% of J. 1

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	P_{out}	X_{out}	P_{av}	X_{av}
1	34,700	106	329	94	12.9	86.8	—	—
2	41,000	66	620	95.5	55.3	74.3	40.7	80.6
3	18,200	66	276	99	73.3	56.9	66.5	65.6
4	10,400	52	192	107	81.7	37.1	78.3	47.1
5	4,100	37	110	115	83.7	29.6	82.8	33.4
6	2,400	26	92	120	84.8	24.9	84.1	27.3
7	2,300	17	135	124.5	86.0	18.7	85.3	21.9
8	1,200	11	110	127	86.5	14.4	86.3	16.1
9	800	7	110	129	86.7	13.8	86.6	14.2
10	700	4	180	130.5	87.2	10.5	87.0	11.5
11	200	3	70	131	87.2	10.5	87.2	10.5
12	100	2	50	131	87.2	10.5	87.2	10.5

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg. ga.)	35.1	29.5	21.9	12.0	1.5

Run B O - 7

Condensing Steam temperature = 128.5°C.

Feed rate = 984 lbs./hr. (by material balance)

= 954 lbs./hr. (by orifice)

Total heat transfer = 68,900 B.t.u./hr. (by steam side)

= 68,800 B.t.u./hr.(by condenser and cooler)

Heat transfer in cooler = 28,000 B.t.u./hr.

Initial boiling point = 60% of J. 7

Jacket	q/A	Δt, °F.	U	Exit temp. (°C.)	P_{out}	X_{out}	P_{av}	X_{av}
1	7,200	120	60	66	—	—	—	—
2	7,100	107	66	72.5	—	—	—	—
3	6,100	95	65	79	—	—	—	—
4	5,700	84	68	84.5	—	—	—	—
5	4,800	75	64	89.5	—	—	—	—
6	4,500	67	68	93.5	—	—	—	—
7	4,800	61	78	96	0.2	33.4	—	—
8	5,500	59	93	96	3.9	30.8	2.1	32.1
9	6,300	57	110	98	7.0	28.5	5.5	29.6
10	10,000	53	189	100	11.7	24.7	9.4	26.6
11	8,400	50	168	102	15.4	21.4	13.5	23.1
12	7,800	45	176	105	18.4	18.5	16.9	20.0

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	10.8	9.9	9.2	7.1	0.6

Run B O - 8

Condensing steam temperature = 140.5°C.

Feed rate = 1,008 lbs./hr. (by material balance)

= 1,015 lbs./hr. (by orifice)

Total heat transfer = 47,000 B.t.u./hr. (by steam side)

= 44,200 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 34,300 B.t.u./hr.

Initial boiling point = 60% of J. 8

Jacket	q/A	Δ t, °F.	U	Exit temp. °C.	P_{out}	X_{out}	P_{av}	X_{av}
1	5,200	124	42	74	—	—	—	—
2	4,900	115	43	79	—	—	—	—
3	4,800	107	45	83.5	—	—	—	—
4	4,100	99	41	88	—	—	—	—
5	3,400	92	37	91.5	—	—	—	—
6	3,700	85	44	95	—	—	—	—
7	3,900	78	51	99.5	—	—	—	—
8	3,100	71	44	103	0.6	20.5	—	—
9	4,600	67	69	104	2.1	19.3	1.4	19.9
10	5,700	65	88	105.5	4.4	17.4	3.2	18.4
11	3,300	61	54	107.5	4.7	17.1	4.5	17.3
12	2,500	57	44	110.5	4.8	17.0	4.8	17.0

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	5.6	5.2	4.7	4.2	1.5

Run B 0 - 11

Condensing steam temperature = 158°C.

Feed rate = 1,078 lbs./hr. (by material balance)

= 1,035 lbs./hr. (by orifice)

Total heat transfer = 58,800 B.t.u./hr. (by steam side)

= 59,700 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 41,600 B.t.u./hr.

Initial boiling point = 60% of J. 7

Jacket	q/A	Δ t, °F.	U	Exit temp. °C.	P_{out}	X_{out}	P_{av}	X_{av}
1	8,800	132	67	86	—	—	—	—
2	6,900	121	57	92	—	—	—	—
3	6,700	111	60	97	—	—	—	—
4	5,700	102	56	102	—	—	—	—
5	5,000	93	54	106.5	—	—	—	—
6	5,100	86	60	110.5	—	—	—	—
7	4,300	78	56	115	0.9	15.2	—	—
8	4,000	71	56	118	1.3	14.9	1.2	15.0
9	1,500	68	22	119	1.5	14.7	1.4	14.8
10	9,500	65	146	121	6.2	10.5	3.8	12.7
11	5,400	61	88	123	7.5	9.2	6.8	9.9
12	5,000	57	88	126	7.7	9.0	7.6	9.1

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	5.2	5.2	4.4	3.6	0.6

Run B O - 12

Condensing steam temperature = 176°C.

Feed rate = 1,065 lbs./hr. (by material balance)

= 974 lbs./hr. (by orifice)

Total heat transfer = 64,800 B.t.u./hr. (by steam side)

= 65,900 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 48,600 B.t.u./hr.

Initial boiling point = 50% of J. 6

Jacket	q/A	Δ t, °F.	U	Exit temp. °C.	P_{out}	X_{out}	P_{av}	X_{av}
1	12,000	147	82	99	—	—	—	—
2	9,200	129	71	107	—	—	—	—
3	7,900	115	69	114	—	—	—	—
4	6,300	105	60	119	—	—	—	—
5	5,500	96	57	124	—	—	—	—
6	(4,700)	89	(53)	126	1.8	10.4	—	—
7	(4,300)	86	(50)	127.5	3.9	8.4	2.9	9.4
8	4,200	83	51	129.5	5.3	7.1	4.5	7.9
9	2,300	80	29	131	5.6	6.8	5.4	7.0
10	9,600	70	137	139	5.7	6.7	5.6	6.8
11	4,800	60	80	143	5.7	6.7	5.7	6.7
12	3,800	54 1/2	70	145.5	6.4	6.0	6.0	6.4

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	6.5	5.5	4.9	3.3	0.4

Run B O - 13

Condensing steam temperature = 139°C.

Feed rate = 1,080 lbs./hr. (by material balance)

= 1,060 lbs./hr. (by orifice)

Total heat transfer = 61,500 B.t.u./hr. (by steam side)

= 60,600 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 27,200 B.t.u./hr.

Initial boiling point = 20% of J. 9

Jacket	q/A	Δ t, °F.	U	Exit temp. °C.	$\bar{P}_{out}$	X_{out}	$\bar{P}_{av}$	X_{av}
1	4,900	122	40	74	—	—	—	—
2	5,300	113	47	79	—	—	—	—
3	5,400	104	52	84	—	—	—	—
4	5,000	95	53	89	—	—	—	—
5	4,600	87	53	93	—	—	—	—
6	4,200	80	53	96.5	—	—	—	—
7	4,100	74	55	100	—	—	—	—
8	4,700	68	69	102.5	—	—	—	—
9	5,200	65	80	103.5	2.1	24.1	0.8	25.1
10	11,600	64	181	104	8.4	18.9	4.6	22.1
11	7,400	61	121	106	11.3	16.2	9.9	17.5
12	6,500	55	118	110.5	12.2	15.4	11.8	15.8

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	11.8	11.0	8.9	8.7	0.8

Run B O - 14

Condensing steam temperature = 149°C.

Feed rate = 1,093 lbs./hr. (by material balance)

= 1033 lbs./hr. (by orifice)

Total heat transfer = 86,100 B.t.u./hr. (by steam side)

= 87,500 B.t.u./hr. (by condenser and cooler

Heat transfer in cooler = 31,600 B.t.u./hr.

Initial boiling point = 60% of J. 6

Jacket	q/A	Δ t, °F.	U	Exit temp. °C.	p_{out}	X_{out}	p_{av}	X_{av}
1	6,000	138	43	76	—	—	—	—
2	6,500	125	52	82.5	—	—	—	—
3	6,600	114	57	88.5	—	—	—	—
4	6,200	104	60	94	—	—	—	—
5	6,000	94	64	99	—	—	—	—
6	6,400	88	73	101	1.6	30.2	—	—
7	8,600	84	102	103.5	4.9	27.8	3.2	29.0
8	10,300	79	130	107	8.4	25,0	6.7	26.4
9	9,700	73	133	110.5	11.5	22.4	10.0	23.7
10	16,500	66	250	114.5	19.8	14.40	15.8	18.4
11	9,200	58	158	119	22.2	11.7	21.0	13.1
12	6,100	50	122	123,5	23.0	10.7	22.6	11.3

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	16.5	14.9	41.9	10.6	0

Run B O - 15

Condensing steam temperature = 162°C.

Feed rate = 970 lbs./hr. (by material balance)

= 1040 lbs./hr. (by orifice)

Total heat transfer = 101,100 B.t.u./hr. (by steam side)

= 95,600 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 36,600 B.t.u./hr.

Initial boiling point = 90% of J. 4

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	p_{out}	x_{out}	p_{av}	x_{av}
1	7,000	143	49	81.5	—	—	—	—
2	7,300	136	54	89	—	—	—	—
3	7,400	123	60	96	—	—	—	—
4	7,700	110	70	103.5	0.1	29.9	—	—
5	8,700	101	86	105	4.6	26.5	2.4	28.3
6	10,900	96	113	109	8.9	23.2	6.8	24.9
7	14,000	89	157	113.5	14.3	18.3	11.6	20.8
8	12,800	79	162	119.5	18.3	14.3	16.3	16.4
9	8,600	68	126	126	19.6	12.9	18.8	13.8
10	13,000	56	232	133	23.1	9.0	21.4	11.0
11	6,400	45	142	138.5	23.9	8.0	23.5	8.5
12	4,900	36	136	142.5	24.3	7.5	24.1	7.8

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	22.0	21.0	18.7	11.6	1.3

Run B O - 16

Condensing steam temperature = 161°C.

Feed rate = 1,011 lbs./hr. (by material balance)

= 1,029 lbs./hr. (by orifice)

Total heat transfer = 123,000 B.t.u./hr. (by steam side)

= 125,000 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 33,900 B.t.u./hr.

Initial boiling point = 10% of J. 4

Jacket	q/A	Δt,°F.	U	Exit temp. °C.	P_{out}	X_{out}	P_{av}	X_{av}
1	10,100	148	68	82	—	—	—	—
2	9,800	131	75	92	—	—	—	—
3	10,800	116	93	100	—	—	—	—
4	18,400	105	175	103	10.0	34.4	4.5	38.2
5	17,900	100	179	106	18.8	27.3	13.5	31.8
6	14,600	93	157	111	24.0	22.3	21.6	24.8
7	18,300	81	226	118	30.5	15.1	27.4	18.7
8	12,500	68	184	126	32.5	12.6	31.4	13.9
9	8,100	55	147	133.5	32.9	12.1	32.7	12.4
10	12,400	43	288	138.5	36.7	6.8	36.4	7.2
11	4,900	35	140	142.5	37.1	6.2	36.9	6.5
12	3,700	29	128	145.5	37.2	6.0	37.2	6.1

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	37.7	36.7	31.4	18.9	3.0

Run B O - 17

Condensing steam temperature = 149.5°C.

Feed rate = 1,020 lbs./hr. (by material balance)

= 1,032 lbs./hr. (by orifice)

Total heat transfer = 123,000 B.t.u./hr. (by steam side)

= 124,000 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 29,400 B.t.u./hr.

Initial boiling point = 60% of J. 4

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	p_{out}	X_{out}	p_{av}	X_{av}
1	10,200	142	72	76	—	—	—	—
2	9,300	122	76	85	—	—	—	—
3	9,300	105	89	95	—	—	—	—
4	11,200	93	120	101	2.5	44.6	—	—
5	18,000	86	210	102.5	12.1	37.4	7.4	40.6
6	13,400	82 1/2	162	103	16.6	34.1	14.3	35.8
7	22,100	75 1/2	293	110	27.8	23.9	22.5	29.0
8	15,000	66 1/2	226	115	33.0	18.0	30.4	21.0
9	9,300	57	163	121	34.6	16.0	33.7	17.1
10	12,700	45 1/2	280	127.5	38.0	11.2	36.9	13.1
11	5,800	36	161	131.5	38.8	10.2	38.4	10.7
12	4,200	31	135	133	40.2	8.0	39.5	9.1

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	36.5	35.7	32.1	20.5	3.7

Run B O - 18

Condensing steam temperature = 176°C.

Feed rate = 1,025 lbs./hr. (by material balance)

= 1,073 lbs./hr. (by orifice)

Total heat transfer = 75,000 B.t.u./hr. (by steam side)

= 74,700 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 48,500 B.t.u./hr.

Initial boiling point = beginning of J. 6

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	p_{out}	X_{out}	p_{av}	X_{av}
1	12,400	146	85	99	—	—	—	—
2	9,000	125	72	109	—	—	—	—
3	8,400	110	75	116	—	—	—	—
4	7,100	100	72	121.5	—	—	—	—
5	6,100	92	66	125	—	—	—	—
6	6,300	85	74	128.5	2.7	10.6	1.3	11.9
7	6,700	79	85	132	3.9	9.5	3.2	10.1
8	5,500	71 1/2	77	136.5	4.1	9.3	4.0	9.4
9	5,700	63 1/2	90	141	4.3	9.1	4.2	9.2
10	8,800	56	157	145	6.1	7.3	5.2	8.2
11	5,100	50	102	147	7.0	6.5	6.6	6.9
12	3,900	45 1/2	86	150.5	7.5	6.0	7.2	6.3

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	10.2	8.3	7.0	3.8	0.5

Run B O - 19

Condensing steam temperature = 176°C.

Feed rate = 1,050 lbs./hr. (by material balance)

= 1,060 lbs./hr. (by orifice)

Total heat transfer = 113,200 B.t.u./hr. (by steam side)

= 117,300 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 44,300 B.t.u./hr.

Initial boiling point = 80% of J. 3

Jacket	q/A	Δt, °F	U	Exit temp. °C	p_{out}	x_{out}	p_{av}	x_{av}
1	9,900	162	61	89	—	—	—	—
2	10,700	145	74	98	—	—	—	—
3	11,900	129	92	107	1.3	28.1	—	—
4	18,600	115	162	113	8.8	22.2	5.1	25.2
5	15,900	104	153	119	14.3	17.2	11.6	19.7
6	11,600	93	125	125.5	16.6	14.9	15.5	16.0
7	16,000	81 1/2	197	132	21.5	9.6	19.0	12.3
8	10,200	68	150	140	22.5	8.4	22.0	8.9
9	7,300	56	130	146	22.6	8.2	22.5	8.4
10	11,900	45	265	152	25.5	4.7	23.6	7.1
11	5,900	36	164	156.5	25.8	4.3	25.6	4.5
12	3,300	29	114	159	26.0	4.0	25.9	4.2

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	30.6	28.9	23.7	14.2	2.2

Run B 0 - 20

Condensing steam temperature = 176 °C.

Feed rate = 1,050 lbs./hr. (by material balance)

= 1,047 lbs./hr. (by orifice)

Total heat transfer = 143,000 B.t.u./hr. (by steam side)

= 150,300 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 39,600 B.t.u./hr.

Initial boiling point = 30% of J. 3

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	p_{out}	X_{out}	p_{av}	X_{av}
1	14,200	173	82	85.5	—	—	—	—
2	17,600	146	120	101	—	—	—	—
3	24,600	128	192	108	10.9	36.0	—	—
4	32,200	114	282	114	26.2	22.7	19.3	29.4
5	21,400	102 1/2	209	121	34.3	13.2	30.5	18.0
6	13,900	87	160	131	35.6	11.6	34.8	12.5
7	15,300	68 1/2	224	142	37.0	9.5	36.2	10.6
8	8,600	53	162	148	37.8	8.2	37.4	8.9
9	5,900	43	137	153	38.2	7.8	38.0	8.1
10	10,200	35	292	157.5	40.4	4.4	39.3	6.1
11	4,300	27	159	161	40.5	4.2	40.4	4.3
12	2,500	22 1/2	111	163	40.6	4.0	40.5	4.1

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.	49.2	4.64	36.6	22.1	4.0

Run B O - 21

Condensing steam temperature = 176°C.

Feed rate = 990 lbs./hr. (by material balance)

= 1,065 lbs./hr. (by orifice)

Total heat transfer = 170,000 B.t.u./hr. (by steam side)

= 165,800 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 33,000 B.t.u/Hr.

Initial boiling point = 60% of J. 2

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	p_{out}	x_{out}	p_{av}	x_{av}
1	18,200	170	107	92	—	—	—	—
2	33,000	135	245	106	9.4	52.0	—	—
3	27,200	119	228	111	23.6	43.1	17.0	47.6
4	37,600	106	354	120	40.8	26.4	33.4	34.7
5	22,000	89	248	129	48.2	16.0	44.9	21.1
6	11,200	74	151	138	49.5	13.9	48.9	15.3
7	13,500	54	250	150	49.7	13.6	49.4	14.1
8	6,500	40	163	155	50.3	12.7	50.0	13.0
9	4,600	31 1/2	146	159	50.5	12.3	50.4	12.3
10	9,000	26	346	161.5	54.4	4.9	52.5	8.5
11	3,100	21	148	164	54.5	4.6	54.4	4.6
12	1,600	18	89	165	54.7	4.0	54.6	4.1

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	66.9	62.4	49.3	30.5	6.5

Run B O - 22

Condensing steam temperature = 139°C.

Feed rate = 1,015 lbs./hr. (by material balance)

= 1,015 lbs./hr. (by orifice)

Total heat transfer = 111,400 B.t.u./hr. (by steam side)

= 112,100 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 24,800 B.t.u./hr.

Initial boiling point = beginning of J. 6

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	p_{out}	X_{out}	p_{av}	X_{av}
1	8,000	138	58	68	—	—	—	—
2	8,300	121	69	76	—	—	—	—
3	7,900	105	75	85	—	—	—	—
4	7,700	91	85	92	—	—	—	—
5	7,600	79	96	98	0.2	45.9	—	—
6	8,600	74	116	98	4.9	43.2	2.7	44.5
7	13,400	72 1/2	185	99.5	11.8	38.8	8.5	41.0
8	16,500	69	240	102	20.2	36.1	15.6	36.0
9	13,900	64	218	105	25.3	27.7	22.3	30.5
10	18,900	57 1/2	330	109	33.5	18.8	29.6	23.3
11	10,700	50	214	113.5	36.8	14.6	35.2	16.6
12	6,100	41 1/2	147	118.5	37.2	14.0	37.0	14.3

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	27.9	27.2	26.0	18.7	3.5

Run B O - 23

Condensing steam temperature = 140°C.

Feed rate = 996 lbs./hr. (by material balance)

= 1,008 lbs./hr.(by orifice)

Total heat transfer = 145,200 B.t.u./hr. (by steam side)

= 145,100 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 20,100 B.t.u./hr.

Initial boiling point = 20% of J. 5

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	p_{out}	x_{out}	p_{av}	x_{av}
1	11,100	146	77	66	—	—	—	—
2	10,700	122	88	79	—	—	—	—
3	10,200	101	101	89	—	—	—	—
4	10,200	85	121	97	—	—	—	—
5	12,900	76	170	98.5	6.4	59.4	2.4	61.0
6	16,700	75	223	98.5	16.7	54.4	11.6	57.0
7	30,600	72 1/2	422	101	33.2	43.1	25.8	48.8
8	21,700	66 1/2	326	105	43.7	32.5	38.3	38.4
9	12,200	59 1/2	205	109	47.0	28.2	45.3	30.5
10	16,600	51	326	114	53.8	17.7	50.6	22.9
11	7,700	43	179	118	55.8	14.0	54.9	15.9
12	4,900	36 1/2	134	121.5	56.3	13.0	56.0	13.6

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	(44.3)	(43.2)	38.9	27.0	5.8

Run B O - 24

Condensing steam temperature = 160°C.

Feed rate = 884 lbs./hr. (by material balance)

= 995 lbs./hr. (by orifice)

Total heat transfer = 169,000 B.t.u./hr. (by steam side)

= 169,000 B.t.u./hr. (by condenser and cooler)

Heat transfer in cooler = 15,000 B.t.u./hr.

Initial boiling point = 50% of J. 2

Jacket	q/A	Δt, °F.	U	Exit temp. °C.	p_{out}	X_{out}	p_{av}	X_{av}
1	21,300	155	138	87	—	—	—	—
2	37,200	112	328	104	14.3	72.0	—	—
3	39,300	95 1/2	411	107	41.2	59.2	30.4	65.6
4	37,400	88	425	112	62.1	36.7	53.8	48.1
5	16,700	76 1/2	218	120	67.5	26.0	65.0	31.3
6	9,100	62	147	128	68.8	23.1	68.1	24.5
7	12,100	47	258	137	71.5	16.0	70.1	19.6
8	5,200	35	149	141	72.0	14.5	71.7	15.3
9	3,900	28	140	145	72.5	13.0	72.1	13.8
10	5,300	21	252	149	73.1	10.7	72.8	11.7
11	2,600	15	173	151	73.5	9.5	73.3	10.1
12	1,500	13	115	151.5	74.2	7.0	73.9	8.1

	Entering	Leaving 1st Pass	Leaving 2nd Pass	Leaving 3rd Pass	Leaving 4th Pass
Pressure (cm.Hg.ga.)	(83.9)	(75.4)	63.7	36.0	7.0

C A L C U L A T E D   D A T A

Section IV. Miscellaneous

Calculated Data of Slade [12]

This data was used in the construction of Figure 30a. The apparatus consisted of a 31-inch length of 0.200-inch i.d. horizontal stainless steel tube, heated by electrical current passing directly through the walls. Thermocouples were located at intervals along the top and bottom of the tube.

Run 1, Series 2, plotted on p. 13c of Slade thesis, data given on p. 35d of Slade thesis. Heat flux = 174,000 B.t.u./(hr.)(Sq.ft.)

Feed rate = 0.55 lbs./min. Entering velocity = 0.675 ft./sec.

Fluid pressure at inlet = 29.7 lbs./sq.in.abs.

Fluid pressure at outlet = 18.7 lbs./sq.in.abs.

Total vaporization = 60.8% of the feed, by weight.

p_{av}	h_{top}	h_{bottom}	$(\Delta t)_{top}$, °F	$(\Delta t)_{bottom}$, °F.
0.1	4380	4460	39.7	39.0
8.2	4070	3800	42.8	45.8
16.3	3800	3950	45.8	44.0
24.3	3350	3950	52.0	44.0
32.5	3380	3390	51.4	51.3
40.6	2460	2700	70.7	64.4
48.7	1880	—	92.6	—
56.8	1300	1270	134.3	137.3

Run 2, Series 2, plotted p. 13d and data given p. 35e of Slade thesis.

Heat flux = 101,500 B.t.u./(hr.)(sq.ft.). Feed rate = 0.27 lbs./min.

Initial velocity = 0.331 ft./sec.

Fluid pressure at inlet = 20.7 lbs./sq.in.abs.

Fluid pressure at outlet = 14.7 lbs./sq.in.abs.

Total vaporization = 75.6% of feed, by weight

p_{av}	h_{top}	h_{bottom}	$(\Delta t)_{top}$, °F	$(\Delta t)_{bottom}$, °F
5.8	3420	3310	29.7	30.7
15.2	3130	3200	32.4	31.7
25.0	3900	3130	35.0	32.4
33.8	2120	3150	47.9	32.2
43.1	2670	2410	38.1	42.1
52.4	1970	1030	51.6	99.0

Table V

Calculated Data for Figure 33

Run	Total q	Sensible q	Net q	Heating Jackets	Boiling Jackets
W 1	273,000	28,000	245,000	1.7	10.3
W 2	244,000	34,000	210,000	2.1	9.9
W 3	227,000	39,000	188,000	2.9	9.1
W 4	417,000	34,000	383,000	1.0	11.0
W 5	423,000	51,000	372,000	2.0	10.0
W 6	245,000	36,000	209,000	2.1	9.9
W 7	426,000	68,000	358,000	2.7	9.3
W 8	200,000	29,000	181,000	1.8	7.2
W 9	376,000	52,000	324,000	1.8	7.2
W 10	308,000	39,000	269,000	1.1	4.9
W 11	465,000	42,000	423,000	0.7	5.3
W 12	358,000	30,000	328,000	0.2	5.8
W 13	552,000	53,000	499,000	0.5	5.5
W 15	616,000	44,000	552,000	0.4	5.6

Run	Initial Δt, °C.	Final Δt, °C.	Average Δt, °C.	U_{av}	Final p
W 1	8	13 1/2	10 1/2	1430	69
W 2	8	13 1/2	11	1210	33
W 3	8	14 1/2	11	1210	20
W 4	15	25	20	1110	95 (rejected)
W 5	11 1/2	23 1/2	17 1/2	1330	41
W 6	8	13 1/2	11	1210	33
W 7	9	22	15 1/2	1580	39
W 8	9 1/2	13 1/2	11 1/2	1400	27
W 9	14	23	18 1/2	1530	32
W 10	19	24 1/2	22	1590	31
W 11	24	33 1/2	29	1750	67
W 12	55	61	58	610	99 (rejected)
W 13	39	50	44 1/2	1290	77 (rejected)
W 15	38	50	44	1420	98 (rejected)

Data of Oliver(10)

Heat transfer to boiling water inside of a vertical steam-heated nickel pipe, 0.495-in. I.D., 20.7-in. long. (cf. Figure 33a).

Run	Initial velocity (F.P.S.)	Overall Coefficient (U)	Overall Δt, (°C.)
D 1	2.8	1230	32.5
D 10	2.8	1380	31.5
E 1	2.8	1860	39.2
E 11	2.8	1970	39.2
F 1	2.8	2260	45.2
G 1	2.8	2280	60.4
I 4	2.8	2430	60.9
E 2	3.2	1830	38.9
C 1	3.6	1480	40.3
C 2	3.6	1750	55.3
D 2	3.6	1400	32.3
D 8	3.6	1580	31.4
E 10	3.6	1870	38.9
F 2	3.6	2170	44.8
F 9	3.6	2190	44.9
G 2	3.6	2140	59.8
E 3	5.7	1740	38.4
D 3	6.2	1360	32.2
D 3 A	6.2	1330	31.7
D 9	6.2	1470	31.0
E 9	6.2	1750	38.2
F 3	6.2	1970	44.4
F 8	6.2	1940	44.4
G 3	6.2	2020	59.1
I 2	6.2	1670	38.7
I 3	6.2	1810	45.3
E 3 A	6.4	1600	38.3
I 1	6.6	1340	31.1
E 4	7.3	1620	38.0
D 4	8.0	1500	31.7
F 4	8.0	1960	44.0
G 4	8.0	1990	58.8
E 8	8.4	1850	37.8
E 5	9.4	1890	37.4
D 5	10.0	1760	29.9
F 5	10.0	2040	43.4
F 10	10.0	2120	43.3
G 5	10.0	2190	58.1
D 6	11.7	1930	29.7
F 6	11.7	2140	43.1
G 6	11.7	2220	58.0
E 6	11.7	2150	37.0
E 7	13.6	2380	36.6
D 7	14.5	2180	29.4
F 7	14.5	2350	42.8
G 7	14.5	2320	57.7
G 8	14.5	2540	57.4
I 5	15.4	2390	58.1

Table VI

Calculated Data for Figure 41

Run	$\frac{q_{total}}{1000}$	$\frac{q_{sensible}}{1000}$	$\frac{q_{net}}{1000}$	Heating Jackets	Boiling Jackets	$\frac{q/A}{1000}$	Final Δt, °C.	U	% Benzene in Product
BO-1	107.0	19.1	87.9	2.8	9.2	10.9	18.5	327	60.5
BO-2	112.9	19.0	93.9	2.6	9.4	11.4	20.5	309	75.5
BO-5	83.6	22.7	60.9	3.2	8.8	7.9	13.0	338	23.0
BO-6	102.2	18.4	83.8	0.6	11.4	8.4	1.0	4660	10.5
BO-7	68.8	33.7	35.1	6.6	5.4	7.4	23.5	175	18.5
BO-8	44.2	28.2	16.0	7.6	4.4	4.1	30.0	76	17.0
BO-11	59.7	36.0	23.7	6.6	5.4	5.0	32.0	87	9.0
BO-12	65.9	38.1	27.8	5.5	6.5	4.9	30.5	89	6.0
BO-13	60.6	32.5	28.1	8.2	3.8	8.4	28.5	164	15.4
BO-14	87.5	30.9	56.6	5.6	6.4	10.1	25.5	220	10.7
BO-15	95.6	25.3	70.3	3.9	8.1	9.9	19.5	282	7.5
BO-16	125.0	28.6	96.4	3.1	8.9	12.3	15.5	430	6.0
BO-17	123.9	31.3	92.6	3.6	8.4	12.6	16.5	424	8.0
BO-18	74.7	37.9	36.8	5.0	7.0	6.0	25.5	131	6.0
BO-19	117.3	26.6	90.7	2.8	9.2	11.2	17.0	365	4.0
BO-20	150.3	34.5	115.8	2.3	9.7	13.6	13.0	582	4.0
BO-21	165.8	34.0	131.8	1.6	10.4	14.4	11.0	725	4.0
BO-22	112.1	34.9	77.2	5.0	7.0	12.5	20.5	339	14.0
BO-23	145.1	39.5	105.6	4.2	7.8	15.4	18.5	462	13.0
BO-24	169.0	35.2	133.8	1.5	10.5	14.5	8.5	947	7.0

Data of Bringardner[4] were taken in a natural convection still, boiling occurring outside of a single steam-heated nickel-plated copper tube (0.54 inch O.D.). "Oil B", as used by Bringardner, is the same oil used in the semi-commercial apparatus.

The following data were interpolated from the results obtained by Bringardner and were used in constructing the broken lines in Figure 41.

% benzene in still	h, at Δt = 30°C.	h at Δt = 50°C.
100	1520	720
92	1060	1000
70	420	520
50	270	300
30	140	185
6	90	

Table VII

Calculated data for Figure 43

Run	1st Pass U_{av}	1st Pass ΔP	1st Pass p_2	2nd Pass U_{av}	2nd Pass ΔP	2nd Pass p_2	3rd Pass U_{av}	3rd Pass ΔP	3rd Pass p_2	4th Pass U_{av}	4th Pass ΔP	4th Pass p_2
W 1			6	1290	1.2	20	1930	4.7	43	1680	10.4	69
W 2			2	1130	1.0	9	1520	4.1	18	1770	12.9	33
W 3			0.1	980	1.6	4	1360	3.6	10	1710	14.7	20
W 4	1170*	1.4	17	1390	4.5	43			75			
W 5			3	1360	2.8	11	1690	11.3	23	1870	29.7	41
W 6			2	1040	1.0	8	1520	3.1	18	1900	13.1	33
W 7			1	1520	2.9	8	1940	15.5	19	2150	30.2	39
W 8				722*	0.6	3	1320	2.1	12	1680	9.8	27
W 9				892*	0.9	3	1660	6.3	14	1990	24.6	32
W 10							1270*	3.5	10	1760	16.3	31
W 11							1520*	5.2	24	1990	28.2	67
W 13							1540*	9.0	38			77
W 15							1670*	12.4	48			98

NOTE: Data rejected if mixture leaving pass $\geqq$ 74% vaporized.

Date rejected if $\Delta P \geqq$ 5 cm. Hg.

* Includes warming data.

Table VIII

Calculated data for Figure 44

Run	1st Pass			2nd Pass			3rd Pass			4th Pass		
	U_{av}	ΔP	p_2	U_{av}	ΔP	p_2	U_{av}	ΔP	p_2	U_{av}	ΔP	p_2
B 1	480	1.5	21	700	5.6	67			93			
B 2	840	4.8	48			75						
B 3	111	2.8	9	58	2.4	24	102	5.9	54			82
B 4	235	0.6	0	340	1.5	11	550	6.3	33	685	12.4	66
B 5	260	1.1	8	340	1.3	28	610	5.9	71			96
B 6	97	1.1	8	57	2.8	21	75	5.4	40	98	10.8	66
B 7	250	0.9	5	360	1.7	26	635	6.7	64			91
B 8	710	5.7	59			87						
B 9	490	6.1	67			93						
B 10	350	5.0	57			85						
B 11	174	2.4	26	80	4.5	46	72	6.5	64			87
B 12	89	1.1	27	48	1.7	50	44	2.8	72			98
B 13	69	0.9	33	36	0.9	60			79			
B 14	201	0.9	2	310	1.3	20	590	5.4	54			85
B 1 A	230	0.9	7	400	1.8	30	610	8.1	71			91
B 2 A	320	1.1	25			83						
B 3 A	360	1.5	64			vap.						
B 4 A	290	0.9	8	420	1.4	33			79			
B 5 A	310	1.1	25			82						
B 6 A	280	1.0	7	400	1.1	32			74			
B 7 A						14	645	6.3	40	640	17.0	73

NOTE: Data rejected if mixture leaving pass is $\geq$ 74% vaporized.

Table IX

Calculated data for Figure 45

	1st Pass		2nd Pass		3rd Pass		4th Pass	
Run	U_{av}	ΔP	U_{av}	ΔP	U_{av}	ΔP	U_{av}	ΔP
BO-7	64	0.9	67	0.7	94	2.1	178	6.5
BO-8	43	0.4	41	0.5	55	0.5	62	2.7
BO-11		0	57	0.8	45	0.8	107	3.0
BO-12	74	1.0	57	0.6	43	1.6	96	2.9
BO-13	46	0.8	53	2.1		0.2	140	7.9
BO-14	51	0.6	66	1.0	122	4.3	177	10.6
BO-15	54	1.0	90	2.3	148	7.1	170	10.3
BO-16	79	1.0	170	5.3	186	12.5	185	15.9
BO-17	79	0.8	164	3.6	227	11.6	192	16.8
BO-18	77	1.9	71	1.3	84	3.2	115	3.3
BO-19	76	1.7	147	5.2	159	9.5	181	12.0
BO-20	131	2.8	217	9.8	174	14.5	187	18.1
BO-21	193	4.5	251	13.1	186	18.8	194	24.0
BO-22	67	0.7	99	1.2	214	7.3	230	15.2
BO-23	89	1.1	171	4.3	318	11.9	213	21.2
BO-24					182	27.7	180	29.0

Table X

Calculated Data for Figure 46

(cf. Page 173)

Run	Pass	P_{av}	ΔP	p_{av}	Δp	T, °C.	A/1000	B/1000	f/1000	p_{in}
W 1	2	96	1.2	13	14	107	4.55	3.67	0.88	6
	3	93	4.7	30	23	106	7.51	2.62	4.89	20
	4	86	10.4	56	26	103	8.30	1.58	6.72	43
W 2	2	100	1.0	6	7	108	2.49	3.98	(neg.)	2
	3	97	4.1	13	9	107	4.58	2.36	2.22	9
	4	90	12.9	24	15	105	7.27	2.13	5.14	18
W 3	2	103	1.6	2	4	109	5.44	6.83	(neg.)	0
	3	100	3.6	7	6	108	3.40	2.92	0.48	4
	4	94	14.7	14	10	106	6.57	2.44	4.13	10
W 4	2	110	4.5	30	26	110	6.46	3.07	3.39	17
	3	103	11.0	58	32	108	7.69	1.88	5.81	43
	4	89	15.6	90	20	104	6.14	0.76	5.38	75
W 5	2	135	2.8	7	8	117	3.87	3.90	(neg.)	3
	3	129	11.3	17	12	115	6.19	2.41	3.78	11
	4	110	29.7	32	18	110	7.45	1.92	5.53	23
W 6	2	98	1.0	5	6	108	2.97	4.10	(neg.)	2
	3	96	3.1	13	10	107	3.47	2.62	0.85	8
	4	90	13.1	25	15	105	7.18	2.05	5.13	18
W 7	2	139	2.9	5	8	118	5.34	5.46	(neg.)	1
	3	132	15.5	13	11	116	10.46	2.89	7.57	8
	4	108	30.2	29	20	110	7.59	2.36	5.23	19
W 8	3	93	2.1	8	9	106	3.85	3.84	0.01	3
	4	88	9.8	20	15	104	6.83	2.56	4.27	12
W 9	3	116	6.3	8	11	112	5.20	4.70	0.50	3
	4	104	24.6	23	18	109	6.40	2.68	3.72	14
W 10	4	95	16.3	20	20	106	6.32	3.42	2.90	10
W 11	4	108	28.2	48	43	109	9.96	3.06	6.90	24
W 12	4	86	12.9	98	18	103	6.75	0.63	6.12	82
W 13	4	112	33.8	66	38	111	8.33	1.97	6.36	38

Table XI

Calculated Data for Figure 47

(cf. page 173)

Run	Pass	P_{av}	ΔP	Pav	ΔP	T,°C.	A/1000	B/1000	f/1000	Pin
B 1	2	94.4	5.6	44	46	89	10.29	3.26	7.03	21
	3	88.1	7.0	88	26	86	6.05	1.01	5.04	67
	4	81.6	6.1	99	7	82	4.38	0.24	4.14	93
B 2	2	109.9	9.8	70	27	93	6.00	1.32	4.68	48
	3	99.6	10.9	89	19	90	4.80	0.73	4.07	75
	4	88.2	11.9	99	6	85	4.23	0.21	4.02	94
B 3	2	104.9	2.4	17	13	91	4.77	2.61	2.16	9
	3	100.8	5.9	34	30	89	5.66	3.01	2.65	24
	4	91.2	13.3	72	28	85	5.51	1.33	4.18	54
B 4	2	101.7	1.5	4	11	90	10.60	9.38	1.22	0.4
	3	97.8	6.3	20	22	89	8.56	3.75	4.81	11
	4	88.4	12.4	53	33	85	5.82	2.12	3.70	33
B 5	2	95.2	1.3	18	20	89	5.28	3.79	1.49	8
	3	91.6	5.9	46	43	87	9.09	3.19	5.90	28
	4	84.1	9.1	91	25	84	6.55	0.94	5.61	71
B 6	2	99.0	2.8	14	13	90	6.27	3.17	3.10	8
	3	94.9	5.4	30	19	88	5.45	2.16	3.29	21
	4	86.8	10.8	55	26	85	5.48	1.61	3.87	40
B 7	2	95.6	1.7	15	21	88	7.20	4.77	2.43	5
	3	91.4	6.7	41	38	87	9.94	3.16	6.78	26
	4	83.5	9.1	86	27	84	5.93	1.07	4.86	64
B 8	2	103.5	8.9	82	28	90	6.45	1.17	5.28	59
	3	96.7	9.3	97	23	92	5.28	0.81	4.37	87
	4	85.1	9.3	100	0	114	4.25	0	4.25	99
B 9	2	101.7	8.5	87	26	89	6.39	1.02	5.37	67
	3	93.1	8.7	100	7	86	5.25	0.24	5.01	93
	4	84.4	8.7	100	0	110	4.46	0	4.46	vapor
B 10	2	97.9	7.4	77	28	91	6.39	1.24	5.15	57
	3	91.1	6.3	97	15	88	4.05	0.53	3.52	85
	4	83.9	8.0	100	0	99	4.45	0	4.45	vapor
B 11	2	98.3	4.5	36	20	89	5.95	1.90	4.05	26
	3	92.8	6.5	57	18	87	5.15	1.08	4.07	46
	4	84.7	9.6	78	23	84	5.13	1.01	4.12	64
B 12	2	84.3	1.7	39	23	84	6.04	2.01	4.03	27
	3	82.0	2.8	63	22	83	5.99	1.19	4.80	50
	4	78.8	3.7	89	26	82	5.39	1.00	4.39	72

Calculated Data for Figure 47 (cont'd)

Run	Pass	P_{av}	ΔP	p_{av}	Δp	T°C.	A/1000	B/1000	f/1000	P_{in}
B 13	2	79.5	0.9	51	26	82	5.55	1.74	3.81	33
	3	78.4	1.3	74	18	81	5.46	0.83	4.63	60
	4	77.0	1.5	97	18	80	4.73	0.63	4.10	79
B 14	2	95.8	1.3	10	18	88	7.97	6.14	1.83	2
	3	92.4	5.4	34	34	87	9.41	3.41	6.00	20
	4	84.5	10.4	78	31	84	7.29	1.36	5.93	54
B 1A	2	97.6	1.8	16	23	89	7.17	4.91	2.26	7
	3	92.7	8.1	47	41	87	10.50	2.98	7.52	30
	4	83.6	10.1	88	20	83	6.36	0.78	5.58	71
B 2A	2	86.1	3.4	51	58	85	9.99	3.88	6.11	25
	3	82.4	4.0	98	17	83	5.87	0.59	5.28	83
	4	78.8	3.2	100	0	92	4.30	0	4.30	vapor
B 3A	2	79.5	1.8	95	36	83	6.24	1.29	4.95	64
	3	78.0	1.2	100	0	93	3.77	0	3.77	vapor
	4	76.8	1.2	100	0	99	3.65	0	3.65	vapor
B 4A	2	96.7	1.4	19	25	89	5.10	4.49	0.61	8
	3	92.1	7.9	53	46	87	9.86	2.96	6.90	33
	4	83.4	9.4	97	21	82	5.89	0.74	5.15	79
B 5A	2	86.8	3.6	52	57	85	9.60	3.74	5.86	25
	3	82.8	4.4	98	18	84	5.95	0.63	5.32	82
	4	79.0	3.2	100	0	96	3.92	0	3.92	vapor
B 6A	2	97.1	1.1	18	25	89	3.71	4.74	(neg.)	7
	3	92.2	8.7	52	42	87	9.69	2.76	6.93	32
	4	83.7	8.3	89	18	85	4.93	0.69	4.24	74
B 7A	3	102.6	6.3	25	26	90	7.89	3.55	4.34	14
	4	90.9	17.0	62	33	87	7.66	1.81	5.85	40

Table XII

Calculated Data for Figure 48

(cf. page 173)

Run	Pass	P_{av}	ΔP	P_{av}	ΔP	T, °C.	A/1000	B/1000	f/1000	Pin
Bo 1	2	97	1.0	5	11	88	12.7	7.5	5.2	1
	3	95	4.8	21	28	88	14.2	4.5	9.7	10
	4	87	12.1	63	40	85	11.1	2.1	9.0	37
BO 2	2	105	0.6	6	11	88	5.7	6.0	(neg.)	2
	3	101	9.5	22	28	88	23.6	4.3	19.3	10
	4	88	16.1	63	36	86	12.3	2.0	10.3	37
BO 5	2	92	1.8	8	18	89	22.4	7.4	15.0	0
	3	88	4.7	32	31	90	14.0	3.3	10.7	18
	4	82	9.3	58	13	96	14.0	0.8	13.2	49
BO 6	2	103	7.6	84	12	111	11.6	0.5	11.1	73
	3	94	9.9	87	3	126	12.8	0.1	12.7	85
	4	83	10.5	88	0	131	11.7	0	11.7	87
BO 7	3	85	2.1	3	8	96	18.2	8.0	10.2	0
	4	80	6.5	13	10	101	12.9	2.6	10.3	7
BO 8	4	79	2.7	6	4	107	10.8	2.1	8.7	2
BO 11	3	81	0.8	2	2	117	11.1	4.5	6.6	0
	4	79	3.0	6	5	122	10.0	2.8	7.2	2
BO 12	3	81	1.6	3	3	129	11.1	2.8	8.3	2
	4	79	2.9	4	1	142	14.2	0.9	13.3	6
BO 13	4	82	7.9	9	9	104	20.3	3.6	16.7	2
BO 14	3	89	4.3	5	10	105	19.8	6.8	13.0	2
	4	83	10.6	17	10	116	13.0	2.0	11.0	12
BO 15	2	96	2.3	4	9	106	14.5	7.2	7.3	0
	3	92	7.1	16	12	117	13.1	2.4	10.7	9
	4	83	10.3	22	3	136	11.8	0.5	11.3	20
BO 16	2	111	5.3	14	22	104	12.8	5.3	7.5	0
	3	102	12.5	29	9	122	12.7	1.1	11.6	24
	4	88	15.9	35	4	141	11.1	0.3	10.8	33
BO 17	2	111	3.6	8	17	102	14.9	7.2	7.7	0
	3	103	11.6	29	18	113	12.0	2.1	9.9	17
	4	89	16.8	37	4	130	11.3	0.3	11.0	35
BO 18	3	82	3.2	6	3	135	13.0	0.2	12.8	3
	4	78	3.3	9	4	147	7.6	1.5	6.1	4

Calculated Data for Figure 48 (cont'd)

Run	Pass	P_{av}	ΔP	P_{av}	ΔP	T, °C.	A/1000	B/1000	f/1000	P_{in}
BO 19	2	103	5.2	10	15	116	15.4	5.2	10.2	1
	3	96	9.5	20	5	136	11.9	0.9	11.0	17
	4	84	12.0	23	3	155	10.9	0.4	10.5	23
BO 20	2	118	9.8	27	22	117	12.4	2.8	9.6	11
	3	107	14.5	34	4	146	11.6	0.4	11.2	36
	4	90	18.1	37	1	159	11.0	0.1	10.9	38
BO 21	2	133	13.1	41	24	124	12.9	2.0	10.9	24
	3	117	18.8	45	2	153	13.7	0.1	13.6	50
	4	96	24.0	49	4	163	12.8	0.2	12.6	51
BO 22	3	100	7.3	15	21	101	15.3	4.8	10.5	5
	4	88	15.2	34	11	111	11.7	1.1	10.6	25
BO 23	2	117	4.3	6	15	99	26.6	8.5	18.1	0
	3	110	11.9	36	28	103	11.5	2.7	8.8	17
	4	94	21.2	51	10	116	11.8	0.7	11.1	47
BO 24	3	126	27.7	65	3	139	19.6	0.2	19.4	69
	4	98	29.0	67	15	150	15.0	0.1	14.9	73

SAMPLE CALCULATIONS

S A M P L E   C A L C U L A T I O N S

I. - Water Run W - 1

<u>Heat transfer (steam side)</u>:
Steam Pressure = 9.7 lbs./sq./in.

Latent heat of vaporization = 953 B.t.u./lb.[(8)]

Flashing factor = 1.028 (From Figure 53)

To get heat losses from jackets as gms. condensate/min., from Figure (53) read 12 gms./min. for jackets, 1,7,8,9,12 and 11 gms./min. for jackets 2,3,4,5,6,10, and 11.

Calculate net gms. condensate / min. by subtracting losses from measured gms./min.

Calculate q/θ (B.t.u./hr.) by formula:

$$Q/\theta = (\text{net gms./min.})(60)(\text{Latent heat})(\text{flashing factor})/454$$

Where 60 = minutes/hour

and 454 = gms./lb.

For Run W-1; $Q/\theta = (\text{net gms./min.})(60)(953)(1.028)/(454) = 130(\text{net gms./min.})$

Area per jacket = $(\pi)(1.062/12)(3.167) = 0.88$ sq.ft.

where (1.062/12) = inside pipe diameter, in feet

3.167 = heated length, in feet = 3 ft. 2 in.

Calculate $q/A\theta = (q/\theta)/0.88$

For jacket 1: Measured gms./min. = 150
Net gms./min.= 150-12 = 138
$q/\theta = 138 \times 130 = 18{,}000$ B.t.u./hr.
$q/A\theta = 18{,}000/0.88 = 20{,}000$ B.t.u./(hr.)(sq.ft.)

(All values of q/θ and $q/A\theta$ reported to the nearest 1000 B.t.u./hr.)

Total heat transfer by steam side = sum of q/θ for all jackets = 0.88 (sum of $q/A\theta$ for all jackets).

Heat transfer (water side):

Condenser water rate (lbs./sec.) is read from Figure (54).

For a head of 16.4 cm., the condenser water rate is 2.2 lbs./sec.

Net heat picked up in condenser =(water rate)(water rise, ^{o}F.)

For W-1; net heat picked up in condenser = (2.2 x 3600)(18.1 x 1.8)

= 258,000 B.t.u./hr.

where 3600 = sec./hr.

and 1.8 = ^{o}F/^{o}C.

Estimated heat losses from water side, based on benzene, boiling at 80^{o}C.

= 1,000 B.t.u./hr. for Separator
+ 1,000 B.t.u./hr. for Condenser
+ 2,000 B.t.u./hr. for Settler.

For water, boiling at 100^{o}C., multiply these heat losses by $\frac{100-25}{80-25} = 1.36$

For runs W-1 to W-9; heat loss = 1.36(1,000+1,000+2,000) = 5,000 B.t.u./hr.

For runs W-10 to W-15(no settler); heat loss = 1.36(1,000 + 1,000)
= 3,000 B.t.u./hr.

Total heat transfer by ~~steam~~ water side = net heat picked up by condenser + heat losses.

For W-1; Total heat transfer by ~~steam~~ water side = 258,000 + 5,000
=263,000 B.t.u./hr.

Fluid Pressures

The pressure drops across the passes, and the static pressure at the end of the fourth pass are given in cm. of Hg., as calculated by the following formulas:

$$\Delta P \text{ across pass (cm.Hg.)} = (12.6/13.6) \text{ (actual reading)} = 0.926 \text{ (actual reading)}$$

For W-1 the actual reading for pressure drop across the fourth pass was 11.2 cm.(mercury-water) which was reported as 0.926 x 11.2 = 10.4 cm. mercury.

At the end of the 4th pass, one arm of the manometer was open to the atmosphere and was not filled with water.

Actual reading = Pressure(cm.Hg.)+(1/2)(actual reading)/ 13.6

$$\text{Pressure(cm.Hg.ga.)} = \left(1 - \frac{1}{2 \times 13.6}\right)\text{(actual reading)} = 0.963\text{(actual reading)}$$

Fluid pressures (cont'd):

For W-1, the actual reading , ~~reading~~ for static pressure at end of fourth pass was 4.1, which was reported as 0.963 x 4.1 = 4.0 cm. Hg. ga.

A minimum pressure drop of 1/2 cm. Hg. was assigned to the first heated pass regardless of readings, because manometer fluctuations made smaller readings questionable.

Cumulative Pressure Drop = cumulative sum of pressure drops across each pass, beginning with 1st pass.

For W-1, cumulative pressure drop.
= 0.5 at end of 1st pass
= 0.5 + 1.2 = 1.7 at end of 2nd pass
= 1.7 + 4.7 = 6.4 at end of 3rd pass
= 6.4 + 10.4 = 16.8 at end of 4th pass

Static fluid pressure at any point = sum of static pressure at end of 4th pass, plus any intervening pressure drops.

For W-1: Static pressure (cm. Hg. ga.) at end of 4th pass = 4.0
Static pressure at end of 3rd pass = 4.0 + 10.4 = 14.4
Static pressure at end of 2nd pass = 14.4 + 4.7 = 19.1, etc.

Calculation of Feed Rate:

$$q = \frac{S_o \, c \sqrt{2g \, H_v}}{\sqrt{1 - (S_o)^2 / (S_1)^2}}$$

Equation 6, page 60 P.C.E.( )

where q = cu. ft./sec., not B.t.u./hr.

Hence:

$$W' = W/3,600 = (q)(\rho) = S_o \, \rho \, c \sqrt{2g \, H_v} \; / \; \sqrt{1 - (D_o / D_\rho)^4}$$

$2g = 64.4$; $\rho = 60.7$ (at average feed temperature of 80°C.)

$$H_v = \text{ft. of water} = (\text{reading}) \, \frac{\text{cm (Hg-H}_2\text{O)}}{} \; 1/30.5 \; \frac{\text{ft (Hg-H}_2\text{O)}}{} (13.6 - 1) \; \frac{\text{ft H}_2\text{O}}{}$$

$$= (\text{reading})/2.42$$

$D_o = 0.213$ in.; $D_\rho = 1.049"$; $\sqrt{1 - (D_o / D_\rho)^4} = 0.999$, say 1.

$$S_o = (\pi/4)(0.213/12)^2 = 1/4040$$

Calculation of Feed Rate: (Cont'd)

$c =$ function $(Re)_o$; Fig. 21, page 61, P.C.E.( )

$$W' = \frac{60.7 \times c \times \sqrt{64.4\,(\text{reading})/2.42}}{4040} = (c)\sqrt{(\text{reading})}/(13.1)$$

$$(Re)_o = D_o V_o \rho/\mu = (0.213/12)(V_o)(60.7)/(0.000672\ z)$$

where z = viscosity of water in centipoise, at feed temperature.

$$V_o = W'/\rho S_o = 4040W'/60.7 = 66.5\ W'$$

Sample Calculation for Run W-1:

Orifice reading = 4.9; assume $c = 0.61$

$W' = (0.61)\sqrt{4.9}\ / 13.1 = 0.103$ lbs./sec; $V_o = 66.5 \times 0.103 = 6.85$ ft./sec.

$z = 0.432$ at 65.5°C.; $(Re)_o = 1605 \times 6.85/0.432 = 25{,}400.$

at $(Re)_o = 25{,}400$: $c = 0.61$ (check)

$W = 3600 \times 0.103 = 372$ lbs./hr.

Weight per cent of feed vaporized (p):

A special labor-saving device is used for the calculation of the per cent of feed vaporized leaving any jacket. This involves first calculating the per cent vaporized if the water were boiling at 1 atmosphere absolute and 100°C. This fictitious per cent vaporized is labelled p'. Subsequently, the mixture at 1 atmos. and p' vaporized is compressed adiabatically to the actual pressure, condensing from p' to the true per cent vaporized, p. p' is the value of per cent vaporized which would exist at any point if the mixture were flashed to atmospheric pressure.

Weight per cent of feed vaporized (p) (cont'd) :

Calling the specific heat of water equal to 1, the heat needed to heat the feed in Run W-1 from the feed temperature (65.5°C.) to 100°C. is (372) (100-65.5) (1.8) = 23,000 B.t.u./hr.

B.t.u./hr. in J.1 = (0.88)x(Q/A) = 0.88 x 20,000 = 18,000 (no vaporization)

B.t.u./hr. in J.2 = 0.88 x 16,000 = 14,000

B.t.u./hr. in first two jackets = 18,000 + 14,000 = 32,000

Net B.t.u./hr. for vaporization at 100°C. in Jacket 2 = 32,000-23,000=9,000

Lbs. water vaporized at 100°C. in J.2 = 9,000/970 = 9.3 lbs./hr.

where 970 = latent heat of vap$^{\underline{n}}$ (B.t.u./lb.) at 100°C.

p' for J.2 = (100%) 9.3/372 = 2-1/2 %

Lbs. water vaporized at 100°C. in each remaining jacket = 0.88(q/A)/970

For J.3; lbs. water vaporized at 100° C. = 0.88 x 19,000/970 = 17.2 lbs./hr.

Cumulative lbs./hr. vaporized at 100°C. in J.3 = 9.3 + 17.2 = 26.5 lbs./hr.

p' for J.3 = (100%) 26.5/372 = 7%

Similarly, for J.12, cumulative lbs./hr. vaporized at 100°C. = 258.1,

and p' for J.12 =(100%) 258.1/372 = 69%

To convert from p' to p use Fig. (56):

For J.2, static pressure = 20 cm.Hg.ga. and p' = 2.5%

p' - p = 1.2%; p = 2.5 - 1.2 = 1% vaporized

For J.3, static pressure = 20 cm. Hg. ga. and p' 7%

p' - p = 1.2%; p =7-1 = 6% vaporized

For Jacket 12, static pressure = 4 cm. Hg. ga. and p' = 69

p' - p = Negligible; p = 69 % vaporized

Calculation of initial boiling point:

Initial boiling point = saturation temperature corresponding to initial pressure = 106.8°C., corresponding to 20.8 cm. Hg. ga. This neglects pressure drop prior to boiling, but saturation temperature leaving 1st pass in Run W-1 is 106.7°C.

Sensible heat = (372)(106.8 - 65.5)(1.8) = 28,000 B.t.u./hr.

q in J. 1 = 18,000 (see above), so no vaporization in Jacket 1.

q in J. 2 = 14,000 (see above)

Sensible heat in J. 2 = 28,000 - 18,000 = 10,000 B.t.u./hr.

Boiling starts at about (100%) 10,000/14,000 = 70% of J.2

Calculation of temperature leaving Jacket 1.:

$(372)(t-65.5)(1.8) = 18{,}000; \quad t = 65.5 + 26.9 = 92.4^{\circ}$ C.

Calculation of average p in Jacket.

Average p in jacket = $(p_{in} + p_{out})/2$.

For J.9, Run W-1; p_{av} = 34 + 43 = 39%, expressed to nearest whole per cent.

If vaporization starts at more than 30% of jacket, do not consider data on the jacket as data for boiling water.

If vaporization starts at, say, 20% of jacket, read p_{av} at mid-point of jacket

For J. 1, Run W-12; p_{av} = 18 by Figure 12.

Calculation of temperature differences:

Temperature differences are read directly from Figure 1 - 14, taking the difference between the condensing steam temperature and the fluid temperature. For jackets which are used entirely for boiling, the temperature difference is taken at the mid-point of the jacket.

For J. 11, Run W-1; $\Delta T = 114.8 - 103.2 = 11.6$, say 11 - 1/2°C.

Calculation of temperature differences:

For jackets which are used entirely for preheating, the temperature difference is taken as the logarithmic average of the initial and final temperature difference.

For J.11, Run W-1: $\Delta T = 114.8 - 65.5 = 49.3^{\circ}C.$, initial

$\Delta T = 114.8 - 92.4 = 22.4^{\circ}C$, final

$(\Delta T)_{e.m.} = (49.3 - 22.4)/\ln(49.3/22.4) = 34.1^{\circ}C.$

Calculation of overall heat transfer coefficient (U):

$q/A = U\Delta t;\quad U = (q/A)/(\Delta t, {}^{\circ}F.)$

For Run W-1: U in J.1 = $20{,}000/(1.8 \times 34.1) = 325$

Calculation of velocities

$W = VA/v;\quad V = v\,W'/A$

v = specific volume in./ft./lbs.

= 0.016 for liquid water

W' = feed rate, lbs./sec. = W/3600

$A = (\pi/4)(1.049/12)^2 = 1/167$

$V = v(W/3600)(167) = vW/21.6$

For W-1, when all liquid; $V = 0.016 \times 372/21.6 = 0.28$ ft./ sec.

If part vapor; $100v = p(v)_{vapor} + (100 - p)(0.016)$

If $p \leqq 10\%$, neglect volume of liquid, and $v = (N)_{vapor}/100$

Get $(v)_{vapor}$ from steam tables (8) for saturated at given pressure

For W-1, J. 4: Fluid pressure = 4 cm. Hg. ga. = 80 cm. Hg. abs.
= 15.5 lbs./sq. in.

$(v)_{vapor}$ at 15.5 lbs./sq. in. = 25.5 cu. ft./lb.

Calculation of velocities,(cont.)

$p = 69\ \%$

$v = (0.69)\ (25.5) = 17.6$ cu. ft./lb.

$V = 17.6 \times 372/21.6 = 300$ ft./sec.

Calculation of average velocity in jacket:

Av velocity taken as geometric mean = $\sqrt{V_{in} \times V_{out}}$

For J.3, Run W-1 ; $V_{av} = \sqrt{3.9 \times 22} = 9.3$ ft./sec.

Sample Calculations

II. - Benzene Run B 9

Calculation of the benzene runs is analogous in almost every respect with calculation of the water ruhs, with the exception of those benzene runs in which the feed was completely vaporized and superheated. This latter occurrence was not encountered in the water runs.

Minor variations are encountered in estimating heat losses.

Heat transfer (steam side):

It is assumed that the un-lagged condensate discharge lines condense all steam evolved when steam condensate is flashed to atmospheric pressure.

Steam pressure = 28 lbs./sq.in.

Latent heat of vaporization = 930 B.t.u/lb. (8)

For Jacket 1; measured condensate = 1940 gms./4 min.

gross heat transfer = (1940/4) (60) (930)/(454) = 59,600 B.t.u./hr.

Estimated heat losses = 500 B.t.u./(hr.) (jacket) (cf. ~~Fig.~~ p. 339A).

net heat transfer = 59,100 B.t.u./hr.

Repeating for 12 jackets, and adding: total heat transfer = 160,000 B.t.u./hr

Heat transfer (water side):

For a head of 4.8 cm., the condenser water rate is 2.88 lbs./sec. (Fig. 55)

Heat picked up in condenser = (2.88) (3600) (1.8) (8.0) = 150,000 B.t.u./hr.

Estimated heat losses = 3000 B.t.u./hr.

(For B6A & B7A heat losses taken as 5000 because of settler drum installation.)

Total heat transfer = 153,000 B.t.u./hr.

Calculation of q/A:

Average total heat transfer = (160,000 + 153,000)/2 = 157,000 B.t.u./hr.

q/A for J.l. = (59,100/0.88) (157,000/160,000)=65,800 B.t.u./(hr.)(sq.ft.)

Calculation of Feed rate (by heat balance):

Feed enters at 45°C (all liquid) and leaves at 124°C (all vapor).

Heat absorbed / lb. of feed = (0.445) (1.8) (80.5-45) + 170 + (0.29) (1.8) (124-43.5)

= 28.4 + 170 + 22.6 = 221 B.t.u./lb.

0.445 = sp. heat of liq. benzene at average temp. of 63°C. (Fig. 50)

170 = latent heat of vaporization of benzene (B.t.u./lb.) at 80.5°C. (Fig. 50)

0.29 = sp. heat of benzene vapor (7).

Feed rate = 157,000/221 = 710 lbs./hr.

Calculation of Feed rate (by orifice):

Same procedure as in water runs, using suitable values for ρ & z, and using an orifice diameter of 0.206 inches instead of 0.213 inches.

For Run B9, calculate feed rate = 660 lbs./hr.

$$\frac{\text{Feed rate (heat balance)}}{\text{Feed rate (orifice)}} = \frac{710}{660} = 1.076 = \text{rate factor.}$$

When feed rate by heat balance can not be calculated because of incomplete vaporization, determine feed rate by orifice and multiply by an average rate factor of 1.08.

Calculation of final boiling point:

When dry, superheated vapor is formed, the following trial and error procedure is used to estimate the final boiling point.

For Run B9, the final temperature is 124°C. Assume final boiling temperature is 88°C.

Sensible heat of vapor = (0.29) (1.8) (124-88) (710) = 13,300 B.t.u./hr.

Heat transfer in last 5 jackets = 0.88 (1800 + 2100 + 3200 + 4500 + 4000)

= 13,700 B.t.u./hr.

Latent heat transfer in J. 8 = 13,700 - 13,300 = 400 B.t.u./hr.

Boiling apparently ends at 400/(0.88) (4,000) = 11% of J. 8

At this point the static pressure = 18 cm. of Hg.ga. (cf. Fig. 35) and the saturation temperature = 88°C. This checks the assumed temperature in this case and it is unnecessary to repeat the calculation.

The rest of the calculations are identical to the calculations for the water runs. In converting from p' to p for benzene, use Fig. 57.

Sample Calculations

III - Benzene - Oil Run BO - 22

Heat Transfer (steam side):

Procedure for Runs BO - 11 to BO - 24 identical with that for water runs. Earlier benzene-oil runs use procedure identical to that of benzene runs. For BO-22; Q_s = 111,400 B.t.u./hr.

Heat Transfer (water side)

Heat picked up by condenser is calculated in the same fashion as for the water runs. For BO-22, the heat picked up by the condenser was 82,300 B.t.u./hr.

cooling water rate (lbs./sec.) is read from Fig.(54)

For a head of 21cm., the cooling water rate is 0.41 lbs./sec.

Heat picked up in cooler = (water rate) (water rise,°F.).

For BO-22; heat picked up in cooler = (0.41 x 3600) (9.35 x 1.8)

= 24,800 B.t.u./hr.

Take estimated heat losses the same as for benzene runs. For Runs BO-1 to BO-12 losses = 3000 B.t.u./hr.

For Runs BO-13 to BO-24 losses = 5000 B.t.u./hr. because of settler drum.

Total heat transfer by water side = heat picked up in condenser + heat picked up in cooler + losses.

For BO-22; total heat transfer by water side = 82,300 + 24,800 + 5,000

=112,100 B.t.u./hr.

q/A for jackets, as calculated above, were multiplied by $\frac{112,100}{111,400}$, basing the heat flux on heat transfer determined by water side.

Fluid Pressures: Calculations identical with those for water runs.

Restriction of minimum pressure drop of 0.5 cm. Hg. per pass not applied.

<u>Calculation of Feed Rate by orifice</u>; Calculation identical to that for water runs, save that appropriate values of ρ and z for the feed mixture are used. For runs BO-1 to BO-12, feed orifice diameter = 0.206 in. instead of 0.213 inches.

<u>Calculation of Feed Rate by Heat and Material Balances</u>:

Feed analysis = 46 °/o benzene

Product analysis = 14°/o benzene.

Basis: 1 lb. of feed.

$$\text{lbs. benzene vaporized} = \left[\left(\frac{46}{100-46}\right)-\left(\frac{14}{100-14}\right)\right] (100-46)$$

$$= 0.372 \text{ lbs.}$$

Feed temperature = 58.5°C.

Product temperature = 118.4°C.

Av. temperature = 88.5°C.

specific heat of feed at 88.5°C = 0.488 (by Fig. 50)

heat of vaporization of benzene at 118.4°C = 156 B.t.u./lb. (by Fig. 50)

Heat absorbed per lb. of feed = 0.488 x 1.8 x (118.4 - 58.5) + 156x0.372

= 110.5 B.t.u.

Feed rate = 112,100/110.5 = 1015 lbs./hr.

<u>Determination of initial boiling point:</u>

	Pressure (cm. Hg. abs.)	Observed Temperature	Apparent Composition	Saturation Temp.
Entering	103.9	58.5	—	—
Leaving 1st Pass	103.2	85.0	—	98.4
Leaving 2nd Pass	102.0	98.1	45°/o	98.0
Leaving 3rd Pass	94.7	105.0	28°/o	95.0
Leaving 4th Pass	79.5	118.4	15°/o	—

Pressure = fluid pressure (cm. Hg. ga.) + 76 = cm. Hg. abs.

Apparent composition = $^o/_o$ benzene in liquid boiling under the specified pressure at the specified temperature, as read from Fig. 51.

Saturation temp. = boiling temp. at specified pressure of a mixture containing 46 $^o/_o$ benzene (feed composition), as read from Fig. 51.

It is concluded that boiling starts near the end of the second pass at a temperature of close to 98.0°C.

To heat feed from 58.5°C to 98.0°C.

$$= 0.478 \times 1.8 \times (98.0-58.5) \times 1015 = 34{,}500 \text{ B.t.u./hr.}$$

where 0.478 = Cp. for feed at average temp. of 78°C. (Fig. 50).

Heat transfer in 1st 5 jackets = 0.88 (sum of q/A for 1st 5 jackets)

$$= 0.88\ (8{,}000 + 8{,}300 + 7{,}900 + 7{,}700 + 7{,}600)$$

$$= 34{,}800$$

Latent heat in fifth Jacket = 34,700 - 34,500 = 300 B.t.u./hr.

$^o/_o$ of 5th Jacket used for boiling = $\frac{300}{0.8 \times 7600}$ = 4.9 $^o/_o$, reported as substantially zero.

<u>Determination of percent vaporized (p)</u>:

The cumulative weight percent vapor is calculated by moving step-wise through the jackets, after having assumed a fluid temperature curve. This curve is drawn on Fig. 39 to pass throught the boiling temperature at the boiling point, and thence through the successive observed temperatures. Based on this temperature gradient, the sensible heat in the jacket is subtracted from the total heat transfer, the difference being used to calculate the lbs. of feed vaporized in the jacket. If the temperature gradient (as drawn) is too steep in one jacket it must be too flat in another jacket. Hence, although the actual vaporization in a jacket lacks a certain amount of precision, the cumulative vaporization is less subject to shape of the temperature gradient.

A sample calculation is presented for J. 12 of Run BO - 22. It has been calculated that 359.9 lbs. of vapor per hour are leaving J. 11 at an indicated temperature of 113.5°C. The mixture leaves J. 12 at a temperature of 118.4°C, and the heat transfer in the jacket = (0.88)(6,100) = 5,400 B.t.u./hr.

Sensible heat transfer =(0.514)(1.8)(118.4-113.5)(1,015)=

4,700 B.t.u./hr.

0.514 = Cp. for 46% benzene mixture at temperature of 116°C.(Fig. 50).

Total heat transfer = 5400 - 4700 = 700 B.t.u./hr.

Pounds vaporized = 700/157 = 4.5 lbs./hr.

157 = latent heat of vaporization of benzene at 118°C.(Fig.50)

Total vaporization = 359.9+4.5 = 364.4 lbs./hr.

Cumulative weight per cent vapor (p") leaving J.12 =

364.4/1015 = 35.9%

This value of 35.9% is corrected below to 37.2%.

Determination of per cent benzene in liquid phase (x):

Let f = % benzene in feed

X = (100%)(f-p)/(100-p) = residual benzene/ residual total liquid.

also; p = 100% (f-X)/(100-X)

If p" out of J. 12 of Run BO - 22 = 35.9%; X" =(46 -35.9)/(100-35.9)=

15.7%

But X for liquid leaving J. 12 = 14%, by analysis.

If X = 14%; p = (46 - 14)/(100 - 14) = 37.2%

This inconsistency arises from cumulative errors in the cumulative calculation of p'', as well as minor inconsistencies in the change of latent heat and specific heat with temperature. Accordingly, values of X'' are calculated corresponding to each value of p''. These values of X'' are then corrected by the equation $(46 - X)/(46 - X'') = (46 - 14)/(46 - 15.7)$. Based on these calculated values of X, corresponding values of p are calculated and reported.

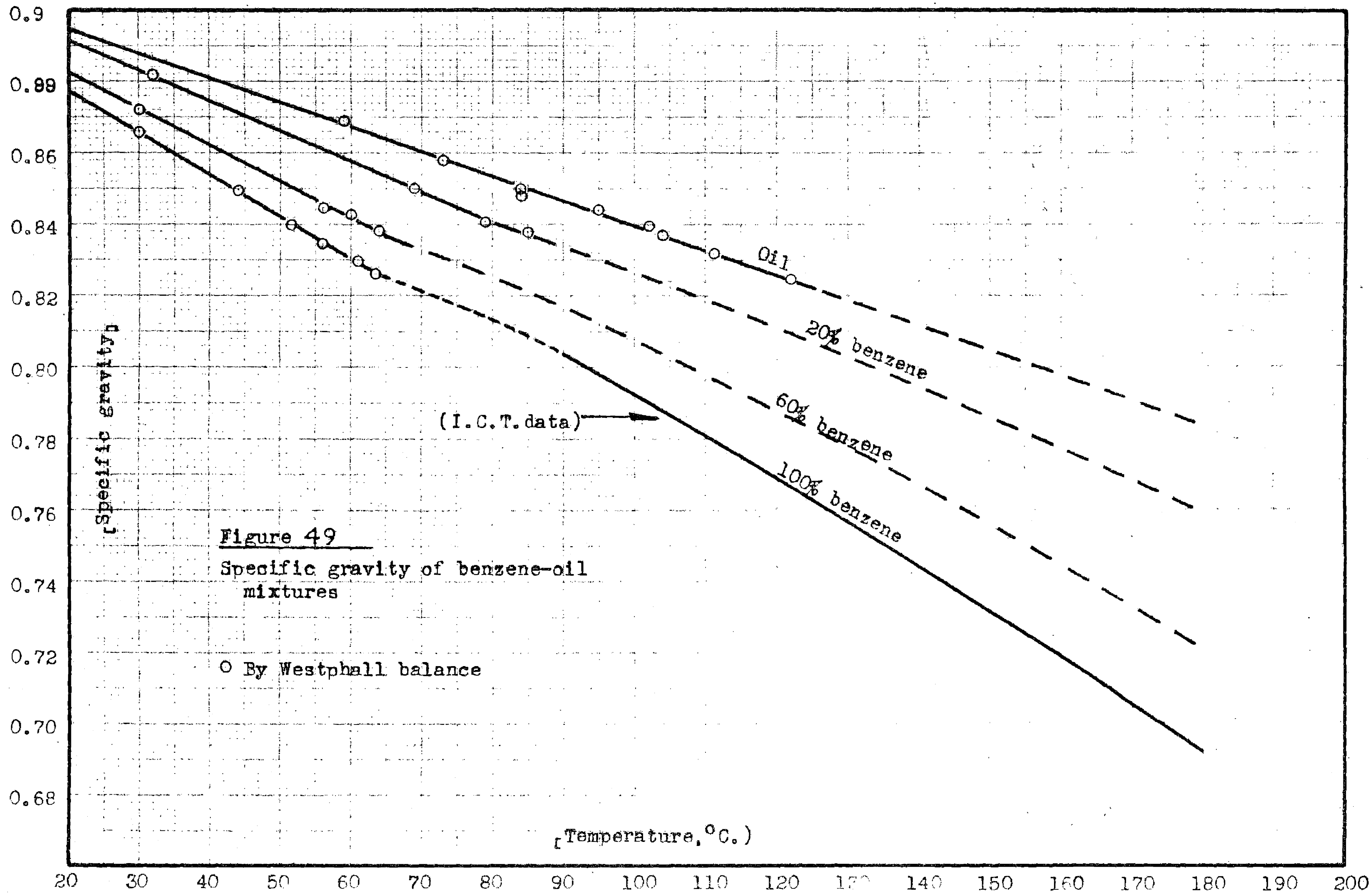


Figure 49
Specific gravity of benzene-oil mixtures

~~Figure 29~~

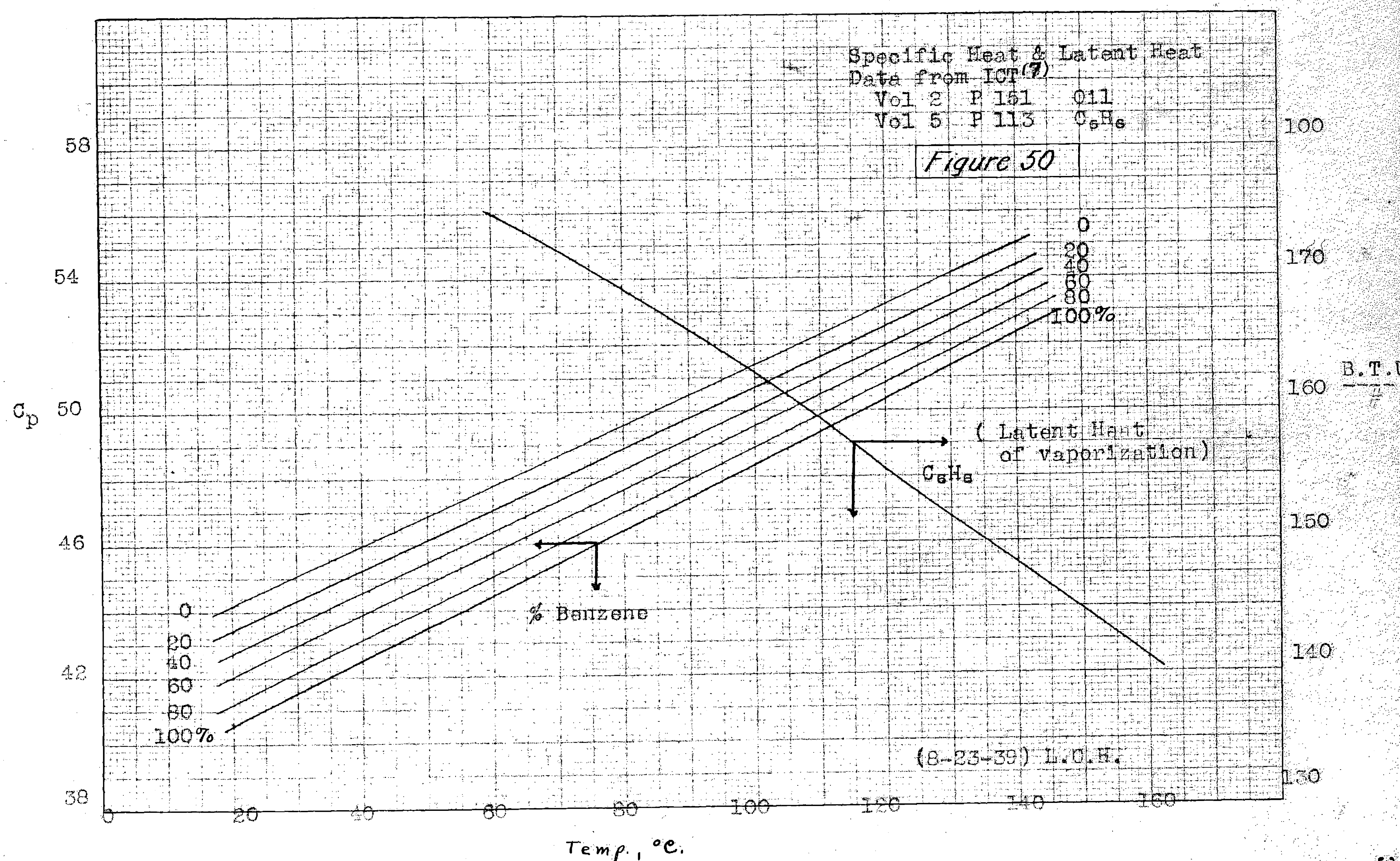
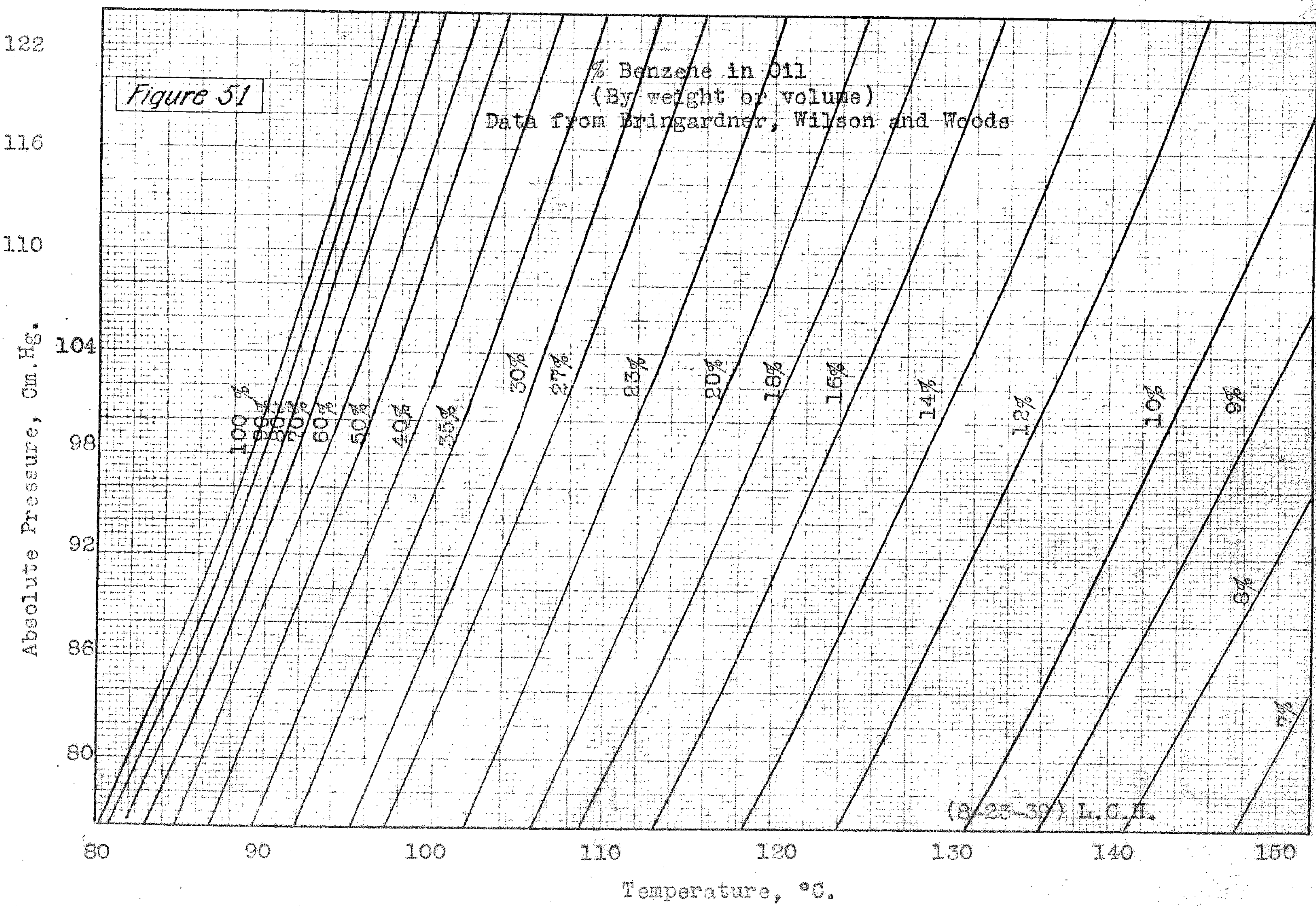


Figure 50

Figure 51


% Benzene in Oil
(By weight or volume)
Data from Bringardner, Wilson and Woods
100%
90%
80%
70%
60%
50%
40%
35%
30%
27%
23%
20%
18%
16%
14%
12%
10%
9%
8%
7%
(8-23-39) L.C.H.
Absolute Pressure, Cm.Hg.
122
116
110
104
98
92
86
80
Temperature, °C.
80
90
100
110
120
130
140
150

Derivation of Figure

The atmospheric boiling point of various benzene-oil mixtures was determined in a 500 cc. flask equipped with a 9-inch water-cooled Leibig condenser, using a 260 cc. sample and using glass beads in the bottom of the still. The fractional thermometer was immersed in the liquid. The still was heated through a sand bath.

Assuming that Raoult's law was applicable to the benzene in a mixture of benzene and oil boiling at atmospheric pressure, and assuming that the oil was completely non-volatile, the apparent molecular weight of the oil could be calculated. It was found that the apparent molecular weight of the oil was substantially constant at about 380 for mixtures containing $40^{o}/_{o}$ to $80^{o}/_{o}$ benzene. When less benzene was present the apparent molecular weight of the oil fell off substantially linearly to a value of 220 at $5^{o}/_{o}$ benzene in the oil.

For pressures above one atmosphere the boiling temperature of benzene was found in the I.C.T. (7). It was assumed that the apparent molecular weight of the oil did not vary with temperature for a given benzene oil mixture, and the vapor pressure of mixtures at elevated temperatures was calculated using Raoult's law for the benzene, assuming the oil non-volatile, and using the above-calculated molecular weights.

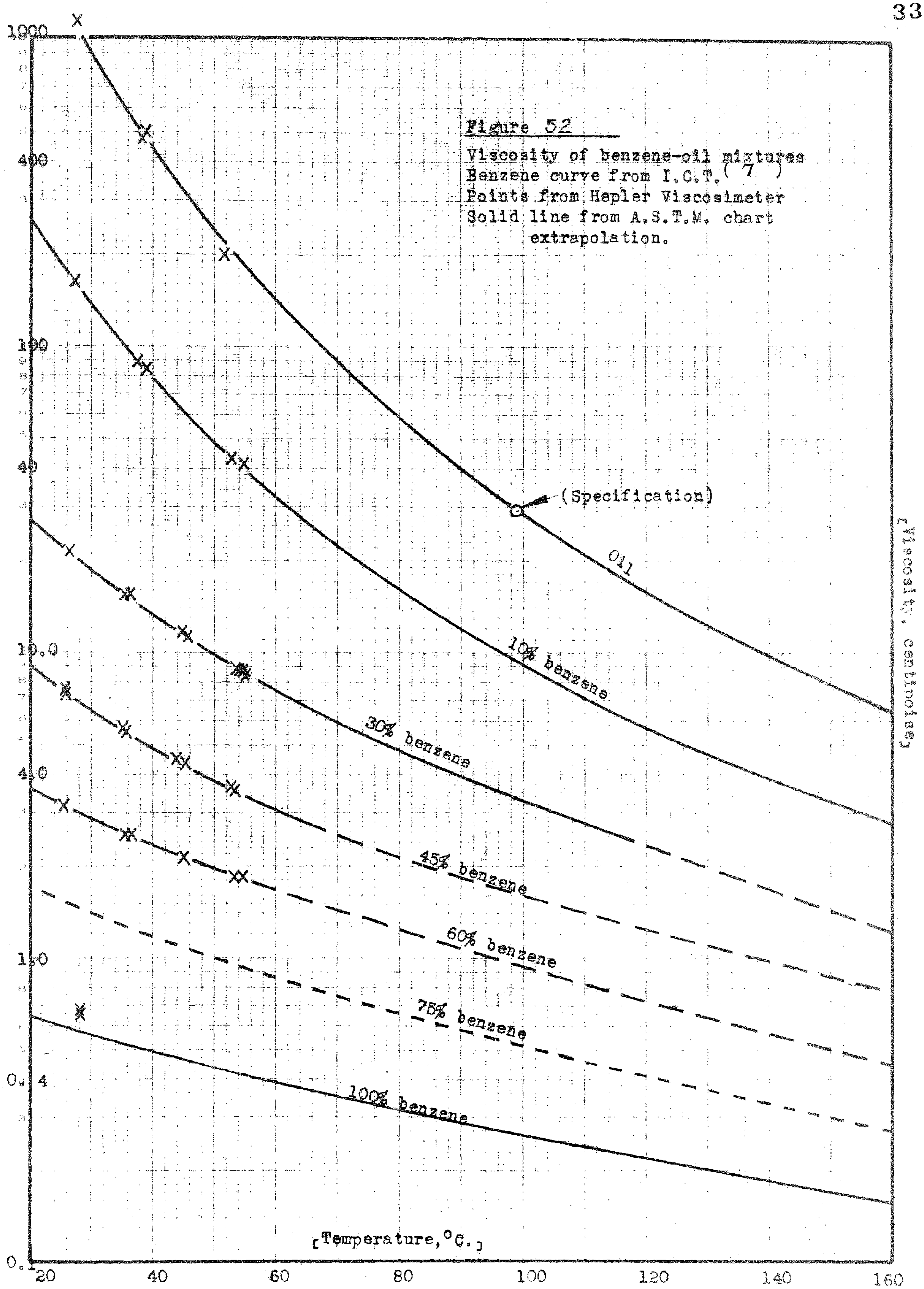


Figure 52
Viscosity of benzene-oil mixtures
Benzene curve from I.C.T.(7)
Points from Hepler Viscosimeter
Solid line from A.S.T.M. chart
extrapolation.
(Specification)
Oil
10% benzene
30% benzene
45% benzene
60% benzene
75% benzene
100% benzene
1000
400
100
40
10.0
4.0
1.0
0.4
0.1
20
40
60
80
100
120
140
160
Temperature, °C.
Viscosity, centipoise

Viscosity data Figure 52

Data on viscosity were taken in a Hepler viscosimeter, industrial model.

Sample	T, °C.	z(c.p.)
Pure benzene	28.2	0.672
	28.2	0.656
60% benzene	25.4	3.18
	25.4	3.12
	25.4	3.18
	35.6	2.52
	36.2	2.51
	44.9	2.12
	44.9	2.14
	53.1	1.81
	54.6	1.84
45% benzene	25.7	7.54
	25.7	7.29
	35.1	5.62
	35.5	5.50
	43.5	4.47
	45.1	4.34
	52.7	3.64
	53.1	3.57
30% benzene	26.2	21.1
	26.4	21.4
	26.4	21.4
	26.4	21.1
	35.2	15.4
	35.5	15.5
	36.1	15.3
	44.6	11.5
	45.2	11.5
	45.8	11.3
	53.4	8.91
	54.4	8.72
	54.5	8.60
	55.0	8.66
	55.1	8.46
10% benzene	27.1	162
	27.2	160
	37.7	88.3
	38.7	83.6
	52.7	42.8
	54.5	41.3

Sample	T, °C.	z(c.p.)
Pure oil	27.5	1128
	27.5	1126
	38.2	475
	38.3	499
	38.8	490
	38.9	497
	51.8	199

I.C.T.,[7] P. 12, Volume V. for pure benzene.

From 0 - 75°C., $z = 1442 / (90 + t^{\circ}C.)^{1.64}$ (1)

From 0 - 190°C., $z = z' \, [\, 1 - 0.0123\,(p - 1)\,]$ (2)

Where z = viscosity (c.p.) of fluid under its own vapor pressure.

z' = viscosity calculated from Equation 1.

p = vapor pressure, in atmospheres.

T °C.	z'	p-1(atm.)	z(c.p.)
20	0.647	0	0.647
30	0.561		0.561
40	0.492		0.492
50	0.436		0.436
60	0.389		0.389
70	0.350		0.350
80	0.318		0.318
90	0.289	0.326	0.288
100	0.264	0.757	0.262
110	0.248	1.29	0.244
120	0.224	1.94	0.219
130	0.208	2.71	0.201
140	0.194	3.64	0.185
150	0.180	4.70	0.170
160	0.169	5.98	0.157

<u>Specifications on oil:</u>
(Humble Oil and Refining Co., May 31, 1938, File H 4.18 - 1052; PA 40,009)

Gravity, °A.P.I.	25.7
O.C.Flash, °F.	555
Vis. @ 210°F., S.S.U.	164
Pour Point, °F.	5

Therefore sp.gr.=0.878
and z = 30.7 c.p. at 99°C.,
using .878
z = 29.4 c.p. at 99°C.,
based on sp.gr.=0.84 @ 99°C.

Viscosities extrapolated to 37 centistokes by A.S.T.M. chart D341-32T.

SUPPLEMENTARY PLOTS AND TABLES

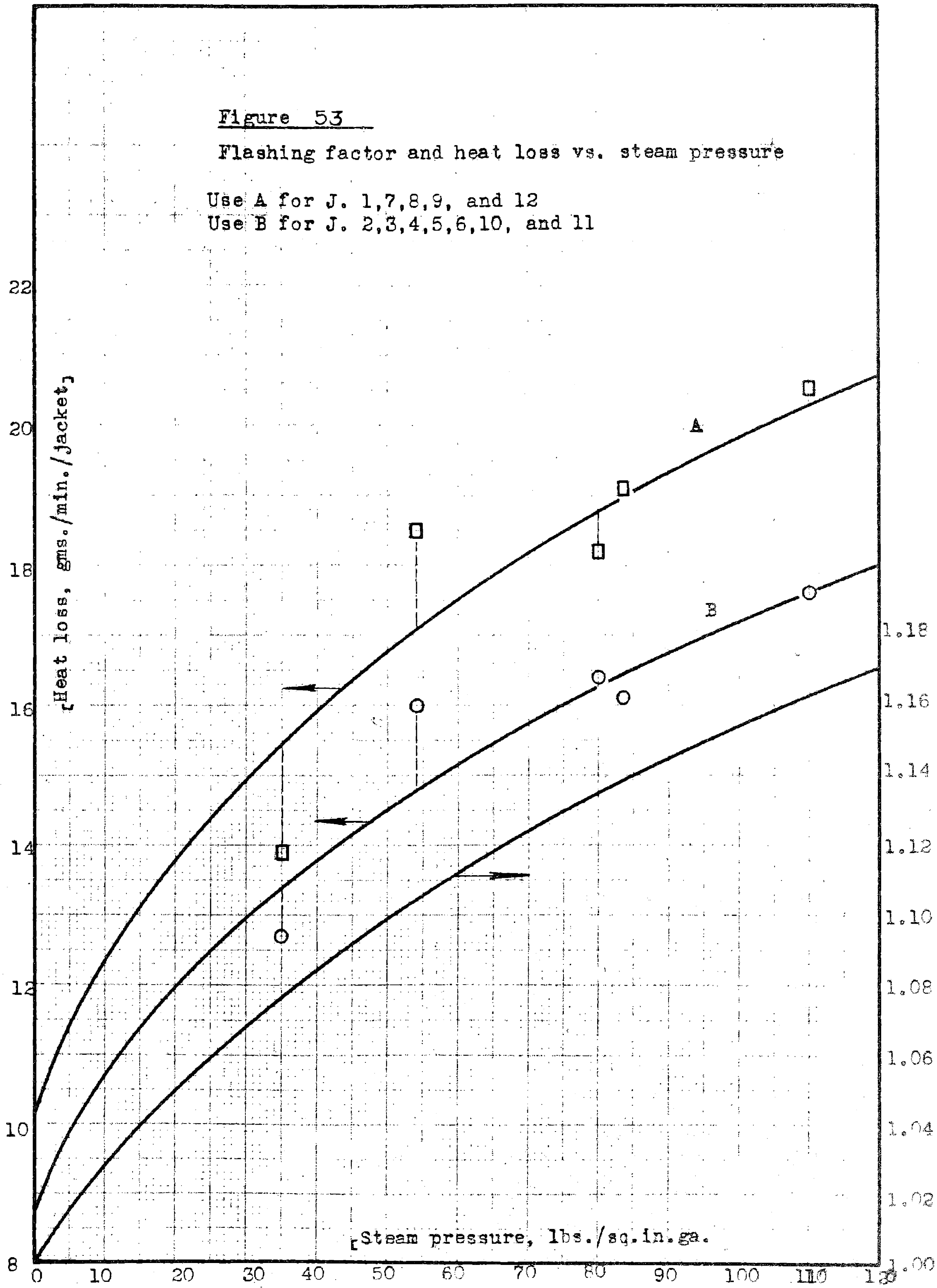


Figure 53
Flashing factor and heat loss vs. steam pressure
Use A for J. 1,7,8,9, and 12
Use B for J. 2,3,4,5,6,10, and 11
Heat loss, gms./min./jacket
Steam pressure, lbs./sq.in.ga.
A
B
8
10
12
14
16
18
20
22
0
10
20
30
40
50
60
70
80
90
100
110
120
1.00
1.02
1.04
1.06
1.08
1.10
1.12
1.14
1.16
1.18

Derivation of Flashing Factor (Figure 53)

Flashing factor = ratio of total condensate formed to residual condensate collected after adiabatic flashing to atmospheric pressure.

Sample calculation for 120 lbs./sq.in.ga. steam pressure

Basis: 1 lb. condensate at 134.7 lbs./sq.in.abs.

Enthalpy = 321.6 B.t.u./lb.[8]

Enthalpy condensate at 212°F.= 180.0

Enthalpy vapor at 212°F. = 1150.2

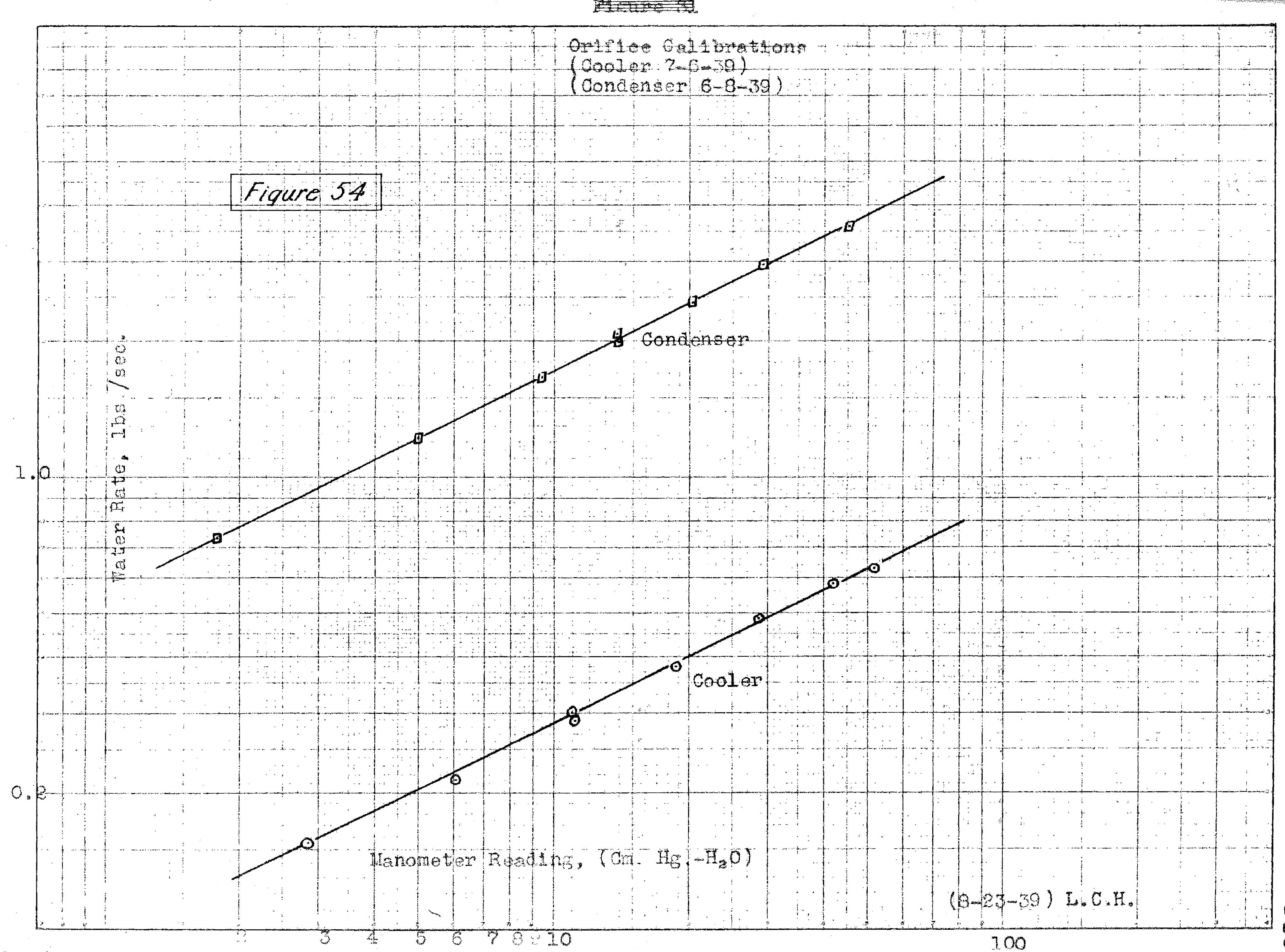
y = lbs. of steam flashed

$321.6 = y(1150.2)+(1-y)1800; \quad y = 0.146$

Flashing factor = $1/(1-y) = 1.171$

Estimated Steam-Side Heat Loss for Benzene Runs

The combined coefficient of radiation and convection from the jacket insulation was assumed to be 2.0 B.t.u./(hr.)(sq.ft.)($^\circ$F.). The exposed surface of the evaporator, as insulated, was estimated at 40 sq. ft., or 3 1/3 sq. ft. per jacket. The surface temperature of the insulation was estimated to be 130°F. when the steam pressure was about 7 lbs./sq.in.ga., this estimate being based upon feeling the insulation with the hand. Hence, taking the room temperature as about 70°F., the heat loss at low steam pressures would be equal to (2.0)(3.33)(130-70), or 400 B.t.u./(hr.) (jacket). At higher steam pressures the heat loss was assumed proportional to the overall temperature difference. Thus, with steam at 28 lbs./sq.in.ga. condensing at 272°F., the estimated heat loss is equal to (400)(272-70)/(233-70), or 500 B.t.u./(hr.)(jacket).


Orifice Calibrations
(Cooler 7-6-39)
(Condenser 6-8-39)
Figure 54
Condenser
Cooler
Water Rate, lbs /sec.
1.0
0.2
Manometer Reading, (Cm. Hg.-H_2O)
3
4
5
6
7
8
10
100
(8-23-39) L.C.H.

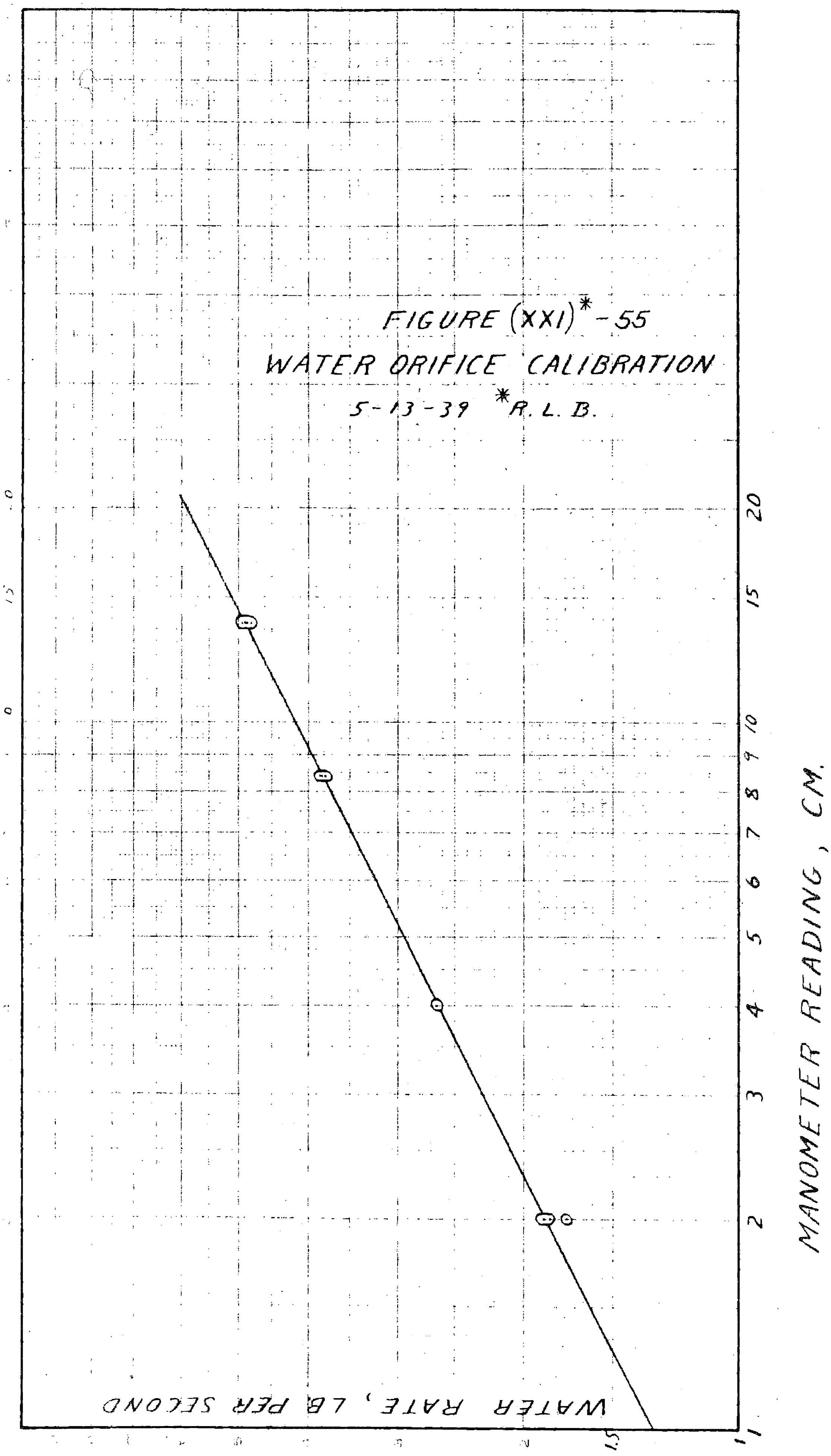
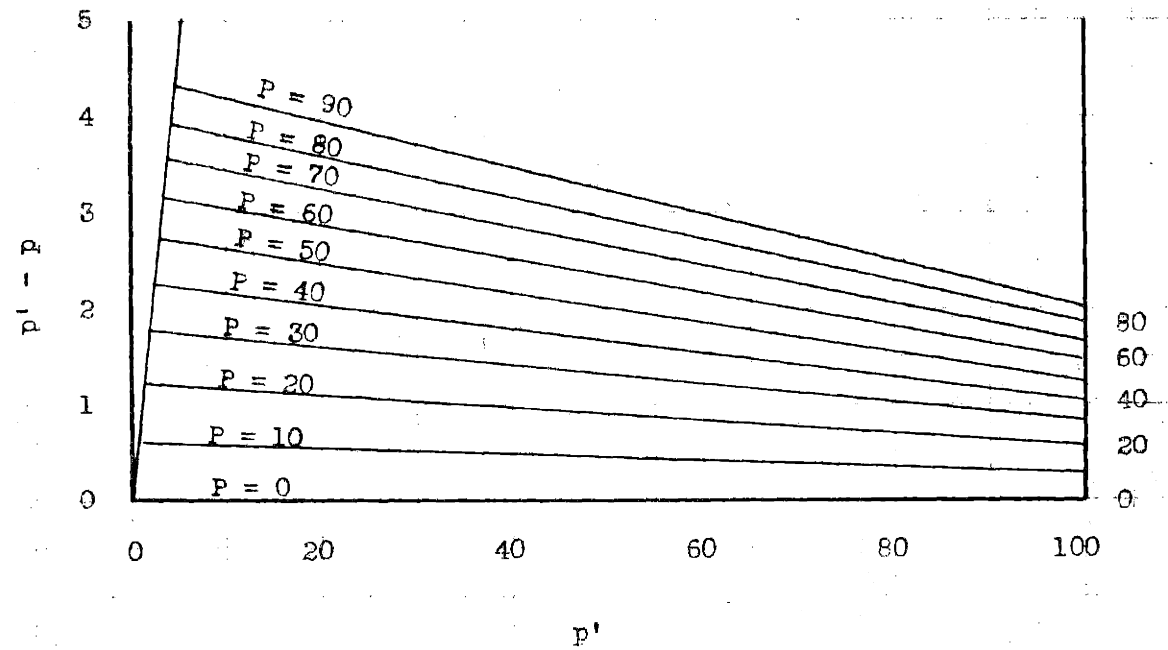


FIGURE (XXI)* - 55
WATER ORIFICE CALIBRATION
5-13-39 *R. L. B.

Figure 56

Plot for correcting from

p' to p, for water



p = Weight % of mixture as vapor at P cm. Hg. ga.

p' = Weight % of mixture as vapor if flashed to 0 cm. Hg. ga.

Derivation of Figure (56)

Basis: 100 lbs. of water at pressure P (cm. Hg. ga.) and temperature T (^{o}F.), and containing p lbs. of vapor. Cool the mixture to 212^{o} F. by lowering the pressure adiabatically, thereby vaporizing x lbs. of water, and resulting in p' lbs. of vapor.

Specific heat of liquid = 1

Specific heat of vapor = 0.45

Latent heat of vaporization at 212^{o}F. = 970 B.t.u./lb.

(1) $970\,x = [(100-p)(1) + p(0.45)](T-212)$

(2) $p' = p + x$

Eliminating p and rearranging:

(3) $x = (100 - 0.55\,p') / [970/(T-212) - 0.55] = (100-0.55p')/\mathcal{A}(P)$

For a given value of T (or P), a plot of x vs. p' is a straight line.

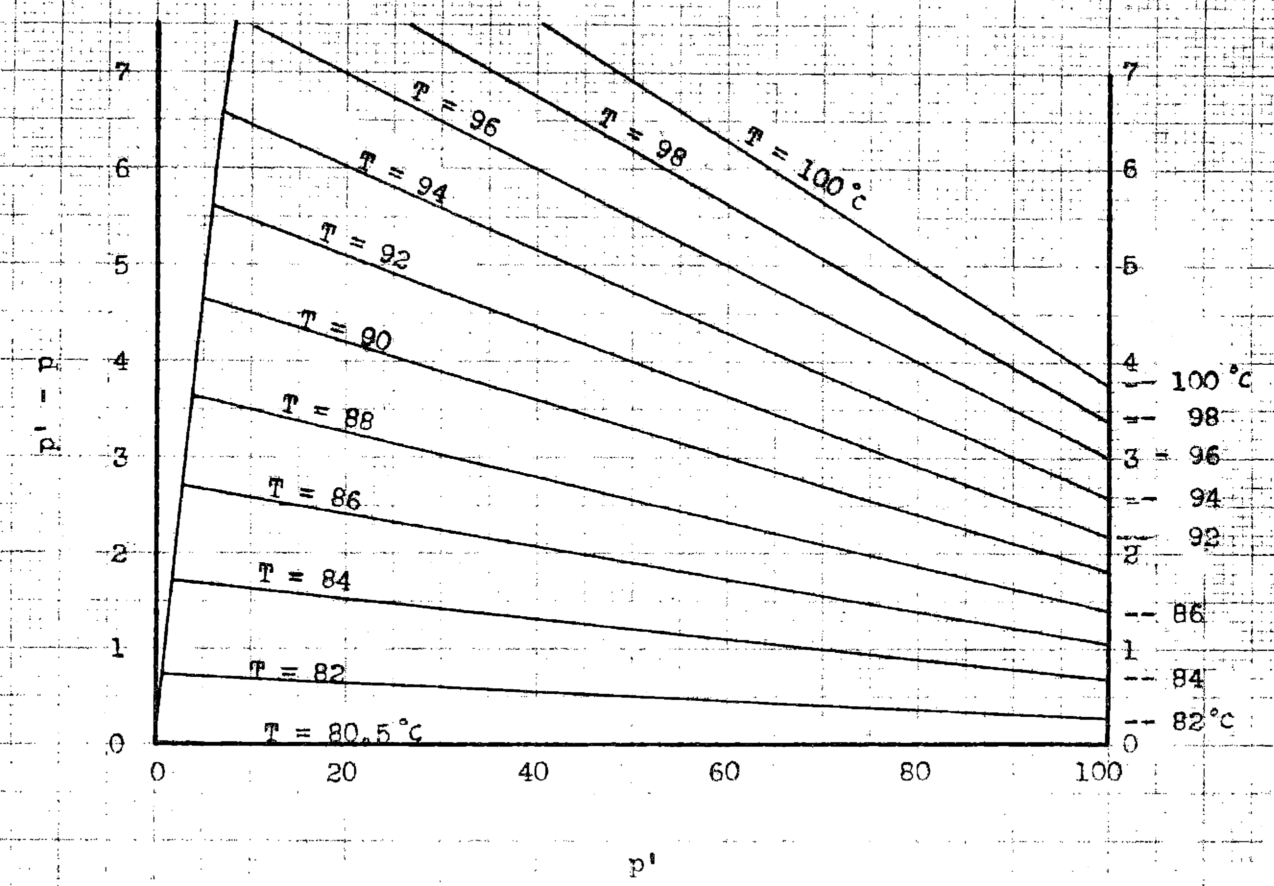
P	0	10	20	30	40	50	60	70	80	90
T	212	218.2	224.1	229.3	234.1	238.7	242.9	247.0	250.7	254.2
$\mathcal{A}$(P)	∞	156	79.5	55.5	43.3	35.8	30.9	27.2	24.6	22.4

Plot ($p' - p = x$) vs. p', showing lines of constant P.

When P = 0; $p' - p = 0$

When P = 30; $p' - p = (100 - 0.55\,p') / 55.5 = 1.8 - .99\,p'$

Figure 57
Plot for correcting from
p' to p, for benzene



Derivation of Figure 57

Basis: 100 lbs. of benzene at pressure P and temperature T °C., and containing p lbs. of vapor. Cool the mixture to 80.5°C. by lowering the pressure adiabatically, thereby vaporizing x lbs. of benzene and resulting in p' lbs. of vapor.

Specific heat of liquid benzene at 80.5°C = 0.465

Specific heat of benzene vapor at 80.5° C. = 0.290

Heat of vaporization at 80.5° C. = 169 B.t.u./lbs.

(1) $169\,x = \left[(100-p)(0.465) + p\,(0.290) \right] (T-80.5)\,(1.8)$

(2) $p' = p + x$

Eliminating p and rearranging:

(3) $x = (46.5 - 0.175\,p') \,/\, \left[93.9/(T-80.5) - 0.175 \right]$

$x = (46.5 - 0.175\,p') \,/\, \mathfrak{H}(P)$

For a given value of T, a plot of x vs. p' is a straight line.

T	80.5	82	84	86	88	90	92	94	96	98	100
$\mathfrak{H}(P)$	∞	62.4	26.6	16.9	12.5	9.70	7.99	6.77	5.83	5.18	4.63

Plot ' p' - p = x) vs. ;', showing lines of constant T.

When $T = 80.5$; $p' - p = 0$

When $T = 90$; $p' - p = (46.5 - 0.175\,p') / 9.70 = 4.80 - .0180\,p'$

TABLE OF NOMENCLATURE

A = heat transfer area (inside), sq.ft., = 0.88 sq.ft./jacket. = $\pi D N$

A,B = terms in Equation 11, page 173.

c = orifice coefficient

D = inside diameter, feet

F = friction head, feet

f = weight per cent benzene in feed

or f = Fanning friction factor

g = acceleration of gravity

H = fluid head, feet

h = film coefficient of heat transfer, B.t.u./(hr.)(sq.ft.)($^{\circ}$F.)

J = abbreviation for Jacket

M = square root of ratio of velocity of liquid phase to velocity of vapor phase

N = pipe length, feet

P = fluid pressure, cm.Hg.ga.

P_{st} = steam pressure, lbs./sq.in.ga.

p = cumulative weight per cent of feed vaporized

q = Q/θ = heat transfer, B.t.u./hr.

q/A = heat flux, B.t.u./(hr.)(sq.ft.)

R = gas low constant

R_e = Reynolds number = $D V \rho$ (μ = $4 W/\pi D \mu$)

S = cross section, sq. ft. = $\pi D^2/4$

T = t = temperature, $^{\circ}$C.

U = overall coefficient of heat transfer, B.t.u./(hr.)(sq.ft.)(^{o}F.)

V = velocity, feet per second

v = specific volume, cu. ft./lb.

W = feed rate, lbs./hr.

X, x = weight per cent benzene in liquid phase

z = viscosity, centipoise

γ = constant in Equation 11, page 173 .

μ = viscosity, lbs./(hr.)(ft.) = 2.42 z

ρ = density, lbs./cu.ft. = 1/v

Subscripts:

f refers to liquid

fl refers to fluid

o refers to orifice

p refers to pipe

st refers to steam

v refers to vapor

Superscripts:

(') specifies units of time in seconds, except for p'

p' refers to that value of p which would be obtained by adiabatically flashing to atmospheric pressure.

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A U T O B I O G R A P H Y

AUTOBIOGRAPHY

The author was born December 10, 1912 in Claremore, Oklahoma. He was educated in the elementary schools of Claremore and was raised in the Presbyterian Church. In 1926 the family moved to Kansas City, Missouri, where the author entered Southwest High School, graduating in 1930.

In 1930 the author entered Stanford University, California, graduating in 1934 with the Bachelor of Arts degree, based upon a major in chemistry. While at Stanford he became affiliated with Theta Delta Chi (social fraternity), Alpha Chi Sigma (professional chemistry fraternity), and Phi Lambda Upsilon (honorary chemistry fraternity). During his fourth year at Stanford he served, on appointment by the dean of students, as a senior sponsor in the freshman dormitory. He received a commission as second lieutenant in the Officer's Reserve Corps, Ordnance Division, and has subsequently been promoted to first lieutenant.

In 1934 the author entered the graduate school of the Department of Chemical Engineering at the Massachusetts Institute of Technology. He enrolled in the Chemical Engineering Practice School in the fall of 1935 and received the degree of Master of Science in 1936.

During the summer of 1936 the author was employed in the Technical Service Division of the Humble Oil and Refining Company, Baytown, Texas.

After serving as an assistant during the spring of 1936, the author was appointed instructor in the Department of Chemical Engineering, M.I.T. He has continued in this position to the summer of 1940, meantime pursuing a course of study leading to the degree of Doctor of Science.

The following articles have been published by the author:

Alignment Chart for Interpreting Orsat Analyses of Flue Gas, W.K.Woods, Chem. Met. Eng., 157, March (1937)

Measuring Friction at High Velocities, W. K. Woods and C. S. Robinson, Army Ordnance, 46-47, July-August (1937)