Condensate Removal by Centrifugal Force

FINAL REPORT

NASA RESEARCH GRANT NsG-59-60

For work conducted during the period
March 1, 1960 to February 28, 1963

Prepared by

K. O. Beatty jr.
Project Director

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RALEIGH, N. C.

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The purpose of this project was to investigate condensation with condensate removal by centrifugal force; specifically, condensation on a flat rotating disk. The work was carried out in the Chemical Engineering laboratories at North Carolina State of the University of North Carolina at Raleigh in cooperation with the Department of Engineering Research.

GENERAL SCOPE OF PROJECT

The project was an outgrowth of experimental work that had been done in 1957-58 in these laboratories. This work consisted of measurements of heat transfer coefficients during condensation of non-aqueous vapors on the upper surface of a 5-inch water cooled disk rotating in a horizontal plane at velocities of 400 to 2400 rpm. Vapors used were methanol, ethanol and refrigerant 113. Heat transfer rates were determined by direct collection of condensate. Coefficients were calculated using surface temperatures measured by thermocouples in the disk surface connected to slip rings on the shaft. Results of this work appear in the published literature (1) and as a doctoral thesis (2).

In general, data had shown reasonable agreement with the predictions from simple Nusselt-type laminar film theory of condensation. Coefficients increased about as the 0.42 power of the rotational speed, i.e., slightly less than the 0.5 power predicted by theory. Measured values of the coefficients had magnitudes about 70% of the predicted values. This meant that coefficients several times those experienced in ordinary gravity-flow condensers could be obtained at rotational speeds of about 2000 rpm. It appears that significantly higher values could be obtained with the higher rotational speeds readily obtained with common rotating machinery. The potential value of such high coefficients in terrestial applications was clear. The possible use of rotational disk condensers in space applications was also obvious since gravity-flow condensers will not function properly under these conditions.

The discrepancy between observed and predicted values for coefficients, while not very large, was sufficient to indicate clearly that present theory was inadequate. This was particularly significant in extrapolations to higher speeds in view of the difference between the observed and predicted exponents on the rotational velocity term. Further experimentation at higher speeds seemed essential. It also seemed essential that experimental variation of some other parameters be made to help in propounding a sounder theoretical basis.

It was, therefore, proposed to build a rotating disk condenser that

(a) would permit direct measurement of the disk surface temperature, vapor temperature, and condensation rate, i.e., provide data for direct calculation of the average heat transfer coefficient on the disk surface,

(b) could operate to 10,000 rpm,

(c) would permit use of disks of several diameters,

(d) could be operated with the vapor chamber at sub-atmospheric or super-atmospheric pressure,

(e) could be operated sealed in closed cycle to permit use of mixtures of condensable and non-condensable gases as well as of pure vapors.

With an apparatus meeting these specifications it is possible to determine the effects of independent variations of rotational speed, fluid properties, disk geometry, temperature difference, gravitational vector, vapor velocity, and diffusional resistance.

Design, construction and calibration of this apparatus were major items in this project. A considerable quantity of data were taken but since the request for extension of the project was rejected, the experimental program fell far short of exploiting the potential of the apparatus which was built.

EXPERIMENTAL APPARATUS

Two rotating condensers were used in this work. One consisted of a rebuilt, slightly modified version of the apparatus used by Nandapurkar. This unit had a 5-inch diameter disk and a small glass covered vapor chamber. It was suitable for operation at or near atmospheric pressure only. As modified, it could be operated for moderate periods of time at 5,000 rpm but seal ring limitations dictated 2,000 rpm as the practical limit for continuous use.

A cross-section sketch of this apparatus is shown in Figure 1. Because of the glass view plate, the apparatus was particularly convenient for condensate film flow pattern studies.

The second unit was much more elaborate. It was designed with a large stainless steel vapor chamber capable of operation at vacuum to 200 psi. Rotational speed of the condenser surface could be varied from 100 to over 5,000 rpm. The shaft-type seal would permit operation to 10,000 rpm with a suitable drive pulley. The condenser disk can be removed and replaced without removal of the shaft from its housing. Disks from 5" to 10" in diameter can be accommodated and a 7-1/2" diameter disk is installed in the apparatus. A cross-section sketch of the disk is shown in Figure 2 and general photographic views of the assembled apparatus are shown in Figures 3 and 4.

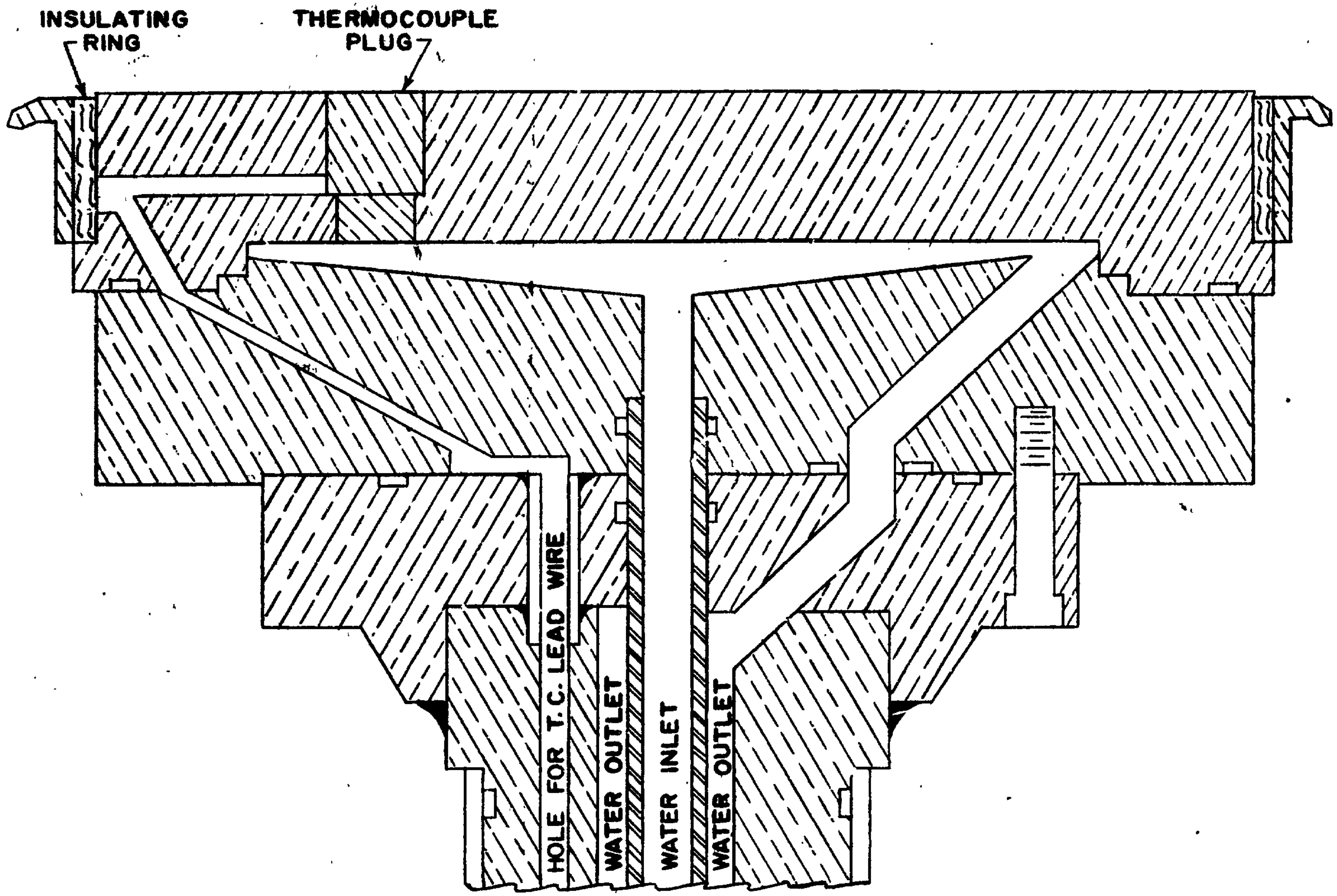


FIGURE 2: CROSS-SECTION OF 7 1/2" DIAMETER DISK LARGE ROTARY CONDENSER

The vapor chamber is mounted on an A-frame with hollow trunnions through the bearings. This support design permits the chamber to be rotated 180 degrees around horizontal axis with only minor piping changes. By rotating the chamber, the plane of the condensing face of the disk may be oriented toward, away from, or at an angle to the surface of the earth. This rotation was deemed necessary in order to be sure that no gravitational effects were influencing the data. A detailed description of the apparatus and its components will be found in the Master of Science thesis of J. A. Merricks (3). Basically, this equipment meets all six of the requirements listed in the preceding section.

EXPERIMENTAL RESULTS

Several significant observations and conclusions were made during the course of this work. In addition, much data was gathered that will contribute to further understanding if it is supplemented by additional experiments and analysis of results. This report will confine itself to those results for which data already obtained form a sound basis.

The results obtained in this project fall into several areas. Since the larger condenser was not brought on stream until the third year of the project, most of the experimental data were obtained on a rebuilt version of the 5" diameter atmospheric condenser which had been used in this laboratory previously. With this condenser, runs were made with pure vapors of methanol, ethanol, refrigerant-113, and water. Runs were also made using methanol vapors with varying percentages of nitrogen or carbon dioxide as non-condensable diluents.

The results for 57 runs using pure methanol at disk rotational speeds of 140 to 5000 rpm are given in Figure 5. Results for 36 runs using ethanol, 29

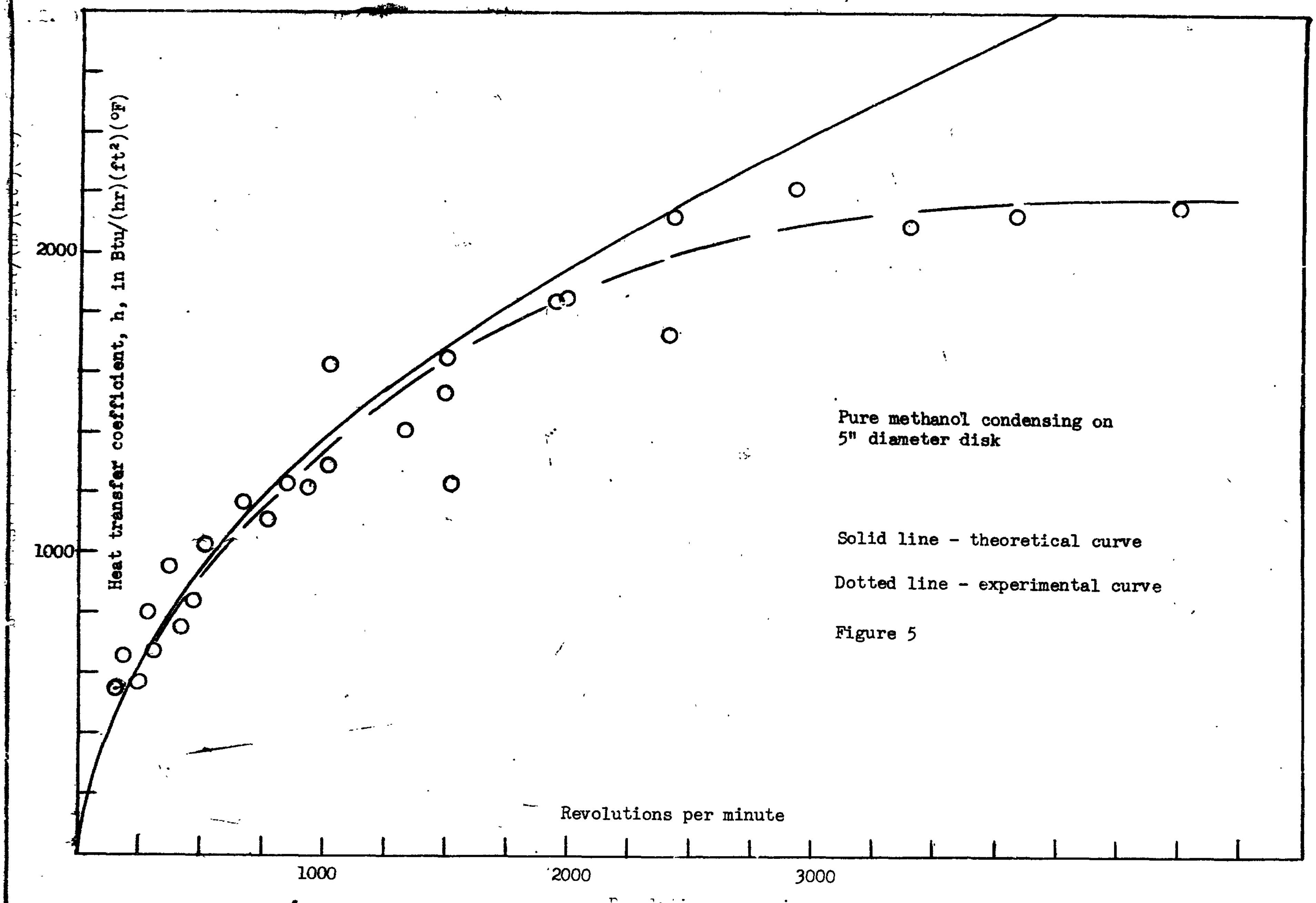

Heat transfer coefficient, h, in Btu/(hr)(ft²)(°F)
2000
1000
Revolutions per minute
1000
2000
3000
Pure methanol condensing on
5" diameter disk
Solid line - theoretical curve
Dotted line - experimental curve

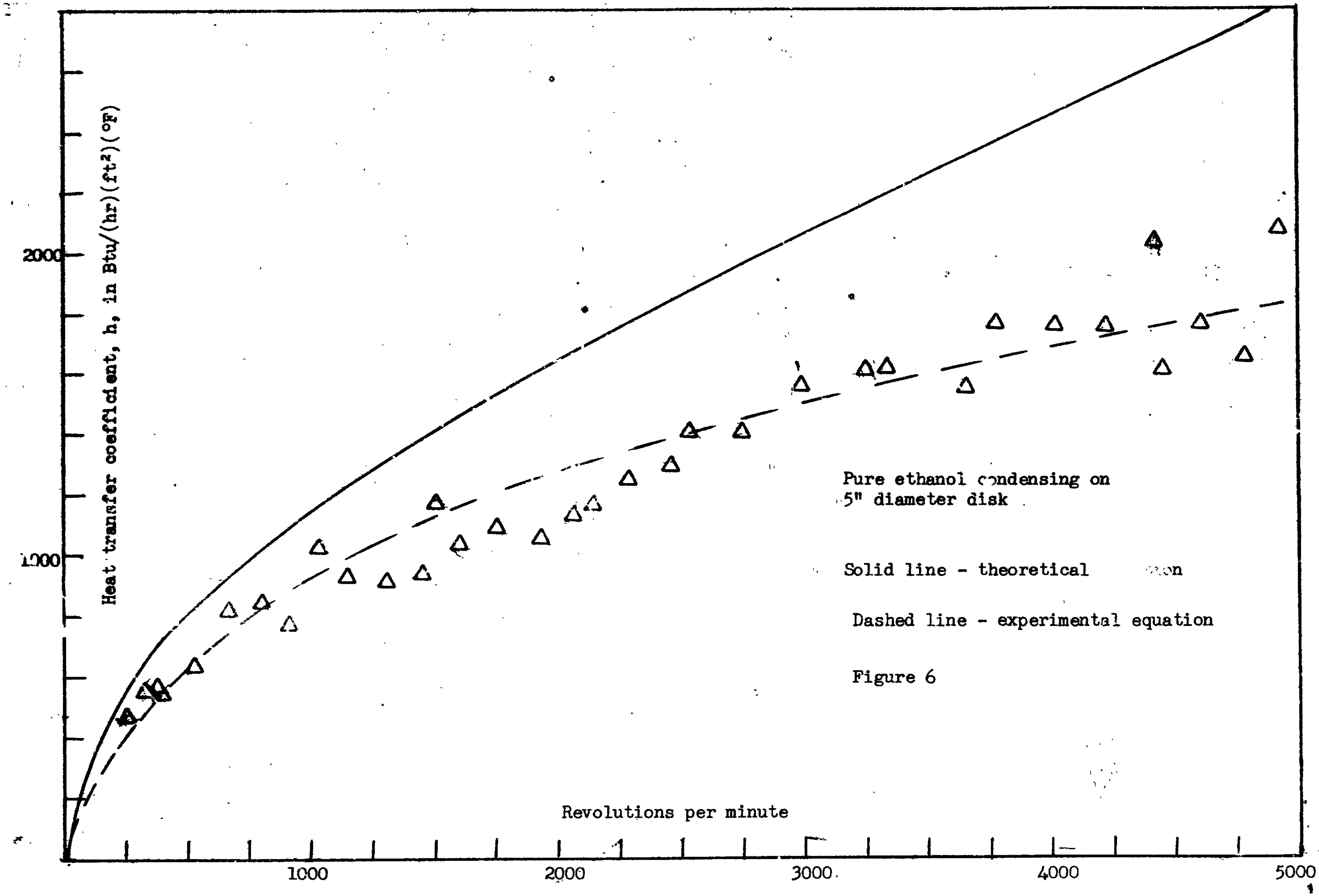
Figure 5

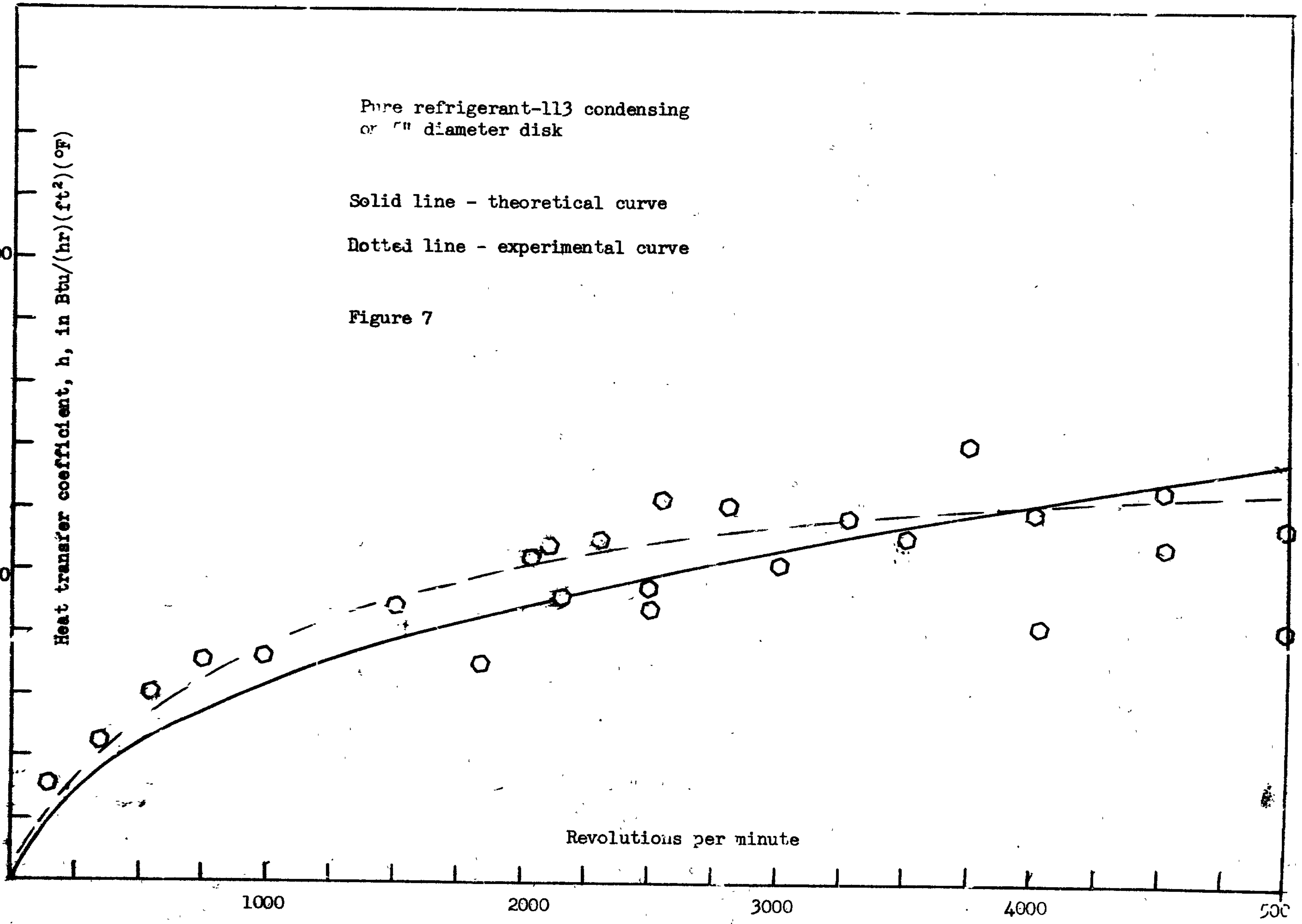
runs using refrigerant-113, and 23 runs using water vapor are given in Figures 6, 7, and 8, respectively.

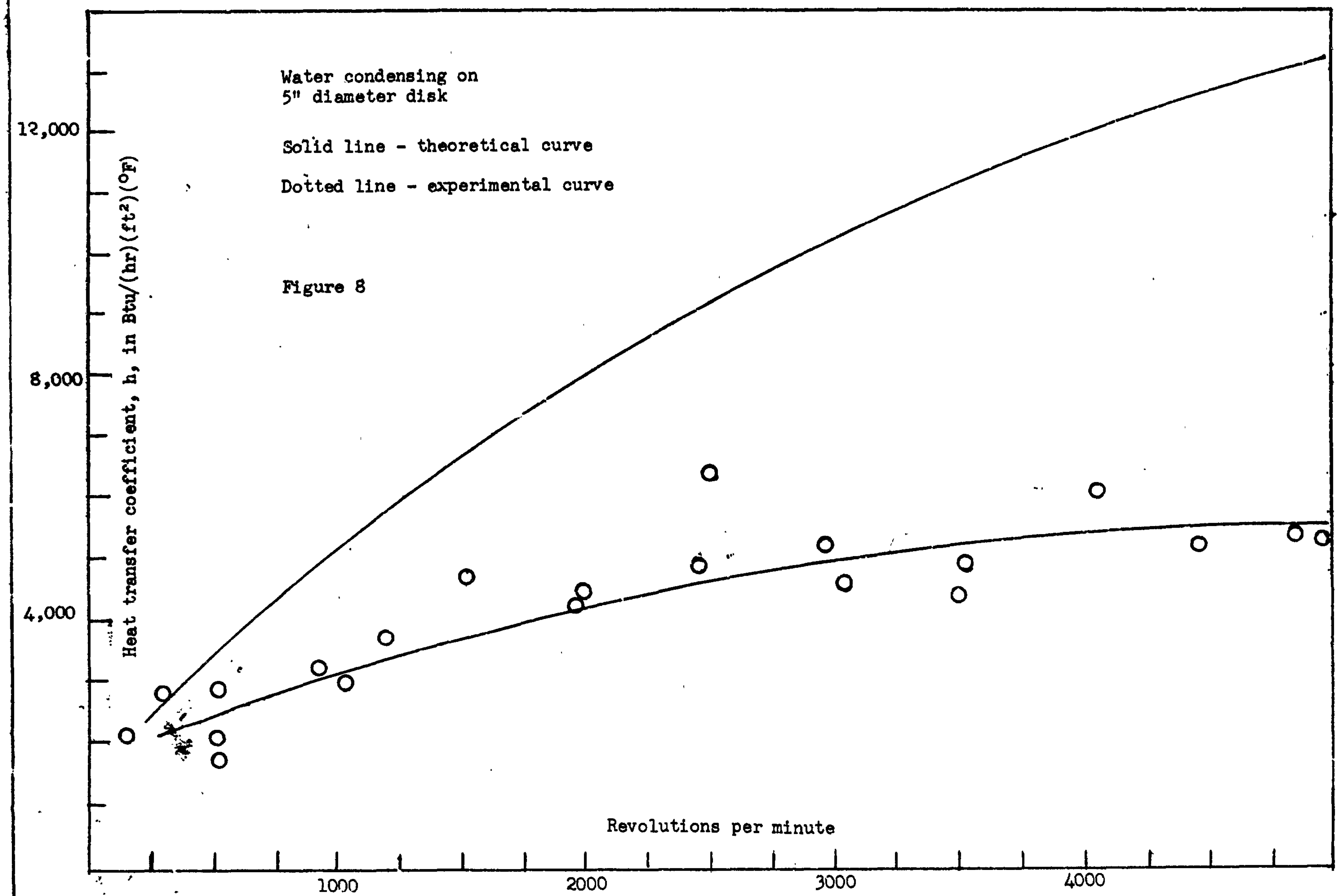
The results obtained using methanol diluted with various amounts of nitrogen are given in Figure 9, and for methanol and carbon dioxide in Figure 10. The rotational speeds for the runs with non-condensables present were limited to a maximum of 1000 rpm. This limitation was made to reduce wear and tear on the apparatus since the data showed that increasing rotatiobal velocities was having only small effects on the heat transfer coefficient.

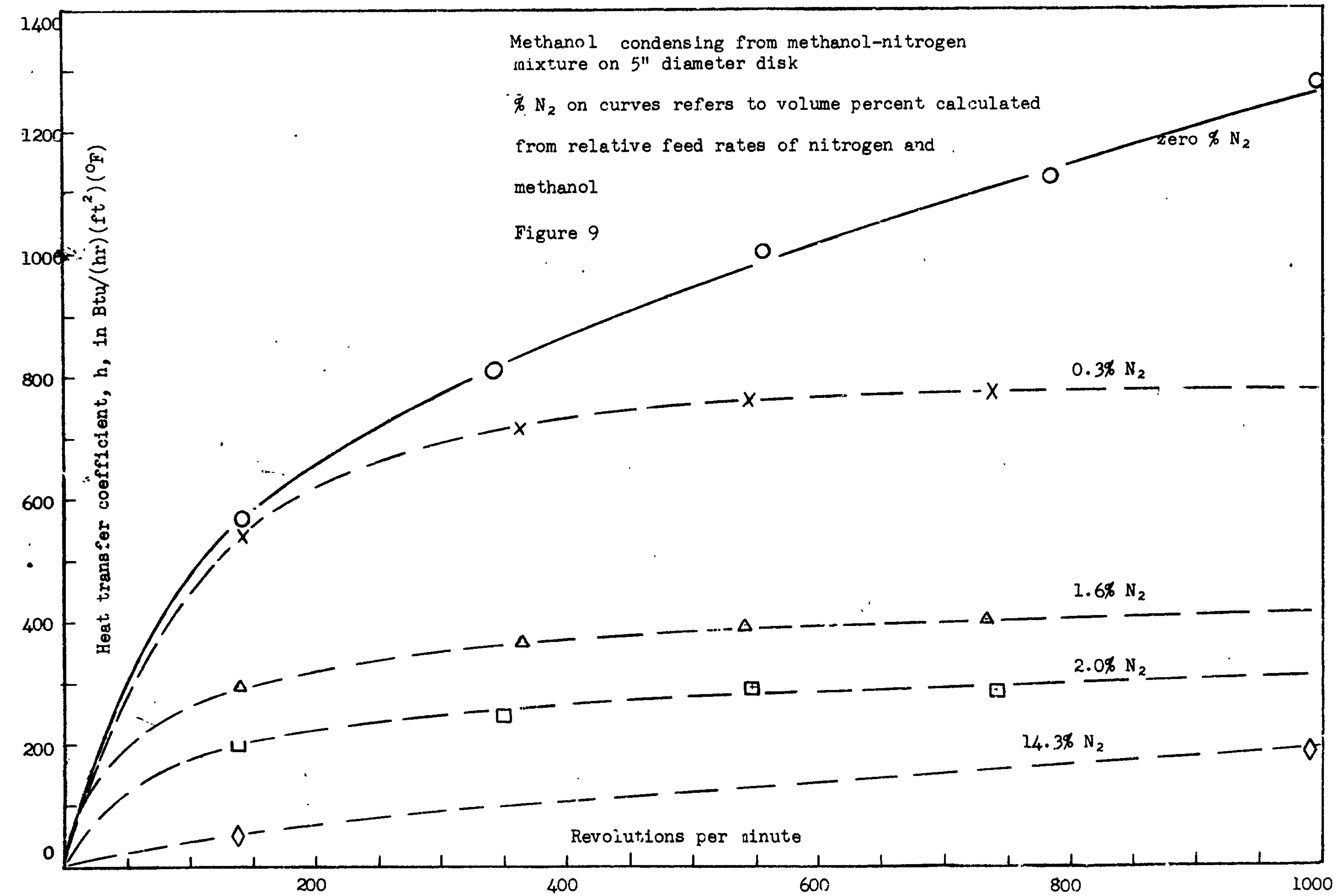
Although much data was obtained on the new condenser with the 7-1/2" disk, there seems little point to reporting the actual values here. Mechanically, the apparatus performed very well but a number of problems arose in obtaining consistent data. Reproducibility of results from day to day left much to be desired. Much of this variability was attributed to the difficulty of adequately purging the large vapor chamber of air and keeping it purged during a series of runs. The large mass of the stainless steel chamber also made it difficult to reach truly steady state operation in less than several hours. There was also some evidence that there might be some pulsating cavitation of the condenser water within the disk, particularly at the higher rotational speed. For these reasons, the data from the runs on the larger condenser are not presented here.

Basically, however, the data for the larger condenser confirm the qualitative trends shown by the smaller condenser. As rotational speeds were increased, heat transfer coefficients increased somewhat more slowly than predicted by theory and appeared to reach a maximum value in the region near 4000 to 5000 rpm. Quantitatively, even the highest heat transfer coefficients obtained with the larger condenser were often as much as twenty percent below that obtained with the smaller condenser. This was true even when purging was

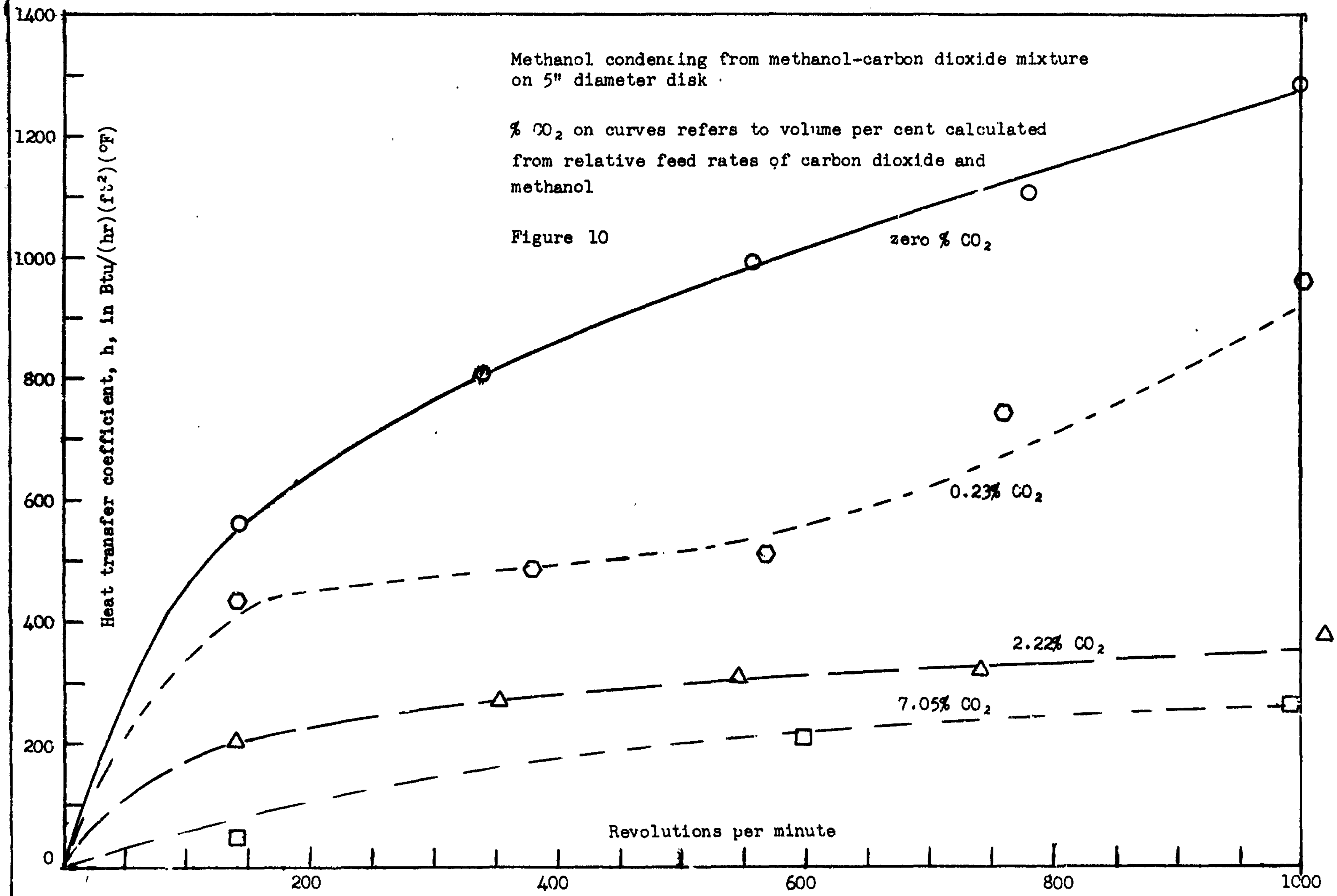

Heat transfer coefficient, h, in Btu/(hr)(ft²)(°F)
2000
1000
Revolutions per minute
1000
2000
3000
4000
5000
Pure ethanol condensing on
5" diameter disk
Solid line - theoretical
Dashed line - experimental equation
Figure 6


Pure refrigerant-113 condensing
or 5" diameter disk
Solid line - theoretical curve
Dotted line - experimental curve
Figure 7
Heat transfer coefficient, h, in Btu/(hr)(ft²)(°F)
2000
1000
Revolutions per minute
1000
2000
3000
4000
500


Water condensing on
5" diameter disk
Solid line - theoretical curve
Dotted line - experimental curve
Heat transfer coefficient, h, in Btu/(hr)(ft²)(°F)
12,000
8,000
4,000
Revolutions per minute
1000
2000
3000
4000
Figure 8



Methanol condensing from methanol-nitrogen mixture on 5" diameter disk
% N_2 on curves refers to volume percent calculated
from relative feed rates of nitrogen and
methanol
Figure 9
zero % N_2
0.3% N_2
1.6% N_2
2.0% N_2
14.3% N_2
Heat transfer coefficient, h, in Btu/(hr)(ft^2)(^{o}F)
Revolutions per minute
0
200
400
600
800
1000
1200
1400



Methanol condensing from methanol-carbon dioxide mixture on 5" diameter disk

% CO_2 on curves refers to volume per cent calculated from relative feed rates of carbon dioxide and methanol

Figure 10

carried out for extended periods of time and great care was taken to see that steady state conditions had been reached when data were being collected. A possible explanation for these lower coefficients will be given in a later section of this report.

THEORY AND DISCUSSION OF RESULTS

The simplest kind of Nusselt-type approach to rotating disk condensation assumes that the condensate is in purely radial laminar flow, that all thermal resistance is caused by conduction through the condensate layer, and that the temperature gradient is a constant throughout the layer. It further assumes that fluid properties are uniform within the condensate layer and that there are no interfacial or end effects. Such a model predicts that the velocity profile of the condensate will be parabolic with a maximum at the vapor-condensate interface.

Continuing the calculation leads to the interesting prediction that the condensate layer should have a uniform thickness over the entire surface of the disk regardless of its radial dimensions.

Even if the assumptions are modified to include the possibility of three dimensional flow (instead of purely radial), the result is the same except for very thick condensate layers. The predicted condensate layer thickness, δ , in terms of the operating parameters is given by Equation (1):

$$\delta = \left(\frac{3\,k\nu\Delta}{2\,\lambda\,\rho}\right)^{1/4} / (\omega)^{1/2} \qquad (1)$$

where k = thermal conductivity of condensate

ν = kinematic viscosity of condensate

ρ = density o condensate

λ = latent heat of vaporization of condensate

Δ = saturation temperature of vapor minus surface temperature of disk

ω = angular rate of rotation of disk

By the assumptions of this theory, the heat transfer coefficient, h, is given by Equation (2)

$$h = \frac{k}{\delta} = 0.904 \left(\frac{k^3 \lambda \rho}{\nu \Delta}\right)^{1/4} (\omega)^{1/2} \qquad (2)$$

As has been mentioned, the Nusselt-type theory has been modified to allow for tangential and axial velocity components for the condensate and to take into account the effect of vapor drag on a smooth condensate surface (4, 5). These modifications do not alter predicted values of coefficients by more than a few per cent under normal operating conditions. Under the conditions where significant deviations are predicted to occur, several other factors not accounted for even in this modified theory would be expected to make important contribution to the values of the heat transfer coefficient. It appears then that the disqrepancy between observed and predicted values of heat transfer must lie in some major error, or errors, in the assumptions underlying the theory. From data obtained on this project it is believed that a major error results from the assumption of a flat vapor-condensate interface.

Visual and photographic evidence have shown conclusively that with disks of 5 inches or greater in diameter and at rotational speeds of 400 rpm or greater, there are surface waves in the condensate layer. The condensate layer thicknesses predicted by theory are of the order of 0.0015 inches at 500 rpm to 0.0005 inches at 5000 rpm. Measured values of the heat transfer coefficient indicate that the actual average thicknesses must be approximately of this magnitude. In order to make any waves in such thin film visible, it is necessary to illuminate the film at a small grazing angle. The photographs in Figures 11 and 12 show clearly that very complex wave patterns, basically circumferential in nature, do indeed exist in the film. The waves amplitudes are certainly of a magnitude comparable to the predicted film thickness and they cover appreciable areas of the disk surface. Consequently, it must be looked on as almost happenstance that a theory which ignores these waves predicts coefficients in reasonable agreement with the observed values.

It is a matter then of both theoretical and empirical concern to determine the nature and the origin of these waves. A series of experiments using methanol, ethanol and refrigerant-113 established that the waves did not cover the entire surface of the disk but existed only in the annular area beyond some critical radius, r_c. The value of this critical radius was shown to decrease with increasing rotational speed and to be somewhat dependent on the fluid that was being condensed (6). As a result of observations with the three fluids mentioned above and at rotational speeds over the range of 400 to 5,000 rpm, it was found that the critical radius could be correlated within $\pm$ 5% by the simple equation

$$\frac{r_c^2\,\omega}{\nu} = 280{,}000 \qquad (3)$$

where ν is the kinematic viscosity of the condensate.

Alternatively, it may be stated that waves will be formed on the disk over those areas where the radius r meets the requirement that

$$\frac{r^2\,\omega}{\nu} \geq 280{,}000 \qquad (4)$$

This correlation suggests that wave formation is the result of exceeding a critical value of some type of Reynolds number.

In gravity-flow condensation, it has been shown that turbulence arises in the condensate layer when its Reynolds number, based on layer thickness, exceeds 2100. This same Reynolds number calculated from the layer thickness when waves are observed to appear on the rotating disk condenser has a value of less than 10. This hardly seems adequate for any type of laminar-turbulent transition. The absence of turbulence in the condensate layer was also confirmed by the introduction of small amounts of dye into the layer. (It should also be noted that turbulence per se would be expected to enhance the heat transfer coefficient not reduce it.)

If the origin of waves is not in the condensate film itself, it seems probable that it must lie in the vapor or in interaction between the vapor and the condensate layer. A critical vapor Reynolds number may be calculated by using the vapor viscosities in place of the condensate viscosity in the left hand side of equation (4). When this is done it is found that the criterion for wave formation is that

$$\frac{r_c^2\,\omega}{\nu_g} = 26{,}000 \qquad (5)$$

where ν_g is the kinematic viscosity of the vapor.

This value of 26,000 is slightly less than 0.1 of the 300,000 figure quoted by Schlichting (7) for the transition Reynolds number on a rotating disk. However, the figure given by Schlichting is based on the outer radius of the disk and no indication is given of what portion of the flow over the disk is turbulent. Furthermore, Schlichting's value is based on a single homogeneous fluid flowing across the solid surface of the disk with zero velocity components in all three directions of the disk surface. In the disk condenser, the vapor-condensate interface has a significant radial component of velocity and, because of the condensation, there is a finite vapor velocity component normal to this interface.

It seems reasonable then to propose the hypothesis that the <u>waves in the condensate layer are produced by turbulence arising in the vapor boundary layer</u> when some critical Reynolds number is exceeded. It is to be expected that waves produced in this manner would reduce the mean radial velocity of the liquid layer and, hence, increase its average thickness. An increase of a few ten-thousandths of an inch in condensate layer thickness would be sufficient to account for the reduced values of coefficient observed. Since this turbulence would increase with increasing rotational speeds, this hypothesis is consistent with the failure of coefficients to increase with rotational speed as rapidly as predicted by Nusselt-type theory.

In waves of small radius of curvature, <u>interfacial tension will play an important part</u>. With the rotational plane of the disk parallel to the earth's surface, gravitational provides the principal force component interacting with interfacial tensions. Despite the enormous radial forces associated with the rotation, these much smaller forces acting normal to the disk have a profound influence on the condensate layer behavior. This is brought out dramatically in the photograph of Figure 13 showing water condensing on a disk rotating at 900 rpm. Under these conditions, the mean radial acceleration is approximately 10 g on this disk. Despite this, it is seen that the flow paths for the condensate follow a maze-like pattern determined in large part by the interfacial tensions among the three phases - vapor, liquid and solid surface.

The Nusselt-type theory previously referred to <u>predicts that the heat transfer coefficient for condensation on a rotating disk should increase as the square root of the rotational speed</u>. Modification of the theory to take into account the non-radial components of condensate velocity do not alter this prediction. The equations for gravity-flow and centrifugal flow are quite similar. Basically the factor g/L in the gravity flow equation is replaced by ω Thus, the coefficient for one foot high vertical surface draining by gravity would be matched by a disk rotating at a speed of 32.2 radians/sec or about 50 rpm. On this basis the coefficient at 5,000 rpm ought to be only about 1/ 0 of this value.

Experimental data, however, are not in agreement with this prediction. The measured values of heat transfer coefficients given in Figures 5, 6, and 7 show that after a rotational speed of about 3,000 rpm is reached, further speed increases have little effect. In fact, there is some indication that increases of speed above 5,000 rpm may cause some decrease in the coefficient. Although the results on the 5-inch disk condenser (i.e. with the small vapor chamber) differ somewhat in magnitude from those with the 7-1/2 inch disk (i.e. with the large vapor chamber apparatus), both sets of data show this characteristic of leveling off in the 4,000 to 5,000 rpm range.

This result is not what would have been anticipated from experience with gravity-flow condensers. In gravity-flow condensers it has been shown that heat transfer coefficients in condensation run consistently higher than predicted values because of three common effects. First, the existence of ripples results in a decrease in the mean condensate layer thickness at a given condensing rate. Second, turbulence and agitation in the condensate layer improve heat transfer over that to be expected with rectilinear laminar flow. Third, when concurrent flow of vapor and condensate exist, the vapor drag thins the condensate layer and improves the heat transfer coefficient.

In the rotating disk condenser, increased rotational speeds produce larger numbers of ripples, give some agitation in the condensate layer, and increase the outward radial component of the vapor velocity. Clearly each of these effects would normally be expected to make the actual coefficient exceed the value that is predicted by Nusselt's theory. The extent of such augmentation should increase as the rotational speed increases but experiment shows just the opposite. The answer to this anomaly would appear to lie primarily in the interactions of the vapor and condensate resulting from non-radial velocity components in the vapor. It seems quite probable that interfacial tension may play a secondary but significant role in the condensate flow patterns. The complexities of predicting the film instabilities for two phase flows have been reviewed by Ostrach and Koestel (8). These authors discuss specifically the problem of instabilities in condensing flows and mention four types of instability as "clearly possible" under these conditions.

The consistently low heat transfer coefficients observed with the new larger condenser may well be associated with vapor-condensate interaction effects. In the small (5", atmospheric pressure) unit, the vapor seal is around the periphery of the disk itself. The vapor chamber is roughly 9 inches in diameter by 3 inches high and the upper surface of the chamber is not parallel (it slopes from vapor inlet down to vapor outlet) to the disk face. In the large unit, the seal is around the drive shaft and a portion of the bottom of the disk as well as the top surface is exposed. Although an annular condensate collection trough surrounds the disk, the inside of the vapor chamber is nearly three times the seven and one-half inch diameter of the disk The top of the vapor chamber is dome shaped and rises to about twenty inches above the disk surface.

Quite clearly the vapor flow patterns induced by the disk rotation could be significantly different in the condensers. It has been proposed in this report that vapor-condensate interface effects play a major role in the deviation between experiment and simple theory. If so, then, these changes in the vapor chamber dimensions may be a significant factor in the observed differences in coefficients obtained in the two condensers. In any event, it must be concluded that for some time, design will have to be based on experiment but theoretical considerations will be essential in deciding on the proper course of experimentation.

In gravity-flow condensation it has been established that even small amounts of non-condensables materially reduce the heat transfer coefficient. For example, Akers et al. (9) found that 9% carbon dioxide in ethanol gave a coefficient of 110 as compared to a value of 800 for condensing pure ethanol vapor at the same heat flux and in the same apparatus. This effect is attributed to accumulation of a film of non-condensable gas on the surface of the condensate. The condensing vapor must then diffuse through this film. The diffusion resistance of this gas film effectively reduces the rate of condensation under a given temperature difference driving force. It has been found that natural convection forces associated with the density difference between the vapor and the gas materially influence the effective gas film thickness. Thus, helium, molecular weight 4, has less effect on the condensation coefficient of ethanol, molecular weight 46, than does carbon dioxide, molecular weight 44.

With a rotating condenser, it would be expected that the motion of the non-condensable gas film would be controlled by the rotational speed and the relative density of vapor and gas. High density gases would flow continuously off the disk edge while low density gases would tend to move inward over the disk surface. This is both an interesting and important phenomenon. There is almost always some non-condensable gas present in any vapor system. In the closed recycle systems, such gases can accumulate as a result of degassing of the materials of construction, chemical reactions, leaks (in vacuum systems), and other processes which liberate gas into the vapor chamber. Where purging is not convenient or possible (e.g. toxic vapors, use on a space vehicle, completely sealed system) an accurate knowledge of the effect of such accumulation and proper design to cope with it are vital.

The data gathered on the effects of non-condensables on rotating condenser performance was very limited. Even in a sealed system, it is not easy to keep the non-condensable gas concentration constant in the main vapor chamber. The results obtained were all on the effect of nitrogen and of carbon dioxide on condensation of methanol. The density of nitrogen is about 10% less than that of methanol so it might be expected that the nitrogen would tend to accumulate on the disk as a result of the centrifugal forces. The density of carbon dioxide is about 40% greater than that of methanol and it should be thrown off the disk. The small amount of data available support this concept. As little as 0.3% N_2 had significant effect on the coefficient and over the range of 400 to 1000 rpm, the coefficient remained nearly constant. With carbon dioxide, however, Figure 10 shows the initial effect is the same as for nitrogen but as the rotational speed increases there is some tendency for coefficients to improve.

The concentration of the non-condensables in the gas film is, of course, higher than the average concentration of these gases in the vapor. If this gas film moves across the disk surface, as has been proposed here, it might be possible to collect this gas enriched vapor and thus remove the gas without need of extensive purging. For this purpose, a rotating disk condenser might be used in a vapor system even though the principal condenser surface was of some other type.

IN CONCLUSION

The rotating disk condenser has been shown to provide very effective condenser surface even at relatively low rotational speeds. As the speed is increased above about 3000 rpm, however, little, if any, enhancement of coefficient is obtained. This is inconsistent with theory which predicts that coefficients should increase as the square root of the rotational speed. There is a real need for both theoretical and experimental work in this area in order that the potential of this type of condenser may be more fully realized.

The additional independent control variable offered by centrifugal force as compared to dependence on gravity alone makes the rotating disk condenser a much more flexible and useful unit for many purposes than more conventional condensers. Even a moderate amount of engineering effort could lead to the design of very simple, effective, and convenient rotating disk condensers.

K. O. Beatty jr.

December 9, 1964

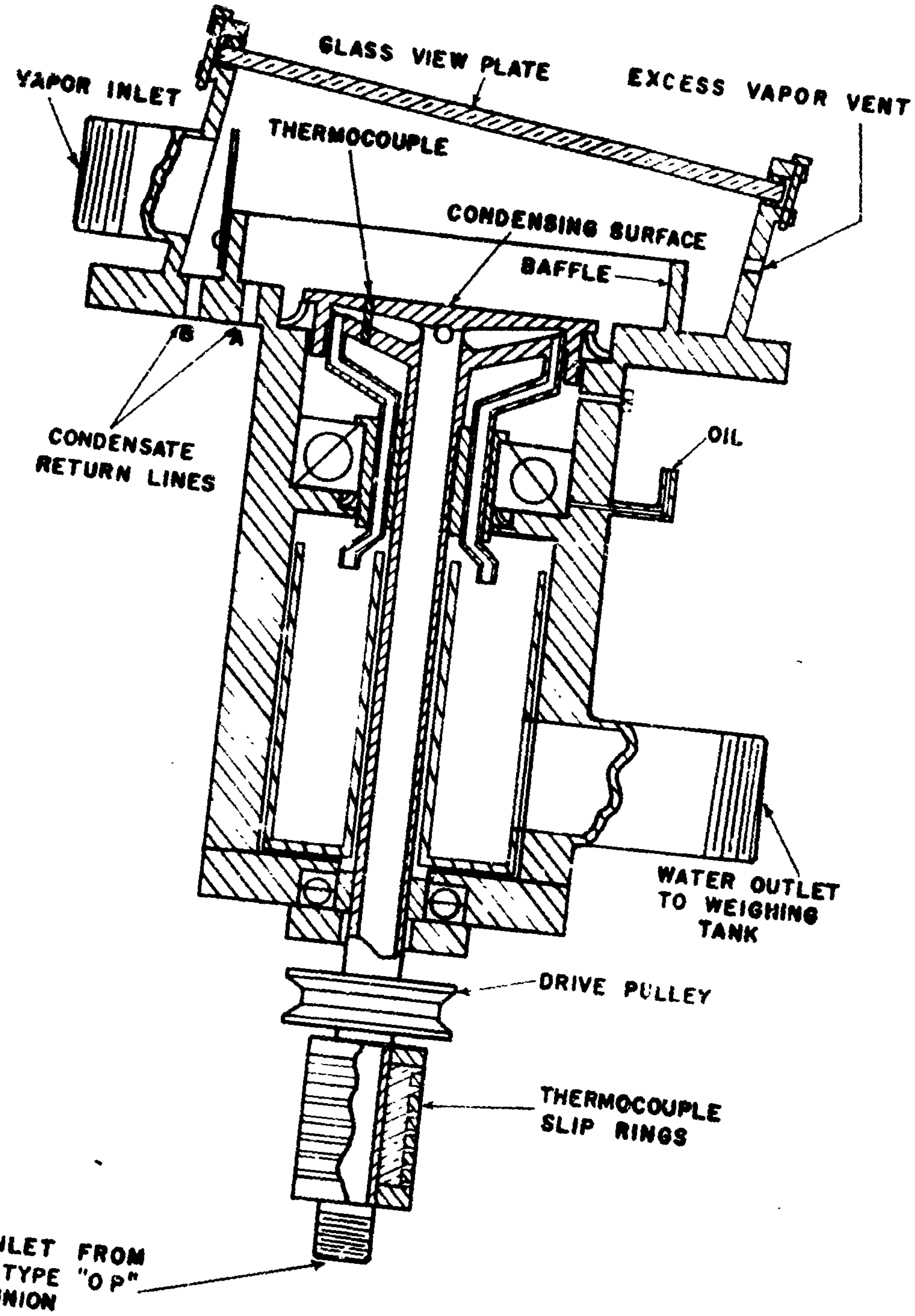
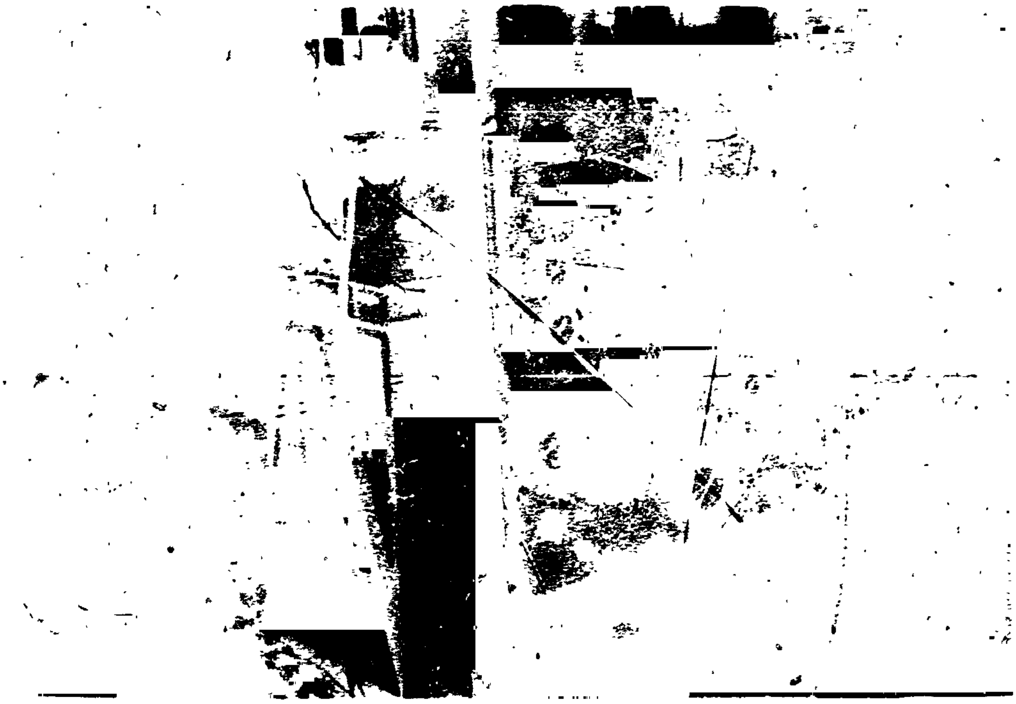


FIGURE I
SECTION VIEW OF ROTATING CONDENSER
(Principal Mechanical Features)



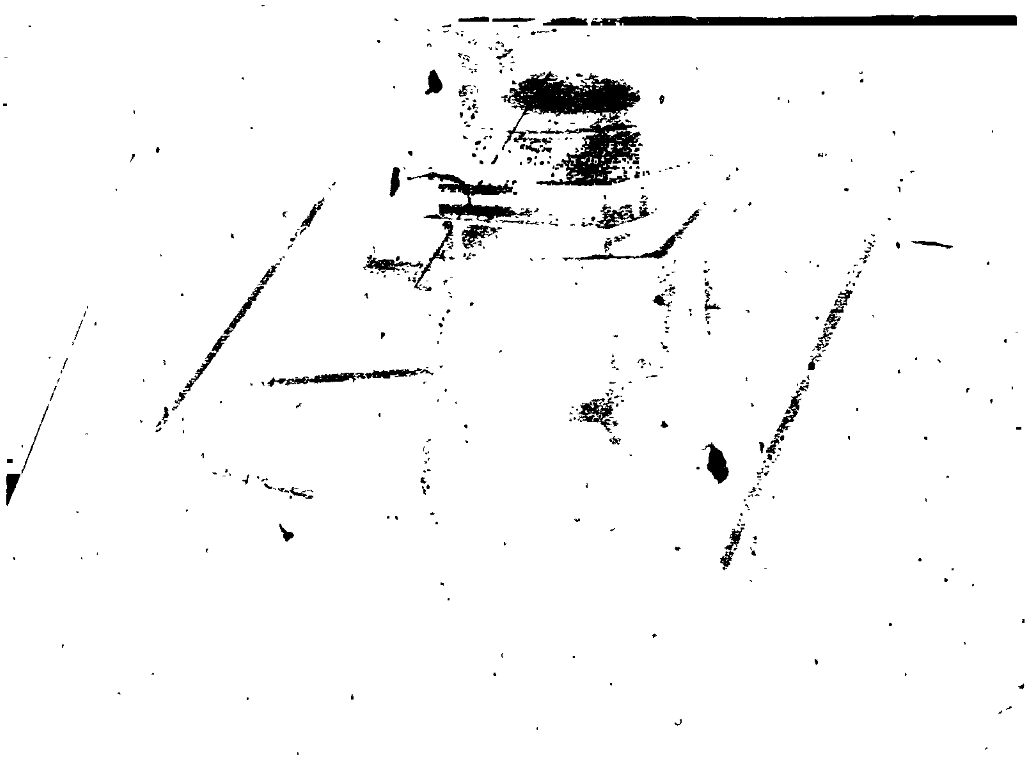


FIGURE 3: VIEWS OF ASSEMBLED STAINLESS STEEL CONDENSER

Top Picture: Front view showing control panel.

Lower Picture: Review showing piping and boiler in lower foreground.

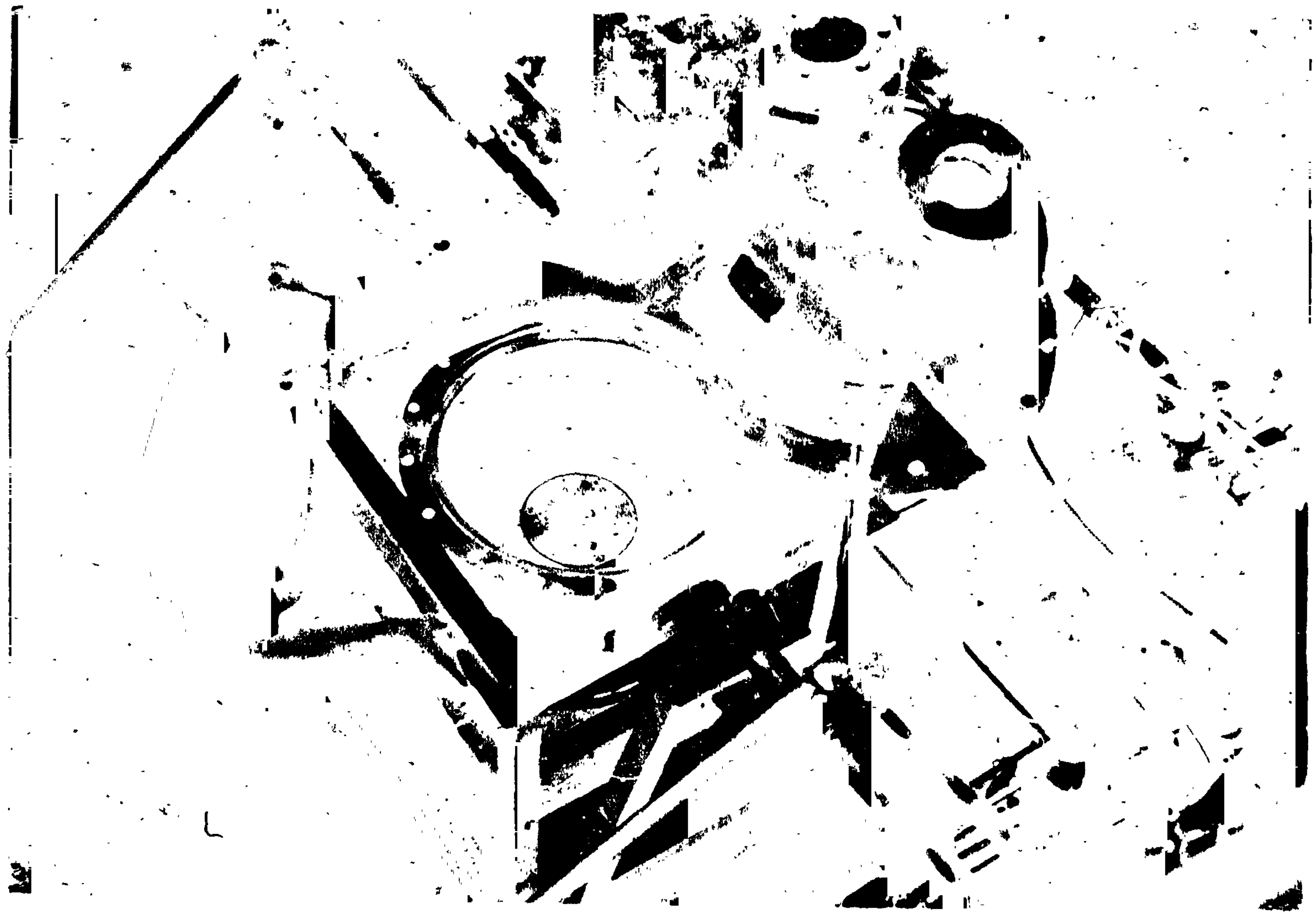


FIGURE 4: View into vapor chamber of stainless steel unit showing 7 1/2" condenser disk.

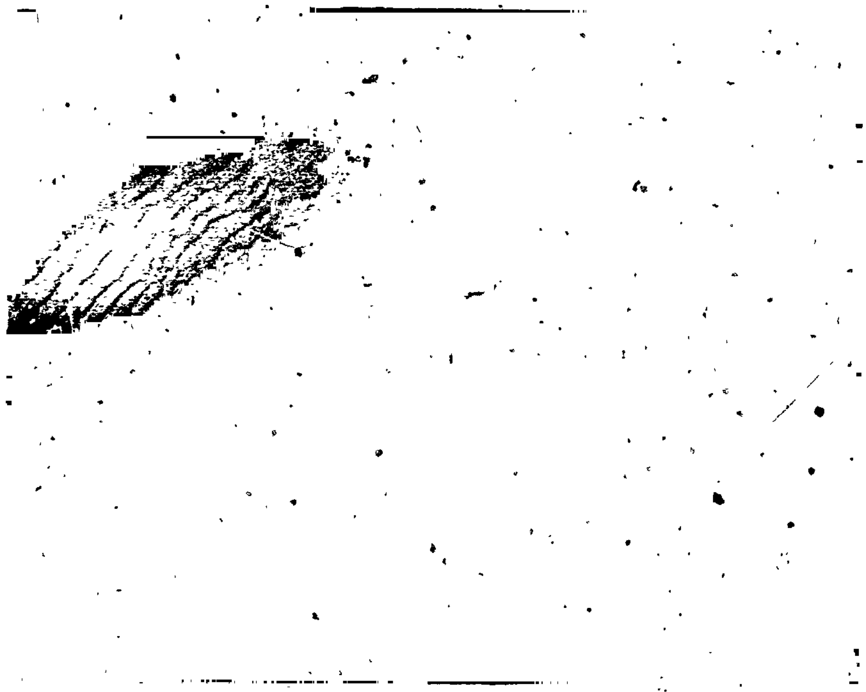


FIGURE 11: Methanol condensing on horizontal disk rotating at 1020 r.p.m. (About 1 1/4X)

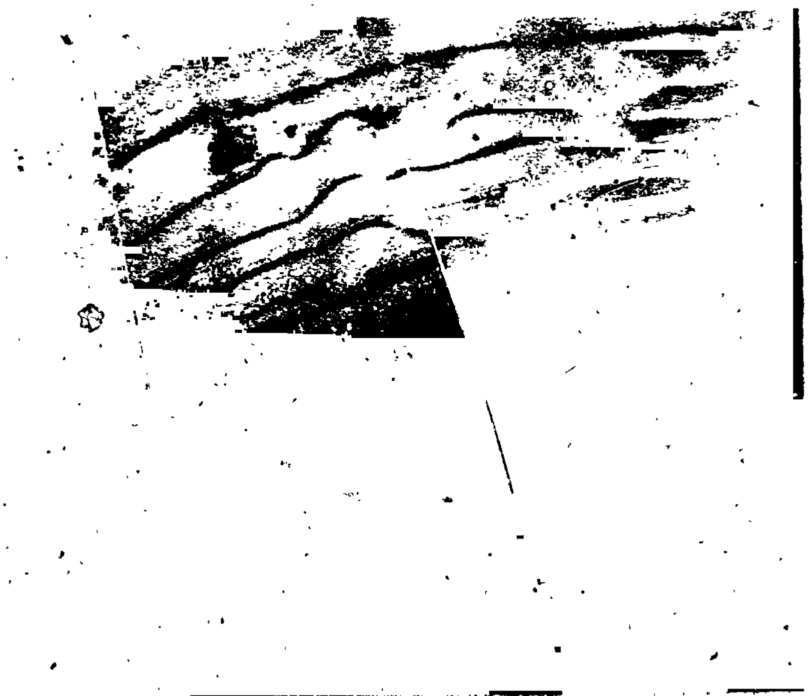


FIGURE 12: Close up of ripples in condensate film of methanol on rotating disk (About 3X)

FIGURE 13: Water condensing on horizontal disk rotating at 900 r.p.m. (About 2X)

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