THE DETERMINATION OF A WATER FILM COEFFICIENT AND A CONDENSING STEAM FILM COEFFICIENT FOR A SINGLE TUBE HEAT EXCHANGER

by

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I. INTRODUCTION

Methods of calculating heat transfer rates between condensing steam and water, separated by metal pipe or tubing, are extremely important to any engineer interested in the field of heat transfer. This is vital knowledge in the design of surface condensers and feedwater heaters.

In this type equipment, the temperature drop across the tube thickness is small and thus difficult to measure accurately. In order to determine each individual resistance to heat transfer by the usual methods, it is necessary to determine very accurately the metal temperature on the steam side and the metal temperature on the water side. For this reason, only overall coefficients of heat transfer are usually calculated. The engineer should keep in mind however, that the overall resistance to heat transfer is numerically equal to the sum of the individual series resistances. The determination of these individual resistances is the subject with which this thesis deals. In a clean tube these are:

1) Condensing steam film resistance
2) Tube resistance
3) Water film resistance

In addition to these three, a dirty tube would contain a fourth item, scale deposit resistance. This scale may be located inside the tube due to impure cooling water, or may be located on the external portion due to oil or other impurities in the steam. In this thesis, new, clean pipe was used to eliminate this fourth variable.

All three of the previously mentioned resistances will vary somewhat with water velocity. Nusselt's theoretical equation shows that steam film resistance varies slightly with water velocity.[1] It is also true that the thermal resistance of metal varies with temperature, and thus the tube resistance would vary slightly as water velocity is varied since the tube temperature would change. It is an established fact that water film resistance is a function of water velocity. From the foregoing, it seems that when water velocity varies in a clean pipe or tube, there are three variables affecting the overall heat transfer coefficient. Actually this is true, but it can be noted that except in the cases of very high water velocity, the water film resistance is many times greater than the sum of the other two resistances. Due to this fact, and since these smaller resistances are only slightly affected by water velocity, it seems reasonable to assume that the sum of these two small resistances may be assumed constant throughout the range of water velocities used in this thesis. Therefore, the results will be based on the premise that only water film resistance varies as water velocity is varied.

From the basic heat transfer laws it can be stated that the reciprocal of the overall heat transfer coefficient is equal to the tube metal resistance plus the steam film resistance plus the water film resistance, or in equation form:

(1) $1/U = L/k(2D_o/D_o+D_i) + 1/h_s + 1/m(V_w)^{0.8}$, where:

U = Overall heat transfer coefficient, Btu/hr-ft^2-F

h_s = Steam film coefficient, Btu/hr-ft^2-F

V_w = Average water velocity, ft/sec

1. McAdams, W. H. , Heat Transmission, Pg. 261, Mc-Graw-Hill Book Company, New York, 1942

L = Tube thickness, feet

k = Thermal resitance of the tube metal, Btu/hr-ft^2-F

D_o = Outside diameter of pipe, inches

D_i = Inside diameter of pipe, inches

m = empirical constant

Since the thermal resistances of the various metals used for heat exchanger tubes have been determined and verified many times, no attempt was made to determine these values in this thesis.

The graphical method which was used consisted of plotting $1/U$ as ordinates versus $1/V_w^{0.8}$, and drawing the best resulting straight line. It can be seen from equation (1) that when $1/V_w^{0.8}$ equals zero, $1/U$ represents the sum of the metal resistance and the steam film resistance. This value of $1/U$ was determined by extending the straight line until $1/V_w^{0.8}$ equaled zero. Since the metal resistance was known, the steam film resistance was determined by subtracting the metal resistance from this value of $1/U$. The steam film coefficient could then be determined since it is the reciprocal of the steam film resistance.

Equation (1) may now be rewritten:

(1a) $\quad 1/U = a + 1/m(V_w)^{0.8}$, where:

a = The intercept of the straight line on the $1/U$ axis when $1/V_w^{0.8}$ equals zero. It also equals the sum of the steam film and metal resistances.

Other terms are as previously defined

It is seen from this equation, that the empieical constant "m" is the reciprocal slope of the straight line. This slope was measured and "m" evaluated. Since the water film coefficient is the reciprocal of the water film resistance, the water film coefficient may be determined at any velocity by substitution in the equation $h_w = m(V_w)^{0.8}$.

II. REVIEW OF LITERATURE

When condensation of a vapor occurs, the film coefficient depends upon whether the condensation is dropwise or filmwise. With dropwise condensation, the condensate collects on the surface of the tube or pipe in drops; with filmwise condensation, the condensate forms a continuous film over the entire surface. Film coefficients for dropwise condensation are usually four to eight times as high as for those for filmwise condensation. The latter type is usually more likely to occur in practice, since dropwise condensation usually takes place only on polished surfaces.[1]

The film coefficient is also affected by the velocity of the vapor. If the vapor strikes the tube surface with a velocity sufficient to sweep away part of the condensate film the thickness of the film is reduced and the film coefficient is increased.

There has been considerable research on the determination of steam film and water film coefficients.

It is very difficult to obtain a satisfactory law for the condensing steam film coefficient.[2] Many experimenters have reported steam film

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1. Stoever, H. J. , Applied Heat Transmission, Pg. 40, McGraw-Hill Book Company, New York, 1941
2. Gaffert, G. A. , Steam Power Stations, Pg. 86, McGraw Hill Book Company, New York, 1946

coefficients ranging from approximately 1000 to 4000 Btu/hr-ft^2-F.[1] An average design figure is 2000 Btu/hr-ft^2-F. Since no satisfactory law has been devised, the designer must rely on past experience and experimental data in choosing an appropriate steam film coefficient.

Many equations for the calculation of water film coefficients have been advanced. The following equation is recommended by McAdams for heating water in a horizontal tube when the Reynolds number exceeds 2100.[2]

(2) $hD/k = 0.023(DG/\mu)^{0.8}\ (c_p\mu/k)^{0.4}$, where:

h = Water film coefficient, Btu/hr-ft^2-F

D = Actual inside pipe diameter, feet

k = Thermal conductivity of water at the average water temperature, Btu/hr-ft^2-F/ft

G = Mass velocity, #/hr-ft^2

μ = Viscosity of water at average water temperature, #/hr-ft

c_p = Specific heat of water at the average water temperature, Btu/#-F

If the range of the water temperature lies between 40 and 220 degrees Fahrenheit this equation may be more simply written:[3]

(2a) $h_w = 150(1 + 0.011t)(V_w)^{0.8}/\ D^{0.2}$, where:

h_w = Water film coefficient, Btu/hr-ft^2-F

t = Average water temperature, F

V_w = Average water velocity, ft/sec

D = Actual inside pipe diameter, inches

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1. McAdams, W. H. , Heat Transmission, Pg. 183, McGraw-Hill Book Company, New York, 1942
2. Op. cit., McAdams, pg. 168
3. Op. cit., McAdams, pg. 183

It can be seen from this equation, that for a given pipe diameter, at a constant water film temperature, that the water film coefficient varies as the eight-tenth power of the water velocity. This relationship is based on turbulent flow so it will not hold for low water velocities resulting in a Reynolds number less than 2100.

Mechanical Engineers Handbook ,edited by L. S. Marks, contains a complete list of thermal resistances of various metals on pages 392 and 637. These tables were used for determening the metal resistance.

The possibility of determening the separate thermal resistances graphically was first suggested by E. E. Wilson. He used this graphical method in conjunction with some experimental data of George Orrok.[1] The curve plotted by Wilson is shown on page 273 of McAdams' Heat Transmission. The data used for Wilson's curve was collected in 1909 by Orrok. Although this data was carefully collected, in some instances the results were erratic,due largely to the fact that in some instances the steam condensed by Orrok entered the condenser highly superheated. This led to lower steam film coefficients than would otherwise be expected. This condition was eliminated in this thesis by having the steam enter the heat exchanger with a quality of approximately 99 per cent.

It is stated in reference material, that in a project of this kind every precaution should be taken to prevent entrainment of air in the steam, since this affects the overall heat transfer coefficient greatly. If air is present in steam, a layer of condensate forms on the cooling

1. Orrok, G. A. , Transactions of the American Society of Mechanical Engineers, 32, 1139 (1910)

surface, and a film of air collects next to the condensate layer. This additional film decreases the apparent steam film coefficient and thus decreases the overall heat transfer rate. Since the steam from the Virginia Polytechnic Institute power plant flows through a deaerating heater, and since a positive pressure was maintained in the heat exchanger at all times, this was not a serious problem.

III. OBJECT OF THE INVESTIGATION

The object of this thesis is to determine the condensing steam film coefficient and the water film coefficient in a single tube heat exchanger at various water velocities and constant steam pressure.

IV. PROCEDURE

A) CONSTRUCTION OF APPARATUS

It was first necessary to construct the equipment necessary to obtain the desired data. This equipment consisted essentially of a shell and tube heat exchanger and the necessary auxilliary apparatus. Circulating water flowed through the pipe while steam condensed in the shell surrounding it.

The shell was constructed of three and one-half inch standard weight wrought steel pipe. Its length was 7.66 feet. Flanges constructed from one-quarter inch steel plate were welded to either end. Companion flanges, also cut from one-quarter inch steel plate, were bolted to their mates by eight, one-half inch machine bolts. These outside flanges were fitted with stuffing boxes which allowed the circulating water pipe to expand, but prevented steam leakage.

It was originally planned to have the circulating water flow through one inch outside diameter copper and admiralty metal condenser tubes. Due to the critical shortage of these metals it was impossible to obtain these tubes. A three-quarter inch standard weight copper pipe was available, so it was used for the first series of tests. For the second series of tests, this tube was replaced with a three-quarter inch extra heavy wrought steel pipe.

In both cases, these pipes passed through the center of the heat exchanger and through the stuffing boxes. A thermometer well was placed in the inlet line to measure the inlet water temperature. Another thermometer well was placed in the outlet side of the pipe to measure the outlet water temperature.

The cooling water pipe was connected by a three-quarter inch pipe to the existing water line which maintains a pressure of approximately 65 psi gage. A gate valve was placed on the inlet pipe so that the heat exchanger pipe could be replaced or cleaned. Flow was controlled by a globe valve located on the outlet side of the line. This assured that the heat exchanger pipe was full of water at all times. The outlet pipe was extended to a point where the flow of water could be conveniently weighed in weigh tanks placed on platform scales.

A thermometer well was placed in the center of the shell to record steam temperature. A pressure gage was connected through a one-eighth inch siphon to the heat exchanger to record steam pressure.

The supply steam was taken from the existing 165 psi gage line.

Since this steam was throttled down to a much lower pressure, it was necessary to desuperheat this steam before allowing it to enter the heat exchanger. This was accomplished by bubbling the steam through water. The desuperheater was constructed from a ten inch wrought steel pipe, standard weight, fourteen inches long. A top and bottom was cut from one-quarter inch steel plate and welded to the large pipe. The top plate of the desuperheater was drilled to allow a three-quarter inch supply line to extend down to within about one inch of the bottom of the tank. This pipe was welded in place and fitted with a globe valve to control steam flow. Another hole was drilled and tapped in the top of the tank and a three-quarter inch pipe screwed into it to serve as a discharge line for the steam. A one-eighth inch water line was tapped into the side of the tank near the bottom in order to supply water with which the desuperheating could be accomplished. A globe valve was placed in this line to control flow. Water glass fittings were than installed so that the water level in the desuperheater could be determined.

A tee was placed in the steam discharge line so that the steam flow could be branched to the heat exchanger and to a throttling calorimeter which measured the steam quality. A mercury manometer was connected to this calorimeter by means of a rubber hose. The calorimeter could be taken out of service by closing a gate valve which was placed in the line.

The steam flowed into the heat exchanger through a hole at one end of the top of the shell. A baffle plate with small holes drilled in it was welded to the inside of the shell directly under the entrance pipe. This served to prevent the steam from impinging directly on the cooling water pipe with too high a velocity, and also helped to distribute the steam equally.

A one-half inch condensate drain line was tapped into the bottom of the opposite end of the shell. This drain led to the hotwell where the condensate was collected. The shell was pitched so that the condensate would flow to the drain by gravity. The drain pipe was arranged so that there was always a leg of water between the shell and hotwell. This was to insure an accurate condensate temperature. A thermometer well was placed in this drain line to record the condensate temperature.

The hotwell was constructed from a four inch standard weight wrought steel pipe, four feet long. The top and bottom was cut from one-quarter inch steel plate and welded in place. A thermometer well was placed in the hotwell about one-third from the bottom of the tank. Gage glass fittings were installed and a 38 inch gage glass was used to indicate water level. A 36 inch steel scale was placed around the water glass. The condensate was drained from the hotwell through a one-half inch pipe. A gate valve was placed in this line to stop and start flow.

The shell was supported about five and one-half feet above floor level by a stand which was constructed mainly from one-half inch pipe.

When the unit was first assembled,a great deal of diffuculty was encountered with leaks in the welded joints. The unit was disassembled and the leaks repaired. Wherever possible, each individual part of the unit was given a hydrostatic test at 65 psi gage water pressure before it was assembled the second time. After this procedure was followed, the unit was assembled and tested at 50 psi gage steam pressure and was found to be free of leaks.

Eighty-five per cent magnesia insulation was used to decrease the heat loss from the heat exchanger. Insulation was placed on the desuperheater, the pipe connections to the calorimeter and shell, the calorimeter, the shell, and on the condensate line leading to the hotwell. The hotwell was not insulated since heat loss from this part of the apparatus would not affect the operation of the unit.

B) TEST PROCEDURE

The heat exchanger was tested with two different pipes conveying the circulating water. The first series of runs consisted of 49 tests using a new, clean, three-quarter inch standard weight copper pipe. After completion of this phase of the experimental work, this pipe was replaced with a new, clean, extra heavy wrought steel pipe. Forty-three test runs were made using this pipe.

Before any data was taken, the platform scales were calibrated by placing standard 50 pound weights on the scales and recording the scale readings. Then the pressure gage was calibrated by the use of a dead weight tester.

For the first run, the water velocity was adjusted to approximately two feet per second. The desuperheater was filled with water until about two inches of water were visible in the gage glass. The steam line was then opened allowing steam to bubble through the water. After a short while, the water reached saturation temperature and steam was discharged to the heat exchanger. The valve to the throttling calorimeter was opened.

The rate of steam flow was controlled to maintain five psi gage steam pressure. The drain valve on the hotwell line was allowed to remain open for two or three minutes so that any air that might be present in the shell would be forced out by the steam. This drain was then closed. The unit was allowed to reach equilibrium conditions before the test was begun. In the various tests, this time varied from ten to forty minutes.

When equilibrium conditions were reached, the test run was started. The following readings were recorded: inlet water temperature, condensate temperature, hotwell temperature, steam temperature, outlet water temperature, calorimeter temperature, calorimeter pressure, calorimeter water correction, steam pressure, tare weight of the cooling water scales, and the zero reading on the hotwell gage glass scale. Pressures and temperatures were recorded every three minutes until the completion of the 15 minute test. During the test, if the hotwell became filled with water, the scale reading was taken and the drum emptied. Another zero reading was recorded and the scale reading taken at the end of the test. Allowance is made in the calculations for the condensate that was lost during the time necessary to drain the hotwell.

The cooling water was weighed for the duration of the test. At high water velocities, one weigh tank would not handle the weight of cooling water flowing in 15 minutes, so two tanks were used. The flow was directed into one of them by use of a floor type flow divider equipped with quick opening valves.

When this run was completed, the water velocity was increased to three feet per second while the pressure was held constant at five psi gage and the test procedure repeated. Test runs were taken at this same pressure, while increasing the water velocity in increments of approximately one foot per second. The maximum velocity reached was about 18 feet per second.

In the next series of runs, the pressure was maintained at ten psi gage and the velocity varied through the same range. The last series of tests for the copper pipe were taken at fifteen psi gage with varying water velocity. After the three series of tests were completed, check runs were made by repeating some of the test runs at each pressure. The copper pipe was then removed and inspected.

Before the steel pipe was placed in the heat exchanger, it was cleaned thoroughly. The protective coating found on most new steel pipes was removed and the pipe was buffed to a shiny finish. It was then placed in the unit.

The test procedure for the steel pipe was similar to that used for the copper pipe. The only difference was that after adjusting the water velocity, test runs were taken at five, ten and fifteen psi gage. Then the water velocity was increased and the test procedure repeated. This was found to be preferable to completing all of the runs at one pressure before running any tests at other pressures, since maintaining a constant water velocity meant that equilibrium conditions were reached in a shorter time. For this steel pipe, the water velocity was varied from about two to thirteen feet per second. After this series of tests was completed, check runs were made by repeating some of the tests at each pressure.

V. METHOD OF CALCULATION

Calculations shown are for test # 47

1. Cooling Water Flow

$$W_w = 4\ F_d$$
$$= 4\ (2540)$$
$$= 10{,}160\ \#/hr$$

where, W_w = Water flow, #/hr
F_d = Water flow, #/15 min

2. Average Water Velocity

$$V_w = W_w\ v\ /\ A$$
$$A = \pi d_i^2/4$$
$$= \pi(0.82/12)^2/4$$
$$= 0.00366\ ft^2$$
$$V_w = \frac{(10{,}160)(0.01604)}{0.00366} =$$
$$= 12.39\ ft/sec$$

where, V_w = Average water velocity, ft/sec
v = Specific volume of the cooling water at its average temperature, $ft^3/\#$
A = Flow area, ft^2
d_i = Inside pipe diameter, ft

3. Temperature Rise of Cooling Water

$$\Delta t = t_o - t_i$$
$$= 89.9 - 56.1$$
$$= 33.8\ F$$

where, Δt = Temperature rise of cooling water, F
t_o = Leaving water temperature, F
t_i = Entering water temperature, F

4. Absolute Steam Pressure

$$P_a = P_g + 0.491\ P_b$$
$$= 15.6 + 0.491\ (28.22)$$
$$= 29.5\ psia$$

where, P_a = Absolute steam pressure , psia
P_g = Gage steam pressure, psig
P_b = Barometric pressure, in. Hg.

5. Absolute Calorimeter Pressure

$$P_{ca} = 0.491\ (P_b + P_{cg} - W_c/13.6)$$
$$= 0.491\ (28.22 + 1.6 - 7.8/13.6)$$
$$= 14.4 \text{ psia}$$

where, P_{ca} = Absolute calorimeter pressure, psia
P_{cg} = Calorimeter gage pressure, in.Hg.
W_c = Water correction, in. H_2O

6. Weight of Steam Condensed

$$W_s = \frac{A_d\ L_c\ C_f}{12\ V_h}$$

$$A_d = \frac{\pi d_i^2}{4} = \frac{\pi (4.03/12)^2}{4}$$
$$= 0.0885 \text{ ft}^2$$

$$C_f = \frac{T}{T - N_d\ T_d} = \frac{900}{900 - 4\ (30)}$$
$$= 1.152$$

$$Ws = \frac{(0.0885)(176)(1.152)}{12\ (0.01696)}$$
$$= 352 \text{ \#/hr}$$

where, Ws = Weight of steam condensed, #/hr
L_c = Total condensate scale reading, in.
A_d = Internal cross sectional area of pipe, ft^2
C_f = Correction factor to allow for condensate lost while draining the hotwell.
V_h = Specific volume of water at hotwell temperature, ft^3/#
d_i = Internal diameter of hotwell pipe, ft
T = Duration of test, sec.
T_d = Time to drain hotwell, Average = 30 sec.
N_d = Number of times hotwell is drained during the run.

7. Heat Gained by the Water

$$Q_w = W_w\ (h_{wo} - h_{wi})$$
$$= 10{,}160\ (\ 57.89 - 24.16\ )$$
$$= 342{,}200 \text{ Btu/hr}$$

where, Q_w = Heat gained by the water, Btu/hr
h_{wo} = Enthalpy of saturated water at leaving water temperature, Btu/#
h_{wi} = Enthalpy of saturated water at entering water temperature, Btu/#

8. Quality of Entering Steam

$$x = \left(\frac{h_c - h_f}{h_{fg}}\right) 100$$

$$= \left(\frac{1154.0 - 217.8}{945.9}\right) 100$$

$$= 99.0\ \%$$

where, x = Quality of entering steam, per cent
h_c = Enthalpy of steam at calorimeter pressure and temperature, Btu/#
h_{fg} = Latent heat of vaporization at heat exchanger steam pressure, Btu/#
h_f = Enthalpy of saturated water at heat exchanger steam pressure, Btu/#

9. Enthalpy of Entering Steam

$$h_s = h_c$$

$$= 1154.0 \text{ Btu/\#}$$

where, h_s = Enthalpy of entering steam, Btu/#

10. Heat Given Up by Steam

$$Q_s = W_s\ (h_s - h_{fc})$$

$$= 352\ (1154.0 - 212.7)$$

$$= 333{,}000 \text{ Btu/hr}$$

where, Q_s = Heat given up by steam, Btu/hr
hf_c = Enthalpy of saturated water at condensate temperature, Btu/#

11. Logarithmic Mean Temperature Difference

$$t_m = \frac{\Delta t_a - \Delta t_b}{\log_e \Delta t_a / \Delta t_b}$$

$$= \frac{191.4 - 157.6}{\log_e 191.4 / 157.6}$$

$$= 173.9 \text{ F}$$

where, t_m = Logarithmic Mean temperature difference, F
$\Delta t_a = t_s - t_{wi}$, F
$\Delta t_b = t_s - t_{wo}$, F
t_s = Steam temperature, F
t_{wi} = Inlet water temperature, F
t_{wo} = Outlet water temperature, F

12. Overall Heat Transfer Coefficient

$$U = \frac{Q_w}{S\ t_m}$$

where, U = Overall heat transfer coefficient, $Btu/hr-ft^2-F$
S = External cooling surface, ft^2
D_o = Outside diameter of cooling pipe, ft.
L = Effective length of cooling pipe, ft.

$$S = \pi D_o L$$

$$= \pi(1.05/12)(7.66)$$

$$= 2.102 \ ft^2$$

$$U = \frac{342,000}{2.102\ (173.9)}$$

$$= 939 \ Btu/hr-ft^2-F$$

13. Reciprocal of Overall Heat Transfer Coefficient

$$1/U = 1/939$$

$$= 0.001606 \ ft^2-hr-F/Btu$$

14. Reciprocal of the Water Velocity to the Eight-Tenths Power

$$1/V_w^{0.8} = 1/\ 12.39^{0.8}$$

$$= 0.134 \ (sec/ft)^{0.8}$$

15. Water Film Coefficient, Test Data

$$h_w = m\ V_w^{0.8}\ (D_o/D_i)$$

$$= 343\ (12.39)^{0.8}\ (1.05/0.824)$$

$$= 3500 \ Btu/hr-ft^2-F$$

where, h_w = Water film coefficient, $Btu/hr-ft^2-F$, based on inside surface
V_w = Average water velocity, ft/sec
m = Reciprocal slope of the straight line 1/U vs. $1/V_w^{0.8}$. For run 47-See curve 3.

16. Steam Film Resistance

$$R_s = a - R_m$$

$$R_m = \frac{L}{k}\left[\frac{2\,D_o}{D_o + D_i}\right]$$

$$= \frac{0.113}{12\,(220)}\left[\frac{2\,(1.05)}{(1.05 + 0.82)}\right]$$

$$= 0.000048 \text{ ft}^2\text{-hr-F/Btu}$$

$$R_s = 0.000635 - 0.000048$$

$$= 0.000587$$

where, R_s = Steam film resistance, Btu/hr-ft^2-F
a = Intercept of the straight line on the 1/U vs. $1/V_w^{0.8}$ curve, For run 47–See Figure 3.
R_m = Thermal resistance of pipe metal, Based on outside surface
L = Pipe thickness, ft.
k = Thermal conductivity of the pipe metal, Btu/hr-ft^2-F/ft: 220 for copper; 26 for steel. (Marks Handbook, pg. 392)
D_o = Outside diameter of cooling water pipe, in.
D_i = Inside diameter of cooling water pipe, in.

17. Steam Film Coefficient

$$h_s = 1/\,R_s$$

$$= 1\,/\,0.000587$$

$$= 1705 \text{ Btu/ hr-ft}^2\text{-F}$$

where, h_s = Steam film coefficient, Btu/hr-ft^2-F

18. Water Film Coefficient, Equation (2a)

$$h_w = 150\left[1 + 0.11t\right] V_w^{0.8}\,/\,D^{0.2}$$

$$= \frac{150\left[1 + 0.11(73)\right]\,12.39^{0.8}}{0.82^{0.2}}$$

$$= 2082 \text{ Btu/hr-ft}^2\text{-F}$$

where, t = Average water temperature, F
D = Actual inside pipe diameter, in.

19. Reynolds Number

$R_n = DG / \mu$

$G = W_w / A$

$= 10{,}160 / 0.00366$

$= 2{,}760{,}000$

$R_n = \dfrac{(0.82/12)(2{,}760{,}000)}{2.276}$

$= 83{,}100$

where, R_n = Reynolds Number
D = Inside Pipe diameter, ft.
G = Mass velocity, #/hr-ft^2
μ = Viscosity of water at average temperature, #/hr-ft (McAdams pg. 413)
W_w = Weight of water flow, #/hr
A = Flow Area, ft^2

20. Nusselt Number

$N_u = h D / k$

$= \dfrac{3500\ (0.82/12)}{0.3452}$

$= 695$

where, N_u = Nusselt Number
h = Experimental water film coefficient, Btu/hr-ft^2-F
D = Inside pipe diameter, ft
k = Thermal conductivity of water at average water temperature, Btu/hr-ft^2-F/ft

21. Prandtl Number

$P_r = c_p\ \mu / k$

$= \dfrac{1\ (2.276)}{0.3452}$

$= 6.59$

where, P_r = Prandtl Number
c_p = Specific heat of water at average water temperature, Btu/#-F

22. Dimensionless Number

$\dfrac{(h D / k)}{(c_p\ \mu / k)^{0.4}}$

$= 695 / 6.59^{0.4}$

$= 327$

VI. RESULTS

The results of this thesis are listed in the following tabulations and tables.

WATER FILM COEFFICIENTS

The water film coefficients are based on the internal surface area and are written in terms of the average water velocity, V_w.

For Flow Through 3/4" Standard Weight Copper Pipe

Steam Pressure	Water Film Coefficient
5 psi gage	419 $V_w^{0.8}$ Btu/hr-ft^2-F
10 psi gage	405 $V_w^{0.8}$ Btu/hr-ft^2-F
15 psi gage	438 $V_w^{0.8}$ Btu/hr-ft^2-F
Average	421 $V_w^{0.8}$ Btu/hr-ft^2-F

For Flow Through 3/4" Extra Heavy Steel Pipe

Steam Pressure	Water Film Coefficient
5 psi gage	414 $V_w^{0.8}$ Btu/hr-ft^2-F
10 psi gage	414 $V_w^{0.8}$ Btu/hr-ft^2-F
15 psi gage	421 $V_w^{0.8}$ Btu/hr-ft^2-F
Average	416 $V_w^{0.8}$ Btu/hr-ft^2-F

STEAM FILM COEFFICIENTS

For Flow Through 3/4" Standard Weight Copper Pipe

Steam Pressure	Steam Film Coefficient
5 psi gage	2215 Btu/hr-ft^2-F
10 psi gage	1990 Btu/hr-ft^2-F
15 psi gage	1705 Btu/hr-ft^2-F
Average	1970 Btu/hr-ft^2-F

For Flow Through 3/4" Extra Heavy Steel Pipe

Steam Pressure	Steam Film Coefficient
5 psi gage	2120 Btu/hr-ft^2-F
10 psi gage	2240 Btu/hr-ft^2-F
15 psi gage	2240 Btu/hr-ft^2-F
Average	2200 Btu/hr-ft^2-F

HEAT EXCHANGER RESULT SHEET — 3/4" Standard Weight Copper Pipe

Test Number	Water Velocity ft/sec	Temp. Rise of H_2O F	Steam Pressure psia	Water Weight #/ hr	Steam Condensed #/hr	Heat Gained by H_2O Btu/hr	Heat Lost by Steam Btu/hr	Log.Mean Temp.Dif. F	U, Btu/hr-ft^2-F	1/U 1/(Btu/hr-ft^2-F)	$1/V_w^{0.8}$ 1/(ft/sec)$^{0.8}$
1	2.02	73.0	19.22	1,644	126	119,800	121,000	126.9	450	0.00222	0.570
2	2.80	66.0	18.57	2,290	159	150,700	153,100	131.2	545	0.00184	0.439
3	3.60	59.0	18.80	2,956	208	173,800	199,500	137.0	604	0.00166	0.359
4	3.95	60.4	19.09	3,232	206	194,800	198,000	134.2	690	0.00145	0.333
5	5.10	53.5	19.40	4,192	241	222,800	231,800	139.0	763	0.00131	0.272
6	5.91	50.2	18.87	4,848	256	243,500	246,000	141.0	821	0.00122	0.241
7	6.80	48.3	18.73	5,552	282	267,800	272,000	142.8	891	0.00112	0.216
8	7.80	45.0	19.38	6,372	303	285,800	292,000	145.9	934	0.00107	0.194
9	9.46	39.8	19.22	7,728	310	306,200	298,200	148.6	981	0.00102	0.166
10	9.75	38.4	19.12	7,960	305	305,000	293,800	150.8	964	0.00104	0.162
11	10.45	38.2	19.11	8,584	324	326,500	312,000	149.8	1038	0.00096	0.152
12	11.73	35.2	18.51	9,628	328	338,000	315,800	151.5	1060	0.00094	0.139
13	13.21	32.1	19.13	10,844	333	347,600	320,000	154.0	1073	0.00093	0.126
14	14.10	30.5	18.63	11,540	352	352,000	338,000	153.4	1090	0.00092	0.120
15	14.71	27.9	18.63	12,064	350	336,000	336,200	154.1	1036	0.00096	0.116
16	18.21	23.9	18.92	14,976	372	342,200	358,000	157.5	1032	0.00097	0.098
30	4.34	52.7	19.01	3,556	175	187,000	168,100	140.2	634	0.00158	0.309
31	8.87	38.4	18.98	7,276	260	278,200	250,000	149.9	888	0.00113	0.177
32	15.90	25.1	19.93	13,024	345	327,800	332,000	159.0	980	0.00102	0.110
34	16.41	27.2	19.79	13,496	362	365,800	348,000	157.7	1100	0.00091	0.107
17	4.02	59.7	23.71	3,284	200	196,000	191,080	146.8	635	0.00157	0.329
18	4.94	57.0	23.71	4,048	226	229,500	215,500	148.5	735	0.00136	0.279
19	5.75	53.6	24.41	4,704	261	251,900	248,500	150.9	794	0.00126	0.246
20	7.00	48.8	24.00	5,748	289	279,800	276,000	153.1	860	0.00116	0.210
21	6.93	48.7	24.15	5,672	283	276,000	270,000	154.8	850	0.00118	0.212

HEAT EXCHANGER RESULT SHEET 3/4" Standard Weight Copper Pipe

Test Number	Water Velocity ft/sec	Temp.Rise of H2O F	Steam Pressure psia	Water Weight #/hr	Steam Condensed #/hr	Heat Gained by H2O Btu/hr	Heat Lost by Steam Btu/hr	Log.Mean Temp.Dif. F	U, Btu/hr-ft^2-F	1/U 1/(Btu/hr-ft^2-F)	$1/V_w^{0.8}$ 1/(ft/sec)$^{0.8}$
22	10.40	39.9	24.10	8,532	343	338,200	328,000	160.0	1008	0.00099	0.154
23	10.80	38.8	24.30	8,888	353	343,800	336,000	160.3	1018	0.00098	0.149
24	11.60	36.3	24.30	9,548	360	346,000	343,800	162.0	1015	0.00098	0.141
25	9.36	40.3	24.15	7,656	328	308,000	312,000	159.0	923	0.00108	0.117
26	12.89	32.9	23.90	10,580	370	346,000	353,800	163.7	1008	0.00099	0.130
27	13.82	29.6	24.20	11,320	348	335,000	332,000	165.2	965	0.00104	0.122
28	15.70	26.4	23.82	12,892	365	340,000	349,000	167.3	965	0.00104	0.111
29	18.80	23.4	24.00	15,420	386	360,000	369,000	168.8	1014	0.00098	0.096
33	16.72	25.8	24.28	13,732	348	353,800	332,000	167.3	1002	0.00099	0.105
35	16.69	27.2	24.80	13,692	388	370,500	370,000	168.1	1048	0.00095	0.105
37	3.12	64.9	23.80	2,522	160	165,000	153,000	143.1	550	0.00182	0.401
38	4.26	56.5	24.10	3,488	206	193,100	197,100	147.1	625	0.00160	0.313
36	3.12	69.7	28.40	2,552	179	177,800	169,800	151.8	557	0.00179	0.401
39	4.39	60.0	28.60	3,580	216	214,000	205,000	156.5	651	0.00154	0.306
40	5.00	56.6	28.50	4,092	226	230,000	214,200	159.1	688	0.00145	0.276
41	6.10	52.2	28.80	5,000	248	260,400	235,800	162.6	764	0.00131	0.235
42	7.05	48.0	28.70	5,772	288	276,000	272,000	165.4	794	0.00126	0.210
43	8.21	44.6	28.75	6,728	304	291,000	288,000	167.7	849	0.00118	0.185
44	9.00	31.3	29.20	7,376	314	304,000	296,000	169.1	856	0.00117	0.172
45	10.29	37.7	28.70	8,416	332	316,500	314,000	170.9	884	0.00113	0.155
46	11.41	36.0	29.40	9,352	320	336,000	304,000	173.0	925	0.00108	0.143
47	12.39	33.8	29.50	10,160	352	342,000	338,000	173.9	939	0.00107	0.134
48	13.21	32.3	29.40	10,868	324	350,200	307,000	174.7	956	0.00105	0.126
49	14.30	29.9	29.20	11,764	354	350,200	336,000	175.9	950	0.00105	0.119

HEAT EXCHANGER RESULT SHEET

3/4" Extra Heavy Steel Pipe

Test Number	Water Velocity ft/sec	Temp.Rise of H_2O F	Steam Pressure psia	Water Weight #/hr	Steam Condensed #/hr	Heat Gained by H_2O Btu/hr	Heat Lost by Steam Btu/hr	Log Mean Temp.Dif. F	U, Btu/hr-ft2-F	1/U 1/(Btu/hr-ft2-F)	$1/V_w^{0.8}$ $1/(ft/sec)^{0.8}$
50	3.73	50.9	18.95	2,468	139	125,200	133,200	136.8	436	0.00229	0.350
53	4.60	51.2	18.90	3,044	163	155,900	157,000	137.1	540	0.00185	0.304
56	5.60	43.2	20.05	3,716	170	160,500	163,100	144.0	530	0.00189	0.252
59	6.84	36.1	18.75	4,540	164	163,100	158,000	146.8	530	0.00189	0.214
62	7.08	36.6	18.85	4,696	183	171,000	174,000	147.9	550	0.00182	0.209
65	8.10	35.4	19.00	5,352	193	189,000	186,000	148.0	609	0.00164	0.187
68	6.10	42.3	19.00	4,044	173	170,300	167,000	144.1	561	0.00178	0.235
69	9.16	32.2	19.00	6,068	195	195,00	188,000	150.2	617	0.00162	0.170
72	10.00	31.0	19.20	6,644	202	204,800	195,000	151.8	643	0.00155	0.159
75	11.11	27.5	18.50	7,376	203	202,000	195,000	153.0	630	0.00159	0.146
78	12.70	25.7	19.00	8,428	212	216,200	205,000	154.9	666	0.00150	0.130
81	2.16	66.1	18.60	1,428	97	94,000	93,000	128.8	348	0.00288	0.540
84	3.04	56.0	18.40	2,016	118	112,100	114,000	135.1	395	0.00254	0.410
87	4.25	48.7	18.90	2,820	140	137,100	135,000	140.0	466	0.00214	0.314
90	5.04	44.5	18.60	3,332	154	147,800	148,000	141.8	495	0.00202	0.274
51	3.52	61.6	23.70	2,324	152	142,900	145,000	142.1	477	0.00210	0.365
54	4.66	56.1	24.30	3,080	179	172,000	171,000	146.7	559	0.00179	0.292
57	5.60	46.1	23.85	3,712	171	170,900	163,100	152.9	530	0.00189	0.252
60	6.90	39.8	24.30	4,576	187	181,800	179,000	157.7	549	0.00182	0.212
63	7.28	39.9	23.60	4,824	192	192,000	184,000	156.1	585	0.00171	0.204
66	8.00	38.1	23.60	5,300	206	201,800	197,000	157.2	609	0.00164	0.189
70	8.86	35.5	23.50	5,856	205	208,000	196,000	159.3	620	0.00161	0.175

HEAT EXCHANGER RESULT SHEET 3/4" Extra Heavy Steel Pipe

Test Number	Water Velocity ft/sec	Temp.Rise of H2O F	Steam Pressure psia	Water Weight #/hr	Steam Condensed #/hr	Heat Gained by H2O Btu/hr	Heat Lost by Steam Btu/hr	Log Mean Temp.Dif. F	U, Btu/hr-ft2-F	1/U 1/(Btu/hr-ft^2-F)	$1/V_w^{0.8}$ 1/(ft/sec)$^{0.8}$
73	10.00	33.1	23.70	6,620	215	219,200	206,000	160.8	650	0.00154	0.159
76	10.98	30.0	23.85	7,292	218	218,000	208,000	163.0	635	0.00157	0.147
79	12.89	27.1	23.90	8,536	224	230,400	215,000	165.1	665	0.00150	0.129
82	2.16	71.2	24.20	1,424	107	101,200	101,800	138.1	348	0.00288	0.540
85	3.10	60.3	24.10	2,056	127	123,500	121,000	144.8	405	0.00246	0.405
88	4.21	52.1	23.10	2,796	152	145,200	145,100	148.0	467	0.00214	0.316
91	5.02	47.8	23.50	3,324	164	158,900	156,500	151.0	500	0.00200	0.276
52	3.56	67.0	28.90	2,348	165	157,000	156,000	150.3	495	0.00202	0.362
55	4.65	60.0	28.70	3,076	197	183,900	188,500	154.6	565	0.00177	0.292
58	5.57	50.2	28.35	3,692	192	187,000	182,000	160.7	554	0.00180	0.253
61	6.78	43.6	28.45	4,488	202	195,000	191,500	165.1	561	0.00178	0.216
64	7.14	43.5	27.90	4,720	204	204,400	194,000	164.6	591	0.00169	0.208
67	8.26	39.9	27.75	5,484	224	218,000	213,500	166.1	625	0.00160	0.185
71	9.16	37.0	27.80	6,080	222	224,000	211,000	169.4	629	0.00159	0.170
74	9.76	35.6	27.80	6,460	224	229,800	212,000	168.8	649	0.00154	0.162
77	10.68	32.3	27.60	7,080	226	228,000	215,000	171.0	635	0.001573	0.150
80	13.00	28.6	27.90	8,612	236	245,800	224,000	173.7	674	0.00148	0.128
83	2.14	75.7	27.90	1,408	117	106,300	111,000	144.0	351	0.00285	0.544
86	3.10	64.2	27.70	2,048	130	131,200	123,700	151.7	412	0.00243	0.405
89	4.27	56.1	28.10	2,832	164	158,100	155,800	158.9	476	0.00210	0.312
92	5.01	51.5	28.20	3,320	170	170,100	162,000	160.6	505	0.00198	0.276

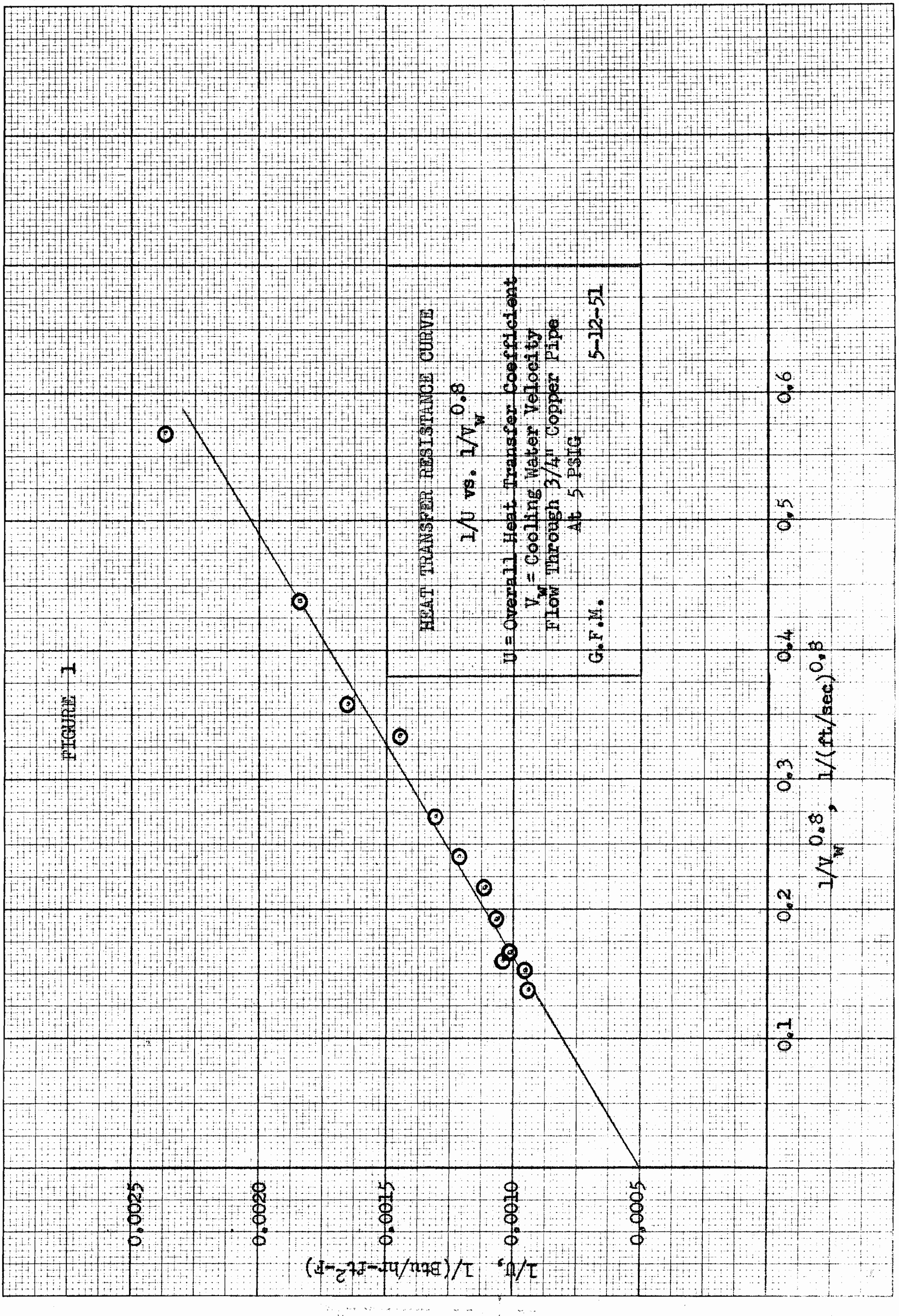


FIGURE 1
HEAT TRANSFER RESISTANCE CURVE
1/U vs. 1/V_w^0.8
U = Overall Heat Transfer Coefficient
V_w = Cooling Water Velocity
Flow Through 3/4" Copper Pipe
At 5 PSIG
G.F.M.
5-12-51
0.1
0.2
0.3
0.4
0.5
0.6
1/V_w^0.8, 1/(ft/sec)^0.8
0.0005
0.0010
0.0015
0.0020
0.0025
1/U, 1/(Btu/hr-ft^2-F)

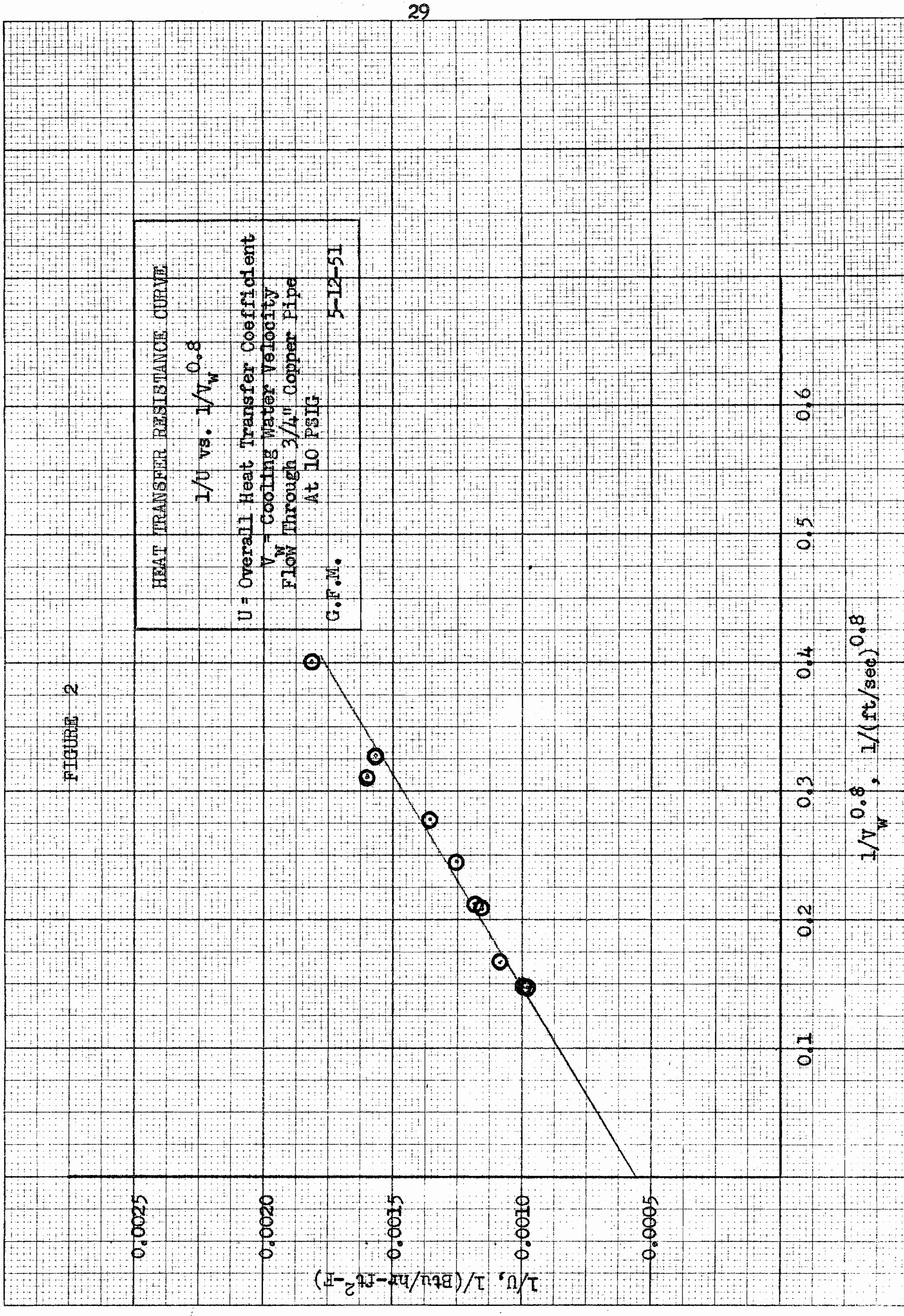

HEAT TRANSFER RESISTANCE CURVE
1/U vs. $1/V_W^{0.8}$
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Copper Pipe
At 10 PSIG
G.F.M.
5-12-51
0.0025
0.0020
0.0015
0.0010
0.0005
1/U, 1/(Btu/hr-ft^2-F)
0.1
0.2
0.3
0.4
0.5
0.6
$1/V_W^{0.8}$, $1/(\text{ft/sec})^{0.8}$

FIGURE 2

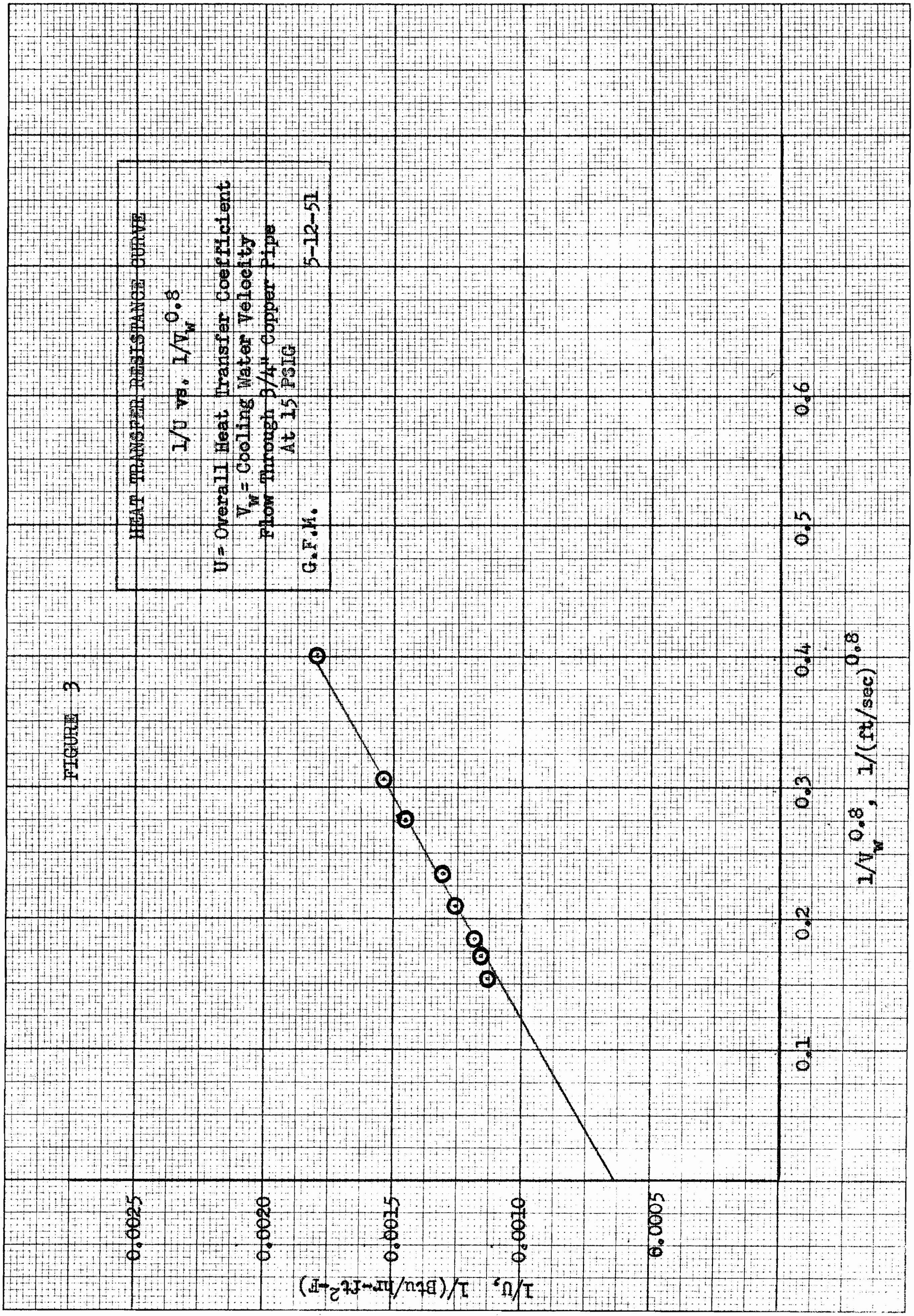


FIGURE 3
HEAT TRANSFER RESISTANCE CURVE
1/U vs. 1/V_W^0.8
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Copper Pipe
At 15 PSIG
G.F.M.
5-12-51
0.0005
0.0010
0.0015
0.0020
0.0025
1/U, 1/(Btu/hr-ft^2-F)
0.1
0.2
0.3
0.4
0.5
0.6
1/V_W^0.8, 1/(ft/sec)^0.8

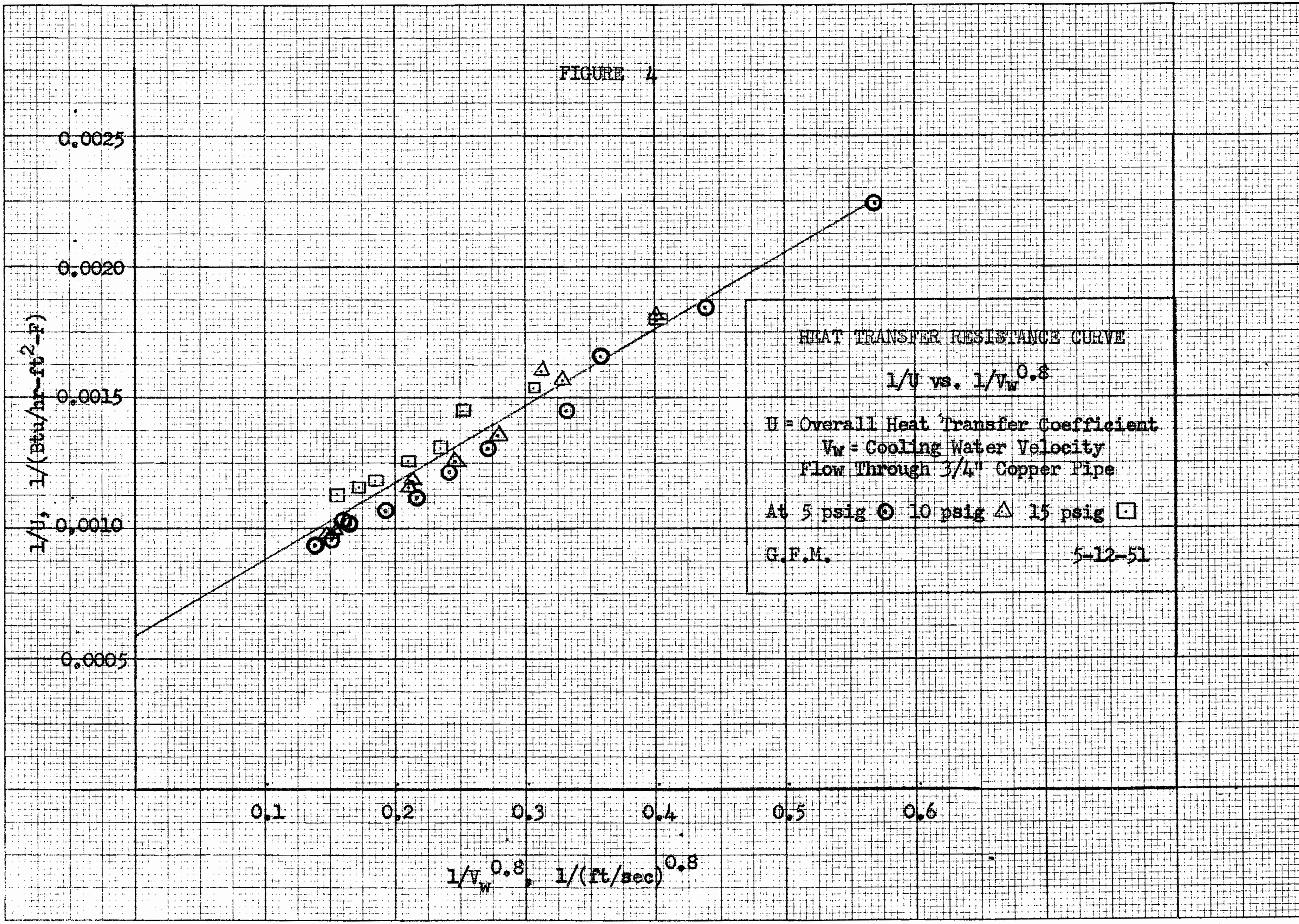

0.0025
0.0020
0.0015
0.0010
0.0005
$1/U$, $1/(Btu/hr\text{-}ft^2\text{-}F)$
0.1
0.2
0.3
0.4
0.5
0.6
$1/V_W^{0.8}$, $1/(ft/sec)^{0.8}$
HEAT TRANSFER RESISTANCE CURVE
$1/U$ vs. $1/V_W^{0.8}$
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Copper Pipe
At 5 psig ⊙ 10 psig △ 15 psig ⊡
G.F.M.
5-12-51

FIGURE 4

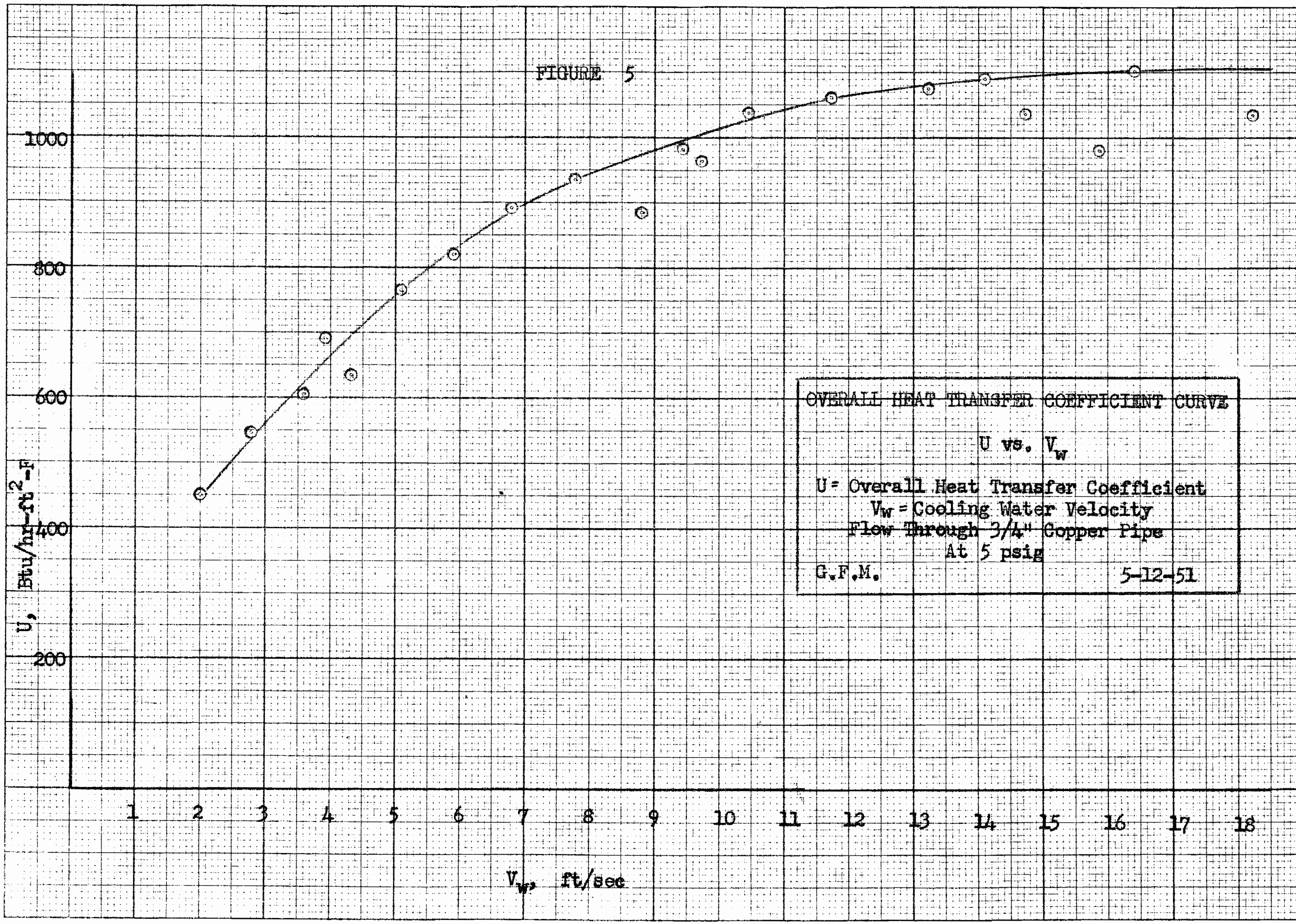

1000
800
600
400
200
U, Btu/hr-ft²-F
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
V_W, ft/sec
OVERALL HEAT TRANSFER COEFFICIENT CURVE
U vs. V_W
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Copper Pipe
At 5 psig
G.F.M.
5-12-51

FIGURE 5

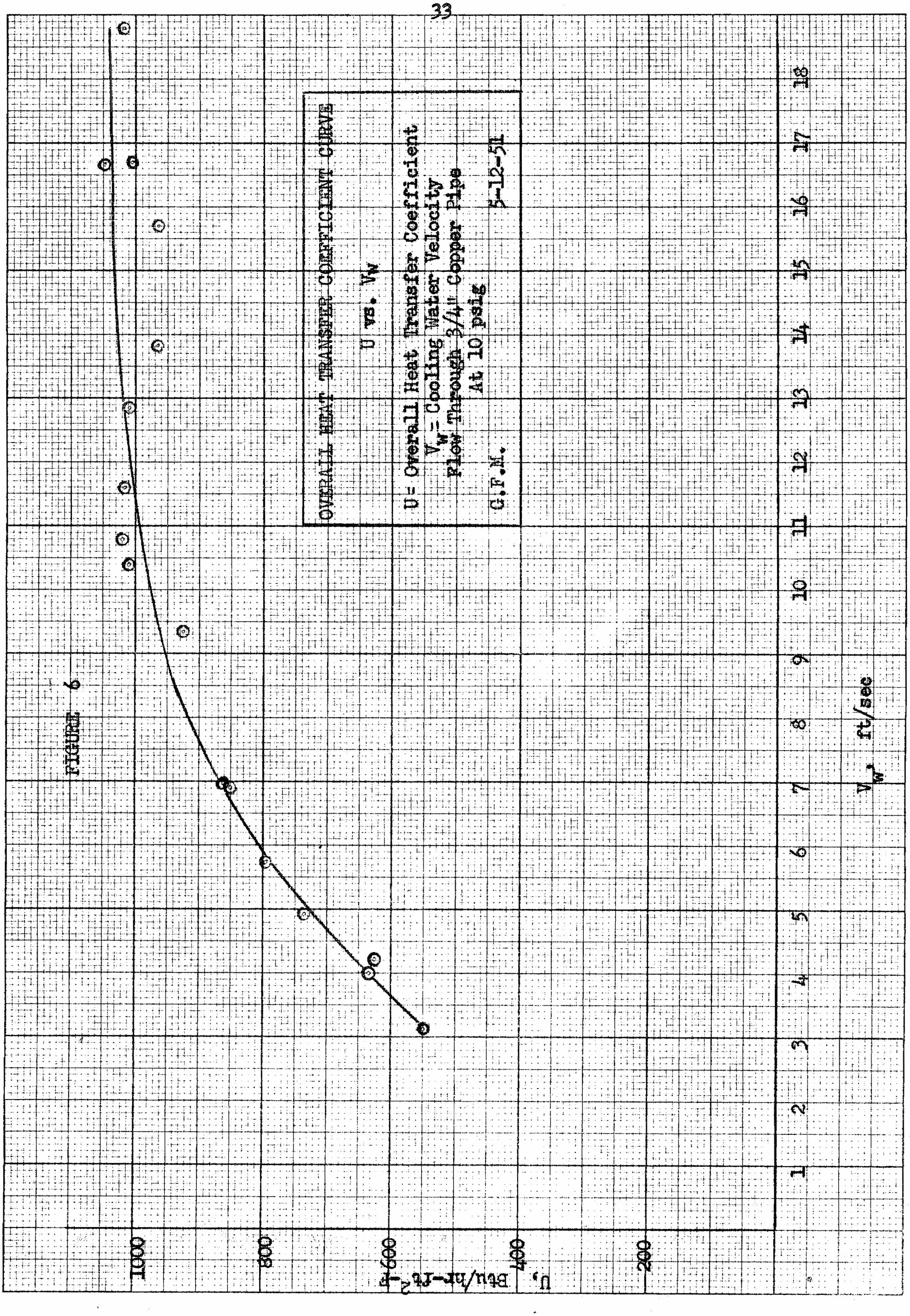
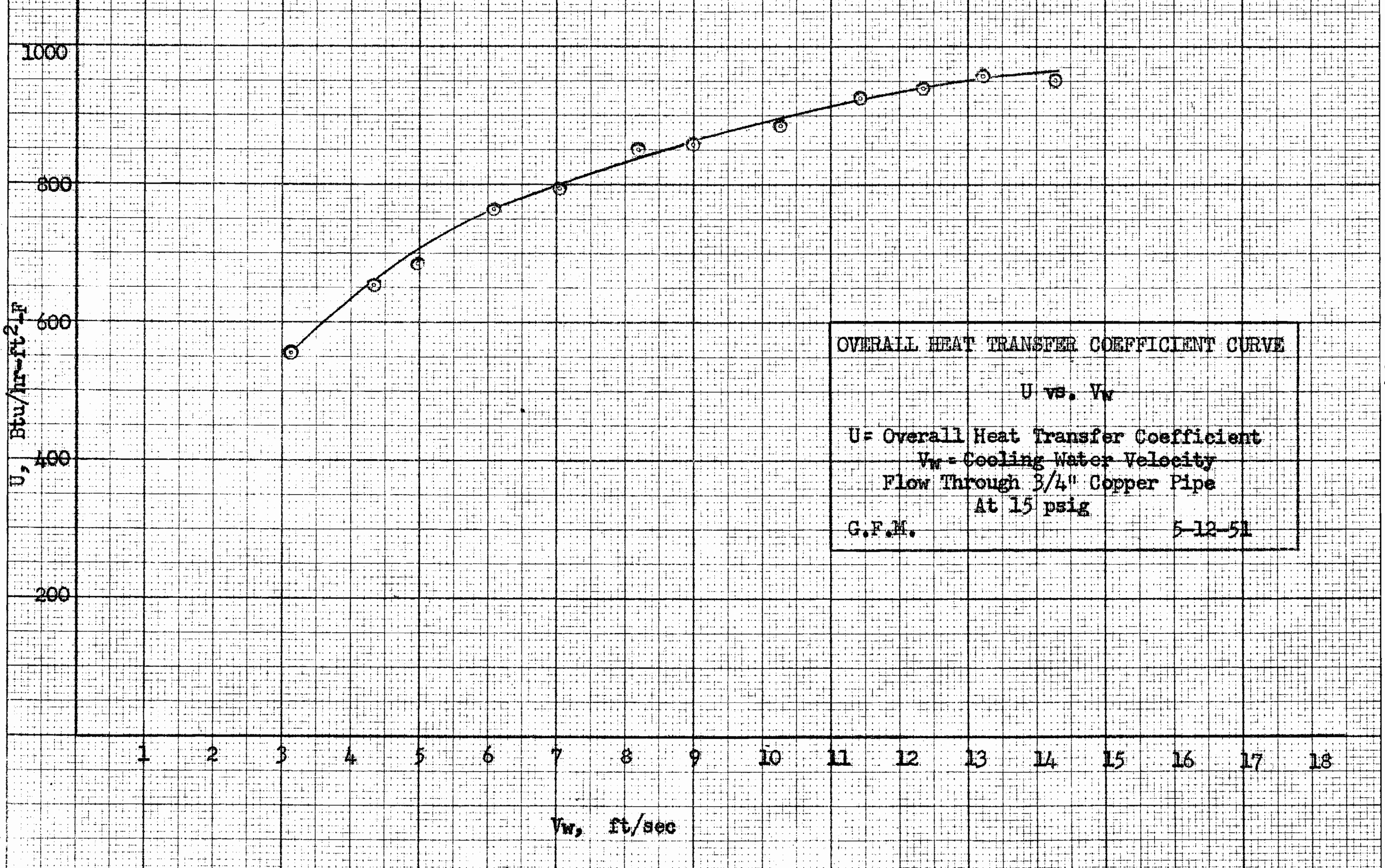


FIGURE 6
OVERALL HEAT TRANSFER COEFFICIENT CURVE
U vs. V_W
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Copper Pipe
At 10 psig
G.F.M.
5-12-51
V_W, ft/sec
U, Btu/hr-ft^2-F
1000
800
600
400
200
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18

FIGURE 7


1000
800
600
400
200
U, Btu/hr-ft2-F
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
Vw, ft/sec
OVERALL HEAT TRANSFER COEFFICIENT CURVE
U vs. Vw
U = Overall Heat Transfer Coefficient
Vw = Cooling Water Velocity
Flow Through 3/4" Copper Pipe
At 15 psig
G.F.M.
5-12-51

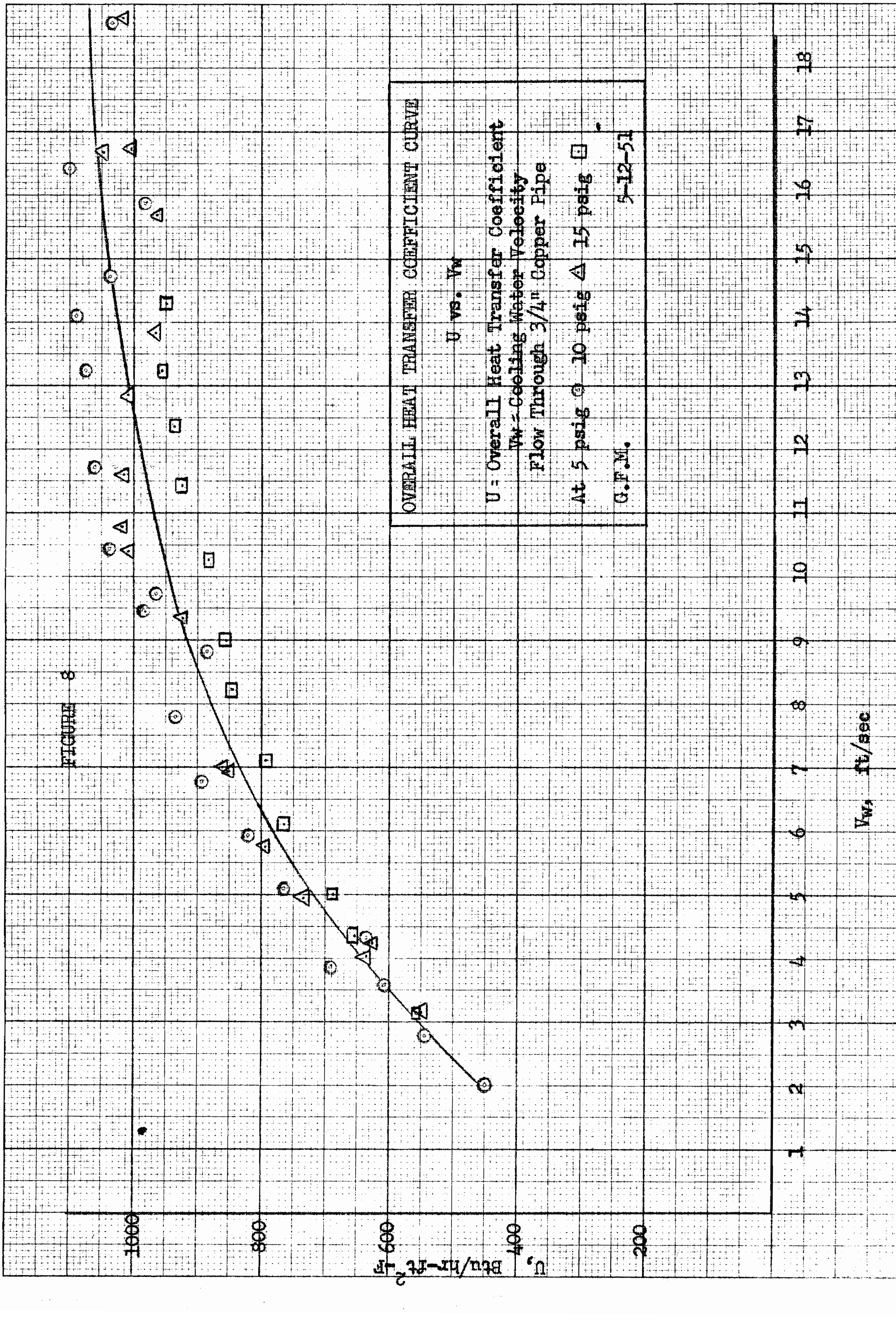


FIGURE 8
OVERALL HEAT TRANSFER COEFFICIENT CURVE
U vs. Vw
U = Overall Heat Transfer Coefficient
Vw = Cooling Water Velocity
Flow Through 3/4" Copper Pipe
At 5 psig ⊙ 10 psig △ 15 psig □
G.F.M.
5-12-51
U, Btu/hr-ft²-°F
1000
800
600
400
200
1
2
3
4
5
6
7
8
9
10
11
12
13
14
15
16
17
18
Vw, ft/sec

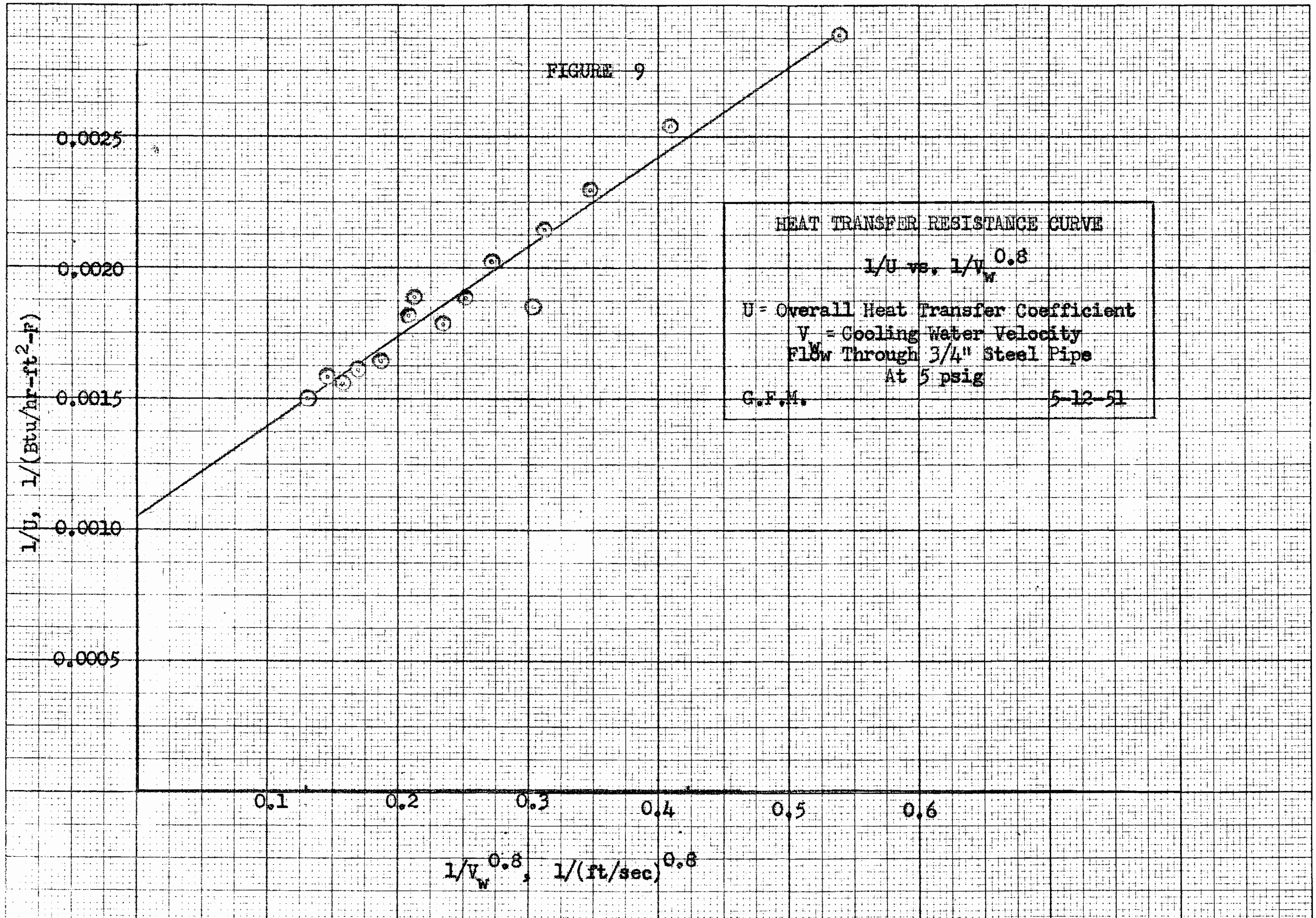


FIGURE 9
HEAT TRANSFER RESISTANCE CURVE
1/U vs. 1/V_W^0.8
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Steel Pipe
At 5 psig
G.F.M.
5-12-51
0.0025
0.0020
0.0015
0.0010
0.0005
1/U, 1/(Btu/hr-ft^2-F)
0.1
0.2
0.3
0.4
0.5
0.6
1/V_W^0.8, 1/(ft/sec)^0.8

FIGURE 10

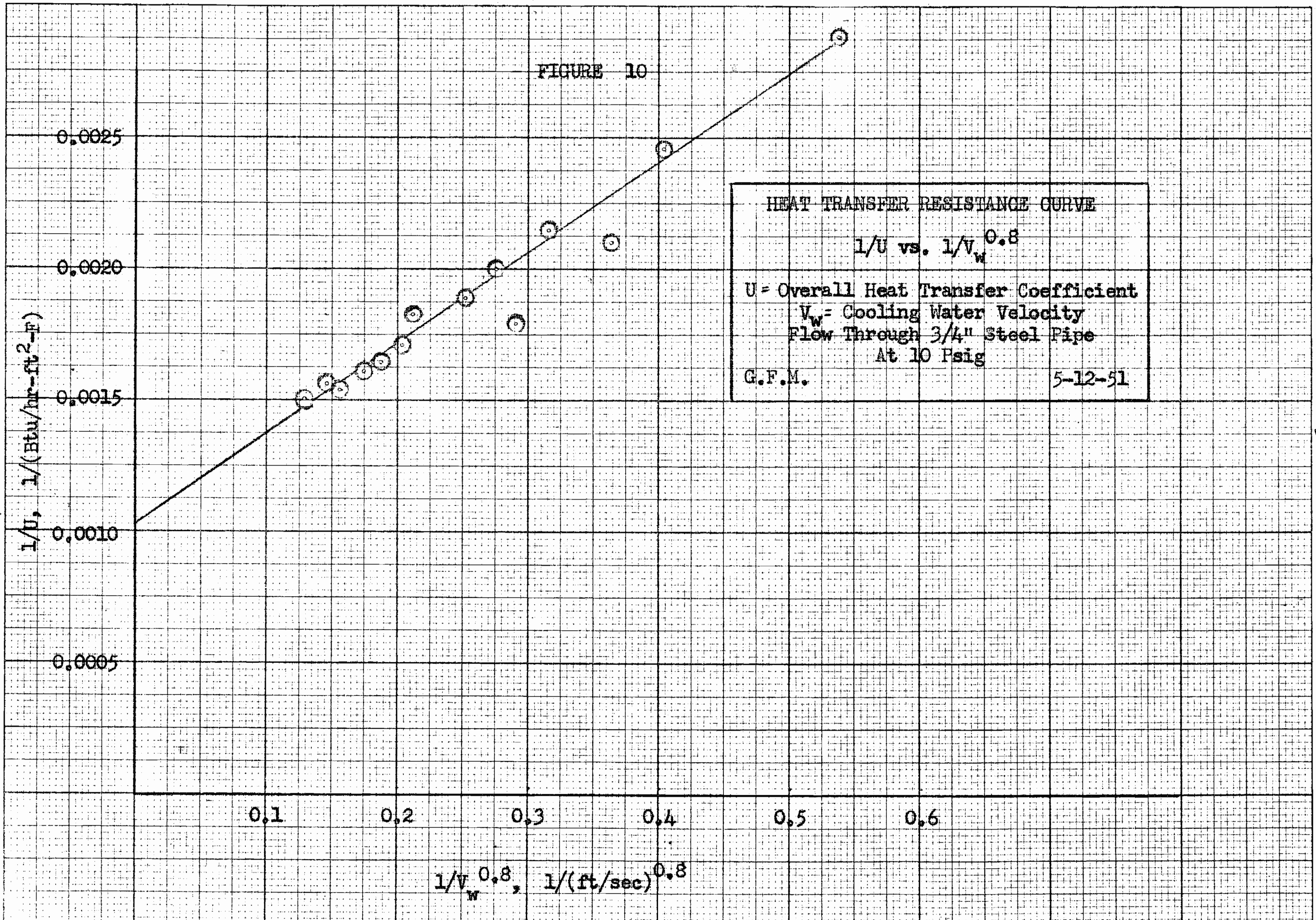
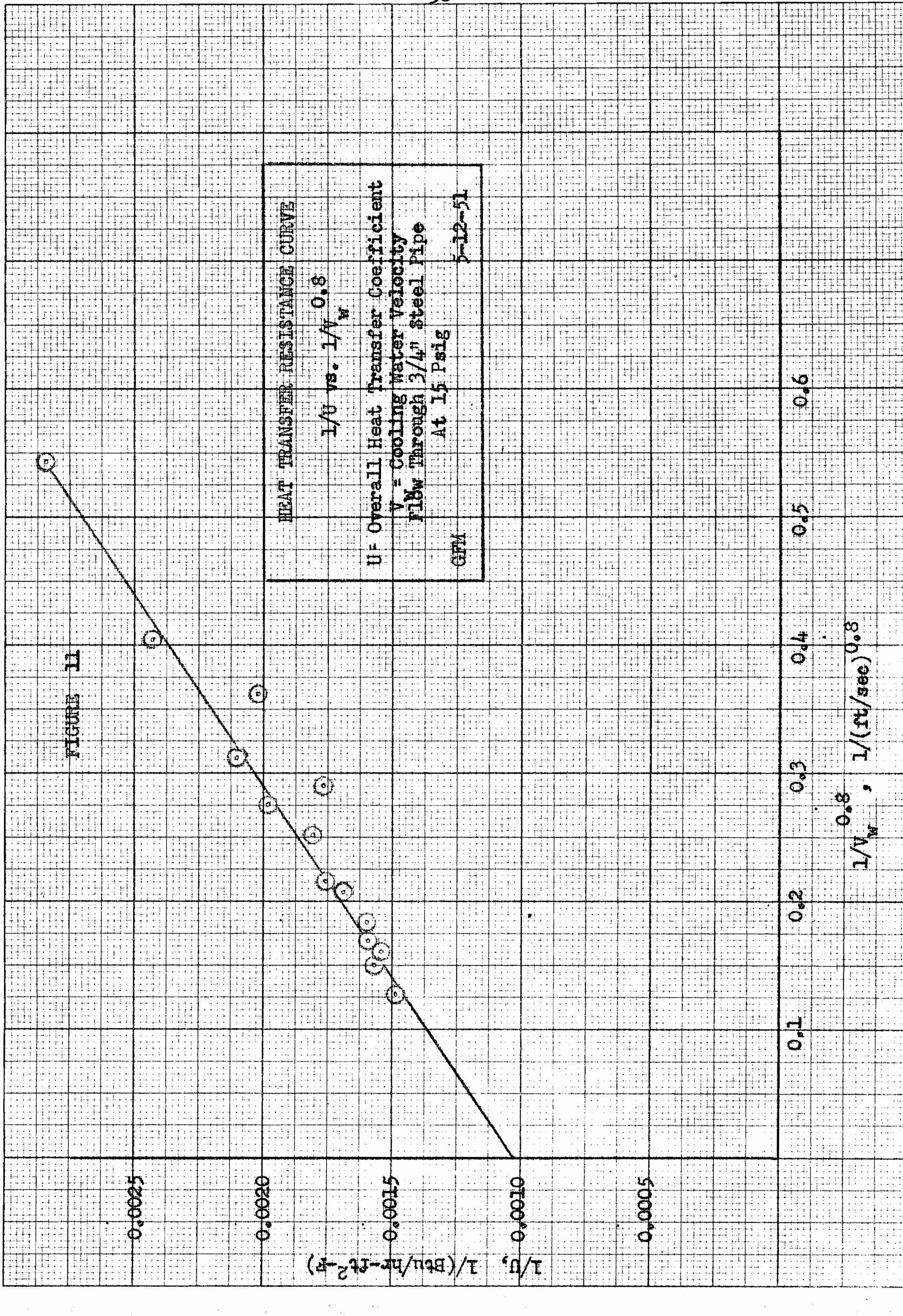

HEAT TRANSFER RESISTANCE CURVE
1/U vs. 1/V_w^0.8
U = Overall Heat Transfer Coefficient
V_w = Cooling Water Velocity
Flow Through 3/4" Steel Pipe
At 10 Psig
G.F.M.
5-12-51
0.0025
0.0020
0.0015
0.0010
0.0005
1/U, 1/(Btu/hr-ft^2-F)
0.1
0.2
0.3
0.4
0.5
0.6
1/V_w^0.8, 1/(ft/sec)^0.8

FIGURE 11


HEAT TRANSFER RESISTANCE CURVE
1/U vs. $1/V_w^{0.8}$
U = Overall Heat Transfer Coefficient
V_w = Cooling Water Velocity
Flow Through 3/4" Steel Pipe
At 15 Psig
GFM
5-12-51
0.1
0.2
0.3
0.4
0.5
0.6
$1/V_w^{0.8}$, $1/(ft/sec)^{0.8}$
0.0005
0.0010
0.0015
0.0020
0.0025
1/U, $1/(Btu/hr\text{-}ft^2\text{-}F)$

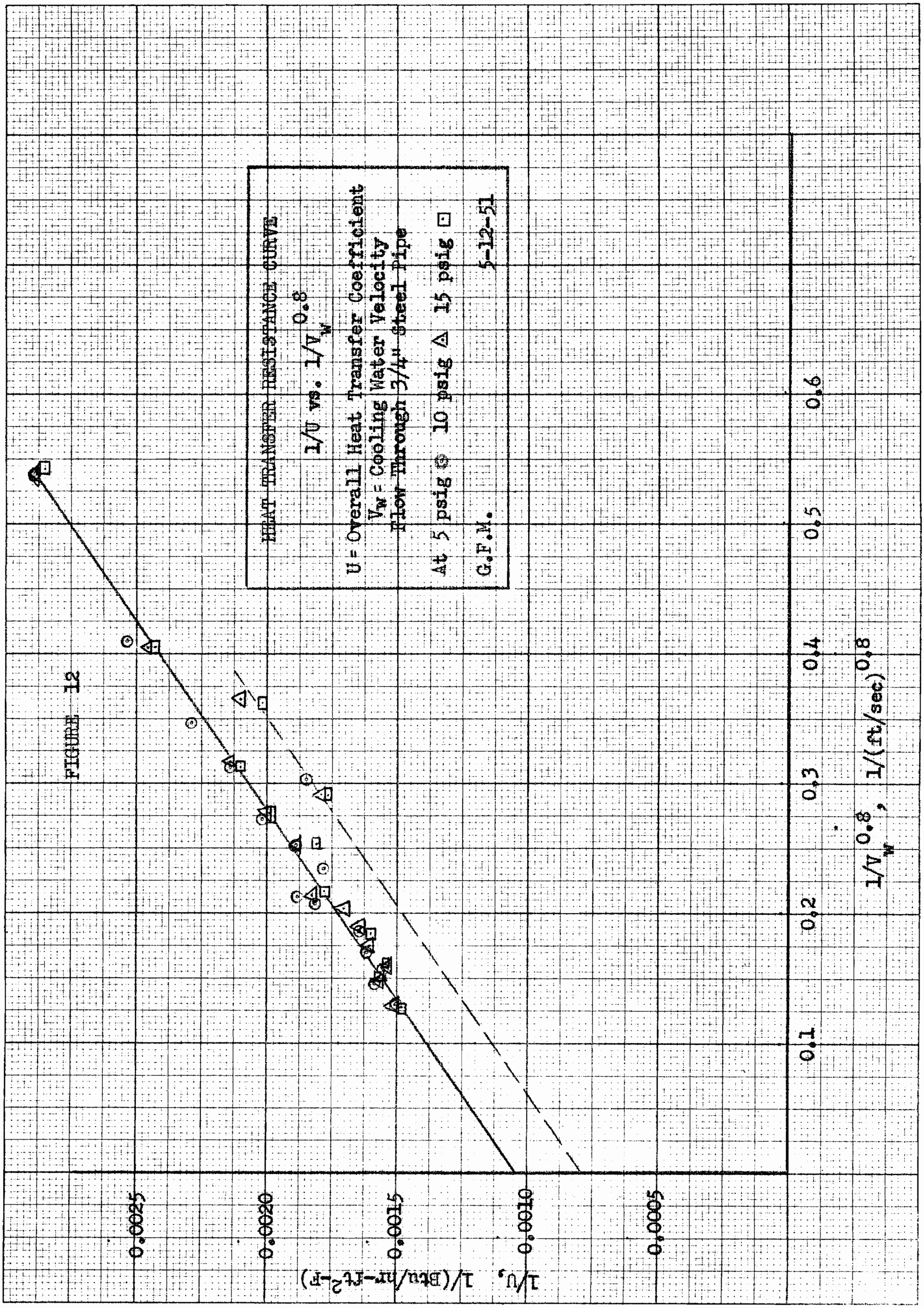


FIGURE 12
HEAT TRANSFER RESISTANCE CURVE
1/U vs. $1/V_W^{0.8}$
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Steel Pipe
At 5 psig ⊙ 10 psig △ 15 psig ⊡
G.F.M.
5-12-51
0.0025
0.0020
0.0015
0.0010
0.0005
1/U, 1/(Btu/hr-ft^2-F)
0.1
0.2
0.3
0.4
0.5
0.6
$1/V_W^{0.8}$, $1/(\text{ft/sec})^{0.8}$

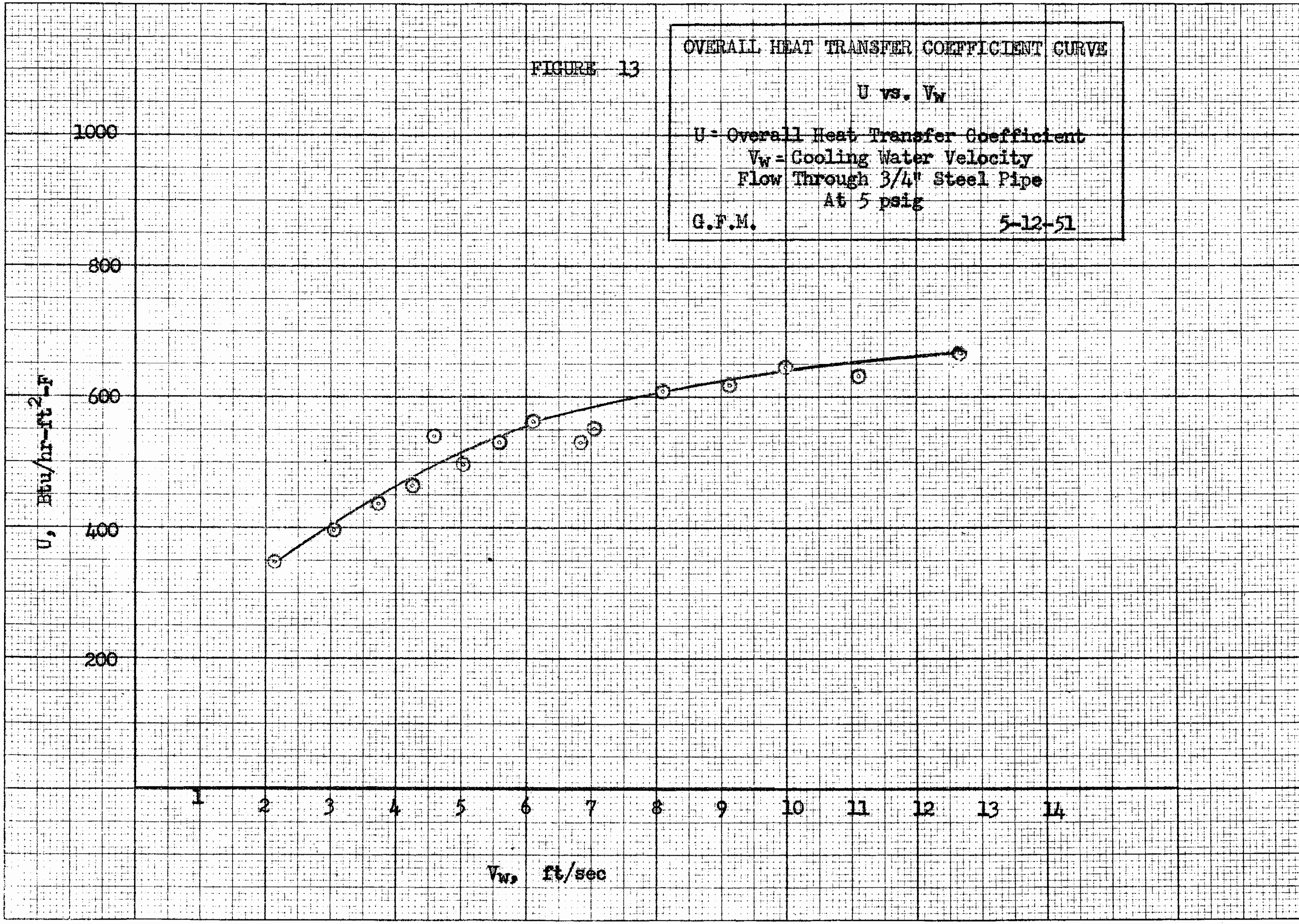
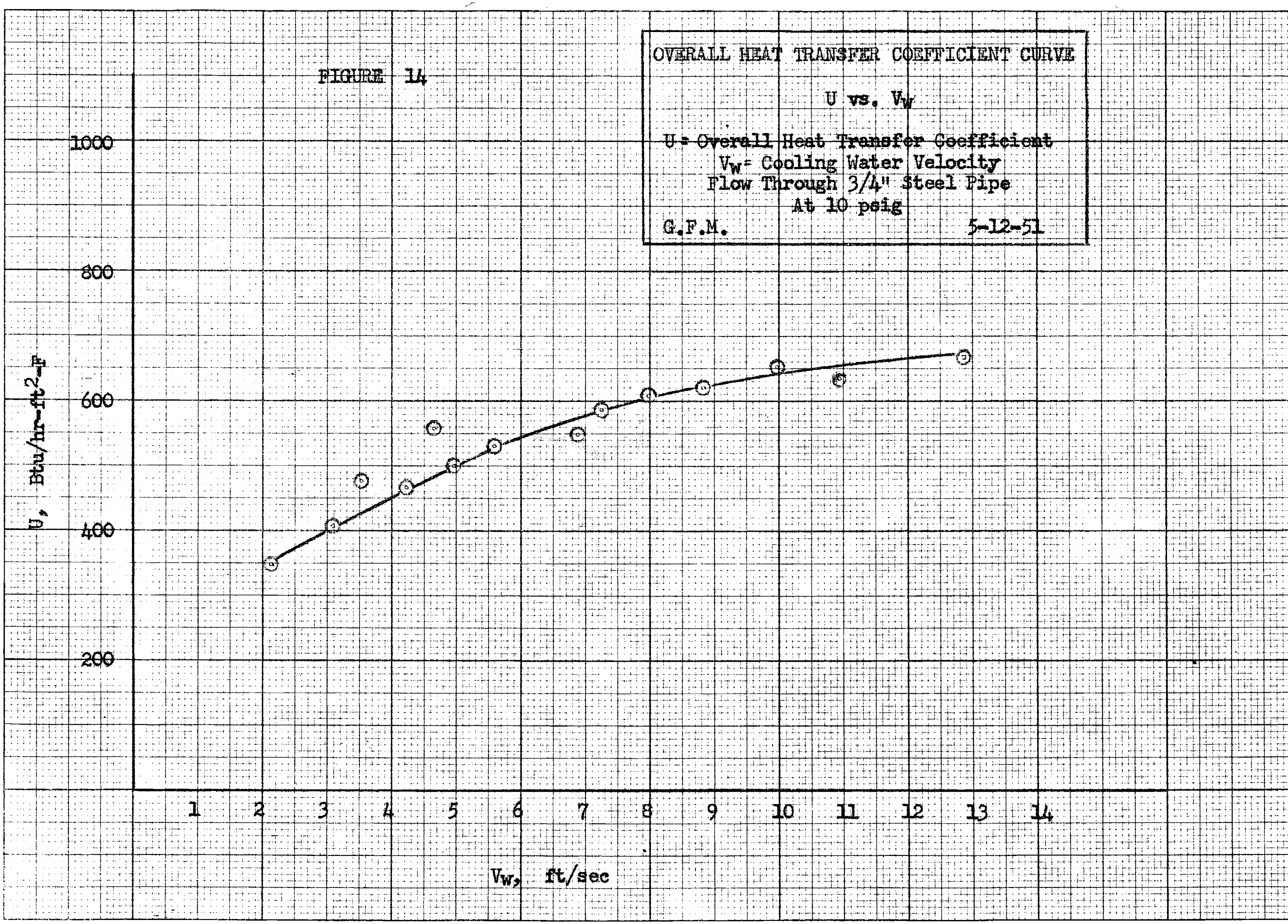


FIGURE 13
OVERALL HEAT TRANSFER COEFFICIENT CURVE
U vs. V_W
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Steel Pipe
At 5 psig
G.F.M.
5-12-51
U, Btu/hr-ft^2-F
1000
800
600
400
200
1
2
3
4
5
6
7
8
9
10
11
12
13
14
V_W, ft/sec

FIGURE 14


OVERALL HEAT TRANSFER COEFFICIENT CURVE
U vs. V_W
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Steel Pipe
At 10 psig
G.F.M.
5-12-51
1000
800
600
400
200
U, Btu/hr-ft^2-F
1
2
3
4
5
6
7
8
9
10
11
12
13
14
V_W, ft/sec

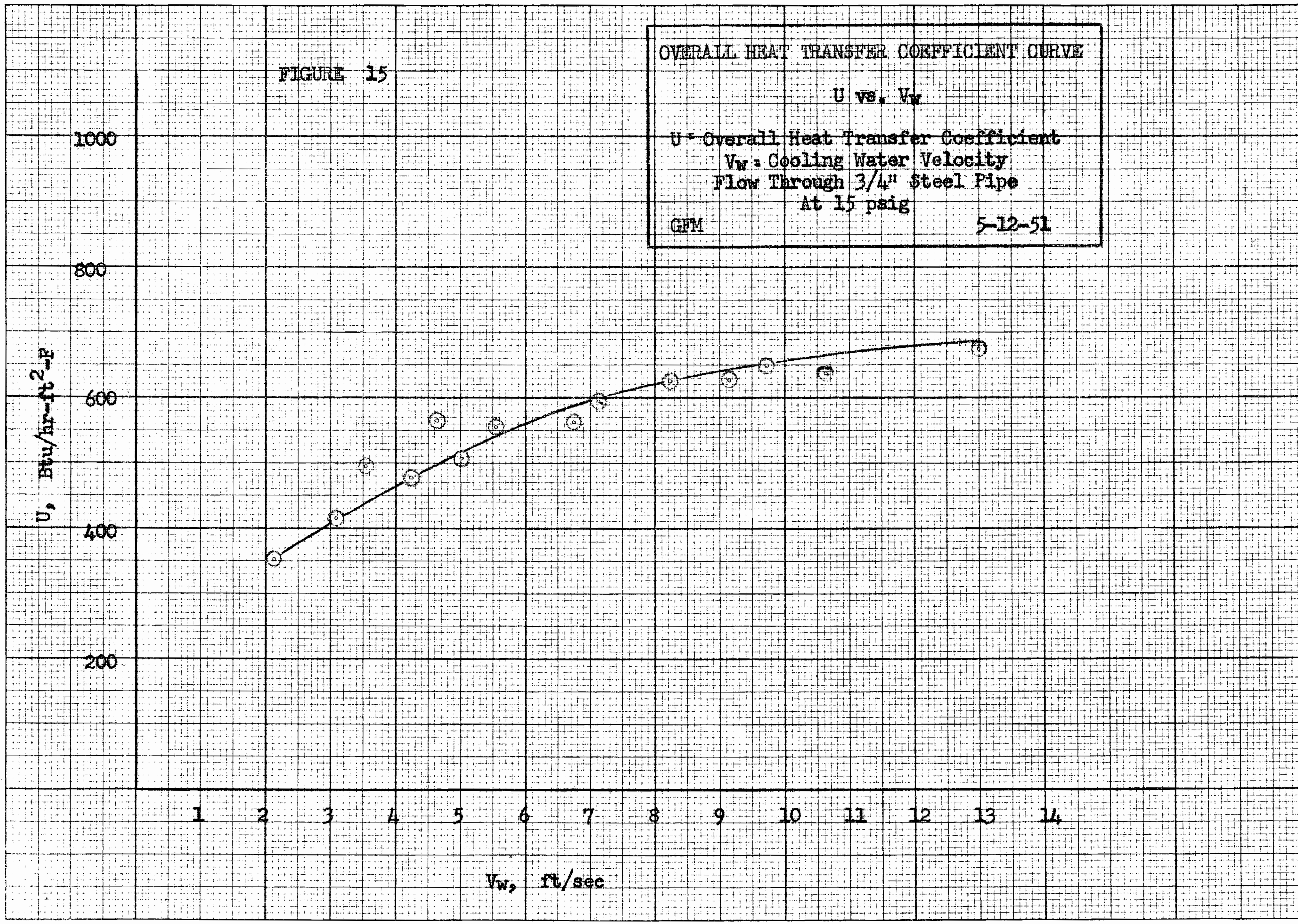
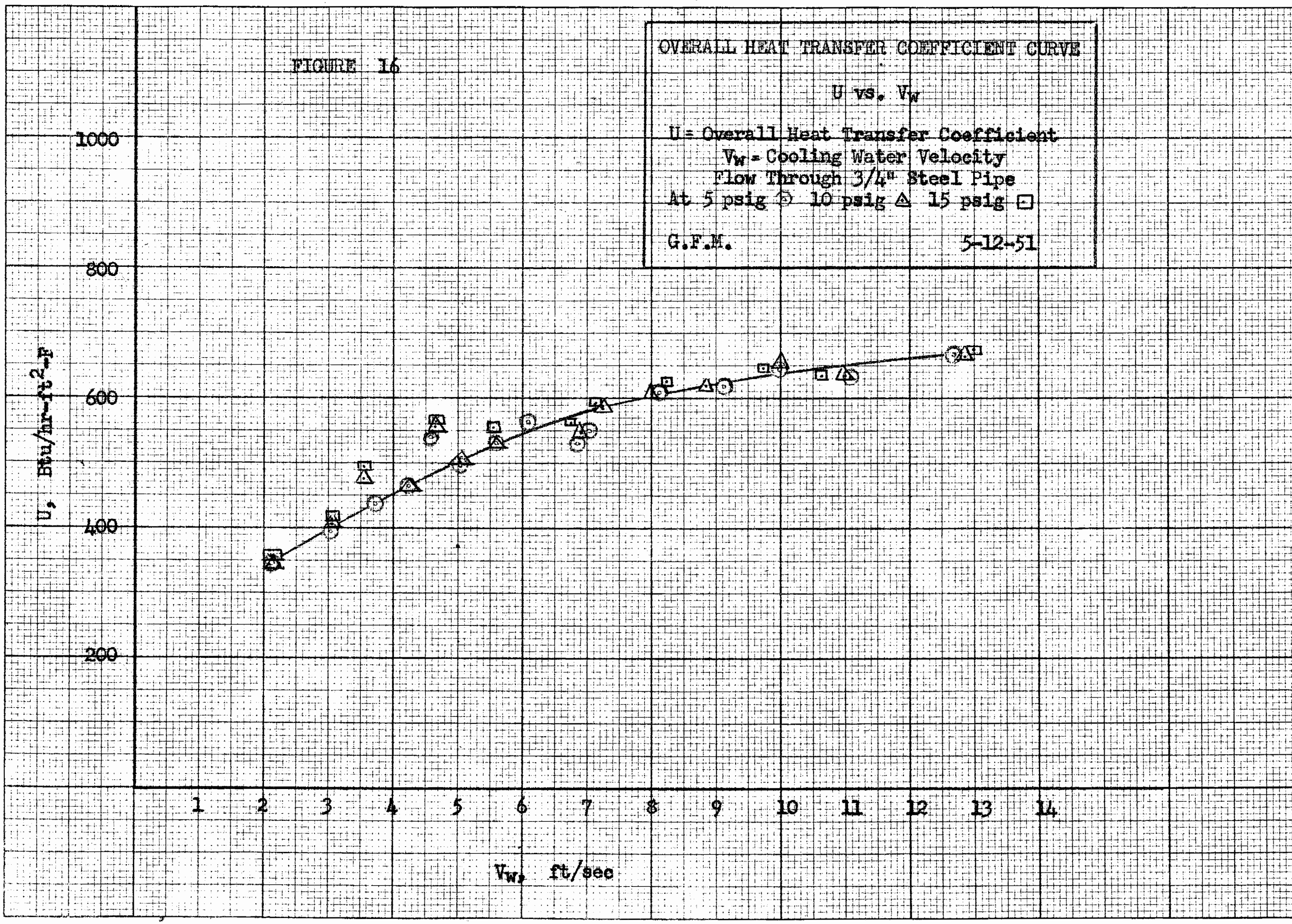


FIGURE 15
OVERALL HEAT TRANSFER COEFFICIENT CURVE
U vs. Vw
U = Overall Heat Transfer Coefficient
Vw = Cooling Water Velocity
Flow Through 3/4" Steel Pipe
At 15 psig
GFM
5-12-51
1000
800
600
400
200
U, Btu/hr-ft2-F
1
2
3
4
5
6
7
8
9
10
11
12
13
14
Vw, ft/sec

FIGURE 16


OVERALL HEAT TRANSFER COEFFICIENT CURVE
U vs. V_W
U = Overall Heat Transfer Coefficient
V_W = Cooling Water Velocity
Flow Through 3/4" Steel Pipe
At 5 psig ⊙ 10 psig △ 15 psig ⊡
G.F.M.
5-12-51
1000
800
600
400
200
U, Btu/hr-ft^2-F
1
2
3
4
5
6
7
8
9
10
11
12
13
14
V_W, ft/sec

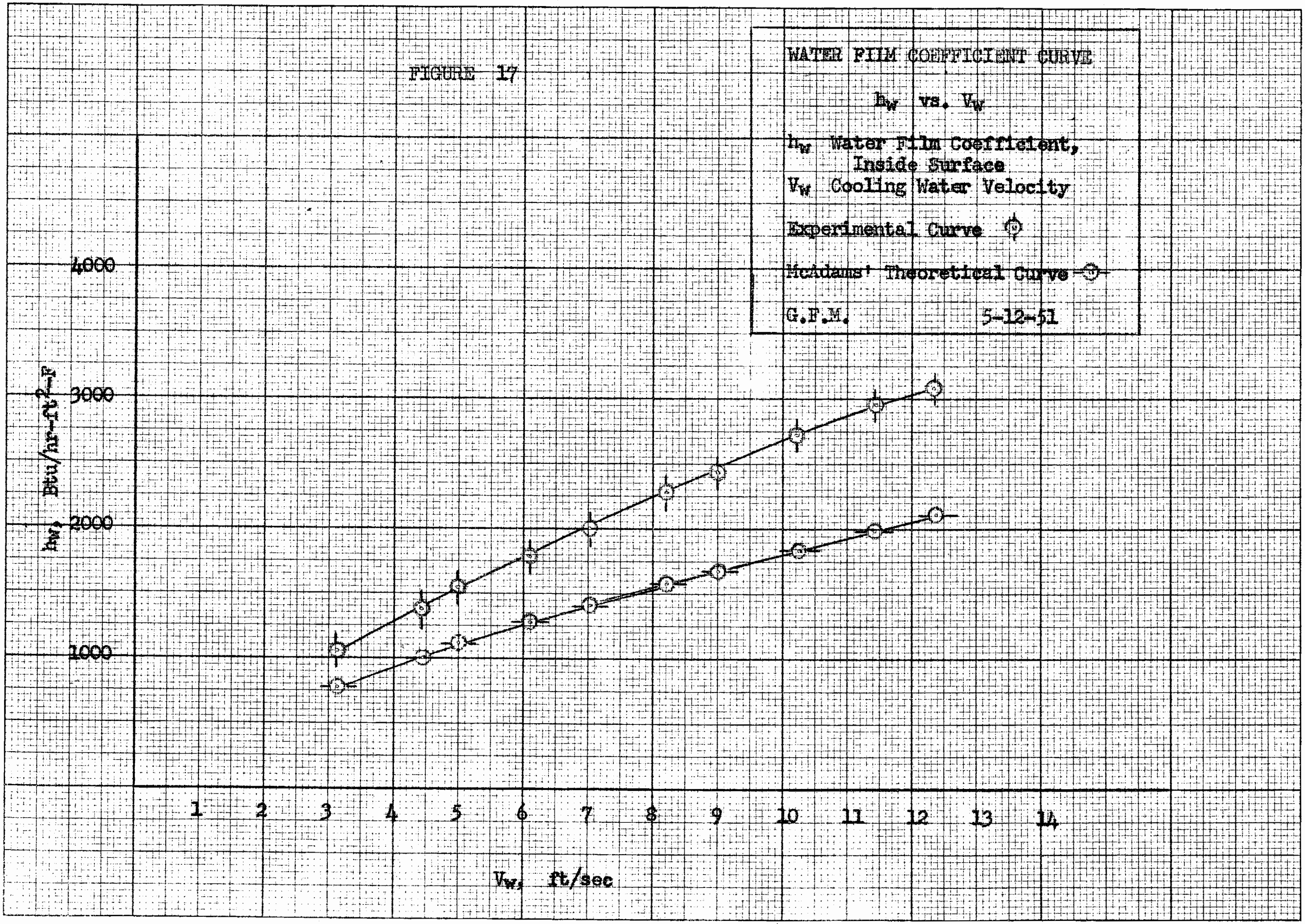
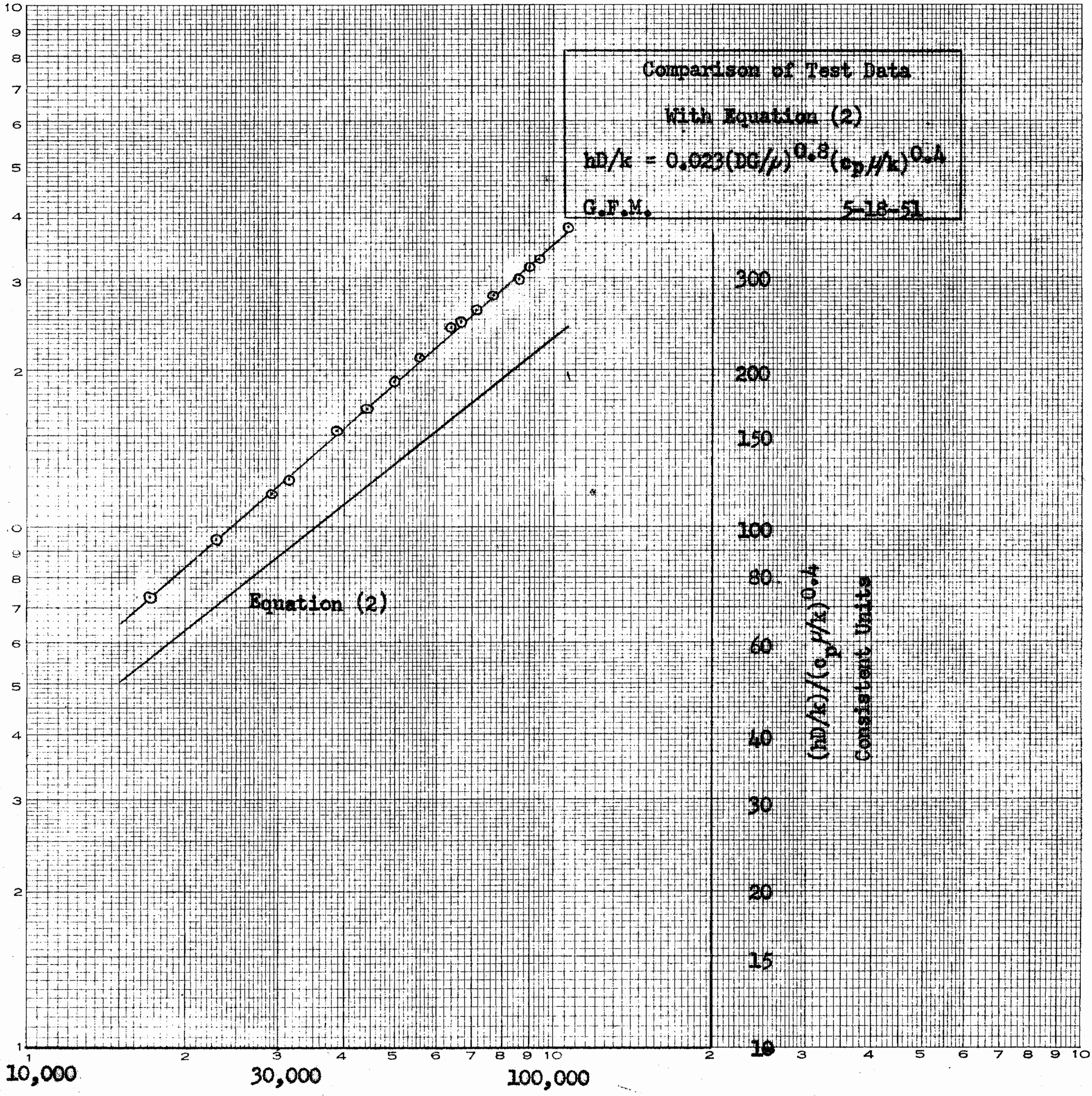

WATER FILM COEFFICIENT CURVE
h_W vs. V_W
h_W Water Film Coefficient, Inside Surface
V_W Cooling Water Velocity
Experimental Curve
McAdams' Theoretical Curve
G.F.M. 5-12-51
4000
3000
2000
1000
h_W, Btu/hr-ft^2-F
1
2
3
4
5
6
7
8
9
10
11
12
13
14
V_W, ft/sec

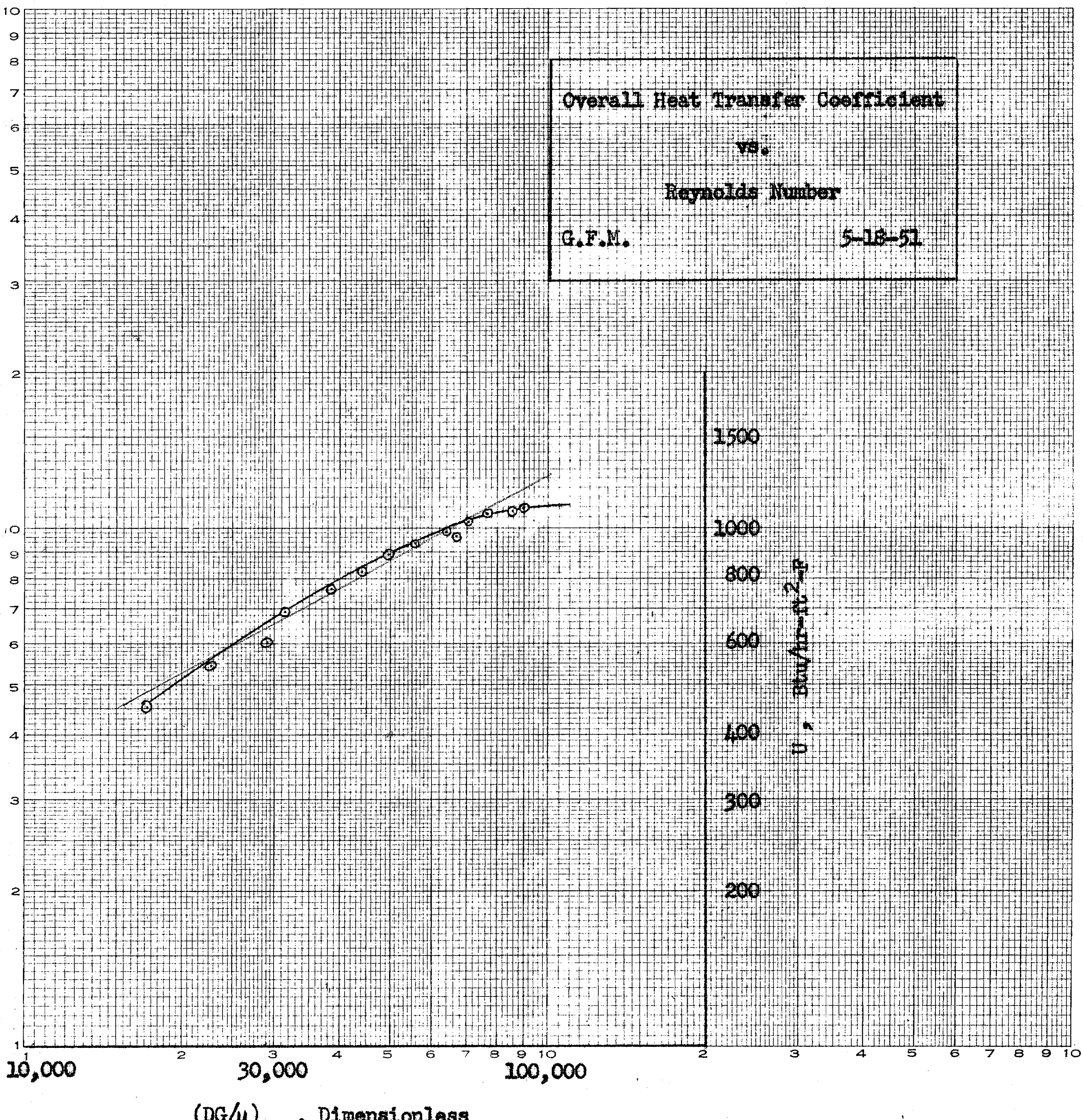
FIGURE 17

FIGURE 18



DG/μ , Consistent Units

FIGURE 19


Overall Heat Transfer Coefficient
vs.
Reynolds Number
G.F.M.
5-18-51
1500
1000
800
600
400
300
200
U , Btu/hr-ft.^{2}-F
10,000
30,000
100,000
(DG/μ) , Dimensionless

VII. DISCUSSION OF RESULTS

Upon completion of the tests on each pipe, the pipe was examined for evidence of scale or sludge deposits. They were found to be clean, so the assumption that the scale resistance was equal to zero was valid.

The overall heat transfer coefficient, U , was based on the heat gained by the water while flowing through the heat exchanger. Therefore, any error in the measurement of the incoming and outgoing water temperature would result in an incorrect value of this coefficient. Due to construction difficulties, it was impossible to place the thermometer wells used to measure the incoming and outgoing water temperature exactly at the entrance and exit of the heat exchanger. This resulted in an error in measurement of the water temperature due to radiation to or from the pipe. However, since the smallest temperature rise of the water was 25 degrees Fahrenheit, this error may be considered negligible.

The pressure and temperature of the steam in the heat exchanger were both recorded. The steam temperature recorded, and the saturation temperature corresponding to the absolute steam pressure agreed within two degrees Fahrenheit in every test. In computing the logarithmic mean temperature difference, the recorded steam temperature was used since the thermometer could be read more accurately than could the pressure gage in the range of operation.

The quality of the incoming steam was measured by a throttling calorimeter. When the steam pressure was 15 psi gage, this quality averaged 99 per cent. During the 10 psi gage and 5 psi gage runs, the steam was not superheated sufficiently to determine the quality. Therefore it was assumed that the quality of the steam entering the heat exchanger was

99 per cent for those runs too.

The condensed steam was measured by recording the water level in the hotwell. The hotwell that was constructed was too small. It should have been large enough to hold all of the steam condensed during the 15 minute test without being drained. It was necessary to drain this hotwell during the test. This meant that during the draining period some of the condensate was not collected. The time necessary to drain the hotwell was determined to be approximately 30 seconds. A correction factor was applied to the total scale reading to account for this lost condensate as shown in the Method of Calculation. During the latter part of the tests, it was discovered that the drain valve from the hotwell had been leaking. This meant that the calculated weight of steam condensed was low for all of the tests after the valve started to leak.

In comparing the heat gained by the water and the heat given up by the steam, the first ten runs indicated a radiation loss of one per cent. In the succeeding runs however, the heat gained by the water was greater than the heat given up by the steam in practically every case. This indicates that the drain valve started to leak after about the tenth run and caused these poor results. Since all of the calculations were based on the heat gained by the water the accuracy of the experimental results was not affected.

The curves plotted showed that as water velocity was increased the overall heat transfer coefficient, U , increased at a diminishing rate until a point was reached where there was no furthur appreciable increase in U with an increase in water velocity. In the two to fourteen feet per second range of water velocity U increased since at higher water velocities the water film resistance becomes smaller, thus allowing a larger amount of heat to flow from the steam to the water. At water velocities above 15 feet per second, U remained relatively constant. At this speed a particle of water remained in the heat exchanger for less than six-tenths of a second so there was insufficient time for the water to absorb more than a certain amount of heat. In the practical range of water velocities, from four to ten feet per second, the water velocity had a very appreciable affect on the overall heat transfer coefficient.

Test runs were taken at five, ten, and fifteen psi gage but the results indicated that the pressure in the heat exchanger had only a slight effect on the overall heat transfer coefficient. This range of steam pressures represents a steam temperature variation of about 20 degrees Fahrenheit. A slightly higher coefficient was obtained at the higher pressures. This can be attributed mainly to the higher temperature difference between the steam and water at higher pressures.

The value of U leveled off at about 1070 Btu/hr-ft^2-F when the copper tube was tested. The maximum coefficient obtained for the extra heavy steel pipe was 760 Btu/hr-ft^2-F. This lower value was to be expected since the steel pipe was thicker and the metal had a higher thermal resistance. A change in surface coefficients affected this latter value of U slightly.

The steam film coefficients determined varied from 1705 to 2215 Btu/hr-ft^2-F. The average of all of the steam film coefficients determined was 2085 Btu/hr-ft^2-F. As stated in the Review of Literature, an average design figure is 2000 Btu/hr-ft^2-F. The tests on the copper pipe indicated that the steam film decreased slightly at increased pressure. This coefficient remained relatively constant at all pressures when the steel pipe was used. Since the maximum variation from 2000 Btu/hr-ft^2-F was plus or minus 15 per cent, this value is an acceptable design figure.

Before the steel pipe was placed in the heat exchanger for testing, the protective lacquer was cleaned off and the pipe polished to a high finish with a buffing wheel. This led to some very interesting results. Immeadiately after the pipe was replaced in the heat exchanger, five test runs were taken. These test runs gave very high values for the overall heat transfer coefficient. The steam film coefficient was computed for these five runs and found to be 4200 Btu/hr-ft^2-F. This was more than twice the average steam film coefficient. This deviation can most logically be explained by assuming that dropwise condensation occured for these first five runs. Dropwise condensation normally gives steam film coefficients four to eight times the magnitude of those accompanying filmwise condensation, and usually occurs only on polished surfaces. It is logical to assume that by the next evening when the next series of runs was taken, the polish on the pipe had diminished to such a degree as to give the normally expected filmwise condensation. Check runs were taken at these velocities and pressures at a later date and the results from these check runs conformed to the trend of the normal curve as shown on Figure 12.

The equations determined for the water film coefficient ranged from 405 $V_w^{0.8}$ to 438 $V_w^{0.8}$ Btu/hr-ft^2-F, where V_w equals the average water velocity in feet per second, and the coefficients are based on the outside surface. The average value for all tests was 418 $V_w^{0.8}$ Btu/hr-ft^2-F.

The average water film coefficent for the copper tube based on the outside surface was 330 $V_w^{0.8}$ Btu/hr-ft^2-F. The extra heavy steel pipe gave a coefficient of 292,$V_w^{0.8}$ Btu/hr-ft^2-F when based on the outside surface. When these coefficients were corrected to inside surface area, they became 421 $V_w^{0.8}$ and 416 $V_w^{0.8}$ Btu/hr-ft^2-F , respectively. This close comparison between pipes of different material and thickness indicates that the expression 418 $V_w^{0.8}$ Btu/hr-ft^2-F accurately evaluates the water film coefficient when the water temperature ranges from 54 to 135 degrees Fahrenheit, and the Reynolds number lies between 17,000 and 100,000 , the limits of this thesis.

The water film coefficient expression determined by this experiment was compared with values obtained from equations (2) and (2a) , found in the Review of Literature. These equations may be used if the Reynolds Number exceeds 2100 as they did in this thesis.

Figure 17 shows the comparison with equation (2a). The test results range from 36 to 48 per cent higher than equation (2a).

Equation (2) may be written as:

$hD / k = a(DG/\mu)^{0.8}(c_p\mu/k)^{0.4}$, where the units are as defined in the Review of Literature. Mcadams suggests that the coefficient "a" in this equation equals 0.023 for liquids being heated inside of a pipe.[1] The experimental value of "a" was determined to be 0.33 from Figure 18. This value is 43 per cent higher than that suggested by McAdams.

These high values do not necessarily suggest that the results of

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1. McAdams, W. H. , HEAT TRANSMISSION, Pg. 169, McGraw Hill Book Company, New York, 1942

this thesis are in error. Other experimenters have reported coefficients much higher than those quoted by McAdams. A curve found in Heat Transmission, by McAdams on page 181 illustrates this. On the basis of these facts it is believed that the value of 0.023 for "a" is conservative.

VIII. CONCLUSIONS

1. The overall heat transfer coefficient for the flow of heat from condensing steam through metallic pipe to water flowing inside the pipe increases as the water velocity increases up to a certain point. Above this velocity, approximately 15 feet per second for a pipe with an internal diameter of 0.82 inches, there is no furthur appreciable increase in overall heat transfer coefficient with an increase in velocity for a pipe with a length over diameter ratio of 100 or less.

2. A steam film coefficient of 2000 Btu/hr-ft^2-F may be used for steam condensing outside of a horizontal pipe.

3. A water film coefficient of 418 $V_w^{0.8}$ Btu/hr-ft^2-F may be used for heating water inside of a pipe when the water temperature ranges between 54 and 135 degrees Fahrenheit and the Reynolds number lies between 17,000 and 100,000.

4. The steam film coefficient for a highly polished pipe is at least twice that for a pipe which is not polished.

IX. RECOMMENDATIONS

1. A hotwell large enough to hold all of the steam condensed in fifteen minutes should be placed on the heat exchanger.

2. As soon as the materials are available, copper and admiralty metal condenser tubes should be placed in the heat exchanger and tests conducted to determine if these thinner tubes would yield the same results as those given in this thesis.

3. An old pipe or condenser tube could be placed in the heat exchanger and the scale resistance determined by the methods used in this thesis.

X. ACKNOWLEDGEMENTS

The assistance and criticism by the thesis committee consisting of Assistant Professor H. P. Marshall, Associate Professor C. H. Long and Mr. R. K. Will of the Mechanical Engineering Department is greatly appreciated.

The author also wishes to thank those students in Advanced Mechanical Engineering Laboratory who aided in the construction of the equipment and the performance of tests.

The help and advice of Mr. Frank Grissim in the construction of certain parts of the heat exchanger was invaluable.

XI. BIBLIOGRAPHY

1. Gaffert, G. A. , STEAM POWER STATIONS, McGraw Hill Book Company, New York, 1946.

2. Keenan, J. H. and Keyes, F. G. , THERMODYNAMIC PROPERTIES OF STEAM , John Wiley and Sons, Inc. , New York, 1936.

3. Marks, L. S. , MECHANICAL ENGINEER'S HANDBOOK , McGraw Hill Book Company, New York, 1941

4. McAdams, W. H. , HEAT TRANSMISSION , McGraw Hill Book Company, New York, 1942

5. Orrok, G. A. , TRANSACTIONS OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS , Volume 32, 1910, American Society of Mechanical Engineers, New York, 1910

6. Stoever, H. J. , APPLIED HEAT TRANSMISSION , McGraw Hill Book Company, New York, 1941

XII. VITA

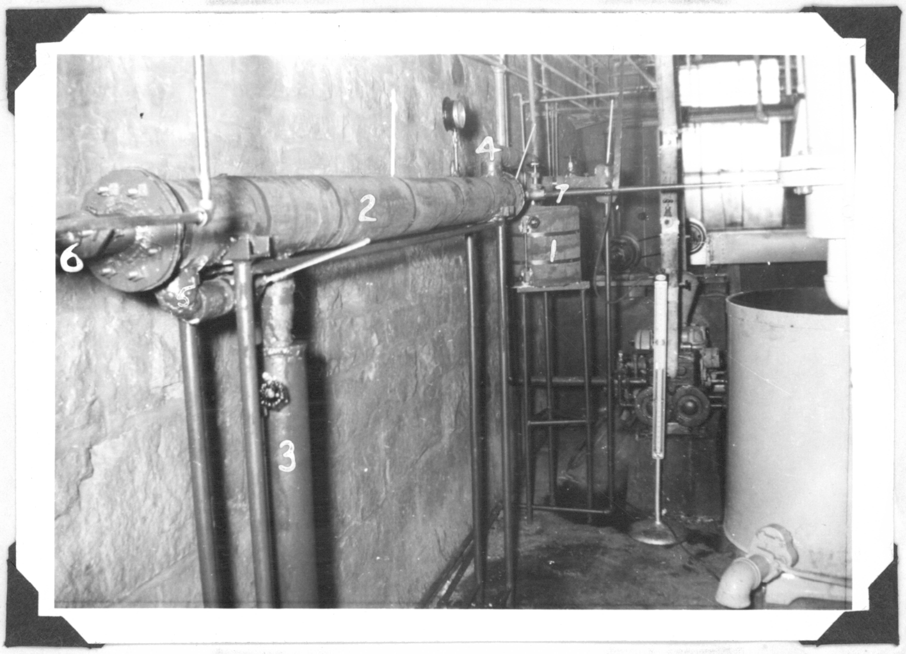
George Franklin Moore was born in Wilmington, Delaware on August 7, 1927. He attended elementary and junior high school in Wilmington and graduated from the Pierre S. duPont High School in the same city in January 1945.

Soon after graduation, he entered the Navy and was discharged in November of the same year. He entered the University of Delaware in January 1946 and completed the requirements for the degree of Bachelor of Mechanical Engineering in January 1949. He was awarded this degree with honors in June 1949.

From January until June 1949, he held the position of instructor in Mechanical Engineering at the University of Delaware. He began work for his graduate degree at this time. He accepted a position as Instructor in Mechanical Engineering at Virginia Polytechnic Institute in the summer of 1949. During the summer of 1950 he worked for the Babcock and Wilcox Company as a service engineer. He remained at Virginia Polytechnic Institute until he completed the requirements for the degree of Master of Science in Mechanical Engineering in June 1951.

XIII. APPENDIX

PHOTOGRAPH OF EQUIPMENT



1. Desuperheater
2. Heat Exchanger
3. Hotwell
4. Steam in
5. Condensate out
6. Water in
7. Water out

HEAT EXCHANGER TEST DATA

			TEMPERATURE, F						PRESSURE			WEIGHT	
Test Number	Barometer in.Hg.	Date	Water In	Condensate	Hot Well	Steam	Water Out	Calorimeter	Calorimeter in.Hg.	Water Corr. in. H2O	Steam psig	Cooling H2O #/ 15 min.	Condensate in./ 15 min.
1	28.12	4-17-51	57.0	223.5	217.8	224.3	130.0	208	0.25	0.0	5.42	411	69
2	28.12	4-17-51	56.0	222.3	219.5	223.0	122.0	208	0.30	0.0	4.77	573	85
3	28.12	4-17-51	55.5	222.5	220.2	224.2	114.5	208	0.35	0.0	5.00	739	108
4	28.07	4-18-51	56.8	222.5	221.6	223.5	117.2	208	0.10	0.0	5.30	808	108
5	28.07	4-18-51	56.7	223.8	223.2	224.8	110.2	208	0.30	1.0	5.60	1048	124
6	28.07	4-18-51	56.0	222.7	222.2	223.8	106.3	208	0.32	1.0	5.07	1202	132
7	28.07	4-18-51	55.8	222.8	222.3	223.4	103.9	208	0.50	1.0	5.43	1388	142
8	28.07	4-18-51	55.4	223.3	223.2	224.8	100.4	208	0.70	1.0	5.58	1593	150
9	27.98	4-19-51	54.6	223.5	223.2	224.8	94.4	207	0.80	2.0	5.45	1932	154
10	27.98	4-19-51	54.6	224.0	223.8	225.0	93.0	209	1.00	2.5	5.35	1990	150
11	27.97	4-19-51	54.5	222.8	222.5	224.2	92.7	212	1.50	2.1	5.36	2146	162
12	27.97	4-19-51	54.2	222.0	221.6	223.3	89.4	209	1.73	2.5	5.09	2407	161
13	27.97	4-19-51	54.0	222.8	222.5	224.2	86.1	212	1.50	2.1	5.56	2711	165
14	27.97	4-19-51	54.0	222.0	221.6	223.3	84.5	209	1.68	2.4	4.99	2885	174
15	28.19	4-20-51	54.9	221.8	222.0	223.6	82.8	208	2.30	3.3	5.01	3016	169
16	28.19	4-20-51	54.8	223.2	223.2	224.3	78.7	210	2.40	3.3	5.30	3744	176
17	28.19	4-20-51	56.8	234.0	233.8	235.2	116.5	208	2.50	3.3	9.90	821	107
18	28.19	4-20-51	56.4	234.3	234.0	236.1	113.4	208	2.40	3.3	9.90	1012	117
19	28.19	4-20-51	56.0	234.3	233.6	236.1	109.6	211	2.40	3.5	10.60	1176	134
20	28.22	4-21-51	56.0	234.0	234.0	234.8	104.8	207	2.60	3.6	10.10	1437	144
21	28.22	4-21-51	56.0	234.5	234.5	235.8	104.7	207	2.50	3.9	10.25	1418	141
22	28.22	4-21-51	55.2	234.0	235.3	235.3	95.1	211	2.50	3.9	10.20	2133	170
23	28.22	4-21-51	55.0	234.0	234.0	235.4	93.8	211	2.50	3.4	10.40	2222	175

HEAT EXCHANGER TEST DATA

Test Number	Barometer in.Hg.	Date	TEMPERATURE, F Water In	Condensate	Hot Well	Steam	Water Out	Calorimeter	PRESSURE Calorimeter in. Hg.	Water Corr. in. H20	Steam psig	WEIGHT Cooling H20 #/ 15 min.	Condensate in./ 15 min.
24	28.22	4-21-51	55.0	234.0	234.0	235.7	91.3	211	2.5	3.9	10.4	2387	173
25	28.09	4-22-51	55.8	234.5	234.0	236.1	96.1	207	2.5	4.5	10.25	1914	163
26	28.09	4-22-51	55.2	223.5	233.0	235.7	88.1	212	2.6	3.5	10.00	2645	177
27	28.09	4-22-51	55.0	233.3	233.0	235.5	84.6	212	2.6	4.6	10.30	2830	173
28	28.09	4-22-51	55.0	233.2	233.0	235.5	81.4	213	2.6	4.6	9.92	3223	174
29	28.09	4-22-51	55.0	234.0	234.0	235.5	78.4	213	2.6	4.6	10.20	3855	185
30	28.25	4-24-51	56.8	223.7	222.0	225.3	109.5	207	2.5	4.7	5.01	889	93
31	28.25	4-24-51	55.2	222.5	222.0	224.6	93.6	207	2.6	4.8	5.14	1819	134
32	28.25	4-24-51	55.0	224.0	224.0	226.0	80.1	207	2.6	4.8	5.95	3256	171
33	28.24	4-24-51	55.0	232.7	233.2	235.3	80.8	208	2.6	4.8	9.90	3433	173
34	28.24	4-24-51	55.0	224.8	223.0	226.8	82.2	207	2.6	4.8	6.19	3374	176
35	28.24	4-24-51	55.0	234.2	234.5	237.0	82.2	214	2.6	4.8	10.90	3423	185
36	28.24	4-24-51	57.0	244.8	243.5	246.2	126.7	212	2.6	4.8	14.50	638	96
37	28.24	4-24-51	57.0	233.7	232.5	235.0	121.9	210	2.6	4.8	9.90	638	86
38	28.17	4-25-51	58.5	235.0	235.7	235.7	115.0	209	0.5	5.1	10.30	872	107
39	28.17	4-25-51	58.3	244.3	244.3	246.5	118.3	215	0.6	6.4	14.80	895	112
40	28.17	4-25-51	58.0	245.2	245.0	246.7	114.6	216	0.6	6.4	14.70	1023	118
41	28.17	4-25-51	57.4	245.2	244.5	247.4	109.6	216	0.7	6.4	15.03	1250	129
42	28.17	4-25-51	57.0	245.2	245.3	247.5	105.0	216	0.7	6.4	14.93	1443	144
43	28.17	4-25-51	56.8	244.5	245.0	247.2	101.4	216	0.7	6.5	15.32	1682	152
44	28.22	4-26-51	56.7	245.0	244.8	247.5	98.0	215	1.1	7.3	15.33	1844	157
45	28.22	4-26-51	56.5	244.7	245.0	246.9	94.2	217	1.3	7.2	14.80	2104	166
46	28.22	4-26-51	56.2	244.7	244.7	248.1	92.2	218	1.3	7.3	15.50	2328	160

HEAT EXCHANGER TEST DATA

			TEMPERATURE, F						PRESSURE			WEIGHT	
Test Number	Barometer in. Hg	Date	Water In	Condensate	Hot Well	Steam	Water Out	Calorimeter	Calorimeter in. Hg.	Water Corr. in. H2O	Steam psig	Cooling H_2O #/ 15 min.	Condensate in./15 min.
47	28.22	4-26-51	56.1	244.3	245.0	247.5	89.9	219	1.60	7.8	15.6	2540	176
48	28.22	4-26-51	56.1	244.2	244.0	247.5	88.4	218	1.65	7.6	15.5	2717	162
49	28.22	4-26-51	56.1	243.7	243.8	247.0	86.0	218	1.70	7.6	15.3	2941	177
50	28.21	5- 1-51	60.1	222.8	221.6	224.0	111.0	208	0.70	7.9	5.1	617	74
51	28.21	5- 1-51	60.4	233.5	233.0	235.6	122.0	208	1.10	8.0	9.8	581	81
52	28.21	5- 1-51	60.7	244.5	244.0	247.1	127.7	211	1.30	8.0	15.0	587	88
53	28.21	5- 1-51	59.8	222.5	222.0	224.1	111.0	210	1.30	8.0	5.0	761	87
54	28.21	5- 1-51	59.9	234.5	233.0	236.3	116.0	210	1.50	8.1	10.4	770	96
55	28.21	5- 1-51	60.3	244.7	244.2	246.7	120.3	214	1.70	8.1	14.8	769	103
56	27.98	5- 2-51	59.0	224.3	223.5	225.6	102.2	208	0.90	9.4	6.3	929	90
57	27.98	5- 2-51	59.0	233.0	232.5	235.7	105.1	208	1.20	9.6	10.1	928	91
58	27.98	5- 2-51	59.0	244.0	243.5	246.0	109.2	209	1.40	9.7	14.5	923	103
59	27.98	5- 2-51	58.6	221.0	219.0	224.2	94.7	210	1.80	9.9	5.0	1135	87
60	27.98	5- 2-51	58.6	234.5	233.0	236.8	98.4	210	2.00	10.0	10.6	1144	100
61	27.98	5- 2-51	58.5	243.5	243.0	246.2	102.1	210	2.20	9.2	14.7	1122	108
62	27.74	5- 3-51	58.0	223.5	223.0	224.5	94.6	210	0.65	5.6	5.3	1174	97
63	27.74	5- 3-51	58.0	233.7	233.0	234.8	97.9	212	0.80	5.6	10.0	1206	102
64	27.74	5- 3-51	58.0	243.5	243.0	245.2	101.5	214	1.00	5.6	14.3	1180	107
65	27.74	5- 3-51	57.6	223.0	222.0	224.0	93.0	210	1.00	5.6	5.4	1338	103
66	27.74	5- 3-51	57.6	234.0	233.0	234.8	95.7	212	1.50	5.6	10.0	1325	108
67	27.74	5- 3-51	57.6	242.5	242.0	244.2	97.5	214	1.30	5.6	14.2	1371	116
68	27.74	5- 3-51	58.0	223.2	222.0	224.2	100.3	210	1.30	5.6	5.4	1011	92
69	27.62	5- 4-51	56.7	223.0	222.0	224.1	88.9	207	0.70	6.8	5.4	1517	104

HEAT EXCHANGER TEST DATA

			TEMPERATURE, F						PRESSURE			WEIGHT	
Test Number	Barometer in. Hg.	Date	Water In	Condensate	Hot Well	Steam	Water Out	Calorimeter	Calorimeter in. Hg.	Water Corr. in. H2O	Steam psig	Cooling H2O #/ 15 min.	Condensate in./ 15 min.
70	27.62	5-4-51	56.5	233.0	232.7	234.3	92.0	207	0.80	6.8	9.9	1464	107
71	27.62	5-4-51	56.5	243.0	242.0	245.0	93.5	209	1.10	8.0	14.2	1520	115
72	27.62	5-4-51	56.5	223.5	222.5	224.3	87.5	210	1.30	9.7	5.6	1661	107
73	27.62	5-4-51	56.9	233.0	232.7	234.6	90.0	212	1.50	9.8	10.1	1655	111
74	27.62	5-4-51	57.2	242.7	243.0	244.7	92.8	214	1.70	9.8	14.2	1615	116
75	27.64	5-5-51	56.5	222.0	221.0	223.4	84.0	208	0.85	4.9	4.9	1844	104
76	27.64	5-5-51	56.4	233.5	233.2	234.8	86.4	208	0.90	9.5	10.2	1823	112
77	27.64	5-5-51	56.4	242.3	242.0	244.0	88.7	212	1.00	9.5	14.0	1770	117
78	27.64	5-5-51	56.2	222.7	222.5	224.2	81.9	210	1.05	9.5	5.4	2107	107
79	27.64	5-5-51	56.2	234.0	234.0	235.4	83.3	212	1.25	9.6	10.3	2134	116
80	27.64	5-5-51	56.2	243.0	243.0	244.5	84.8	214	1.45	9.7	14.3	2153	123
81	27.73	5-5-51	59.8	223.7	221.7	224.2	125.9	208	2.00	10.2	5.0	357	53
82	27.73	5-5-51	59.5	235.0	232.5	236.3	130.7	207	2.25	10.3	10.6	356	59
83	27.73	5-5-51	59.5	243.0	241.0	244.5	135.2	208	2.50	10.5	14.3	352	65
84	27.73	5-5-51	58.7	222.5	221.0	223.7	114.7	208	2.60	10.5	4.8	504	65
85	27.73	5-5-51	58.6	234.0	232.0	235.7	118.9	208	2.65	9.6	10.5	514	70
86	27.73	5-5-51	58.6	242.5	242.0	244.3	122.8	208	2.80	10.6	14.1	512	70
87	27.96	5-6-51	58.0	223.0	222.0	223.7	106.7	208	0.80	9.5	5.2	705	75
88	27.96	5-6-51	58.0	232.0	232.0	233.9	110.1	208	1.10	9.6	9.4	699	81
89	27.96	5-6-51	58.0	243.5	243.3	245.3	114.1	208	1.30	9.7	14.4	704	88
90	27.96	5-6-51	58.0	222.0	222.0	223.2	102.5	210	1.30	9.7	4.9	833	82
91	27.96	5-6-51	58.0	232.7	231.0	234.1	105.8	209	1.70	9.9	9.8	831	87
92	27.96	5-6-51	58.0	244.0	243.0	245.7	109.5	213	1.90	10.0	14.5	830	91

ABSTRACT

The object of this thesis was to determine water film coefficients and condensing steam film coefficients for a single tube heat exchanger.

A shell and tube apparatus was constructed and these coefficients were determined by Wilsons graphical method. Test runs were made at various pressures and water velocities.

It was determined that for flow through a horizontal tube the water film coefficient closely approximates 416 $V_w^{0.8}$ Btu/hr-ft^2-F, where V_w equals water velocity in feet per second, and the Reynolds number lies between 17,000 and 100,000.

It was also found that an average condensing steam film coefficient for filmwise condensation was 2000 Btu/hr-ft^2-F. It was discovered that this coefficient is much higher if the condensing surface is highly polished.