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ALLOWABLE TEMPERATURE RISE IN TUBES OF THE PILES: PRECAUTIONS AGAINST BOILING

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September 27, 1945

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TO: W. E. JORDAN

FROM: W. K. WOODS

## ALLOWABLE TEMPERATURE RISE IN TUBES OF THE PILES: PRECAUTIONS AGAINST BOILING


**Summary:**

In design of the pile it was considered advisable never to impose so great a heat load on any tube that the available header pressure would be insufficient to sweep the tube free of vapor if boiling should accidentally be initiated in the tube. In the face of this restriction, the following maximum temperature rise (°C.) should be adhered to:

| Header Pressure lbs./sq. in. | Orifice Dia. - In. | 0.240 | 0.200 | 0.175 | 0.140 |
|---|---|---|---|---|---|
| 350 | | 65 | 76 | 88 | -- |
| 200 | | 48 | 56 | 65 | 85 |
| 100 | | 33 | 39 | 45 | 62 |

The above figures are based upon tubes containing solid cylindrical dummy slugs and film sufficient to cause a 40 lb./sq. in. change in pressure drop at 20.4 g.p.m., and provided with water at a temperature of 20°C. The limiting temperature rise may be increased by 3°C. when the inlet water temperature is lowered from 20°C. to 5°C., and may be increased by an additional 3-9°C. for film-free tubes containing low resistance perforated aluminum dummy slugs (see Figure 2). The header pressure referred to is the pressure as measured in the control room.


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**Discussion:**

The flow characteristics for tubes equipped with two different sizes of orifices and operated at two different heat loads are shown in Figure 1. These curves were computed by methods described in the Appendix. The straight-line portion of the curves in the region of high flow rate represent the pressure drop encountered when no vaporization occurs in the tube. The high pressure drops encountered at low flow rates are caused by the partial vaporization of water in the tube. At low flow rates the curves for the two different orifice sizes are practically superimposed, for the resistance of the orifice becomes negligible in comparison with the resistance of the tube.

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Boiling in the pile is particularly objectionable because of the extreme sensitivity of the pile reactivity to variation in the weight of water in the pile. Incidental objections involve accelerated corrosion and the possibility of mechanical damage resulting from water hammer effects. Unstable conditions would be encountered during operation under conditions where the slope of the characteristic curve is negative and there would exist a danger of "burning out" tubes in the pile. To assure against any possibility of establishing boiling in the pile, the heat load on any tube (as measured by rise in water temperature through the tube) should be limited to a value such that the highest pressure drop indicated by the boiling branch of the corresponding characteristic curve is less than the available pressure drop. Thus, for the tube described in Figure 1 (low inlet water temperature, high resistance dummy slugs, and considerable film) provided with a 0.240 inch orifice the temperature rise should not exceed 68°C. when the available pressure drop is 329 lbs./sq. in., and should not exceed 36°C. when the header pressure is 81 lbs./sq. in. The maximum pressure drop indicated by the boiling branch of the characteristic curve has been arbitrarily limited to 77% of the available pressure drop. The header pressure as measured in the control room is 18 lbs./sq. in. greater than the corresponding pressure drop.

For low resistance tubes (film-free and containing perforated dummy slugs) the entire characteristic curve is lower than that shown in Figure 1. Hence, for a given heat load the required header pressure is lowered but the associated flow rate and temperature rise is substantially unchanged from that indicated by Figure 1. For example, a low resistance tube equipped with a 0.240 inch orifice may be permitted a temperature rise of 66°C. when the available pressure drop is only 281 lbs./sq. in.

For a given heat load, an increase in inlet water temperature raises the boiling branch of the characteristic curve (because of the increase in amount of vapor at a given flow rate) but lowers the non-boiling branch of the curve (because of the decrease in liquid viscosity at the higher temperature levels). Both effects are small, but the fact that they are in opposite directions makes the effect noticeable.

The above results are shown graphically in Figure 2, together with additional data for one intermediate heat load and the two intermediate orifice sizes.

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APPENDIX I

METHOD OF CALCULATION

Reference A - Woods and Worthington to Greenewalt, April 1, 1943
Reference B - Doc. 7-2092 - Woods to Squires, July 16, 1945

The general method of analysis of pressure drop for steam-water mixtures flowing through the tubes of the pile is discussed in detail in the report "Boiling in the Pile", Woods and Worthington to Greenewalt, April 1, 1943. This appendix is intended primarily to serve as a record of the numerical figures used in the calculations.

The overall pressure drop for non-boiling conditions was determined by the methods of Document 7-2092, - in particular, by use of the equation given in Appendix I of that report.

For boiling conditions, the general method of attack involves starting at known outlet conditions and working in a step-wise manner back through the tube. The rate of heat generation was assumed to be expressible by the following equation:

$$q = 0.5 + 0.505 \sin (7.06 \ell)^{\circ} \qquad (1)$$

where q is the cumulative heat generation up to any point $\ell$ in the active zone of the tube, expressed as a fraction of the total heat generation; and $\ell$ is the distance in feet from the mid-point of the tube, taken as positive in a direction downstream from the mid-point. (Note: This equation is derived from assuming that the active zone contains 32 slugs and assuming a 35 cm. reflector.)

Conditions at the Outlet End of the Pig-Tail

Except at relatively low heat loads, critical conditions exist at the outlet end of the pig-tail, as discussed in reference A. These conditions may be rapidly evaluated by means of the attached Figure 3, which is reproduced from a Technical Division, Engineering Department report. Since the inside diameter of the pig-tail is 0.550 inch, the abscissa value of Figure 3 is obtained from the equation:

$$G_c = 84 \text{ (g.p.m.)} \qquad (2)$$

The critical pressure as obtained from Figure 3 is taken to be the pressure at the outlet end of the pig-tail as long as it is greater than 22 lbs./sq. in. abs.; otherwise the pressure is assumed to be 22 lbs./sq. in. abs. This value is arrived at by considering the uppermost tubes of the pile, which are subjected to a static head of 17 feet of water above atmospheric pressure. These tubes are about 30 feet above the gages in the control room; the gage pressure in the control room is obtained by adding 13 lbs./sq. in. static head and subtracting 15 lbs./sq. in. barometric pressure, for a net correction of minus 2 lbs./sq. in.

Pressure Drop through the Pig-Tail

For boiling conditions, most of the pressure drop through the pig tail is attributable to kinetic energy effects. The effect of wall traction in the

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spiral pig-tail is known only approximately, so the effect of variation in Reynolds Number is neglected. It should be noted that the minimum cross-sectional area occurs in the pig-tail itself, rather than adjacent to the thermocouple in the outlet nozzle, - else critical conditions would be encountered near the thermocouple.

Taking the inside cross-sectional area of the pig-tail equal to 0.237 sq. in., the kinetic energy in the pig-tail is given by the expression:

$$K = (1/778)\ (V^2/2g) = 7.3\ (Wv)^2 \quad (3)$$

Where K is the kinetic energy, B.T.U./lb.; W is the flow rate, lbs./sec., and v is the average specific volume, cu. ft./lb.



The pressure drop in the pig-tail is given by the relation:

$$\Delta P = R\ W^2\ v_{av} + 78.8\ W^2\ \Delta v \quad (4)$$

Where $\Delta P$ is the pressure drop in lbs./sq. in.

The pressure drop due to kinetic energy changes on entrance to the pig-tail is given by the relation:

$$\Delta P' = 31.2\ W^2\ v_{av} \quad (4a)$$

The quantity R is then evaluated by using the data of reference B, obtaining:

$$R\ W^2/62.3 + 31.2\ W^2/62.3 = 0.060\ (60\ W/8.33)^2;\ R = 163 \quad (4b)$$

<u>Pressure Drop in Inactive Zone of the Tube; Case A. Solid Dummy Slugs</u>

The kinetic energy in the annulus (area = 0.351 sq. in.) is given by the relation:

$$K = 3.36\ (Wv)^2 \quad (5)$$

This quantity is usually negligible because of the low values for specific volume of the steam-water mixture upstream from the pig-tail.

The pressure drop in the annulus is given by the relation:

$$\Delta P = R\ W^2\ v_{av}\ N + 36.2\ W^2\ \Delta v \quad (6)$$

The quantity R is then evaluated by using the data of reference B, obtaining for clean slugs:

$$R\ W^2\ 40.3/62.3 = 0.662 \times 1.320\ (Z)^{0.2}\ (60\ W/8.33)^2$$

$$R = 47.1\ (Z/W)^{0.2} \quad (7A)$$

Hence:

$$\Delta P = 47\ (Z)^{0.2}\ (W)^{1.8}\ v_{av}\ N + 36.2\ (W)^2\ \Delta v \quad (6A)$$

Where Z is the viscosity in centipoise and N is the length of the inactive zone, taken as 8.6 feet for each end of the tube.

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Pressure Drop in Inactive Zone; Case B. Perforated Dummy Slugs

This case is handled in the same manner as the preceding section except for the assignment of a fixed (and different) value to R N of equation 6.

As reported in reference B, $r_o = 1.32$ for a full column of slugs of which about 57% is due to the active zone. For a column containing an active zone plus perforated dummy slugs (in combination, of course, with jacketed lead slugs), $r_o = 1.00$ and $r_o$ for the dummy slugs is about equal to $1.00 - 0.57 \times 1.32 = 0.25$.



Hence, evaluating R N in equation 6:

$$(R\ N)\ W^2/62.3 = 0.25 \times 0.662 \times (60\ W/8.33)^{1.8}\ (Z)^{0.2} \qquad (7B)$$

$$R\ N = 360\ (Z/W)^{0.2} \text{ for both ends, or } 180\ (Z/W)^{0.2} \text{ for each end.}$$

Hence:

$$\Delta P = 180\ (Z)^{0.2}\ (W)^{1.8}\ (v_{av}) + 36.2\ W^2\ (\Delta v) \qquad (6B)$$

Pressure Drop in Active Zone

For a film-free active zone, equation 5 - 6A are applicable.

For a fouled tube it is assumed arbitrarily that the film is distrubuted uniformly along the length of the tube and the change in cross-section is neglected insofar as its effect on the kinetic energy term of equations 5 or 6. Hence, in accordance with the data of reference B, the value of R given in equation 7A is increased by the factor: $1 + (F/150)/(0.57)\ (1.32)$. For a value of F equal to 40 lbs./sq. in., this factor becomes 1.35, and

$$\Delta P = 63.5\ (Z)^{0.2}\ (W)^{1.8}\ v_{av}\ N + 36.2\ (W)^2\ \Delta v \qquad (6C)$$

Treatment of Viscous Flow

At very low flow rates and in the upstream portion of the tube before boiling has commenced, streamline flow may be encountered. In this case the pressure drop is computed by the above formulas for turbulent flow and the result is then multiplied by the factor:

$$(\Delta P)_s/(\Delta P)_t = (Re)_c^{0.8}/(Re)^{0.8} \qquad (8)$$

in which Re is the Reynolds Number based on the hydraulic diameter, subscripts s and t refer to streamline and turbulent flow, respectively, and subscript c refers to the critical Reynolds Number at which the value of the factor becomes unity. The quantity, Re, is equal to 7800 W/Z (based on a hydraulic diameter of 0.153 inches) and $(Re)_c$ is about equal to 2500 according to Appendix VI of Document No. 3-2567.

Hence:

$$(\Delta P)_s \simeq (0.321\ Z/W)^{0.8} (\Delta P)_t \qquad (8A)$$

to be used when Z (viscosity in centipoise) is greater than 3.12 W.

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**Determination of Physical Properties**

The weight fraction of water vaporized at any point under any pressure is determined by the equation:

$$(y)\,(h)_{fg} = (h)_o + Q - (h)_f - K \qquad (9)$$

Where $y$ = weight fraction of water vaporized
$(h)_{fg}$ = enthalpy of vaporization at the specified pressure, B.T.U./lb.
$(h)_o$ = enthalpy of water at inlet temperature, B.T.U./lb.
$(h)_f$ = enthalpy of water saturated at the specified pressure, B.T.U./lb.
$Q$ = heat added to water during passage through tube up to the specified point, B.T.U./lb.
$K$ = kinetic energy of mixture as given by equations 3 or 5.

Values of h were determined from "The Thermodynamic Properties of Steam", by Keenan and Keyes.

The specific volume of a vapor-liquid mixture was determined by the relation:

$$V = (y)\,(V)_g + (1-y)\,(V)_f \qquad (10)$$

Where subscripts g and $f$ refer to saturated vapor and liquid at the specified pressure.

The effective viscosity of a vapor-liquid mixture is given by the relation:

$$(1/Z) = (y)\,(1/Z)_g + (1-y)\,(1/Z)_f \qquad (11)$$

**APPENDIX II**

**SAMPLE CALCULATION**



Basis: Heat load = 10,000 C.H.U./min. = 316 KW

Flow rate = 2 g.p.m. of water at 5°C (41°F)

$W = 2 \times 8.33/60 = 0.278$ lbs./sec.
$Q = 1.8 \times 10{,}000/60 \times 0.278 = 1080$
$(h)_o = 9$ B.T.U./lb.

Abscissa value for Figure 3 = 84 x 2 = 168 (by equation 2)
Ordinate value for Figure 3 = 1080 + 9 = 1089
Critical pressure (by Figure 3) = 40 lbs./sq. in. at outlet end of pig-tail.

At outlet end of pig-tail, assume K = 46 B.T.U./lb.; then by equation 9

$$y \times 934 = 1089 - 236 - 46;\quad y = 0.864$$

By equation 10:

$$V = 0.864 \times 10.50 + 0.136 \times 0.017 = 9.07 \text{ cu. ft./lb.}$$

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By equation 3:

$K = 7.3\ (0.278 \times 9.07)^2 = 46$ B.T.U./lb., confirming the above assumption.

Now let the pressure at a given point in the pig-tail be equal to, say, 73 lbs./sq. in. and determine the fraction (n) of the total wall traction which is encountered downstream from this point. By the above methods, it can be developed that at this pressure, $K = 16$, $y = 0.880$, and $V = 5.25$.

Hence:

$\Delta V = 2.82$ and $V_{av} = 7.66$. By equations 4 and 4b:

$73 - 40 = 163\ (n)\ (0.278)^2\ (7.66) + 78.8\ (0.278)^2\ (2.82)$;
$n = 0.164$

For rest of pig-tail, pressure drop may be approximated by using outlet value of V and disregarding kinetic energy term in equation 4.

$\Delta P \doteq (163)\ (1-0.164)\ (0.278)^2\ (5.25) \doteq 56$; $P \doteq 129$ lbs./sq. in.
At 129 lbs./sq. in., $K = 5$, $y = 0.878$, $V = 3.06$. Hence, $\Delta V = 2.19$ and $V_{av} = 4.16$.

$\Delta P = (163)\ (1-0.164)\ (0.278)^2\ (4.16) + 78.8\ (0.278)^2\ (2.19) =$ 57 lbs./sq. in. and the pressure at the inlet to the pig-tail is computed to be 130 lbs./sq. in. abs.

Note that it was inadvisable in this case to compute the pressure drop across the pig-tail in one step because of uncertainty in knowing how to average terminal values of 9.07 and 3.06 for specific volumes.

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## APPENDIX III

### TABULAR RESULTS

Tabulated below are header pressures (as measured in the control room) for the characteristic curves used to determine the points shown in Figure 2. A heat load of 100% is defined as the heat required to raise 20 g.p.m. of water through 60°C. A low resistance tube is defined as having 32 film-free active slugs, 6 jacketed lead slugs, and 21 perforated aluminum slugs; a high resistance tube is defined as having 32 active slugs with film sufficient to cause a 40-lb./sq. in. increase in pressure drop when the water rate is 20.4 g.p.m. and the water viscosity is 1 centipoise, and containing 33 jacketed lead slugs.

High Resistance Tubes - Inlet Temperature = 5°C - Heat Load = 100%

| Flow Rate (g.p.m.) \ Orifice Diameter | 0.240 | 0.200 | 0.175 | 0.140 |
|---|---|---|---|---|
| 20 | 440 | 594 | 806 | 1557 |
| 12 | 175 | 230 | 306 | 577 |
| 9 | 143 | 174 | 217 | 369 |
| 6 | 212 | 225 | 245 | 312 |
| 3 | 271 | 275 | 279 | 296 |
| 2 | 248 | 249 | 251 | 259 |

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High Resistance Tubes - Inlet Temperature = 5°C - Heat Load = 50%

| Flow Rate (g.p.m.) \ Orifice Diameter | 0.240 | 0.200 | 0.175 | 0.140 |
|---|---|---|---|---|
| 10 | 132 | 170 | 225 | 411 |
| 6 | 62 | 75 | 95 | 162 |
| 3 | 126 | 130 | 134 | 151 |
| 2 | 146 | 147 | 149 | 157 |
| 1 | 129 | 129 | 129 | 131 |

High Resistance Tubes - Inlet Temperature = 5°C - Heat Load = 25%

| Flow Rate (g.p.m.) \ Orifice Diameter | 0.240 | 0.200 | 0.175 | 0.140 |
|---|---|---|---|---|
| 5 | 49 | 59 | 72 | 119 |
| 3 | 30 | 34 | 38 | 55 |
| 2 | 59 | 60 | 62 | 70 |
| 1 | 80 | 80 | 80 | 82 |
| 1/2 | 65 | 65 | 65 | 66 |

Low Resistance Tubes - Inlet Temperature = 5°C - Heat Load = 100%

| Flow Rate (g.p.m.) \ Orifice Diameter | 0.240 | 0.200 | 0.175 | 0.140 |
|---|---|---|---|---|
| 20 | 357 | 511 | 723 | 1474 |
| 12 | 144 | 199 | 275 | 546 |
| 9 | 123 | 154 | 197 | 349 |
| 6 | 191 | 204 | 224 | 291 |
| 3 | 234 | 238 | 242 | 259 |
| 2 | 211 | 212 | 214 | 222 |

Low Resistance Tubes - Inlet Temperature = 5°C - Heat Load = 25%

| Flow Rate (g.p.m.) \ Orifice Diameter | 0.240 | 0.200 | 0.175 | 0.140 |
|---|---|---|---|---|
| 5 | 43 | 53 | 66 | 113 |
| 3 | 27 | 31 | 35 | 52 |
| 2 | 52 | 53 | 55 | 63 |
| 1 | 68 | 68 | 68 | 70 |
| 1/2 | <65 | <65 | <65 | <66 |

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High Resistance Tubes - Inlet Temperature = 20°C - Heat Load = 100%

| Flow Rate (g.p.m.) \ Orifice Diameter | 0.240 | 0.200 | 0.175 | 0.140 |
|---|---|---|---|---|
| 20 | 426 | 580 | 792 | 1543 |
| 14.2 | 228 | 305 | 412 | 792 |
| 9 | 151 | 182 | 225 | 377 |
| 6 | 236 | 249 | 269 | 336 |
| 3 | 279 | 283 | 287 | 304 |

High Resistance Tubes - Inlet Temperature = 20°C - Heat Load = 25%

| Flow Rate (g.p.m.) \ Orifice Diameter | 0.240 | 0.200 | 0.175 | 0.140 |
|---|---|---|---|---|
| 5 | 48 | 58 | 71 | 118 |
| 3.55 | 33 | 38 | 45 | 68 |
| 2 | 69 | 70 | 72 | 80 |
| 1 | 83 | 83 | 83 | 85 |

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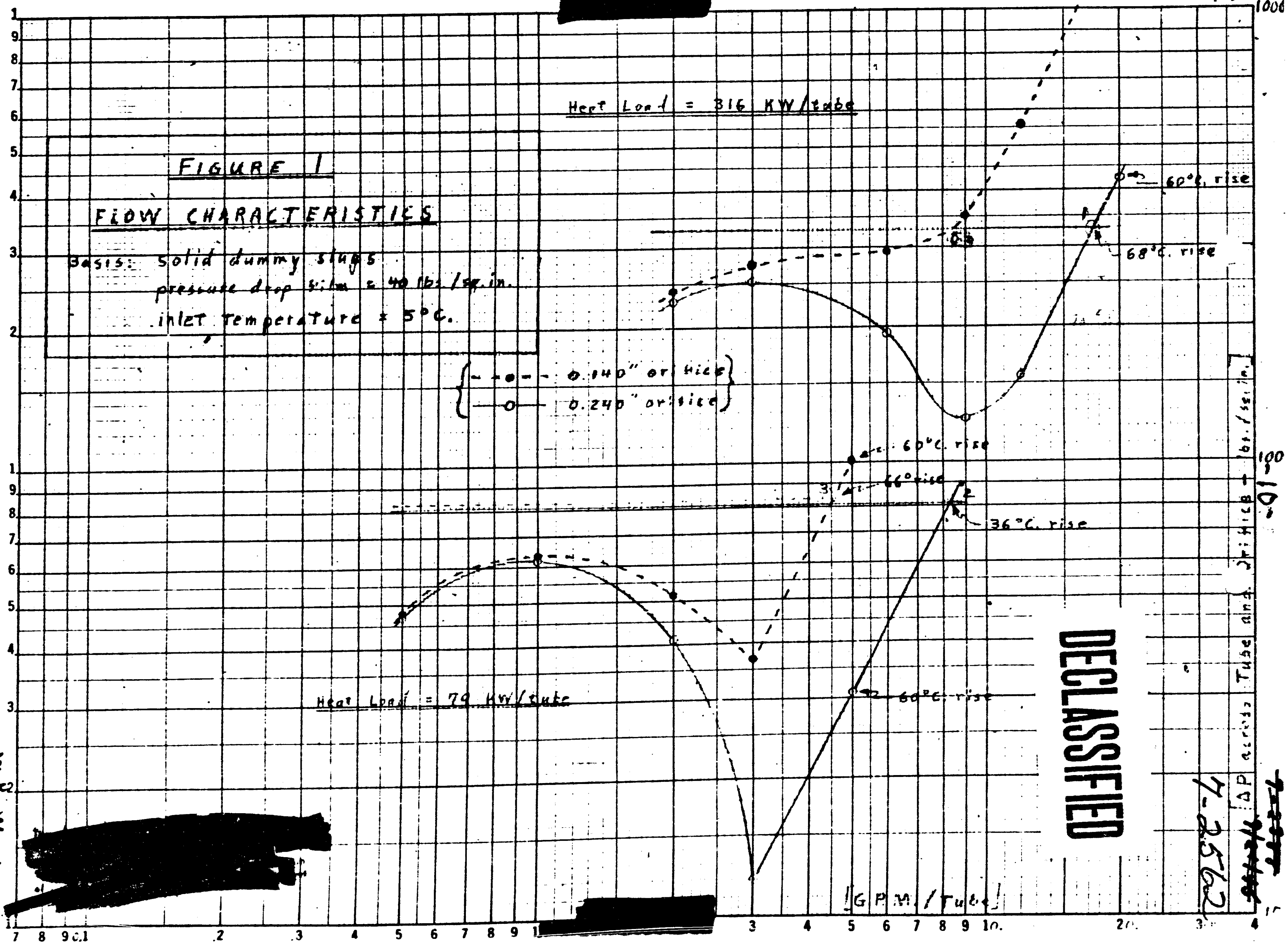


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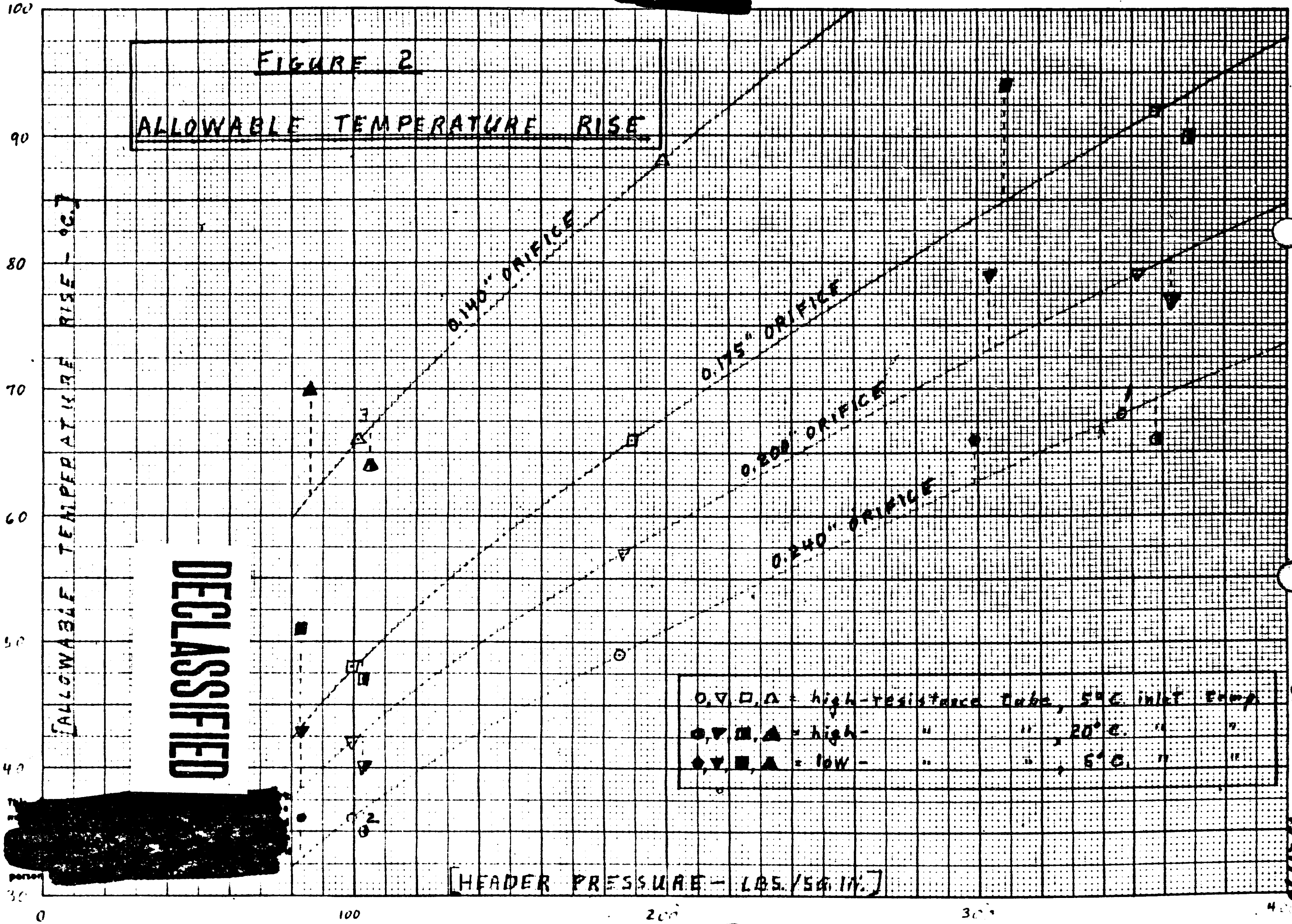


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