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MASTER

HIGH-TEMPERATURE FURNACE FOR HIGH-PRESSURE
ARGON GAS OPERATION

C. D. St. Onge
H. A. Pohto

UNION CARBIDE CORPORATION
NUCLEAR DIVISION
OAK RIDGE Y-12 PLANT

operated for the **ATOMIC ENERGY COMMISSION** *under* **U. S. GOVERNMENT Contract W-7405 eng 26**

**UNION
CARBIDE**

OAK RIDGE Y-12 PLANT
P. O. Box Y
OAK RIDGE, TENNESSEE 37830

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UNION CARBIDE CORPORATION
Nuclear Division

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Contract W-7405-eng-26
With the US Atomic Energy Commission

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ABSTRACT

A furnace has been designed that would operate in an existing Y-12 Plant pressure vessel at 1,700° C and 20,000 psi in an argon atmosphere. The furnace could be constructed at a relatively low cost, and would have a reasonable service life.

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SUMMARY

For the past three years, the Y-12 Plant has been operating high-temperature furnaces in a pressurized argon atmosphere. During this time, considerable progress has been made in the design of furnaces.

The Plant is presently operating a furnace with a hot zone that is 20 inches in diameter and 60 inches long at a temperature of 1,000° C and a pressure of 15,000 psi. The steady-state heat loss for the furnace under these conditions is 65 kw.

This report covers the proposed design of a furnace that would operate in an existing pressure vessel at 1,700° C and 20,000 psi in an argon atmosphere. The proposed furnace could be constructed at a relatively low cost and would have a reasonable service life.

INTRODUCTION

The furnace, shown in Figures 1 and 2, would have a hot zone 14 inches in diameter and 36 inches long. The steady-state heat loss through the furnace under the operating conditions of 1,700° C and 20,000 psi in an argon atmosphere has been calculated to be approximately 50 kw.

The total estimated cost to build and install the furnace is \$12,500 and would require approximately five months for detailed design, fabrication, and installation into the existing pressure vessel.

Figure 1. HIGH-TEMPERATURE FURNACE DESIGN (1,700°C).

Figure 2: HEATER ASSEMBLY.

HIGH-TEMPERATURE FURNACE FOR HIGH-PRESSURE GAS OPERATION

FURNACE DESIGN

The furnace assembly, consisting of the base and bell subassemblies, will be operated in an existing pressure vessel that is 27 1/2 inches in inside diameter and 88 inches deep.

The base, containing the molybdenum resistance strip heating elements, is installed in the vessel from the top and is not removed from the vessel except for inspection or element replacement. The base is insulated with 95 percent alumina bricks.

The bell, containing the part, hearth, and thermocouples, is removed by means of an overhead hoist after each run. Thermocouple wires are taken from the hot zone, down between the hearth support and inside shell, out under the gas seal, and up the outside of the bell to the wire connectors through the top closure of the vessel. Once the bell is removed from the vessel, the workpiece and wires can readily be removed from the assembly by disconnecting the hearth and support from the bell.

Past experience has indicated that the inner shell should be impervious to the gas flow.

Due to the effectiveness of the gas seal, the temperature in the area of the seal is much lower than the temperature of the hot zone.

Heating of the furnace work zone is accomplished primarily by gas convection from the heating elements. A thin-wall ceramic baffle, sitting on the hearth, provides an annular gap that directs the gas flow down the inner wall and back to the heating elements. Insulation in the bell assembly consists of three concentric 95 percent alumina shells, each separated by an annular gas gap.

The temperature distributions through the wall, top, and base were calculated and are given in the section, "Heat-Flow Calculations". Heat is removed from the furnace by forced water cooling of the inside cooling surface of the existing pressure vessel.⁽¹⁾

HEAT-LOSS EVALUATION

Data on the properties of argon at 20,000 psi were taken from work previously done on furnace design⁽²⁾ and are given in Figure 3.

The physical properties of the other materials were taken from published sources^(3 - 6) and are shown plotted as a function of temperature in Figure 4.

Figure 3. ESTIMATED PROPERTIES OF PURE ARGON AT 20,000 PSIA.

Figure 4. THERMAL PROPERTIES OF MATERIALS.

The general method of heat-loss evaluation used was similar to that reported previously.(2)

HEAT-FLOW CALCULATIONS

The mode of heat flow through the furnace from the hot zone to the heat sink can best be described by the following diagram:

The heat loss was independently evaluated for heat flow in three directions: (1) radially outward, (2) vertically upward, (3) vertically downward. The heat transfer was broken into three separate regions: side, top, and bottom. Heat flow between these regions (ie, side to top and side to bottom) was not considered in this analysis. Some heat flow will exist between regions but would require a considerable amount of work to evaluate. For purposes of this study, this flow was not considered significant.

The free convection heat-transfer rate across the film of argon was calculated using Equations 7-1 and 8-11 of Brown and Marco.(4)

$$q = hA\Delta t, \text{ and} \quad (1)$$

$$h = C_1 K \left(\frac{g\beta\rho^2 C_p \Delta t}{\mu K} \right)^{1/3} \quad (2)$$

The conduction heat-transfer rate for the solid materials was calculated using Equation 3-6 of Brown and Marco(4) for the plane surfaces and Equation 3-30 for the cylindrical surfaces:

$$q = \frac{AK_m \Delta t}{X} \text{ (plane), and} \quad (3)$$

$$q = \frac{2\pi l K_m \Delta t}{\ln(D_2/D_1)} \text{ (cylindrical).} \quad (4)$$

The radiant heat-transfer rate across the gas gaps was calculated using Equations 8 and 9 of Gambill:(2)

$$q = \frac{\sigma A (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1} \text{ (plane gap), and} \quad (5)$$

$$q = \frac{\sigma A_1 (T_1^4 - T_2^4)}{\frac{1}{\epsilon_1} + \left(\frac{D_1}{D_2}\right)\left(\frac{1}{\epsilon_2} - 1\right)} \text{ (radial gap).} \quad (6)$$

The free convection heat-transfer rate across the gas gaps was calculated using Equations 4 and 5 of Gambill:(2)

$$q = \frac{K_e A \Delta t}{\delta} \text{ (plane gap), and} \quad (7)$$

$$q = \frac{2 \pi K_e l \Delta t}{\ln \left(\frac{D_2}{D_1} \right)} \text{ (radial gap).} \quad (8)$$

The value of K_e was calculated by Equations 14.49a and 14.49b of Grober, Erk, and Grigull:(7)

$$K_e = 0.11 K (N_{Ra})^{0.29}; \text{ for } 6,000 < N_{Ra} < 10^6, \text{ and} \quad (9)$$

$$K_e = 0.40 K (N_{Ra})^{0.20}; \text{ for } 10^6 < N_{Ra} < 10^8. \quad (10)$$

The forced convection heat-transfer rate across the water film was calculated using Equations 7-1 and 7-14 of Brown and Marco:(4)

$$q = h A \Delta t, \text{ and} \quad (1)$$

$$h = 0.00134 (t + 100) \frac{V^{0.8}}{D_e^{0.2}}. \quad (11)$$

The values for V and D_e were taken from previous work on the flows in the cooling plates and liner.⁽¹⁾

The argon velocity in the annular gaps, listed in Table 1, was calculated from Equation 13 of Gambill:⁽²⁾

$$V = (\delta g \beta \Delta t)^{1/2}. \quad (12)$$

Table 1
HEAT-FLOW FACTORS - RADIAL HEAT FLOW OUTWARD

Gap	Percent Radiation	N_{Ra}	K_e/K	Δt (°F)	Velocity (ft/sec)
3 - 4	52.5	3.24×10^6	8.00	267	0.240
5 - 6	24.1	1.39×10^7	10.71	276	0.306
7 - 8	5.8	3.15×10^7	12.58	225	0.339
9 - 10	0.3	4.36×10^7	13.40	189	0.367

Since the physical properties of the gas, as well as those of the solid materials, depended upon the temperature, an iterative solution was necessary. A separate heat flow for each of the three directions was assumed and the calculations were started. The inside furnace temperature of 1,700° C was used as the starting point, and the heat flow changed until the calculated value for the heat-sink temperature fell within the range of 45 - 60° F.

Heat Flow Radially Outward

The temperature distribution calculated for a radial heat flow of 44.7 kw is indicated in Figure 5. Other information pertaining to the radial flow is reported in Table 1.

The calculations show large temperature gradients across the inner two cylinders. The effect of this temperature gradient is discussed in the section, "Thermal-Stress Calculations".

Heat Flow Vertically Upward

The temperature distribution calculated for a heat flow of 2.43 kw is given in Figure 6. Other heat-flow information is listed in Table 2.

The calculations were first made with an argon gap of four inches between the top of the bell assembly, Plane 9, and the surface of the cooling plate, Plane 11.

Figure 5. RADIAL HEAT FLOW OUTWARD - SIDE REGIONS.

Table 2
HEAT-FLOW FACTORS - VERTICAL HEAT FLOW UPWARD

Gap	Percent Radiation	N_{Ra}	K_e/K	Δt (° F)
3 - 4	64.9	1.95×10^7	11.4	213
5 - 6	39.5	8.37×10^7	15.4	294
7 - 8	12.7	2.58×10^8	19.3	320
10 - 11	0.0	4.34×10^8	21.4	269

Some clearance is necessary between these surfaces in order that the thermocouple wires are not damaged when the vessel is closed. With a four-inch argon gap, the heat flow was only slightly lower than the 2.43 kw, but the surface temperature of the bell lid was much higher. The addition of a conductive plate, such as aluminum, over the center portion of the bell increased the heat flow slightly, but lowered the temperature distribution more in line with that calculated for the radial flow.

Region	Material	Plane	Temperature ($^{\circ}\text{F}$)
1 - 2	Argon	1	3,100
2 - 3	1 1/2" Alumina	2	2,763
3 - 4	1/2" Argon	3	2,168
4 - 5	1 1/2" Alumina	4	1,955
5 - 6	1/2" Argon	5	1,424
6 - 7	1 1/2" Alumina	6	1,130
7 - 8	1/2" Argon	7	705
8 - 9	1/2" Type 304L SS	8	385
9 - 10	3 1/2" Aluminum	9	360
10 - 11	1/2" Argon	10	345
11 - 12	1" HY-80 Steel	11	76
12 - 13	Water	12	56
		13	48

Figure 6. VERTICAL HEAT FLOW UPWARD - TOP REGIONS.

Heat Flow Vertically Downward

The temperature distribution calculated for a downward heat flow of 2.73 kw is given in Figure 7. Additional information on the heat flow is reported in Table 3.

In calculating the downward flow, the same equations were used for convection across the plane layers of gas as used in the upward flow. This analysis should give a higher value for the heat flow than normally would exist. From Table 3, it is clear that the large gap and high Δt causes a very large N_{Ra} between the bottom of the furnace, Plane 4, and the top of the lower cooling plate, Plane 5.

This large gap could be reduced by two inches by lowering the furnace base with the addition of two inches more of alumina brick between Planes 2 and 3. If the furnace is used with the existing five-inch gap, convection from the underside of

Region	Material	Plane	Temperature ($^{\circ}$ F)
1 - 2	Argon	1	3,100
2 - 3	5" Alumina Brick	2	2,898
3 - 4	1/4" Type 304L SS	3	757
4 - 5	Argon	4	745
5 - 6	1" HY-80 Steel	5	90
6 - 7	Water	6	68
		7	59

Figure 7. VERTICAL HEAT FLOW DOWNWARD - BOTTOM REGIONS.

Table 3
HEAT-FLOW FACTORS - VERTICAL HEAT FLOW DOWNWARD

Gap	Percent Radiation	N_{Ra}	K_e/K	Δt ($^{\circ}$ F)
4 - 5	19.2	$6.78 \times 10^{11}(1)$	92.7	655

(1) The large N_{Ra} indicates turbulent rather than laminar natural convection.

the furnace base, due to the high temperature at Plane 4, may increase the temperature in the radial gas gap between Surfaces 9 and 10.

THERMAL-STRESS CALCULATIONS

Since the temperature is higher on the inside surface, the stresses in the cylinders will be compressive on the inner surface and tensile on the outer surface.

The thermal stresses in the two inner cylinders were calculated using the formulas of Roark. (8) Tension at the outer surface:

$$S_t = \frac{\Delta t a E}{2 (1 - \nu) \ln \left(\frac{R_2}{R_1} \right)} \left(1 - \frac{2R_1^2}{R_2^2 - R_1^2} \ln \frac{R_2}{R_1} \right). \quad (13)$$

Compression at the inner surface:

$$S_t = \frac{\Delta t a E}{2 (1 - \nu) \ln \left(\frac{R_2}{R_1} \right)} \left(1 - \frac{2R_2^2}{R_2^2 - R_1^2} \ln \frac{R_2}{R_1} \right). \quad (14)$$

The values for a , E , and ν are not in good agreement for the 95 percent alumina. If typical values listed for the material are used in Equations 13 and 14, values of stress beyond the rupture strength of the material are obtained. This does not mean that the shells will fail during the first cycle of use. Experience has shown that small cracks occur and progressively get larger during repeated cycling. At elevated temperatures, the castable becomes somewhat plastic under the stresses and has a tendency to flow.

The problem was discussed with several manufacturers and consumers of large alumina castable crucibles used in the steel industry for melting metal. In many cases, the material was used under more severe conditions than those in this design, and a reasonable service life was obtained.

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SYMBOLS

A	cross-sectional area normal to heat flow (ft^2).
C_p	specific heat of gas ($0.125 \text{ Btu/lb-}^\circ \text{F}$).
C_1	constant, depending on position of surface; dimensionless (from Brown and Marco).
D	diameter (ft).
E	Young's modulus of elasticity (lbs/in^2).
g	gravitational constant (ft/hr^2).
h	film coefficient ($\text{Btu/hr-ft}^2\text{-}^\circ \text{F}$).
K	thermal conductivity ($\text{Btu/hr-ft-}^\circ \text{F}$).
l	length (ft).
L	characteristic length in Grashof modulus (ft).
N_{Gr}	Grashof modulus, $L^3 \rho^2 g \beta \Delta t / \mu^2$, dimensionless.
N_{Pr}	Prandtl modulus, $C_p \mu / K$, dimensionless.
N_{Ra}	Rayleigh modulus (N_{Gr})(N_{Pr}), dimensionless.
q	heat transfer rate (Btu/hr).
R	radius (inches).
S	thermal stress (lbs/in^2).
t	average water temperature ($^\circ \text{F}$).
Δt	temperature difference ($^\circ \text{F}$).
T	absolute temperature ($^\circ \text{R}$).
V	velocity (ft/hr).
X	thickness (ft).

- α coefficient of thermal expansion (in/in-° F).
 β volumetric coefficient of expansion (1/° F).
 δ gas gap (ft).
 ϵ total normal radiant emissivity, dimensionless.
 μ viscosity of gas (lbs/hr-ft).
 ρ density of gas (lbs/ft³).
 σ Stefan-Boltzmann radiation constant (0.173×10^{-8} Btu/hr-ft²-° R⁴).
 ν Poisson's ratio, dimensionless.

SUBSCRIPTS

- e effective or equivalent value.
1 of inner surface.
2 of outer surface.
m mean value.
t in tangential direction.