Martin Marietta Energy Systems, Inc., Central Engineering Services
Technical Programs and Services

CORRELATION OF HEAT TRANSFER IN A CYLINDER CONTAINING URANIUM HEXAFLUORIDE ENGULFED IN A FIRE

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CONTENTS

LIST OF FIGURES	v
LIST OF TABLES	vii
ACKNOWLEDGEMENTS	ix
ABSTRACT	xi
1. INTRODUCTION AND SUMMARY	1
2. HEAT TRANSFER OVER THE CYLINDER EXTERIOR	5
2.1 RADIATIVE HEAT TRANSFER TO THE CYLINDER EXTERIOR	5
2.2 NATURAL CONVECTION OVER THE CYLINDER EXTERIOR	6
3. HEAT TRANSFER FROM THE CYLINDER TO CONTENTS	9
3.1 HEAT TRANSFER TO THE SOLID UF_6 BULK	9
3.2 HEAT TRANSFER TO THE OVERFLOWING UF_6 LIQUID	13
3.3 HEAT TRANSFER TO THE UF_6 VAPOR	14
4. HEAT TRANSFER BETWEEN URANIUM HEXAFLUORIDE PHASES	17
4.1 HEAT TRANSFER BETWEEN LIQUID AND SOLID UF_6 IN GAP REGION	17
4.2 HEAT TRANSFER BETWEEN LIQUID UF_6 AND SUBMERGED UF_6 SOLID	18
4.3 HEAT TRANSFER BETWEEN VAPOR UF_6 AND THE SOLID BULK	19
4.4 HEAT TRANSFER BETWEEN UF_6 VAPOR AND LIQUID UF_6	20
5. URANIUM HEXAFLUORIDE POOL BOILING	23
5.1 FREE CONVECTION BOILING	23
5.2 NUCLEATE BOILING	25
5.3 TRANSITION BOILING	27
5.4 FILM BOILING	27
5.5 EFFECT OF THE ROHSENOW CONSTANT ON THE UF_6 BOILING CURVE	28

5.6 EFFECT OF PRESSURE ON THE UF_6 BOILING CURVE	28
5.7 EFFECT OF SUBCOOLING ON THE UF_6 BOILING CURVE	29
5.8 SUMMARY OF BOILING HEAT TRANSFER COEFFICIENT CORRELATIONS .	30
REFERENCES	33

LIST OF FIGURES

1.1	Schematic of heat transfer mechanisms in UF_6 cylinders	2
2.1	Heat transfer coefficients (h_1 and h_2) over the exterior of a horizontal cylinder exposed to air	6
3.1	Heat transfer coefficients (h_3 and h_4) across UF_6 vapor gap due to convection and conduction	12
3.2	Heat transfer coefficients ($h_3 + h_4$ and h_5) in an annular gap containing UF_6 vapor	12
3.3	Radiative heat transfer coefficients (h_6 and h_7) between the cylinder and UF_6	14
3.4	Natural convective heat transfer coefficient (h_8) between the cylinder and UF_6 vapor	15
4.1	Convective heat transfer coefficient (h_9) between UF_6 liquid and solid bulk in gap region	18
4.2	Convective heat transfer coefficient (h_{10}) between UF_6 liquid and submerged UF_6 solid	19
4.3	Convective heat transfer coefficient (h_{11}) between UF_6 vapor and solid bulk	20
4.4	Convective heat transfer coefficient (h_{12}) between UF_6 vapor and liquid UF_6	21
5.1	Typical boiling curve: surface heat flux as a function of excess temperature	23
5.2	Pool boiling curves for UF_6 as a function of the Rohsenow constant	28
5.3	Pool boiling curves for UF_6 as a function of pressure	29
5.4	Critical and Leidenfrost heat fluxes as a function of pressure	30
5.5	Critical heat flux as a function of subcooling	31

LIST OF TABLES

1.1	Modes of heat transfer in UF_6 cylinders	3
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ABSTRACT

Transient heat transfer/stress analysis models are currently being developed to evaluate the response of cylinders containing uranium hexafluoride (UF_6) to fire accident scenarios. In order to accurately predict temperatures within the cylinder, and ultimately elapsed time to failure, the heat transfer to and within the cylinder must be well characterized. This report contains a complete set of heat transfer correlations required for such a model. Correlations are presented for predicting heat transfer rates over the cylinder exterior (radiative exchange and natural convection), from the cylinder interior to the various phases of UF_6 (solid, liquid, and vapor) in the cylinder, between UF_6 phases in the cylinder, and during UF_6 liquid boiling. The heat transfer coefficients predicted by these correlations were chosen based on best engineering judgement and have not yet been compared to data from actual cylinder fire tests.

1. INTRODUCTION AND SUMMARY

Models are currently being developed to evaluate the response of UF_6 cylinders to fire accident scenarios. In order to accurately predict temperatures and pressures within the cylinder, and ultimately elapsed time to failure, the heat transfer to and within the cylinder must be well characterized. The purpose of this report is to develop and present correlations for determining the heat transfer rates associated with the UF_6 cylinders. The requisite thermophysical properties of UF_6 are discussed in detail in a report entitled *Correlation of the Thermophysical Properties of Uranium Hexafluoride Over a Wide Range of Temperature and Pressure*.¹

Chapter 2 contains correlations for predicting heat transfer rates over the exterior of the UF_6 cylinder, including radiative exchange and natural convection. Chapter 3 contains correlations for estimating the heat transfer rates from the cylinder interior to the various phases of UF_6 (solid, liquid, and vapor) contained in the cylinder. Chapter 4 contains correlations required for predicting the natural convective heat transfer between the various phases of UF_6 in the cylinder. Chapter 5 presents correlations for predicting boiling heat transfer coefficients for the various boiling regimes (free convection, nucleate, transition, and film boiling) of UF_6 liquid. The effect of several variables including the Rohsenow constant, pressure, and liquid subcooling on the UF_6 boiling curve are also investigated in Chapter 5.

These various modes of heat transfer associated with the UF_6 cylinders are illustrated in Fig. 1.1. In order to depict all the possible heat transfer modes, two configurations are shown. The first (top) configuration represents the cylinder early in the heating cycle prior to liquid overflow (i. e., the liquid level is below the top of the solid bulk). The second (lower) configuration represents the cylinder later in the heating cycle following liquid overflow of the solid bulk. Table 1.1 describes the various modes of heat transfer and refers the reader to the section of the report which describes it.

It should be noted that the correlations presented in this report were chosen based on best engineering judgement. The heat transfer coefficients predicted by these correlations have not yet been compared to data from actual fire tests, although this is anticipated in the future. This report is basically intended to be a catalog of the heat transfer correlations needed for the construction of a transient heat transfer/stress analysis model of a UF_6 cylinder engulfed in a fire or exposed to other external heat input.

Fig. 1.1. Schematic of heat transfer mechanisms in UF_6 cylinders.

Table 1.1. Modes of Heat Transfer in UF₆ Cylinders

Symbol	Description of Heat Transfer Mechanism	Section
q ₁	Radiant heat transfer from surroundings to cylinder exterior	2.1
q ₂	Natural convection from surroundings to cylinder exterior	2.2
q ₃	Natural convection across annulus containing UF ₆ vapor	3.1
q ₄	Conduction across annulus containing UF ₆ vapor	3.1
q ₅	Radiation between cylinder interior and solid UF ₆	3.1
q ₆	Radiation between upper cylinder interior and solid UF ₆ bulk prior to liquid overflow	3.1
q ₇	Radiation between upper cylinder interior and overflowing UF ₆ liquid	3.2
q ₈	Natural convection from cylinder interior to UF ₆ vapor	3.3
q ₉	Natural convection between UF ₆ liquid in annulus and UF ₆ solid bulk	4.1
q ₁₀	Natural convection between UF ₆ liquid and top surface of UF ₆ solid bulk	4.2
q ₁₁	Natural convection between UF ₆ vapor and top surface of UF ₆ solid bulk	4.3
q ₁₂	Natural convection between UF ₆ vapor and top surface of overflowing UF ₆ liquid	4.4
q ₁₃	Liquid UF ₆ boiling	5.1-5.4

2. HEAT TRANSFER OVER THE CYLINDER EXTERIOR

Heat transfer from the surroundings to the cylinder exterior during a fire accident scenario occurs by radiative exchange as well as natural convection. Correlations for predicting these heat transfer rates are presented in the following sections.

2.1 RADIATIVE HEAT TRANSFER TO THE CYLINDER EXTERIOR

The radiative heat transfer from the surroundings to the cylinder, q_1 , is determined by

$$q_1 = AF_e\sigma(T_{fire}^4 - T_{cyl}^4), \quad (2.1)$$

where A = cylinder surface area,
 F_e = emissivity factor,
 σ = Stefan-Boltzmann constant,
 T_{fire} = fire temperature (absolute),
 T_{cyl} = cylinder surface temperature (absolute).

The emissivity factor, F_e , is approximated by³

$$F_e = \frac{1}{\frac{1}{\epsilon_{fire}} + \frac{1}{\epsilon_{cyl}} - 1}, \quad (2.2)$$

where ϵ_{fire} = emissivity of the fire,
 ϵ_{cyl} = emissivity of cylinder.

For comparison to other heat transfer coefficient values, a radiative heat transfer coefficient, h_1 , is calculated as

$$h_1 = \frac{q_1}{A(T_{fire} - T_{cyl})}. \quad (2.3)$$

The radiative heat transfer coefficient as a function of cylinder surface temperature is shown in Fig. 2.1 assuming a fire temperature of 1475°F and fire and cylinder emissivity values of 1.0 and 0.9, respectively, as prescribed in 10 CFR 71.²

Fig. 2.1. Heat transfer coefficients (h_1 and h_2) over the exterior of a horizontal cylinder exposed to air.

2.2 NATURAL CONVECTION OVER THE CYLINDER EXTERIOR

The natural convective heat transfer to the cylinder exterior, h_2 , may be approximated by the following correlation recommended by Churchill and Chu:⁴

$$h_2 = \frac{k}{D} \left[0.6 + \frac{0.387 Ra_D^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}} \right]^2, \quad (2.4)$$

where k = thermal conductivity of air,
 D = diameter of cylinder,
 Ra_D = Rayleigh number,
 Pr = Prandtl number of air.

The Rayleigh number is defined as

$$Ra_D = \frac{g\beta\Delta TD^3}{\nu\alpha}, \quad (2.5)$$

where g = acceleration due to gravity,
 β = coefficient of expansion of air,
 $\Delta T = T_{\text{fire}} - T_{\text{cyl}}$,
 ν = kinematic viscosity of air,
 α = thermal diffusivity of air.

Equation 2.4 is valid for $10^5 < Ra_D < 10^{12}$.

The heat transfer coefficient due to natural convection to air is shown as a function of cylinder surface temperature in Fig. 2.1. A cylinder diameter of 4.0 ft and a fire temperature of 1475°F were assumed. Properties of air were taken from Incropera and DeWitt.⁵ As evident from Fig. 2.1, radiation is the dominant mode of heat transfer to the cylinder exterior during a fire. Radiation heat transfer coefficients are at least an order of magnitude higher than corresponding convection heat transfer coefficients. These results are consistent with observations made by Mansfield.¹⁰ Mansfield observed that the ratio of radiative to convective heat flux is approximately 9:1 for large pool fires.

The effect of assuming forced convection as opposed to natural convection over the cylinder exterior has been investigated. Based on hydrocarbon pool fire data¹¹, initial vertical velocities of approximately 13 ft/s (4 m/s) are achieved in a pool fire. Using this velocity and an appropriate correlation for forced convection over a cylinder⁵, forced convection heat transfer coefficients were calculated. The forced convection values were found to be of the same order of magnitude as those predicted by natural convection correlations. Therefore, it can be concluded that the convective heat transfer over the cylinder exterior may be accurately predicted by the natural convection correlation presented earlier in this section.

3. HEAT TRANSFER FROM THE CYLINDER TO CONTENTS

During heating of the cylinder, heat transfer occurs between the cylinder and all three phases of UF_6 (solid, liquid, and vapor). Prior to liquid overflow, heat transfer to the solid UF_6 bulk occurs across a vapor gap on the bottom portion of the cylinder. Radiative heat transfer also takes place between the upper cylinder interior and the UF_6 solid top surface. Heat transfer from the cylinder to liquid UF_6 in contact with the cylinder occurs by natural convection and/or boiling which will be discussed in Chapter 5. Following liquid overflow, radiative heat exchange occurs between the upper portion of the cylinder and the overflowing liquid surface. Heat transfer to the UF_6 vapor occurs by natural convection along the upper interior of the cylinder.

3.1 HEAT TRANSFER TO THE SOLID UF_6 BULK

Heat transfer to the solid UF_6 bulk occurs by convection and conduction as well as radiant exchange across an annular gap containing UF_6 vapor. Kuehn and Goldstein have correlated equations for convection and conduction between horizontal circular cylinders.⁶ The following relation can be used to approximate the Nusselt number for heat transfer by natural convection between an inner and outer horizontal cylinder:

$$Nu_{Di,conv} = \frac{2}{\ln \left[\frac{1 + 2/X}{1 - 2/Y} \right]}, \quad (3.1)$$

where

$$X = [(0.518 Ra_{Di}^{1/4} [1 + (0.559/Pr)^{3/5}]^{-5/12})^{15} + (0.1 Ra_{Di}^{1/3})^{15}]^{1/15}, \quad (3.2)$$

$$Y = \left[\left(\left[\left(\frac{2}{1 - e^{-0.25}} \right)^{5/3} + (0.587 Gr_{Do}^{1/4})^{5/3} \right]^{3/5} \right)^{15} + (0.1 Ra_{Do}^{1/3})^{15} \right]^{1/15}, \quad (3.3)$$

and

$$G = \left[\left(1 + \frac{0.6}{Pr^{0.7}} \right)^{-5} + (0.4 + 2.6 Pr^{0.7})^{-5} \right]^{-1/5}. \quad (3.4)$$

Ra_{Di} is the Rayleigh number based on the inner diameter (D_i), Ra_{Do} is the Rayleigh number based on the outer diameter (D_o), and Pr represents the Prandtl number. Using this nomenclature, the gap thickness is defined as

$$t_{gap} = \frac{D_o - D_i}{2} \quad (3.5)$$

All properties in Eqs. 3.1 - 3.4 are evaluated at the film temperature (T_{film}) calculated as

$$T_{film} = \frac{T_{cyl} + T_{solid}}{2} \quad (3.6)$$

where T_{cyl} = inner cylinder surface temperature,
 T_{solid} = UF_6 solid surface temperature.

For heat transfer across the annulus due to conduction, Kuehn and Goldstein recommend the following correlation for Nusselt number:⁶

$$Nu_{Di,cond} = \frac{2}{\cosh^{-1} \left[\frac{(D_i^2 + D_o^2)}{2D_i D_o} \right]} \quad (3.7)$$

The value of the overall Nusselt number for heat transfer across an annular gap (convection + conduction) valid at any Rayleigh number is estimated by

$$Nu_{Di} = [Nu_{Di,cond}^{15} + Nu_{Di,conv}^{15}]^{1/15} \quad (3.8)$$

The corresponding heat transfer coefficient is calculated as

$$h_{3+4} = \frac{Nu_{Di} k}{D_i} \quad (3.9)$$

Heat is also transferred across the vapor gap by radiative exchange. This radiative heat transfer is calculated as

$$q_5 = A_{sol} F_e \sigma (T_{cyl}^4 - T_{sol}^4) , \quad (3.10)$$

where A_{sol} = surface area of solid UF_6 exposed to UF_6 vapor,
 F_e = emissivity factor,
 σ = Stefan-Boltzmann constant,
 T_{cyl} = inner cylinder surface temperature (absolute),
 T_{sol} = solid UF_6 temperature (absolute).

The emissivity factor, F_e , can be estimated as³

$$F_e = \frac{1}{\frac{1}{\epsilon_{cyl}} + \frac{1}{\epsilon_{sol}} - 1} , \quad (3.11)$$

where ϵ_{cyl} = emissivity of cylinder,
 ϵ_{sol} = emissivity of solid UF_6 .

Equation 3.11 assumes that $A_{sol} \approx A_{cyl}$. The corresponding heat transfer coefficient for this radiative exchange is

$$h_5 = \frac{q_5}{A_{sol} (T_{cyl} - T_{sol})} . \quad (3.12)$$

Figure 3.1 illustrates the relative magnitudes of the heat transfer coefficients due to convection and conduction across a gap containing UF_6 vapor as a function of gap width. A film temperature of 400°F, the corresponding vapor pressure at 400°F, and an outer diameter of 48 in (i. e., 48X cylinder) were assumed in order to generate the data appearing in Fig. 3.1. As seen from the data, convection heat transfer is the dominant mode of heat transfer across the vapor gap. Even at very small gap widths (1/8 in), the convective heat transfer is an order of magnitude greater than the heat transfer due to conduction across the gap. Figure 3.2 presents the combined heat transfer coefficient (convection and conduction) in an annular region containing UF_6 vapor as a function of vapor film temperature and pressure. An outer diameter of 48 in, a gap width of 1/4 in, and a solid UF_6 temperature of 100°F were used to generate this data. As seen from Fig. 3.2, the heat transfer across the gap due to convection and conduction is a strong function of pressure. Values of the combined heat transfer coefficient range from approximately 1 to 20 Btu/hr-ft²-°F. The radiative heat transfer coefficient across the vapor gap (h_5) is also shown in Fig. 3.2 as a function of film temperature. Values of this heat transfer coefficient range from approximately 1 to 3 Btu/hr-ft²-°F.

Prior to liquid overflow, radiative heat exchange also occurs between the top cylinder interior and the solid UF_6 surface and may be calculated as

Fig. 3.1. Heat transfer coefficients (h_3 and h_4) across UF_6 vapor gap due to convection and conduction.

Fig. 3.2. Heat transfer coefficients ($h_3 + h_4$ and h_3) in an annular gap containing UF_6 vapor.

$$q_6 = \frac{A_{sol} \sigma (T_{cyl}^4 - T_{sol}^4)}{\frac{1}{\epsilon_{sol}} + \frac{A_{sol}}{A_{cyl}} \left(\frac{1}{\epsilon_{cyl}} - 1 \right)}, \quad (3.13)$$

where A_{cyl} = surface area of upper cylinder interior above solid surface exposed to UF_6 vapor.

The corresponding heat transfer coefficient is calculated as

$$h_6 = \frac{q_6}{A_{sol} (T_{cyl} - T_{sol})}. \quad (3.14)$$

This heat transfer coefficient is shown as a function of cylinder temperature in Fig 3.3. This radiative heat transfer coefficients shown as a function of cylinder temperature in Fig. 3.3. Plots are presented for UF_6 solid temperatures of 70 and 147.3°F. Typical values of steel emissivity range from approximately 0.07 to 0.81.³ For illustrative purposes, a value of 0.6 was arbitrarily chosen to construct the plots. The solid UF_6 was assumed to have an emissivity of 0.9.⁹

3.2 HEAT TRANSFER TO THE OVERFLOWING UF_6 LIQUID

Heat transfer from the cylinder to the UF_6 liquid in contact with the cylinder wall occurs by boiling which is discussed in Chapter 5. Once the liquid has overflown the solid UF_6 bulk, heat is transferred to the liquid UF_6 surface by radiation. This radiative heat transfer is calculated as

$$q_7 = \frac{A_{liq} \sigma (T_{cyl}^4 - T_{liq}^4)}{\frac{1}{\epsilon_{liq}} + \frac{A_{liq}}{A_{cyl}} \left(\frac{1}{\epsilon_{cyl}} - 1 \right)}, \quad (3.15)$$

where T_{liq} = liquid UF_6 surface temperature,
 ϵ_{liq} = emissivity of liquid UF_6 ,
 A_{liq} = surface area of liquid UF_6 exposed to UF_6 vapor,
 A_{cyl} = surface area of upper cylinder interior above liquid surface exposed to UF_6 vapor.

The corresponding heat transfer coefficient is calculated as

$$h_7 = \frac{q_7}{A_{liq} (T_{cyl} - T_{liq})}. \quad (3.16)$$

Fig. 3.3. Radiative heat transfer coefficients (h_r and h_{rj}) between the cylinder and UF_6 .

This radiative heat transfer coefficient is shown as a function of cylinder temperature in Fig. 3.3. Plots are presented for liquid UF_6 temperatures of 147.3 and 446.3°F (triple point and critical temperatures). Cylinder and liquid emissivities of 0.6 and 0.9, respectively, were used to generate these plots. Heat transfer coefficients for this mode of heat transfer range from approximately 0.8 to 11.4 Btu/hr-ft²-°F.

3.3 HEAT TRANSFER TO THE UF_6 VAPOR

Heat transfer from the cylinder to UF_6 vapor will occur along the upper cylinder interior. Kuehn and Goldstein recommend the following correlation for natural convection heat transfer to a fluid inside a horizontal cylinder at any Rayleigh number:⁶

$$Nu_{Do,conv} = \frac{2}{-\ln \left[1 - \frac{2}{Z} \right]}, \quad (3.17)$$

where

$$Z = \left[\left(\left(\frac{2}{1 - e^{-0.25}} \right)^{5/3} + (0.587 Ra_{Do}^{1/4})^{5/3} \right)^{3/5} + (0.1 Ra_{Do}^{1/3})^{15} \right]^{1/15} \quad (3.18)$$

The corresponding heat transfer coefficient is calculated as

$$h_g = \frac{Nu_{D_o, conv} k}{D_o} \quad (3.19)$$

Properties used to calculate the Rayleigh number should be evaluated at the film temperature $[T_{film} = (T_{cyl} + T_{vap})/2]$. The natural convection heat transfer coefficient over the upper cylinder interior is presented graphically in Fig. 3.4 as a function of the vapor film temperature and pressure. As seen in Fig. 3.4, this natural convective heat transfer coefficient is very sensitive to pressure. Values range from approximately 1 to 24 Btu/hr-ft²-°F.

Fig. 3.4. Natural convective heat transfer coefficient (h_g) between the cylinder and UF₆ vapor.

4. HEAT TRANSFER BETWEEN URANIUM HEXAFLUORIDE PHASES

Heat is transferred by natural convection between the three UF_6 phases inside the cylinder. Each heat transfer mechanism is discussed below.

4.1 HEAT TRANSFER BETWEEN LIQUID AND SOLID UF_6 IN GAP REGION

The heat which is transferred from the UF_6 liquid to the bulk solid in the annular region is effectively natural convection over a cylinder. Kuehn and Goldstein recommend the following correlation for natural convection heat transfer from a horizontal cylinder valid at any Rayleigh and Prandtl number:⁶

$$Nu_D = \frac{2}{\ln \left[1 + \frac{2}{W^{1/15}} \right]}, \quad (4.1)$$

where

$$W = \left(0.518 Ra_D^{1/4} \left[1 + \left(\frac{0.559}{Pr} \right)^{3/5} \right]^{-5/12} \right)^{15} + (0.1 Ra_D^{1/3})^{15}. \quad (4.2)$$

where Ra_D = Rayleigh number based on diameter,
 Pr = Prandtl number.

The corresponding heat transfer coefficient is calculated as

$$h_g = \frac{Nu_D k}{D}, \quad (4.3)$$

where k = thermal conductivity of liquid UF_6 ,
 D = diameter of solid UF_6 lump.

Values of the natural convection heat transfer coefficient calculated from the above equations are presented in Fig. 4.1 as a function of the liquid temperature. A diameter of 4 ft and a solid UF_6 temperature of 147.3°F were used to generate this data. The effect of pressure on this heat transfer coefficient is negligible. The value of this heat transfer coefficient ranges from approximately 25 to 200 $\text{Btu/hr-ft}^2\text{-}^\circ\text{F}$.

Fig. 4.1. Convective heat transfer coefficient (h_g) between UF_6 liquid and solid bulk in gap region.

4.2 HEAT TRANSFER BETWEEN LIQUID UF_6 AND SUBMERGED UF_6 SOLID

If heat is continually added to the cylinder, the liquid UF_6 will eventually overflow the solid. After this occurs, heat is transferred from the UF_6 liquid to the submerged solid. This heat transfer is approximated as natural convection over the upper surface of a horizontal flat plate. The Nusselt number for natural convection over the upper surface of a cool horizontal plate can be approximated by ⁵

$$Nu_L = 0.27Ra_L^{1/4} \quad (4.4)$$

Equation 4.3 is valid for $10^5 < Ra_L < 10^{10}$. The characteristic length, L , used in the Rayleigh number calculation is defined as

$$L = \frac{A_s}{P}, \quad (4.5)$$

where A_s = solid surface area exposed to liquid UF_6 ,
 P = perimeter of solid exposed to liquid UF_6 .

The corresponding heat transfer coefficient is calculated as

$$h_{10} = \frac{Nu_L k}{L} \quad (4.6)$$

In this calculation, the solid surface is approximated as a flat surface. The natural heat transfer coefficient between liquid UF_6 and the submerged solid bulk is presented graphically in Fig. 4.2 as a function of the liquid film temperature. A characteristic length of 1.11 ft and a solid UF_6 temperature of 147.3°F were used to generate this data. The effect of pressure on this liquid convective heat transfer coefficient is negligible. As seen from Fig. 4.2, the values of this heat transfer coefficient range from approximately 7.5 to 27.5 Btu/hr-ft²-°F.

4.3 HEAT TRANSFER BETWEEN VAPOR UF_6 AND THE SOLID BULK

Heat is transferred from the UF_6 vapor to the solid bulk prior to liquid UF_6 overflow. This natural convective heat transfer may be calculated from Eqs. 4.4 to 4.6. The natural heat transfer coefficient at the vapor-solid interface is shown as a function of the vapor film temperature in Fig. 4.3. A characteristic length of 1.11 ft and a solid UF_6 temperature of 100°F were used to generate this data. The convective heat transfer coefficient (h_{11}) between the vapor and solid UF_6 ranges from 0.6 to 8 Btu/hr-ft²-°F over the range of temperature and pressure considered.

Fig. 4.2. Convective heat transfer coefficient (h_{10}) between UF_6 liquid and submerged UF_6 solid.

Fig. 4.3. Convective heat transfer coefficient (h_{11}) between UF_6 vapor and solid bulk.

4.4 HEAT TRANSFER BETWEEN UF_6 VAPOR AND LIQUID UF_6

Once the liquid UF_6 overflows the solid bulk, natural convective heat transfer will occur at the vapor-liquid interface. In this case, the liquid surface is actually flat and therefore the heat transfer may be calculated by Eq. 4.6. This natural convection heat transfer coefficient (h_{12}) is presented graphically as a function of vapor film temperature in Fig. 4.4. A characteristic length of 1.11 ft and a liquid UF_6 temperature of 147.3°F were used to generate the data in Fig. 4.4. The convective heat transfer between the UF_6 vapor and liquid UF_6 ranges from approximately 0.2 to 8.1 $\text{Btu/hr-ft}^2\text{-}^{\circ}\text{F}$ over the range of temperature and pressure considered.

Fig. 4.4. Convective heat transfer coefficient (h_{12}) between UF_6 vapor and liquid UF_6 .

5. URANIUM HEXAFLUORIDE POOL BOILING

Heating of UF_6 storage cylinders at pressures greater than or equal to 22.046 psia will lead to boiling of UF_6 liquid. Therefore, as a part of the thermal analysis of the UF_6 cylinders, correlations for the boiling heat transfer coefficients must be obtained for each possible boiling regime. Although boiling data for common substances such as water and hydrocarbons are abundant in the literature, such data are nonexistent for UF_6 . Therefore, a boiling curve for UF_6 must be constructed from existing boiling correlations found in the literature.

Figure 5.1 represents a typical pool boiling curve. actual boiling data would result in a smooth, more rounded plot than the one shown in Fig. 5.1; however, the segments of the boiling curve between points A, B, C, and D are very nearly linear on a log-log plot. Correlations for the various boiling regimes (free convection, nucleate, transition, and film boiling) are discussed in Sections 5.1 to 5.4. The effects of surface condition, pressure, and liquid subcooling are discussed in Sections 5.5 to 5.7. The complete set of boiling correlations is then summarized in Section 5.8.

5.1 FREE CONVECTION BOILING

The excess temperature, ΔT_e , is defined as

$$\Delta T_e = T_{\text{wall}} - T_{\text{sat}}, \quad (5.1)$$

Fig. 5.1. Typical boiling curve: surface heat flux as a function of excess temperature.

where T_{wall} = cylinder wall temperature,
 T_{sat} = saturation temperature.

For small values of ΔT_e , the heat transfer is controlled by natural convection effects. It is assumed that the majority of liquid UF_6 boiling in the cylinder will occur in an annular region between the cylinder shell and the solid UF_6 bulk. Some liquid UF_6 boiling will occur above the solid bulk. However, the natural convection heat transfer coefficients for this situation are not expected to vary significantly from those predicted for the annulus. Correlations for natural convection heat transfer between horizontal circular cylinders (i.e., annulus) are presented in a report by Kuehn and Goldstein (see Section 3.1).⁶ Data on UF_6 obtained from these equations were correlated to obtain the following expressions for natural convection heat transfer between the inside cylinder wall and liquid UF_6 at various pressures:

$$p = 25 \text{ psia} \quad h_{\text{NC}} = 22.596 \Delta T_e^{0.3161}, \quad (5.2)$$

$$p = 50 \text{ psia} \quad h_{\text{NC}} = 22.299 \Delta T_e^{0.3196}, \quad (5.3)$$

$$p = 100 \text{ psia} \quad h_{\text{NC}} = 21.688 \Delta T_e^{0.3228}, \quad (5.4)$$

$$p = 250 \text{ psia} \quad h_{\text{NC}} = 20.910 \Delta T_e^{0.3282}, \quad (5.5)$$

$$p = 400 \text{ psia} \quad h_{\text{NC}} = 20.796 \Delta T_e^{0.3308}, \quad (5.6)$$

where h_{NC} , the natural convective heat transfer coefficient, is in Btu/hr-ft²-°F. Equations 5.2 - 5.6 were combined to obtain the following expression for natural convection boiling of UF_6 as a function of excess temperature and pressure, p , in psia:

$$h_{\text{NC}} = a \Delta T_e^b, \quad (5.7)$$

where

$$a = 22.878 - 1.296 \times 10^{-2} p + 1.949 \times 10^{-5} p^2, \quad (5.8)$$

and

$$b = 0.3151 + 8.189 \times 10^{-5} p - 1.075 \times 10^{-7} p^2. \quad (5.9)$$

The corresponding heat flux may be determined by

$$q_{NC}'' = a \Delta T_e^{b+1}. \quad (5.10)$$

It should be noted that the natural convective regime is affected by liquid subcooling (i.e., liquid below the saturation temperature). The effect of subcooling on the position of the boiling curve appears to depend on the convection geometry.⁸ This effect was not considered in the present study.

5.2 NUCLEATE BOILING

At a certain value of ΔT_e , termed the onset of nucleate boiling, vapor bubbles will begin to form and the heat transfer coefficient will rise rapidly with increasing ΔT_e . Nucleate boiling continues until a maximum value of heat flux, known as the critical heat flux (Point B in Fig. 5.1), is reached. At this latter point, a considerable amount of vapor is being formed and liquid is no longer continuously wetting the surface. The most widely accepted and used correlation for nucleate boiling was developed by Rohsenow⁷

$$q_{NB}'' = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{g_c \sigma} \right]^{1/2} \left[\frac{c_{p,l} \Delta T_e}{C_{s,f} h_{fg} Pr_l^{1.7}} \right]^3, \quad (5.11)$$

where

- μ_l = liquid dynamic viscosity,
- h_{fg} = heat of vaporization,
- g = acceleration due to gravity,
- g_c = conversion factor,
- ρ_l = liquid density,
- ρ_v = vapor density,
- σ = surface tension,
- $c_{p,l}$ = liquid specific heat,
- $C_{s,f}$ = Rohsenow boiling constant,
- Pr_l = liquid Prandtl number.

Equation 5.11 is of the form

$$q_{NB}'' = \frac{C_b \Delta T_e^3}{C_{s,f}^3}, \quad (5.12)$$

where

$$C_b = \mu_l h_{fg} \left[\frac{g(\rho_l - \rho_v)}{g_c \sigma} \right]^{1/2} \left[\frac{c_{p,l}}{h_{fg} Pr_l^{1.7}} \right]^3 \quad (5.13)$$

All properties in the preceding equation are evaluated at the saturation temperature of the liquid. The value of the constant $C_{s,f}$ depends on the surface-liquid combination. Representative values of the Rohsenow constant range from 0.0025 to 0.1.⁵ High values of $C_{s,f}$ imply smooth, polished surfaces while lower values are characteristic of more rough, pitted surfaces.

The value of excess temperature at boiling incipience, ΔT_{ONB} (Point A in Fig 5.1), can be determined by setting Equations 5.10 and 5.12 equal to one another and solving for ΔT_e . The result is

$$\Delta T_{ONB} = \left[\frac{a C_{s,f}^3}{C_b^3} \right]^{\frac{1}{2-b}} \quad (5.14)$$

Zuber obtained an expression for the critical heat flux of the form⁷

$$q_{\max}'' = \left(\frac{\pi}{24} \right) h_{fg} \rho_v \left[\frac{\sigma g g_c (\rho_l - \rho_v)}{\rho_v^2} \right]^{1/4} \left[\frac{\rho_l + \rho_v}{\rho_l} \right]^{1/2} \quad (5.15)$$

Replacing the Zuber constant ($\pi/24$) by an experimental value of 0.149 and approximating the last term with unity, Equation 5.15 becomes⁵

$$q_{\max}'' = 0.149 h_{fg} \rho_v \left[\frac{\sigma g g_c (\rho_l - \rho_v)}{\rho_v^2} \right]^{1/4} \quad (5.16)$$

All properties in Eq. 5.16 are evaluated at the liquid saturation temperature. The temperature difference at the critical heat flux can be calculated from the following expression:

$$\Delta T_{CHF} = \left[\frac{q_{\max}'' C_{s,f}^3}{C_b^3} \right]^{1/3} \quad (5.17)$$

5.3 TRANSITION BOILING

As the excess temperature increases beyond the value associated with the critical heat flux, the boiling heat transfer coefficient decreases. This decrease in heat transfer rate is due to a vapor blanket of low thermal conductivity which forms on the surface. The heat flux will continue to decrease until a minimum value, called the Leidenfrost point (Point C in Fig. 5.1), is reached. Berenson arrived at the following expression for the excess temperature at the Leidenfrost point ⁸

$$\Delta T_L = \frac{0.127 \rho_{vf} h_{fg}}{k_{vf}} \left[\frac{g(\rho_l - \rho_v)}{\rho_l + \rho_v} \right]^{2/3} \left[\frac{g_c \sigma}{g(\rho_l - \rho_v)} \right]^{1/2} \left[\frac{\mu_{vf}}{g(\rho_l - \rho_v)} \right]^{1/3} \quad (5.18)$$

All properties are evaluated at the saturation temperature with the exception of ρ_{vf} , k_{vf} , and μ_{vf} which are evaluated at the film temperature $[T_f = (T_{wall} + T_{sat})/2]$.

5.4 FILM BOILING

Once the excess temperature exceeds the value at the Leidenfrost point, the heat flux begins to increase as radiation through the vapor film becomes significant. The heat flux at burnout (Point D in Fig. 5.1) is calculated as

$$q_{BO}'' = F_e \sigma (T_m^4 - T_{sat}^4) \quad (5.19)$$

where

$$F_e = \frac{1}{\frac{1}{\epsilon_{cyl}} + \frac{1}{\epsilon_{UF_6}} - 1} \quad (5.20)$$

and
 ϵ_{cyl} = emissivity of cylinder material,
 ϵ_{UF_6} = emissivity of liquid UF_6 ,
 σ = Stefan-Boltzmann constant,
 T_m = melting temperature of the cylinder material (absolute).

The value of the excess temperature at burnout is

$$\Delta T_{BO} = T_m - T_{sat} \quad (5.21)$$

5.5 EFFECT OF THE ROHSENOW CONSTANT ON THE UF_6 BOILING CURVE

The value of the Rohsenow constant affects the location of points A and B in Figure 5.1. Although the value of the critical heat flux is not affected by the value of $C_{s,f}$, the temperature difference at which it occurs is affected. A higher value of $C_{s,f}$ will tend to move point A further up the free convection curve, forcing point B to the right also. The opposite is true if a lower value of $C_{s,f}$ is chosen. Figure 5.2 illustrates the effect of the Rohsenow constant on the UF_6 boiling curve. Figure 5.2 was generated assuming a constant pressure of 22.046 psia. The steel was assumed to have a melting temperature of 2500°F and an emissivity of 0.6. The emissivity of liquid UF_6 was assumed to be 0.9.⁹ As seen from Fig. 5.2, the temperature difference at the onset of nucleate boiling can vary from 0.19 to 11.60°F depending on the value of $C_{s,f}$. The range of $C_{s,f}$ (0.0025-0.025) covers the range of most values for various surface-liquid combinations found in the literature.^{5,7,8} For the same liquid, high values of $C_{s,f}$ imply smooth, polished surfaces while lower values are characteristic of more rough, pitted surfaces. The critical heat flux value at the triple point pressure of 22.04 psia is approximately 86,077 BTU/hr-ft². The corresponding temperature difference ranges from 6.18 to 61.75°F.

5.6 EFFECT OF PRESSURE ON THE UF_6 BOILING CURVE

The UF_6 pressure will affect the appearance of the pool boiling curve since it has a direct effect on the saturation temperature as well as many of the thermal properties. The pool boiling curves for UF_6 at pressures of 22.046 psia, 150 psia, 300 psia, 450 psia, and 600 psia are shown in Figure 5.3. A value of 0.0025 was assumed for the Rohsenow constant. Plots of critical heat flux and Leidenfrost heat flux

Fig. 5.2. Pool boiling curves for UF_6 as a function of the Rohsenow constant.

Fig. 5.3. Pool boiling curves for UF_6 as a function of pressure.

as a function of pressure are shown in Figure 5.4. The critical heat flux, which has a value of approximately 86,077 BTU/hr-ft² at the triple point pressure, attains a maximum value of approximately 140,600 BTU/hr-ft² at a pressure of 200 psia. The critical heat flux value drops to approximately 44,500 BTU/hr-ft² at a pressure of 600 psia. The Leidenfrost heat flux reaches a maximum value of approximately 1,000 BTU/hr-ft² at a pressure of 300 psia. This value is reduced considerably, to approximately 300 BTU/hr-ft², at a pressure of 600 psia. Since both h_{fg} and $(\rho_l - \rho_v)$ have values of zero at the critical point, the critical and Leidenfrost heat flux values also approach zero as the pressure approaches the critical pressure (669 psia).

5.7 EFFECT OF SUBCOOLING ON THE UF_6 BOILING CURVE

If the bulk temperature of the liquid UF_6 is less than the saturation temperature, subcooled pool boiling occurs. The critical heat flux is increased by liquid subcooling as suggested by the following expression by Ivey and Morris ⁸

$$\frac{q''_{crit,sub}}{q''_{crit,sat}} = 1 + \frac{0.1}{\rho_v h_{fg}} \left(\frac{\rho_v}{\rho_l} \right)^{1/4} c_{p,l} \rho_l \Delta T_{sub} \quad (5.22)$$

Fig. 5.4. Critical and Leidenfrost heat fluxes as a function of pressure.

where

$$\Delta T_{sub} = T_{sat} - T_{liq} , \quad (5.23)$$

and T_{liq} = bulk liquid temperature.

Figure 5.5 illustrates the effect of liquid subcooling on the critical heat flux. Since the pressure must be greater than the triple point pressure (22.04 psia) for subcooling to be possible, a pressure of 50 psia was arbitrarily chosen to generate Figure 5.5. As seen from Figure 5.5, the critical heat flux is increased approximately 10% over the saturated value if the bulk UF_6 temperature is 10°F lower than the saturation temperature.

5.8 SUMMARY OF BOILING HEAT TRANSFER COEFFICIENT CORRELATIONS

The following correlation is recommended for natural convection boiling heat transfer:

$$h_{NC} = a \Delta T_e^b , \quad (5.24)$$

where values of a and b are obtained from Eqs. 5.8 and 5.9, respectively. UF_6 free convection boiling heat transfer coefficients can reach a maximum of approximately 50 BTU/hr-ft²-°F.

Fig. 5.5. Critical heat flux as a function of subcooling.

The following correlation is recommended for nucleate boiling:

$$h_{NB} = \frac{C_b \Delta T_e^2}{C_{sf}^3}, \quad (5.25)$$

where the value of C_b is obtained from Eq. 5.13. Heat transfer coefficients in the nucleate boiling region can range from 4 to 95,000 BTU/hr-ft²-°F.

The following interpolation is recommended for transition boiling:

$$h_{TB} = \frac{10^{-m \log \Delta T_e + n}}{\Delta T_e}, \quad (5.26)$$

where

$$m = \frac{\log(q''_{\max}) - \log(q''_L)}{\log(\Delta T_{CHF}) - \log(\Delta T_L)} , \quad (5.27)$$

and

$$n = \log(q''_{\max}) - m \log(\Delta T_{CHF}) . \quad (5.28)$$

Heat transfer coefficients in the transition boiling region can range from 1 to 95000 BTU/hr-ft²-°F.

The following interpolation is recommended for film boiling:

$$h_{FB} = \frac{10^{-u \log \Delta T_e + v}}{\Delta T_e} , \quad (5.29)$$

where

$$u = \frac{\log(q''_{BO}) - \log(q''_L)}{\log(\Delta T_{BO}) - \log(\Delta T_L)} , \quad (5.30)$$

and

$$v = \log(q''_{BO}) - u \log(\Delta T_{BO}) . \quad (5.31)$$

Heat transfer coefficients in the film boiling region are relatively low, approximately 1-35 BTU/hr-ft²-°F.

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