An Evaluation of Loss of HPCI Room Cooler at Dresden Station

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Abstract

The purpose of this analysis is to perform HPCI room heatup calculations subsequent to loss of room coolers in response to questions raised by NRC reviewers during review of previously submitted calculations. The principal features of this reanalysis are the use of ASHRAE based natural convection heat transfer coefficients, explicit modeling of the HPCI room walls including through wall heat transfer, and consideration of potential leakage due to postulated failure of the gland exhaust subsystem.

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1. Introduction

The purpose of this calculation is to respond to questions raised during the review of previously submitted calculations [Reference 1]. Specifically, quantification of the impact of using reduced heat transfer coefficients on the HPCI room surfaces was requested. Additionally, questions were raised concerning the environmental qualification of the HPCI gland exhaust system, and quantification of the impact of postulated steam leakage from the seals was requested. Finally, an interest in the real margins existing in the room heatup calculations was communicated. This calculation has been prepared to document analyses performed in response to these requests.

2. Description of Analysis

All analyses represented in this report were performed with the RELAP 4 Mod 6 computer code as installed on the CECo computer system. This version of the code contains an equation of state for air and has the ability to transport air between nodes. The code has a default minimum heat transfer coefficient of 5.0 BTU/HR/FT2-F, which necessitates adjustment of the geometry of the heat slabs to simulate alternative convection coefficients.

2.1 HPCI Room Heatup with Reduced Convection Coefficients

This case consists of a recalculation of the original analysis, but with a reduced surface heat transfer coefficient. The initial room temperature was set at 120 F versus the 140 F initial condition utilized previously [Reference 1]. This was done to maintain consistency with prior Dresden and Quad Cities calculations. The original calculation assumed a single heat slab of one foot thickness representing the walls, floor, and ceiling areas. The slab was assumed to be single sided for conservatism. The surface heat transfer coefficient used originally was 5.0 BTU/HR/FT2-F consistent with analyses assumptions for the LPCI and Core Spray rooms. A survey of natural convection heat transfer coefficients returned suggested values of 1-5 BTU/HR/FT2-F (see Appendix). A value of one was selected for this case, and the base deck was rerun. A diagram of this model is shown in Figure 1.

2.2 HPCI Room Heatup with Detailed Wall Models

In order to demonstrate the conservative nature of the original calculation, a modification to the original model was prepared, subdividing the heat structures representing the walls, floor and ceiling of the HPCI room into six separate heat structures. This model is shown in Figure 2. The walls were explicitly modeled to their actual thickness (3 feet for walls and ceiling, 4 feet for the floor) and two sided conduction was modeled. Heat transfer to the soil was allowed, although at a greatly reduced heat transfer coefficient (0.05 BTU/HR/FT2-F). The other surfaces used overall heat transfer coefficients recommended by ASHRAE. These coefficients ranged from 1.08 for the floor to 1.63 for the ceiling. The wall heat transfer coefficient was 1.46 BTU/HR/FT2-F. This case is considered to be most representative of anticipated room heatup while still maintaining conservatism relative to the heat load and limiting through wall conduction. A boundary condition of 104 F is assumed for all rooms adjacent to the HPCI room (affects 2 walls and the ceiling), consistent with previous analyses.

2.3 HPCI Room Heatup with Postulated Gland Seal Leakage

An additional set of cases was run to provide insight into the room heatup behavior resulting from postulated leakage of gland seal steam. A review of the UFSAR (Section 6.2.5.3.3.5) indicated that for a fully pressurized turbine casing with a locked turbine rotor, the gland condenser would over pressurize due to loss of cooling water and seal steam would be released to the room. The leakage for this case is 2160 lb/hr, and the HPCI system is assumed to isolate on high room temperature within 15 minutes.[Reference 2] Figure 3 provides a diagram of the gland seal system.

In this scenario, the turbine is assumed to be continuously operating, and cooling water would be supplied to the gland exhaust condenser. The gland exhaust fan is not currently on the EQ list and could be postulated to fail eventually due to elevated room temperatures. In the event of gland exhaust fan failure, the removal of non-condensable from the condenser would be impaired, which would lead to a degradation of the condensation heat transfer. For the purposes of this evaluation, a constant value of seal leakage to the HPCI room of 10% of the locked rotor scenario was analyzed. In reality, one would expect little or no leakage provided cooling water is maintained to the condenser, and the leakage would be anticipated to develop slowly over extended time intervals.

Since significant steam leakage was postulated in this case, the use of a condensing coefficient for the heat transfer slabs is warranted. Several values were utilized ranging from the ASHRAE convection coefficients used in the above analysis to 2.0 BTU/HR/FT2-F (the default minimum Uchida correlation value for high air/water ratios) to 5.0 BTU/HR/FT2-F (judged to be the most reasonable average value over time). [Reference 3]

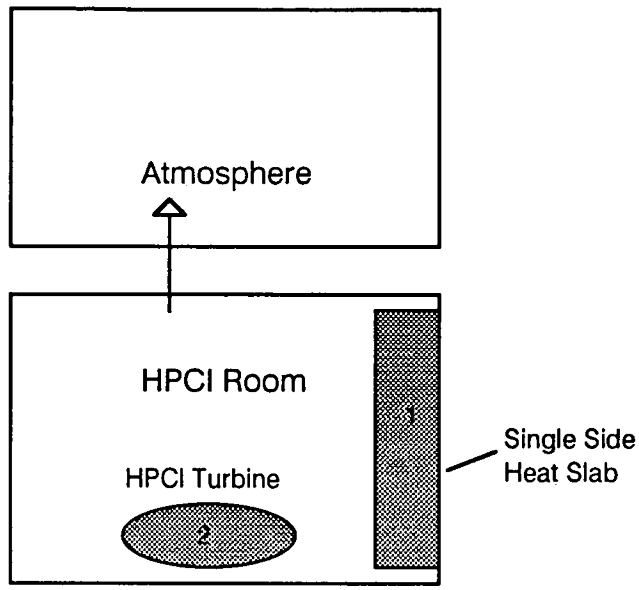


Figure 1 Original HPCI Room Model

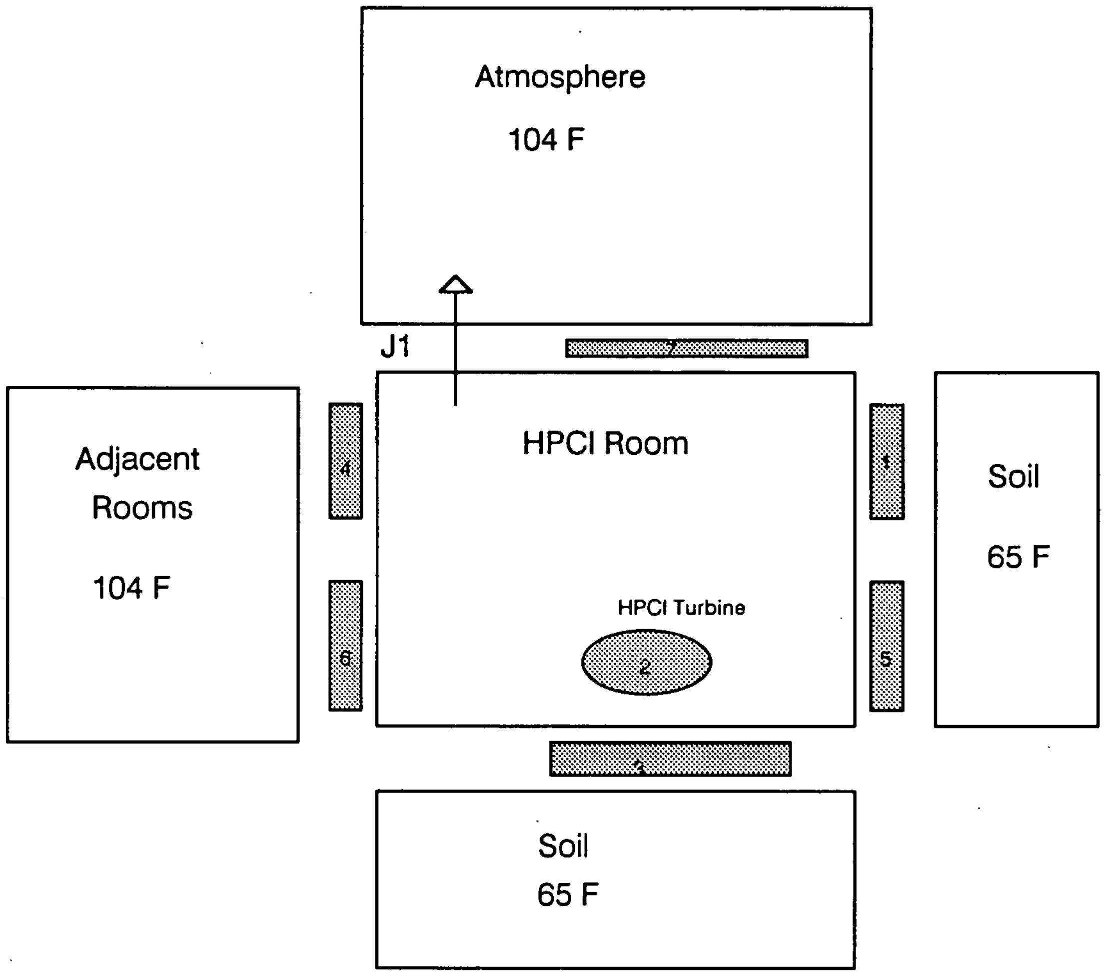


Figure 2 Diagram of Revised HPCI Room Model

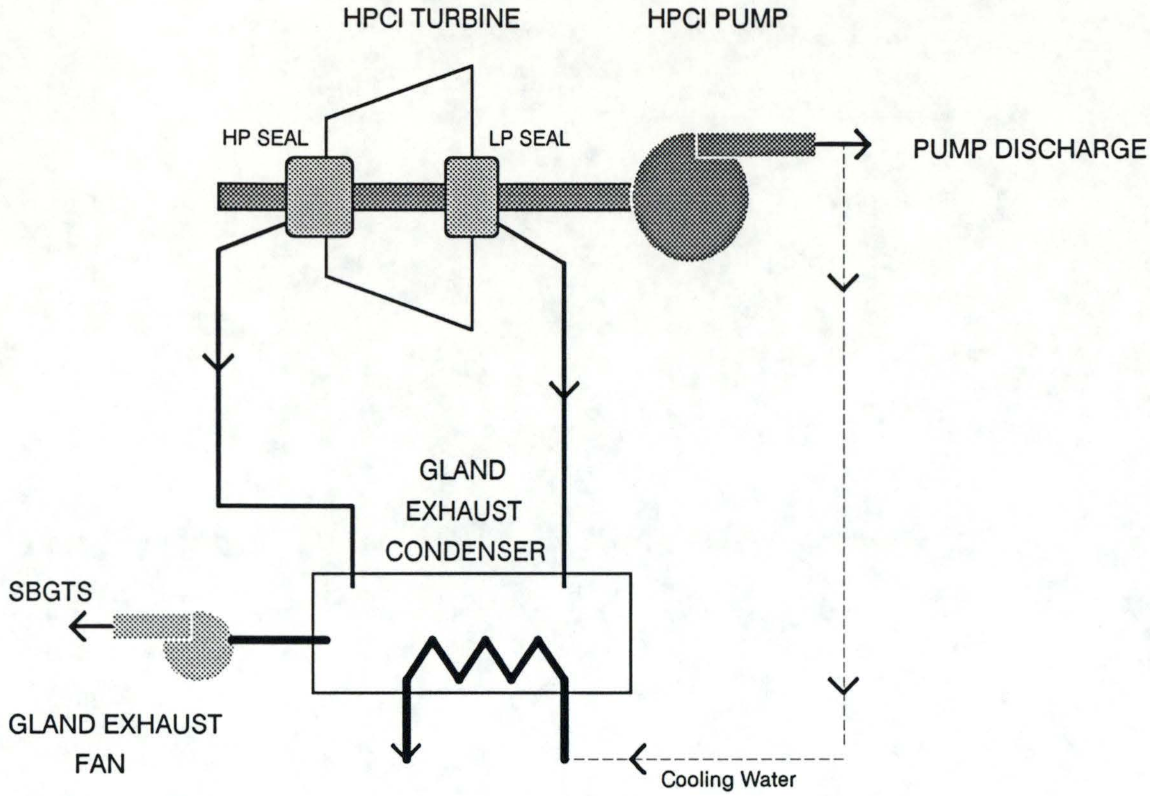


Figure 3 Gland Seal System Flowpaths

3.0 Results of Calculations

3.1 HPCI Room Heatup with Reduced Convection Coefficients

This case resulted in a heatup of the room slightly faster than previously reported. In this case the temperature reached the 185 F limit value at slightly over 40 hours. (Note: Reference 1 reported reaching 185 in approximately 40 hours starting at 140 F, a highly conservative initial temperature; 120 F was employed as an initial temperature in this analysis throughout) The result is shown in Figure 4. As noted above, this case employed a heat transfer coefficient of 1.0 BTU/HR/FT2-F, representing the minimum value of heat transfer that would be anticipated. Note that actual HPCI operations times of less than four hours are actually anticipated. The lack of impact using reduced heat transfer coefficients is believed to result from the concrete conductivity being the limiting parameter in this problem. This is illustrated graphically in the Appendix.

3.2 HPCI Room Heatup with Detailed Wall Models

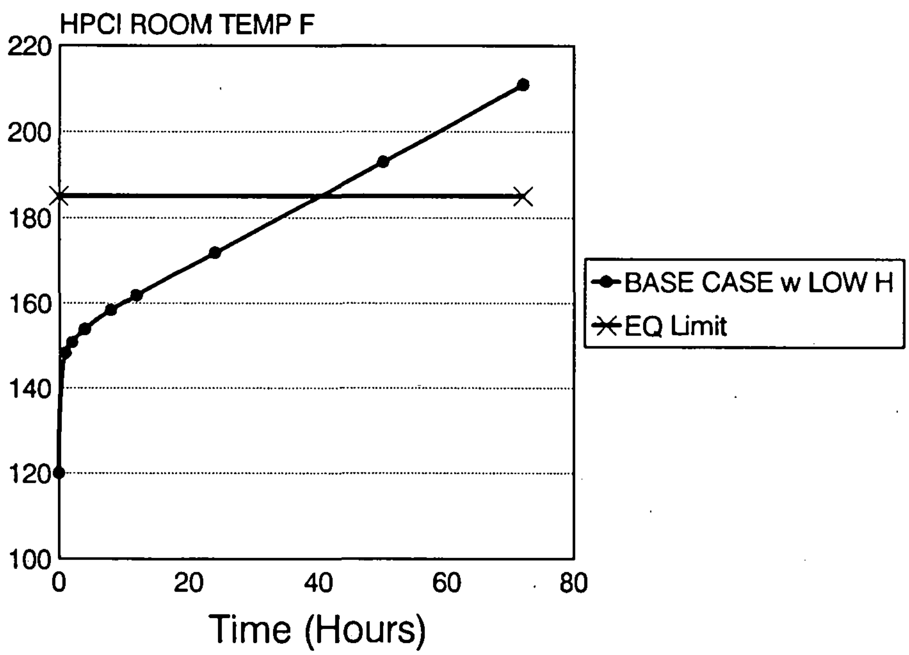
This case was performed to demonstrate the conservatism inherent in the original model. The original model did not credit through wall heat transfer, and assumed that the heat slab thickness was one foot. In reality, the wall thickness is generally 3 foot thick, and through wall heat transfer is expected. In this case the coefficient of heat transfer to the soil was kept deliberately small for conservatism. The selection of other overall heat transfer coefficients was based on ASHRAE tables for still air with non reflective surfaces. The results of this calculation demonstrate that the heatup of the room to 185 F would take nearly three days. This result is shown in Figure 5.

3.3 HPCI Room Heatup with Postulated Gland Seal Leakage

This case builds on the previous case to include the effects of the seal steam leakage into the room following postulated loss of the gland exhaust fan. For this calculation, a seal leakage of 216 lb/hr was assumed, representing 10% of the locked rotor steam leakage rate for a faulted condenser and limiting pressure at both ends of the turbine casing. Since steam addition would lead to condensation on structures, a series of heat transfer coefficients were utilized. Cases of no condensation, minimal condensation, and anticipated condensation were run. The results of these cases are shown in Figure 6. The results demonstrate that the shape of the heatup curve is heavily influenced by the heat transfer coefficient, but the endpoint temperature is relatively constant. These results clearly demonstrate that significant time (12-24 hours) is required to reach EQ limits, even with a high rate of steam leakage to the room.

HPCI ROOM HEATUP

Base Model with Low HTC

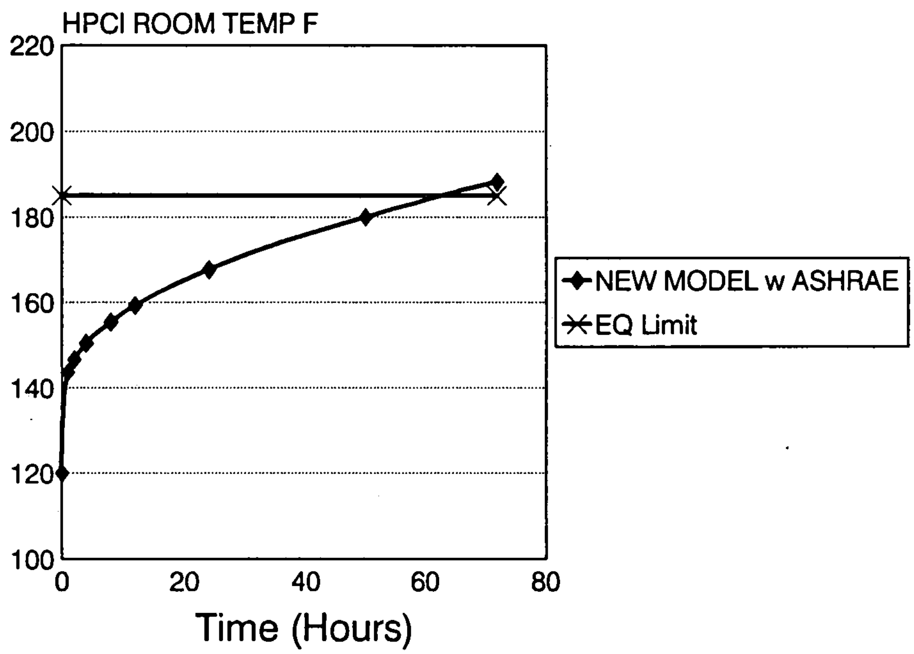


1.0 BTU/HR/FT2-F

Figure 4. Base Case with Reduced Heat Transfer Coefficient

HPCI ROOM HEATUP

Revised Model with Through Wall HT

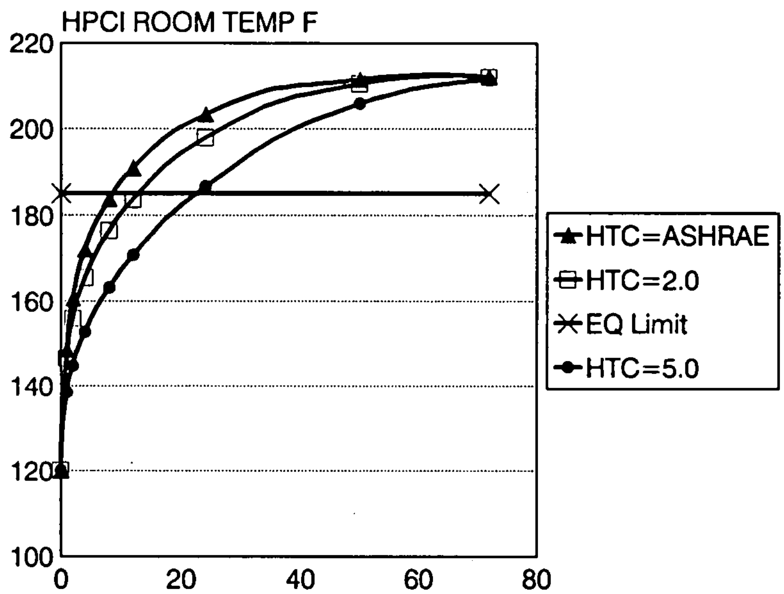


ASHRAE Coefficients

Figure 5. Revised Model with ASHRAE Coefficients

HPCI ROOM HEATUP

Revised Model with Gland Seal Leakage



Range of HT Coefficients

Figure 6. HPCI Room Response to Gland Seal Condenser Leakage

4.0 Conclusions

The HPCI room heatup has been reexamined for a variety of postulated conditions. The original analysis has been shown to be conservative. The more detailed consideration of heat transfer structures illustrates that HPCI operation without ventilation or room coolers could continue for extended periods of time, measured in days, well beyond anticipated operational requirements (less than four hours). These calculations have been performed in a conservative fashion, and assume continuous HPCI operation with heat load to the room constant at the capacity of the room cooler. In the seal steam leakage case, constant seal leakage from start of the HPCI was assumed, ignoring both the anticipated cyclic operation of the HPCI and the time required to achieve degradation of condensation in the gland exhaust condenser. This clearly bounds the manner in which HPCI would operate in transient and accident scenarios, and demonstrates that operation of the HPCI without room coolers does not compromise the operability of the HPCI system.

References

1. "ECCS Pump Room Transient Response to Loss of Room Cooler for Dresden Station Units 2 and 3" , RSA-D-90-01.

2. Dresden UFSAR, current to 8/92.

3. "CONTEMPT 4/Mod5: An Improvement to CONTEMPT 4/Mod4 Multicompartment Containment System Analysis Program for Ice Containment Analysis" , NUREG/CR-4001, BNL-NUREG-51824, C. C. Lin. Section 3.9.1.2.2

Listing of Computer Cases

Job Identifier	Job Number	Case Description
NFSKR1	JO 9141	Base Case with Reduced HT HTC=1.0 BTU/HR/FT2-F
NFSKR2	JO 9062	Revised model no leak HTC=ASHRAE
NFSKR3	JO 8722	Revised model with seal leak HTC=ASHRAE
NFSKR4	JO 8745	Revised model with seal leak HTC=2.0 BTU/HR/FT2-F
NFSKR5	JO 8776	Revised model with seal leak HTC=5.0 BTU/HR/FT2-F

Appendix

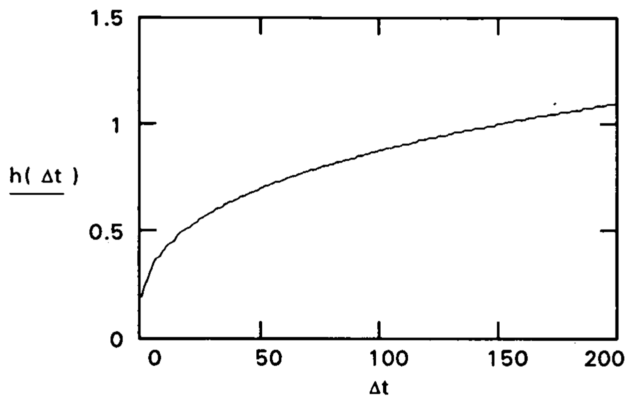
The following pages provide additional discussion of surface heat transfer coefficients in forced and natural circulation systems.

Heat transfer coefficients for air

1. ASHRAE 1981 p. 2.12 Vertical Plates, Table 5, Natural Convection with air
Vertical Plate, Large, Turbulent Range

$\Delta t := 1 .. 200$ Temperature difference in degrees F

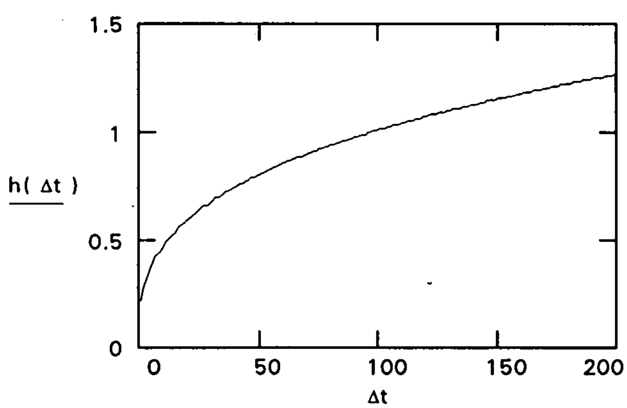
$h(\Delta t) := 0.19 \cdot \Delta t^{0.33}$ Heat Transfer Coefficient in BTU/HR/FT2-F



2. ASHRAE Horizontal Plates facing upward when heated or downward when cooled

$\Delta t := 1 .. 200$ Temperature difference in degrees F

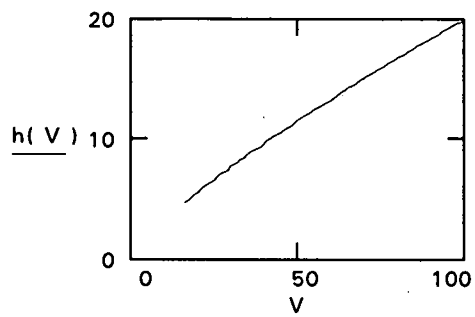
$h(\Delta t) := 0.22 \cdot \Delta t^{0.33}$ Heat Transfer Coefficient in BTU/HR/FT2-F



3. ASHRAE 1981, p 2.15, Table 6, Sec IVa, Forced Convection, vertical plane surfaces velocity of 16 to 100 fps at room temperature

$V := 16 .. 100$ Velocity in Feet/sec

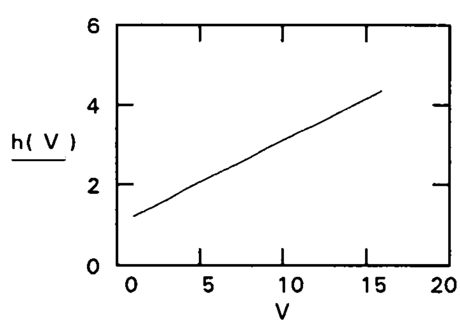
$h(V) := 0.5 \cdot V^{0.8}$ Heat Transfer Coefficient in BTU/HR/FT2-F



4. ASHRAE 1981, p 2.15 Table 6, Sec IVb, Forced Convection, vertical plane surfaces velocity of < 16 fps at room temperature

$V := 1 .. 16$ Velocity in Feet/sec

$h(V) := 0.99 + 0.21 \cdot V$ Heat Transfer Coefficient in BTU/HR/FT2-F



5. Incropera & DeWitt, Introduction to Heat Transfer, 1985.
The free convection heat transfer coefficients for air range from 5 to 25 W/m2-K (.88 to 4.4 Btu/hr-ft2-F)

6. Todreas and Kazimi, Nuclear Systems I, 1989. Table 10-2, p 418
Typical natural convection coefficients for low pressure gas range from 1-5 BTU/hr-ft2-F

7. ASHRAE 1981 Chapter 23 Table 1, p 23.12, Surface Conductances and Resistances for Air

Still Air, Heat Transfer Coefficient in BTU/HR/FT2-F

Horizontal Upward Heat Flow for Non Reflective Surface $h = 1.63$
Vertical Horizontal Heat Flow for Non Reflective Surface $h = 1.46$
Horizontal Downward Heat Flow for Non Reflective Surface $h = 1.08$

These values are employed in the analysis cases described in this report.

Surface Heat Transfer Impact on Overall Heat Transfer Coefficient

The purpose of this calculation is to demonstrate the effect of varying the surface heat transfer coefficients on a typical heat slab such as a HPCI wall. The steady state expression for overall heat transfer coefficient is set up and a variety of surface heat transfer coefficients are employed. A typical concrete wall of 3 foot thickness is assumed. Slab geometry renders the area constant and it is removed from the expression. Three values of outside heat transfer coefficients are used to illustrate the impact of the other side heat transfer.

$k := 1.05 \qquad t := 3$

Let h run from .2 to 5

$h := .2, .4 .. 5$ Heat Transfer Coefficient in BTU/HR/FT2-F

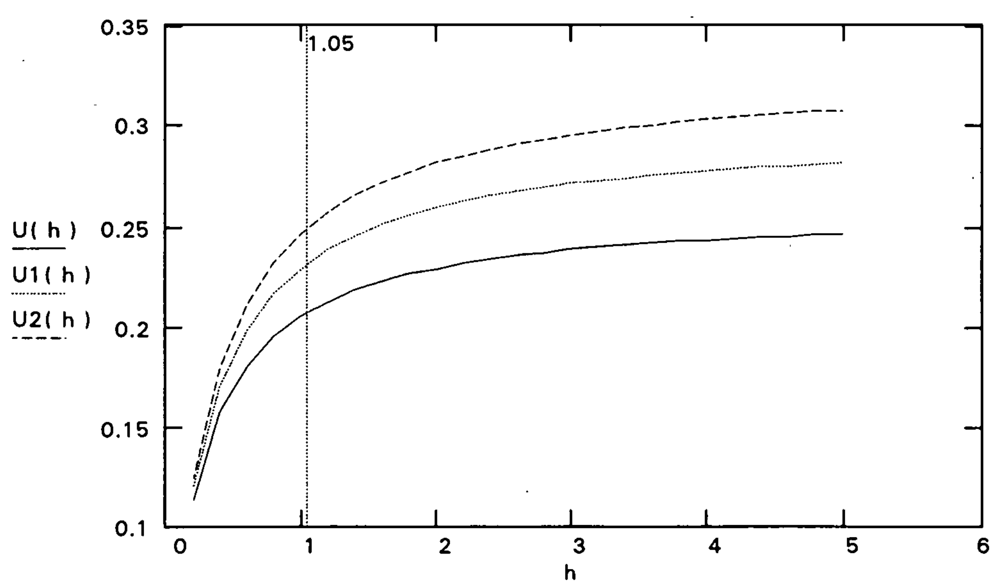
Try outside HTC values for 1,2,5 BTU/HR/FT^2 F

$ho := 1 \qquad ho1 := 2 \qquad ho2 := 5$

Solve the expression for UA, note that A is in all denominator terms and is cancelled

$$U(h) := \frac{1}{\left(\frac{1}{h}+\frac{t}{k}+\frac{1}{ho}\right)} \qquad U1(h) := \frac{1}{\left(\frac{1}{h}+\frac{t}{k}+\frac{1}{ho1}\right)} \qquad U2(h) := \frac{1}{\left(\frac{1}{h}+\frac{t}{k}+\frac{1}{ho2}\right)}$$

Displaying the overall U as a function of h. Note that a reference line representing the thermal conductivity of concrete is added. This is significant in that changes in h show small impact when above this value. (ie. the problem becomes conduction rather than convection limited)



Surface HTC