REHAU MONTANA ECOSMART HOUSE PROJECT
Bozeman, MT
RMEH 03 Test Report

Radiant Floor Heat Transfer Coefficient (HTC): Empirical vs. Theoretical Values

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05/15/2015

Executive Summary

An experiment was performed at the REHAU Montana Ecosmart House (RMEH) in Bozeman in order to evaluate the radiant floor heat transfer coefficient (HTC) from both empirical and theoretical perspectives. Theoretical values were found and calculated according to European standards and ranged from 1.6 to 1.9 Btu/hr·ft^2·°F. Empirical HTC values from experimental data with respect to boiler and floor heat output values ranged from 1.8 - 3.6 Btu/hr·ft^2·°F and 1.2 - 1.4 Btu/hr·ft^2·°F, respectively. Calculations details can be found in subsequent sections of this report.

Experiment Description

The purpose of this experiment was to determine the steady-state floor temperature necessary to achieve different setpoint temperatures and compare the theoretical radiant floor heat transfer coefficient (HTC) to the actual (empirical) HTC. The theoretical values were taken from the European standards, EN 1264 parts 2 and 5 and EN 15377-1, ranging from 1.6 to 1.9 Btu/hr·ft^2·°F. During this experiment a gas-fired condensing boiler was used as the heat source. Heat was distributed throughout the REHAU Montana Ecosmart House (RMEH) using a radiant floor system controlled by room setpoint temperatures. Buffer tank reset control logic was configured within the building control system, REHAU Smart Controls (RSC), to adjust the radiant zone supply temperatures based on space temperature setpoints and outdoor temperature measurements. Each room setpoint temperature scenario was run for a 3-day period. Scenarios were performed at 65, 68, 72 and 75°F room setpoint temperatures.

Results

Collected floor temperatures were weighted by square footage and averaged for each setpoint scenario and adjusted according to a temperature verification field test with an infrared (IR) thermometer. Boiler and floor heat outputs were calculated based on natural gas consumption and methods outlined in chapter 6 of the *ASHRAE Handbook 2012 – HVAC Systems and Equipment* [1]. Summaries of the calculated and published HTC values are listed in Table 1.

Table 1. Steady-state Floor Temperatures, Heat Output and HTC Results

Set-point (°F)	Boiler Heat Output (Btu/hr ft^2)	Floor Heat Output (Btu/hr ft^2)	Steady-State Floor Temp. (°F) (Adjusted)	Zone Temp. (°F) (Adjusted)	Temp. Diff. (°F) (Floor-Zone)	Theoretical HTC (Btu/hr·ft^2·°F)		Empirical HTC (Btu/hr·ft^2·°F)	
						EN 1264-5	EN 1264-2, EN 15377	Based on Boiler Output	Based on Floor Output
65	4.0	1.4	66.9	65.7	1.2	1.9	1.6	3.6	1.2
68	4.6	2.7	70.4	68.3	2.1	1.9	1.7	2.3	1.3
72	5.6	2.1	73.8	72.1	1.6	1.9	1.7	3.5	1.3
75	7.1	5.9	78.0	73.8	4.2	1.9	1.8	1.8	1.4

Both boiler and floor heat output increased steadily with an increase in setpoint as depicted by the 2nd-order polynomial trend lines shown in Figure 1. These results correlate with the increase in net gas consumption reported in the RMEH 02-003 test report. Differences between boiler and floor heat outputs can be attributed to various heat losses occurring in the uninsulated pipes within the mechanical room, pipe distribution to zones, and thermal mass absorption to name a few. According to this study approximately 50% of the heat produced from the boiler was lost before reaching individual radiant zones.

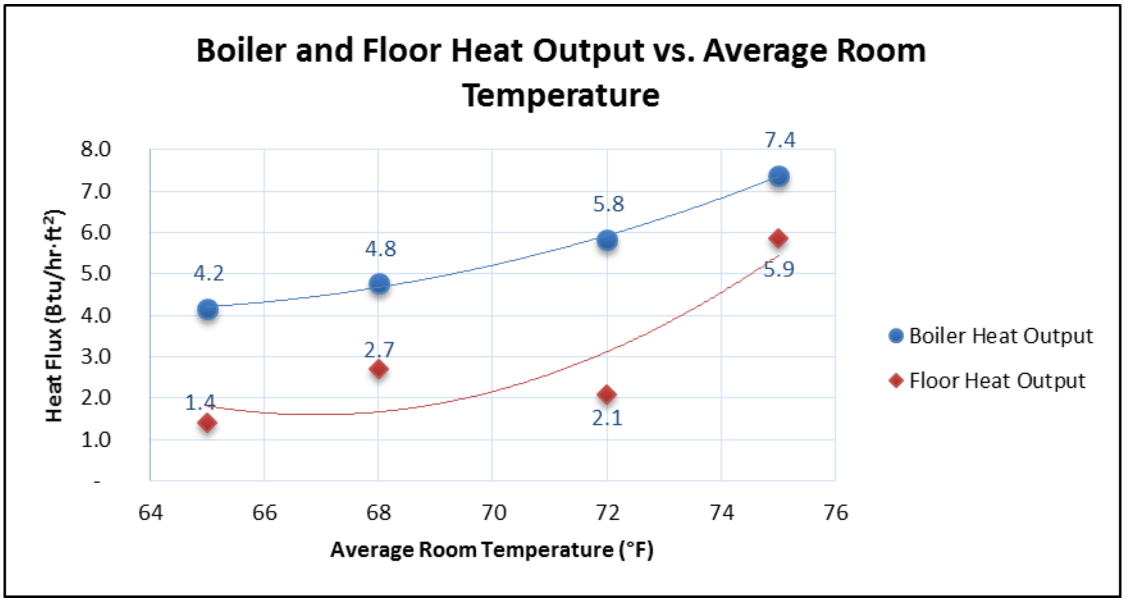

Figure 1. Boiler and Floor Heat Output vs. Average Room Temperature

Calculated HTC values from experimental data with respect to boiler and floor heat output values ranged from 1.8 - 3.6 Btu/hr·ft^2·°F and 1.2 - 1.4 Btu/hr·ft^2·°F, respectively. The method used to calculate the floor heat output was comprised of a radiant component and a natural convection component as shown in the HTC calculations section of this report. Generally a higher HTC signifies better system performance at a given temperature setpoint because less temperature differential is required to deliver the same amount of energy to a radiant slab. Thermal resistance of the floor material has little influence on the overall radiant HTC although it is important when determining the temperature gradient between the heat carrying fluid and floor surface. A graphical representation of the empirical and theoretical HTC values versus average room temperatures is shown in Figure 2.

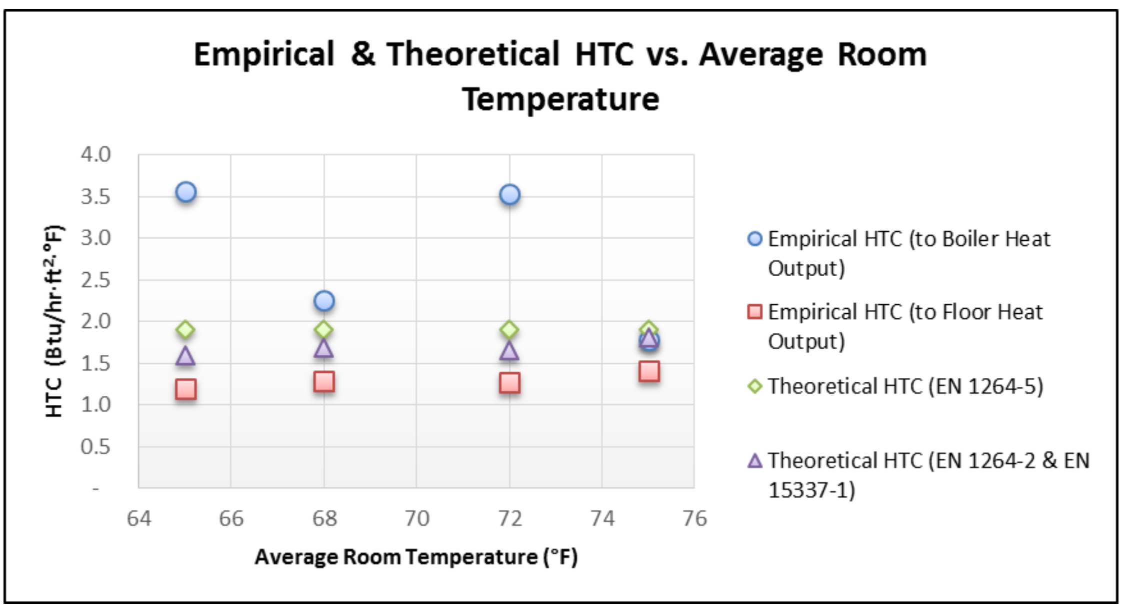

Figure 2. Empirical and Theoretical HTC Values at Different Setpoint Temperatures

The empirical HTC with respect to the floor heat output was noticeably more consistent when compared to the empirical HTC based on boiler heat output. The calculated HTC with respect to boiler output was a function of the boiler output itself and the difference between the average whole house floor and space temperatures. A clear correlation between the HTC based on boiler output was not observed. However, the empirical HTC based on floor output values did trend more closely with theoretical values commonly used in general practice of radiant floor design.

Floor Temperature Calculations

Average room and floor temperatures for each setpoint scenario were collected using RSC sensors and are shown in Table 2. Analysis of this data found multiple instances where the floor temperature was less than the room temperature. From a thermodynamic perspective this is impossible, given that the only source of heat was supplied through the radiant floor system. The test setup had doors between rooms closed to prevent heat migration between zones. Some heat gain within the rooms could be attributed to solar gains through windows during the day, but this phenomenon was even discovered in data points recorded in the middle of the night.

Table 2. Radiant Zone Average Floor and Room Temperatures

ZONE	65°F Setpoint		68°F Setpoint		72°F Setpoint		75°F Setpoint	
	Mean Floor Temp. (°F)	Mean Room Temp. (°F)	Mean Floor Temp. (°F)	Mean Room Temp. (°F)	Mean Floor Temp. (°F)	Mean Room Temp. (°F)	Mean Floor Temp. (°F)	Mean Room Temp. (°F)
Rad Zone LL1RAD- Meeting Room	58.0*	67.9*	58.4*	69.4*	60.0*	72.4*	65.3*	74.2*
Rad Zone LL3RAD- Studio / Bathroom	60.3*	62.6*	66.6	65.5	71.0	70.8	73.1	71.0
Rad Zone LL6RAD – Storage	61.9*	62.6*	65.8	65.5	68.9*	70.8*	72.6	71.0
Rad Zone ML1RAD - Front Entry and Half Bath	67.0	62.9	69.8	67.9	73.4	72.0	74.8	74.9
Rad Zone ML3RAD- Study	63.8*	64.2*	69.6	68.5	73.9	72.4	77.2	73.8
Rad Zone ML4RAD - Dining and Living Rooms	55.9*	65.6*	63.2*	68.8*	67.0*	72.7*	71.0*	74.2*
Rad Zone ML5RAD – Kitchen	61.5	65.6	66.0	68.8	71.2	72.7	73.5	74.2
Rad Zone ML6RAD – Laundry	60.7*	64.9*	65.0*	67.7*	69.5*	71.8*	73.6*	74.3*
Rad Zone UL1RAD - Master Bedroom	65.2	65.1	70.9	68.6	73.5	72.0	78.5	74.0
Rad Zone UL2RAD - Master Bath	67.7	64.1	70.6	67.2	73.5	71.1	78.2	74.0
Rad Zone UL3RAD - Daughters Living Area	62.2*	65.0*	66.3*	67.8*	71.2*	72.2*	74.9	74.2
Rad Zone UL4RAD - Daughters Bed Room	55.4*	65.1*	59.7*	68.6*	63.0*	72.0*	65.1*	74.0*
Rad Zone UL5RAD - Guest Bedroom and Bath	64.5*	66.6*	66.7*	68.6*	71.2*	73.0*	73.1*	74.4*
Rad Zone UL6RAD - Daughters Bath	67.3	64.1	71.4	66.9	75.6	71.2	79.8	73.6
Rad Zone UL7RAD – Hallway	64.9	64.9	68.5	67.9	73.3	71.6	80.2	73.9

Note: Temperatures with an * signify that the floor temperature was less than room temperature.

This suggests that some of the floor or room sensors might not be providing accurate measurements. This might be caused by the location of some floor sensors, close to walls, where the temperature is not representative of the average radiant floor temperature. Commissioning of the control system has not been performed to date at the RMEH; therefore, MSU personnel spent a considerable amount of time checking the accuracy of the RSC sensors. Since room sensors controlled each radiant zone, the lack of calibration of some RSC sensors did not prevent zones from achieving temperature setpoints. With regards to obtaining meaningful empirical values for the radiant HTCs, it was important to have representative floor temperature data. Figure 3 serves as an example of this discrepancy between floor and room temperatures.

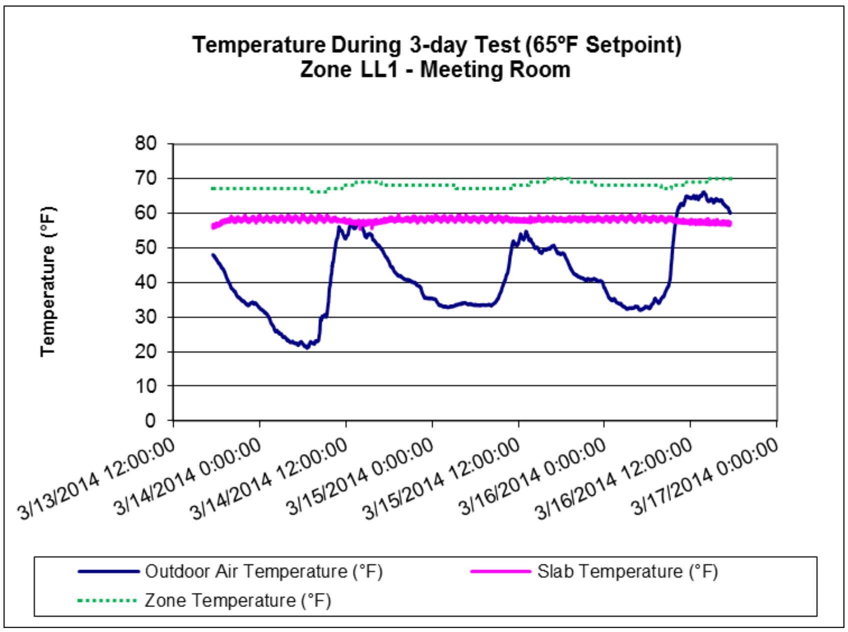

Figure 3. Example of Floor to Room Temperature Discrepancy

As a result of these discrepancies a validation test was performed to compare floor temperatures measured with external devices to those given by the RSC sensors. This field test was carried out on March 14th, 2014 and was concurrent with the RMEH 02-003, 65°F setpoint test.

Devices used for this test consisted of an infrared (IR) thermometer set to an emissivity of 0.95, a digital thermometer with a Type T thermocouple, and a NIST-traceable calibrated thermohygrometer. Readings from the IR and digital thermometers were consistently less than a degree Fahrenheit apart. Based on the repeatable finding from this temperature verification test, these values were used to calculate the temperature difference between the RSC sensors and actual floor temperatures. Some radiant zones were found to have large variations in floor temperature reading between the RSC sensors and the IR thermometer. A summary of the temperature differences between the IR thermometer and RSC sensors for each radiant zone are listed in Table 3. Detailed information about the results of the floor temperature validation test can be provided in a separate file upon request (Test RMEH 03-001 IR-TC Temp Test).

Table 3. Temperature differences Between RSC sensors and IR Thermometer

Zone	Temperature difference (°F)	
	Floor [IR test – RSC]	Room [IR test – RSC]
Rad Zone LL3RAD - Studio / Bathroom	11.6	4.7
Rad Zone LL1RAD - Meeting Room	10.8	-1.4
Rad Zone ML4RAD - Dining and Living Rooms	9.1	-0.8
Rad Zone ML3RAD - Study	7.3	1.6
Rad Zone UL4RAD - Daughters Bed Room	6.6	1.9
Rad Zone ML1RAD - Front Entry and Half Bath	6.4	-1.9
Rad Zone ML6RAD - Laundry	5.2	0.0
Rad Zone ML5RAD - Kitchen	4.3	-0.8
Rad Zone LL6RAD - Storage	3.2	2.6
Rad Zone UL5RAD - Guest Bedroom and Bath	1.2	0.0
Rad Zone UL3RAD - Daughters Living Area	0.8	-1.6
Rad Zone UL6RAD - Daughters Bath	0.5	-2.4
Rad Zone UL1RAD - Master Bedroom	0.0	0.0
Rad Zone UL2RAD - Master Bath	-1.5	0.7
Rad Zone UL7RAD - Hallway	-1.7	0.3

Upon review of the data and verification of the RSC sensors, two programming errors in the RSC system were found in zones ML1 – Front Entry and UL4 – Daughter's Bedroom. Zone ML1 had the room and floor sensor point addresses switched in the control system resulting in the zone being controlled by floor temperature and not by room temperature as intended. With respect to zone UL4, this room never achieved setpoint. This was due to the room being controlled by the wrong thermostat, i.e. the one in the adjacent zone (Zone UL3 – Common Living Area). Since doors were kept closed to prevent heat migration, this caused the room to be always under heated. Both errors were resolved and subsequent analysis excluded these zones from the HTC calculations.

Adjustments were made to the RSC data to account for the measured differences between the IR thermometer and RSC sensors. This resulted in a data set, Table 4, where floor temperatures were consistently above room temperatures and aligned more closely with the expected performance of a radiant floor system. In addition, overall temperature averages were weighted by square footage. It was observed that significantly fewer irregularities occurred after the temperature adjustments, resulting in only a few zones producing mean floor temperatures below their corresponding room temperatures.

Table 4. Adjusted Radiant Zone Average Floor and Room Temperatures

Zone	Zone Area (Sq.Ft.)	Area Weight Ratio	65°F Setpoint		68°F Setpoint		72°F Setpoint		75°F Setpoint	
			Mean Floor Temp. (°F)	Mean Room Temp. (°F)	Mean Floor Temp. (°F)	Mean Room Temp. (°F)	Mean Floor Temp. (°F)	Mean Room Temp. (°F)	Mean Floor Temp. (°F)	Mean Room Temp. (°F)
Rad Zone LL1RAD - Meeting Room	1057	0.300	68.8	67.9	69.2*	69.4*	70.8*	72.4*	76.1	74.2
Rad Zone LL3RAD - Studio / Bathroom	209	0.059	71.9	62.6	78.2	65.5	82.6	70.8	84.7	71.0
Rad Zone LL6RAD - Storage	165	0.047	65.1	62.6	69.0	65.5	72.1	70.8	75.8	71.0

Rad Zone ML3RAD - Study	181	0.051	71.1	64.2	76.9	68.5	81.2	72.4	84.5	73.8
Rad Zone ML4RAD - Dining and Living Rooms	477	0.135	65.0*	65.6*	72.3	68.8	76.1	72.7	80.1	74.2
Rad Zone ML5RAD - Kitchen	198	0.056	65.8	65.6	70.3	68.8	75.5	72.7	77.8	74.2
Rad Zone ML6RAD - Laundry	112	0.032	65.9	64.9	70.2	67.7	74.7	71.8	78.8	74.3
Rad Zone UL1RAD - Master Bedroom	191	0.054	65.2	65.1	70.9	68.6	73.5	72.0	78.5	74.0
Rad Zone UL2RAD - Master Bath	198	0.056	66.2	64.1	69.1	67.2	72.0	71.1	76.7	74.0
Rad Zone UL3RAD - Daughters Living Area	250	0.071	63.0*	65.0*	67.1*	67.8*	72.0*	72.2*	75.7	74.2
Rad Zone UL5RAD - Guest Bedroom and Bath	232	0.066	65.7*	66.6*	67.9*	68.6*	72.4*	73.0*	74.3*	74.4*
Rad Zone UL6RAD - Daughters Bath	76	0.022	67.8	64.1	71.9	66.9	76.1	71.2	80.3	73.6
Rad Zone UL7RAD - Hallway	176	0.050	63.2*	64.9*	66.8*	67.9*	71.6	71.6	78.5	73.9
Total Weighted Average Temp.	**3522**	**Σ=1**	**66.9**	**65.7**	**70.4**	**68.3**	**73.8**	**72.1**	**78.0**	**73.8**
Temp. Diff. (Floor – Room)			**1.2**		**2.1**		**1.6**		**4.2**	

Note: Zones ML1 and UL4 were excluded from this calculations due to RSC programming errors yielding unusable data from those zones. Temperatures with an * signify that the floor temperature was less than room temperature.

The plot shown in Figure 3 represented the original RSC sensor data. Figure 4 shows the adjusted RSC data based on the calculated offset values from the floor temperature verification testing. For this particular instance the floor temperatures exceeded room temperatures as expected. A complete set of graphs representing the original and adjusted floor and room temperatures can be provided in a separate file upon request (Test RMEH 03-001 IR-TC Temp Test).

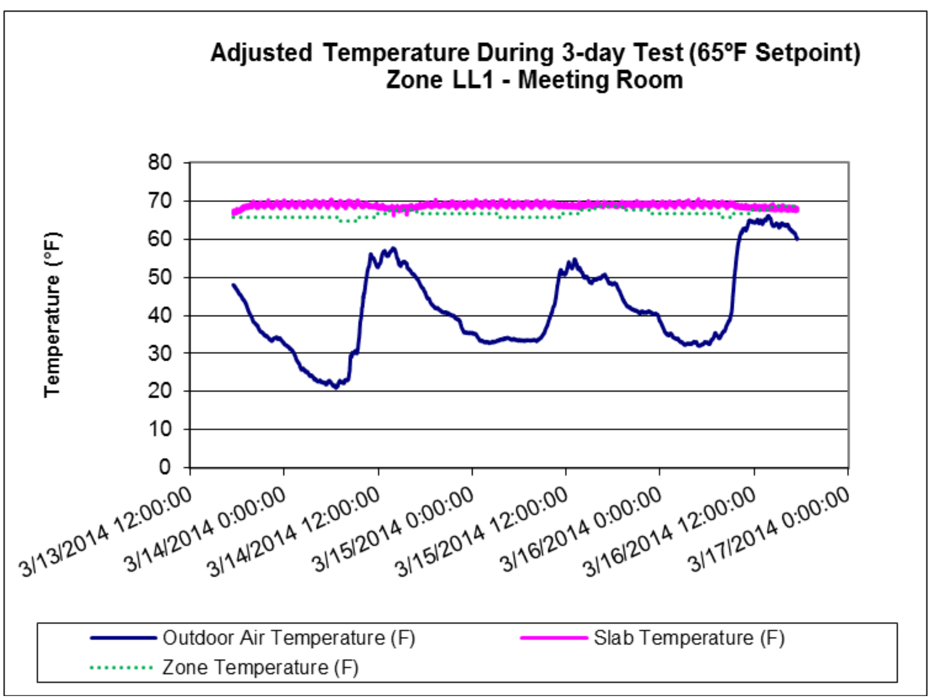

Figure 4. Example of Floor to Room Adjusted Temperatures

HTC Calculations

Theoretical values of HTC were taken from EN 1264-5, EN 1264-2 and EN 15377 (Cholewa et al, 2013). HTC value from EN 1264-5 is constant and equal to

$$HTC_{EN\ 1264-5} = \ 10.8\ W/m^2 \cdot K = 1.9\ Btu/hr \cdot ft^2 \cdot °F$$

A more elaborated HTC is given in EN 1264-2 and EN 15377, and is the result of the following equation:

$$HTC_{EN\ 1264-2} = \ 8.92 \cdot (T_s - T_{op})^{0.1}$$

Where,
T_{op} = operative temperature (°F)
T_s = radiant surface temperature (°F)

In order to calculate the empirical Heat Transfer Coefficient (HTC) of the radiant floors, the following general formulas were used (Cholewa et al, 2013):

$$h_{total} = \frac{q_{total}}{T_{op} - T_s}$$

Where,
h_{total} (HTC) = radiant heat transfer coefficient (Btu/hr·ft^2·°F)
q_{total} = heat output per unit area (Btu/hr·ft^2)
T_{op} = operative temperature (°F)
T_s = radiant surface temperature (°F)

In calculating T_{op}, mean radiant temperature (MRT) comes into play; but due to the complexity of calculating a weighted and averaged MRT for the whole house, it was assumed that $T_{op} = T_a$, where T_a is the ambient temperature as measured by room sensors.

Total heat output (q_{total}) can be looked at from two perspectives. One is the total heat output by the boiler, which can be calculated by looking at the boiler gas consumption during the test period (Wujek, 2010).

$$Heat\ Output\ per\ unit\ area\ = \frac{NG \cdot HV_{NG} \cdot \eta_{hs}}{Time\ span \cdot Radiant\ Area}$$

Where,
NG = natural gas consumption (standard ft^3)
HV_{NG} = heating value of the natural gas (Btu/ft^3)
η_{hs} = boiler thermal efficiency (dimensionless)
Time span = total time of the experiment (hours)
Radiant Area = effective area of the radiant panel excluding obstructions like kitchen islands (ft^2).

Natural gas (NG) consumption was calculated from boiler gas meter readings. The heating value of the natural gas (HV_{NG}) for the test period was calculated by multiplying the average BTU factor (Btu/ft^3) with the conversion pressure factor found in the utility bills. Northwestern Energy utility bills can be provided in a separate file upon request (Test RMEH 03-001 Utility Bills).

$$HV_{NG} = 1020.591 \cdot 0.8483221 = 866.1\ Btu/ft^3$$

The thermal efficiency of the condensing boiler was assumed to be the manufacturer published annual utilization efficiency (AFUE) of the boiler, which is 96% (Triangle Tube – Prestige Solo Lit., 2011). The time span of the experiment was 72 hours and the radiant area was calculated to be 3552 ft^2.

It was determined that the calculated HTC do not account for the different floor arrangements, therefore, it should be considered an averaged radiant HTC for the whole house. *It is recommended that separate tests be performed on each floor individually in order to assess possible differences in HTC values depending on different floor constructions.*

For example, the net gas consumption was 1456 Standard Cubic Feet (SCF) during the 68 °F setpoint test. A heat output per unit area of 4.8 Btu/hr·ft^2 and a room to floor temperature difference of 2.1°F was calculated. This resulted in a HTC value of 2.3 Btu/hr·ft^2·°F for the 68 °F setpoint test.

$$HTC = \frac{4.8}{2.1} = 2.3\ \frac{Btu}{hr\ ft^2\ °F}$$

However, there were ample differences in the HTC calculated for the different setpoints, ranging from 1.8 Btu/hr·ft^2·°F for 75°F, to 3.6 for 72°F. The average HTC value for the four setpoints was 2.8 Btu/hr·ft^2·°F (see Table 1).

These estimated HTC values are above the published HTC values used in common practice (1.6 to 1.9 Btu/hr·ft^2·°F). This might be connected to the fact that the total heat supplied to the system through the boiler is not the same as the total heat actually emitted by the radiant panel (Cholewa et al, 2013). Therefore, it is important to estimate the real amount of heat emitted by the radiant surface. This total heat can be estimated by methods outlined in chapter 6 of the ASHRAE Handbook 2012 – HVAC Systems and Equipment [1].

$$q_{total} = q_r + q_c$$

Where,
q_{total} = total heat flux by radiant panel (Btu/hr·ft^2·°F)
q_r = net heat flux because of thermal radiation on active (heated or cooled) panel surface (Btu/hr·ft^2·°F)
q_c = heat flux from natural convection (Btu/hr·ft^2·°F)

The heat flux from thermal radiation can be approximated by

$$q_r = 0.15\times10^{-8}\left[\left(t_p + 459.67\right)^4 - (AUST + 459.67)^4\right]$$

Where,
T_p = effective panel surface temperature, °F
AUST = area-weighted temperature of all indoor surfaces of walls, ceiling, floor, windows, doors, etc. (excluding active panel surfaces), °F

In calculating AUST, the temperature of outdoor exposed surfaces has influence. For this approximation it was assumed that AUST = ambient temperature (T_a), given the high level of insulation and thermal mass of the RMEH. T_p is equal to T_s (radiant surface temperature as read from the sensors) as was used before in this report. The empirical HTC values with respect to boiler heat output and floor heat output are provided in Table 5.

The heat flux due to natural convection is approximately

$$q_c = 0.31\left|t_p - t_a\right|^{0.31}\left(t_p - t_a\right)$$

Table 5. Empirical HTC Based on Boiler and Floor Heat Outputs

Setpoint (°F)	Boiler Heat Output per unit area (Btu/hr ft^2)	Radiation heat flux (Btu/hr ft^2)	Convection heat flux (Btu/hr ft^2)	Floor Heat Output per unit area (Btu/hr ft^2)	Temp. Diff. ($t_p - t_a$) (°F)	Empirical HTC (Btu/hr·ft^2·°F)	
						Based on Boiler Output	Based on Floor Output
65	4.0	1.02	0.38	1.40	1.2	3.6	1.2
68	4.6	1.87	0.82	2.70	2.1	2.3	1.3
72	5.6	1.49	0.60	2.09	1.6	3.5	1.3
75	7.1	3.85	2.01	5.86	4.2	1.8	1.4

It can be noticed that the HTC calculated with respect to floor radiant emission was very consistent, around 1.3 Btu/hr·ft^2·°F, and around half the average HTC calculated with respect to boiler output (2.8 Btu/hr·ft^2·°F). It was concluded that approximately 50% of the boiler's heat output was lost before to it reached the radiant zones throughout the RMEH. This could be viewed as a significant loss of energy, but for this specific installation it does not appear to be unreasonable given the temperature losses that can occur due to uninsulated pipes in mechanical room, pipe distribution, thermal mass absorption, etc.

This study is based on adjusted temperature data that accounted only for the temperature differences between RSC sensors and field measurement devices for the 65°F temperature setpoint test. Additional temperature testing and calibration of the RSC sensors would be advised along with performing this experiment again to come up with more consistent conclusions about the radiant heat transfer coefficient at the RMEH.

Appendix A. Test Schedule Sheet

Test Number:	RMEH 03-001		
Description:	Evaluate calculated radiant floor HTC to actual HTC values by comparing empirical floor temperatures and heat output values with theoretical values.		
Objectives:			
	1 Determine actual floor temps to achieve room setpoint temperatures		
	2 Run scenarios for 65, 68, 72F, 75F room setpoint temperatures		
	3		
	4		
	5		
Data Collection Parameters:		**Description**	**Source**
	1	OA Temp	RSC
	2	Zone Set Point Temp	RSC
	3	Zone Actual Temp	RSC
	4	Slab Sensor Temp	RSC
	5	Slab Set Point Temp	RSC
	6	HDD	MSU
	7	Boiler Gas Usage	MSU
	8		
	9		
	10		
	11		
	12		
	13		
	14		
	15		
	16		
	17		
	18		
Test Duration:			
	Length	3 day test at each setpoint temp	
	Start Date		
	End Date		
Deliverables:			
	1 Determine steady state floor temperature based on setpoint temp		
	2 Calculate HTC values (theoretical v. actual)		
	3		
	4		
	5		
Notes:			
MSU Notes:	testing in January 2014		

Appendix B. Experiment Notes

Date of experiment: RMEH 03-001 took place between 3/13/14 - 4/5/14, concurrently with test RMEH 02-003.

Data for experiment RMEH 03-001 was collected during the following dates:

- Scenario 1 – 65°F Setpoint: 13/Mar/2014 – 16/Mar/2014
- Scenario 2 – 68°F Setpoint: 17/Mar/2015 – 20/Mar/2014
- Scenario 3 – 72°F Setpoint: 02/Apr/2014 – 05/Apr/2014
- Scenario 4 – 75°F Setpoint: 29/Mar/2014 – 01/Apr/2014

Appendix C. Data Collection Parameters

RSC and National Instruments (NI) data acquisition systems were used to collect data for this experiment. Data was collected for the following points:

<u>RSC Data Points</u>

- Outdoor Air Temperature
- Zone Setpoint Temperature
- Zone Actual Temperature
- Slab Sensor Temperature
- Buffer Tank Temperature
- Buffer Tank Reset Setpoint Temperature
- Mixing Valve Reset Setpoint Temperature

<u>NI Data Points</u>

- Gas Consumption from Boiler

Appendix D. References

1. ASHRAE (2012). *2012 ASHRAE Handbook - HVAC Systems and Equipment*. Chapter 6.

2. European Standard (2009). *EN 1264-2 - Water based surface embedded heating and cooling systems – Part 2: Floor Heating: Prove methods for the determination of the thermal output using calculations and test methods*. Berlin: CEN.

3. European Standard (2009). *EN 1264-5 - Water based surface embedded heating and cooling systems – Part 5: Heating and cooling surfaces embedded in floors, ceilings and wall – Determination of the thermal output*. Berlin: CEN.

4. European Standard (2008). *EN 15377-1 - Heating systems in buildings - Design of embedded water based surface heating and cooling systems - Part 1: Determination of the design heating and cooling capacity*. Berlin: CEN.

5. Cholewa, T., Rosinski, M., Spik, Z., Dudzinska, M. R., Siuta-Olcha, A. (2013). On the heat transfer coefficients between heated/cooled radiant floor and room. *Energy and Buildings 66:*599–606.

6. Wujek, J.B. and Dagostino F. R. (2010). *Mechanical and Electrical Systems in Architecture, Engineering and Construction.* 5th Edition. Upper Saddle River, New Jersey: Pearson.