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Using Radiant Cooled Floors to Condition Large Spaces and Maintain Comfort Conditions

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ABSTRACT

This paper describes the development of a hybrid conditioning system that creates a comfortable indoor environment in a building. The operation of a variable-volume displacement conditioning system and a radiant cooled floor have been optimized to reduce the building load. Control strategies were developed that optimize energy consumption and contain moisture levels within specified limits. The development of conditioning-only occupied zones is shown and how the overall energy consumption is reduced. Its application in a large airport is described.

INTRODUCTION

Radiant cooled floors have been successfully used to maintain comfort conditions in many buildings, but their use as part of a hybrid system to condition a large airport has not yet been undertaken. To further complicate matters, the airport is in Thailand, which necessitates a careful control of humidity within the conditioned spaces. Another complexity added to the design was the load fluctuation due to the diversity of the airport.

In recent years, several studies have been made on the concept of radiant cooled floors. Borressen (1994) and Simmonds (1993, 1994) reported on a radiant cooled floor design for a museum, and Schlappman (1996) reported on the functioning of the radiant floor system two years after it started operating. Holmes and Wilson (1996) reported on a slab cooling system being used to condition a building in London.

Many papers have been written on improved comfort conditions using radiant systems for both heating and cooling. (Olesen 1997; Simmonds 1993, 1994). Large areas have previously been conditioned using radiant systems, but a 200,000 m^2 radiant cooled floor of the 500,00 m^2 International Airport in Bangkok proved quite a challenge. Traditionally, airports are conditioned by ventilation systems that vary either flow or temperature or both to the space. Simmonds (1996a, 1996b) described the design process for the displacement ventilation system proposed for this airport. Holst et al. (1998a, 1998b) described the advantages of using a radiant cooled floor together with a variable-volume displacement system. This report also showed how the envelope design was optimized to reduce the heat guide to the space and reduce energy consumption.

BUILDING CONFIGURATION

The main terminal building is rectangular in shape and is constructed from single clear glazing and PTFE materials. The roof of the terminal and parts of the concourses are shaded, or partly shaded, by solar shading devices and roof overhangs. However, a majority of the concourses have no external shading devices. The PTFE material does provide a sufficient barrier for direct solar, but the single clear glazing offers very little resistance.

The thermal resistance of both PTFE and single glazing is relatively low, and the conductive heat gain to the space would be quite high if traditional indoor temperatures of 24°C were to be maintained. During April, the warmest month, ambient temperatures vary from 34°C to 36°C and there would be a 10°C to 12°C temperature difference across the envelope. A variable-volume displacement system was designed to maintain the required temperature in the occupied zone (24°C). Stratification would be enhanced by this system so that temperatures on the inside of the building envelope would be

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nearly equal to the ambient conditions, virtually eliminating the convective heat gain to the space. The design and evaluation of the variable-volume displacement system have been discussed by Simmonds (1996a, 1996b).

BUILDING LOAD

The calculation of peak loads for a building of this magnitude are obviously complex. In total, four building load and simulation tools were used to determine and cross-reference the loads. The load to the concourse was 97 W/m^2. Holst and Welfouder (1998) showed that the radiant cooled floor can absorb up to about 50 W/m^2 of short- and longwave radiation. Convection and conduction components provide another 30 W/m^2 of cooling from the radiant floor. Designing a radiant floor to handle a capacity of 80 W/m^2 is critical. The water supply temperature is limited to 13°C because of floor surface condensation risk. Preliminary analysis using a 17°C return temperature showed that the mass flow and temperature differential were inadequate for the required 80 W/m^2 capacity.

The Bangkok climate, which has high air temperatures and solar radiation, leads to enormous solar gains and, therefore, to high cooling loads and low comfort because of high operative temperatures in highly transparent buildings. The optimization process for the membrane roof construction and the glazing leads to a highly reduced solar transmission and, therefore, to low solar-induced cooling loads in the concourses of the new Bangkok International Airport. Table 1 describes the material properties. Table 2 lists the solar radiation, longwave radiation, and air flows between zones. Table 3 notes the internal loads.

DESIGN PROCEDURE

The design approach was rather unique in that two authors (Holst and Reuss) were working with a team in Germany and Simmonds and Gaw and a team were working in America on the practical design of the airport.

Holst used TRNSYS and CFD to simulate the indoor environmental conditions and to model envelope design. The envelope of the concourses consists of a PTFE membrane and glass section. Special attention was paid to the coating of the glazing. The inside surface of the laminated glass had a low-e coating. The low emissivity would redirect the cool radiation effect from the floor back into the space. This can be identified as follows:

$$A_1\, \varepsilon_1\, \sigma\, T_1^4 = A_2\, \varepsilon_2\, \sigma\, T_2^4$$

where

A_1 = area of surface 1,
A_2 = area of surface 2,
ε_1 = emissivity of surface 1,
ε_2 = emissivity of surface 2,
σ = Stefan-Boltzmann constant,
T_1 = temperature of surface 1,
T_2 = temperature of surface 2.

TABLE 1
Material Properties

Element	Construction	U-Factor
Floor	5 cm insulation, conductivity 0.04 W/m·K	0.66 W/m^2·K
Wall (Glass Concourses)	Laminated glass with low-e coating inside	4 W/m^2·K
Wall (Glass Terminal)	Single glazing	5 W/m^2·K
Wall (Glass Offices)	Insulated glass	1.5 W/m^2·K
Roof (Glass Concourses)	Laminated glass with low-e coating inside	4 W/m^2·K
Roof (Membrane Concourses)	Glass fiber PTFE + baffles + inner membrane	2.5 W/m^2·K
Roof (Glass Terminal)	Laminated glass with low-e coating inside	4 W/m^2·K
Roof (Glass Offices)	Roof + insulated glass	0.91 W/m^2·K

TABLE 2
Solar Radiation, Longwave Radiation, and Air Flow Between Zones

Peak external solar radiation	1100 W/m^2
Mean solar transmission, membrane	0.02
Mean solar transmission, fritted glass (roof)	0.035
Mean solar transmission, typical concourse roof (membrane + fritted glass)	0.028
Mean solar transmission, airside center roof (membrane + fritted glass)	0.026
Mean solar transmission, airside center cross section roof (fritted glass)	0.037
Mean solar transmission, terminal skylights + louvers	0.009
Mean solar transmission, IGU roof office block	0.65
Emissivity of low-e coated glass	0.17
Emissivity of scratch-resistant low-e coated foil on membrane roof	0.37
Mean emissivity of typical concourse roof (glass + membrane)	0.32
Mean emissivity of airside center cross section roof (glass)	0.17
Mean emissivity of terminal skylights (glass + construction)	0.25
Coupling between conditioned zone and upper unconditioned stratified zone	30°C
1 air exchange per hour (Terminal Hall, Level 4 Arrivals Hall)	0.2 air exchanges per hour

TABLE 3
Internal Loads for Typical Sections

Description	Persons		Light	Equipment
	(m^2/person)	(W/m^2)	(W/m^2)	(W/m^2)
Circulation Corridor - with people mover - without people mover	 19 19	 4.74 4.74	 10 10	 5 0
Holdroom	3	30	10	0
Office	14	6.43	15	20
Retail	3	30	35	10
Atrium Transfer Lounge Business Lounge Central Waiting Lounge (like Holdroom)	10 10 5 3	9 9 18 30	10 10 15 10	0 5 10 0
Employee Facilities	14	6.43	10	15
Concessions	5	18	15	15
Bus Gate Vest.	10	9	10	0
Bus Gate	3	30	10	0
Airline Office	10	9	15	15
Arrival Hall/Baggage Reclaim	5	18	15	5
Baggage Re-check	15	6	15	15
Hotel	5	18	15	10
Custom Gate	10	9	15	15
Custom Office	10	9	15	15
Departure Hall	4	22.5	10	5
Gallery	5	18	10	5
Kitchen	15	6	15	30
Meeter/Greeter Lobby	5	18	15	5
Lobby	19	4.74	10	0
Ramp	10	9	10	0
Restaurant	5	18	15	5
Restaurant/Lounges/Shops	6	18	20	10

The reduction in emissivity of surface 2, the glazing, from 0.84 to 0.12 improves the radiation effect from the floor.

Once the heat load to the space had been determined, the conditioning system was developed. The loads for the concourse were 97 W/m^2, and the conditioning systems to remove this heat from the space was divided as follows:

Radiant floor, 80W/m^2

Ventilation air, 17W/m^2*

The ventilation air volumetric supply rate was calculated using a supply temperature of 18°C. The coils for the air-handling units could produce 16°C, which could be supplied to the space by increasing the temperature difference between required space conditions and air supply temperatures—any fluctuation in the space load could be quickly picked up by the air system while the radiant floor followed. The extra cooling capacity of the air could also be used as a safety measure in the event that the humidity levels were too high.

Results of the analysis provided by Holst show that the floor can absorb up to about 50 W/m^2 of short- and long wave radiation; convection and conduction components provide the other 30 W/m^2 of cooling from the floor. Because 50 W/m^2 could be absorbed into the floor, the pipe layout and mass flow of water became critical. The water supply temperature was limited to 13°C because of the condensation risk at the radiant floor surface.

TABLE 4
Cooling Power of Radiant Floor

Description	Peak Power of Floor Cooling Refer. to Cooled Area (W/m^2)	Tube Spacing (mm)	Cooled Area to Total Floor Area (%)	Peak Power of Floor Cooling Refer. to Total Area (W/m^2)
Holdrooms	80	150	80	64
Circulation Level 2	70	200	64	45
Circulation Level 3	80	150	69	55
Waiting Lounges	80	150	91	72
Terminal Level 4	70	200	86	60
Terminal Level 6	70	200	84	59
Terminal Level 7	70	200	90	63

Preliminary analysis simulated a system using a 17°C return temperature. The mass flow and temperature differential were inadequate for the 80 W/m^2 capacity. The maximum return water temperature was increased to 19°C and the internal pipe diameter was 20 mm. This resulted in lower mass flow and improved heat transfer to the water flowing through the floor.

To control the cooling output from the radiant floor, a controller would only control the supply water temperature at a constant 13°C. The return water temperature can vary between 13°C (no load) and 19°C (full load). The flow through the floor will be constant. Table 4 shows the different radiant cooled floor capacities.

Design temperatures were:

Design air temperature	24°C
Ambient design temperature	36°C
Mean floor temperature (mix of cooled + noncooled areas)	27°C
Design mean inside surface temperature of roof construction concourses	55°C
Design mean inside surface temperature of roof construction terminal skylights	45°C

The outside air demand was:

Open areas	17 m^3/h per person
Enclosed areas	26 m^3/h per person
Airline offices	34 m^3/h per person

MATERIALS DEVELOPMENT

The glazing system for the concourses consists of two 6 mm glass layers. A clear float glass on the outside with a double ceramic fritting reduces the solar transmission. The fritting is white toward the outside to reach a high solar reflection and black toward the room side. The laminate layer contains a coated sun protection foil to achieve a better selectivity (i.e., daylight/solar transmittance). The inside glass pane is tinted and low-e coated on the surface facing the room.

The fritting pattern varies from 76% fritting to 20% from the roof panes to the vertical panes. The low-e coating on the inside glass pane is a standard pyrolytic coating, which is a state-of-the-art coating used for heat protection glass. Due to the low-e coating on the inside surface, the total solar energy reaching the floor surface is reduced dramatically.

Figure 1 shows the effect of low-emission IR coating on the internal conditions. The standard situation without floor cooling and low-e coating has a PMV of 1.85 and PPD of 70%. A low-e coating does not improve conditions. The addition of floor cooling improves conditions to PMV = 1.05 and PPD = 31%. Results obtained from simulating a radiant cooled floor and a low-e coating on the inside of the glass improve conditions to PMV = 0.44 and PPD = 10%.

RESULTS

To simplify calculating the thermal load for the entire airport, the complex was divided into representative zones.

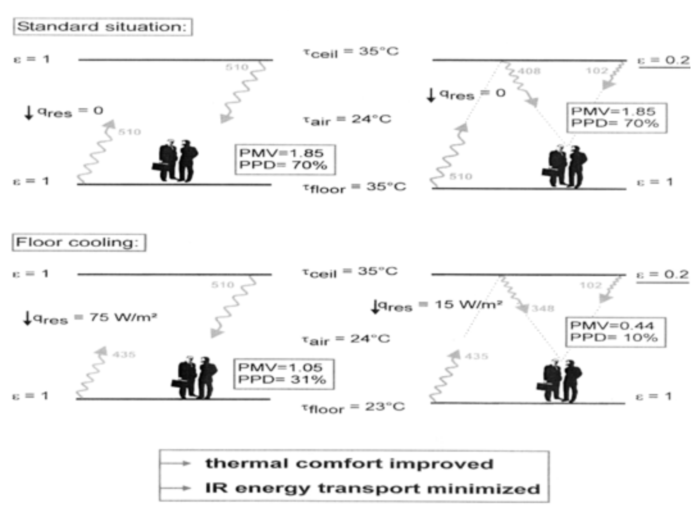

Figure 1 *The effect of low emission IR coating.*

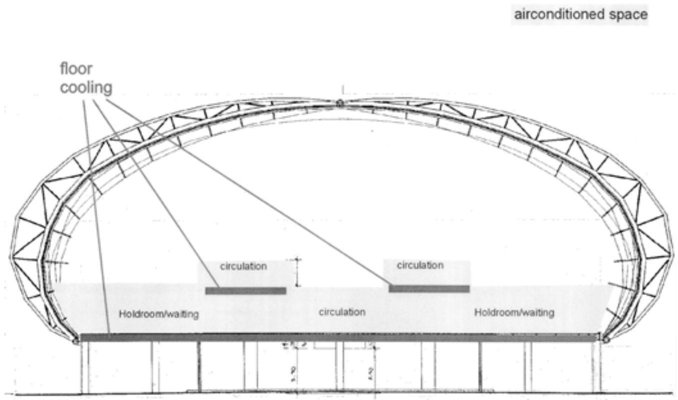

Figure 2 *New Bangkok International Airport, Case 1—37/119 segments.*

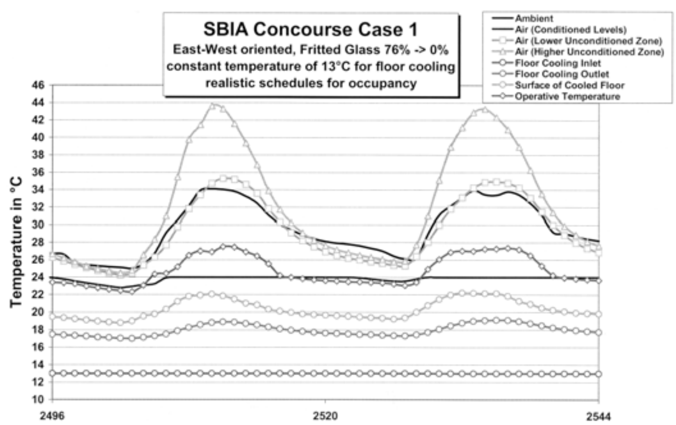

Figure 3 *SBIA concourse, case 1.*

Figure 2 shows the first typical concourse examined. The section shows the holdrooms and the arrivals corridor at the highest level. Level 1 has floor cooling over 71% of its area, and level 2 has floor cooling over 58% of the floor area.

At ambient temperatures of up to 34°C, the maximum internal temperature underneath the roof reaches 50°C, but when the floor cooling system is being supplied with a constant inlet water temperature of 13°C, the operative temperature in the occupied zone is 27°C and the air temperature is 24°C (see Figure 3). The peak cooling load for the segment being investigated is 165 kW, whereby the floor cooling removes 90 kW of heat from the space. The latent load for dehumidifying the outside air is 35 kW. There is 17 m^3/h fresh air per person, and the sensible load to cool outdoor air and recirculating air and maintain space conditions at 24°C is 150 kW (see Figure 4). The result of this simulation for an area of 1593 m^2 shows that the peak load per square meter is 103 W/m^2 because there is 1082 m^2 of floor cooling in this segment and the radiant cooled floor removes 83 W/m^2 of the load.

For the terminal building, the total cooling load has its maximum at 9500 kW and the floor cooling does not exceed 3000 kW because of the external shading of the trellis roof (Figure 5). The maximum operative temperature is only 26°C because the roof is shaded by the trellis roof blades and does not get as warm as the roof in the concourses (Figure 6). The average operative temperature is about 24°C throughout the day, which provides a very comfortable indoor environment. For this large area, the combination of floor cooling and variable-volume displacement ventilation consumes a considerably lower amount of energy and provides a higher degree of comfort.

Compared to the original concept (Figure 7), the peak cooling load was reduced by 35% in the optimized concept (Figure 8).For the entire airport, a cooling energy consumption of 191 GWh per year is estimated, which means 513 kWh

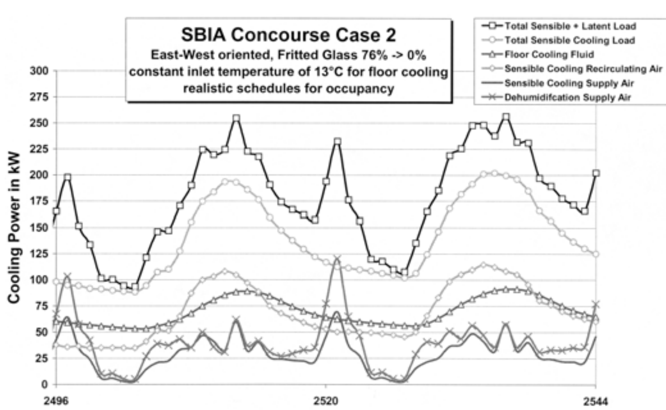

Figure 4 *SBIA concourse, case 2.*

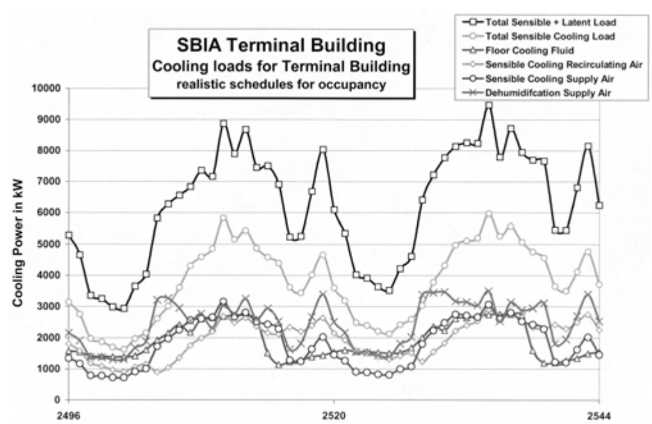

Figure 5 *SBIA terminal building cooling loads.*

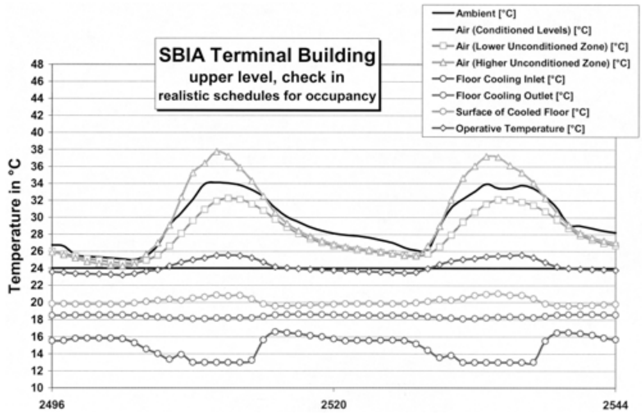

Figure 6 *SBIA terminal building upper level.*

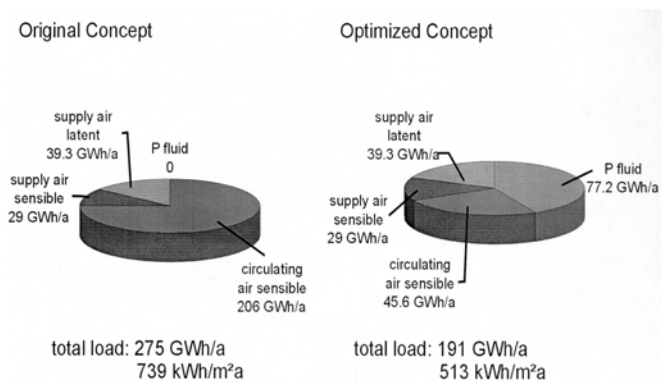

Figure 7 *SBIA comparison of cooling loads for the entire airport.*

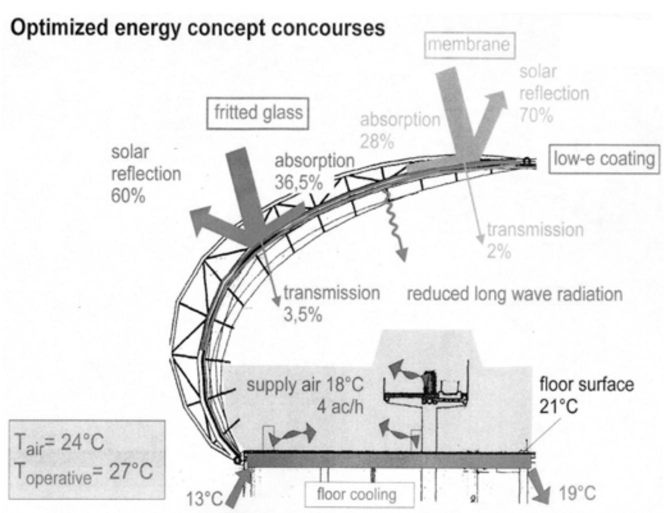

Figure 8 *Optimized energy concept concourses for SBIA.*

for each conditioned square meter of floor area per year. This is a reduction of 226 kWh/m^2 energy savings per year. Bearing in mind that the airport has a total floor area of 550,000 m^2, the total annual savings are considerable.

CONCLUSION

The Bangkok climate and the architectural design of the new airport would normally lead to enormous solar and heat gains requiring high cooling loads and producing low comfort conditions. Therefore, the operation of a variable-volume displacement conditioning system and a radiant cooled floor were developed and optimized to reduce the building load.

The introduction of a floor cooling system covers one part of the cooling load, which leads to a reduction of air changes for the circulating air cooling system and also leads to lower surface and operative temperatures. Since the floor cooling removes the solar gains absorbed on the floor, it allows the anticipated stratification of the hot air in the upper volume of the concourse. The thermal comfort is improved and the infrared energy transport from the roof to the bottom level is lowered by introducing low-e coatings on the inside surface of the hot concourse building envelope. An optimized concept for the concourses with an operative temperature of 27°C was developed. The placing of the air inlets and the air exhausts makes the stratification work for the standard concourse C (see Figure 8).

REFERENCES

Borresen, B. 1994. Floor cooling in atrium. Velta Congress, St. Christophe, Austria (in German).

Holmes, M., and A. Wilson. 1996. Assessment of the performance of ventilated floor thermal storage system. *ASHRAE Transactions* 102 (1).

Holst, S., and T. Welfouder. 1998. Calculation of peak cooling loads. Murphy/Jain Architects, 29 May.

Holst, S., T. Lechner, S. Reuss, and M. Selulen. 1998a. Concept and verification by material and function evaluation. Murphy/Jain Architects, 29 January.

Holst, S., T. Lechner, S. Reuss, and M. Selulen. 1998b. Climate and energy concept, concourses. Murphy/Jain Architects, 4 October.

Olesen, B.W. 1997. Possibilities and limitations of radiant floor cooling. *ASHRAE Transactions* 103 (1).

Simmonds, P. 1993. Thermal comfort and optimal energy use. *ASHRAE Transactions* 99 (1): 1037-1048.

Simmonds, P. 1994. Control strategies for combined heating and cooling radiant systems. *ASHRAE Transactions* 100 (1).

Simmonds, P. 1996a. Using CFD to analyze temperature stratification in a large airport building. ASME Fluid Conference, San Diego.

Simmonds, P. 1996a. Creating a micro-climate in a large airport building to reduce energy consumption. ASHRAE Conference on Buildings in Hot and Humid Climates. Ft. Worth.

Schlappman. 1996. Report on the Groninger Museum radiant floor system. University of Stuttgant (in German).

DISCUSSION

Philip Haves, Lawrence Berkeley National Laboratory, Berkeley, Calif.: When calculating the floor cooling from direct absorption of solar radiation in worst cases, where there is a very high occupant density, did you allow for shading of the floor by the occupants and their hand baggage, etc.?

Did you modify you comfort calculation procedure to allow for the radiant coupling between occupants when the occupant density is very high, e.g. in check-in lines? Is there a need for new calculations and design tools to deal with such situations?

Bernard Blazewicz, Principal Engineer, Merck and Co., Sommerset, NJ: At what floor temperature did condensation occur?