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 creating comfortable indoor environment in the building. The operation of a variable volume displacement conditioning system and a radiant cooled floor have been developed and 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.

Introduction

Radiant cooled floors have been successfully used to maintain comfort conditions in many buildings. The use of a radiant cooled floor as part of a hybrid system to condition a large airport has not as 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.

Background

In recent years, several studies have been made on the concept of radiant cooled floors. Borressen (1994), and Simmonds (1993), Simmonds (1994) reported on a radiant cooled floor design for a museum and, Schlappmann (1996) reported on the functioning of the radiant floor system two years after it started operating. Holmes, et. al., (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 (1990,1997), Simmonds (1993, 1994, 1996). Large areas have previously been conditioned using radiant systems, but a 200,000m^2 radiant cooled floor of a 500,000m^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 (1996) Described the design process for the displacement ventilation system proposed for this airport. Holst, et al (1997, 1998) described the advantages of using a radiant cooled floor together with the variable volume displacement system. This report also showed how the envelope design was optimized to reduce the heat gain to the space and reduce energy consumption.

Building Configuration

The building envelope for this building will to be constructed from clear glazing and Teflon materials. The roof of the main terminal and parts of the concourses are shaded or partially shaded by roof shading devices. However, a majority of the concourses have no external shading devices. Teflon does provide a sufficient barrier for direct solar, but the glazing offers very little solar resistance.

The thermal resistance of both Teflon and glass are relatively low and the conductive heat gain to the space would be enormous if a traditional indoor temperature of 24 °C have to be maintained. Using a variable air volume system traditionally employed at most large airports necessitates large volumes of air being supplied to the space maintaining the required temperature requires temperature. However due to the large volumes to the spaces creating a well mixed flow and no stratification. Air conditioning systems must supply volumes of conditioned air to a room in order to maintain conditions within specified limits. The environment requirements for a building can usually be met by different air conditioning systems. But it is easier to achieve the required results within given economical constraints with a variable system. The variable air volume system, varies the air volume being supplied to the space in accordance to the required conditions. By varying the air flow instead of constant air flow, the consumption of energy can be reduced. The air flow to the various zones are individually reset when the loads change. The maximum air flow is not normally required in all zones

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simultaneously. Although the air supply system is designed for maximum operation. These systems can consume considerable amounts of energy most of which is consumed conditioning unoccupied space.

During April, the warmest month, ambient temperatures vary from 34-36 °C. The temperature difference across the building envelope would be 10-12k. A variable volume displacement system was designed to maintain the required temperatures in the occupied zone (24C). Stratification would be enhanced by this system so that temperatures on the inside of the building envelope would be nearly equal to the ambient conditions virtually eliminating the convective heat gain to the spaces. The variable volume displacement system design and evaluation has been reported by Simmonds (1995,1996).

Building Load

The calculation of Peak loads for an building of this magnitude are obviously complex. In total 4 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 et al 1998 showed that the radiant cooled can absorb up to about 50 w/m^2 of short and long wave 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 80w/ m^2 capacity.

Design Concept

The Bangkok climate, showing 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 2 shows the material properties and basic assumptions. Table 4 shows the solar radiation, long wave radiation and air flows between zones. Table 4 shows the internal loads.

Design Procedure

The design approach was rather unique in that 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 teflon 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 re-direct the cool radiation effect from the floor back into the space. This can be identified as follows:

$A_1\,\text{å}_1\,\text{ó}\,T_1^4 = A_2\,\text{å}_2\,\text{ó}\,T_2^4$

Where A_1 = Area of Surface 1
A_2 = Area of Surface 2
å_1 = Emissivity of Surface 1
å_2 = Emissivity of Surface 2
ó = Stefan Boltemanns Constant
T_1 = Temperature of Surface 1
T_2 = Temperature of Surface 2

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 being developed. The loads for the concourse were 97W/m^2, 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.
The analysis results show that the floor can absorb up to about 50W/m^2 of short and long wave radiation, convection and conduction components provide the other 30W/m^2 of cooling from the floor. Because 50W/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. Preliminary analysis simulated a system using a 17°C return temperature. The mass flow and temperature differential were inadequate for the 80W/m^2 capacity. The maximum return water temperature was increased to 19°C and the internal pipe diameter was 20mm. 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^{0}C , the return water temperature can vary between 13^{0}C (no load) and 19^{0}C (full load). The flow through the floor will be constant. Table 1 shows the different radiant cooled floor capacities.

Table 1 COOLING POWER OF RADIANT FLOOR

Description	Peak Power of Floor Cooling Refer. to Cooled Area [W/m2]	Tube Spacing [mm]	Cooled Area to Total Floor Area [%]	Peak Power of Floor Cooling Refer. to Total Area [W/m2]
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

Ventilation

The outdoor air AHU will supply variable outdoor air quantities at a constant temperature to the zone AHU's. The volume of outdoor air will be varied depending upon the CO_2 levels metered in each zone, down to a fixed minimum position. The resulting zone conditions will be reported back to a central controller that can (if required) reset the outdoor air supply temperature to the zone AHU's, when the zones are lightly occupied. Each zone AHU can supply air at 16C to the space via the displacement diffusers (The air supply temperature to the space will be 18^{0}C). This supply temperature of 16^{0}C is crucial for the moisture control in the space which should not rise above 60%. The moisture content of space air at 26^{0}C and 60% is 11.2g/kg. The moisture content of the supply air at 16^{0}C is 10.00g/kg, which provides a buffer of 1.2g/kg. This buffer will absorb the moisture given off by the occupants (1200persons*65g/h.person=78kg/h) that will result in a moisture content of 10.78g/kg , which is below the maximum level of 11.2g/kg. These calculations are based on the maximum cooling load. The return air from the zone to the space should be returned as low as possible , this will reduce the cooling capacity of the zone AHU (the return air at say 30^0C is mixed with outdoor air at 10^0C in the mixing plenum of the zone AHU and this mixed air is conditioned to the required supply temperature of 16^0C). If the return air is extracted high in the space then the return air temperature will be in the region of 36^0C (due to the stratitification in the space enhanced by the displacement ventilation system) which will increase the cooling capacity of the zone AHU. When the load in the space drops either through no solar load such as nighttime or through no occupancy. Then a thermostat will proportionally reduce the volume of air to the space, the space temperature will still remain between 25-26^0C but it will require less energy to do so.

Table 2 Material Properties + Basic Assumptions

Transmission gains through envelope:

Element	Construction	U-Value
Floor	5 cm Insulation, Conductivity 0.04 W/mK	0.66 $W/^2mK$
Wall (Glass Concourses)	Laminated Glass with Low-e Coating Inside	4 W/m^2K
Wall (Glass Terminal)	Single Glazing	5 W/m^2K
Wall (Glass Offices)	Insulated Glass	1.5 W/m^2K
Roof (Glass Concourses)	Laminated Glass with Low-e Coating Inside	4 W/m^2K
Roof (Membrane Concourses)	Glass Fibre PTFE + Baffles + Inner Membrane	2.5 W/m^2K
Roof (Glass Terminal)	Laminated Glass with Low-e Coating Inside	4 W/m^2K
Roof (Glass Offices)	Roof + Insulated Glass	0.91 W/m^2K

Temperatures

Design air temperature	24°C
Ambient design temperature	36°C
Mean floor temperature (mix of cooled + non-cooled 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

Table 3 Solar Radiation, Long Wave Radiation and Air Flow Between Zones

Peak external solar radiation	1100 W/m2
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 exchangers per hour)

Table 4 Internal Loads for Typical Sections

Description	Persons		Light	Equipment
	[m^2/Pers]	[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
Busgate Vest.	10	9	10	0
Busgate	3	30	10	0

Internal Loads for Special Sections

Description	Persons		Light	Equipment
	[m^2/Pers]	[W/m^2]	[W/m^2]	[W/m^2]
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

Outside Air Demand

Open Areas	17 m^3/h Pers
Enclosed Areas	26 m^3/h Pers
Airline Offices	34 m^3/h Pers

Materials Development

The glazing system for the concourses consists of two 6mm glass layers. A clear float glass on the outside with a double ceramic fritting reduces the solar transmission. The fritting is white towards the outside to reach a high solar reflection and black towards 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 as hard-coating 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=1.85 and PPD=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 whole airport. The complex was divided into representative zones. Case 1 shows the first typical concourse examined (figure 2). The section shows the holdrooms and the arrivals corridor at the higher 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 temperature of 34°C the maximum internal temperature underneath the roof reaches 50°C,, but when the floor cooling system 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 165kW, whereby the floor cooling removes 90 kW of heat from the space. The latent load for dehumidifying the outside air is 35kW. There is 17m^3/h of fresh air per person and the sensible cooling load to cool outdoor air and recirculating air and maintain space conditions at 24°C is 150kW see Figure 4. The result of this simulation for an area of 1593m^2 shows that the peak load per m^{2} is 103w/m2, because there is 1082m2 of floor cooling in this segment the radiant cooled floor removes 83w/m2 of the load.

For the terminal building, the total cooling load has its maximum at 9500kW and the floor cooling does not exceed 3000kW 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 24C 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 considerable 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/a per year is estimated, which means 513kWh/m2.a for each conditioned square meter floor area per year. This is a reduction of 226kWh/m2.a energy savings. Bearing in mind that the airport has total floor area of 550,00m2, the tital annual savings are considerable.

Conclusion

The introduction of a floor cooling system covers one part of the cooling load which leads to a reduction or air changes for the circulating air cooling system. Besides the floor cooling system 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. All together, an optimized concept for the concourses with operative temperatures 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 10)

The Bangkok climate and the architectural design would normally lead to enormous solar and heats gains requiring high cooling loads and producing low comfort conditions.

Applying an optimization process tot he membrane and glazing reduces the solar transmission and therefore low solar induced cooling loads. The radiant floor cooling system removes a majority of the cooling load which reduces the air volumes normally associated with a traditional all air conditioning system. The radiant cooled floor reduces surface and operative temperatures and also directly absorbs a large portion of the solar load. The variable volume displacement ventilation system enhances stratification which reduces a majority of the convective heat gain to the space.

Thermal comfort conditions in the occupied zone are improve and the infra energy transport from the roof to the occupant is improved by the introduction of low coatings on the inside surfaces of the gazing.

The operative temperature in the occupied zone will be lower than 27°C throughout the building.

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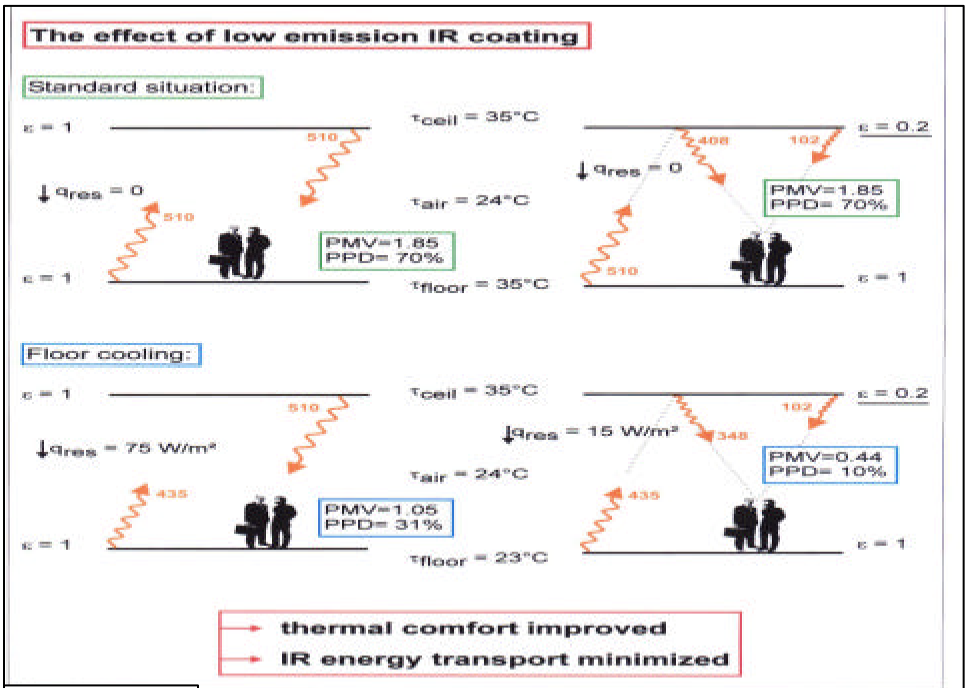


Figure 1

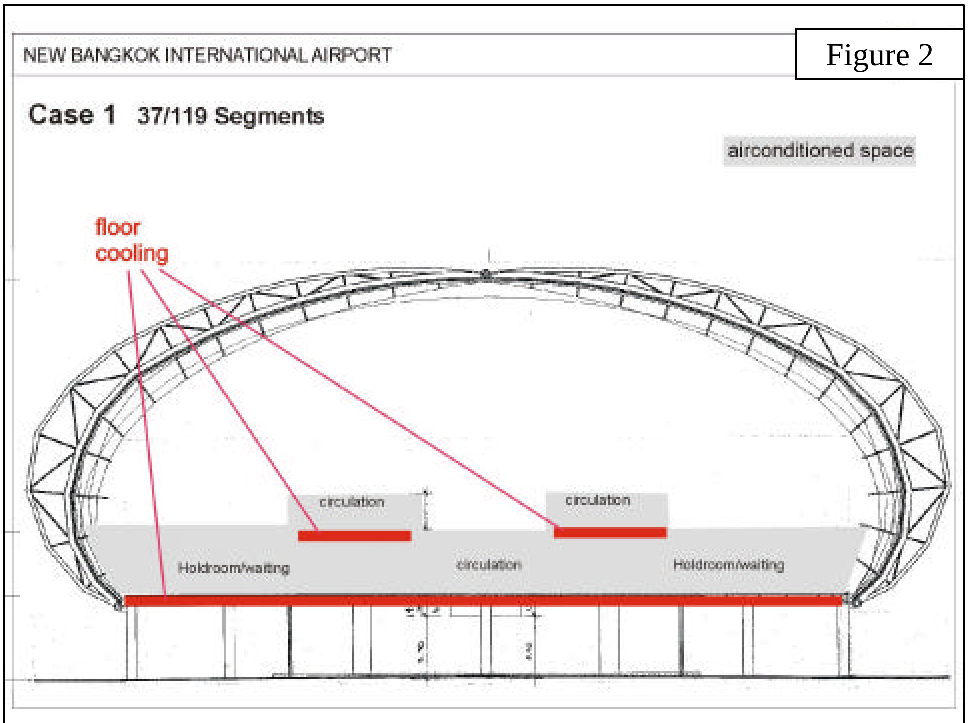


Figure 2

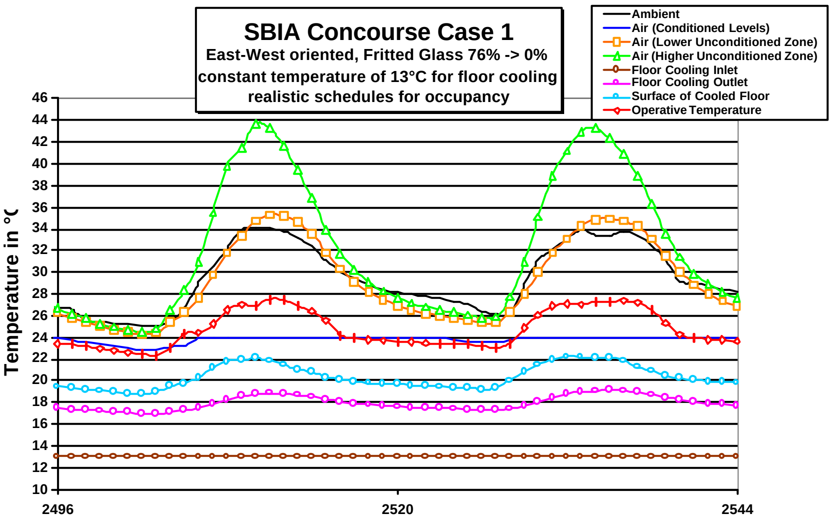


Figure 3

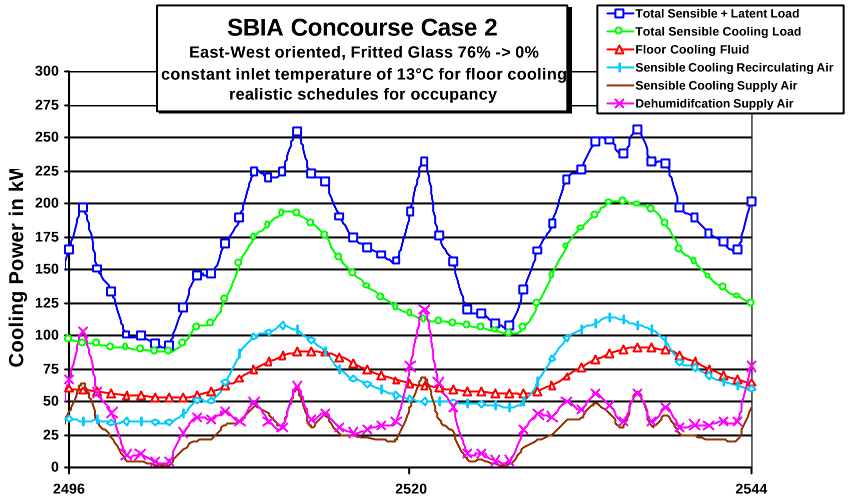
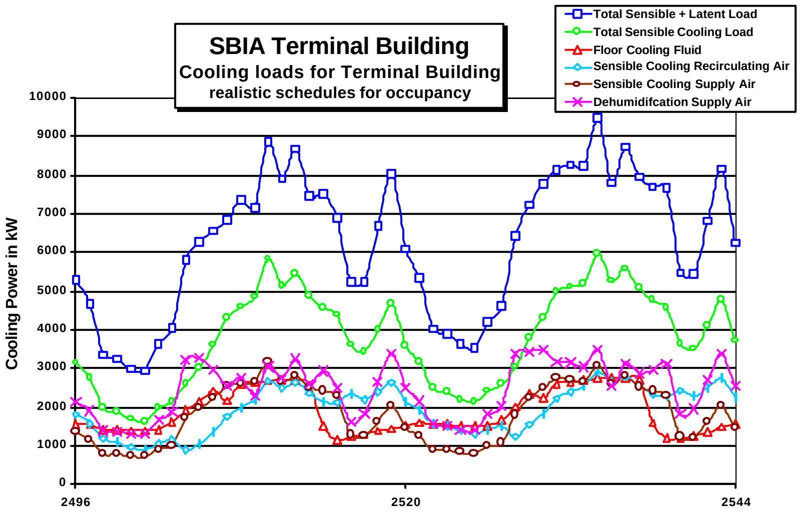


Figure 4

Figure 5



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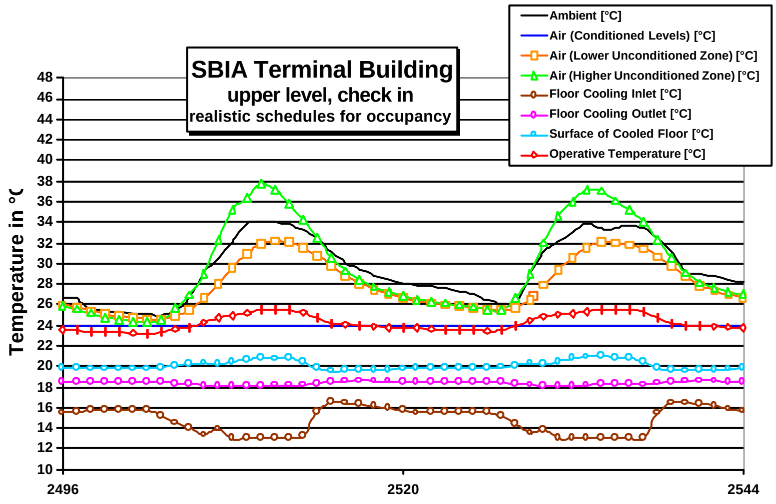

Figure 7

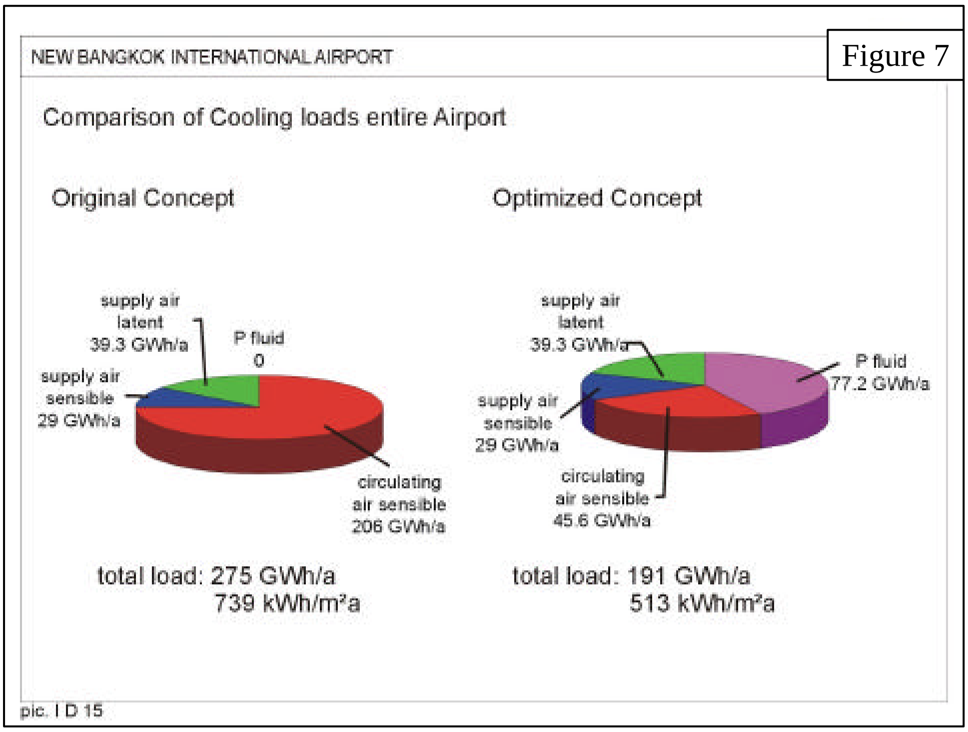

FIGURE 8