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DEVELOPMENT OF A DUCTLESS AIR SUPPLY SYSTEM USING LOW TEMPERATURE AIR

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ABSTRACT

This paper proposes a new ductless air supply system with a ceiling plenum chamber using low temperature air as a secondary HVAC system for an ice thermal storage system. The proposed air supply system mixes low temperature air with return air from a room using a mixing fan unit (MFU), pressurizes a plenum chamber with the mixed air and supplies the air to the occupied room from diffusers on the ceiling. The purpose of this study is to develop a new HVAC system to utilise low temperature air, to prevent cold draught and dew condensation, to keep thermal comfort in the room and to save fan energy consumption. Room temperature is fed back to controllers that regulate fan motors of an air handling unit and an MFU with inverters. Low temperature air from an air-handling unit is controlled at 10°C by regulating flow rate of chilled glycol. A 7m and 10m and 3m height room is provided to evaluate the proposed system. 24-hour time series data are obtained by data loggers. Vertical temperature distribution is within setpoint $\pm$ 1K and horizontal temperature is controlled between 25.7 and 26.2°C at 1.5m height above floor. Time-averaged PMV at the centre of the room is +0.24 and PPD is 6.2%. 24-hour time-averaged temperature is 26.2°C at the sensor for control and the root mean square error is 0.18K. Temperature difference between mixed and room air is averaged to 9K. It prevents cold draught due to low temperature air supplied directly to the room from ceiling-mounted air diffusers. Air velocity at the centre of the room is averaged to 0.22m/s and turbulence intensity is 8.6%. The results meet ISO7730-1994. This system is also expected to reduce the fan energy consumption by 70%. In addition, use of night electric power for ice storage systems saves energy cost of a chiller. The ductless air supply system using low temperature air is appreciated satisfactory for thermal comfort and energy saving.

KEYWORDS

Ductless, Ice Storage Systems, Air Supply, Large Temperature Difference, Energy Saving, Variable-Air-Volume (VAV), Ceiling Plenum Chamber, Low Temperature Air, PMV, PPD

INTRODUCTION

Ice storage systems are widely applied to shift on-peak cooling load to off-peak nighttime. Low temperature water and air is available in ice storage systems. In a decade, energy saving strategy has been studied to use large temperature difference water and air by using low temperature water and air. It should be noted that problems in an application of low temperature air supply are dew condensation and thermal discomfort. To solve the problems, the authors proposed a new air supply system using a ceiling plenum chamber and mixing fan units (MFUs). MFUs mix low temperature supply air from an air-handling unit with return air from the occupied room, and pressurise the ceiling plenum chamber with the mixed air. The mixed air is supplied to a room from diffusers mounted on the ceiling. The mixed air temperature is almost between 16 and 18 °C. The final temperature differences between supply air and room air are from 8 to 10K. The air supply system uses the insulated plenum chamber instead of branch ductworks in order to lead air from MFUs to the ceiling-mounted diffusers. The proposed system reduces first cost and ductwork spaces above ceiling.
It may also be helpful to renewal works.

To control room temperature, supply air volume varies according to the thermal load. This control strategy is expected to save fan power energy consumption using variable-air-volume (VAV) air-handling unit and MFUs.

This paper will show that the proposed air supply system satisfies thermal comfort conditions in a occupied room according to ISO standard 7730 and helps fan energy saving.

EXPERIMENTAL SETUP AND TEST CONDITIONS

Tests are conducted in a full-scale room simulator. The test room has dimensions of 7m by 10m by 3m high and is furnished with simulated perimeter load heaters. 6 slot diffusers and 6 circular diffusers are mounted on the ceiling. Plenum chamber upper the ceiling is fully insulated with fibreglass boards. See Figure 1 for the room tested.

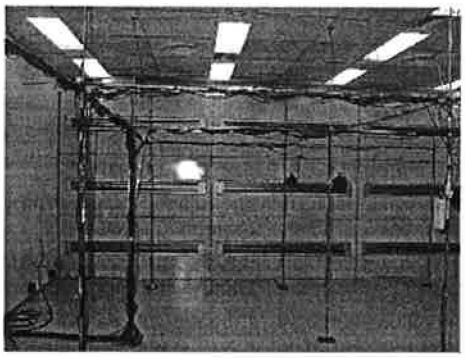


Figure 1: The room tested for the experiments

To give equivalent cooling load through the envelope wall and windows, electric heaters are installed on the wall. The heating capacity of the heaters can be adjusted continuously (maximum 5kW).

Air temperature and velocity are measured using T-type thermocouples and thermal anemometers. Mean radiant temperature is measured by T-type thermocouples inserted in globe thermometers. Power consumption of a brine chiller, air-handling unit fan, electric heaters on the wall and lighting is calculated by pulse counters. Data are scanned by data loggers and transported to a PC through RS-232C serial port every minute.

PID controllers vary fan speed of the air-handling unit and MFUs using pulse-width-modulated (PWM) inverters. They achieve variable-air-volume control for the room air temperature. Maximum total air volume of the slot and circular diffusers mounted on the ceiling is $2630m^3/h$. It is 12.5 air changes per hour. To maintain the requirement for fresh air, the minimum air volume is set as 30Hz of inverter frequency for the air-handling unit and MFUs.

A brine chiller produces 2°C glycol as simulated low temperature water from an ice storage system. Glycol is supplied to the cooling coil of the air-handling unit. Heating operation is also available to use an electric heater installed in a brine tank. To supply low temperature air of 10°C from the air-handling unit, flow rate of glycol through the cooling coil is controlled by two-way valves. A variable-air-volume (VAV) air-handling unit and MFUs control room temperature with inverters as shown in Figure 2.

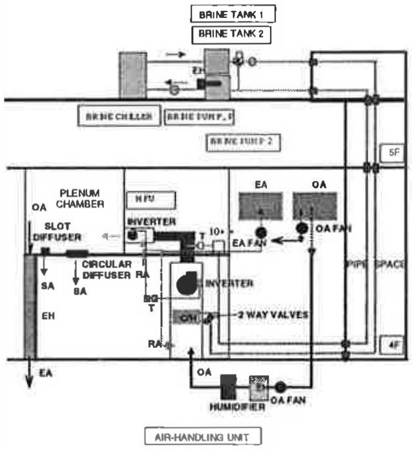


Figure 2: Ductless Air Supply System

Air temperature is measured vertically at the surface of the floor, 0.1m, 0.5m, 1.0m, 1.5m, 2.0m and 2.5m above floor, at the lower and upper surface of the ceiling and at 0.5m above the ceiling (3.5m above the floor) horizontally in 12 points of the room tested. Figure 3(a) shows the horizontal location of measurement points. Radiant temperature is measured at points of No.2, 5, 8 and 11 at 1.5m above floor.

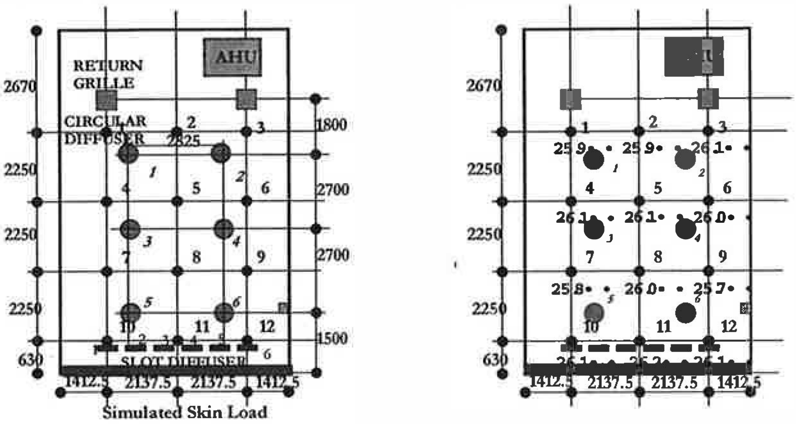
EXPERIMENTAL RESULTS

The main goal of the experiments is to demonstrate the thermal comfort and energy saving which this new air supply system can achieve. The following results are obtained by experimental data of cooling mode. The heaters give 2.5kW cooling load as the perimeter load. Cooling load of overhead lighting is totally 1.6kW. Setpoints of 25 and 26• are given to the room air temperature. For all cases, tests are conducted over 24-hour periods continuously.

Horizontal Air Temperature Profiles

Horizontal temperature profile in the room tested is described in Figure 3(b). Setpoint of the room temperature is 26°C. Horizontal room temperature difference at 1.5m above floor is kept within 0.5K.

in a steady-state. Space-averaged room temperature of 12 points at 1.5m above floor is 26.0^{0}C. Such a tendency is also observed at a different setpoint. It may be evaluated favourable for thermal comfort.



(a) Measurement Points and Diffusers Location

(b) Horizontal Temperature Profile (1.5m above Floor)

Figure 3: Horizontal Temperature Profiles

Vertical Air Temperature Profiles

Vertical temperature profiles in the room tested are presented in Figure 4. Temperature difference at 0.1m and 1.1m above floor is less than 2K at each setpoint temperature. According to ISO 7730, the vertical air temperature difference between 0.1m and 1.1m above floor shall be less than 3K. The experimental results for 12 measuring points meet the recommendation of ISO7730.

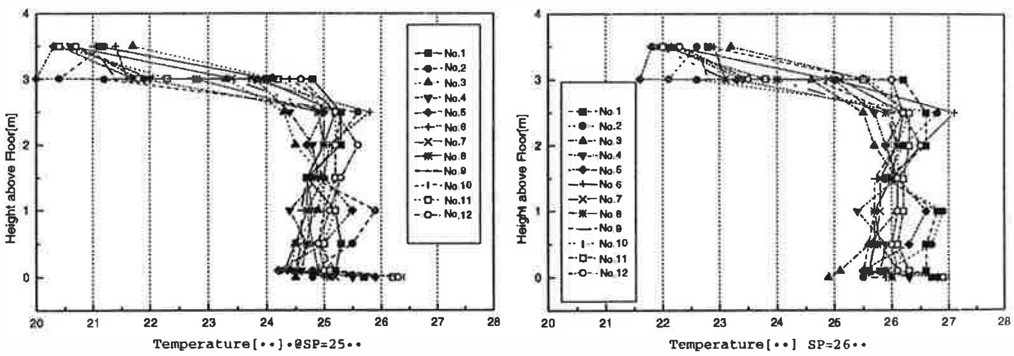


Figure 4: Vertical Temperature Profiles

Stability of the Room Temperature

Time series data of air temperature and humidity over 24-hour are presented together in Figure 5. Fluctuations of the room air temperature are within 0.5• from the setpoint of the room air temperature at the centre of the room tested. The standard deviation of the room air temperature is 0.18K. The room temperature is controlled considerably at a constant level. Thus, controllability of the room temperature is evaluated to be excellent.

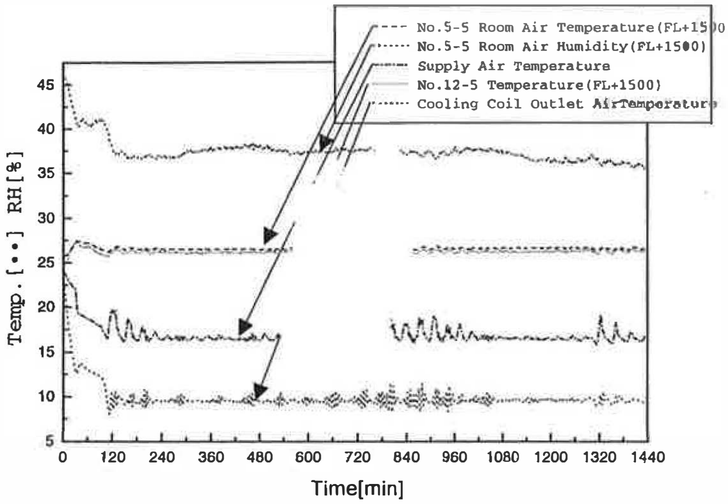


Figure 5: Time Series Data of Temperature and Humidity

Air Velocity

Mean air velocity in the centre of the room tested is 0.22m/s and the standard deviation is 0.018m/s over 24-hour period. It may be estimated rather constant in spite of fluctuations of the supply air velocity caused by the fan of air-handling unit. Turbulence intensity (Tu) is defined as a percentage of the standard deviation divided by mean air velocity. In this case Tu is 8.6%. Allowable air velocity at Tu of 10% is 0.32m/s at 26• according to ISO 7730. It meets the recommendation of ISO 7730.

Energy Consumption

The energy consumption considered here is based on the operation of the air-handling unit fan and MFUs. A VAV air-handling unit and MFUs are expected to reduce fan power. Fan power is assumed to be proportional to squares of fan frequency by using inverters. Fan power consumption for the air-handling unit is 5.8kWh over 24-hour period. For the MFUs, power consumption is not measured here, but it is estimated approximately the same value as the air-handling unit. The nominal fan power output of the air-handling unit is 1.5kW and that of the MFUs is 1.5kW. The conventional constant volume air-handling unit might consume 1.5kW over 24 hours and it might lead to 36kWh. The energy consumption of 5.8kWh plus 5.8kWh is nearly 30% of that, compared to the conventional air supply system.

Time-averaged cooling load is calculated by measuring the flow rate and temperature difference of chilled water. Mean cooling load over 24-hour period is estimated 50% of the cooling capacity of the air-handling unit. It is assumed to be equal to a seasonal cooling load factor of a commercial building. Thus, the test condition is evaluated to be equivalent to a real building.

DISCUSSIONS

Thermal Comfort

Thermal comfort in the room tested is assessed according to ISO7730 and ANSI/ASHRAE55. PMV is calculated according to ISO7730. Room air temperature, relative humidity, mean radiant temperature and air velocity are measured. Activity level and clothing of occupants are given as 1.2Met and 0.6clo in summer conditions.

The room air temperature, relative humidity, mean radiant temperature and air velocity averaged over 1-hour period in a steady-state is 26.1•, 35.6%,26.0• and 0.21m/s relatively in the centre of the room tested, when the setpoint is 26•. PMV is +0.24 and PPD is 6.2%, while ISO7730 recommends −0.5<PMV<+0.5 and PPD<10%.

Economical Evaluation

A variable-air-volume air supply system can reduce the fan energy consumption using the proposed ductless system. In addition, the proposed system can work as a secondary HVAC system for an ice storage cooling system, which employs night electric power at lower price. In Japan, night power for thermal storage systems is charged at about 1/5 compared to daytime.

CONCLUSIONS

A new ductless air supply system using large temperature difference air is proposed to optimise secondary HVAC systems for ice storage systems. Two goals has been accomplished during this study:

1. Thermal comfort in the room tested is appreciated satisfactory according to ISO 7730-1994 in summer conditions.
2. Fan power consumption can be reduced by using a VAV air-handling unit and MFUs by 70%. The proposed system can save energy cost for fan power.

The proposed air supply system may be expanded to a wider range of buildings.

References

ISO Standard 7730 (1994). Moderate thermal environments - Determination of the PMV and PPD indices and specification of the conditions for thermal comfort.

ANSI/ASHRAE 55 (1992). Thermal Environmental Conditions for Human Occupancy, ASHRAE, USA.

Knebel D. E. and John D. A. (1993). Cold Air Distribution, Application, and Field Evaluation of a Nozzle-Type Diffuser. *ASHRAE Transactions* **99:Pt.1,** 1337-1348.

Elleson J. S. (1993). Energy Use of Fan-Powered Mixing Boxes with Cold Air Distribution. *ASHRAE Transactions* **99:Pt.1,** 1349-1358.

Chow W.K., Wong L.T., Chan K.T. and Yiu J.M.K. (1994). Experimental Studies on the Airflow Characteristics of Air-conditioned Spaces. *ASHRAE Transactions* **100:Pt.1,** 256-263.