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A Study of Thermal Comfort Control Using Least Enthalpy Estimator on HVAC system

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Abstract—**Thermal comfort is the condition of mind that expresses satisfaction with complex thermal factors. Mostly, human thermal comfort depends on four environmental parameters and two personal parameters. This paper proposes an example, especially in typical summer clothing during sedentary and moderate activity at mean air velocity, that the slope of the effective temperature line as thermal comfort level, which is composed of air temperature and relative humidity, is defined.**

Not only were we interested in thermal comfort of room, but also energy saving. This paper also proposes a least enthalpy estimator (LEE) that combines the definition of thermal comfort level and the theory of enthalpy into a load predicting way to search suitable a temperature and relative humidity setting. Due to the change of humidity in the space, the temperature will be revised reversely toward a new setting. It is helpful to keep on the thermal comfort level and to determine the proper settings opportunely for HVAC control system with LEE.

I. INTRODUCTION

There are many thermal comfort models established. However, thermal comfort model mainly depends upon the combination of the complex interaction of dry-bulb temperature, mean radiant temperature, air velocity, relative humidity, clothing insulation, and activity level[1]. Hamdi and Lachiver[2] indicated that there are three main reasons such that thermal comfort calculation is complex, and that the thermal comfort sensation of human is subjective, and that the sensation of thermal comfort is with many variables, which are hard to measure with precision. Meanwhile, since the activity level and clothing strongly depend upon personal preference and only air temperature and humidity could be controlled in conventional HVAC systems, it is hard to implement the mathematical thermal comfort models in actual environment.

According to the study results by Chamra et al. [3], if the activity level, clothing insulation, and air velocity were constant and the mean radiant temperature is assumed as air temperature, the temperature and relative humidity have effect on the thermal comfort sensation and the temperature instead of the relative humidity is the dominant factor. The lower the relative humidity, the higher the dry bulb temperature could be for the same thermal comfort. Thus, a lower relative humidity can compensate for a higher dry-bulb temperature without affecting thermal comfort sensation.

Henderson et al. [4] indicated that comfort control tends to decrease the consumed energy of HVAC systems that maintain low humidity and penalize systems that maintain high humidity. Generally, the effective temperature indicates the rate at which dry-bulb temperature can be increased with decreasing relative humidity while maintaining the same thermal comfort. The rational reason for energy savings is that by lowering relative humidity in the space, a higher dry-bulb temperature is allowable while maintaining constant comfort. As a result of the higher temperature set point, the consumed energy of the system could be reduced.

The use of effective temperature is suitable for thermal comfort control and energy saving. However, thermal comfort condition in the space is time varying. The setting of dry-bulb temperature and relative humidity should be properly arranged for constantly changing space conditions. Based on the thermal comfort requirements, the first thing for saving energy of HVAC systems is to analyze the space thermal load. The load change of room can be treated as the alteration of air enthalpy in the space. Therefore, the theory of enthalpy could be suited to predict future air conditions of space. In this paper, a least enthalpy estimator (LEE) is proposed which utilizes the concept of thermal comfort and the theory of enthalpy to provide a suitable setting on the effective temperature line for use by HVAC control system to achieve thermal comfort and energy saving purposes.

The paper is organized as follows. In Section 2, interpreting a case of thermal comfort control criterion. In Section 3, introducing the Least Enthalpy Estimator (LEE). In Section 4, the solution of the least enthalpy difference is decided by LEE. The decisions of LEE are presented as a simulation result.

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II. THERMAL COMFORT CONTROL CRITERION

A. Factors Influencing Thermal Comfort

Most models of heat exchanges between the body and the environment refer to classic heat transfer theory. Both steady-state energy balance and two-node transient energy balance are two widespread models and proposed in ASHRAE Fundamentals [1]. These models assume that the body is uniform while heat production of the body transfer to the environment. The steady-state model developed by Fanger (1970) assumes that the body is in a state of thermal equilibrium with negligible heat storage. At steady state, the rate of heat production equals the rate of heat dissipation. A two-node model developed by Gagge *et al*. (1971) represents the body as the body core and the skin layer. A transient energy balance declares that the rate of heat storage equals the net rate of heat gain minus the heat loss.

ASHRAE 55-1992[5] defined thermal comfort as the condition of mind that occupants satisfy with the thermal environment. Since heat production of the body is proportioned to human activity level directly, heat exchange between the body and the environment should be operated to avoid discomfort under a small body temperature difference. There are many discussed factors influencing people thermal comfort and response in thermal space. According to both models describing above, the majority of influencing thermal comfort condition will depend upon four environmental factors and two personal factors. The environmental factors are air temperature, mean radiant temperature, humidity, and air velocity. The personal factors that influence thermal comfort are activity level and clothing insulation.

To realize a thermal comfort environment, it is helpful to understand that the influencing combination of thermal comfort. Activity level and clothing insulation have obvious personal effects on thermal comfort. It is difficult to detail these personal factors in thermal space. However, in some typical indoor applications, people wear similar clothing insulation for the similar activity in the same climate. Both activity level and clothing insulation could be assumed as constant. In this paper, we also assume that the mean radiant temperature could be approximated as the ambient air temperature. Air velocity has its effect on thermal comfort as well. This factor is always under control on HVAC system. It is able to treat the air temperature and the humidity with HVAC apparatus.

A standard set of conditions representative of typical indoor applications is used to define as a thermal comfort zone for summer in ASHRAE 55-1992, which is intended to provide acceptable conditions for occupants wearing typical indoor clothing at near sedentary activity and it requires that the air velocity is no greater than 0.25 m/s. Only both air temperature and relative humidity affect variables of thermal comfort control in this case.

B. Thermal Comfort Zone in ASHRAE 55

The comfort zones on psychrometric chart for summer and winter are shown in ASHRAE 55-1992 for sedentary and moderate activity, and wearing typical indoor clothing in the same climate. Since lines of constant comfort in ASHRAE 55-1992 should be identical, the effective temperature is applied to this case.

The effective temperature (ET*) integrates temperature and humidity into a single widespread environmental index. Houghton and Yaglou (1923) developed the original effective temperature in ASHVE. Gagge et al. (1971) introduced a revised effective temperature, which is the same sensible plus latent heat loss, that is the dry-bulb temperature at 50% rh.

C. Optimum Thermal Comfort Line

The purpose of ASHRAE 55-1992 is to specify the combinations of indoor space environment and personal factors that will produce thermal environmental conditions acceptable to 80% or more of the occupants within a space. Table I gives the optimum operative temperature for people in typical summer clothing during sedentary and moderate activity, which indicates that the metabolic rate is less than 1.2 met, at 50% rh and mean air velocity $\leq$0.15 m/s. The operative temperature is similar to the ambient air temperature, while the mean radiant temperature is approximated as the ambient air temperature. During the summer season, typical clothing in commercial establishments consists of light slacks and short sleeves shirt, which the clothing insulation values is about 0.5 clo.

TABLE I
OPTIMUM OPERATIVE TEMPERATURE FOR PEOPLE DURING LIGHT, PRIMARILY SEDENTARY ACTIVITY ($\leq$1.2MET) AT 50% RH AND MEAN AIR SPEED $\leq$0.15M/S (SOURCE: ANSI/ASHRAE STANDARD 55-1992)

Season	Description of Typical Clothing	I_{cl}	Optimum Operative temperature
Summer	Light Slacks and Short-sleeve Shirt	0.5 clo	24.5 ^{o}C

The constant effective temperature could express the same thermal comfort level of people. For the definite control value, we adopt the optimum temperature as a typical constant effective temperature line (24.5ET*, 30%~60% rh) on thermal comfort zone in ASHRAE 55-1992. This paper proposed that a thermal comfort control case aimed at the optimum thermal comfort line on the psychrometric chart is shown as Fig. 1. The dry-bulb temperature of this line varies from 24.225^{o}C to 25.085 ^{o}C. Correspondingly, the relative humidity of this line varies from 60% rh to 30% rh.

III. LEAST ENTHALPY ESTIMATOR (LEE)

In regard to the office air-conditioning conditions, typical indoor clothing and activity, in summer we adopt the effective temperature as a constant thermal comfort level in ASHRAE. According to this thermal comfort definition, it is suitable to manipulate both temperature and relative humidity concurrently. Not only were we concerned with thermal comfort of people, but also energy saving. Based on the optimum thermal comfort control line, the rational hypothesis to perform energy saving is that raising temperature by lowering space relative humidity. It is available to save energy and to keep thermal comfort in the space.

Enthalpy is a measure of heat content of substances. The load change of room thermal condition being treated is the alteration of enthalpy of air. It is suitable for finding the amount of heat necessary for certain processes. The theory of enthalpy could be good for predicting the future condition of air in thermal space [6].

The state point of the air can be plot on the psychrometric chart with any two properties. The enthalpy of a mixture of perfect gases equals the sum of the individual partial enthalpies of the components. So the enthalpy of moist air could be written the sum of the individual partial enthalpies of two components as dry air and saturated water vapor at the state, expressed as (1).

$$\begin{aligned} h &= h_{dryair} + W h_g \\ &= 1.006\, t + W\,(2501 + 1.805\, t) \\ &= h_{sensible} + h_{latent} \end{aligned} \tag{1}$$

where t is the dry-bulb temperature, ^{o}C;
W is the humidity ratio, kg/kg;
h_{dryair} is the specific enthalpy for dry air;
h_g is the enthalpy for saturated water vapor at temperature of the mixture.

A. Portion of Sensible Heat and Latent Heat

It is necessary actually to utilize energy for the treatment of moist air in HVAC system. Basically, the enthalpy difference of moist air could symbolize the treatment of moist air from state A to state B as the consumed energy of HVAC system. Here is a cooling and humidifying example in HVAC system. State A on psychrometric chart stands for the initial air condition of the space. State B is the final condition of the air leaving HVAC system. Each enthalpy of state A and B was given in Table II.

The enthalpy difference of the moist air from state A to state B was shown as portion of sensible heat and latent heat below.

Δh = 49.08 - 49.81 = - 0.73 kJ/kg
Δh_L = 24.01 - 22.22 = 1.79 kJ/kg
Δh_S = 25.07 - 27.59 = - 2.52 kJ/kg

TABLE II
STATES OF COOLING AND HUMIDIFYING PROCESS

Symbol	State A: 27 ^{o}C, 40% rh of air	State B: 24.5 ^{o}C, 50% rh of air
Enthalpy (h)	49.81 kJ/kg	49.08 kJ/kg
Latent heat (h_L)	22.22 kJ/kg	24.01 kJ/kg
Sensible heat (h_S)	27.59 kJ/kg	25.07 kJ/kg

However, energy is consumed by HVAC apparatus, which treat air condition for thermal comfort of space. In HVAC system, all of four processes such as cooling, heating, humidifying, and dehumidification need to consume energy. The consumed energy, represented by the latent heat change and the sensible heat change, could be shown:

Removing sensible heat (cooling): 27.59 - 25.07 = 2.52 kJ/kg
Adding latent heat (humidifying): 24.01 - 22.22 = 1.79 kJ/kg
Total consumed energy : 2.52+1.79 = 4.31 kJ/kg

The portion of sensible heat could imply the consumed energy that the apparatus of cooling or heating performs. The portion of latent heat also implies the consumed energy that the apparatus of humidifying or dehumidification performs. The enthalpy difference between two states of moist air in the space can be distinguished between sensible heat change and latent heat change.

B. Least Enthalpy Difference

The optimum comfort line (24.5ET*), which is a combination of temperature and relative humidity, is an example of thermal comfort control setting in this paper. We utilize the enthalpy relation between measured space condition (***t_R***, ***W_R***) and the optimum comfort line to get a thermal setting pair, including temperature and humidity. (1) can solve the enthalpy of air in thermal space with dry-bulb temperature and humidity; meanwhile, the operations of the four processes all need to consume energy in the HVAC system. The enthalpy difference could be written as (2), revised from (1). The first part is the portion of sensible heat and the second part is the portion of latent heat in this equation.

$$\begin{aligned} E(j) &= \left|1.006\times[t_2(j)-t_R]+1.805\times[t_2(j)W_2(j)-t_R W_R]\right| \\ &\quad + \left|2501\times[W_2(j)-W_R]\right| \end{aligned} \tag{2}$$

where ***$E(j)$*** is the enthalpy difference from space condition to point ***j*** on the optimum comfort line ;
t_R and ***W_R*** are temperature and humidity ratio of the space thermal condition in the space;
$t_2(j)$ and ***$W_2(j)$*** are temperature and humidity ratio of point ***j*** on the optimum comfort line;

For saving energy and thermal comfort control, this paper designs a Least Enthalpy Estimator (LEE). The enthalpy difference between the space state to the optimum comfort line could be calculated by (2). The point on the optimum comfort line with the least enthalpy difference will be picked up as a setting pair, including temperature and humidity by LEE. According to this setting pair, the HVAC control system can afford a suitable treatment of air to deal with the future space air condition, which takes account on both thermal comfort and energy saving.

IV. SOLUTION OF LEAST ENTHALPY ESTIMATOR

Since the optimum thermal comfort line on the psychrometric chart is defined as Fig. 1. The dry-bulb temperature (***t_2***) of this line varies from ***t_2^0*** to ***t_2^N***. The relative humidity of this line also varies from 60% rh that the corresponding humidity ratio is W_2^0, to 30% rh that the corresponding humidity ratio is W_2^N. The humidity ratio (W_2) of this line is in the interval [0.01134, 0.00588]. Here we assumed that the thermal condition at point j=k, which is (W_2^k, t_2^k) as below.

$$\forall W_2 \in [0.01134,\ 0.00588];$$
$$let\ W_2^{\,0} = 0.01134;\quad W_2^{\,N} = W_2^{\,0} - N\Delta W_2 = 0.00588;$$
$$W_2^{\,k} = W_2^{\,0} - k\Delta W_2;\ k = 0, 1, \cdots, N\ ;\ \Delta W_2 > 0.$$
$$\forall t_2 \in [24.225,\ 25.085];$$
$$let\ t_2^{\,0} = 24.225;\quad t_2^{\,N} = t_2^{\,0} + N\Delta t_2 = 25.085;$$
$$t_2^{\,k} = t_2^{\,0} + k\Delta t_2;\ \ k = 0, 1, \cdots, N\ ;\ \Delta t_2 > 0.$$

There are three zones, which depend on the humidity ratio of the space thermal state (W_R) on the psychrometric chart, shown as Fig. 1. The first zone is expressed as the region while $W_R > W_2^0$. The second zone is the region, which W_R is in the interval [W_2^0, W_2^N]. The humidity ratio of the third zone is less than W_2^N. The enthalpy difference of LEE will be dominated by the function of the humidity ratio, expressed as (3).

$$E(k) = \left|1.006\,(t_2^{\,k} - t_R) + 1.805\,(W_2^{\,k} \times t_2^{\,k} - W_R t_R)\right| + \left|2501\,(W_2^{\,k} - W_R)\right|\ \ ; k = 0,1,\cdots, N. \tag{3}$$

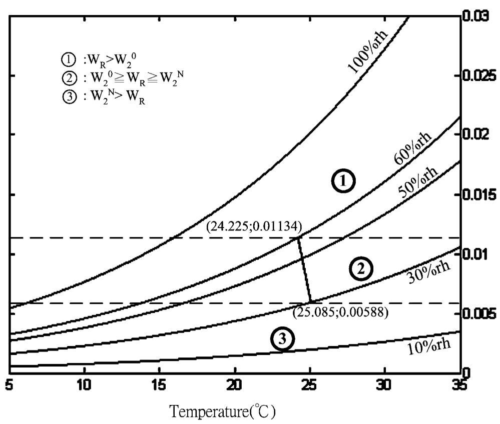

Fig. 1. Three decision zones of LEE

A. *Least Enthalpy Difference at* W_2^0

The region of $W_R > W_2^0$ in Fig. 1 represents that the air needs to be treated toward the optimum thermal comfort line. The enthalpy difference between space state and the optimum thermal comfort line is calculated by (2). According to (2), function of the enthalpy difference is an increasing function. The least enthalpy difference between space state and the optimum thermal comfort line is located at point j=0, which is (t_2^0, W_2^0). The proof of the least enthalpy difference expresses as below.

$$Since\ \ W_R > W_2^{\,0} > W_2^{\,k}\ \ and\ \ |X + Y| \le |X| + |Y|;$$
$$\lim_{\substack{\Delta t_2 \to 0 \\ \Delta W_2 \to 0}} (E(0) - E(1))$$
$$\le \lim_{\Delta W_2 \to 0} \{\left|1.006\,(t_2^{\,0} - t_R) + 1.805[(W_2^{\,0} - \Delta W_2)t_2^{\,0} - W_R t_R]\right| + 1.805\,t_2^{\,0}\Delta W_2 - 2501\,(W_2^{\,0} - W_R) - \left|1.006\,(t_2^{\,0} - t_R) + 1.805[(W_2^{\,0} - \Delta W_2)t_2^{\,0} - W_R t_R]\right| + 2501\,(W_2^{\,0} - W_R) - 2501\,\Delta W_2\}$$
$$= \lim_{\Delta W_2 \to 0} (1.805\,t_2^{\,0} - 2501)\Delta W_2 = 0^- < 0 \quad (\because 2501 >> 1.805\,t_2^{\,0})$$
$$\therefore E(0) < E(1)$$

and

$$\lim_{\substack{\Delta t_2 \to 0 \\ \Delta W_2 \to 0}} (E(k) - E(k+1))$$
$$\le \lim_{\Delta W_2 \to 0} (1.805\,t_2^{\,0} - 2501)\Delta W_2 = 0^- < 0 \quad (\because 2501 >> 1.805\,t_2^{\,0})$$
$$\therefore E(k) < E(k+1)$$

That is, $E(k)$, k=0, …,N is an increasing function while $W_R > W_2^0$, shown as (4).

$$\therefore E(0) < E(1) < \cdots < E(k) < E(k+1) < \cdots < E(N) \tag{4}$$

The least enthalpy difference is located at point j=0, which is (24.225 °C, 0.01134 kg/kg) on the psychrometric chart. There are two cases about the space states calculated with LEE. The first state (R1) and the second state (R2), which are (26 °C, 60% rh) and (30 °C, 50% rh) respectively, approached the optimum thermal comfort line. The values of (2) are shown as Fig. 2. The setting pair will be selected at (24.225 °C, 60% rh).

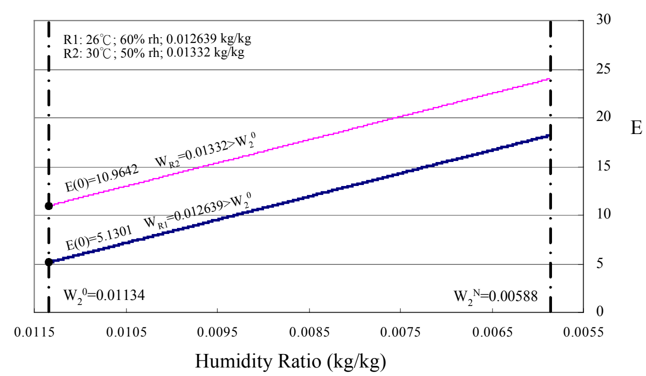

Fig. 2. Enthalpy difference of cases while $W_R > W_2^0$

B. *Least Enthalpy Difference at* W_2^n

In zone 2, shown as in Fig. 1, we assume that W_R is the same as at W_2^n, which is located in the interval [0.01134, 0.00588]. According to (2), function of the enthalpy difference is a decreasing function while $W_2^k \ge W_2^n$; meanwhile, it is an increasing function while $W_2^k \le W_2^n$. That is, the least enthalpy difference between space state and the optimum thermal comfort line will be located at point j=n while $W_R = W_2^n$.

The proof of the least enthalpy difference expresses as below.

$$Since\ W_2^k \le W_R\ and\ -|X+Y| \ge -|X|-|Y|;$$

$$\lim_{\substack{\Delta t_2 \to 0 \\ \Delta W_2 \to 0}} (E(0)-E(1))$$

$$\ge \lim_{\Delta W_2 \to 0} (2501-1.805 t_2^{\ 0}) \Delta W_2 = 0^+ > 0 \qquad (\because 2501 >> 1.805 t_2^{\ 0})$$

$$\therefore E(0) > E(1)$$

and

$$Since\ W_2^{\ k} \le W_2^{\ n} = W_R,\ n > k;\ and\ -|X+Y| \ge -|X|-|Y|.$$

$$\therefore \lim_{\substack{\Delta t_2 \to 0 \\ \Delta W_2 \to 0}} (E(k)-E(k+1))$$

$$\ge \lim_{\Delta W_2 \to 0} (2501-1.805 t_2^{\ 0}) \Delta W_2 = 0^+ > 0 \qquad (\because 2501 >> 1.805 t_2^{\ 0})$$

$$\therefore E(k) > E(k+1)$$

That is, $\boldsymbol{E(k)}$, k=0, …,n is a decreasing function while $\boldsymbol{W_2^k \ge W_R}$, shown as (5).

$$\therefore E(0) > E(1) > \cdots > E(k) > E(k+1) > \cdots > E(n) \qquad (5)$$

With the same reason as the proof of the first zone while $\boldsymbol{W_R > W_2^0}$, Function of $\boldsymbol{E(m)}$, m=n, …,N will be a increasing function while $\boldsymbol{W_2^m \le W_2^n = W_R}$, shown as (6). That is,

$$E(n) < E(n+1) < \cdots < E(m) < E(m+1) < \cdots < E(N) \qquad (6)$$

So $\boldsymbol{E(j)}$ is a decreasing function, while 0≤j≤n; meanwhile, is an increasing function, while N≥j≥n. (7) is shown as below.

$$E(0) > E(1) > \cdots > E(n-1) > E(n) < E(n+1) < \cdots < E(n-1) < E(N) \qquad (7)$$

Due to (7), E(n) will be the least enthalpy difference. That is, the setting pair is located at point j=n, which is ($\boldsymbol{t_2^n}$, $\boldsymbol{W_2^n}$) on the psychrometric chart. There are three cases about the space states calculated with LEE. The first state (R3), which is (20 °C, 60% rh), approached the optimum thermal comfort line. The value of (2) is shown as Fig. 3. The setting pair will be selected at (24.63 °C, 45.3% rh). The second state (R4, 26 °C, 40% rh) and the third state (R5, 24.5 °C, 50% rh), which are approached the optimum thermal comfort line. The values of (2) are also shown as Fig. 3. The setting pairs will be selected at (24.69 °C, 43.2% rh) and (24.5 °C, 50% rh) respectively, which are located at the optimum thermal comfort line.

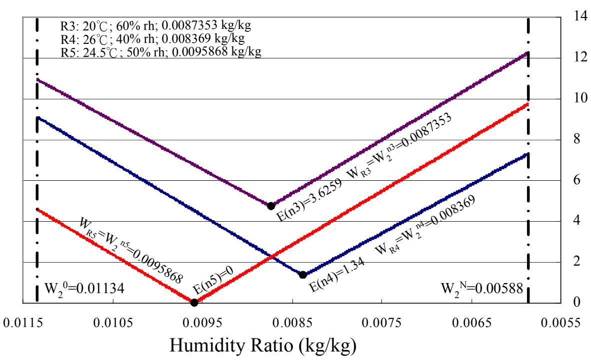

Fig. 3. Enthalpy difference of cases while $\boldsymbol{W_2^0 \ge W_R \ge W_2^N}$

C. *Least Enthalpy Difference at* W_2^N

The region of $\boldsymbol{W_2^N > W_R}$ in Fig. 1 represents that the air humidity ratio in the space is less than the humidity ratio at (25.085 °C, 30% rh). According to (2), function of the enthalpy difference between space state and the optimum thermal comfort line is a decreasing function. The least enthalpy difference between space state and the optimum thermal comfort line is located at point j=N, which is also the smallest humidity ratio of the optimum thermal comfort line.

The proof of the least enthalpy difference, while $\boldsymbol{W_2^N \ge W_R}$, expresses as the region while $\boldsymbol{W_2^k \ge W_R}$, proofed as preceding section. That is, $\boldsymbol{E(j)}$, j=0, …,N is a decreasing function while $\boldsymbol{W_2^N > W_R}$, shown as (8).

$$E(0) > E(1) > \cdots > E(k) > E(k+1) > \cdots > E(N) \qquad (8)$$

The least enthalpy difference is located at point j=N, which is (25.085 °C, 0.00588 kg/kg). There are two cases about the space states calculated with LEE. The first state (R6), which is (22 °C, 20% rh) approached the optimum thermal comfort line. The value of (2) is shown as Fig. 4. The setting pair will be selected at (25.085 °C, 30% rh). The second state (R7) is (28 °C, 22% rh). The value of (2) is also shown as Fig. 4. The setting pair will be selected at (25.085 °C, 30% rh) as well.

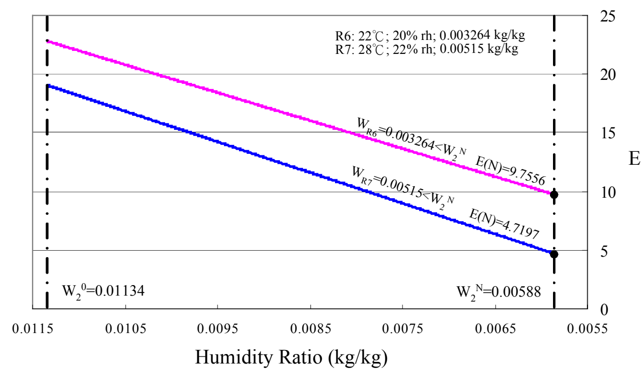

Fig. 4. Enthalpy difference of cases while $\boldsymbol{W_R < W_2^N}$

D. *Decisions of LEE*

According to the discussions of LEE, the solutions of the least enthalpy difference are dominated by the humidity ratio of space thermal condition. The humidity ratio of the setting pair that is decided by LEE is W_2^S, shown as (9). The corresponding temperature on the optimum thermal comfort line is t_2^S. The solution of LEE is located at ($\boldsymbol{t_2^S}$, $\boldsymbol{W_2^S}$). This solution could apply for the HVAC control system as a setting. According to this setting pair, the HVAC control system will do the suitable behavior depending upon its controller which could be any applicable controller such as fuzzy, NN or even PID.

$$W_2^S = \begin{cases} 0.01134\,, & W_R \geq 0.01134 \\ W_R\,, & 0.01134 \geq W_R \geq 0.00588 \\ 0.00588, & W_R \leq 0.00588 \end{cases} \tag{9}$$

E. Simulations of LEE on Psychrometric Chart

The simulations of LEE, which makes a decision to approach the optimum thermal comfort line from seven space thermal states respectively, are shown as Fig. 5.

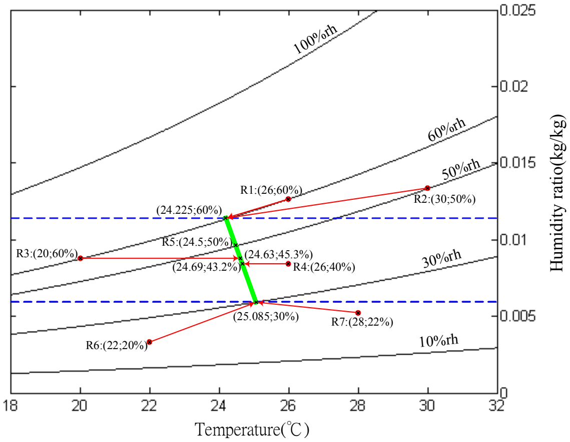

Fig. 5. Simulations of LEE decision

There are two cases R1 and R2 in zone 1. These cases need to treat air condition with cooling and dehumidification. Obviously, the consumed energy of R2 approaching to the optimum thermal comfort line for air treatment with HVAC apparatus will be larger than R1. Meanwhile, The residual of the least enthalpy difference of R2 is also greater than R1 by LEE decision. So the algorithm of LEE is aimed at energy saving as in the actual behaviors of HVAC system. The two cases, R6 and R7, in zone 3 are the same as cases in zone 1.

There are three cases in zone 2. Those are R3 (20 ^{o}C, 60% rh), R4 (26 ^{o}C, 40% rh) and R5 (24.5 ^{o}C, 50% rh) respectively. It is the same situation as the cases in zone 1. Besides, R5 is on the optimum thermal comfort line as well. The residual of R5 is zero by LEE. It is suitable to maintain space thermal comfort condition with LEE.

This simulation shows a reasonable setting, which could respond fast to resist the worst case of space thermal condition and finely adjust to fit the gradual change by LEE.

V. Conclusion

Since people stay indoor over a long period of their business time in a typical sedentary and moderate activity level with a similar clothing insulation during the same season. By integrating the enthalpy theory and the definition of thermal comfort control level, the proposed LEE can opportunely determine a suitable thermal setting pair for HVAC control systems. Thus, a proper set point on the optimum comfort line is picked depend on the measured space temperature and relative humidity. According to this set point, the actuators of a HVAC system will follow the proper set point and reliably track the setting of both indoor temperature and relative humidity to maintain a comfortable space dynamically.

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