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Understanding the Impact of Variable Refrigerant Temperature Controls on Energy Efficiency and Occupant Comfort

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Executive Summary

This paper investigates the effect of variable evaporating temperatures on occupant comfort and system efficiency. Through the use of accepted thermodynamic principles we demonstrate that the efficiency of a VRV/VRF system using such technology will increase at part load conditions and that increased off-coil temperatures have the potential to decrease occupant dissatisfaction in thermal comfort.

Introduction

Whether you are an owner, operator, prospective buyer, or installer of any HVAC system, your primary concerns about the system likely relate to its energy consumption and its ability to maintain occupant comfort. A great deal of time is spent designing buildings, selecting systems, and implementing control strategies to meet these goals, but typically the refrigerant evaporating and condensing temperatures are not utilized as a mechanism to control energy efficiency and occupant comfort.

Variable refrigerant flow (VRF) systems offer a highly flexible and efficient solution to providing heating and cooling to a space. These systems benefit from energy efficient inverter-driven compressors, and have the ability to meet varying individual heating and cooling loads throughout a building without the losses or construction challenges of other systems.

Building Applications and Requirements

In the past, HVAC equipment has been rated, advertised, and purchased on the basis of energy efficiency tested at a single operating temperature under full load – when outside temperatures are at the extreme and space conditioning needs are maximum. In reality, HVAC equipment rarely operates under full load. As such, it is critical to understand the extent of partial load operation within buildings and the impact on annual energy consumption.

I. Several decades ago, governments around the world began setting minimum efficiency requirements (e.g. Energy Efficiency Ratio or EER) for HVAC equipment operating at full capacity and a single temperature condition. Since many HVAC systems rarely operate at full capacity, newer standards have been developed to express equipment performance under part-load conditions. Integrated Energy Efficiency Ratio (IEER) in the US and European Seasonal Energy Efficiency Ratio (ESEER) in Europe are used to express average energy efficiency over more realistic operating conditions. Table 1 shows how the IEER and ESEER use a weighting function to integrate part load performance in a representative manner. Both metrics provide a more realistic efficiency value based on industry-accepted regional averages of part-load operation; the weighting coefficients show just how infrequently systems operate under full-load conditions. In addition to these rating procedures, the more complicated calculations for Seasonal Energy Efficiency Ratio (SEER) and Heating Seasonal Performance Factor (HSPF) as defined in AHRI's 210/240 standard express equipment performance under the even more specific varying operating conditions of a particular climatic region.

Table 1: ESEER and IEER Calculation Procedures

	$ESEER = C_A*EER_A + C_B*EER_B + C_C*EER_C + C_D*EER_D$		$IEER = C_A*EER_A + C_B*EER_B + C_C*EER_C + C_D*EER_D$	
	Ambient T [°C] / Load [%]	C [%]	Ambient T [°F] / Load [%]	C [%]
A	35°C / 100%	3	95°F / 100%	2
B	30°C / 75%	33	81.5°F / 75%	61.7
C	25°C / 50%	41	68°F / 50%	23.8
D	20°C] / 25%	23	65°F / 25%	12.5

II. Along with hourly variability in cooling load, the latent (dehumidification) load in a building also changes over time. HVAC equipment must be sized in buildings to provide both sensible and latent cooling at the rated full-load conditions. Humidity must be removed from the air in the cooling season to maintain occupant comfort, and the heat exchanger temperature must be below the dew point in order to condense moisture out of the air. Figure 1 shows this process on the psychrometric chart; air is cooled sensibly from 1-2, moisture is removed from 2-3 and the air is reheated to a comfortable state in 3-4. When dehumidification loads are not high, a conventional system will often still operate with a low evaporating temperature (below the dewpoint). In these scenarios, it is possible to increase the evaporating temperature above the dew point to still provide sensible cooling, but significantly increase the energy-efficiency by reducing the pressure lift of the system. Process 5-6 shows a more mild cooling condition where the air can be cooled sensibly with a higher evaporator temperature and no latent load.

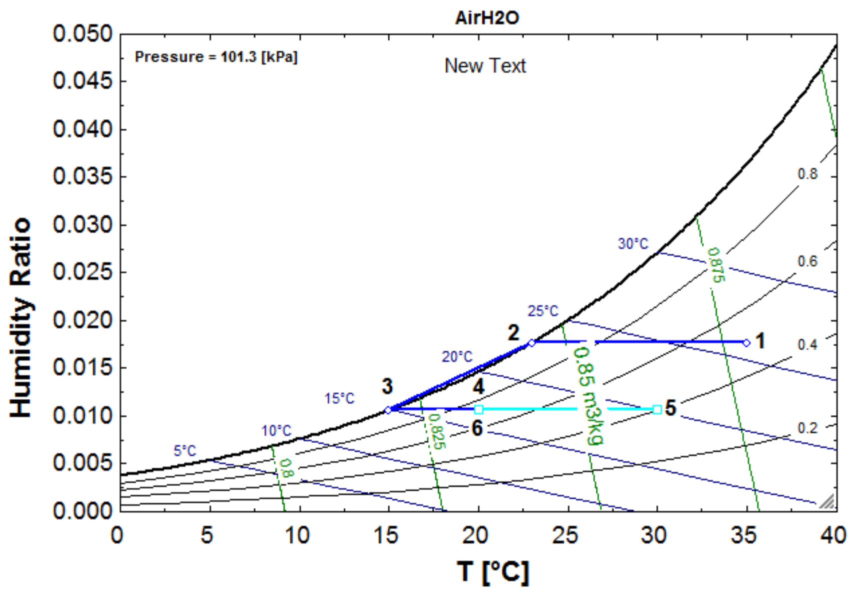

Figure 1: Psychrometric Chart for Cooling

Basic Thermodynamic Principles

As discussed, a vapor compression cycle in the real world doesn't always operate under full-load; here we'll show how several different variables can control the performance of the cycle. Figure 2 shows the operation of a simple vapor compression system under air conditioning conditions: refrigerant is compressed from process 1-2, the condenser releases heat in 2-3, liquid refrigerant expands across an expansion device in 3-4, and absorbs heat from in the evaporator in 4-1. The cooling capacity of the evaporator equals the refrigerant mass flow rate times the enthalpy difference between points 1 and 4 (Eqn. 1). Compressor power consumption is expressed by the refrigerant flowrate times the corresponding enthalpy difference (Eqn. 2).

$Q_{cooling} = \dot{m}_{ref}*(h_{evap,out} - h_{evap,in})$ *Eqn. 1*

$W_{compressor} = \dot{m}_{ref}*(h_{comp,out} - h_{comp,in})$ *Eqn. 2*

The conventional approach to modulate capacity in part-load conditions is to reduce the compressor speed and thus refrigerant flow rate. The capacity and power consumption decrease but the evaporating temperature does not change and thus the pressure lift seen by the compressor remains the same (and efficiency remains comparatively low). Equation 3 shows the cooling capacity in terms of the air side. Here, we see that we can reduce cooling capacity by decreasing the heat transfer coefficient U_{HX} (by reducing air or refrigerant flow rates), by decreasing the heat exchanger surface area, or by decreasing the temperature difference between air and evaporating refrigerant (increasing Te).

$$Q_{cooling} = U_{HX}(\dot{m}_{ref}, \dot{m}_{air}, \ldots) * A_{HX} * (T_{air} - T_e) \qquad \text{Eqn. 3}$$

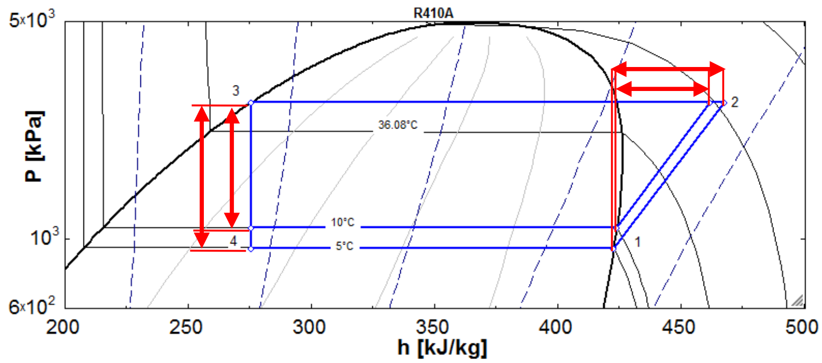

Figure 2: R410A Air Conditioning Cycle P-h Diagram

Figure 2 shows an example vapor compression cycle at two different evaporating temperatures. The vertical red lines show the difference in pressure lift from increasing the evaporating temperature from 5 to 10°C and the horizontal lines show the corresponding reduction in compressor power input. In a scenario where the dehumidification load is not significant, it is practical to increase the evaporating temperature in this way to improve efficiency and still provide comfort. At a given compressor speed, a cycle will always operate with a higher energy efficiency when the evaporating temperature is increased.

Figure 3 shows an example of the conventional approach to part-load control, where the compressor's speed is reduced to decrease capacity and as a consequence efficiency may change due to the design of the compressor. In Figure 4, the part-load condition is met by decreasing compressor speed and also increasing evaporator temperature. Because the capacity is less when the evaporator temperature is higher (Eqn. 3), the compressor speed needs to be higher than in Figure 3 where the evaporator temperature was fixed. Figure 4 shows how the efficiency can be improved in part-load conditions by adjusting both compressor speed and evaporating temperature.

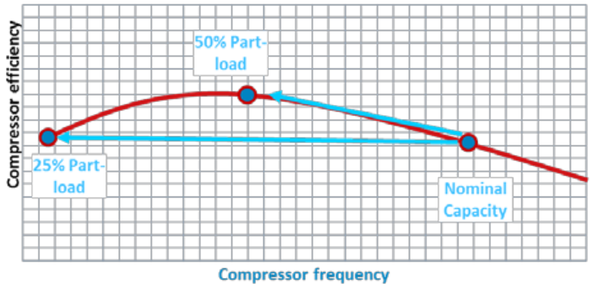

Figure 3: Conventional Part-load Control

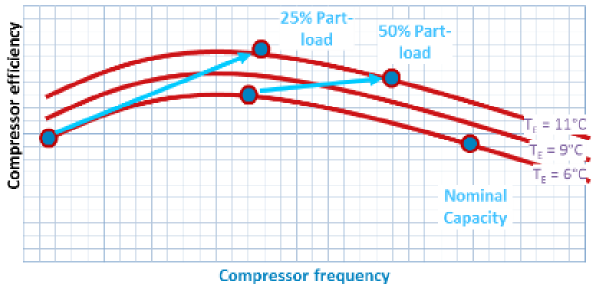

Figure 4: Part-load Operation with Increased Evaporator Temp

Occupant Thermal Comfort

The thermal comfort of occupants should not be sacrificed as we strive to improve the energy efficiency of VRF systems. Although there are multiple factors contributing to human beings thermal sensations, air temperature, humidity ratio and supply air flow velocity are certainly among the most important parameters. Occupants can directly control the three parameters through thermostat setting and fan speed settings, therefore the bulk air conditions in a conditioned space are typically well controlled.

However, the local thermal condition around discharge air outlets are usually hard to control. Figure 5 shows predicted percentage of people feeling uncomfortable as a function of supply air temperature. The predicted percentage of dissatisfied (PPD) is an index defined in ASHRAE standard 55 (2013) to quantitatively predict the percentage of thermally dissatisfied people under a certain indoor condition. The index takes indoor temperature, humidity ratio, air velocity and other factors into consideration and is formulated based on human subject tests/surveys. To obtain the PPD, predicted mean vote (PMV) has to be calculated first using the equations defined in ASHRAE Standard 55. Typically, the PMV index falls between -3 and +3. The lower the value, the colder the thermal sensation to occupants. PMV value of 0 means neutral/comfortable. The relationship between PMV and PPD is shown in Figure 6. In the case of cooling, about 50% of people tends to feel too cold once the discharge air temperature drops below 21°C. In the case of VRT, the evaporating temperature of the system is increased to save energy. As a result, the supply air temperature will be increased too. The increase of supply air temperature effectively improves the thermal comfort of the conditioned space since less percentage of occupant is predicted to be uncomfortable. If the supply air temperature can be increased by 4K (25°C), 95% of the occupants are thermally comfortable. In winter, the situation is reversed in the sense that too high discharge temperature will cause a higher percentage of dissatisfaction. A variable refrigerant temperature control in winter will decrease the condensing temperature, and as a result, the supply air temperature is also decreased. As it shown in Figure 7, when the discharge temperature decreases by 8K, VRT control can effectively control the dissatisfaction rate under 5%.

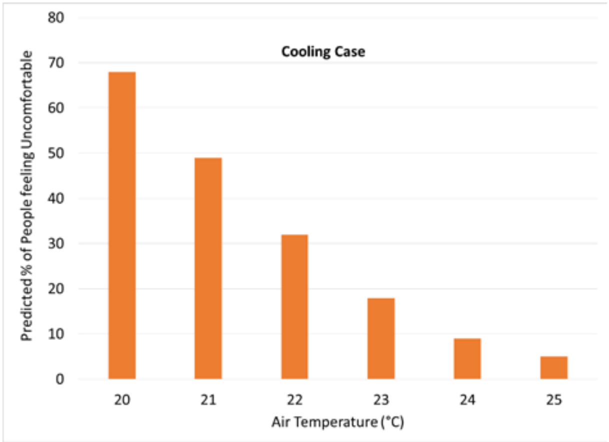

Figure 5: Thermal comfort of various air temperature (cooling)

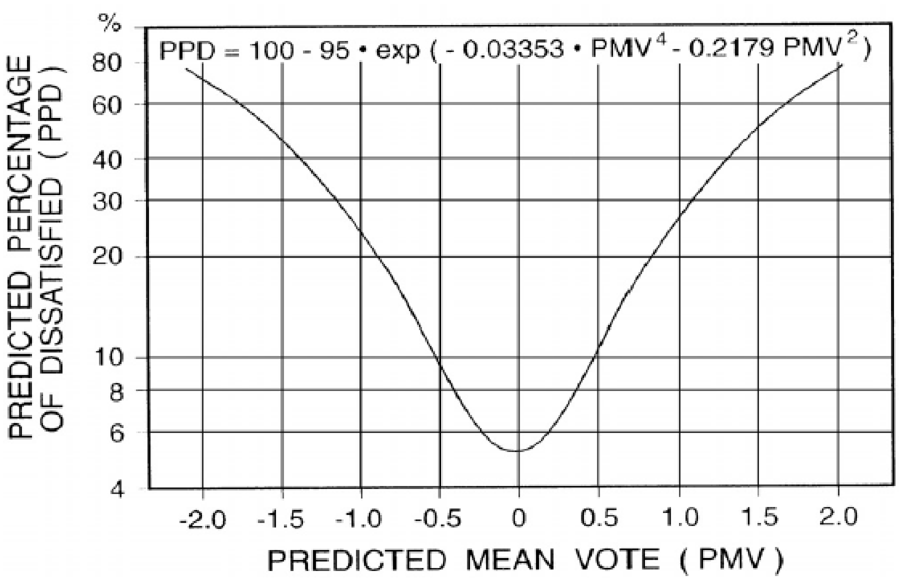

Figure 6: Predicted percentage dissatisfied (PPD) as a function of predicted mean vote (PMV) (source: ASHRAE standard 55 (2013))

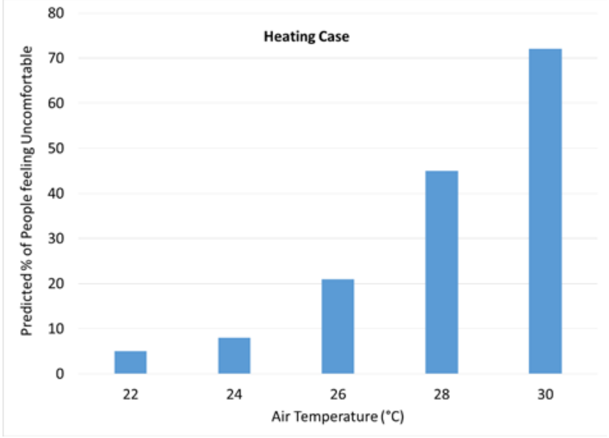

Figure 7: Thermal comfort of various air temperature (heating)

VRT can bring another benefit of reducing vertical temperature difference. The indoor units of most split systems are installed either close to the ceiling or in the ceiling (cassette units). This setup can bring a vertical temperature difference in the space. In summer, the discharge temperature may be as low as 10°C while the set temperature in room is usually higher than 18°C. In winter, the temperature difference is even higher. Moreover, since the warm air flows to occupants head while their feet are colder, the thermal dissatisfaction is even worse. The VRT increases supply air temperature in summer and on the contrary, reduces supply air temperature in winter. Consequently, the vertical temperature difference is reduced and better thermal comfort is expected.

Variable Refrigerant Temperature Control

In order to realize the benefits of variable refrigerant evaporating/condensing temperature, a control strategy must be able to decrease compressor speed and increase evaporating temperature when cooling load is lower than the rated capacity (or decrease condensing temperature in heating mode). This will result in greater efficiency than the conventional technique of only reducing compressor speed in part-load conditions. However, it is essential for the system to also retain the capability to operate at the designed low evaporating temperature when sensible loads are high and/or the occupants demand a fast response to changing loads. It is desirable for users to have multiple control settings to choose between maximum efficiency and minimum response time to meet the required setpoint especially in high humidity conditions.

In order to demonstrate the energy savings potential of variable refrigerant temperature control, an idealized vapor compression cycle like the one shown in Figure 2 is simulated under part load conditions. In simulations for the 'conventional' system, the evaporating temperature is fixed at 6°C, while the 'varying temperature' system allows the evaporating temperature to increase to as much as 13°C in the part-load condition. Table 2 summarizes the performance of these two systems. The simple model assumes a 10°C approach temperature difference between the ambient air and condensing temperature and a fixed 65% compressor isentropic efficiency. The system with varying refrigerant temperature has significantly improved efficiency under part-load conditions which are weighted heavily by the ESEER, thus the ESEER rating is improved by 28% by allowing refrigerant temperature to vary during part load conditions. Evaporating temperatures around 6°C are typical in conventional systems and VRT systems would be capable of increasing the evaporating temperature to 16°C. It is therefore reasonable to conclude that an actual system may increase the ESEER by 20-30% by varying the refrigerant temperature.

Table 2: Energy Savings Estimation

Ambient T [°C] / Load [%]	Conventional System Te [°C] / EER [W/W]	Varying Temperature Te [°C] / EER [W/W]
35°C / 100%	6°C / 3.4	6°C / 3.4
30°C / 75%	6°C / 4.1	9°C / 4.6
25°C / 50%	6°C / 5.0	11°C / 6.3
20°C] / 25%	6°C / 6.3	13°C / 9.4
ESEER	5.0	6.4

Summary

VRF and other variable-speed systems have the capability to reduce energy consumption by operating more efficiently in part-load conditions. Varying refrigerant temperature (by increasing evaporating temperature in cooling or decreasing condensing temperature in heating) can further improve energy efficiency in part-load conditions by reducing the pressure lift and resulting compressor power. In addition to energy efficiency gains, the supply air temperatures can result in equal or greater occupant comfort.