Evaluation of Mechanical Dehumidification Concepts (Part 2)

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Evaluation of Mechanical Dehumidification Concepts (Part 2)

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ABSTRACT

In recent years, humidity concerns have gained increasing attention in the air conditioning industry. In part, this increased attention is the result of high indoor air quality standards and newly introduced legislation and industry regulations. Humidity concerns have become even more visible since the introduction of alternate refrigerants with superior thermo-physical properties that may adversely affect system dehumidification capability. Although various dehumidification concepts have found their way into standard equipment offerings, the choice of the system type and configuration is not obvious and entails detailed analysis of multiple operational parameters. This paper analyzes general trends of essential performance characteristics and evaluates system design sensitivity over a range of environmental conditions. The conclusion reached is that the reheat method utilizing a two-phase refrigerant mixture may provide fairly advanced and flexible approach to address a wide spectrum of potential applications in terms of latent and sensible capacity demands while preserving system functionality and reliability.

1. INTRODUCTION

In recent years, humidity concerns have gained increasing attention in the air conditioning industry. In part, this increased attention is the result of high indoor air quality standards and newly introduced legislation and industry regulations. In addition, if certain conditions related to humidity exist, equipment applications can actually promote the growth of mold and bacteria. These conditions include: 1) High rates of internal humidity generation; 2) Buildings with insufficient insulation and poor construction; 3) Humid outdoor environments combined with high fresh air circulation requirements. These factors add another degree of complexity to the humidity issues, often resulting in lengthy and costly litigations for the original equipment manufacturers and consulting firms. Also, the introduction of alternate refrigerants with superior evaporation thermo-physical properties, such as R410A, (Lemmon et al., 2002) results in relatively high evaporation heat transfer coefficients and elevated saturation temperatures, which in turn may negatively impact system dehumidification ability at certain operating conditions.

It is not surprising that various dehumidification concepts and techniques have found their way into standard equipment offerings. Such methods have been classified by several authors (Harriman et al., 2001) and are based on the system type (mechanical, electrical, desiccant, etc.) or internal construction (integrated or add-on). Hybrid systems have been considered as well.

The selection of system type and configuration is not always obvious and entails detailed analysis of multiple characteristics, such as performance, life-cycle cost (partially defined by the system runtime and efficiency), reliability, control complexity, design flexibility, expandability, similarity to existing equipment currently in use, serviceability, etc. Mechanical dehumidification systems, especially the configurations utilizing the primary refrigerant circulating throughout the cycle, are often the category of first choice. The attractiveness of these systems is enhanced by relative application simplicity, solution flexibility with respect to treatment of both indoor and outdoor air streams, and elegant design options.
Although the category of mechanical dehumidification systems employing primary refrigerant is relatively narrow, numerous design schematics and arrangements are available within the class. In order to select a proper system type, the fundamental question of the application requirements must be addressed. The system conceptual design strongly depends upon a range of indoor and outdoor environments, or stated differently, upon the relative significance of sensible and latent load components over an array of operating conditions. Although a universal solution is desired, most of the systems are geared towards one end of the design spectrum or the other. The author of this paper makes an attempt to evaluate two fairly popular mechanical reheat/dehumidification concepts introduced to the market for air conditioning and heat pump applications: sequential hot gas scheme (Whinery and Chapin, 2002) and two-phase mixture scheme (Bussjager, 2004).

2. SYSTEM BEHAVIOR & SENSITIVITY

It is essential to analyze each dehumidification concept not just at the rating point but also at off-design conditions in terms of system reaction and sensitivity to various environmental conditions and operational parameters. Issues such as performance degradation, potential system malfunctioning, reliability and required changes in the control logic must be carefully evaluated. The following parameters are considered the most critical for system operation and are examined in the section below: ambient temperature, indoor relative humidity, outdoor airflow, indoor temperature, and indoor airflow. To eliminate the system specifics, the results are presented in a normalized form as ratios to system performance in the conventional cooling mode of operation at identical conditions. In other words, the analysis will show augmentation each dehumidification concept is able to provide in comparison to the conventional system schematic at each set of conditions.

2.1 Ambient Temperature

Air conditioning systems must function at various ambient conditions and should be able to sustain the desired performance in the dehumidification mode of operation as well. Since the warm liquid design (Bussjager, 1997) naturally becomes a part of the two-phase mixture schematic, we let the control logic switch to this operating mode at ambient temperatures above 95°F (35°C).

For the sequential hot gas system, the temperature difference between the refrigerant exiting the compressor and the indoor air stream leaving the evaporator varies with the ambient temperature. The higher the ambient temperature is, the greater the amount of heat rejected into the indoor air stream by the reheat coil. At the same time, the refrigerant vapor quality entering the condenser drops and the condenser fraction of the overall amount of heat rejected by the system diminishes accordingly. Since the refrigerant condensation temperature increases with the ambient temperature and the system subcooling decreases, the evaporator enthalpy difference and its capacity subsequently decrease. Although both components of the evaporator capacity diminish, the latent capacity drops at a much faster rate, since the evaporation temperature temperature increases as well (a result of the condensation temperature/subcooling phenomenon described above). At off-design ambient conditions, above and below the zero (neutral) sensible capacity point, the system will correspondingly provide heating or cooling. The farther away from the neutral point (on the ambient temperature scale) the system operates, the more undesired heating or cooling will be supplied as a by-product of the dehumidification cycle. Finally, the system latent efficiency (a ratio of the evaporator latent capacity to the total power consumed by the system) drops with the ambient temperature increase.

The two-phase mixture system reacts in a different way to variations in the ambient temperature. As in the hot gas approach, the condensation temperature parallels the ambient temperature change. This change in condensation temperature causes the reheat coil capacity to follow a similar trend, since the latter is essentially defined by the difference in the indoor air and the refrigerant condensation temperatures. One of the major benefits of the two-phase mixture design is an increase in the subcooling of the refrigerant leaving the reheat coil as the ambient temperature increases. (Note that the subcooling of the refrigerant exiting the condenser still follows the downward trend.) This is drastically different from the hot gas approach and allows the evaporator performance boost at ambient temperatures higher than the design point temperature. Although both latent and sensible components of the evaporator capacity improve with the ambient temperature, the latent component increases at a greater rate, since the evaporating temperature falls with the overall system subcooling augmentation. While the evaporator sensible
performance is enhanced at higher ambient temperatures, it cannot entirely compensate for the reheat coil capacity increase. Nevertheless, the sensible capacity deviation from the neutral design point becomes smaller than in the hot gas approach. The increase in system latent performance as the ambient temperature increases is of great advantage, since more moisture usually needs to be removed from air at higher ambient temperatures. Lastly, the system latent efficiency also increases, following the latent capacity trend.

As noted above, with the two-phase mixture method, the system subcooling is progressively decreasing with the ambient temperature. This decrease in subcooling is the results of the temperature difference reduction and subsequent decline in the heat transfer potential of the reheat coil. Thus, in some cases, one of the available head pressure control options may need to be activated to prevent potential malfunctioning of the expansion device. Although this design is more sensitive to ambient temperature variations, the system will gradually recover, leading to an order of magnitude boost in the latent performance and similar adequate sensible cooling capacity reduction, as desired at lower ambient temperatures. Also, at lower ambient temperatures, the two-phase mixture system can be switched to a heating and dehumidification mode of operation by closing the refrigerant passage through the condenser and allowing a major portion of refrigerant to flow and condense in the reheat coil (bypassing the main condenser).

The trends described above are presented in a graphical form in Figure 1. For illustrative purposes only, the design point of the neutral sensible system capacity is selected for the ambient temperature of about 85°F (29.4°C) (the negative numbers correspond to heating) while the indoor space is kept at 80°F (26.7°C) dry-bulb and 67°F (19.4°C) wet-bulb conditions. As can be seen from the graph, the latent performance for both the hot gas and the two-phase mixture systems is almost identical at the design point. Although all the latent performance trends are upwards (on a normalized basis), the actual latent capacity of the hot gas design decreases with the ambient temperature rise and in reality is just slightly higher than the latent performance for the conventional cooling system. Obviously, the design point of the neutral sensible system capacity and the required latent capacity are at the designer’s discretion and can be chosen at any ambient temperature.
At relatively high ambient temperatures, both sensible and latent components of the system capacity are required to satisfy increased cooling and dehumidification demands. In such ambient conditions, the hot gas design would switch to the conventional cooling mode of operation. The two-phase mixture system, however, has the option either to transition naturally to the warm liquid mode (by completely closing the condenser bypass valve) or to operate in the conventional cooling mode. Let’s assume that the trigger point for the system controls occurs at ambient temperatures near 95°F (35°C). If the cooling mode is executed, both systems perform identically and all the aforementioned performance ratios become equal to unity. Although for the hot gas system the sensible system capacity is significantly enhanced, the latent performance drops in comparison to the dehumidification cycle. This is the penalty paid for the condenser coil size and system subcooling reduction as well as for the condensation and evaporation temperature increase. On the other hand, for the two-phase mixture system, the latent performance is slightly improved when the transition to the warm liquid method occurs while the sensible system capacity reaches reasonably high levels of about 75% of the conventional cooling system (see Figure 1). Therefore, the two-phase mixture system inherently seems to have a higher degree of flexibility in design and in satisfying various latent and sensible load demands.

![Relative Humidity Effect](image)

**Figure 2**

### 2.2 Indoor Relative Humidity

It is not uncommon for a dehumidification system to operate in environments with varying indoor humidity levels. The system should be able to respond adequately to humidity changes by removing sufficient amounts of moisture in order to keep the conditioned space within the comfort zone or recommended specification. Both the hot gas and the two-phase mixture designs react similarly to variations in indoor humidity and exhibit identical performance trends. As expected, the evaporation temperature increases with the relative humidity lift, followed by an insignificant rise in the condensation temperature and the condenser subcooling. However, there are some differences between the hot gas system and the two-phase mixture system. With the two-phase mixture design, although the condenser subcooling stays relatively flat, the overall system subcooling does decrease at higher humidity levels, unless head pressure control is activated. In general, the two-phase mixture system is more sensitive to humidity level changes (although the subcooling stayed within the predetermined limits). This phenomenon occurs as a result of: 1) a rise in the evaporation temperature when the humidity level rises, 2) a subsequent increase in the refrigerant mass flow rate, and 3) a reduction in the temperature difference and the heat transfer potential of the reheat coil. Although the total evaporator capacity increases slightly with the relative humidity, its sensible component decreases (see Figure 2). Since the reheat coil performance stays relatively flat, the overall system sensible capacity decreases with the relative humidity elevation. The evaporator latent performance for both systems increases at the same rate, exhibiting an equal ability to remove moisture from the indoor air stream. The most

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noticeable enhancement in the evaporator latent performance, in comparison to the conventional system, is achieved at the lower end of the relative humidity spectrum. This increase in latent performance is the result of the improved reheat coil performance due to lower evaporation temperatures. Although Figure 2 represents the data at the specific indoor-outdoor environment, these performance trends are almost identical at other conditions.

2.3 Outdoor Airflow
Outdoor (condenser) airflow affects many of the system operational parameters and should be given special attention in the evaluation of the dehumidification system performance. Although at the present time the vast majority of air conditioning systems are not configured for the variable speed operation, the trend of achieving higher efficiencies and reducing the lifetime cost of the equipment is becoming one of the most critical concerns in the industry. In addition, many applications use various means of outdoor airflow adjustment to achieve the required head pressure control. Also, operating the system in dirty environments may vary condenser coil airside impedance, causing a change in the fan operating point and the amount of delivered airflow. Thus, it becomes important to analyze the effect of the outdoor airflow on the dehumidification system performance.

As expected, in the hot gas approach, the condensation temperature and subsequently the reheat coil performance decrease with increased outdoor airflow. As a result, the vapor quality of the refrigerant entering the condenser is increased. At the same time, the augmented airflow promotes heat transfer enhancement and temperature approach reduction in the condenser. These two contradictory trends influence the condenser subcooling, the enthalpy of the refrigerant entering the evaporator, and the evaporation temperature. In most cases, the system subcooling decreases and the evaporation temperature remains almost flat with the change in the airflow. The evaporator sensible and latent capacities do not exhibit any noticeable deviations. Since the reheat coil performance decreases with the airflow enhancement, total system sensible capacity displays an upward trend. Additionally, due to the condensation temperature reduction, the system power consumption diminishes, followed by a boost in the system latent efficiency. The performance trends outlined above make the hot gas configuration somewhat less responsive to head pressure control by condenser airflow adjustment.

In the two-phase mixture design, the condensation temperature and the reheat coil performance reveal an identical behavior to the hot gas system discussed above. The refrigerant mixing point at the condenser exit retains an equivalent vapor quality. As a result, the condenser subcooling as well as the overall system subcooling decreases with the outdoor airflow augmentation, due to reduced temperature difference and heat transfer potential in the system.
condenser and reheat coil. In most cases, the enthalpy of the refrigerant entering the evaporator diminishes, causing the evaporation temperature to exhibit an upward trend as airflow increases. The evaporator sensible and latent performances decrease, followed by an increase in the total system sensible capacity. A reduction in power consumption and system latent efficiency can be observed as well. Thus, the two-phase mixture concept is, in some respect, more sensitive and adaptable to the head pressure control by the means of airflow adjustment.

As shown in Figure 3, the rates of change in essential performance characteristics for both dehumidification schematics are almost identical (this is true for all conditions). The discrepancy in absolute values of relative latent performance for the two systems is related to the fact that the condenser airflow analysis is performed at low ambient temperatures, far away from a performance parity point, which is, as stated above, at the system designer’s freedom. Figure 3 also shows that the most improvement in the dehumidification ability, relative to the conventional system, is achieved at the lower end of the airflow spectrum, since the reheat coil plays a more significant role in the latent performance enhancement at higher condensation temperatures.

2.4 Indoor Temperature
Indoor (conditioned space) temperature is highly dependent on the application, so it is important to evaluate the dehumidification system performance at various indoor dry-bulb temperatures. As expected, for both the hot gas and the two-phase mixture approaches, the evaporation and condensation temperatures increase as the indoor temperature rises. Although both temperatures exhibit an upward trend, the rate of change for the evaporation temperature is more pronounced. As a result, reheat coil performance decreases slightly as the indoor temperature increases. This phenomenon occurs as the result of two counteracting processes: the decreased heat transfer potential in the reheat heat exchanger diminishes its capacity, but an increased refrigerant flow rate through the coil boosts its performance. The overall effect is negative.

In the hot gas system, the condenser subcooling is generally enhanced with the rise in indoor temperature, since the condensation temperature and subsequently the heat transfer potential in the condenser are enhanced, and in spite of an increase in the refrigerant flow circulating through the system. Also, the increased refrigerant flow rate and system subcooling lead to the evaporator latent and sensible capacity augmentation with the indoor dry-bulb temperature boost. The latent performance is enhanced at a much higher rate than the sensible performance, if the relative humidity in the conditioned space is kept constant, leading to increased amount of moisture in the air stream at higher dry-bulb temperatures. Also, the abovementioned phenomena, along with the reheat coil capacity degradation, contribute to the augmentation of the overall system sensible performance. Consequently, the overall system capacity, the power consumption and the latent efficiency exhibit upward trends as the indoor temperature increases.

In the two-phase mixture design, subcooling at the exit of the reheat coil decreases as the indoor temperature rises. The condenser, located upstream of the reheat coil (on the refrigerant path), cannot compensate for its diminishing performance in terms of reduced temperature difference and increased refrigerant flow rate. In the extremely rare cases in which the indoor temperatures are outside of the conventionally acceptable range of operation, head pressure control may be required. The evaporator latent performance is changing at a lower rate with the indoor temperature (although in the same direction as in the hot gas design), while the evaporator sensible capacity remains almost flat, causing the total system sensible capacity to improve slightly due to the reduction in the reheat coil performance. Finally, the power consumption and the system latent efficiency increase with the indoor dry-bulb temperature elevation, but similarly, at a lower rate in comparison to the hot gas system.

The trends of the essential normalized system parameters discussed above are shown on Figure 4 at constant ambient temperature and indoor relative humidity. (These trends are similar over a wide spectrum of ambient temperatures and indoor relative humidity.) It is clearly seen that although the system sensible capacity remains practically flat for the two-phase mixture schematic at the wide range of indoor dry-bulb temperatures, its latent performance diminishes at the high end of the indoor temperature spectrum (relative to the conventional system). Without the implementation of head pressure control, the latent performance of the two-phase mixture system
becomes almost equal to the conventional system operation at these higher temperatures. An activation of head pressure control will remedy the situation and boost the two-phase mixture system performance to a desired level.

![Indoor Temperature Effect](image)

**Indoor Temperature Effect**

Indoor Dry-Bulb Temperature, [°F]

Figure 4

**2.5 Indoor Airflow**

Similar to outdoor airflow, indoor (evaporator) airflow can vary for a number of reasons. For instance, variable air volume (VAV) and variable volume temperature (VVT) systems are found in multiple applications in the industry. In addition, plugged filters or customized ductwork may cause a shift in the fan operating point, even for the preset airflow systems. Thus, the effect of indoor airflow on the dehumidification system performance must be evaluated.

In the hot gas system, the evaporation temperature changes in unison with the indoor airflow, while the condensation temperature remains constant. Furthermore, the system subcooling diminishes slightly with the airflow increase, since the higher refrigerant flow circulates through the system. The reheat heat exchanger performance is generally augmented with the airflow enhancement, primarily due to a boost in air and refrigerant flow rates, and in spite of a reduction in the temperature difference between refrigerant and air in the reheat coil. The overall system sensible capacity increases as the airflow increases, since the evaporator sensible cooling performance is improving at a higher rate than the reheat coil heating performance. (The evaporator is not affected by the temperature difference between the indoor air and the condensing refrigerant.) Although the total evaporator capacity increases with the airflow and the evaporation temperature rise, its sensible component is improving at a higher rate, leaving only a rapidly decreasing remainder for the latent component. Decreasing latent performance, combined with the reheat coil performance improvement rate outpacing the evaporator performance, causes a decrease in total system capacity with the indoor airflow boost. Also, power consumption increases and the system latent efficiency decreases with the airflow enhancement.

Generally speaking, the two-phase mixture design reveals similar behavior and follows identical trends, with the exception that the reheat coil subcooling increases with the airflow supplied to the conditioned space. This is attributed to the fact that the system subcooling enhancement is not restricted by the ambient air temperature but is defined by the refrigerant and the airflow rates and the temperature difference in the reheat coil between the condensing refrigerant and the air exiting the evaporator. It can be seen from Figure 5 that both dehumidification schematics perform in a similar fashion with no anticipated problems or system malfunctions. The most significant improvement is demonstrated at the high end of the airflow spectrum, since the reheat coil performance...
enhancement in terms of the system subcooling is the most profound in that region. Finally, it must be noted that although the latent performance graphs in Figure 5 display an upward trend (relative to the conventional system), the actual tendencies of the latent system characteristics, in the absolute sense, follow the downward slope.

### Indoor Airflow Effect

<table>
<thead>
<tr>
<th>Indoor Airflow Effect</th>
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<tr>
<td><strong>Normalized Performance</strong> (to Conventional System)</td>
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<tr>
<td>1.5</td>
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<tr>
<td>System sensible capacity (2-ph mixture)</td>
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<tr>
<td>Evaporator latent capacity (2-ph mixture)</td>
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<tr>
<td>System latent EER (2-ph mixture)</td>
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<tr>
<td>System sensible capacity (hot gas)</td>
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<td>Evaporator latent capacity (hot gas)</td>
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<td>System latent EER (hot gas)</td>
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#### 3. CONCLUSIONS

A dehumidification/reheat design concept should be selected based on the requirements of a particular application in terms of the cooling and heating needs and the moisture removal criterion. The comparison analysis conducted in this paper for the hot gas reheat design and the two-phase mixture reheat design permits to conclude that the two-phase mixture approach may possess several appealing features and provide adequate coverage for a wide spectrum of potential applications. Although the critical performance characteristics in this study have been normalized by the conventional system parameters and general conclusions have been drawn, some dependence on the system configuration and design remains hidden, even in a non-dimensional representation. For instance, this influence would manifest itself through the conventional system efficiency values (and the means of altering the basic refrigerant cycle to achieve these efficiency levels) as well as through the utilized refrigerant type. Having said that, it is still the author’s belief that the general performance trends and a relative position of the characteristic curves remain unchanged.

### REFERENCES


