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INTEGRATION OF DEHUMIDIFICATION INTO LIFE CYCLE SYSTEM DESIGN

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ABSTRACT

When designing an air conditioning system, many commercial and physical parameters must be determined and variables accommodated. The nonlinearities between them are significant. A system which gives excellent performance at one of the infinity of possible combinations of internal load and external weather may produce unacceptable conditions at some other seemingly more benign combination. A system selected to satisfy those occasional peak loads may fail to perform satisfactorily during the more common part load conditions. An extensive study of the intensive properties of air passing through a dehumidifier coil has led to the development of the family of air conditioning systems presented here. The integrated LFV/HCV system with staging is described over its full operating range and is compared with conventional design solutions. A high driving potential outside air pretreatment system, applicable to both temperate and tropical climates, is shown to improve indoor air quality and reduce both capital and operating costs.

INTRODUCTION

In the course of the last 15 years of research in the design of comfort air conditioning systems for large buildings there has been very little improvement in the performance of conventional systems other than the marginal gains derived from integration with building management systems. We attribute this in large part to the fact that the designer's contribution to the selection of the all important dehumidifier is very small. It is selected mainly on the basis of peak heat load performance. Often a functional specification is written and the detailed task of selection is left to the supplier of the dehumidifier. As indicated in previous papers (see for example Shaw and Luxton 1988), within the air conditioning design community there is considerable misunderstanding of the heat and mass transfer performance of the dehumidifier. The cost of the air conditioning system, the energy requirements of the system, the health of the occupants of the building and the reputation of the engineering profession depend very much on the performance of the dehumidifier and how it interfaces with the total system. Comfort is a hard state to define. It is difficult to quantify the full effects of being sealed in a faulty artificial climate for 8 hours a day. Some progress has been made and this conference evidences the multidisciplinary nature of the problem. New and improved Standards and Codes of Practice are appearing and the first step toward solution - the recognition that the problem exists - has been taken. Nevertheless, ***we have to conclude that very few existing conventional designs can meet the ASHRAE Specifications for Human Occupancy (ASHRAE, 1989a) and for Indoor Air Quality (ASHRAE, 1989b) during any significant proportion of their operating hours***. To the best of

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our knowledge there is no conventional system of air conditioning based on mechanical refrigeration which will meet existing engineering standards for performance and energy efficiency where the system is required to serve widely varying climates or loads.

THE SOLUTION - BACK TO BASICS

A major step toward a solution to the situation described above becomes obvious if we go back to first principles. The heart of the air conditioning system is the dehumidifier and we must understand its performance over the entire range of its operation. In previous papers (Shaw 1982; Shaw and Luxton 1988), many misinterpretations of the basic principles have been indicated. They have arisen because the bases on which the principles have classically been formulated differ from those encountered in air conditioning. For example, consider the perfectly correct statement to be found in most heat transfer textbooks that an **increase** in the airstream velocity of moist air through a dehumidifier will result in increased condensation of moisture. Behind the statement are the assumptions that there is no constraint on the mass flow rate of the airstream, that there is no relationship between the latent heat load and the sensible heat load which must be offset by air conditioning systems, and that no constraint is placed on the temperature difference between air supplied to and returned from an air-conditioned space. In air conditioning what actually applies is very different.

If we hold both entry conditions to the coil and coil surface area constant, and compare the behaviour of a low face velocity coil to that of a conventional face velocity coil, a **decrease** in the moist airstream velocity through a dehumidifier will result in:

- **increased** condensation of moisture **per unit mass flow of dry air**;
- a process line drawn on a psychrometric chart which has a larger gradient and approaches a straight line as the velocity of the airstream approaches zero;
- a mean air-side wetted surface temperature which reduces as the air flow velocity decreases;
- a Reynolds number of the air flow which indicates that the flow is viscosity controlled, i.e. **the air flow is not turbulent** (Luxton and Shaw 1991) a conclusion which also applies for conventional face velocities;
- an air side heat transfer coefficient which is lower;
- a ratio of mass transfer to heat transfer which is greater;
- a coil which has fewer rows of depth;
- reduced fan power requirements since the pressure drop through the dehumidifier is reduced;
- reduced fan room noise.

In earlier papers we have also established the importance of maintaining a relatively high coolant velocity in air conditioning design (Sekhar et al. 1989; Shaw et al. 1992). Here, on the coolant side a high Reynolds number and a high heat transfer coefficient biases the temperature of the wetted outside surface toward that of the coolant. In the referenced series of papers, together with a number of associated papers and patents, we have described the evolution of the Low Face Velocity/High Coolant Velocity (LFV/HCV) based methods of air conditioning. The effect of the LFV/HCV combination on the coil surface temperature is discussed fully in Shaw and Luxton (1990).

Staging to maintain high coolant velocity

When the room heat loads and/or the outside air conditions vary, the accepted means of adjusting coil capacity to follow a reducing heat load is to throttle the chilled water flow rate. However this action, if allowed to proceed unchecked, will result in most cases in an increase in room humidity. The maximum acceptable room humidity is either defined by a client's specifications or by industry Standards. For example, ASHRAE Standard 55-89 considers 60% relative humidity as the upper bound for human comfort. As the sensible heat load decreases the RH in the room rises progressively and the 60% boundary may be reached. This signals that the dehumidifier is too large for the part load duty. The problem of the dehumidifier being too large is that the coolant flow through it must be reduced to a trickle to compensate. When this happens the coil surface temperature rises and dehumidification potential is lost. A "run-away" room humidity can result. By sensing coolant velocity, or an appropriate surrogate, it is possible to close down several circuits selectively to cause a uniformly distributed reduction in the coil depth, i.e. in the effective coil surface area. In the case of a VAV system a Supply Air Thermostat would then respond by adjusting the coolant flow to preserve the supply air temperature, ensuring that the now smaller coil satisfies the sensible load requirement. However, the coolant velocity in the reduced number of active circuits will now be higher and the coil surface temperature will be biased towards the coolant temperature, so restoring the dehumidification potential. We term this change in size "**staging**". Its use together with the general concepts of the LFV/HCV design method results, in practical design terms, in a dehumidifier which will simultaneously offset both the sensible and latent heat loads over the full operating range without the need for supplementary humidity control.

In some cases, as illustrated by the Standard Air Conditioning System of Table 1, no staging is necessary. However, when staging is not used, as in conventional practice, it is particularly important to understand the relationship between part load performance and peak load performance. Consider the case of a designer selecting a return air dehumidifier for the Standard Air Conditioning System of Table 1. "Rule of thumb" based on conventional wisdom would dictate a water temperature rise at peak load of 7°C (12·6°F) or higher. The resulting system may well show acceptable peak load performance. Part load conditions are however, likely to be those which promote a "sick" building. For a system which does not employ staging, and within a particular stage of one that does, falling loads are invariably associated not only with a fall in water velocity, but also with an increase in water temperature rise, which results in a decreased potential for dehumidification. If the potential is allowed to decline to a sufficiently low level, or if as often happens the surface temperature of a significant portion of the coil actually exceeds the dew point temperature of the air, excessive room humidity will result. It is the part load, rather than peak load conditions, which will be the norm during the life cycle of an air conditioning system. These can be produced by an infinite number of combinations of internal load and external weather conditions; it is accordingly essential that the designer must analyse the spectrum of load conditions to which his building will be subjected, and identify those which are likely to be associated with unacceptable conditions, (Luxton, Shaw and Sekhar 1989). For the example described, satisfactory performance across the entire operating range of the coil can be attained if the water temperature rise at peak load is 3.2°C (5.8°F) or less. For the Prestige Air Conditioning System (Table 1), satisfactory performance at the critical part-load condition is attained by reducing the active coil size, i.e. "staging", resulting in a higher part-load water velocity, a reduced water temperature rise, and a lower room humidity.

Usually one stage change is sufficient. In a particularly demanding example involving a lecture theatre in the tropics, two stage changes were required to meet comfort standards across the entire operating range of the system which encompasses a wide range of occupancy levels. Outside air conditions are invariably humid with the dewpoint seldom falling below 23°C (73°F) and frequently exceeding 26°C (79°F). In Figure 1, a plot of Room Sensible Heat Load versus Room Relative Humidity illustrates the effect of staging on the humidity within the lecture theatre, and compares it with the conventional VAV and CAV systems, in which staging is omitted. (It should be noted that the inability of the conventional system to maintain a sufficiently low dewpoint requires that a higher supply air temperature be used to avoid condensation on supply air diffusers). Figure 2 is a "**performance map**" showing the dehumidification performance of the system across its entire operating range, specified for this example in terms of outside air conditions and occupancy levels. The performance map depicts the performance of the coil in the case of decreasing loads from the full size coil operating at or near the peak condition (stage 1), to the intermediate stage 2 where a mid-size coil can satisfy the loads, to stage 3 in which the minimum size coil is all that is needed. (Note however that Stages 3 and 2 represent the largest proportion of the operating range). The conventional VAV and CAV performances are depicted for the case of full student attendance only (150 students). A computer simulation was used to generate Figure 2. These results are also presented in tabular form in Shaw et al. (1992). The changeover process between stages is both smooth and stable. Supply air temperature remains constant across a changeover, only the chilled water circuiting changing to place a bound on humidity with decreasing loads, and to ensure adequate refrigeration capacity with increasing loads.

Given a complete occupancy schedule and weather/time of day data, it would be possible to use the performance map to compute an accurate running cost for the lecture theatre air conditioning plant during a full year of operation. In the absence of more complete information, Shaw et al. (1992) performed an analysis of energy usage for the lecture theatre using several combinations of outside air conditions and occupancy levels, together with conservative estimates regarding the periods over which those conditions were pertinent. With all systems satisfying the same psychrometric conditions in the lecture theatre, the annual running cost of the conventional VAV system was calculated to be 37% higher than that for the LFV/HCV system, and that of the CAV system a factor of two greater.

THE HIGH DRIVING POTENTIAL (HDP), OUTSIDE AIR PRETREATMENT SYSTEM

It was found that by applying basic principles of heat and mass transfer, conventional methods of pretreatment of outside air could be markedly improved through a new and more effective outside air pretreatment system. The goal is to maximise the difference between the partial pressure of the air entering the outside air dehumidifier, and that of the water vapour at the wetted external surfaces of the dehumidifier coil. This is achieved by minimising the chilled water temperature rise through the dehumidifier and maximising the chilled water velocity, thus maintaining a high potential for dehumidification through the dehumidifier. The cold, dry outside air which results contributes towards meeting the room loads allowing smaller units within the building envelope. Conventional precooling, as currently practised in tropical climates, aims more to reduce the dry bulb temperature of the outside air than to dehumidify it, though some dehumidification does occur. The pretreatment plant is characterised by a modest chilled water flow rate which results in a chilled water temperature rise through the dehumidifier coil of around 8°C (14.4°F) to satisfy the entry requirements of the chiller. Precooling is seldom practised in temperate climates. Instead the driving potential available in the outside air is usually degraded by mixing with the system return air prior to passing through the dehumidifier. Consequently, an excessively large dehumidifier coil is required to cool and dehumidify the entering air to the supply air temperature and to offset the room sensible and latent heat loads. In conditions which produce low room sensible heat ratios, the consequence is that the room humidity may frequently fail to satisfy that which is acceptable for comfort and health.

One of the major problems of air conditioning systems is their common failure to satisfy the ventilation requirements during part load operation. This problem is particularly prevalent in the case of variable air volume systems serving a number of zones. VAV systems are otherwise preferred to CAV systems because of their inherently lower energy consumption (Figure 1). The problem is particularly serious during part load conditions for a building having a constant population density, especially in the lowest loaded zones. A building which requires 15% outside air at peak sensible heat load would require 30% outside air when the sensible heat load reduces to 50% of its peak value. If, as is common in the sick buildings deriving from the past quarter century, the outside air dampers are fixed such that the ventilation/return air ratio remains approximately constant during VAV operation, there is an inevitable tendency for stuffiness, especially in the lower loaded zones such as a library. Such conditions do not pose a problem for the new pretreatment method as there is benefit in maintaining the quantity of treated outside air as this air contributes significantly to satisfying the room load over the full operating range. Plant rooms can then be smaller, or even eliminated as return air treatment units have very modest duties and could be accommodated in the ceiling plenum. While performance gains are achievable by using LFV/HCV technology in the return air units to complement the HDP methodology, this is not essential because of the reduced dehumidification duty imposed on them.

The theory underlying the new pretreatment method and the techniques for its implementation address the dynamics of the problems arising from changing climatic conditions, changing ventilation requirements, and changing room sensible and latent heat loads. The water temperature rise through the outside air coil used in conventional precooling practice would preclude its further use elsewhere in the building. In the HDP outside air pretreatment system:

- The temperature rise of the chilled water in the outside air dehumidifier is small as the quantity of chilled water passing through it is large, the actual quantity being determined by the sum of the simultaneous demands of all the air handling units in the section of the building served by the pretreatment plant.
- At part load this high flow through the pretreatment plant can be maintained if the internal latent loads in the building are large.
- The chilled water path through the dehumidifier is kept short to minimise the water temperature rise and the pressure drop. The water temperature rise can be as low as 1°C (1·8°F) allowing the dehumidifier coil to present a low mean surface temperature to the airstream.
- The capital cost and energy penalties associated with the increased outside air quantities required by ASHRAE Standard 62-1989 are largely offset as the outside air performs part of the treatment of the return air.

The result is a system which resolves many of the problems to which conventional systems are prone when subjected to widely ranging operating conditions and ventilation requirements. During critical humid part-load conditions when the room sensible heat ratio is low and the outside air condition has a high humidity ratio, the outside air entry condition is close to the saturation curve and the HDP pretreatment system is especially beneficial.

Chilled water and air paths for a typical HDP system are shown schematically in Figure 3. The underlying psychrometric principles of such a system, when applied to a multistorey office building in the tropics, are illustrated in Figure 4. The process lines shown are appropriate to a system operating at 50% sensible heat load. The path of the coil condition curve effectively follows the saturation line which is the steepest thermodynamically possible path. At the critical part-load condition, the dew-point temperature of the air leaving the coil is 8·9°C (48°F) and the return air coil condition curve is very shallow, i.e. is operating close to dry. The outside air coil removes 10·4 g moisture per kg dry air (0.0104 lbm/lbm), as opposed to a mere 0·75 g/kg (0·00075 lbm/lbm) in the case of the return air coil. Mixing the pretreated outside air with the treated return air produces supply air having a lower dew point temperature than is otherwise attainable using conventional technology. The deep dehumidification offered by the outside air heat exchanger thus offers an efficient replacement for techniques such as overcooling and reheating where the maintenance of indoor air humidities within the ASHRAE comfort zone during periods of high dew point ambient conditions is required.

When systems are being designed using the conventional arrangement in which a mixture of the outside air and the return air is passed through a common coil, combinations of outside air conditions and room loads can arise which require rates of dehumidification per unit of sensible cooling which, even if they are possible thermodynamically, require such a low face velocity as to be economically non-viable. Such conditions are by no means restricted to the tropics. In the writers' experience they commonly occur during part-load operation in temperate climates as the result of high diurnal temperature swings, highly diverse weather conditions and variable internal loads. The HDP outside air pretreatment system described here is particularly applicable to such situations.

In the system shown in Figure 3, a bypass line connecting the chilled water supply and return risers enables a regulating valve to maintain a constant volume flow of chilled water to the outside air heat exchanger during part load conditions while the distributed air handling units are throttling the chilled water during periods of reduced room sensible load. This is precisely at the time during which the room sensible heat ratios approach their minimum values. Maintaining a constant chilled water flow rate allows the room RH to be maintained with only a minor parasitic penalty in terms of pump power consumption, particularly when the outside air heat exchanger is located in close proximity to the chillers. The higher water temperature rises in the (throttled) distributed handling units then serve to bring the mixture of the return flows into the range accepted by the chiller. The capital advantage enjoyed by the deep pretreatment system results from the reduced size of the terminal air handling units reduced alienation of rentable floor space and a significant downsizing of chilled water risers and chilled water pumps. The operating cost advantage is because chilled water pumping energy is significantly reduced, and, unlike conventional central plant systems, the quantity of air being moved is modest and with it is being moved the potential to reduce loads in the distributed units. Note that no return air riser is required, spill being effected via washrooms and toilets at each level. The value of the enhanced psychrometric and ventilation performance of the system is more difficult to determine; it is likely to be some function of the rentable value of the space in the building.

CONCLUSIONS

Air conditioning design requires an understanding of the dehumidifier performance and its relationship to variations in climatic range and room loads. The fact that the rôle of the dehumidifier is simultaneously to offset both sensible and latent heat loads and that these loads share a nonlinear interdependency imposes a heavy responsibility on the designer. In addition, demanding new ventilation and psychrometric performance criteria must now be met while both capital and operating costs are minimised. In this paper we have described and shown the advantages of two broad classes of systems, both of which originate from our research into dehumidifier performance characteristics and their relationship to overall system performance.

The heat and mass transfer advantages of the Low Face Velocity/High Coolant Velocity air conditioning system have been briefly explained. Part load conditions are shown to be a crucial though often neglected aspect of the design process. From a consideration of the coil condition curve it is clear that a VAV system is better suited to handling low room sensible heat ratio conditions than is a CAV system because it necessarily maintains a colder coil surface temperature. In the LFV/HCV system this characteristic is significantly enhanced by switching out some coolant circuits to reduce the effective coil surface area as the sensible load decreases. The ability of the LFV/HCV based VAV system to handle an extremely diverse range of load conditions has been demonstrated by reference to a particularly demanding application, that of a lecture theatre in the tropics.

The designer is often faced with the situation where architectural considerations place a constraint on the space available for air conditioning equipment. Even where such constraints are not overly restrictive, the opportunity to specify more compact equipment, thus increasing rentable floor space, might be seen as a reason for choosing one system in preference to another. A new methodology for precooling outside air which offers both superior performance and more compact return air handling units has been described. Use of the proposed High Driving Potential (HDP) precooling technique in conjunction with distributed return air units using conventional face velocities has been shown to satisfy rigorous comfort standards. A further increase in dehumidification performance can be achieved if needed by using LFV/HCV return air units. As part-load design in temperate climates is comparable in concept with peak load design is tropical climates, the new technology may be applied with advantage in both climatic situations.

To summarise, it has been shown that by paying appropriate attention to the fundamental principles of heat and mass transfer in the dehumidifier coil, it has been possible to develop a new family of air conditioning system designs which for the first time can satisfy the most demanding comfort specifications, in any environment over the full operating range of the system, while minimising both capital and operating costs in achieving that performance.

ACKNOWLEDGMENT

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TABLE 1.
Performance comparison of a fixed stage coil with a 2-stage coil serving the same heat load.

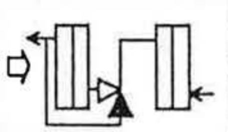
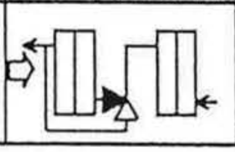
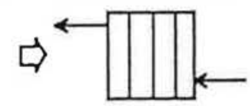
	"Prestige" Air Conditioning System Return Air Coil - 2 Stages		"Standard" Air Conditioning System Return Air Coil - Single Stage	
Performance Data	Peak Load Stage 1	Part Load Stage 2	Peak Load Fixed Size	Part Load Fixed Size
Room RH %	48.6	49.1	49.5	54.0
Water Temp. Rise	4.5°C (8.1°F)	2.0°C (3.6°F)	3.2°C (5.8°F)	7.6°C (13.7°F)
Ch. Water	4.6 Lps (9.8 cfm)	4.7 Lps (10 cfm)	6.5 Lps (13.8 cfm)	1.2 Lps (2.5 cfm)
Face Velocity	1.68 m/s (5.51 ft/s)	0.77 m/s (2.53 ft/s)	2.24 m/s (7.35 ft/s)	1.03 m/s (3.38 ft/s)
Room Sens	77.4 kW (264164 Btu/h)	39.0 kW (133106 Btu/h)	77.4 kW (264164 Btu/h)	39.0 kW (133106 Btu/h)
Schematic of Coil Arrangement				

FIGURE 1.
Effect of staging of LFV/HCV system compared with conventional VAV and CAV systems.

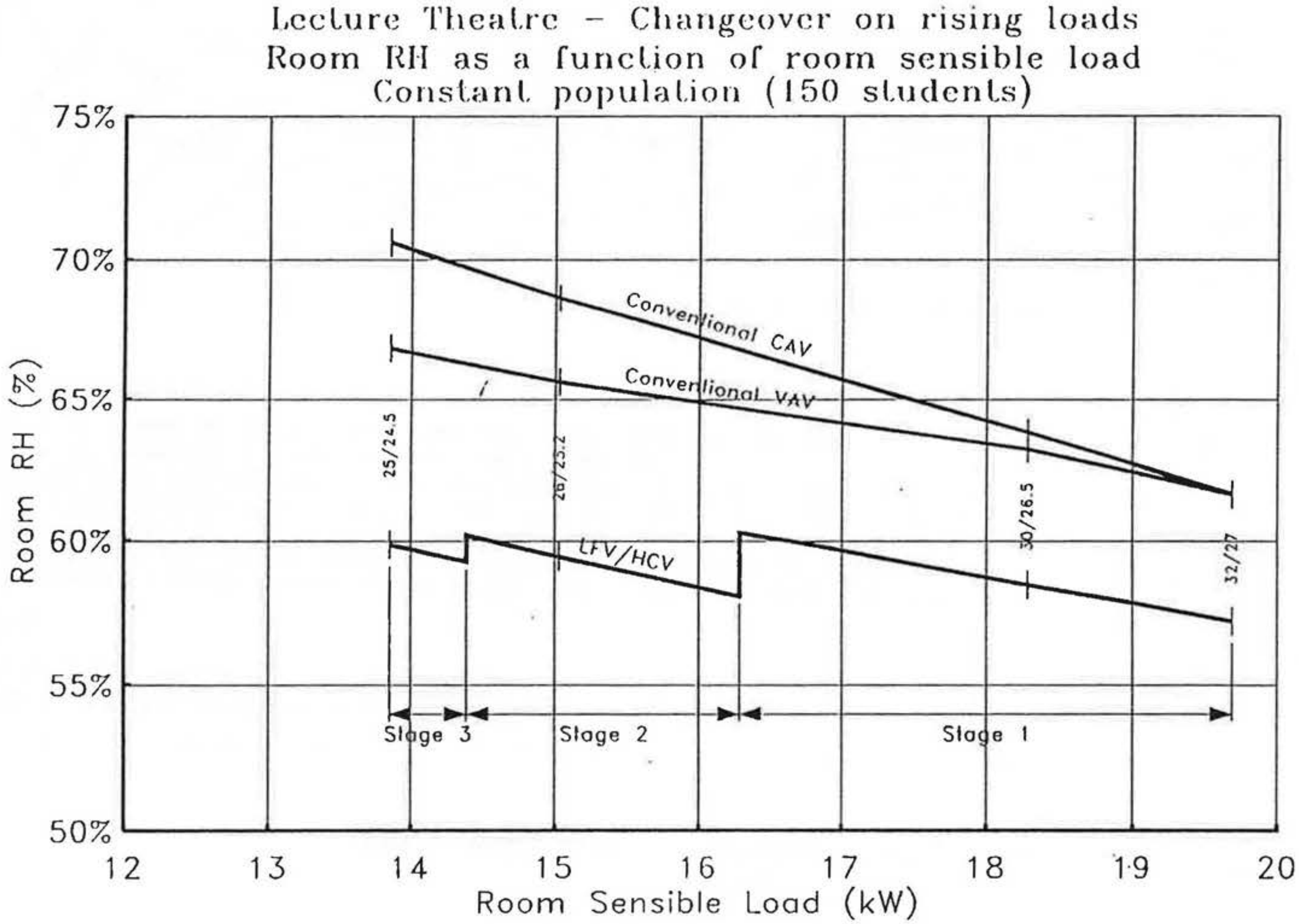


FIGURE 2.
Performance map for a lecture theatre in a tropical climate.

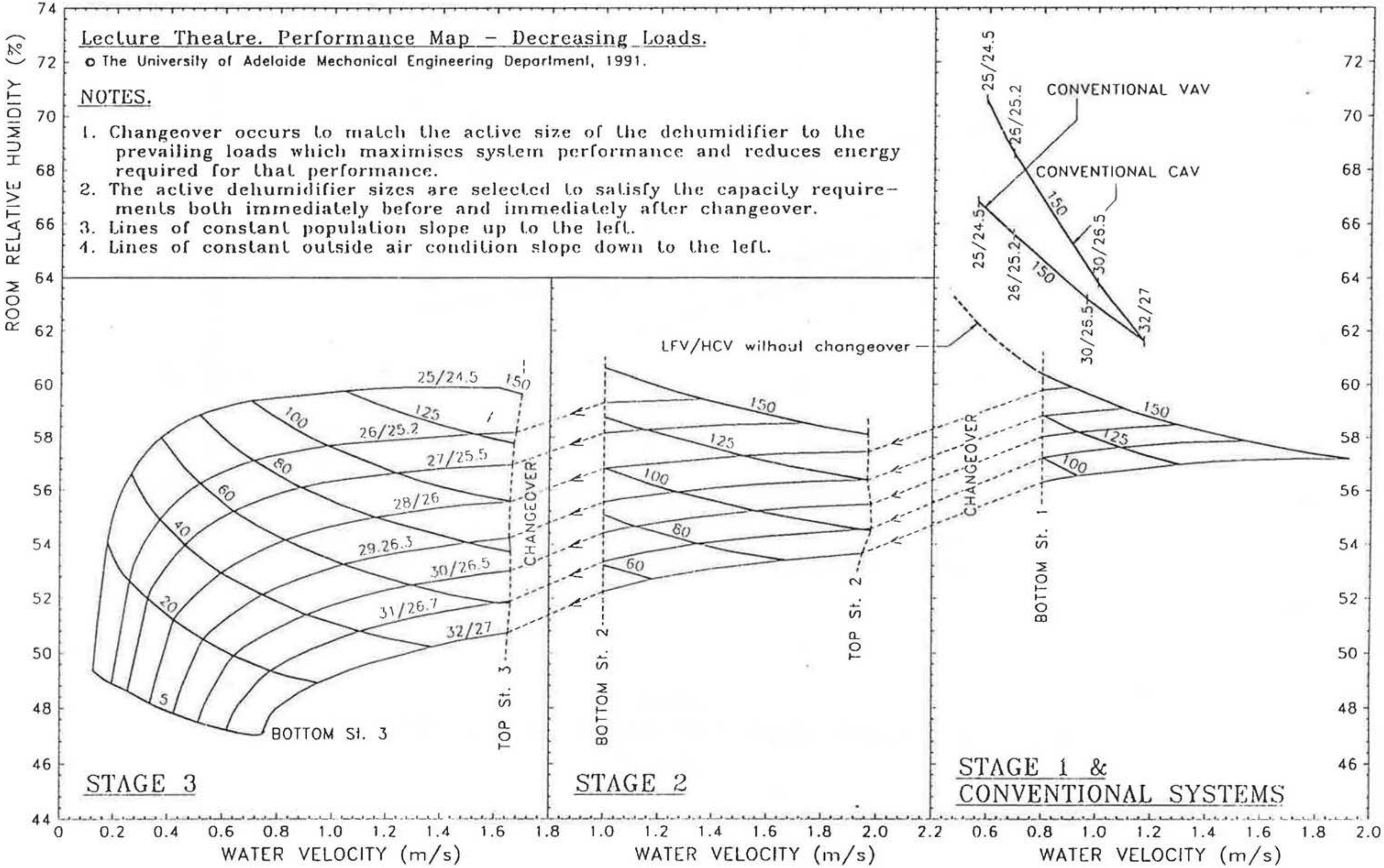


FIGURE 3.
Schematic layout of an HDP system applied to a multistorey building.

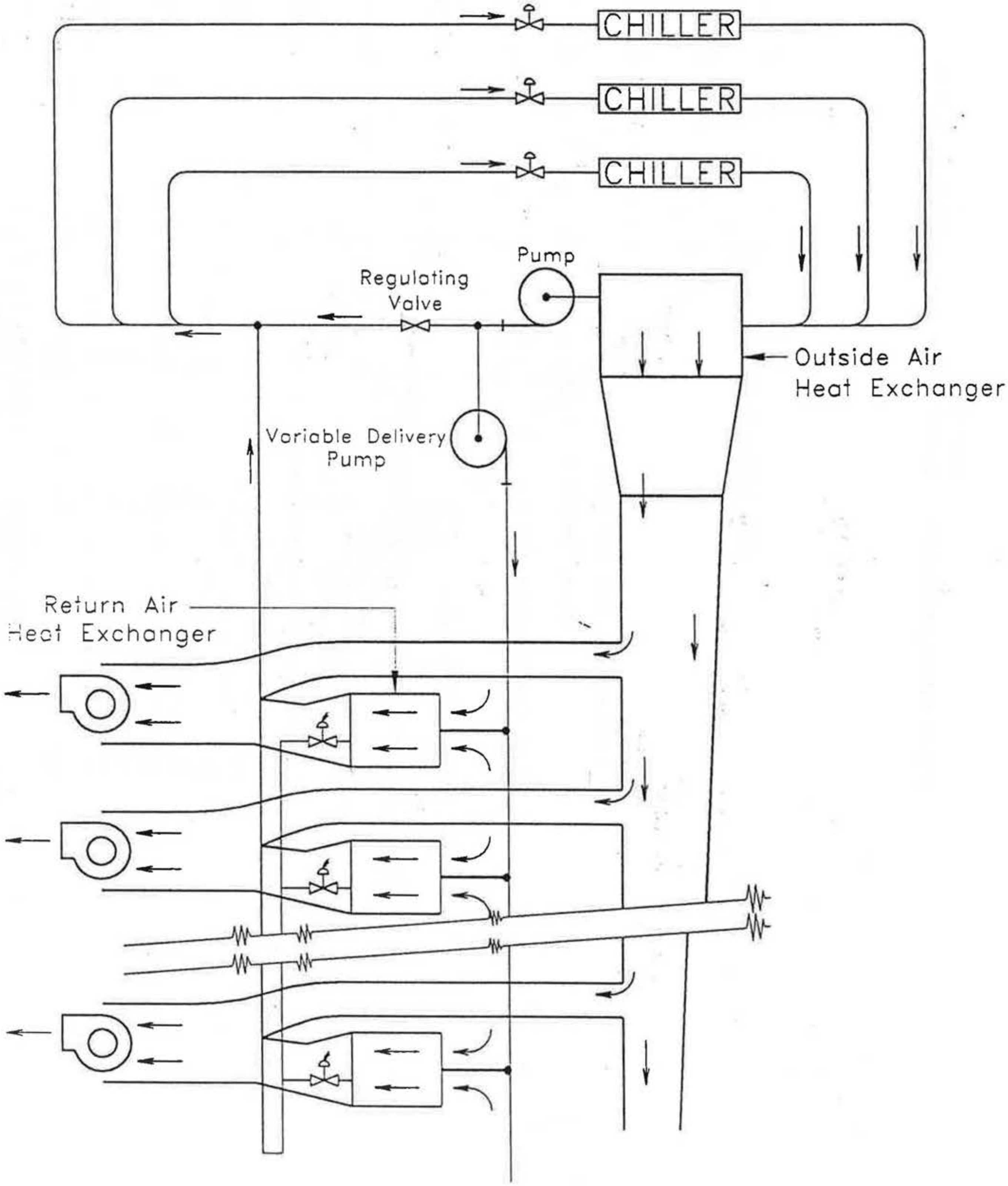


FIGURE 4.
Psychrometric performance of an HDP system under part-load conditions.