SELECTION AND CONTROL OF HVAC SYSTEMS FOR OPTIMAL LIFE-CYCLE EFFICIENCY USING SIMULATION TECHNIQUES.

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ABSTRACT
Design of air conditioning systems for efficient and effective part-load operation poses a considerable challenge, but one which is frequently not appreciated by the designer tied to traditional design practices. A plant simulation programme used in conjunction with a load calculation programme to assess rapidly the impact of equipment selection and control strategies on plant performance is described with the aid of an example. Use of the High Driving Potential (HDP) system provides the designer with the opportunity to implement a range of energy-saving strategies for optimising life-cycle plant operation. The outcomes are contrasted with those of the traditional energy wasteful strategies used to obtain acceptable part-load operation with conventional systems.

INTRODUCTION
The objective of a comfort air conditioning system is to maintain sufficient ventilation to restrict contaminant concentrations to acceptable levels within the conditioned space, while also achieving dry-bulb temperature and humidity which is within limits determined by health and comfort conditions. What constitutes an acceptable operating band for the psychrometric conditions within a space is to some extent open to subjective interpretation. ASHRAE Standard 55-1989, "*Thermal environmental conditions for human occupancy*", specifies a summer comfort zone in which the effective temperature lies between 22·8 and 26°C, and the relative humidity lies between 30% and 60%. There is some evidence (Busch, 1992) that, in tropical climates at least, higher dry-bulb temperatures than those specified by ASHRAE Standard 55-1989 may be acceptable. On the other hand, the authors have argued (Luxton and Marshallsay, 1998) that health considerations dictate that relative humidity should be restricted to a much narrower band; a design target of 45-55% would seem to be optimal from the point of view of both health and comfort.

In an air conditioning system the dehumidifier coil performs the ***dual*** function of removing latent and sensible heat from the air stream. Reliable measurement of relative humidity has traditionally been costly, and conventional strategies for controlling relative humidity explicitly through the use of overcooling and reheat add a heavy energy consumption burden. For this reason, in a typical installation dry-bulb temperature alone is controlled explicitly; thus humidity must be maintained within acceptable limits ***by design***. Attainment of satisfactory operating conditions at peak load is seldom a problem, provided the dehumidifier coil has sufficient capacity. Part-load conditions, which by definition are the norm, pose a far more challenging design problem, partly because they are frequently associated with a decrease in sensible heat ratio, but more fundamentally because of the manner in which coil operating characteristics alter in response to changing loads. However conventional design practice concentrates on design for peak conditions, with the far more common part-load conditions receiving scant attention. While inappropriate wording of contracts is often the reason for this situation, limitations in the design tools available in most consulting offices constrain change. Some automation is available in the form of load calculation programmes of varying degrees of accuracy and sophistication, proprietary coil selection programmes, and spreadsheets. However, the designer is still faced with the task of transcribing data manually from one software component to another, while the all-important task of estimating the psychrometric performance of a system is almost always performed using a ruler and pencil on a psychrometric chart; a procedure which is extremely labour-intensive, and does not generalise with any degree of confidence to systems serving more than one zone. Consequently, there is a severe disincentive to evaluating system performance at more than a few operating conditions, or to exploring more than a very limited subset of the available design solutions. If satisfactory system performance is to be guaranteed across the full range of operation, and if any attempt is made to optimise the selection of equipment and strategies for its operation, a holistic approach to the task of system specification and design is required.

While it is logical that the designer should seek to minimise the capital and operating costs of the system, and environmental impact should dictate that the latter is the more important consideration, sadly, the mind-set of the industry correlates operating efficiency with higher first cost and the structure of the industry militates minimisation of first cost. It will take some time for this situation to change but a significant constraint on change is the lack of efficient design tools that can alter the mind-set. In a conventional design framework capital cost reductions are often achieved by compromising both comfort specifications across a significant portion of the operating range and plant efficiency. A common consequence of design compromise is increased energy consumption It has been demonstrated (Luxton and Marshallsay, 1998) that by carefully considering the physical principles which underlie the air conditioning process, and using these principles to craft a suitable set of design tools, it is possible to reconcile the apparently conflicting considerations involved in air conditioning design. The design process may be looked at as an exercise in minimising an objective function representing the cost of owning the system (formulated as an appropriately weighted sum of the capital and operating costs), subject to a set of constraints, the most important of which will be those imposed by the comfort specifications and those by architectural considerations. These concepts, as they currently stand are too fuzzy to admit a precise mathematical formulation. However, the ability to evaluate rapidly the effect of design decisions on the performance of a system at critical points in its operating envelope does provide the designer with the opportunity to identify a near-optimal solution in an efficient manner. Indeed, the overview of system performance available may well facilitate identification of capital savings which will not be apparent through examination of a restricted subset of the design space.

There are two major software components involved in the design approach advocated herein; a load calculation programme, and the ZEBRA air conditioning simulation code. The essential input to ZEBRA is a set of loads, pertinent to building operation for a period of perhaps one year, with internal climate conditions set to the desired target values. It is the authors' contention that ***the load profile for a building should represent a mandatory component of the specifications for any major air conditioning project***. ZEBRA differs from other extant HVAC simulation codes such as SIMBAD (Husaunndée et al., 1997), and the plant simulation components of energy analysis codes such as ESP-r (Clarke, 1985) in that it models the *steady-state* thermal performance of air conditioning systems in response to variations in imposed loads and ambient conditions. This can be justified on the grounds that the time constants associated with the building envelope and ambient conditions are considerably longer than those which govern the behaviour of the air conditioning plant. Thus we can effect a decoupling between the building envelope and the plant, which is entirely in accordance with our requirement to be able to make a rapid assessment of the effect on plant performance of changes in plant specifications, operating parameters and operating schedules in the presence of a *known* load schedule. Once the equipment has been chosen, and its set-points and operating schedules selected, it is assumed that a control system can be designed to deliver the desired performance; packages such as SIMBAD exist for just this purpose.

STRUCTURE AND ALGORITHMS

The ZEBRA package has been designed according to object-oriented design principles, and is implemented using the C++ programming language. The structure of the programme has been described in some detail elsewhere (Marshallsay, 1996; Marshallsay and Luxton, 1997). At the component level, the model defines a set of classes representing the individual equipment items, the most important of which are the coil and the fan.

Class Coil provides a framework for simulating the performance of chilled water cooling coils. The heat and mass transfer characteristics of the dehumidifying coil are simulated by a dual-potential model, in which the bulk heat and mass transfer processes occurring across an incremental portion of the coil surface are modelled as a series combination of two potentials:

a) The rate of heat and mass transfer between the moist air stream and the coil surface is governed by the enthalpy difference between the air stream and the air in contact with the coil surface. If the dew point of the coil surface falls below that of the air stream, condensation occurs.
b) Heat transfer between the coil (or condensate) surface and the coolant is driven by the corresponding temperature difference.

To estimate the heat and mass transfer rates within the coil as a whole, the surface area is considered to be covered by two non-overlapping portions, one of which is fully dry, while the other is fully wet. The heat and mass transfer relation for the fully wet portion of the coil is derived from experimental test data using the AU method (Marshallsay, 1996), and expressed in the form of $St.Pr^{2/3}$ vs. Re. For the fully dry portion a data reduction technique based on that specified in ARI Standard 410-81, "*Standard for Forced-Circulation Air-Cooling and Air-Heating Coils*" is used. Given the condition of the air and coolant entering the coil, and an estimate of the total coil capacity ($\tilde{q}_t$), the corresponding wet and dry coil surface areas ($\tilde{A}_w$ and $\tilde{A}_d$) can be calculated using the model; the actual coil capacity can be found by satisfying the requirement that the estimated coil surface area must equal the actual surface area (A_o):

$$f(\tilde{q}_t) \equiv A_o - \tilde{A}_d - \tilde{A}_w = 0 \qquad (1)$$

following which the condition of the working fluids leaving the coil can be calculated. Air and water-side pressure drops are also calculated (Marshallsay, 1996). Additional data structures make provision for simple coil components to be combined in series and parallel with respect to both the air and water streams.

Fan performance is simulated within the framework provided by class Fan, subclasses providing for operations specific to centrifugal and axial fan types. Fan characteristics are derived from manufacturers' data, and represented as least-squares cubic spline fits. Class Fan also provides member functions to find the fan operating point for a specified duty using the non-dimensional fan characteristics; provision is made for fan control by riding the fan curve, by discharge dampers, or by variable speed fan motor.

Air handling units are represented by objects of class AHU, or one of its subclasses, each of which may contain 0 or 1 fan, and 0 or more coils. Flexibility in circuiting is an essential feature which, in particular, facilitates simulation of High Driving Potential (HDP) units, in which the outdoor air and return air coils are treated using separate coils (Luxton and Marshallsay, 1998).

Air conditioning systems are represented by objects of class System, each of which references one object of class AHU (or a derived class). A fundamental distinction is made between systems in which the referenced air handling unit is a stand-alone unit, treating both outdoor air and return air in a common casing, and those in which it is a central outdoor air treatment, or primary unit, serving one or more secondary or terminal units. The air handling unit in the first case, and each secondary unit in the second case will serve one or more conditioned areas (objects of class Zone).

The steady-state operating state for a system can be defined as that state at which the heat and mass transfer processes occurring within the system are in balance. In other words, commencing at some point in the system, it must be possible to take a parcel of air through one circuit, subject it to the appropriate psychrometric and mixing processes along the way, and return it to its initial psychrometric state. As noted above, it is customary to provide explicit control of dry-bulb temperature in an air conditioning system. Thus, if the dry-bulb temperature of the system is set at its desired target at the beginning of a circuit, it can be returned to that dry-bulb temperature at the end of the circuit by iterative modulation of the chilled water flow rate until the desired condition is reached. Humidity ratio is not normally subject to explicit control. However, if $\tilde{W}$ is an estimate for the humidity ratio at the beginning of the cycle, and the air is taken through a circuit and returned to a point at which the dry-bulb temperature balances, then the corresponding humidity ratio $\tilde{W}'$ is a function of $\tilde{W}$, and the equilibrium humidity ratio can be found by solving

$$f\left(\tilde{W}\right) \equiv \tilde{W}' - \tilde{W} = 0\,. \qquad (2)$$

In practice, it is usually most convenient to use the supply-air condition as the balance point. For a variable air volume (VAV) system the supply-air temperature is thermostatically controlled; for a constant air volume (CAV) system it is uniquely determined by the thermostat settings for the zones served.

A number of extensions of the basic algorithm are possible. For instance, in a conventional system overcooling and reheat are sometimes used to constrain zone relative humidity to lie at or below a specified limit. The solution without reheat is found as above, and if this constraint is not violated, this represents the desired solution. If the constraint would otherwise be violated, reheat will be invoked so that the constraint will be satisfied exactly. The target condition is now fully specified, and it is a simple matter to calculate the overcooling and reheat required to meet the condition. A more interesting situation arises if the target dry-bulb temperature setting cannot be met. This situation usually arises as a result of specifying coils with insufficient capacity, and would typically be encountered in the early stages of a design study. It can also occur when excess capacity is available. Consider the case of an HDP system in which the chilled water flow rate through the outdoor air coil is fixed, while that through the return air coil may be modulated to control dry-bulb temperature. If the load falls beyond the point at which the valve controlling the return air coil is fully closed, control over dry-bulb temperature will be lost. In either of these cases the solution procedure described above is no longer valid. The chilled water flow rate through the coil (or coils) is known, and a solution becomes possible if we can specify a functional relationship between zone dry-bulb temperature and load. For a system serving *n* zones, let ($\tilde{t}_i$, $\tilde{W}_i$) be an estimate for the condition of the air in zone *i*, and let ($\tilde{t}_i'$, $\tilde{W}_i'$) be the resulting condition when the air has been taken through one cycle. Define $\delta t \equiv \tilde{t}_i' - \tilde{t}_i$ and $\delta W \equiv \tilde{W}_i' - \tilde{W}_i$. For a CAV system the air flow rates for the various zones are fixed, and the equilibrium condition for the system can be found by solving the equation

$$f\left(\tilde{t}\,,\tilde{W}\right) \equiv \sum_{i=1}^{n}\left[\left(\delta t_i\right)^2 + \left(\delta W_i\right)^2\right] = 0\,. \qquad (3)$$

For a VAV system, the air flow to each zone will be modulated in an attempt to minimise the departure of the zone dry-bulb temperature from its thermostat setting. Let Q_i be the air flow to zone *i*, and let Δt_i be the difference between the zone dry-bulb temperature and its thermostat setting. Define $\delta t_{S/A} \equiv \tilde{t}'_{S/A} - \tilde{t}_{S/A}$ and $\delta W_{S/A} \equiv \tilde{W}'_{S/A} - \tilde{W}_{S/A}$, based on supply-air conditions. Then, the equilibrium condition can be found by minimising

$$\sum_{i=1}^{n} (\Delta t_i)^2 \qquad (4)$$

as a function of Q_i, *i* = 1, …, *n*, and subject to the constraints $|\delta t_{S/A}| \le \varepsilon_t$, and $|\delta W_{S/A}| \le \varepsilon_W$, where ε_t and ε_W are suitable tolerances. It is also necessary to constrain the air flow rates so that $Q_i \ge Q_{v,i}$, *i* = 1, …, *n*, and $\sum Q_i \le Q_F$, where $Q_{v,i}$ is the ventilation rate for zone *i*, and Q_F is a constraint imposed by the fan capacity.

For an HDP system control of dry-bulb temperature can be achieved by modulating the chilled water flow through one coil only (usually the return-air coil), while maintaining a fixed flow rate through the other coil. Alternatively, and usually less effectively, the flow rate through the two coils can be modulated in unison using a single valve. If explicit control of humidity is also desired, an extra degree of freedom is afforded by modulating the flow through the two coils independently. We can constrain the humidity ratio within a band of arbitrary width. To find a (non-unique) solution for a given set of operating conditions, the flow through one coil (usually the outdoor-air coil) is maintained fixed at the position determined by the last operating condition, and a solution is found using equation (2). If the solution violates either the upper or the lower humidity constraint, a new solution must be found which satisfies exactly that constraint. Since the desired psychrometric conditions are now completely specified, it is straightforward to calculate the valve settings for the two coils which will produce the desired conditions. Note that no overcooling or reheat is used; ***this is a minimal energy solution to the problem of explicit control of humidity***.

The above algorithms have been stated for the case of a system with a stand-alone air-handling unit; in most cases they generalise readily to the case of a system with central outdoor air treatment.

At the top level, the model defines a class HVACProject, which provides the central conduit for accessing all other components of the model.

LOAD CALCULATION PROGRAMME INTERFACE

ZEBRA provides a mechanism for handling load sequences at arbitrary time intervals in a custom format, known as a *metafile*. A filter has been implemented to facilitate conversion to metafile format of hourly load data sequences generated by the ENER-WIN building simulation programme developed at Texas A&M University (Degelman and Soebarto, 1995). Similar filters are proposed to handle input files from a number of other building simulation codes. The essential data required at each time step are the outdoor air condition, defined in terms of dry-bulb and dew-point temperature and atmospheric pressure, and for each zone

- Sensible load
- Latent load
- Outdoor air flow rate
- Dry-bulb temperature (a point within the thermostat dead-band)
- Load slope (kW/°C)

This last quantity is a measure of the rate at which the sensible cooling load for the zone departs from its given value, assuming that the deviation is a linear function of zone dry-bulb temperature. This is required to estimate the operating point of a system which has capacity that is insufficient, or is in excess of that necessary, to offset a particular load condition as described in the above section. Note that this value is required during the design process to indicate under-sizing of equipment, and to identify problems with scheduling; an approximate value will suffice provided it has the correct sign and order of magnitude.

Once created, a metafile can be loaded. Upon loading, the simultaneous peak sensible load for the system is calculated and stored, as are the individual peak sensible loads for the various zones. In the case of a CAV system, the sum of the individual zone peak loads determines the air flow, given a user-specified temperature rise through the zones (defaults to 9°C). The state of the zones and ambient conditions is best set to those pertaining to the simultaneous peak load as this corresponds to peak air flow and is the correct condition for sizing coils for both VAV and CAV systems. Fans and coils may be selected from the database to provide the required duty. Sizing of equipment items remains largely a subjective procedure relying on the experience of the designer. However, the ZEBRA user interface incorporates a number of features to assist the designer in the initial specification of equipment items, and in its subsequent refinement. These include:

- Default values for all fields where this is logically feasible.
- Assistants to guide the user in specifying zone ventilation requirements, in selecting fans for

efficient operation, and in selecting coil dimensions and circuiting to provide desired coolant and air face velocities.
- "Smart" dialogue boxes, in which altering one field will cause all dependent fields to be updated to maintain internal consistency at all times.

Having selected equipment items with sufficient capacity for the required duty, the user may run a number of time sequences from within the metafile to check the adequacy of the equipment selection across the operating range of the system, and to design strategies to optimise its part-load performance. The package supports a number of media which may be selected as desired for monitoring, tabulation and display of simulation results. These include:

- an internal report generator, which is updated dynamically to display the state of the simulation;
- a psychrometric chart display programme, which is updated dynamically to display the state of a selected system;
- an interface to the Excel spreadsheet programme. An extensive range of performance parameters are written into a workbook during the course of a simulation and are available for further analysis;
- a graphics programme, which may be used to display selected time sequences for a range of parameters on completion of a run.

While it is possible to run a simulation covering the entire cooling season, it is undoubtedly more efficient in the context of a routine design exercise to restrict attention to a suitably selected set of subsequences from the metafile. Systematic identification of those points in the load sequence which are critical from the viewpoint of the design process remains an active area of research (Koptchev and Luxton, 1996).

CONTROL STRATEGIES

Two strategies are available to deal with problems at part-load as they arise, and to optimise system part-load performance:

- *Scheduling* refers to the time-based control of plant operating hours at the global level, and of valve settings and chilled water flow rates at the unit level.
- *Staging* (Luxton and Marshallsay, 1998) is a load-based control mechanism necessarily implemented at the unit level. Typically, staging uses a surrogate for load, such as water pressure drop across a coil, to trigger a control action such as deactivating certain circuits within a coil to maintain dehumidification potential.

ZEBRA allows simulation of the effects of both scheduling and staging, either separately or in combination. Where both are implemented, staging will only be activated when scheduled. Both strategies are applicable within the context of either conventional or HDP design disciplines. The particular characteristics of each indicates that staging is a useful means of maintaining dehumidification potential for a conventional system, while scheduling is to be preferred as a control mechanism for HDP systems. In conventional systems, where outdoor air and return air are treated together through a common coil, staging allows the dehumidification potential of the coil to be revitalised as the sensible load decreases. On the other hand, for an HDP system with a fixed chilled water flow rate through the outdoor air coil, decreasing load results in decreasing humidity levels. If these fall to unacceptably low levels, or if the load falls to the point where chilled water flow to the return air coil is completely closed off, a scheduling strategy based upon increasing chilled water temperature and decreasing chilled water flow rates will usually provide adequate control, with associated benefits in terms of reduced pumping and chiller operating costs.

Within the ZEBRA scheduling model, the user is able to establish at the project level default weekday and weekend schedules for plant operation (on/off) and chilled water temperature. These default schedules remain in effect throughout the cooling season, except where the user elects to provide a custom schedule for specific months. The project plant operation schedules may be overridden at the system level, thus taking care of the case where certain systems within the building are required to operate over a period which differs from the norm. At the unit level schedules may be specified for chilled water flow rate (L/s), valve settings (modulating/fixed, with % open for the latter case; separately specifiable outdoor air and return air coil valves in the case of HDP systems) and staging (on/off). As for project schedules, default schedules are established at the unit level, and these may be customised on a month-by-month basis as desired.

By use of the tools provided, it should be possible to specify a set of schedules which will in most cases provide adequate plant operation throughout the cooling season. In climates where the weather is subject to significant day-to-day fluctuations, in Southern Australia, a pre-specified set of schedules cannot be expected to provide optimal operation. The authors are collaborating with the Australian Bureau of Meteorology to develop a scheduling scheme which will adapt to anticipated local weather conditions (Marshallsay et al., 1999).

CASE STUDY

The use of ZEBRA as a design tool is illustrated through a simple example. The test building is a typical 20-storey office featuring extensive areas of double-glazed reflective glass. The building is oriented at 45° to north, and has a square plan with each wall being 36·7 metres wide. In this example, attention will be restricted to floors 1 to 18, each of which has a floor plan as shown in figure 1, and comprises five conditioned zones, together with an unconditioned service area. These floors are conditioned on a floor-by-floor basis using VAV HDP units. The building has an occupancy of 1 person per 10 square metres, with a ventilation rate of 10 L/s/person.

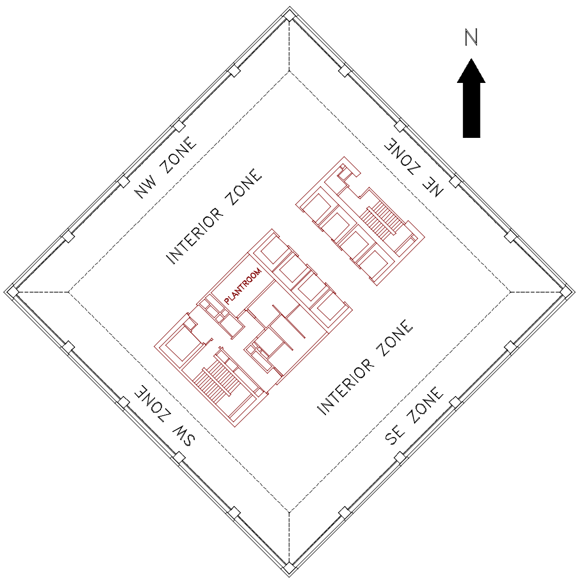

Figure 1. Plan of floors 1-18 of the test building.

For the sake of the present exercise, the building was considered to be located in Jakarta (lat. 6·1° S; long. 106·5° E), a city having a tropical climate. The performance of the building over a 12-month period was simulated using ENER-WIN, a synthetic weather record for this period being generated using ENER-WIN's internal weather generator. The resulting hourly load record was converted into a metafile by ZEBRA which generates a project automatically. The simultaneous peak load for the building occurred at 16:00 on 4th October, and corresponded to an outdoor air condition of 35·5°C db/23·6°C wb.

The unit was designed for a chilled water temperature of 6·5°C and a peak load water consumption of 4 L/s, of which all passed through the outdoor air (OA) coil. Flow through the return air (RA) coil was modulated to achieve the target zone dry-bulb temperature. OA and RA coils were selected to provide the required capacity at peak load; the specifications of the coils selected are shown in Table 1. Essentially, in an HDP design we seek an OA coil which will provide a coil-off dry-bulb temperature of 9-11°C, and an RA coil which will operate with a near-dry surface. The psychrometric chart component associated with ZEBRA provides a rapid means of assessing the suitability of candidate solutions in terms of their psychrometric performance. The selection is also subject to constraints imposed by a number of other factors, the most important of which are probably air face velocity, and the chilled water velocity, pressure drop and temperature rise. The first is determined by the overall coil dimensions and the remainder by the chilled water circuiting. Note that a fin density of 6 fins/inch has been selected for the OA coil, while a somewhat higher density (9 fins/inch) has been selected for the RA coil as it operates with a near-dry surface.

	OA	RA
Tube dia. (in)	⅝	⅝
Width (mm)	1800	2400
Height (mm)	457 (12 tubes)	1600 (42 tubes)
Fins/inch	6	9
Rows	6	4
Circuits	18	21
Pass/circuit	4	8

Table 1. Specifications of coils selected for the example.

Using this configuration, plant operation for the week of 4-8 October 1999 was simulated using a synthetic weather sequence. Since the climate is tropical, air conditioning is required throughout the year, and to prevent a build-up of moisture within the building envelope, the plant operates 24 hours/day. With the designated chilled water temperature and flow rate maintained throughout the period, the zone conditions shown in figure 2 were obtained. However, a problem occurs outside working hours because the occupancy and outdoor air temperatures fall and the very dry air off the outdoor air then dominates the cooling. In these circumstances zone relative humidity falls, and for a period in the early morning it is significantly lower than the lower limit of 45% identified as desirable in the Introduction to the paper. Since the occupancy of the building during this period is very low, this may in fact be both acceptable and advantageous as fabrics can be "pre-dehumidified" to provide a buffer when staff arrive en masse. This early morning flushing will also prevent the growth of dust mites and allergen products. However, should this not be required, plant relative humidity can be constrained to a much narrower band by the use of scheduling. In the present case, the following schedule was implemented:

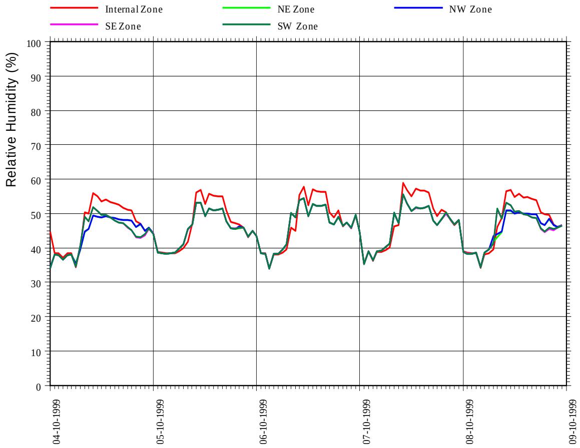

Figure 2. Zone conditions for the example with chilled water temperature and flow rate constant throughout the test period.

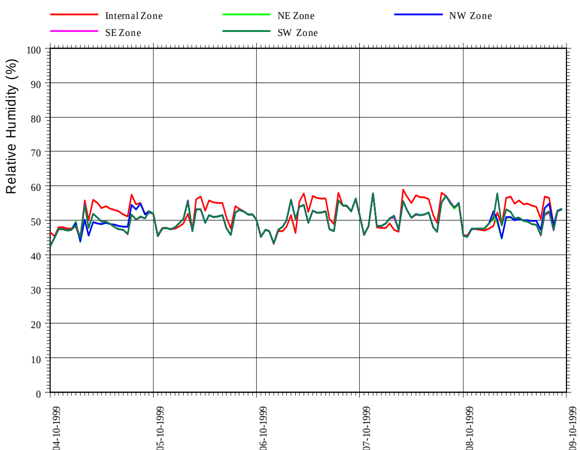

Figure 3. Zone conditions for the example with scheduling implemented as described in the text.

- Chilled water temperature was increased to 9° between 18:15 and 08:15.
- Chilled water flow rate was decreased to 3 L/s between 18:15 and 1:15 and to 2 L/s between 1:15 and 7:45.

In both cases energy consumption is reduced. The consequences for zone RH are as shown in figure 3. This schedule is perhaps not optimal, but does go a considerable way towards assisting the designer to meet the objectives of the design brief. The subject of scheduling will be taken up again in the Conclusions.

ZEBRA and its associated components provide the user with an extensive range of information relating to plant performance and zone conditions. Additional examples are provided by figure 4 which shows the temperature of the chilled water entering and leaving the OA and RA coils, and figure 5 which shows fan efficiency. The ABB Type CYD 490 fan

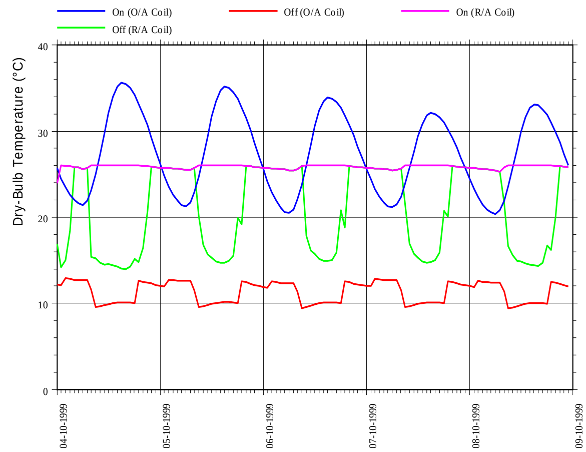

Figure 4. Dry-bulb temperature of air entering and leaving the outdoor air and return air coils for the example.

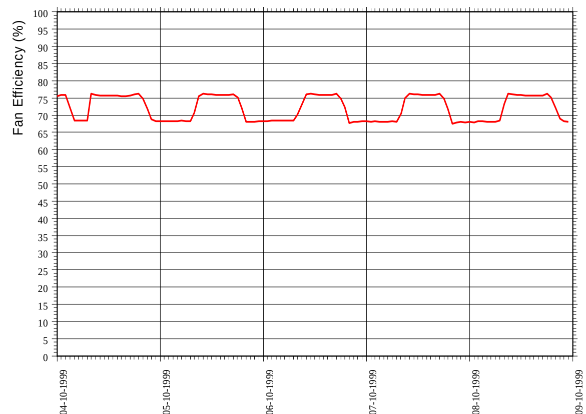

Figure 5. Fan efficiency obtained using an ABB Type 450 fan.

was selected from the ZEBRA database via a "smart" dialogue box which permits the designer to "test drive" candidate fans over their projected range of operation.

CONCLUSIONS.

The authors have demonstrated how an air conditioning plant simulation programme (ZEBRA) can be used in conjunction with a load calculation programme to explore the consequences of equipment selection and of control strategies for system performance ***across the entire operating range***, with a view to optimising system performance. The approach has been illustrated by an example located in the tropics, which demonstrates the desirability of implementing an additional control strategy (in this case scheduling). It is emphasised that part-load operation is liable to cause problems for all types of air conditioning system. For conventional systems these are associated with loss of dehumidifying potential; for HDP systems excessive dehumidification at part load is the problem. With the HDP the extra degree of

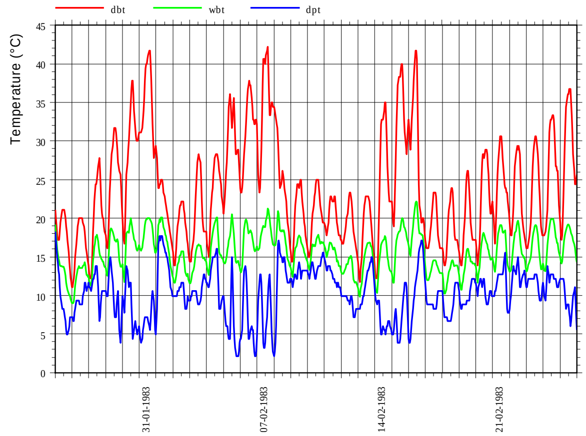

Figure 6. Excerpt from the Adelaide weather record for January-February 1983.

freedom available allows the designer to improve part-load performance ***and*** at the same time to reduce energy costs. Traditional solutions to part-load problems with conventional systems necessarily incur an energy penalty. In some cases this may far exceed the base cost of running the system. The practice of staging by de-activating segments of the coil at part-load provides a partial solution, but is less effective and less robust than is the extra degree of freedom available in an HDP system.

Tropical climates are characterised by small annual and diurnal temperature ranges and this permits a fixed schedule to be applied with a high degree of confidence. By way of contrast, the reader's attention is drawn to figure 6, which is taken from the Adelaide weather records for early 1983. This sequence is extreme (some of the state's worst bush-fires were associated with the third set of peaks), but is nevertheless not atypical of South Australian summer weather patterns. The dynamic range is enormous, and designing systems to operate effectively in this environment is a challenge of some magnitude. The authors are currently employing the tools described above to study the use of the afore-mentioned control strategies to optimise plant operation under such conditions. Among extensions to the basic strategies which are considered are the following:

- Development of adaptive scheduling based on forecast weather data, as previously described.
- Generalisation of the practice of staging. What is proposed is that as the chilled water flow through the return air coil reduces, the flow through the outdoor air coil would be incrementally throttled to maintain a load on the return air coil.

As a final note, the added comfort provided inherently by an HDP system designed according to the methods outlined above is purchased at a price that is ***lower*** than that for the conventional, inherently uncomfortable system, and substantially lower than the reheat systems.

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