

AIR CONDITIONING CONTROL SYTEM FOR VARIABLE EVAPORATOR TEMPERATURE

This invention relates to air conditioning improvements.

BACKGROUND OF THE INVENTION

An object of this invention is to assist in reducing energy required to
5 effect air conditioning while obtaining high performance standards over
the full operating range. Environment control by air conditioning means
involves more than simply increase or decrease of the total capacity, but
rather, dependent on the relative values of latent heat which are to be
simultaneously offset in a manner that the lag in the process of mass
10 transfer behind heat transfer does not contribute an energy penalty.

The problem to which this invention is directed is the problem of how to
reduce energy required to effect acceptable refrigerant air conditioning
for comfort purposes. The system presented here resolves the chaos
arising from the contradictory effects of numerous variables in infinite
15 combinations which are involved in refrigerated air conditioning design.
Two of these unrelated variables, sensible and latent heat appear in both
the room treated air and the outside air introduced for ventilation.
Sensible cooling and dehumidification pass through unrelated
processes. Heat transfer can occur with only a dry bulb temperature
20 difference between the air and the cooling coil whereas mass transfer will
not proceed at all without contact with a cooling surface which is below
that of the dew point temperature of the moist air. To put it in terms of the
partial pressure of the water vapour in the treated air, dehumidification
from the treated air will not proceed until there is a partial pressure
25 difference with the wetted interface of the cooling surface.

PRIOR ART

Closest prior art known to the Applicant are publications of Australian
Patents entitled "Method of Air Conditioning" AU-B-49875/79 (597757),

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"Air Conditioner and Method of Dehumidifier Control" AU-B-81946/87 (662336), and "Air Conditioning for Humid Climates" AU-B-18873/92.

BRIEF SUMMARY OF THE INVENTION

In one form the invention can be said to reside in an air conditioning system including refrigeration means to effect a highest evaporator temperature for its refrigeration cycle when it treats either directly or indirectly cooling and dehumidifying of air to be conditioned, dry bulb temperature measuring means to measure the temperature of air within a space to be conditioned, humidity measuring means to measure humidity of air within the space to be conditioned, and control means adapted to offset both sensible and latent heat loads of the air at each of the climatic and interior building load conditions over the operating range of the air conditioned system, and control means to establish humidity performance in the treated space at a high end of an acceptable range, and, if a chiller is employed, at the high end of the acceptable coolant coil velocity performance range, where the higher of these two properties reach its set point value, in combination with the sensible heat and latent heat loads being offset by the refrigeration cycle, will be generating the value of an evaporator temperature set point which will be as high as reasonably feasible to minimise input energy to the refrigeration compressor and maintain high standard performance conditions for the particular operating condition in the range and on a change of load, will automatically vary the evaporator temperature set point so that it is compatible with each of all the sensible and latent heat loads occurring over the operating range of the system, the control means employed would at each operating condition in the range determine the specific volume at the saturated suction temperature of the refrigerant leaving the evaporator to establish the higher of the said two constant space humidity or coil coolant velocity set points and then calculate the evaporator set point which in the case of this embodiment drives the variable speed control of the compressor motor.

In an alternative description there is proposed method of capacity control of an air conditioning system including a compressor and evaporator in a refrigeration cycle, heat exchanger means, coolant flow conduits, control



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equipment including compressor, flow control means, proportional plus
integral control output equipment and final control means said method
including generating an evaporator pressure/temperature set point for the
evaporator as high as reasonably feasible for each of the operating internal
5 and external heat and moisture load conditions occurring over a total air
conditioning range.

In a further alternative description there is proposed a refrigerant air
conditioning arrangement of a type including refrigeration means to provide a
cooling evaporator either directly for air to be conditioned or indirectly for the
10 air to be conditioned through a chiller, dry bulb temperature measuring means
to measure the temperature of air within a space being air conditioned, control
means connected to the dry bulb temperature measuring means effecting
control of the refrigerant flow and/or coolant flow through the area of cooling
surface available to the air being cooled, further characterised in that there are
15 included humidity measuring means to measure humidity of air within the
space being air conditioned, and further control means to effect a change in
the evaporator temperature in response to the treated space humidity
measuring means or the coolant velocity through the coolant coils interfacing
with the treated air means to maintain a refrigerant evaporator temperature
20 which is as high as reasonably feasible for each of the operating internal and
external heat and moisture load conditions occurring over a total air
conditioning range.

This invention is applicable to both constant air volume and variable air
volume refrigerant air conditioning systems, and enables higher
25 efficiencies in such air conditioning systems over that which is currently
the case.

This is achieved, by setting control means to use a highest feasible
evaporator temperature specific for each operating condition in the range
within the bounds of good performance standards including fluid flow,
30 heat and mass transfer principles as a priority.

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In preference the method includes utilising a control system to effect humidity control and coolant velocity control to use set points representing a design space humidity or design coolant velocity through the tubes of the coils, whichever is the greater, with reference to their
5 respective set points. Thereby this control system will establish a highest acceptable evaporator temperature set point for that particular operating condition in the air conditioning range.

In the absence of chillers, when the evaporator interfaces directly with the treated airstream there is only a humidity set point required. However
10 when a humidity of air in the space to be conditioned does not reach its set point, as may be the case during low humid outside air conditions, in lieu of a secondary coolant velocity set point, a minimum speed setting of the compressor determined by the compressor manufacturer can be supplied within a variable speed control device as is employed in the
15 embodiment of this invention.

In explaining this invention I observe that there is complex incompatibility in relationships between heat and mass transfer, limitations applied to maintain turbulent coolant flow performance at the heat exchangers and the problems arising when a ratio of internal
20 sensible heat loads to internal total heat loads present a design problem. Reducing total heat loads are not necessarily associated with reducing volume flow rate at the suction of the compressor and increasing total heat loads are not necessarily associated with increasing the volume flow rate at the suction of the compressor.

25 According to this invention there can now be provided a system of capacity control of refrigerant air conditioning which will control whether a volume flow rate of the refrigerant rises or falls on a change in detected climatic and internal heat and moisture conditions. Should it fall, input energy is reduced and treated space humidity or coolant velocity is
30 increased. According to this invention there is established a lowest

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volume flow rate to maintain required preselected conditioned air characteristics.

- Factors which can pose problems in capacity control include lag between simultaneous dehumidification and cooling processes, and
- 5 instability arising from coolant flow which has fallen from turbulent to transition flow when a total load falls to about 30% to 50% below peak cooling loads. There is also an undesirable effect of high coolant flow rising above known standards. The invention proposes having a control priority of achieving a highest evaporator temperature feasible for
- 10 reduced input energy while overcoming these limitations.

- The capacity control system in accord with this invention is compatible with numerous conventional arrangements. Through the use of embodiments employing an open reciprocating compressor with variable speed control equipment, the system has been demonstrated for both a
- 15 constant air volume and variable air volume system. In addition to speed control it is compatible with compressors employing cylinder unloading and rotary screw compressors equipped with sliding valves. In the case of centrifugal compressors it is compatible with pre-rotating vanes and also multiple flow temperature systems.

- 20 Use of systems dependant solely on throttling the refrigerant as the capacity control means are not preferred nor hot gas bypass because of their implicit inefficiency but can in the wider concept of this invention be used.

- Relevant to the degree of energy saved is the arrangement of the coils
- 25 and their circuiting. In one arrangement recommended and used in an embodiment of this invention as in Figure 8a and Figure 8c, the outside air coil is employed at what it can do better than the return air coil. It is selected to not only cool and dehumidify the outside air to the supply temperature of treated air that is introduced into the air conditioned space
- 30 but to have the capacity to dehumidify the air to offset all or a large part of the internal latent heat loads when these are high, and also part of the

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internal sensible heat loads. This is achieved by employing all or most of the coolant required during the peak simultaneous load to flow first through the outside air coil before flowing on to serve the return air coils.

In this manner the temperature rise of the coolant through the outside air coil is drastically reduced and the mass transfer at the wetted interface surfaces of this outside air coil during periods of high moisture content in the air is as high as feasible. In this manner I create a coolant flow which approaches a sink temperature as indicated in Figure 1(a) in contrast to a conventional outside air coil in the case as indicated in Figure 1(b). As a consequence in the case of Figure 1a, the low dew point temperature of outside air leaving its coil, on mixing with the return air leaving its coil results in a reduced dew point temperature of the supply air to the treated spaces. The return air coil is smaller since the outside air coil handles part of the room sensible heat load as well as part of or all of the latent heat load.

By this arrangement not only is the lag of mass transfer behind heat transfer overcome, but the reduction in treated space humidity permits the evaporator temperature to increase to a higher value which is limited only by design maximum set points for space humidity or a design maximum set point for secondary coolant flow, whichever is the greater. According to the capacity control system in accord with this invention, the output from this set point determines the evaporator temperature set point.

There are two major factors of significance namely, the capacity control system and the arrangement of the coils and circuits which link the capacity control system to the cooling and dehumidifying coils. However other arrangements may also be admissible. A conventional temperate climate arrangement which is also recommended particularly when the outside air percent of supply air is low, is to simply mix the outside air with return air and then passing it through a combined cooling coil as indicated in Figure 8b and 8d.

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The arrangement described above where a separate outside air coil and separate return air coil is used can however be misapplied. If the outside air consists of 20% of the supply air, in order to offset 1 gram of moisture per kg. of dry air in the treated space, the outside air would have to

5 condense 5 grams of moisture per kg of dry air. If the outside air consisted of only 10% of the supply air in order to reduce 1 gram of moisture per kg of dry air, 10 grams of moisture would have had to be condensed from the outside air.

In lieu of the use of the outside air and return air 2 coil arrangement

10 described in Table 1 for peak load condition on row 4a, when the combined outside air and return air flow rates are employed in a single coil arrangement as in Figure 8b, having the same geometry and depth of row equivalent to Figure 8a the "off" condition is, in significant figures the same as the supply air condition in the 2 coil arrangement described

15 above.

During dry outside air conditions, even with high outside air temperatures the system according to this invention without any additional equipment takes advantage of the low outside air dew point temperatures. On

20 mixing with the return air the internal latent heat loads in whole or in part are offset and consequently reduce the use of energy to offset the internal latent heat loads. The high return air coil coolant temperature prevents or reduces dehumidification by the refrigeration system. This is further described herein under the heading of "humidity economy cycle".

The contribution derived from this invention can best be revealed when

25 the capacity control system of this invention is assessed at the time the system is at a balance point. At such times in the absence of this invention, the humidity within the treated space may be at an acceptable humidity below the maximum acceptable design set point in the humidity range and below the maximum acceptable design set point in the

30 secondary coolant velocity range. Since neither of these two controlled values is at its respective maximum set point the evaporator



temperature/pressure is not at its preferred design setting and the control system in this embodiment of the invention increases the evaporator temperature by raising the higher of these two values to its respective control set point. This will not upset the balance point except favourably in that there may be a rise in the humidity of the treated space to a higher humidity design level and thus a reduction in the energy necessary to dehumidify the treated space. Apart from this reduction due to a higher humidity design level, the mass flow rate at the suction of the compressor at the lower evaporator temperature and the mass flow rate at the suction of the compressor when at its higher design evaporator temperature for its particular condition would be close to equal. It is only the volume flow rate entering the suction of the compressor when it is at a lower evaporator temperature which will be greater than the volume flow rate entering the suction of compressor when it is at the higher evaporator temperature. Thus this capacity control system is not invasive to the traditional control methods such as regulating the flow rate of the coolant when the dry bulb temperature of a treated space rises or falls. It simply regulates the position of the evaporator temperature to be at a feasible maximum. (It is assumed that the coolant velocity set point is below the humidity set point in this particular example).

In the case when the treated air transfers heat and moisture directly toward the evaporator, the compressor speed must be above the minimum permissible for the particular compressor employed. A minimum speed setting is supplied within the Variable Speed Control device. There is no fixed coil which has a fixed supply coolant temperature that can service a complete range of say from peak to 35% of peak load by establishing the highest evaporator set point.

Consider the peak load indicated in Table 1 where a full range in 12 steps from peak to min is considered for a CAV system. The coolant supply temperature is 7°C. The peak refrigerant load is equal to top row labelled 4 rows (4a) in Table 1. This value, the sum of total output sensible and latent heat loads, is 16.51 kW + 9.13 kW = 25.64 kW and

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employs a 4 row deep coil. Now consider the 2 row deep coil labelled (2d) which has the identical face area and circuit lengths. The coolant supply temperature is 7°C. The total output sensible and latent heat load is 7.64 kW + 6.75 kW = 14.39 kW. Thus the 2 row deep coil, precisely half
 5 of the 4 row deep coil, has a part load percentage of $(14.39 / 25.64) \times 100 = 56.12\%$ of peak. It has a water velocity of 0.21 m/s. Clearly if this had been run as a 4 row deep coil the water velocity would have been well below 0.2 m/s in unstable transition flow mode.

Similarly consider the fixed 4 row-deep coils of Table 5. Here the peak
 10 load is indicated in Table 5, where a range in 20 steps from a to s plus row O' is considered for a Variable Air Volume system. The peak load, top row "a" has a coolant supply temperature of 7°C. The peak load refrigeration load is equal to 28.1 kW. Now consider the same 4 row deep coil labelled Table 5 row O'. It too has a coolant supply
 15 temperature of 7°C. Its total refrigeration load is equal to 13.8 kW. This is $(13.8/28.1)(100) = 49.1\%$ of peak. Its coolant velocity is far below the acceptable water velocity of 0.20 m/s. It is equal to 0.1 m/s and is no longer in turbulent flow and is unacceptable. Now compare the condition of row O with the very similar but unsatisfactory performing condition of
 20 row O', at supply temperature of 7°C. The major difference is that row O has a coolant supply temperature of 9°C and water velocity is acceptable at 0.2 m/s. It should be noted that the only reason the 4 row coil performance of rows p, q and r has a sufficiently high return air coil water velocity is that similar to row o, the capacity control system of the
 25 invention had increased the chilled water supply temperature from 7°C. Otherwise, like the performance of row o', the system would have performed unstably at water velocities well below 0.2 m/s.

Table 5, row s performance is a very special case involving an outside air
 30 condition which has a dew point temperature below that of the dew point temperature of the supply air. This will be discussed when the new "Humidity Economy Cycle" of this system will be explained in conjunction with Tables 3 and 4.

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BRIEF DESCRIPTION OF THE PREFERRED EMBODIMENTS

Examples of the invention are described hereunder in some detail with reference to, and are illustrated in, the accompanying drawings and charts wherein:

- 5 Fig 1a refers to the performance of the outside air coil in the arrangement of separate outside air and return air coils discussed in detail above. It is of advantage where a significant outside air percent of the supply air of about 15% or more is required. Otherwise a conventional combined outside air plus return air coil should be
10 considered. Fig 1b represents a conventional outside air coil performance where the coolant does not flow on to the return air coil;

- Fig 2 shows this pressure/enthalpy diagram to demonstrate the higher evaporator temperature/pressure that can be achieved through this invention over numerous conditions over its operating range and
15 thus substantially reduces the enthalpy difference across the compressor;

- Fig 3 is a compressor horsepower selector graph on which is compared a higher suction temperature, 10.0°C (50°F) with a lower suction temperature, -1.1°C (30°F) when the output refrigeration capacity is 35.2 kW (10 tons of refrigeration) or 120,000 Btu/Hr to indicate that
20 there is a substantial reduction in the brake horsepower required by the higher suction temperature cycle. The higher suction temperature cycle requires only 9.1 horsepower, 6.8 kW, whereas the lower suction temperature cycle requires 11.4 horsepower, 8.5 kW, to deliver the same output refrigeration capacity of 35.2 kW;

- 25 Fig 4 is a graph of compressor performance on which is related the higher suction temperature of Fig 3 to a 616 RPM of the compressor and lower suction temperature of Fig 3 to a 914 RPM of the compressor when an open reciprocating compressor of this embodiment is compared for the same output refrigeration capacity of 35.2 kW (10 tons of refrigeration)
30 or 120,000 Btu/Hr;



Fig 5 is a performance diagram derived from Fig 4. It points to the means by which the control system automatically selects the highest admissible evaporator temperature through regulating the RPM of the open compressor in this embodiment of the invention. It is through this means that both reduced energy input is assured while maintaining the treated space conditions to high engineering performance standards. Fig 5 will be discussed in detail under section describing the control system;

Fig 6 shows the controllers Set Point Diagram which constrains the evaporator temperature to either a maximum acceptable Humidity Set Point or maximum Coolant Velocity Set Point, whichever is the higher of the two values and thus achieves reduced energy input to the refrigeration plant by establishing the highest evaporator set point while maintaining high standards of performance;

Fig 7a represents a block diagram of the Control System of this invention when the evaporator coil interfaces directly with the airstream.

Fig 7b is a block diagram of the invention's Control System linked to a constant air volume system employing a chiller. The range of the system from peak to minimum heat load performance is wide requiring 3 stages of active coil sizes so that they are compatible with the sizes of the heat and moisture loads;

Fig 7c is a block diagram of the inventions Control System linked to a variable air volume system employing a chiller. Cooling coil sizes are also indicated for a very wide range in Fig 7c. Table 5 only indicates data for a fixed 4 row deep coil that has a low range, requiring only a fixed return air cooling coil;

Fig 8 are psychometric chart sketches displaying the air flow paths of two arrangements;

Figs 8a and 8c represent the 2 coil arrangement described in the embodiment of this invention. Figs 8b and 8d represent another



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acceptable arrangement wherein a single combined cooling coil of outside air and return air is employed;

5 Figs 9a and 9b present Weather Bureau Data for Singapore and Adelaide respectively, marked by curves establishing ambient design loci for dry bulb day time temperatures. They are displayed together to indicate two very differing climates in their moisture content which can be handled by the same air conditioning design system of this invention when the overall total refrigerating capacity is the same; and

10 Fig 10 is a diagrammatic representation of a variable speed control module.

The term reasonably feasible is utilised herein with respect to air-conditioning performance standards. This invention has been written on the assumption that the standards of the American Society of Heating, Refrigerating and Air conditioning Engineers Inc, ASHRAE, will be
15 observed and also the following limits apply. The control system allows designers to select their preferred operating conditions within the range provided by these standards.

	Relative Humidity Maximum	60%
	Relative Humidity Minimum	30%
20	Cooling Coil Chilled Water Velocity Maximum	2.7 m/s
	Cooling Coil Chilled Water Velocity Minimum	0.15 m/s
	Ventilation IAQ (depends on various factors)	ASHRAE 62-1989
	Coolant Supply Temperature	7°C to 17°C
	Evaporator Temperature	-1.1°C to 15°C
25	Air Pressure Drop Through Cooling Coils Maximum	200 Pa
	Air Face Velocity Maximum	3 m/s
	Chilled Water Pressure Drop Maximum	60 kPa
	Compressor speed minimum and maximum variable, determined by manufacturer.	

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In the event that these limitations are not to control the equipment parameters as for example when relative humidity may be permitted to rise to 70% of the energy savings are greater than if the upper limit of 60% is observed.

- 5 In this embodiment of the invention the design is achieved by regulating the compressor speed to obtain the highest evaporator temperature that is compatible for each specific condition of heat and moisture load over the full operating range. This embodiment of the invention employs an open positive displacement reciprocating compressor with variable
10 speed control (VSC) to relate to performance at all outside air and internal heat and moisture load conditions. See Figure 10. The cost and size of modern variable speed equipment is competitive in this application.

- The important influence in obtaining maximum energy conservation
15 through this system occurs when the control system acts to establish a high evaporator temperature as for example by means of regulating the compressor RPM. The pressure-enthalpy diagram of Fig 2 relates to a standard refrigeration cycle. When the evaporator temperature/pressure is raised the performance of the cycle moves up from the refrigeration
20 effect of $h_1 - h_4$ to $h_1' - h_4'$. This relates to the output refrigeration capacity required to offset the heat and moisture loads.

- The energy input referred to achieve this is the work of the compressor. When the evaporator temperature/pressure is raised the work of the compressor per unit mass of refrigeration through compressor is reduced
25 by the difference between the value of $(h_2 - h_1) - (h_2' - h_1')$.

- Under conditions of a constant compressor RPM a fall in load would send a signal to the expansion valve of Fig 2 to throttle the refrigerant to a lower pressure. One example is indicated in Fig 5 where condition A moves to B, resulting in an excess input refrigeration capacity for this
30 refrigeration requirement. This path is precisely opposite to the purpose



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of this invention, wherein energy is saved through the increase of the evaporator temperature and the decrease in the compressor pressure differential when A moves to C. This is best understood through the term COP (coefficient of performance) for this ratio, refrigeration effect is the

5 numerator and the enthalpy difference across the compressor is the denominator:

COP =

$$\frac{\text{Output Refrigeration Capacity}}{\text{Input Refrigeration Capacity}} = \frac{\text{Useful Cooling \& Dehumidifying}}{\text{The Work Necessary to Achieve this}}$$

- 10 Often there is a failure when evaporator temperatures and coolant temperatures are not controlled to be compatible during many variable simultaneous conditions arising over the climatic range. In this embodiment the problem is resolved.

THE CAPACITY CONTROL SYSTEM

- 15 Reference is now made to the control system, illustrated in Figs 6, and 7a, b, c. The benefit of energy savings is accessible to each combination of heat load occurring over the full operating range according to the control system described below:

- 20 The control system comprises the essential components of room humidity control and coolant velocity control, each generating an evaporator pressure set point (that is, a point beyond which evaporator pressure will not increase).

- 25 The individual control outputs are proportional plus integral that continuously readjusts the output value to obtain either of the two input signals at a specified maximum value.

The higher of the two output values at any instant is the required set point to achieve minimum energy input to the refrigeration plant.

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Fig 6 is a Control Vs Set point Diagram and Fig 7 is a Block Diagram of Control System.

Fig 7a represents the simple case where the evaporator interfaces directly with the treated air. Fig 7b describes a system employing a
5 chiller in association with a constant air volume system. The return air coil has 3 active stages. Fig 7c describes a system employing a chiller in association with a variable volume system. Fig 7b relates to the performance reported in Table 1, Fig 7c to the performance in Table 5.

Through this control system the means are imparted to automatically
10 select whether the evaporator temperature should rise or fall.

In Figs 6, 7a and 7b a humidity sensor 10 is located within a treated space, and a coolant velocity sensor 11 in a coolant line. Each is associated with a respective set point 12a and 12b which imparts limits on the values as described above. They are then fed to a discriminating
15 device 13 which is responsive only to the higher of the two signals, and adjusts the evaporator pressure 14 or chiller control 14 in this embodiment to provide an evaporator pressure which maintains the higher of the two set points at its maximum set point condition. This constitutes a proportional control plus integral action. The sensors may
20 be electrical, pneumatic or hydraulic, or any combination thereof. For example, if electric the device 13 may need to be only diodes. If pneumatic, device 13 may be, for example, a Honeywell selector relay RD405. The arrangement is essentially of mathematical functions, and need not comprise separate discrete devices. A microprocessor may be
25 used to achieve the same functions. A room thermostat 17 with valves MV17a, b and c for Constant Air Volume System, (Fig 7b) and damper 18 plus a supply air thermostat 19 associated with valves MV17a, b and c are indicated for Variable Air Volume system (Fig 7c).

In comparing energy requirements for different COP values, the percent
30 of greater input energy required by the lower COP value over that of the

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higher COP value when relating to the same output refrigeration capacity is determined as follows:

If the lower COP = 2 and higher COP = 3

$$\begin{array}{lcl} \text{per cent of additional} & = & 100 \frac{(\text{COP}=3-\text{COP}=2)}{\text{COP}=2} = 100 \frac{(3-2)}{2} = \\ 5 \text{ energy required by COP}=2 & & \end{array}$$

50% more input energy for COP2

A higher evaporator temperature reduces heat transfer across the heat exchangers. In the case where there is a chiller supplying a cooling coil interfacing with the treated air there is a corresponding rise in the coolant temperature which is offset by conventional controls. A conventional modulating valve opens to increase the coolant flow through the cooling coil, thus maintaining the treated space dry bulb temperature at its design setting. The higher evaporator temperature causes a rise of humidity in the treated space. The control system described above and illustrated in Figs 6, 7a, 7b and 7c prevents it from rising above a humidity design set point and a coolant velocity set point. In this manner the control system provides the maximum COP and the least Input Refrigeration Capacity compatible with performances at good engineering standards.

In the case of a direct expansion air conditioning system where the evaporator interfaces directly with the treated air in lieu of the secondary coolant velocity set point a minimum speed control set point is supplied within the Variable Speed Control Device.

When there is a fall in output refrigeration capacity there is not necessarily a fall in compressor speed and a rise in evaporator temperature as is indicated in Fig 5 in the change from point A to C, though this direction is preferred from the point of reduction in input energy. For example, a fall in output refrigeration capacity from set point E at 670 RPM may find the control system signalling for a rise in the compressor RPM to 700 RPM and a reduction in evaporator temperature



as is indicated in a change to point C, or it may signal for a fall in the compressor RPM and a rise in evaporator temperature to point F at 616 RPM. Similarly a rise in output refrigeration capacity may require either a fall or rise in the compressor RPM as from point C to E or C to D in Fig 5.

- 5 The system according to one or more embodiment is to respond to the simultaneous needs for heat and mass transfer and automatically select a correct choice of compressor speed to offset both the sensible and latent heat loads at each condition over the full operating range and to do so at minimum energy input.
- 10 For a given load a rise in evaporator temperature is indicative of a condition wherein both the space humidity and the coolant velocity are below their design set points. A fall in the evaporator temperature is a response to a signal to either increase dehumidification or decrease coolant velocity.
- 15 This system of air conditioning is compatible with other compressor systems as well as the system of compressor speed adjustment as described in this embodiment selected to demonstrate this invention. See Figure 10 for the diagram of the variable speed control module.

These other systems include:

- 20 (a) Rotary Screw Compressors equipped with sliding valves for capacity control. These have large capacity coverage but there is some loss of efficiency in capacity reduction.
- (b) Open centrifugal machine having automatic speed control from steam and gas turbines. Control of surging is a factor here.
- 25 (c) Centrifugal Compressors with pre-rotating vanes.
- (d) Multiple flow temperature systems are also to be considered.

Staging and flash gas removal are compatible with the system.

An objective is efficient compressors exhibiting high coefficients of performance wherein the evaporator temperatures can be maintained as



high as humidity and coolant flow conditions permit within the treated space. To meet the system conditions the evaporator temperature is a variable and the temperature of the coolant chilled water or brine, etc is also a variable as demonstrated in the embodiment of this invention, as illustrated in Tables 1 and 5. Reference is also made to a section hereunder on zoning which relates to the coolant temperature variations.

In the embodiments presented here in Table 1 and Table 5 a single peak size range of approximately 28 kW is presented. However any size range can be developed having the same performance characteristics. Thus a consulting engineer or a manufacturer can develop a complete line of coverage for all size ranges.

For example employing the range selected here as a datum, the range can be doubled by simply doubling the width of the heat exchangers and doubling the number of circuits to maintain the same coolant velocity range and coolant pressure drop range. Obviously the treated air face area has also been doubled maintaining again the same face velocity and the same air pressure drop. Similarly the number of vertically spaced tubes could be doubled and again number of circuits doubled thus in combination with the doubled width the datum size range has been quadrupled. By similarly halving width, halving tube height of the datum size results in a total of eight size ranges from minimum to maximum. Thus a universal system covering all desired sizes can be formed.

In a system which is sensitively responding to room sensible and latent heat loads to reduce energy input it is useful to consider the matter of zoning in applications such as high rise buildings. In most buildings simultaneous variations in Room Sensible Heat Ratio are not very large. In these cases small differences can readily be handled by selecting the maximum humidity set point of the control system about 2% lower in terms of relative humidity than would be necessary for a single zone system. For example a set point of 58% RH(relative humidity) would



prevent conditions which would allow a rise in its RH to 60% in the area where the RH is not sensed. An examination of Table 5 indicates that most of the time the Control System high coolant velocity set point prevents the zone to reach 60% RH. A careful examination of zoning for this system and the employment of multi-chiller systems can lead to the benefits of this invention in high rise building applications. Table 5 indicates the employment of VAV system in a multizone situation as described above where the variation of treated space humidity is minor.

An additional means for applying this system to multizone buildings is to sense the dew point temperature or the Relative Humidity on Equipment by the main control system in all zones and applying the lowest design humidity set point reading to affect the control of evaporator temperature. Thus with only a minor loss of energy savings, a considerable advantage in system energy savings remain.

15 A HUMIDITY ECONOMY CYCLE

The conditions encountered by the cooling and dehumidifying coils vary very much over a climatic range. There are numerous occasions where energy can be reduced by preventing dehumidification by the refrigeration system.

20 For designers identifying economy cycles as associated with reducing sensible heat loads it may appear unusual to find that through the invention an economy cycle can occur at outside air conditions as high as 35°C dry bulb temperature. Furthermore no special requirement or control is necessary. It occurs automatically in this system to result in a large reduction of both output and input refrigeration capacity by offsetting all or part of the treated space latent heat loads.

A "humidity economy cycle" can be a part of the system during non-humid weather when the outside air is below the dew point of the supply air, the inlet and outlet refrigeration capacities can be reduced well below present practice. Conventional design pays little attention to energy



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conservation during dry weather even though it is a condition that occurs frequently in many climates. The coolant temperature is determined in accord with this invention by the capacity control system which selects the value of the coolant velocity or the room humidity to obtain an output set point value. This results in a coolant velocity temperature that is as high as feasible for most efficient performance. During non-humid weather high coolant temperatures exceeding the dew point temperature of the supply air to the treated spaces will often occur. Consequently dehumidification from the supply air by the refrigeration system will be prevented. Thus the latent heat loads of the treated spaces would be offset at no energy cost since under such conditions the cooling coils are dry. Tables 3 and 4 indicate the case of a zero latent heat load penalty on the refrigeration system. Tables 3 and 4 indicate in detail the performance of the humidity economy cycle during an outside air dry bulb temperature of 30°C. Table 3 details performance at an average room latent heat load bearing a Room Sensible Heat Ratio of 0.85 and Table 4, at a very high room latent load having a Room Sensible Heat Ratio of 0.70. This "Humidity Economy Cycle" is relevant for both the arrangements of this embodiment where a separate return air coil and a separate outside air coil is employed (Fig 8a and 8c) or in the simpler arrangement where the return air is first combined with the outside air to pass through a single coil, (Fig 8b and 8d). The cycle will occur as long as the supply air to the treated space has a dew point temperature equal or below the temperature of the coolant. Tables 3 and 4 high coolant temperatures of 14.2C and 13.0C are indicative of not only a savings of refrigeration capacity to offset room latent heat loads but also to increase the COP of the system and further considerably reduce the input energy to the refrigeration system.

Modifying the system to carry out the invention has the advantage of also providing a means for automatic reduction of output refrigeration capacity:

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The design engineering standards for comfort air conditioning accepts room humidifies over a wide stipulated range. In accord with this invention, it is preferred to select the design humidity set point of the control system at or near the high side of this range. This results in a higher enthalpy value of the air leaving the cooling coil and to a reduced enthalpy difference. For example on a 35°C dry bulb temperature day Table 2B indicates a rise in relative humidity to 58.8% at a chilled water supply temperature of 11°C. Table 2B indicates that when this performance is compared with the 49% room relative humidity room at a chilled water supply of 7°C. there is 10.5% greater output refrigeration capacity required.

In the "The Capacity Control System" reference was made to a horizontal line in Fig 5 which represented a line along which the Capacity Control System locates the balance point for a particular operating condition in the air conditioning range. However where there is a reduction of output refrigeration capacity this line slopes down to the right reflecting the reduction in output energy as the relative humidity within the treated space rises to a higher set point. This fall in the output refrigeration capacity again further reduces the input energy of the compressor.

A further advantage is that the cooling coil can have an increased range of cooling which resolves unstable flow conditions:

This is a further advantage when working to higher evaporator temperature conditions when coolants such as chilled water are employed. As is commonly the case with conventional systems coolant flow rates through coils fall drastically at part load conditions. Numerous tests indicate that in most cases the Reynolds number and water velocity changes from desired turbulent flow to unstable transition flow is at a point of about 30% to 50% below peak load. In the case of this invention the range of the cooling coil size is enlarged. This occurs as a result of the rise of the evaporator temperature which causes a greater demand for the warmer coolant as is indicated above in the section relating to

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reduction of input refrigeration capacity. Accordingly, the range of the coil is enlarged. In Table 2A the part load condition for the return air coil water velocity is 0.35 m/s when the chilled water supply temperature is 7°C. This condition occurs here at 62% of the total peak load when the

5 chilled water velocity is approaching its minimum 0.2 m/s value. For the identical system part load condition, when the chilled water supply temperature is 11.2°C, the chilled water velocity is 2.26 m/s. The invention enables performance to be tailored for each operating condition with each design having broad climatic coverage. There are

10 further advantages in addition to energy conservation. In the process of maintaining the highest evaporator temperature for each individual operating condition it overcomes contradictions arising from many unrelated variables. The compressor is compatible with its refrigeration cycle. The variation of size of the heat loads is compatible with the

15 variation in compressor RPM. A new sensitivity to each specific operating condition is gained.

The above advantages have been found to be extendable to cover different climatic areas using the same design. A VAV system presented hereunder which covers many different climatic ranges having widely

20 different room sensible heat ratios. Humid, tropical Singapore and hot, dry Adelaide can be served by the identical selection. The difference becomes evident when reference is made to psychometric charts displaying for Singapore and Adelaide, Fig 9 conditions for ambient design loci exceeding 10% of operating hours for day time. An

25 examination of the respective humidity ratios for these areas reveal how their design conditions are very dissimilar. The humidity ratio for Singapore design locus is between 21 and 22 g/kg. The humidity ratio for Adelaide is between 10 and 12 g/kg/. The broad coverage of this design is such that these two very dissimilar climates are efficiently

30 served by the identical selection without any energy or performance penalty as long as both systems are compatible with the range of sizes of the heat loads.

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In Table 5, row b represents peak Singapore condition and row e represents peak Adelaide condition.

The energy conserved through this invention involves consideration of performance of a system over its full operating range. To meet this purpose, numerous representative conditions for tropical and temperate climates have been studied. There is no single answer to how much energy is saved since it involves comparisons with numerous conventional systems having different energy and performance characteristics. Furthermore each climate has different lengths of peak and part load needs. For example a dry climate would have the enormous advantage of the humidity economy cycle described above whereas a tropical climate would have a cooling season that covers the full year of which 25% of the time it is below 27°C dry bulb temperature when the coolant supply temperature would rise from 7°C to above 10°C when this invention is applied.

Data for open type reciprocating compressors are used hereunder to assist in an appraisal of the invention. This is taken from a reliable publication, the "Terry Engineering Manual for Belt Driven Equipment". Reference is made to two Terry performance graphs drawn for R22 Refrigerant, Fig 4, (Terry Fig 15E) and Fig 3, (Terry Fig 21). Comparison is made with the performance at an evaporator temperature of -1.1°C (30°F) with that of an evaporator temperature of 10°C (50°F) when in both cases they serve the same Output Refrigeration Capacity of 35.2kW, (120,000 Btu/Hr or 10 tons of Refrigeration. The condensing temperature will be assumed as constant at 43.3 C (110 F). From Fig 4 (Terry Fig 15E), follow the arrow from the upper set of curves to the lower set ending at the value of 35.2kW, (120,000 Btu/Hr). The RPM of the compressor at 10 C (50 F) evaporator temperature is 616 and for -1.1 C (30 F) run, it is 914. This variation in RPM is within the manufacturer's approved limits. Fig 3 (Terry Fig 21), will now indicate the input energy that the two runs required to satisfy the Output Refrigeration Capacity of 35.2kW, (120,000

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- Btu/Hr or as listed in the Figure, 10 tons of refrigeration). Enter the upper graph ordinate at 10 C (50 F) and -1.1 C (30 F) and move to the 43.3 C (110 F) curve condensing temperature line. Move down to the lower graph Output Refrigeration Capacity value 35.2kW (10 tons of Refrigerant
- 5 is what is listed in the graph). Now continue to the right ordinate where "Brake Horsepower Required" is indicated.

- The 10 C (50 F) evaporator temperature run has an input power requirement of 6.7kW (9BHP) and a COP = 5.24, the -1.1 C (30 F) evaporator temperature run has an input power requirement of 8.5kW,
- 10 (11.4 BHP) and a COP = 4.14.

The energy Input Refrigeration Capacity for the -1.1 C (30 F) run is 26.7% greater than the 10 C (50 F) run when handling the same heat loads. Few designers are aware of the enormous energy savings achieved through higher evaporator temperature.

- 15 The above results can also be indicated on the capacity versus evaporator temperature plot of Fig 5 which is derived from Fig 4. Consider a conventional hermetic reciprocating compressor operating at a fixed RPM of 914 serving a higher heat load at some point A of Fig 5. On a fall of load to 35.2kW the action of the expansion valve would be to
- 20 throttle the refrigerant down the 914 RPM condensing unit line to point B, rather than in the case of this system where the control system would select the highest evaporator temperature which would offset the sensible and latent heat loads occurring at that instant, point C in this example, at 616 RPM and 10 C (50 F) unless one of the two capacity
- 25 control set points are reached earlier.

SUMMARY OF ADVANTAGES

Tables 1 and 5 (CAV and VAV systems) indicate that in some prior art it is common for unstable flow conditions of coolant to exist at low flow rates.

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It is common for air conditioning design to fail to determine the details of performance over a range. These details have been partially displayed in Table 1, for a CAV system, and Table 5, for a VAV system. It is known that unstable performance can occur as discussed in the sections above

5 regarding part load performance below 50% of peak load. A further example is illustrated in Table 5 run o representing a part load condition performing at a minimum acceptable chilled water velocity through the return air coil. This occurs at 0.2 m/s and in this case is at a temperature of 9 C. The identical condition was repeated at run o' at bottom of Table 5

10 representing a conventional system at 7 C chilled water supply temperature. It has a water velocity of 0.1 m/s through the return air coil. This is at too low a Reynolds Number and is in transition flow which performs unstably.

The above is only one of many examples. There are other digressions from standard practice that similarly occur, often to overcome problems.

15 An unacceptable solution is reduction of the outside air intake requirement for ventilation. The use of desiccant wheels is an expensive replacement, and over-cooling and reheating is also expensive and considerable discomfort arises when reheating is omitted.

20 Clearly a comparative appraisal cannot be given here. Instead a range is presented referring to the percent over the full climatic range of the cooling system. When at all conditions, engineering standards are satisfied, it is estimated that combined energy savings are in range of 40% to 60%, comprising:

25 (a) Input Refrigeration Capacity Control System which maintains an evaporator temperature as high as feasible 10% to 26% including the humidity economy cycle.

(b) Output Refrigeration Capacity Reduction (based on humidity set point in treated space being maintained high in the acceptable performance range) 5% to 12%

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- (c) Output Refrigeration Capacity Reduction (based on humidity economy cycle) with reduction of the energy required to offset the treated space latent heat loads 5% to 20%
- (d) Based on the arrangement of the components, their sizing and circuiting in order to maintain an evaporator temperature as high as reasonably feasible with performance at good engineering standards 20% to 40%.
- (e) When we relate to energy requirements of conventional systems we must make allowance for correction of the energy employed by these systems under conditions of satisfying performance standards as we do in this invention. Therefore a correction factor is applicable. 10% to 20%

Obviously the factors listed above in a to e are not additive. As indicated above, estimate that energy savings will be approximately within the 40 to 60% range.

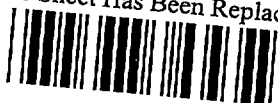
In addition there are capital cost savings, reduced design time, reduced service space, and performance meeting engineering standards over full range.

Further advantages include:

- (a) All conditions which can occur at any point within the operating range are treated individually with an evaporator temperature and coolant supply temperature which is automatically selected for that particular condition.
- (b) System performance which can be tailored to the individual conditions occurring extends the system capacity to handle a wider climatic range.

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- (c) System does not wastefully over cool or reheat at any time. The sensible heat and latent heat loads are offset in the proportion at which they occur.
- (d) System when arranged according to this invention maintains
5 coolant performance in turbulent mode and does not reduce coolant flow to the unstable transition mode.
- (e) Simple controls are not invasive. They operate to improve energy savings for most conventional air conditioning systems.
- (f) System adheres to high Engineering Performance Standards
10 (ASHRAE Standards were applied in the described embodiments)
- (g) The invention is compatible with VAV and CAV designs and with direct expansion air conditioning.
- (h) The described systems do not intrude adversely on any acceptable design as long as provision exists for variation to the
15 evaporator temperature and as long as unnecessary throttling or inefficient capacity control methods are not employed.

In summary:

The energy savings go hand in hand with improved performance.

- It has been demonstrated that by means of a control system associated
20 with maintaining the maximum evaporator temperature compatible with high engineering standards of performance, the complex problems, which arise from the presence of numerous unrelated variables, have been resolved. These variables occur simultaneously and successively with the changes in outside air and internal sensible and latent heat
25 loads. They can best be controlled when each point in time within the operating range is individually addressed.

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In all cases where the system is employed the key objective is to maximise the evaporator temperature in order to conserve both Input and Output Refrigeration Capacity. In the embodiments described an open type compressor was employed. The speed of the compressor is varied responsive to the Evaporator Temperature/Pressure Set Point. When heat and moisture loads fall or rise the speed is reduced or increased (not necessarily respectively) until either one of the treated space, maximum relative humidity design set points, or maximum coolant velocity design set points, is reached, whereupon the capacity control has efficiently established the balance point for that operating condition. As a result the compressor performance closely relates to the existing moisture loads. In lieu of simply throttling the refrigerant to reduce the evaporator temperature on a fall of heat loads or other inefficient capacity control means, the balance point of the system is located at the highest evaporator temperature compatible with the heat and moisture loads occurring at that time. Thus assuring minimum input energy by the refrigeration compressor.

In this design the fluid flow characteristic of turbulent flow is maintained by means of a coolant velocity design maximum. There is also a minimum limit which varies the active size of the cooling coils to put the size of the load where very wide range occur between peak and minor loads.

A very important advantage is associated with the variations in coolant supply temperatures. The range of acceptable cooling coil performance between peak to minimum is very much increased.

Furthermore, the system has been shown to have the capacity to adjust its performance to suit each individual condition including variations in climate and room sensible heat ratio through the flexibility gained from having a variable coolant supply temperature and a variable evaporator temperature/pressure.

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- Input Refrigeration Capacity is a value not readily visible to designers. In this presentation the significance of a variable evaporator temperature and a corresponding variable coolant temperature has been presented. Systems employing stage compression, intercooling, flash gas removal and reduction of condensing temperatures are applicable to this invention. An important principle has been applied here and numerous advantages derived from the use of this principle to air conditioning design. By working to the highest feasible evaporator temperature (and thereby coolant temperature), compatible with good performance to achieve the lowest feasible energy input requirement, there is thermodynamically a greater energy savings with this increase of evaporator pressure, and maintaining condenser pressure constant, than maintaining evaporator pressure constant and decreasing condenser pressure.
- It is estimated that the energy required by conventional systems is 40 to 60% more than what is necessary for this design.

DESCRIPTIONS OF THE PREFERRED EMBODIMENTS

- Embodiments of the invention are described hereunder in some detail with reference to and are illustrated in the accompanying Tables and Drawings, wherein:

Table 1 indicates an example of performance over a range where the invention is applied to a constant air volume system utilising chilled water fed to an outside air coil first as described in detail above.

- Table 2a compares the same sensible heat load conditions at different chilled water supply temperatures. Table 2a indicates a supply coolant temperature of 7 C having a total output refrigeration load of 15.82kW whereas at 11.2 C this load is 15.25 kW a difference of only 3.6% saving. The total saving is not visible here since it depends on the input energy to the compressor. The saving is very much greater approximately 20 per

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cent when the input energy is considered. Perhaps this is the reason so little attention is given to maintaining a high evaporator pressure.

Table 2b is a similar comparison to table 2a. It has a more visible savings of 10.6%. See analysis under section: "Reduction of Output

- 5 Refrigeration Capacity". In addition it has a high input energy saving due to its higher C.O.P.

Table 3 represents a Humidity Economy Cycle indicating a room latent heat load of zero energy penalty in offsetting the room latent heat losses. It indicates the effect of the capacity control system to drive the coolant

- 10 temperature upwards in 8 steps, to achieve excellent performance when the chilled water temperature is doubled from 7 C to 14.2 C at a very much higher evaporator pressure resulting in a very much reduced input energy. Both energy savings in offsetting the total latent heat load without any energy penalty and input energy savings are realised.

- 15 Table 4 again indicates Humidity Economy Cycle similar to Table 3 with a very high room latent heat load that is again fully offset with zero energy penalty

Table 5 indicates an example of performance over a range where the invention is applied to a variable air volume system utilising chilled water

20 fed to an outside air coil first. The same comparison as for a constant air volume system as in Table 1. See conventional row o' performance compared with row o performance of invention under 'SUMMARY OF ADVANTAGES' above.

- 25 Figure 1a compares the performance of the outside air coil in the arrangement of separate outside air and return air coils of this embodiment with Fig 1b which represents a conventional outside air coil application.

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Fig 2 graphically shows how the raising of pressure (and thereby refrigerant temperature) can result in a savings of energy.

Fig 3 shows graphically how an increase in suction temperature from -1.1 C to 10 C (30 F to 50 F) can reduce the Brake Horsepower Requirement
5 by some 26.7%.

Fig 4 can be read in conjunction with Fig 3 and illustrates how the compressor speed for the 30 °F condition is 914 RPM and for the 50°F condition it is 616 RPM for the same refrigeration capacity load of 35.2 kW (10 tons of refrigeration).

10 Fig 5 which is derived from Fig 4, illustrates the savings that can be achieved under conditions of increased suction temperature for the same refrigeration output.

Fig 6 is a graph showing how humidity proportional plus integral output
15 set point and coolant velocity proportional plus integral output set point will generate a high evaporator pressure set point and consequently a high coolant supply temperature as used in the invention.

Fig 7a is a block diagram of the evaporator pressure control system. Here the evaporator interchanges directly with the treated air. It shows the control arrangement which is used in the invention, wherein the
20 humidity is sensed to provide a proportional plus integral output set point.

Fig 7b and Fig 7c are block diagrams of the coolant supply temperature control system. Here the evaporator temperature indirectly exchanges with the treated air. It shows the control arrangement which is used in the invention, wherein the humidity is sensed to provide a proportional plus
25 integral output set point and the velocity of the coolant is sensed to provide a proportional plus integral set point, the two outputs passing through a comparator and the maximum of those two outputs establishing an evaporator pressure set point indirectly through the coolant supply



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temperature that adjusts the variable speed of the compressor so that the evaporator pressure set point is compatible with the higher of the two output limits.

Fig 7b is used to control the capacity in a constant air volume air conditioning system, employing a chiller

In this embodiment employing a four row deep return air cooling coil, all valves are open when at peak load. Modulating Valve 17a throttles until the treated space portion of load it shares with the outside air coil falls to 65% of load approximately when Valve 17a closes Valve 17b begins to throttle to about 35% of load.

When Valve 17b closes then Valve 17c begins to throttle to minimum load at approximately 25% of peak load. Valve 17c is not closed during the full operating range and therefore the velocity sensor is located at this particular valve. One scenario, If room is above 24°C at start up all valves will be open, if room above 23.3 °C valve 17b and 17c will be open and if above 22.6°C, then only valve 17c will be open.

There is no transition or laminar flow in the major active coil portion and no large step changes in the control operation.

Table 1 represents a constant air volume system performance over a range corresponding to Fig 7b.

Fig 7c is a drawing which corresponds to Fig 7b where a variable air volume configuration employing a chiller is presented. Note though 3 modulating valves are shown for a very wide range climatic condition, table 5 employed only one valve.

The valves modulate to maintain to maintain a constant supply air temperature.

Table 5 represents Fig 7c performance over a range.

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Fig 8 indicates psychometric chart sketches of 2 arrangements. Figs 8a and 8c represent the 2 coils arrangement employed in the embodiment of this invention and Figs 8b and 8d represent a single coil arrangement that is also applicable to the invention.

- 5 Fig 9a and Fig 9b indicate present Weather Bureau Data for Singapore and Adelaide respectively marked by curves establishing design loci for the dry bulb day time temperatures. Two very different climates such as these can be treated by the same design by this invention.

Fig 10 shows diagrammatically a variable speed control module.

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TABLE 1

Configuration: Draw-through Type: Typical Floor
 Separate O/A and R/A coils: Parallel airflow, series waterflow
 CAV System

Chilled water fed to O/A coil first.
 Chilled water flow rate through O/A coil fixed at 1.55lps.

Air Flow (ASHRAE Std. Conditions) - Supply Air: 1020.0 lps. Outside Air 200.0 lps.
 Room Dry Bulb Temperature 24.0°C.

Active Coil Depth	Ch. Water Supply Temp. °C Flg 8a (22)	Ret Air Coil Water Vel m/s	Ch. Water Flow Rate lps	Relative Humidity %	Output Room Sens. Refrig. capacity kW	Room Sensible Heat Ratio	Tot. Output Sensible Refr. capacity kW	Tot. Output Latent Heat Capacity kW	Outside Air Condition dbt/wbt °C
4 Rows (4a)	7.00	2.23	1.56	51.2	12.9	0.88	16.51	9.13	35.0/28.0
4 Rows (4b)	9.20	1.96	1.37	56.7	11.0	0.87	14.25	8.28	33.5/27.5
4 Rows (4c)	11.20*	2.26	1.58	57.4	10.5	0.88	12.89	2.37	30.0/21.0
4 Rows (4d)	9.00	0.70	0.49	59.8	10.0	0.86	13.00	7.55	32.5/27.0
4 Rows (4e)	8.00	0.25	0.17	60.6	8.5	0.82	11.37	4.67	32.0/24.0
2 Rows (2a)	8.00	1.96	0.69	58.1	8.5	0.82	11.37	4.96	32.0/24.0
2 Rows (2b)	14.20	2.17	0.76	47.1	6.0	0.85	8.38	0.00	30.0/17.0
2 Rows (2c)	13.00	1.46	0.51	60.3	6.0	0.70	8.38	0.00	30.0/17.0
2 Rows (2d)	7.00	0.21	0.07	59.7	5.8	0.75	7.64	6.75	28.0/24.9
1 Row (1a)	7.00	2.11	0.37	58.7	5.8	0.75	7.64	6.86	28.0/24.9
1 Row (1b)	10.00	0.43	0.08	59.4	4.4	0.80	4.85	6.24	24.0/24.0
1 Row (1c)	7.00	0.30	0.05	59.6	4.4	0.69	5.31	6.94	24.0/24.0

*See Table 2a for comparison of this identical condition with performance when 7°C chilled water in lieu of the 11.2°C chilled water temperature is supplied by the chiller indicated here.

It should be noted that the maximum control system set point limit determining the evaporator temperature/pressure is underlined. (See columns entitled Return Air Coil Velocity and Relative Humidity to determine which value is at maximum limit.)

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TABLE 2A
PERFORMANCE COMPARED AT SAME SENSIBLE HEAT LOAD CONDITIONS
AT DIFFERENT CHILLED WATER SUPPLY TEMPERATURE

Outside Air at 30°C dbt, 21.0 wbt

Chilled water supply Temp °C To OA Coll	Room dry bulb Temp °C	Room wet bulb Temp °C	Ret Air Coll W.v. m/s	Output Room Sensible Ref Cap kW	Room Sensible Heat Ratio	Room Relative Humidity %	Total Output Latent Heat Ref Cap kW	Total Output Ref Cap kW	Room Dew Point Temp °C
7	24.0	17.4	0.35	10.5	0.88	52.4	2.95	15.82	13.65
11.2	24.0	18.2	2.26	10.5	0.88	57.4	2.37	15.25	15.07

TABLE 2B
SIMILAR COMPARISON AS ABOVE

Outside Air at 35°C dbt, 21.5 wbt

Chilled Water Supply Temp °C	Room dry bulb Temp °C	Room wet bulb Temp °C	Ret Air Coll W.v. m/s	Output Room Sensible Ref Cap kW	Room Sensible Heat Ratio	Relative Humidity in Room %	Total Output Latent Heat Ref Capacity kW	Total Output Ref Cap kW	Room Dew Point Temp °C
7	24.0	16.9	0.19	6.0	0.85	49.0	2.1	11.72	12.64
11	24.0	18.4	0.66	6.0	0.85	58.8	1.0	10.61	15.44

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TABLE 3
A HUMIDITY ECONOMY CYCLE
 Outside Air at 30°C dbt, 17°C wbt
 and
 Room Sensible Heat Ratio = 0.85

Chilled Water Supply Temp °C	Room dbt °C	Room wbt °C	Return Air Coil Water Velocity m/s	Room Sensible Heat load kW	Room Sensible Heat Ratio	Room Relative Humidity %	Latent Heat kWR	Total Heat kWR
7	24.0	16.6	0.15	6.0	0.85	46.2	0.10	8.5
8	24.0	16.6	0.19	6.0	0.85	47.1	0.0	8.4
9	24.0	16.6	0.23	6.0	0.85	47.1	0.0	8.4
10	24.0	16.6	0.28	6.0	0.85	47.1	0.0	8.4
11	24.0	16.6	0.35	6.0	0.85	47.1	0.0	8.4
12	24.0	16.6	0.47	6.0	0.85	47.1	0.0	8.4
13	24.0	16.6	0.70	6.0	0.85	47.1	0.0	8.4
14.2	24.0	16.6	2.17	6.0	0.85	47.1	0.0	8.4

TABLE 4
A HUMIDITY ECONOMY CYCLE
 Outside Air at 30°C dbt, 17°C wbt
 and
 Room Sensible Heat Ratio = 0.70

Chilled Water Supply Temp °C	Room dbt °C	Room wbt °C	R/A Coil Ch Water Velocity m/s	Room Sensible Heat Load kWR	Room Sensible Heat Ratio	Room Relative Humidity %	Latent Heat Load kWR	Total Heat kWR
7	24	18.51	0.182	6.0	0.70	59.4	0.11	8.49
12	24	18.65	0.783	6.0	0.70	60.3	0.00	8.39
13	24	18.65	1.464	6.0	0.70	60.3	0.00	8.39

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TABLE 5

Configuration: Draw-through Type: Typical Floor
Separate O/A and R/A coils: Parallel airflow, series waterflow, 4 row active depth of OA and RA coil
VAV SYSTEM

Chilled water fed to O/A coil first

Chilled water flow rate through O/A coil fixed at 1.8 Lps

Air Flow (ASHRAE Std. Conditions) - Supply Air - 1020.0 Lps Outside Air 200.0 Lps

Variable Air - 400 Lps to 1060 Lps

Room Dry Bulb Temperature 24.0 C

Room Supply Air Dry Bulb Temperature 14.4 C

	Comments	Ch. Water Supply Temp C	Ret Air Coil Water Vel m/s	Ch. Water Flow Rate lps	Relative Humidity %	Output Room Sens Refrig. Capacity kW	Room Sensible Heat Ratio	Tot Output Sensible Refr. Capacity kW	Tot Output Latent Heat Capacity kW	Tot Output Refrigeration Capacity kW	Outside Air Cond. Dbt/wbt C
a	Singapore Peak	7.0	2.0	1.000	57.8	12.5	0.7	16.1	12.0	28.1	35/28
b		7.7	2.1	1.100	53.2	12.5	0.8	15.7	10.8	26.5	33/28
c		8.5	1.9	1.000	50.2	11.9	0.9	15.5	8.8	24.3	35/28
d	Adelaide Peak	8.5	2.5	1.300	58.9	9.5	0.7	11.7	8.5	20.2	30/25
e		8.5	2.0	1.100	52.9	12.5	0.8	16.9	3.0	19.9	38/21.5
f		7.0	0.6	0.300	59.4	9.5	0.7	11.2	8.0	19.2	28/24
g	Humidity economy cycle	7.0	0.5	0.300	59.8	9.0	0.7	9.7	8.8	18.4	24/24
h		10.0	2.7	1.400	54.7	9.5	0.8	11.2	6.8	18.0	28/24
i		9.0	2.2	1.200	50.4	12.5	0.9	16.1	1.0	17.2	35/19
j	Humidity economy cycle	9.5	1.0	0.500	60.0	8.0	0.7	8.6	8.3	16.9	24/24
k		10.5	2.3	1.200	51.9	9.0	0.9	9.7	6.8	16.5	24/24
l		9.0	2.2	1.200	52.0	12.5	0.8	14.9	1.5	16.4	30/17
m	Humidity economy cycle	7.0	0.2	0.100	60.8	7.0	0.7	7.5	7.6	15.1	24/24
n		10.0	2.1	1.100	52.3	10.6	0.8	12.9	1.0	13.9	30/17
o		9.0	0.2	0.100	60.9	6.0	0.7	6.4	7.2	13.6	24/24
p	Humidity economy cycle	11.0	2.4	1.200	52.2	5.0	0.8	5.3	6.5	11.8	24/24
q		11.0	2.4	1.200	52.2	9.0	0.8	11.1	0.6	11.7	30/17
r		11.0	0.2	0.100	59.4	4.5	0.8	4.8	6.3	11.1	24/24
s	Humidity economy cycle	13.4	2.7	1.400	47.3	4.7	0.8	6.4	0.0	6.4	30/17
a'	Unstable Conventional	7.0	0.1	0.006	59.3	6.0	0.8	6.4	7.4	13.8	24/24

It should be noted that the maximum control system set point limit determining the evaporator temperature is underlined. (See column entitled Return Air Coil Water Velocity and Relative Humidity to determine which value is at maximum limit.)





CLAIMS

1. An air conditioning system including refrigeration means to effect a highest evaporator temperature for its refrigeration cycle when it treats either directly or indirectly cooling and dehumidifying of air to be conditioned, dry bulb temperature measuring means to measure the temperature of air within a space to be conditioned, humidity measuring means to measure humidity of air within the space to be conditioned, and control means adapted to offset both sensible and latent heat loads of the air at each of the climatic and interior building load conditions over the operating range of the air conditioned system, and control means to establish set points by the designers for humidity performance in the treated space at a high end of an acceptable range, and, if a chiller is employed, at the high end of the acceptable coolant coil velocity performance range, where the higher of these two changing properties reaching its preselected set point value, in combination with the sensible heat and latent heat loads being offset by the refrigeration cycle, will be selecting the value of an evaporator temperature set point which will be as high as reasonably feasible to minimise input energy to the refrigeration compressor and maintain high standard performance conditions for the particular operating condition in the range and on a change of load, will automatically vary the evaporator temperature set point so that it is compatible with each of all the sensible and latent heat loads occurring over the operating range of the system, the control means employed being to change the specific volume at the saturated suction temperature of the refrigerant leaving the evaporator to establish the higher of the said two constant space humidity or coil coolant velocity set points and thus to effect a high evaporator temperature/pressure set point..
2. A method of capacity control of an air conditioning system including a compressor and evaporator in a refrigeration cycle, heat exchanger means, coolant flow conduits, control equipment including compressor, flow control means, proportional plus integral control output equipment and final control means said method including generating an evaporator





pressure/temperature set point for the evaporator as high as reasonably feasible for each of the operating internal and external heat and moisture load conditions occurring over a total air conditioning range.

3. A refrigerant air conditioning arrangement of a type including
5 refrigeration means to provide an evaporator either directly for air to be conditioned or indirectly for the air to be conditioned through a chiller, dry bulb temperature measuring means to measure the temperature of air within a space being air conditioned, control means
10 concurrent with temperature measuring means of the coolant supply leaving the chiller or the evaporator measuring means effecting control either of the velocity of coolant passing through the cooling means and/or the area of cooling surface available to the air being cooled, further characterised in that there are included humidity
15 measuring means to measure humidity of air within the space being air conditioned, and further control means to effect a change in the evaporator temperature in response to the higher of the humidity measuring means or the velocity of coolant means whereby to maintain a refrigerant evaporator temperature which is as high as reasonably feasible for each of the operating internal and external
20 heat and moisture load conditions occurring over a total air conditioning range.
4. A refrigerant air conditioning system as in claim 3 further characterised in that the control means to effect a change in the temperature of the refrigerant evaporator includes means to achieve a
25 temperature change by effecting a change in the speed of a motor driving the refrigerant compressor to a speed directed in response to the control means.
5. A refrigerant air conditioning system as in either one of preceding claim 3 or 4 further characterised in that the control means include
30 means to control the flow rate of coolant through the chiller and cooling coils and in the event of directing a lower than selected flow





rate, effecting a reduction in the extent of chiller and coolant coil size to be used.

6. A refrigerant air conditioning system as in any one of preceding claims 3,4 or 5 further characterised in that there are means to effect
5 a change in the evaporator temperature/pressure in response to directions from the control means.
7. A method of effecting refrigerant air conditioning including using refrigeration means to provide a cooling evaporator either directly for air to be conditioned or indirectly for the air to be conditioned through
10 a chiller, dry bulb temperature measuring means to measure the temperature of air within a space being air conditioned, control means connected to the dry bulb temperature measuring means effecting control either of the velocity of air passing the cooling means and/or the area of cooling surface available to the air being cooled, further
15 characterised in that there are included humidity measuring means to measure humidity of air within the space being air conditioned, and further control means effect a change in the evaporator temperature in response to the humidity measuring means whereby maintaining a refrigerant evaporator temperature which is as high as reasonably
20 feasible for each of the operating internal and external heat and moisture load conditions occurring over a total air conditioning range.
8. A method of effecting refrigerant air conditioning as in claim 7 further characterised in that the refrigeration means to effect a selected extent of cooling by a refrigerant evaporator includes a compressor and
25 means which effect a change in the speed of a motor driving the motor to a speed directed in response to the control means.
9. A method of effecting refrigerant air conditioning as in either one of preceding claim 7 or 8 further characterised in that the control means include means to control the flow rate of coolant through the chiller
30 and in the event of directing a lower than selected flow rate, effecting a deactivation of a portion of the evaporator coils to be used.





- 10.A method of effecting refrigerant air conditioning system as in any one of preceding claims 6, 7 or 8 further characterised in that there are means effecting a change in the pressure of the evaporator in response to directions from the control means.
- 5 11. A method of effecting air conditioning of a space including refrigeration means to effect a selected extent of cooling by a refrigerant evaporator to effect either directly or indirectly a cooling of air to be conditioned, temperature measuring means to measure the temperature of air within a space being air conditioned, humidity
10 measuring means to measure humidity of air within the space being air conditioned and control means to effect a change in an amount of air being subject to the cooling means, and further to control an extent of cooling effect available from the evaporator, the control means effecting control of the air conditioning elements by effecting a
15 priority in changing the controllable elements in response to a change in input so as to direct these toward maintaining a refrigerant evaporator temperature which is enabled thereby to be set to a set point as high as reasonably feasible in response to any of a range of temperature and humidity.
- 20 12.A method of effecting air conditioning of a space as in claim 7 wherein said proportional and integral control output equipment is coupled to said humidity control means and also to said coolant velocity control means where chillers are employed, said method further comprising establishing the governing set point for establishing the governing
25 evaporator temperature determined by the humidity control means or the coolant velocity means set point value which is first reached after start up and after transfer of heat and moisture loads to the controlled environment and thereby in turn controlling said evaporator pressure/temperature to be as high as reasonably feasible for the
30 particular balance point in the range.

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13.A method according to claim 12 further comprising continuously readjusting, the performance of the refrigeration system on change of loads so that said maximum set points of humidity or coolant velocity generate an evaporator pressure/temperature set point and thereby a coolant supply temperature set point where chillers are employed to be as high as reasonably feasible.

14.A method according to 12 further including resetting one of input evaporator pressure signal, and input coolant supply temperature signal so as to maintain evaporator temperature compatible with the greater of said maximum humidity and coolant velocity set points. effecting said resetting by varying input capacity of a compressor, by varying one or more of the following: speed, sliding valves, pre-rotating vanes, or cylinder unloading devices whereby to in turn adjust its capacity to achieve an evaporator set point to each of all specific environmental operating conditions in said range.

15.A method according to 14 wherein said compressor has a range of speeds between a minimum and a maximum, speed control means coupled to said compressor, and wherein there is no secondary coolant nor chiller but treated air interfaces with said evaporator in a heat exchange, said method comprising establishing as a minimum set point said minimum compressor speed, such that in the case where said design input value of humidity set point is not reached, the evaporator temperature will be determined by the compressor at the manufacturers minimum speed setting and be supplied within the variable speed control device or equivalent control.

16.A method according to 14 including a capacity control system establishing one of said coolant input values to reach its design set point and thereby a balance point having a mass flow rate of refrigerant which is generally constant for each specific refrigeration output load, and when the proportional plus integral output calculated set point in combination with the process variable signal from either





- the evaporator pressure sensor or in the case of chiller from the coolant supply temperature sensor signals a reduction in the compressor revolutions per minute it is at a relatively high specific volume and is at a relatively low suction pressure, if the treated space humidity is below its set point and the coolant velocity is also below its set point then readjusting said flow rate to a relatively low specific volume and a suction pressure which is relatively high and thus generate an evaporator temperature/pressure set point as high as reasonably feasible for its particular operating condition based on having established the higher of the treated space humidity or coolant velocity set points, and if this signal had demanded an increase in compressor rpm, then this lower evaporator temperature would also be as high as is reasonably feasible in order to maintain high performance standards.
- 15 17.A method according to claim 1 wherein the configuration comprises a single coil with outside air and return air combined together to from the supply air to the treated space.
- 20 18.A method according to any one of the preceding method claims further including an outside air coil and a return air coil, further including cooling and dehumidifying outside air through said outside air coil, , passing most but not all return air leaving air from treated space through said return air coil and mixing with said cooled and dehumidified outside air as the remainder of said supply air, and passing coolant first through said outside air coil and subsequently through said return air coil.
- 25 19.A method according to any one of the preceding method claims further including a single coil with outside air and return air combined to pass through a single coil to form the supply air to the treated space, said method including the steps of mixing low dew point temperature outside air with return air to produce said supply air, enabling the low dew point temperature of the outside air to offset at
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no energy or part energy cost of the latent heat loads in the treated space due to the high evaporator and coolant temperatures resulting from the high evaporator set point from this humidity economy cycle.

- 20.A method according to claim 7 wherein upon maintaining the highest evaporator temperature feasible when it treats either directly or indirectly the air to be conditioned, the effect of the higher treated space evaporator set points in addition to reducing the input energy to the compressor also reduces the output refrigeration capacity over lower treated space humidity set point as for the identical sensible and latent heat loads to the system.
- 21.The method according to claim 7 enables performance to be tailored for each operating condition, with each design having broad climatic coverage due to the flexibility of the system to offset the infinite combinations of internal heat and moisture loads over wide climatic range, with the consequence the same system has a universal application to fit numerous climates with the limitation being only that of size range a universal system covering all desired size ranges can readily be established.
22. A method according to any one of the preceding method claims further including an outside air coil and a return air coil, and employing during non-humid weather when outside air is below dew point temperature of the supply air to the treated space, said method including the steps of mixing low dew point temperature leaving outside air with leaving return air to produce said supply air, enabling the low dew point temperature of the outside air to offset at no energy cost all or part of the latent heat load in the treated space due to the high evaporator and coolant temperatures resulting from the high evaporator temperature set point making possible this economy cycle.
- 23.A variable air volume refrigerant air conditioning system comprising a compressor, a refrigerant evaporator, a condenser, a chiller for chilling said liquid, and said chilled liquid means for controlling the





supply air dry bulb temperature and air impelled by impeller, conduit means directing flow of said impelled air through air volume dampers for regulating treated space dry bulb temperature, and means to effect sufficient velocity of both refrigerant through said evaporator and
5 chilled liquid through said heat exchangers to retain turbulent flow of said chilled liquid through said heat exchangers.

24. A method of air conditioning as in any one of the preceding method claims further characterised in that the system is a constant air volume air conditioning system and includes a compressor, a
10 refrigerant evaporator, a condenser, an air flow impeller with air cooled in said evaporator in absence of a chiller, and said, means for controlling treated space dry bulb temperature.

25. A method of air conditioning as in any one of the preceding method claims further characterised in that the system includes a heat
15 exchanger having a coil of variable active size.

26. A method of air conditioning as in any one of the preceding method claims further characterised in that the system maintains coolant within an heat exchanger at a turbulent velocity.

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