

VAV Diffuser Controls

Technical Field

Embodiments of the disclosure relate to a method of controlling an HVAC system.

5 **Background Art**

Controls for multi-zone VAV (variable air volume) diffusers in HVAC (heating, ventilation and air conditioning) applications are increasingly being designed for improved energy savings. Areas receiving particular attention are economiser operation and low vs high supply air temperature to save fan energy and chiller energy, respectively.

In relation to operating at optimum supply air temperature, for a given capacity, reducing the supply air temperature in *mechanical cooling mode* results in fan energy savings due to the reduced airflow rate required, but increases mechanical cooling plant energy consumption due to the need to cool the supply air to a lower temperature. The balance between these two is complex. It depends on multiple factors, including plant and fan efficiency, climate, humidity, load on the overall HVAC system (fan power consumption increases with the cube of the airflow rate), etc. Many international guidelines for good HVAC practice recommend setting the supply air temperature as a function of outdoor air temperature, with a higher supply air temperature setpoint being called for at low outdoor air temperatures when less cooling is required, thereby saving on cooling plant energy when additional fan energy savings would be minimal due to the low airflow rates required to deliver the small cooling capacities that are likely to be called for. So-called “trim and respond” setpoint reset logic is typically used to set supply airflow rate and supply air temperature, typically by adjusting fan static pressure and supply air temperature setpoints to cover the load of the “critical” (or worst case) zone so that the electronic VAV diffuser that serves that zone (or group of diffusers serving the worst-case zones) – often referred to as the “index diffuser” - operates with its VAV damper substantially fully open to reduce pressure drop (and hence fan energy) whilst satisfying the thermal load in the space. The VAV diffuser damper to this (or these) critical zone is almost fully open, but nevertheless slightly throttled, due to the trim-and-reset controls operation. This wastes energy, as a slightly greater air pressure is required to overcome the extra pressure of a critical (ie index) diffuser not operating fully open.

All non-index VAV diffusers (ie, those in non-critical zones), by definition, operate with their VAV dampers partially throttled to avoid over-cooling, as each such diffuser is located in a higher pressure location in the duct system than the pressure required for the diffuser to supply its required airflow rate when operating with its VAV damper fully open. Over-cooling may, nevertheless, occur if, for example, a VAV damper is prevented from throttling far enough due to indoor air quality criteria dictating a higher airflow rate than is required to cool the space.

In *economiser mode*, outdoor air, in preference to recirculated air, is drawn into the air

handling unit for cooling purposes. This typically occurs whenever cooling is required and the drybulb temperature or the enthalpy of the outdoor air is lower than that of the return air and the outdoor air is not too cold to cause over-cooling. If the outdoor air is not cool enough to cool the space then so-called “integrated economiser operation” provides supplementary cooling from the mechanical plant. Again, the right balance between supply air temperature (to which the mechanical plant will cool if this setpoint is lower than the outdoor air temperature) and supply airflow rate needs to be found to maximise energy savings. Furthermore, free-cooling potential should ideally be optimised to prevent supplementary cooling from being called for if simply increasing fan speed to raise the supply airflow rate will deliver the cooling capacity required. However, current controls systems or building management and control systems (BMCS), the latter also known as building management and control systems (BMCS), that reset supply air temperature do not fully achieve this goal.

In *mechanical cooling mode* (both with or without the economiser active) the supply air temperature setpoint may be adjusted to suit other criteria, such as dehumidification (which in a humid climate requires a reasonably low supply air temperature setpoint – typically of at least 15 °C - to prevent discomfort due to mugginess and to prevent mould growth) or over-cooling of non-critical zones (which may be averted by raising the supply air temperature, on condition that this will not result in insufficient cooling capacity to the critical zone).

Many studies suggest that supply air temperature reset based on outdoor air temperature provides poor energy optimisation. Additionally, energy may be wasted due to calibration issues between the outdoor air temperature sensor and supply air temperature sensor, especially as calibration tends to drift over time, which, in economiser mode, may lead to a false load being applied to the mechanical cooling plant or to return air being mixed into the outdoor air when not required, in each case wasting energy.

Furthermore, current supply air temperature and pressure reset strategies do not currently allow the optimum balance between these two parameters to be established because system static operating pressure is typically determined at a point in one or more ducts well removed from the VAV diffusers delivering the supply air to each space. It is the VAV diffusers that need to be operated at their minimum permissible pressure, whenever possible, if fan energy savings are to be maximised, but this is not possible if the static pressure at the diffusers is not known.

A further issue faced by many VAV diffuser systems is that of thermal comfort. While resetting supply air temperature and supply airflow rate, and hence supply air pressure, allow energy savings to be achieved relative to operating at fixed supply air temperature and pressure, changes to these parameters change the cooling or heating capacity delivered to each zone for a given VAV damper setting, causing zone temperature fluctuations which may lead to both discomfort and so-called hunting of the control system involving under-shoot and over-shoot as it then tries to re-establish an equilibrium state.

The above shortcomings of electronic VAV diffuser controls systems of the prior art are some of the areas addressed by the invention.

Summary

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An embodiment relates to a method of controlling an HVAC system, the HVAC system comprising an air handling unit to produce a supply airflow, the air handling unit being connected via a ducting system to a plurality of VAV diffuser units, each VAV diffuser unit influencing air temperature in a corresponding zone by delivering supply air to the zone, wherein

- 10 the air handling unit comprises at least one supply air fan or motorised supply air damper; and
- each diffuser unit comprises a diffuser, a motorised variable damper to alter a diffuser airflow rate of a diffuser airflow delivered by the corresponding diffuser unit into the corresponding zone, and a diffuser pressure sensor for determining a pressure of the diffuser
- 15 airflow; wherein
- the HVAC system further comprises a plurality of zone temperature sensors;
- the HVAC system further comprising at least one control unit connected to the supply air fan or motorised supply air damper, each diffuser unit motorised variable damper, each zone temperature sensor and each diffuser pressure sensor;
- 20 the method comprising determining a target thermal capacity for a selected diffuser unit with reference to a target diffuser airflow rate and a target diffuser temperature differential and controlling the motorised variable damper of the selected diffuser unit with reference to the determined target thermal capacity for the selected diffuser unit.

- 25 The control unit may further comprise a plurality of micro-controllers that communicate with one another.

 The plurality of micro-controllers may comprise micro-controllers provided on each of the diffuser units.

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 The air handling unit may further comprise at least one heat exchanger, and the at least one control unit may be connected to the at least one heat exchanger.

 The method may further comprise, for the selected diffuser unit:

 determining a zone temperature setpoint;

determining a zone air temperature with a corresponding zone temperature sensor;

determining a diffuser unit control output in dependence on the zone temperature setpoint and the determined zone air temperature;

determining the target diffuser airflow rate in dependence on a diffuser design airflow rate and the diffuser unit control output for the selected diffuser unit;

determining a diffuser design temperature differential;

determining the target diffuser air temperature differential in dependence on the diffuser design temperature differential and the diffuser unit control output for the selected diffuser unit; and

determining the diffuser target diffuser airflow rate in dependence on the diffuser target temperature differential, or determining the diffuser target temperature differential in dependence on the diffuser target diffuser airflow rate.

The design temperature differential may be determined in relation to a diffuser supply air temperature.

The diffuser supply air temperature may be determined by a sensor incorporated into the selected diffuser unit.

The diffuser supply air temperature may be determined by a sensor located remote from the selected diffuser unit.

The target thermal capacity may be calculated according to the following formula:

$$C\%D = D\%SdTR * D\%AR$$

where C%D is the target thermal capacity expressed as a percentage of maximum diffuser unit control output, D%SdTR is the target diffuser air temperature differential expressed as a percentage of the diffuser design temperature differential, and D%AR is the target diffuser airflow rate expressed as a percentage of the diffuser design airflow rate.

The method may further comprise determining a plurality of sets of control variables, each set comprising a target factored diffuser airflow rate and a target factored diffuser temperature differential, in which each set satisfies the predetermined target thermal capacity for the selected diffuser unit.

The HVAC system may further comprise an economiser damper system for selectively mixing an air supply from outside the HVAC system with the supply air when the HVAC system is in an economiser mode, the method further comprising, when the HVAC system is in an economiser mode:

- 5 each set of control variables satisfies the predetermined target thermal capacity for the selected diffuser unit additionally comprises
 - a target factored economiser deadband;
 - determining the target factored economiser deadband; and
 - controlling the economiser damper system with reference to said target factored economiser deadband.

- 10 The method may further comprise selecting one of said plurality of sets of control variables according to one or more criteria.

The criteria may include one or more of humidity, over-cooling, over-heating, outdoor air temperature, energy conservation, indoor air quality, noise, excess pressure.

- 15 The control unit may further control one or more of: a supply air temperature, supply air pressure, and supply air fan speed or damper position of the motorised supply air damper; to alter or maintain a temperature of at least one zone.

The control unit may control the supply air pressure to alter or maintain a temperature of at least one zone when the motorised variable damper of the diffuser unit serving the at least one zone is fully open.

- 20 The control unit may control the supply air temperature to alter or maintain a temperature of at least one zone when the motorised variable damper of the diffuser unit serving the at least one zone is fully open.

The method may further comprise, for the selected diffuser unit, adjusting the motorised variable damper to deliver an adjusted target factored damper airflow rate.

- 25 The adjusted target factored damper airflow rate may be determined with reference to an actual temperature differential relative to a target factored diffuser air temperature differential for the diffuser unit, and an actual pressure at the diffuser unit relative to a target factored diffuser pressure.

The method may further comprise, for a selected diffuser unit with fully open motorised variable damper, adjusting the target factored diffuser pressure as a function of the target factored diffuser airflow rate.

5 The method may further comprise, for each diffuser unit, determining a diffuser temperature offset request and a diffuser pressure offset request.

For each diffuser unit, the diffuser temperature offset request may be determined by subtracting the actual temperature differential from the target factored diffuser temperature differential, or if the actual pressure is greater than the target factored diffuser pressure then from the target factored diffuser temperature differential multiplied by the square root of the target factored
10 diffuser pressure divided by the square root of the actual pressure.

The diffuser pressure offset request may be determined by subtracting the actual pressure from the target factored diffuser pressure, or if the actual temperature differential is greater than the target factored diffuser temperature differential then from the target factored diffuser pressure multiplied by the square of the target factored diffuser temperature differential divided by the square of the actual
15 temperature differential.

The method may further comprise realising, for each diffuser unit, a diffuser pressure offset request of zero when the target factored diffuser pressure is equal to the actual pressure and the target factored diffuser temperature differential is greater than or equal to the actual temperature differential.

20 The method may further comprise realising a diffuser temperature offset request of zero when the target factored diffuser temperature differential is equal to the actual temperature differential and the target factored diffuser pressure is greater than or equal to the actual pressure.

The method may further comprise realising, for each diffuser unit, a diffuser pressure offset request of zero when the target factored diffuser pressure is greater than the actual pressure and when the target factored diffuser pressure divided by the actual pressure is equal to the square of the actual
25 temperature differential divided by the square of the target factored temperature differential.

The method may further comprise realising a diffuser temperature offset request of zero when the target factored diffuser temperature differential is greater than the actual temperature differential and the target factored diffuser temperature differential divided by the actual temperature differential

is equal to the square root of the actual pressure divided by the square root of the target factored diffuser pressure.

The method may further comprise ranking each diffuser temperature offset request in a temperature rank and ranking each diffuser pressure offset request in a pressure rank.

5 The HVAC system may further comprises an air supply controller with proportional-integral (PI), proportional-integral-derivative (PID) or nudge-and-wait control, and with the setpoint set to zero, and with the process value designated as the highest diffuser pressure offset request from the pressure rank.

10 The HVAC system may further comprise a thermal capacity controller with proportional-integral (PI), proportional-integral-derivative (PID) or nudge-and-wait control, and with the setpoint set to zero, and where the direction of the thermal capacity control is set by a requirement for system cooling or system heating, with the former designating the process value to be the lowest diffuser temperature offset request from the temperature rank and the latter designating the process value to be the highest diffuser temperature offset request from the temperature rank.

15 The requirement for system cooling or system heating may be determined by a vote of the diffuser units.

20 The HVAC system may further comprise an air supply target from the air supply controller, wherein the air supply target determines a supply air pressure setpoint for the air handling unit in relation to an air handling unit supply air pressures sensor, or directly determines the fan speed of the at least one supply air fan.

The HVAC system may further comprise a cooling or heating capacity target from the thermal capacity controller, wherein the cooling or heating capacity target determines a supply air temperature setpoint for the air handling unit in relation to an air handling unit supply air temperature sensor, or directly determines the cooling or heating output of the at least one heat exchanger.

25 The method may further comprise determining an economiser damper temperature setpoint with reference to a target economiser deadband for each diffuser unit.

The method may further comprise, for each diffuser unit, subtracting the actual temperature from the economiser damper temperature setpoint to determine an economiser offset request.

The method may further comprise ranking each diffuser unit economiser offset request in an economiser rank.

5 The HVAC system may further comprise an economiser controller with proportional-integral (PI), proportional-integral-derivative (PID) or nudge-and-wait control, and with the setpoint set to zero, and with the process value designated as the lowest economiser offset request from the economiser rank.

10 The method may further comprise an economiser target from the economiser controller, wherein the economiser target determines an economiser supply air temperature setpoint for the economiser damper system in relation to the air handling unit supply air temperature sensor, or directly determines an economiser damper system output to directly modulate the economiser damper system.

15 **Detailed Description of the Drawings**

In the following detailed description, reference is made to accompanying drawings, which are not to scale and which form a part of the detailed description.

20 The same part number is used for the same part if it appears across multiple figures.

 The illustrative embodiments described in the detailed description, depicted in the drawings and defined in the claims, are not intended to be limiting. Other embodiments may be utilised, and other changes may be made without departing from the spirit or scope of the subject matter presented.

25 It will be readily understood that the aspects of the present disclosure, as generally described herein and illustrated in the drawings, can be arranged, substituted, combined, separated and designed in a wide variety of different configurations, all of which are contemplated in this disclosure.

 Embodiments will now be described, by way of example only, with reference to the accompanying drawings in which:

30 Figure 1 is a diagram illustrating a schematic of an air handling unit (AHU) supplying air to a room or zone via an electronic VAV diffuser;

Figure 2a is a diagram illustrating the operation of a room temperature proportional-integral (PI) controller for cooling and heating of the room or zone depicted in figure 1;

5 Figure 2b is a diagram illustrating the diffuser cooling demand request range for the diffuser depicted in figure 1 in response to a PI control output from the controller in figure 2a;

10 Figure 3a is a diagram illustrating the diffuser temperature differential request range associated with the cooling demand range in figure 2b;

15 Figures 3b is a diagram illustrating the diffuser airflow rate request range associated with the cooling demand range in figure 2b;

20 Figure 3c is a diagram illustrating the diffuser pressure request range of the supply air associated with cooling demand range in figure 2b;

25 Figure 3d is a diagram illustrating the diffuser damper demand range of the supply air associated with cooling demand range in figure 2b;

30 Figure 3e is a diagram illustrating the diffuser temperature differential request range associated with the cooling demand range in figure 2b if economiser status is active.

35 Figure 3f is a diagram illustrating diffuser airflow rate request range associated with the cooling demand range in figure 2b if economiser status is active;

40 Figure 4a is a diagram illustrating a schematic of an air handling unit (AHU) supplying air to a room or zone via an electronic VAV diffuser if economiser mode is active and supplementary mechanical cooling is required;

45 Figure 4b is a diagram illustrating economiser PI controller output for figure 4a;

50 Figure 5a is a diagram illustrating a schematic of an air handling unit (AHU) supplying air to a room or zone via an electronic VAV diffuser if economiser mode is active and 100% free-cooling potential is available;

55 Figure 5b is a diagram illustrating economiser PI controller output for figure 5a;

Figure 6a is a diagram illustrating a schematic of an air handling unit (AHU) supplying air to a room or zone via an electronic VAV diffuser if economiser mode is active and excessive free-cooling potential is available;

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Figure 6b is a diagram illustrating economiser PI controller output for figure 6a.

Figure 7a is a diagram illustrating a schematic of an air handling unit (AHU) supplying air to a room or zone via an electronic VAV diffuser if economiser mode is active and return air damper leakage occurs.

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Figure 7b is a diagram illustrating economiser PI controller output for figure 7a;

15 Detailed Description of the Embodiments

The embodiments, as described herein, relate generally to an AHU supplying air to a multi-zone VAV system comprising multiple electronic VAV diffusers or VAV terminal units.

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For reasons of simplicity, the text below refers to, and the accompanying illustrations show, just one of multiple diffuser units serving a multi-zone VAV system, each diffuser unit being an electronic VAV swirl diffuser as, for example, described in International (PCT) Patent Application No. PCT/AU2021/050923. This diffuser has integrated sensors for supply air temperature, supply air static pressure, and room air temperature. It will be appreciated by persons skilled in the art that each VAV damper need not be integrated into the diffuser unit (for example, it may be a VAV terminal unit located upstream of one or more diffusers), and that the diffuser unit need not discharge with swirl (for example, it may be a four-way blow diffuser or a multi-cone diffuser).

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For reasons of simplicity, methods of determining diffuser minimum VAV damper and AHU minimum outdoor air damper positions or airflow rates for indoor air quality purposes are not discussed. It is to be assumed, in operation, that these dampers do not close to minimum positions that are less than required to satisfy indoor air quality criteria and standards.

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While multiple figures show the diffuser unit installed in a ceiling, it is to be understood that the diffuser unit need not be installed in a ceiling. It may, for example, be installed in a wall or floor, or be freely suspended.

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The illustrations below describe a controls embodiment in room cooling mode, but it is to be understood that the concepts presented apply to room heating mode as well.

5 Reference is made to proportional-integral (PI) controls. It will be appreciated by a person skilled in the art that embodiments with proportional (P) control or proportional-integral-derivative (PID) control also fall within the spirit or scope of the description.

10 It will, further, be appreciated by persons skilled in the art that numerous variations and/or modifications may be made as shown in the specific embodiments without departing from the spirit or scope of the description. The present embodiments are, therefore, to be considered in all respects as illustrative and not restrictive.

Figure 1 is a diagram illustrating a schematic of an *air handling unit (AHU) 1* comprising *return*
15 *air fan 2* drawing air from *return air filter 3* and discharging *return air 2a* via both *return air chamber 24a* through *exhaust air damper 4* to *outdoors 23* and *return air damper 5* to *mixed air chamber 24*, and additionally comprising *supply air fan 7* drawing air via *supply air filter 8* from *mixed air chamber 24* fed by both *outdoor air damper 6* from *outdoors 23* and *return air damper 5* (wherein an *economiser damper system* comprises *exhaust air damper 4*, *return air damper 5* and *outdoor air damper 6*), and in
20 turn discharging *supply air 7a* through *heat exchanger 9* into *supply plenum 10* and across *AHU supply temperature sensor 11* and *AHU supply pressure sensor 11a* before passing from *supply plenum 10* into *supply duct 12* and *branch duct 12a* to flow into *diffuser unit 14* (as well as other *diffuser units 14*, not shown) to be discharged as *diffuser supply air 14a* to mix into *room or zone 20* prior to returning via *return air grille 21* and *return air duct 22* to AHU 1 via *return air filter 3*. In some embodiments, AHU 1 is
25 not equipped with *AHU supply temperature sensor 11* and *AHU supply pressure sensor 11a*.

Diffuser unit 14 is an electronic VAV swirl diffuser comprising *diffuser swirl face 15*, *VAV damper 16* including a motorised *actuator* such as an electronic stepper motor (not shown) and *connection box 13* containing *diffuser static pressure sensor (Pst.d) 17* and *diffuser supply temperature sensor (Ts.d) 18*. *Room air temperature sensor (Troom) 19* may additionally be contained within *diffuser*
30 *unit 14* if, for example, fed by an induction system (not shown) to draw *room or zone 20* air across the sensor, or one or more of the sensors may be located in *room or zone 20*.

It will be apparent to a person skilled in the art that some embodiments of the invention may comprise a central plant fan supplying air to a variable air volume (VAV) unit with diffuser units 14 connected to it. The variable air volume unit comprises a motorised supply air damper that varies the
35 *airflow rate supplied by the central plant fan to the diffuser units 14. In such an instance, the variable air volume unit would serve the function of supply air fan 7 and may be controlled as if it were supply air fan*

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In a preferred embodiment, *diffuser static pressure sensor (Pst.d)* measures the static air pressure within *diffuser unit 14* upstream of *VAV damper 16* relative to the pressure of *room or zone 20*.

In some embodiments, this measurement may, instead, be relative to the pressure in the ceiling void

5 (not shown) that *diffuser unit 14* is located in.

Diffuser controller 90 communicates via *comms cable 91* with each *diffuser unit 14* to receive inputs from respective *diffuser static pressure sensor 17*, *diffuser supply temperature sensor 18*, and *room temperature sensor 19*, as well as to provide outputs to *VAV damper 16* for each *diffuser unit 14*.

10 In some embodiments not all *diffuser units 14* are equipped with *diffuser static pressure sensor 17* or with *diffuser supply temperature sensor 18*.

Diffuser controller 90 communicates via *comms cable 92* with *plant controller 93* (potentially via a Building Management and Control System ("BMCS"), or with either *diffuser controller 90* or *plant controller 93*, or both, being contained within a BMCS, not shown), which in turn communicates via *comms cable 94* with *AHU 1* to receive inputs from, potentially amongst others, *AHU supply pressure sensor 11a* (if present) and *AHU supply temperature sensor 11* (if present), as well as to provide outputs to at least *supply air fan 7*, *return air fan 2* (if present), *heat exchanger 9*, and if equipped with an economiser then *outdoor air damper 6*, *return air damper 5*, and *exhaust air damper 4* (if present). Not shown are inputs to *plant controller 93* from additional sensors for outdoor and return air temperature, as well as outdoor and return air enthalpy, which may be used by *plant controller 93* when activating and deactivating *economiser mode* operation.

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Instead of *comms cables 91, 92* and *94*, some or all communications could be wireless (not shown).

In some embodiments, *diffuser controller 90* and *plant controller 93* are combined into a single controller or BMCS, or either one of them may be integrated into the BMCS.

25 In some embodiments, *diffuser controller 90* comprises a plurality of micro-controllers. Each micro-controller may serve a *diffuser unit 14* and may be located in its respective *diffuser unit 14*, and may be networked to one another wirelessly or via *comms cable 91* which may be a daisy-chain or controls loop cable connecting the plurality of *diffuser units 14* to one another.

In operation, *diffuser controller 90* places all *diffuser units 14* (other than those *diffuser units 14* downstream of a trim heater, as described below) either into *room cooling mode* or *room heating mode*, typically in response to a majority vote from all *diffuser units 14* served by *AHU 1* based on temperature inputs from each *room temperature sensor 19* serving each *room or zone 20*. It will be apparent to a person skilled in the art that a trim heater, not shown, could be located in *branch duct 12a* to serve *diffuser unit 14*, or in *duct 12* to serve a multitude of *diffuser units 14*, in which case *diffuser controller 90* may designate those *diffuser units 14* downstream of the trim heater as being in *trim heating mode* even if all other *diffuser units 14* are in *room cooling mode*. The controls described below

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can readily be adapted for such *trim heating mode* control.

In operation, *plant controller 93* determines fan speed for *supply air fan 7*, based on *AHU supply pressure sensor 11a* (if present) and a corresponding *AHU pressure setpoint* from *diffuser controller 90*, or by direct fan speed control from *diffuser controller 90*. Additionally, *plant controller 93* determines *AHU mechanical cooling mode* or *AHU mechanical heating mode* for *AHU 1*, and controls a respective *mechanical cooling output* or *mechanical heating output* from *heat exchanger 9* (shown as a single heat exchanger, but in some embodiments could equally well consist of a plurality of heat exchangers; for example, one for cooling and one for heating), either from *diffuser controller 90* directly (with *diffuser controller 90* in that case instructing *plant controller 93* of the relevant *AHU mechanical cooling mode* or *AHU heating mode*) or based on *AHU supply temperature sensor 11* in relation to an *AHU supply temperature setpoint* (with *AHU mechanical cooling mode* or *AHU mechanical heating mode* for *AHU 1* being selected by *plant controller 93* to suit) which is reset by *diffuser controller 90* to maintain *room or zone 20* temperature setpoint. Furthermore, in *AHU cooling mode*, *plant controller 93* activates and deactivates *economiser* operation (if *AHU 1* is equipped with an *economiser*) and during *economiser* operation determines damper modulation of *outdoor air damper 6*, *return air damper 5*, and *exhaust air damper 4* either directly or based on inputs from *AHU supply temperature sensor 11* (if present) and a corresponding *economiser damper temperature setpoint* from *diffuser controller 90*. If supplementary mechanical cooling is required to achieve integrated *economiser* operation then *outdoor air damper 6* and *exhaust air damper 4* are commanded fully open and *return air damper 5* fully closed, and *plant controller 93* determines *mechanical cooling output* from *heat exchanger 9* based on *AHU supply air temperature sensor 11* or *diffuser supply air temperature sensor 18*, as well as a corresponding *AHU supply temperature setpoint* from *diffuser controller 90*, or *diffuser controller 90* directly commands *plant controller 93* of the required *mechanical cooling output* from *heat exchanger 9*. The *AHU supply temperature setpoint* is greater than the *economiser damper temperature setpoint*. The two are separated by a *dynamic economiser deadband*, preferably of at least 1 K *minimum economiser deadband* magnitude, determined by *diffuser controller 90*.

When *room cooling mode* has been activated by *diffuser controller 90*, a supply air temperature of less than *room or zone 20* air temperature is required. It should be noted that *heat exchanger 9* may, when controlled by *plant controller 93* to the required *AHU supply temperature setpoint*, be controlled to either a cooling or a heating output (the latter, for example, if a high proportion of *outdoor air 23* that is too cold is supplied by *outdoor air damper 6*). In this case, *heat exchanger 9* will either be controlled by *plant controller 93* to operate in *AHU mechanical cooling mode* to cool air from *mixed air chamber 24* down to the *AHU supply cooling setpoint* (for example if both return air and outdoor air are warm) or to operate in *AHU mechanical heating mode* to warm air from *mixed air chamber 24* up to the *AHU supply heating setpoint* (for example if outdoor air is cold and a

large percentage of outdoor air is required). In other words, in *room cooling mode*, *fan supply air 7a* may be either cooled or heated to a *diffuser supply air 14a* temperature that is less than *room or zone 20* temperature so as to cool *room or zone 20* down to setpoint temperature.

In contrast, if *diffuser controller 90* directly determines the cooling or heating output of *heat exchanger 9* then *AHU mechanical cooling mode* or *AHU mechanical heating mode* will correspond to the *diffuser cooling mode* or *diffuser heating mode*, respectively. Such direct control by *diffuser controller 90* of the cooling or heating output of *heat exchanger 9* should, therefore, be avoided in applications, such as auditoria, which may require supply air with a high proportion of, or, indeed, with 100 percent outdoor air content. Such direct control is better suited to applications that normally include an overwhelming return air proportion (like most commercial office applications).

Figure 2a is a diagram illustrating *room temperature proportional-integral (PI) controller* for cooling and heating of *room or zone 20* depicted in figure 1, with *cooling PI output 25d* being denoted in percent on the vertical axis. The *global room temperature setpoint* is shown as 22.5°C with a deadband of 1 K magnitude, resulting in a *cooling setpoint of room or zone 20* of 23°C and a *heating setpoint of room or zone 20* of 22°C, which are controlled to when *diffuser controller 90* activates *room cooling mode* or *room heating mode*, respectively. *Cooling PI curve 25* and *heating PI curve 26* are each shown with a 1.5 K proportional band by way of example only. Other proportional band values, including ones that differ between cooling and heating, may be used. In the following figures it is to be assumed that *diffuser controller 90* has activated *room cooling mode*. While PI control has been depicted it will be apparent to a person skilled in the art that proportional-integral-derivative (PID) or proportional (P) control could also be used.

Figure 2b is a diagram illustrating *cooling percent demand (C%D) 25d'* of *diffuser unit 14* relative to *cooling PI output 25d* derived from figure 2a. The relationship is shown as proportional (P) so that, for example, a 50% *cooling PI output 25d* in figure 2a will result in a 50% *C%D 25d'* from *diffuser unit 14*. Consequently, if a heat load is applied to *room or zone 20* in figure 1 such that a 50% cooling capacity will achieve room temperature steady state at the cooling setpoint of 23°C in figure 2a then, over time, the integral function of *cooling PI curve 25* in figure 2a will move the curve left, as depicted by arrow 27, to *adjusted cooling PI curve 25a*, resulting in a 50% *cooling PI output 25d*, as shown by circle 25b, which in turn will result in a 50% *C%D 25d'* in figure 2b. If the *diffuser design capacity* of *diffuser unit 14* is, for example, 1000 W then a *C%D 25d'* of 50% will call for a *cooling capacity demand* of 500 W from *diffuser unit 14*.

While *C%D 25d'* is shown in figure 2b as being proportional (P) to cooling PI output 25d, it is to be understood that this relationship need not be linear.

Figures 3a and 3b are graphs illustrating *diffuser percent supply temperature differential request* ($D\%SdTR$) 30 and *diffuser percent airflow rate request* ($D\%AR$) 31 of *diffuser supply air 14a* as a function of *cooling percentage demand* $C\%D$ 25d' determined in figure 2b. $D\%SdTR$ 30 is expressed relative to the *diffuser design temperature differential* ($DDdT$), which is the temperature differential that produces the *diffuser design capacity* at *diffuser design airflow rate* (DDA). $DDdT$ is the design temperature differential between the *diffuser design supply air temperature*, measured at *diffuser supply air temperature sensor 18*, and the temperature of the air in *room or zone 20*. $D\%SdTR$ 30 and $DDdT$ are negative when cooling and positive when heating. Given that sensible cooling or heating capacity is equal to temperature differential multiplied by airflow rate,

therefore,

$$C\%D = D\%SdTR * D\%AR$$

The portion of *maximum magnitude percent dT* ($MaxM\%dT$) 28a in figure 3a that spans from 0% to 40% $C\%D$ 25d' is shown as a square root function from 0% to 100% $D\%SdTR$ 30 relative to $C\%D$ 25d'. Therefore, $D\%AR$ 31 for *minimum airflow rate curve 28a1* is also a square root function for 0% to 40% $C\%D$ 25d' in figure 3b, and is derived from the equation:

$$D\%AR = C\%D / D\%SdTR$$

Diffuser percent airflow rate request ($D\%AR$) 31 is expressed relative to the *diffuser design airflow rate* (DDA) required of *diffuser unit 14* to provide *diffuser design cooling or heating capacity* to *room or zone 20*. For example, if the DDA is 100 L/s then *diffuser unit 14* will have a *diffuser airflow rate request* of 100 L/s when $D\%AR$ 31 is equal to 100%.

The portion of *minimum magnitude percent dT* ($MinM\%dT$) 28b in figure 3a that spans from 0% to 23% $C\%D$ 25d' is shown as a square root function, resulting in $D\%AR$ 31 for *maximum airflow rate curve 28b1* also being a square root function for 0% to 23% $C\%D$ 25d' in figure 3b.

The portion of *maximum airflow rate curve 28b1* in figure 3b that spans from 23% to 100% $C\%D$ 25d' equates to 100% $D\%AR$ 31 (ie corresponding to DDA). Any requested $C\%D$ 25d' value in the range 23% to 100% will be satisfied if *diffuser unit 14* delivers this $D\%AR$ 31 of 100% at a $D\%SdTR$ 30 that corresponds to the same $C\%D$ 25d' value on *minimum magnitude percent dT* ($MinM\%dT$) 28b in figure 3a. For any $C\%D$ 25d' value from 23% to 100%, the corresponding value on the *minimum magnitude*

percent dT ($MinM\%dT$) 28b is equal to the $C\%D$ 25d' value. For example, a $C\%D$ 25d' of 50% (due to a cooling PI output of 50%, as shown in figure 2a) can be realised at a $D\%AR$ 31 of 100% (circle 31b) delivered by diffuser unit 14 at a $D\%SdTR$ of -50% (circle 30b).

5 The portion of *maximum magnitude percent dT* ($MaxM\%dT$) 28a in figure 3a that spans from 40% to 100% $C\%D$ 25d' equates to -100% $D\%SdTR$ 30 (ie to $DDdT$). Any requested $C\%D$ 25d' value in the range 40% to 100% will be satisfied if diffuser unit 14 delivers this $D\%SdTR$ 30 of -100% at a $D\%AR$ 31 on *minimum airflow rate curve* 28a1 in figure 3b that corresponds to the same $C\%D$ 25d' value. For example, a $C\%D$ 25d' of 50% (due to a cooling PI output of 50%, as shown in figure 2a) can be realised at
 10 a $D\%SdTR$ 30 of -100% (circle 30a) delivered by diffuser unit 14 at a $D\%AR$ 31 of 50% (circle 31a).

In figures 3a and 3b, *temperature differential factor* ($F.td$) 32 values vary from 0 on *minimum magnitude percent dT* ($MinM\%dT$) 28b to 1 on *maximum magnitude percent dT* ($MaxM\%dT$) 28a, and from 0 on *maximum airflow rate curve* 28b1 to 1 on *minimum airflow rate curve* 28a1, respectively. For
 15 a given value of $C\%D$ 25d', any pair of $D\%SdTR$ 30 and $D\%AR$ 31 values that satisfy *temperature differential factor* ($F.td$) 32 values of an equal factor value in figures 3a and 3b will deliver the requested $C\%D$ 25d' value.

For example, for an $F.td$ 32 of 0.3, a $C\%D$ 25d' of 50% is satisfied by a $D\%SdTR$ 30 of 64% (circle 30c) in combination with a $D\%AR$ 31 of 78% (circle 31c).

20 For diffuser unit 14 in room cooling mode or in room heating mode, the *temperature differential factor* ($F.td$) 32 is determined by the following $F.td$ equation:

$F.td = 1 - F.d + F.d * F.i$, where
 $F.d$ is a *decreasing factor* (0 to 1), and
 25 $F.i$ is an *increasing factor* (0 to 1).

Note:

- If $F.d$ and $F.i$ are both zero, then $F.td = 1$
- If $F.d = 1$ and $F.i = 0$, then $F.td = 0$
- 30 • If $F.i = 1$, then $F.td = 1$ (regardless of the $F.d$ value)

$F.td$ 32 is a global factor from 0 to 1 that applies to all diffuser units 14 supplied by AHU 1. For each diffuser unit 14, it increases or decreases the magnitude of the *diffuser supply air temperature differential request* $DSdTR$ (and hence varies the associated *diffuser supply air temperature request* at
 35 each *diffuser supply air temperature sensor* 18 relative to associated *room air temperature sensor* 19)

and it offsets this change by lowering or raising, respectively, the associated *diffuser airflow rate request* for each *diffuser unit 14* to deliver the requested *C%D 25d'* value from figure 2b. The larger the difference between *diffuser design pressure (DDP)* and *diffuser minimum pressure (DPmin)* the greater the ability to apply *F.td 32*.

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For a given *C%D* value:

1. *Diffuser percent supply temperature differential request (D%SdTR) 30* is defined as:

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$$D\%SdTR = \text{Max}M\%dT - (1 - F.td) * (\text{Max}M\%dT - \text{Min}M\%dT), \text{ where}$$

$$F.td = 1 - F.d + F.d * F.i$$

2. *Diffuser percent airflow rate request (D%AR) 31* is defined as:

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$$D\%AR = C\%D * / \text{ABS}(D\%SdTR)$$

3. *Diffuser supply temperature differential request (DSdTR)* is defined as:

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$$DSdTR = \text{ABS}(DDdT) * [\text{Max}M\%dT - (1 - F.td) * (\text{Max}M\%dT - \text{Min}M\%dT)], \text{ where}$$

$$F.td = 1 - F.d + F.d * F.i$$

DDdT is set equal to *diffuser design cooling dT* if *room cooling mode* is active, and

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DDdT is set equal to *diffuser design heating dT* if *room heating mode* is active.

Decreasing factor *F.d* has the effect of decreasing *F.td 32* to all *diffuser units 14*, unless *F.i = 1*, thereby decreasing the magnitude of *D%SdTR 30* whilst increasing *D%AR 31* (other than at the two end-points of *C%D 25d'*, discussed below) to maintain a *constant C%D 25d'* value at each *diffuser unit 14*. The value of *F.d* used in the *F.td equation* is the highest magnitude factor (0 to 1) from any *diffuser unit 14* function that produces a decreasing factor, typically via proportional (P) or proportional-integral (PI) control. For example, if a *room or zone 20* is over-cooled as a result of *VAV damper 16* delivering more *diffuser supply air 14a* to satisfy indoor air quality requirements than is required to satisfy cooling demand, then the P or PI output of a "decreasing factor for over-cooling" function may be used to increase *F.d*, which, in turn, reduces *F.td 32* so that both the temperature setpoint for *diffuser supply air*

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temperature sensor 18 and $D\%AR$ 31 are raised (other than at the two end-points of $C\%D$ 25d', and other than if $F.i = 1$, both discussed below) for all *diffuser units* 14 in a bid to reduce over-cooling in any room or zone 20, and this would occur whilst maintaining a constant $C\%D$ 25d' value (assuming steady-state operation) so as not to change *diffuser sensible cooling capacity* delivered to room or zone 20. A

5 "decreasing factor for indoor air quality" is a further example of a decreasing factor that, in this case, would be used to increase the airflow rate of *diffuser supply airflow* 14a without changing the thermal capacity provided, to thereby improve indoor air quality (eg to maintain the CO_2 level in room or zone 20 within prescribed limits) so as save energy by not increasing the outdoor air flowrate demand from AHU 1.

10 Conversely to *decreasing factor* $F.d$, *increasing factor* $F.i$ has the effect of increasing $F.td$ 32 to all *diffuser units* 14, thereby increasing the magnitude of $D\%SdTR$ 30 whilst decreasing $D\%AR$ 31 (other than at the two end-points of $C\%D$ 25d', discussed below) to maintain a constant $C\%D$ 25d' at each *diffuser unit* 14. $F.i$ is the highest magnitude factor (0 to 1) from any *diffuser unit* 14 function that produces an increasing factor, typically via proportional (P) or proportional-integral (PI) control. For

15 example, if $F.td$ is less than 1 due to an active *decreasing factor* $F.d$ (resulting in a raised supply air temperature to all *diffuser units* 14), which in turn causes the humidity in a room or zone 20 to rise above a predetermined threshold (eg 15°C dewpoint), then the P or PI output of an "increasing factor for dehumidification" function (assuming that humidity is measured) serving that room or zone 20 will increase $F.i$, and hence $F.td$ 32, so that both the temperature setpoint for *diffuser supply air temperature*

20 sensor 18 and *diffuser* $D\%AR$ 31 are reduced for all *diffuser units* 14 in a bid to reduce the humidity level in all rooms or zones 20, and this would occur whilst maintaining a constant $C\%D$ 25d' value (assuming steady-state operation) so as not to change *diffuser sensible cooling capacity* delivered. Other examples of $F.i$ include an "increasing factor to reduce system pressure" and an "increasing factor to reduce diffuser noise".

25 $F.td$ 32 has the effect of increasing or decreasing the magnitude of $D\%SdTR$ 30, and of offsetting this by an equivalent decrease or increase in $D\%AR$ 31, respectively, for all points other than the two end-points (ie at 0% and 100%) of $C\%D$ 25d', and it has a diminishing effect as each of these end-points is approached. Consequently, 0% and 100% $C\%D$ 25d' (with the latter corresponding to the

30 *diffuser design capacity* of *diffuser unit* 14) may be achieved regardless of the $F.td$ 32 factor value.

Because a change in the magnitude of $D\%SdTR$ 30 brought about by a change in $F.td$ 32 is offset by an equivalent but opposite change in *percent airflow rate request* 31, $C\%D$ 25d' remains constant (assuming steady-state operating conditions) regardless of changes to $F.td$ 32. In other words, *diffuser unit* 14 delivers a constant *diffuser sensible cooling capacity* across the full factor range of $F.td$

35 32.

In figure 3a, the magnitude of $D\%SdTR$ 30 increases to maximum magnitude as $C\%D$ 25d' increases to 40%. A $C\%D$ 25d' value other than 40% may be used for this to be achieved. For example, in room heating mode (not shown) of room or zone 20, if heating is supplied by heat exchanger 9 then it may be preferable for $D\%SdTR$ 30 to only reach its maximum magnitude when heating percent demand reaches 100% so as to minimise stratification of warm air in room or zone 20 under part load conditions. Under part-load conditions, this will increase the airflow rate and reduce the supply air temperature, reducing buoyancy and improving mixing of warm supply air to a low level, but at the expense of increased fan energy. In contrast, if heating is not by heat exchanger 9, but by a trim heater in duct 12 or branch duct 12a (not shown), then retaining 40% $C\%D$ 25d' for the point at which $D\%SdTR$ reaches its maximum magnitude may be preferable, as this will minimise diffuser supply air 14a airflow rate, and hence the amount of energy required by the trim heater to heat pre-cooled AHU supply air 9a.

Figures 3c and 3d are graphs illustrating diffuser percent pressure request ($D\%PR$) 33 relative to diffuser design pressure (DDP) when operating with a fully open VAV damper 16, and diffuser percent damper flow request ($D\%DFR$) 34 when operating at diffuser minimum pressure ($DPmin$), respectively, of a diffuser unit 14, each as a function of cooling percentage demand 25d' determined in figure 2b. These two graphs combined make up $D\%AR$ 31 in figure 3b, and are derived from it.

Diffuser percent maximum pressure ($D\%Pmax$) 28b1' is defined by a diffuser percent pressure request 33 value of 100% when diffuser unit 14 is operating at its design airflow rate (DDA) with a fully open VAV damper 16. This equates to the diffuser design pressure (DDP). Diffuser percent minimum pressure ($D\%Pmin$) 28a2 is the minimum percent pressure at which diffuser unit 14 may operate, and is shown in figure 3c to have a diffuser percent pressure request 33 value of 33% (circle 35a). For example, a diffuser unit 14 that has a diffuser design pressure (DDP) of 30 Pa and a diffuser minimum pressure ($DPmin$) of 10 Pa has that minimum corresponding to a diffuser percent pressure request 33 value of 33% (ie 10 Pa is 33% of 30 Pa), as shown in figure 3c, circle 35a. Since pressure is proportional to airflow rate squared, the equivalent $D\%AR$ value in figure 3b is 57%, shown by dashed line diffuser percent airflow rate @ minimum pressure request ($D\%MinPAR$) 28a2'', which is the airflow rate percent delivered by a fully open VAV damper 16 operating at diffuser minimum pressure ($DPmin$) as measured by diffuser static pressure sensor ($Pst.d$) 17.

In the above example where $C\%D$ 25d' is equal to 50%, $D\%PR$ 33 is equal to 33% (circle 35a).

Diffuser percent damper flow request ($D\%DFR$) 34 is the percent flow rate request relative to the flow rate of a fully open VAV damper 16 operating at diffuser minimum pressure ($DPmin$) as measured by diffuser supply pressure sensor 17.

In the above example where $C\%D$ 25d' is equal to 50%, $D\%DFR$ 34 is equal to 87% (circle 36a).

Any $D\%AR$ 31 in figure 3b that is greater than *diffuser percent airflow rate @ minimum pressure request ($D\%MinPAR$) 28a2''* will require $D\%DFR$ 34 in figure 3d to be equal to 100% (ie VAV damper 16 is fully open) and a $D\%PR$ 33 value in figure 3c equal to the following:

$$D\%PR = SQ(D\%AR)$$

Any $D\%AR$ 31 in figure 3b that is less than or equal to *diffuser percent airflow rate @ minimum pressure request ($D\%MinPAR$) 28a2''* (ie the airflow rate delivered by a fully open VAV damper 16 operating at *diffuser minimum pressure ($DPmin$)* as measured by *diffuser supply pressure sensor 17*) will require $D\%PR$ 33 in figure 3c to be equal to *diffuser percent minimum pressure ($D\%Pmin$) 28a2* and a $D\%DFR$ 34 value in figure 3d equal to the following:

$$D\%DFR = D\%AR / D\%MinPAR.$$

Diffuser adjusted percent damper flow request ($DA\%DFR$) is the *diffuser percent damper flow request ($D\%DFR$ 34)* adjusted to achieve the $C\%D$ required of *diffuser unit 14* once both the actual temperature differential relative to the requested temperature differential and the actual pressure relative to the requested pressure at *diffuser unit 14* are corrected for:

$$DA\%DFR = D\%DFR * D\%SdTR * DDdT / (TSA - TRA) * (D\%PR * DDP / DPA)^{0.5}, \text{ where:}$$

$DDdT$ is equal to *diffuser design cooling dT* or *diffuser design heating dT*, for *diffuser cooling mode* or *diffuser heating mode*, respectively.

TSA is the *diffuser actual supply air temperature*, as measured by *diffuser supply air temperature sensor ($T.sd$) 18*.

TRA is the *actual room temperature*, as measured by *room temperature sensor ($Troom$) 19*.

DDP is equal to *diffuser design cooling pressure* or *diffuser design heating pressure*, for *diffuser cooling mode* or *diffuser heating mode*, respectively.

DPA is the *diffuser actual pressure*, as measured by *diffuser static pressure sensor ($Pst.d$) 17*.

For a given $C\%D$, $DA\%DFR$ maintains constant sensible cooling or heating capacity delivery to room or zone 20 regardless of fluctuations in *actual diffuser supply air temperature TSA* or *diffuser actual supply air pressure PSA*.

5 For *diffuser unit 14*, the *diffuser percent damper request (D%DR)* of *VAV damper 16* relative to the fully open position is determined by the following formula:

$$D\%DR = DM * DA\%DFR, \text{ where:}$$

10 DM is a *damper mapping function*, determined experimentally for *diffuser unit 14*, relating $DA\%DFR$ to $D\%DR$.

The *diffuser flow (DF)* rate delivered by *VAV damper 16* of a *VAV diffuser unit 14* is determined by the equation:

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$$DF = DA\%DFR * DFP, \text{ where}$$

DFP is determined experimentally for *diffuser unit 14*. It is the flow rate to static pressure relationship (as measured by *diffuser static pressure sensor 17*) of *VAV diffuser unit 14* when
20 *VAV damper 16* is fully open.

A *diffuser unit 14* may either be a *master* or a *slave* diffuser. If a *room or zone 20* is too large to be served by a single *diffuser unit 14* then multiple *diffuser units 14* may be required. However, it may
25 be preferable for these *diffuser units 14* not to work independently of one another, as each will try to achieve independent thermostatic temperature control of the same *room or zone 20*. In this case one *diffuser unit 14* may be assigned “master” status, and for the control of *VAV damper position*, *temperature differential request*, *pressure request* and associated offsets for *room or zone 20* to be from this *master diffuser unit 14* only. *Diffuser controller 90* issues the relevant *VAV damper 16* position
30 request to the *master diffuser unit 14* as well as to the *slave diffuser units 14* of that *room or zone 20*. Control functions for the *slave diffuser units 14* implement the *VAV damper 16* position request either directly (ie they mimic the *VAV damper position* of the *master diffuser unit 14*) or substantially (ie they modify the *VAV damper position* of the *master diffuser unit 14*). The latter may apply if, for example, the *slave diffuser units 14* include integrated temperature sensors for *room or zone 20* air temperature, so
35 that proportional (P) control may be used, relative to the average *room or zone 20* air temperature, to

provide proportional adjustment based on the integrated *room or zone 20* air temperature sensor located in each *slave diffuser unit 14* to account for local temperature variations in *room or zone 20*.

5 *Diffuser unit 14* can either be a *lead diffuser* or a *follower diffuser*, unless it is an *excluded diffuser*, such as a diffuser that has been designated as a *rogue diffuser* or is considered to be a *potentially rogue diffuser* (described further below). A *lead diffuser* can be any one of a *lead temperature diffuser*, a *lead pressure diffuser*, or both a *lead temperature and lead pressure diffuser*.

10 A *lead pressure diffuser* (or group of *lead pressure diffusers*, discussed below) determines the *AHU pressure setpoint* in relation to *AHU supply air pressure sensor 11a* (which *supply air fan 7* is separately controlled to, typically via PI control) or directly determines the *fan speed setpoint* of *supply air fan 7*.

15 A *lead pressure diffuser 14* substantially always has its *VAV damper 16* fully open unless the required airflow rate is less than the airflow rate that fully open *VAV damper 16* would deliver when *lead pressure diffuser unit 14* is operating at *diffuser minimum static pressure* (typically 10 Pa). To achieve this, other than when the *lead pressure diffuser 14* is operating at *diffuser minimum static pressure*, the *diffuser actual supply air pressure PSA* (ie actual static pressure) at the *lead pressure diffuser 14* may be controlled (via *AHU pressure setpoint* adjustment or by *fan speed* adjustment) to
20 equal that pressure that delivers the *diffuser adjusted percent damper flow request (DA%DFR)* when the *VAV damper 16* is fully open.

The change (if any) in static pressure required for the *VAV damper 16* of *diffuser unit 14* to be fully open can be determined by establishing a *diffuser offset pressure request* (positive, zero or
25 negative) to be added to the *diffuser actual supply air pressure PSA* as measured by *diffuser static pressure sensor 17*. If the *diffuser unit 14* is the *lead pressure diffuser*, then the *diffuser offset pressure request* can be used to alter the *AHU pressure setpoint*, as measured by *AHU supply pressure sensor 11a* (if present), or to directly drive the supply air fan speed, in each case to provide to *lead pressure diffuser unit 14* a *diffuser actual supply air pressure PSA* at *diffuser static pressure sensor 17* that delivers an
30 airflow rate to *diffuser unit 14* equal to the *diffuser adjusted percent damper flow request (DA%DFR)* when *VAV damper 16* is fully open.

The *diffuser offset pressure request* for each *diffuser unit 14* may be established through the following process:

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Diffuser actual pressure (measured by diffuser static pressure sensor 17) is subtracted from

diffuser adjusted pressure request to determine diffuser pressure offset request, where:

1. *Diffuser adjusted pressure request is equal to:*

- a. *Diffuser pressure request (DPR), if diffuser temperature differential request divided by diffuser actual temperature differential is greater than or equal to 1; or*
- b. *Diffuser pressure request (DPR) multiplied by diffuser temperature differential request squared, and divided by diffuser actual temperature differential squared, if diffuser temperature differential request divided by diffuser actual temperature differential is less than 1;*

and

2. *Diffuser pressure request (DPR) is equal to $D\%PR_{33}$ multiplied by diffuser design pressure (DDP);*

Diffuser adjusted pressure request is limited not to exceed diffuser design pressure (DDP) and not to be less than diffuser minimum static pressure (DPmin).

A lead temperature diffuser 14 substantially always has its VAV damper 16 fully open unless doing so would exceed the diffuser design airflow rate DDA.

A lead temperature diffuser (or a group of lead temperature diffusers, discussed below) may determine the AHU supply temperature setpoint in relation to AHU supply air temperature sensor 11 (or in relation to diffuser supply air temperature sensor 18) for AHU mechanical cooling mode or AHU mechanical heating mode; or it may directly determine the cooling or heating capacity output of heat exchanger 9, in which case the AHU mechanical cooling mode or AHU mechanical heating mode will correspond to the diffuser cooling mode or diffuser heating mode, respectively. If economiser active is enabled (discussed further below) then it may determine both the AHU supply temperature setpoint as measured by AHU supply temperature sensor 11 (or, alternatively, directly determine the cooling output of heat exchanger 9) and the economiser damper temperature setpoint in relation to AHU supply air temperature sensor 11, or it may directly drive the economiser damper assembly (4, 5 and 6 in figure 1).

In diffuser cooling mode, the AHU supply temperature setpoint ensures that the temperature of diffuser supply air 14a is less than or equal to the air temperature of room or zone 20. It, therefore, may be set by diffuser controller 90 to have an AHU maximum supply temperature setpoint equal to the

cooling setpoint of room or zone 20 (ie, equal to the *global room temperature setpoint* plus the deadband divided by 2, which for previous examples used may be calculated as 22.5°C plus 1 K divided by 2, which equals 23°C).

In *diffuser heating mode*, the *AHU supply temperature setpoint* ensures that the temperature of *diffuser supply air 14a* is greater than or equal to the air temperature of room or zone 20. It, therefore, may be set by *diffuser controller 90* to have an *AHU minimum supply temperature setpoint* equal to the *heating setpoint of room or zone 20* (ie, equal to the *global room temperature setpoint* plus the deadband divided by 2, which for previous examples used may be calculated as 22.5°C minus 1 K divided by 2, which equals 22°C).

The change (if any) in supply air temperature required for the *VAV damper 16* of a *diffuser unit 14* to be fully open can be determined by establishing a *diffuser offset temperature request* (positive, zero or negative) to be added to the *diffuser actual supply air temperature TSA* as measured by *diffuser temperature sensor 18*. If the *diffuser unit 14* is the *lead pressure diffuser*, then the *diffuser offset temperature request* can be used to alter the *AHU supply temperature setpoint*, as measured by *AHU supply temperature sensor 11* (if present), or alternatively to directly determine the cooling or heating output of *heat exchanger 9*, to control the supply air temperature delivered to *lead temperature diffuser 14*, as measured by *diffuser supply temperature sensor 18*, so that the *diffuser actual temperature differential* is equal to a *diffuser adjusted supply temperature differential request*. The *diffuser adjusted supply temperature differential request* is the *diffuser supply air temperature request* adjusted so that *diffuser unit 14* delivers a cooling or heating capacity equal to the *cooling percent demand C%D* (or *heating percent demand*, as the case may be) at an absolute temperature differential that is less than the absolute *diffuser supply temperature differential request* if the *diffuser actual pressure 17* is greater than the *diffuser adjusted pressure request* (ie if excess static pressure is available at *diffuser unit 14*) so as to minimise cooling/heating plant energy consumption by reducing the requested absolute temperature differential and compensating by delivering an increased airflow rate from *lead temperature diffuser unit 14*, but without requiring an increase in *diffuser actual supply air pressure PSA*, as this would waste fan energy. For fully open *VAV damper 16*, the available increase in diffuser airflow rate is proportional to the square root of the difference between the *diffuser adjusted pressure request* and the *diffuser actual supply air pressure PSA*, and hence the *diffuser requested supply air temperature differential* may be reduced accordingly. The increase in airflow rate and corresponding reduction in temperature differential for *lead temperature diffuser 14* is limited to ensure that the *lead temperature diffuser 14* airflow rate does not exceed the *diffuser design airflow rate DDA*.

The *diffuser offset temperature request* for each *diffuser unit 14* is determined through the following process:

Diffuser actual temperature differential (the temperature at *diffuser supply temperature sensor 18* minus *room air temperature 19*) is subtracted from *diffuser adjusted supply temperature differential request* to determine *diffuser temperature offset request*, where:

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1. *Diffuser adjusted supply temperature differential request* is equal to:

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- a. *Diffuser supply temperature differential request (DSdTR)*, if *diffuser pressure request (DPR)* divided by *actual diffuser pressure (DPA)* measured by *diffuser static pressure sensor (Pst.d) 17* is greater than or equal to 1; or
- b. *Diffuser supply temperature differential request (DSdTR)* multiplied by the square root of *diffuser pressure request (DPR)* divided by the square root of *actual diffuser pressure (DPA)* (measured by *diffuser static pressure sensor 17*), if *diffuser pressure request (DPR)* divided by *actual diffuser pressure (DPA)* is less than 1;

15

and

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2. *Diffuser supply temperature differential request (DSdTR)* is equal to *D%SdTR 30* multiplied by *diffuser cooling design temperature differential (-ve)* or *diffuser heating design temperature differential (+ve)*, as the case may be.

and

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3. The magnitude of *diffuser adjusted supply air temperature differential request* is limited to never exceed the magnitude of *DDdT*.

From the above, it follows that:

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1. A *follower diffuser 14* always has its *VAV damper 16* partially throttled, other than at the transition of becoming, or ceasing to be, a *lead diffuser*; and

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2. The *diffuser pressure offset* of a *follower diffuser 14* or of a *lead temperature diffuser 14* (unless the *lead pressure diffuser* and *lead temperature diffuser* are one and the same diffuser) is always less (ie more negative) than the *diffuser pressure offset* of the *lead pressure diffuser*; and the *diffuser temperature offset* of a *follower diffuser 14* or of a *lead pressure diffuser 14* (unless the *lead pressure diffuser* and *lead temperature diffuser* are one and the same diffuser) is always less

(ie more negative) than or greater (ie more positive) than the *diffuser temperature offset* of the *lead temperature diffuser* in *heating mode* or *cooling mode*, respectively; and

3. In steady state, the *pressure offset* and the *temperature offset* of the *lead pressure diffuser* and *lead temperature diffuser* are substantially equal to zero.

A diffuser unit control system in accordance with embodiments of the invention described allows both optimised supply air pressure and optimised supply air temperature to be achieved even when the lead pressure and lead temperature diffusers are different diffuser units, by offsetting excessive supply-to-room temperature differentials received by a diffuser unit with a reduced diffuser pressure request from that diffuser unit, and similarly by offsetting excessive pressure received by a diffuser unit with a reduced supply-to-room temperature differential request from that diffuser unit. Optimised supply air pressure ensures that the VAV damper of at least one diffuser unit (the lead pressure diffuser) operates fully open unless the required airflow rate of the diffuser unit is less than the airflow rate that the diffuser unit's fully open VAV damper would deliver when the diffuser unit is operating at its minimum permissible static pressure (typically 10 Pa). This minimises system static pressure, and hence reduces fan energy. Optimised supply air temperature ensures that the supply-to-room temperature differential request of the lead temperature diffuser unit is reduced if there is excess supply pressure available at the diffuser unit (assuming that it is not also the lead pressure diffuser unit) to provide the required thermal capacity by further opening the VAV damper of that diffuser unit. This may allow the AHU to deliver a warmer supply air temperature in cooling mode (or a lower one in heating mode) than would otherwise be called for by the lead temperature diffuser, thereby saving chiller or heating plant energy. The combination of optimal supply pressure and optimal supply temperature achieved via respective temperature and pressure offsets is particularly advantageous in non-static regain (such as equal friction) duct design applications and where duct related temperature losses differ between diffuser units, as happens in most HVAC applications. In contrast, design guides and technical brochures for VAV diffuser units of the prior art typically call for static regain (also called equal pressure) duct design to each diffuser (with a tolerance between diffusers of just +10% and -20% static pressure) and associated controls do not compensate by raising or lowering supply air temperature requirements if excess pressure is available to the diffuser unit with the lowest or highest supply temperature demand in cooling or heating mode, respectively. This wastes chiller and heating plant energy, as it calls for unnecessarily cold supply air when cooling (wasting energy on unrequired latent cooling), and unnecessarily hot supply air when heating (making supply air even more buoyant, which increases stratification of heat to a high level, reducing heating effectiveness at a low level – where it is required).

For all *diffuser units 14*, the *diffuser temperature offset requests* and the *diffuser pressure offset requests* are each ranked from lowest (most -ve) to highest (most +ve) in a *temperature rank* and a *pressure rank*, respectively, although the position of each may be weighted, for example for diffuser priority (whether it serves a high priority, medium priority or low priority *room or zone 20*) in a *weighted temperature rank* and a *weighted pressure rank*, other than *rogue diffusers*, which may be excluded from the *rank*, and *potentially rogue diffusers*, which may be removed from the *rank*. *Rogue diffusers* (eg diffusers that have been identified as being undersized, faulty, or to have sensors that are out of calibration) may be designated as *rogue diffusers* so as not to contribute erroneously to the *ranking lists*, and the *offsets (pressure, temperature, etc)* from *potentially rogue diffusers* (that group of diffusers that is most likely to include diffusers that ought to be designated, but have not been identified as, as *rogue diffusers*) may be automatically removed from the *ranking lists* by virtue of being in a top percentile of the most positive or most negative value *offsets* that is therefore deemed as most likely to include *potentially rogue diffusers*.

The *diffuser unit 14* with the highest (most +ve) *pressure offset request* is the *diffuser unit 14* that has the greatest demand for an increase in supply pressure, or the least demand for a decrease in supply pressure, and is therefore that *diffuser unit 14* that *AHU supply air 9a* pressure and associated *supply air fan 7* speed ought to be controlled to, unless it is an *excluded diffuser* or a diffuser that is ranked lower in the *weighted pressure rank* due to being a lower priority diffuser. For this reason, the greatest (most +ve) *ranking value* in the *pressure rank* or *weighted pressure rank*, excluding *excluded diffusers*, is selected to be controlled to, with the *diffuser unit 14* that this *pressure offset request* has come from thereby automatically becoming the *lead pressure diffuser* that is controlled to and that, therefore, operates with its *VAV damper 16* substantially fully open (unless doing so would exceed the *diffuser design airflow rate DDA*). The highest (most +ve) *pressure offset request* from the *pressure rank* or the *weighted pressure rank* is then entered as a variable into either an *AHU supply fan speed PI controller* or an *AHU supply pressure PI controller*, in each case with zero as the controller setpoint, to determine either an *AHU supply fan speed PI output* or an *AHU supply pressure PI output* to directly determine the fan speed of *supply fan 7*, in which case it is known as the *AHU supply fan speed setpoint*, or to determine the supply pressure setpoint as measured by *AHU supply pressure sensor 11a* for *supply fan 7*, in which case it is referred to as the *AHU supply pressure setpoint*. The rate of change of the *AHU supply fan speed PI output* or the *AHU supply pressure PI output* (as the case may be) may be limited to prevent sudden changes to *supply fan 7* speed or *pressure setpoints* to prevent surging or overshoot by *supply fan 7*. Alternatively, if averaging is to be used (discussed below) then the *diffuser units 14* with the highest (most +ve) *ranking values* are designated as a *lead group of pressure diffusers*, and an *average pressure offset request* for the *lead group of pressure diffuser units 14* is entered as a variable into either the *AHU supply fan speed PI controller* or the *AHU supply pressure PI controller*.

The highest (most +ve) *pressure offset request* from the *pressure rank* or the *weighted pressure rank* may, furthermore, be entered from the relevant *pressure rank* as a variable into a *pressure relief PI controller* with a negative value (such as, for example, -5, which would correspond to an overpressure of 5 Pa) as the controller setpoint (also called the “process value”) to determine a *minimum diffuser damper position or airflow rate PI output* in percent that is communicated to all *diffuser units 14* that then determines or overrides the minimum VAV *damper 16* position relative to fully open, or to the minimum airflow rate relative to the *diffuser design airflow rate DDA*, of each *diffuser unit 14* to provide PI control that bleeds, through *diffuser units 14*, over-pressure in excess of 5 Pa, in this example, in relation to the *lead pressure diffuser*. The *minimum diffuser damper position or airflow rate PI output*, in turn, may be weighted by each *diffuser unit 14* according to its priority (eg a low priority diffuser may have this value multiplied by 2; a medium priority diffuser may have this value multiplied by 1.5; and a high priority diffuser may not have this value multiplied at all) so as to weight over-pressure relief in favour of lower priority *diffuser units 14*. Pressure relief may be required if *supply air fan 7*, for example, is limited to a minimum speed that delivers a greater rate of *AHU supply air 9a* than the combined minimum airflow rate of all *diffuser units 14* (eg the minimum airflow rate of direct expansion (DX) AHU systems is often limited to 40% of AHU design airflow rate, whereas each *diffuser unit 14* may be set to a minimum airflow rate of 20% relative to its *diffuser design airflow rate DDA*).

If *diffusers units 14* are connected to a fan-coil unit (FCU), rather than an AHU, then fan speed control (if any) is usually staged, typically in three steps. The speed control described above for *supply air fan 7* is suitable for continuously variable fan speed control, but not for staged fan speed control or constant fan speed FCU or AHU applications.

For constant *supply air fan 7* speed applications the *AHU supply fan speed PI controller* or *AHU supply pressure PI controller* is not used. Instead, only the *pressure relief PI controller* is used, preferably controlling to a process value of zero.

For staged *supply air fan 7* speed applications the *AHU supply fan speed PI controller* or *AHU supply pressure PI controller* is not used. Instead, stepped control is used, stepping up to the next highest fan speed whenever the highest (most +ve) *pressure offset request* from the *pressure rank* or *weighted pressure rank* is greater than zero, and stepping down whenever the highest (most +ve) *pressure offset request* from the *pressure rank* or *weighted pressure rank* is less than a predetermined negative value, the magnitude of which is substantially equal to the maximum change in static pressure that a single step fan speed change would cause at any given airflow rate. (This can be determined directly from FCU or AHU fan curves.) For example, if a single step change in fan speed can result in a maximum change in fan static pressure of 30 Pa whilst fan airflow rate remains constant then the speed of *supply air fan 7* should be controlled to step down to the next lowest fan speed whenever the highest

(most +ve) *pressure offset request* from the *pressure rank* or *weighted pressure rank* is less than (more -ve than) -30. Additionally, the process value of the *pressure relief PI controller* should be set to the same value (-30 in the above example) to provide over-pressure relief through *diffuser units 14* once the lowest stepped speed of *supply air fan 7* has been reached.

5 In *diffuser heating mode*, the *diffuser unit 14* with the highest (most +ve) *ranking value* in the *temperature rank* or *weighted temperature rank*, or in *diffuser cooling mode*, the *diffuser unit 14* with the lowest (most -ve) *ranking value* in the *temperature rank* or *weighted temperature rank*, is the *diffuser unit 14* (or *weighted diffuser unit 14*) – excluding *excluded diffusers* - that has the greatest demand for an increase or decrease, respectively, or the least demand for a decrease or increase,
10 respectively, in *supply temperature*, and is, therefore, that *diffuser unit 14* that *AHU supply temperature setpoint* or relevant heating or cooling output request to *heat exchanger 9* ought to be controlled to. It is for this reason that the greatest (most +ve) *ranking value* in *diffuser heating mode*, or the lowest (most -ve) *ranking value* in *diffuser cooling mode*, is controlled to from the *temperature rank* or *weighted temperature rank*, with the *diffuser unit 14* that this temperature offset request has come
15 from thereby automatically becoming the lead temperature diffuser that is controlled to. The highest (most +ve) or lowest (most -ve) *temperature offset request* for *diffuser heating mode* or *diffuser cooling mode*, respectively, is thus entered as a variable from the *temperature rank* or *weighted temperature rank* into either an *AHU supply temperature setpoint PI controller* or an *AHU cooling/heating capacity controller*, in each case with zero as the controller setpoint, to determine either the *AHU supply*
20 *temperature setpoint* (used as the *AHU supply air heating setpoint* or *AHU supply air cooling setpoint* in *AHU mechanical heating mode* or *AHU mechanical cooling mode*, and associates this with an *AHU mechanical heating PI function* and an *AHU mechanical cooling PI function*, respectively, in relation to *AHU supply air temperature sensor 11* or in relation to *diffuser supply air temperature sensor 18*), or to directly determine the heating or cooling capacity output of *heat exchanger 9* and correspondingly
25 designating *AHU mechanical heating mode* or *AHU mechanical cooling mode* to match *diffuser heating mode* or *diffuser cooling mode*, respectively. Alternatively, if averaging is to be used (discussed below) then the *diffuser units 14* with the highest (most +ve) *ranking values* are selected as the *lead group of temperature diffusers*, and the *average temperature offset request* for the *lead group of temperature diffuser units 14* is entered as a variable into either the *AHU cooling/heating capacity controller* or the
30 *AHU supply temperature controller*.

 The setpoint outputs from the *AHU supply temperature setpoint PI controller* and the *AHU supply pressure setpoint PI controller* are cascade setpoints for *AHU supply air temperature setpoint control* and *AHU supply pressure setpoint control*, which means that any heat gain or loss, or pressure
35 loss, respectively, between *AHU 1* and the *lead diffuser unit determining the AHU supply temperature*

setpoint or *AHU supply pressure setpoint*, as the case may be, is automatically adjusted for.

5 It is often desirable to create a *lead temperature diffuser group* and a *lead pressure diffuser group diffuser*, rather than having a single *lead temperature diffuser* and a single *lead pressure diffuser*, so that *AHU supply air 9a* temperature and pressure requests are determined by an average of a plurality of *lead temperature diffusers* and an average of a plurality of *lead pressure diffusers*. This reduces the risk of plant demand (and hence plant energy consumption) being solely determined by a
10 diffuser that may be atypical (it may have an unusually high demand, perhaps because a warm kettle is affecting the temperature sensor of that *room or zone 20*) or due to a *diffuser unit 14* that may be somewhat undersized and hence tends to request a high capacity most of the time, even when the load on all other *room or zone 20s* served by AHU 1 is low and especially if the zone served by the *diffuser unit 14* is of low importance (eg it is a transient space, such as a corridor).

15 Where proportional-integral (PI) control, as opposed to proportional (P) only, is used for *room or zone 20* temperature control, averaging the *diffuser supply air 14a* temperature and pressure requests would ordinarily lead to some of the *diffuser units 14* in the *lead temperature group* and the *lead pressure group* being unsatisfied. This would occur because the *diffuser unit 14* with the highest PI output of its *lead group* would continue to request a greater response from the system (known as
20 “integral windup”), as its PI output would be greater than its *lead group* average, even if the average across all *diffuser units 14* included in that *lead group* were satisfied. To alleviate this, when a *diffuser unit 14* is included in the *lead temperature group* or *lead pressure diffuser group*, as the case may be, the average error between *room or zone 20* setpoint temperature and the air temperature of *room or zone 20* is calculated across all the *diffuser units 14* included in that *lead group*. This error is used to
25 calculate an updated integral value for the *room or zone 20* temperature PI control of each *diffuser unit 14* in the applicable *lead group*. This way, if the *lead group* of *diffuser units 14* included in the average is not in aggregate satisfied then the *room or zone 20* PI controllers of the *diffuser units 14* in that *lead group* will increase their response together. When a *diffuser unit 14* leaves the applicable *lead group*, its integral is released to operate as normal.

30

If *economiser status* is active, then the deadband between the *AHU supply temperature setpoint* and *economiser damper temperature setpoint* may be set equal to *economiser deadband request EDR* (the determination of which is discussed below), in which case the *AHU supply temperature setpoint* is typically less than the *room or zone 20* temperature. *Outdoor air 23* of a temperature that is
35 less than this setpoint not only has lower energy content than *return air 24a* but is also cold enough to

provide cooling to *room or zone 20*. The deadband between the *AHU supply temperature setpoint* and the *economiser damper temperature setpoint* is the temperature range within which *outdoor air 23*, as measured by *AHU supply temperature sensor 11* (or in an alternative embodiment by *diffuser supply temperature sensor 18*), provides 100% free-cooling to fully cover the thermal load of *room or zone 20*,
5 as *outdoor air 23* is drawn through *fully open outdoor air damper 6* with neither mechanical cooling from *heat exchanger 9* nor mixing of warmer return air by *return air damper 5* required or permitted (the former would waste plant energy cooling *AHU supply air 9a*; the latter would waste fan energy recirculating return air).

10 If *economiser status* is active, then the *AHU supply temperature setpoint* and *economiser damper temperature setpoint* may be assigned by *plant controller 93* or *diffuser controller 90* as follows:

1. For *supplementary mechanical cooling operation*, the *temperature offset request* with the lowest (most -ve) value in the *temperature rank* or *weighted temperature rank* is controlled to,
15 with the *diffuser unit 14* that this *temperature offset request* has come from automatically becoming the *lead temperature diffuser* (ie that is controlled to). The lowest (most -ve) *temperature offset request* is thus entered as a variable from the *temperature rank* or *weighted temperature rank* into either an *AHU supply temperature setpoint PI controller* or an *AHU cooling capacity PI controller*, in each case with zero as the controller setpoint, to
20 determine either the *AHU supply temperature setpoint* (used as the *AHU supply air cooling setpoint* and associates this with the *AHU mechanical cooling PI function* in relation to *AHU supply air temperature sensor 11* or to *diffuser supply air temperature sensor 18*), or to directly determine the cooling capacity output of *heat exchanger 9*.
2. For *economiser damper operation*, an *economiser offset request* with the lowest (most -ve)
25 value in an *economiser rank* or *weighted economiser rank* is controlled to, wherein for each *diffuser unit 14* the *economiser offset* is determined by subtracting the supply air temperature as measured by *diffuser temperature sensor 18* from the *diffuser economiser temperature differential request DEdTR* minus the *minimum economiser deadband* (preferably of 1 K, discussed further below). The *diffuser unit 14* that this *economiser offset request* has come
30 from automatically becomes the *lead economiser diffuser* that is controlled to, this being the same *diffuser unit 14* determined as the *lead temperature diffuser* for *supplementary mechanical cooling operation* above. The lowest (most -ve) *economiser offset request* is thus entered as a variable from the *economiser rank* or *weighted economiser rank* into an *economiser damper PI controller* (either an *economiser temperature setpoint PI controller* or an
35 *economiser damper PI controller*, in each case with zero as the controller setpoint, to determine either the *economiser temperature setpoint* in relation to *AHU supply air*

temperature sensor 11, or to directly determine economiser damper modulation from 0% economiser damper to 100% economiser damper).

5

Plant controller 93 or diffuser controller 90 issues an economiser active status as long as AHU mechanical cooling mode is active and the outdoor air temperature and/or enthalpy conditions favour economiser operation and the economiser position request from the economiser damper PI controller is not for outdoor air damper 6 to be fully closed.

10

Plant controller 93 or diffuser controller 90 may determine economiser position requests for outdoor air damper 23 as follows:

1. *If economiser status is not active then an economiser position request of fully closed is issued for outdoor air damper 23.*
2. *If economiser status is active and AHU supply air 9a temperature, as measured by AHU supply air temperature sensor 11 (or in an alternative embodiment by diffuser supply temperature sensor 18), is greater than or equal to the economiser damper temperature setpoint then an economiser position request of fully open is issued for outdoor air damper 23.*
3. *If economiser status is active and AHU supply air 9a temperature, as measured by AHU supply air temperature sensor 11 (or in an alternative embodiment by diffuser supply temperature sensor 18), is less than the economiser damper temperature setpoint then plant controller 93 or diffuser controller 90 activates the economiser damper PI function to modulate the economiser position request for outdoor air damper 23 between fully open for an AHU supply air 9a temperature (as measured by AHU supply air temperature sensor 11, or in an alternative embodiment by diffuser supply temperature sensor 18) above the economiser damper temperature setpoint, and fully closed for an AHU supply air 9a temperature below the economiser damper temperature setpoint.*

30

The assignment, by plant controller 93 or diffuser controller 90, of the AHU supply temperature setpoint, the economiser damper temperature setpoint, as well as the designation of the AHU mechanical cooling PI function, the AHU mechanical heating PI function, and the economiser damper PI function, may result in the following AHU supply air 9a temperature control as measured by AHU supply air temperature sensor 11 (or in an alternative embodiment by diffuser supply temperature sensor 18), and economiser position requests for outdoor air damper 23:

35

If *economiser status* is not active and AHU supply air 9a temperature exceeds the AHU supply temperature setpoint then an *economiser position request* of fully closed is issued for outdoor air damper 23, a heating output of no mechanical heating is issued to heat exchanger 9, and the AHU mechanical cooling PI function is activated to regulate the cooling output of heat exchanger 9 to maintain the temperature of AHU supply air 9a at the AHU supply temperature setpoint.

If *economiser status* is active and AHU supply air 9a temperature exceeds the AHU supply temperature setpoint then an *economiser position request* of fully open is issued for outdoor air damper 23, a heating output of no mechanical heating is issued to heat exchanger 9, and the AHU mechanical cooling PI function is activated to regulate the cooling output of heat exchanger 9 to maintain the temperature of AHU supply air 9a at the AHU supply temperature setpoint.

If *economiser status* is active and AHU supply air 9a temperature is between the AHU supply temperature setpoint and the *economiser damper temperature setpoint* then an *economiser position request* of fully open is issued for outdoor air damper 23 and a mechanical cooling output of no mechanical cooling and a mechanical heating output of no mechanical heating are issued for heat exchanger 9.

If *economiser status* is not active and AHU supply air 9a temperature is less than the AHU supply temperature setpoint then an *economiser position request* of fully closed is issued for outdoor air damper 23, a mechanical cooling output of no mechanical cooling is issued for heat exchanger 9, and the AHU mechanical heating PI function is activated to regulate the heating output of heat exchanger 9 to maintain the temperature of AHU supply air 9a at the AHU supply temperature setpoint.

If *economiser status* is active and AHU supply air 9a temperature is less than the *economiser damper temperature setpoint* then a mechanical cooling output of no mechanical cooling and a mechanical heating output of no mechanical heating are issued for heat exchanger 9, and the *economiser PI function* is activated to regulate outdoor air damper 23 to maintain the temperature of AHU supply air 9a at the *economiser damper temperature setpoint*.

If *economiser status* is active and the *economiser damper PI function* requests outdoor air damper 6 to fully close then plant controller 93 or diffuser controller 90 will terminate the *economiser active status*. This is because *mechanical heating mode* is required. This will occur, for example, if the requested outdoor air damper 6 position by the *economiser damper PI output* is less than the minimum damper position for outdoor air damper 6 required to satisfy indoor air quality. As the temperature of AHU supply air 9a will then be less than the *economiser damper temperature setpoint* plant controller 93 will switch from *mechanical cooling mode* to *mechanical heating mode* upon entering *economiser not active status*.

Figures 3e and 3f are graphs illustrating operation of $F.td$ when *economiser* status is *active*, and determination of *economiser deadband request (EDR)* required for *economiser* operation.

When *economiser status* is *active* the *economiser dynamic deadband*, as determined by *diffuser controller 90*, between the *AHU supply temperature setpoint* and *economiser damper temperature setpoint* is set to a *minimum economiser deadband* magnitude of 1 K (adjustable). However, *diffuser controller 90* may adjust the *AHU supply temperature setpoint* and the *economiser damper temperature setpoint* dynamically to maximise *EDR* to, in turn, maximise free-cooling potential, as follows.

If *economiser status* is *active* then *temperature differential factor $F.td$ 32* is redefined as:

$$F.td = F.i, \text{ where}$$

$F.i$ is the *increasing factor* (0 to 1)

A further factor, *economiser damper factor $F.ed$ 32'* (0 to 1) is defined as:

$$F.ed = 1 - F.d + F.d * F.i, \text{ where}$$

$F.d$ is the *decreasing factor* (0 to 1), and

$F.i$ is the *increasing factor* (0 to 1).

20

The above are used to determine *economiser deadband factor $F.db$* , where

$$F.db = (F.ed - F.td), \text{ and}$$

$F.ed \geq F.td$, as $F.db$ is a factor from 0 to 1.

25

Note:

- If $F.d$ and $F.i$ are both 0, then $F.td = 0$ and $F.db = 1$
- If $F.d = 1$ and $F.i = 0$, then $F.td = 0$ and $F.db = 0$
- If $F.i = 1$, then $F.td = 1$ and $F.db = 0$ (regardless of the $F.d$ value)

30

The above, in turn, determine for each *diffuser unit 14* the *diffuser supply air temperature differential request $DSdTR$* and *diffuser economiser temperature differential request $DEdTR$* when *economiser status* is *active*.

35

When *economiser status* is *active*, *diffuser supply air temperature differential request $DSdTR$*

for each *diffuser unit 14* is determined, for a given C%D value, by redefining *diffuser percent supply temperature differential request (D%SdTR)* and *diffuser supply temperature differential request (DSdTR)*, respectively, as:

$$D\%SdTR = MinM\%dT + F.td * (MaxM\%dT - MinM\%dT); \text{ and}$$

$$DSdTR = ABS(DDdT) * [MinM\%dT + F.td * (MaxM\%dT - MinM\%dT)]$$

The *diffuser economiser temperature differential request DEdTR* is set equal to *diffuser supply air temperature differential request DSdTR* minus *economiser deadband request (EDR)*, where

EDR is defined as

$$EDR = ABS(DDdT) * [[MaxM\%dT - (1 - F.ed) * (MaxM\%dT - MinM\%dT)] - [MaxM\%dT - (1 - F.td) * (MaxM\%dT - MinM\%dT)]]$$

Figures 4a and 4b are diagrams illustrating an example of economiser operation for an embodiment of the invention. It is to be assumed that *plant controller 93* has provided *diffuser controller 90* with an *economiser active status*, and that:

the *outdoor air temperature 23* is 20°C;
the *return air chamber 24a* air temperature is 23°C;
the *room or zone 20* air temperature is 23°C; and
the *diffuser design temperature differential DDdT* is -12 K.

If the *cooling PI output 25d* in figure 2a is 50% in order to maintain *room or zone 20* at a room air temperature of 23°C then *cooling percent demand (C%D) 25d'* in figure 3e will result in:

$$MinM\%dT \text{ 28b} = -50\% \text{ (circle 30b); and } MaxM\%dT \text{ 28a} = -100\% \text{ (circle 30a)}$$

For the sake of simplicity, assume that *diffuser unit 14* is both a *lead temperature* and a *lead pressure diffuser*, and that *actual diffuser supply air temperature differential* equals the *diffuser supply temperature differential request*, and that *actual diffuser pressure* equals the *diffuser pressure request*.

If $F.td = 0$ (ie no increasing factor functions active) and $F.ed = 1$ (ie no decreasing factor functions active) and assuming no heat pick-up in *duct 12* and *branch duct 12a*, then:

AHU upper supply temperature setpoint = 17°C (ie 23°C + 50% of -12 K) ; and

AHU lower supply temperature setpoint = 11°C (ie 23 °C + 100% of -12 K).

5 Therefore (refer to figure 4b):

AHU supply air cooling setpoint = 17°C; and

Economiser damper temperature setpoint = 11°C.

10 For the *outdoor air temperature 23* of 23°C relative to the *AHU supply air cooling setpoint* of 17°C at *AHU supply air temperature sensor 11*, the integral function of *mechanical cooling PI curve 85* will move *cooling PI curve 85* left (arrow 87) over time until it reaches steady state, assumed here as *steady state mechanical cooling PI curve 85'* with a *mechanical cooling PI output* of 50% (circle 85b) providing 50% cooling output from *heat exchanger 9* to achieve the setpoint of 17°C *AHU supply air*
15 *temperature*.

As the *economiser status* is *active* and the *AHU supply air 9a* temperature of 17°C is greater than the *economiser damper temperature setpoint* of 11°C, *outdoor air damper 6* will receive an economiser request to fully open (circle 86b at 100% outdoor air damper output) from *outdoor damper*
20 *PI curve 86*, and associated *return air damper 5* and *exhaust air damper 4* will receive economiser requests to fully close and fully open, respectively, as shown in figure 4a.

Figures 5a and 5b are a further illustration of the above example if the outdoor air temperature 23 has changed to 14°C, with all other parameters remaining unchanged.

25 In this instance, the *outdoor air temperature 23* of 14°C lies in the 6 K *economiser deadband* between the *AHU supply air cooling setpoint* of 17°C and the *economiser damper temperature setpoint* of 11°C at *AHU supply air temperature sensor 11*. The integral functions of both the *mechanical cooling PI curve 85* and the *outdoor damper PI curve 86* will remain at zero. This will result in a *cooling PI output* of 0% (circle 85c) from *mechanical cooling PI curve 85* providing 0% cooling output from *heat exchanger*
30 *9*. *Outdoor air damper 6* will receive an economiser request to fully open (circle 86b at 100% outdoor air damper output) from *outdoor damper PI curve 86*, and associated *return air damper 5* and *exhaust air damper 4* will receive economiser requests to fully close and fully open, respectively, as shown in figure 5a.

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Figures 6a and 6b are a further illustration of the above example if the outdoor air

temperature 23 has changed to 9°C, with all other parameters remaining unchanged.

In this instance, the *outdoor air temperature 23* of 9°C is less than the *economiser damper temperature setpoint* of 11°C measured at *AHU supply air temperature sensor 11*. The integral function of the *mechanical cooling PI curve 85* will remain at zero, resulting in a *cooling PI output* of 0% (circle 85c) requesting 0% cooling output from *heat exchanger 9*. In contrast, the integral function of *outdoor damper PI curve 86* will shift this curve to the right over time until it reaches steady state, assumed here as *steady state outdoor damper PI curve 86'* requesting an *outdoor damper PI output* of 85% (circle 86c), mixing 85% warm return air of 23°C with 15% cool outdoor air of 9°C to maintain the 11°C *economiser damper temperature setpoint* as measured by *AHU supply air temperature sensor 11*. *Outdoor air damper 6* and *exhaust air damper 4* will each receive *economiser requests* from *plant controller 93* to modulate to suit, as shown in figure 6a.

Figures 7a and 7b illustrate the same example as figures 5a and 5b, but with *return air leakage 23''* of 23°C (potentially due to worn damper blade seals over time) mixing with *outdoor airflow 23'* of 14°C. Assuming 10% leakage, this will raise the temperature of *mixed airflow 24'* to approximately 15°C. If heat from *supply air fan 7* (in previous examples assumed to be zero for simplicity) adds a further 1 K to the air temperature, then the *supply fan air 7a* temperature onto *heat exchanger 9* will be approximately 16°C. Advantageously, this does not create a false load on the mechanical plant, as the *mechanical cooling PI output 85a* to *heat exchanger 9* remains 0 (circle 85c) because *AHU supply air temperature sensor* continues to measure a temperature within the deadband of 6 K. This is compliant with the energy code ASHRAE Standard 90.1-2019 Section 9.5.1.3, which requires that controls shall not be capable of creating a false load in the mechanical cooling system.

Figures 8a and 8b illustrate a prior art comparative example of the example shown in figures 7a and 7b in accordance with the invention.

The prior art typically modulates *economiser damper operation* as a first stage, and *mechanical cooling* as a second stage, to a *single AHU supply air temperature setpoint*, as measured by *AHU supply air temperature sensor Ts.ahu 11*. An example is shown in figure 8b in which, for *stage 1*, *outdoor air damper PI output* is varied from 0% to 100% as *economiser PI output* varies from 0% to 50%, and in which, for *stage 2*, *mechanical cooling PI output* is varied from 0% to 100% as *economiser PI output* varies from 50% to 100%.

The *single AHU supply air temperature setpoint* may be fixed in some prior art embodiments and may be reset in some other prior art embodiments. In even further prior art embodiments, the *single AHU supply air temperature setpoint* may be released by *plant controller 93* when *outdoor air 23* temperature, as measured by *outdoor air temperature sensor 23''* permits (based on system cooling and

dehumidification requirements determined by *plant controller 93*) to “float” according to the formula:

$$S.AHU.Ts.sp = T.aa + T.offset, \text{ where}$$

5 *S.AHU.TS.sp* is the *single AHU supply air temperature setpoint* for *AHU supply air temperature 9a* as measured by *AHU supply air temperature T.s.ahu 11*;

T.aa is the temperature of *outdoor air 23* as measured by *outdoor air temperature sensor T.aa 23*”; and

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T.offset is a fixed temperature differential offset, typically set to 1 K.

The above is a bid to maximise free-cooling potential whenever possible. However, 100% free-cooling potential is seldom achievable with this strategy, which tends to either mix in unnecessary
 15 return air (wasting fan energy) or creates a false load on the mechanical cooling plant (wasting chiller energy). The reasons for this are outlined below.

In an ideal world, *T.aa 23*” and *Ts.ahu 11* would each be perfectly calibrated relative to one another (ie there would be no temperature measurement errors between them), *return air damper 5*
 20 would seal perfectly, and *supply air fan 7* would produce no heat gain. In the real world, however, these idealisations are not true. Given that ASHRAE Standard 90.1-2019 Section 9.5.1.3, requires that controls shall not be capable of creating a false load in the mechanical cooling system, *T.offset* is added to *T.aa 23* to accommodate heat gain from *supply air fan 7* and for calibration errors between *T.aa 23* and
 25 *Ts.ahu 11*, so as to reduce the risk of creating a false mechanical load and wasting energy running the mechanical cooling plant when it is not required. This aim is not always achieved, however, and if achieved, results in fan energy being wasted recirculating *return air 5* when not required.

Factors affecting *T.offset* include:

30 HVAC air temperature sensors from reputable brands typically have a calibration error within ± 0.3 K, with drift of ± 0.1 K over a year.

The heat gain from a supply air fan typically varies as the square of the airflow rate. Fan heat gain is usually approximately 1 K at full fan speed (although this depends on fan efficiency, system pressure, etc), which means that it is approximately one quarter of this value (equating to about 0.25 K)
 35 at about 50% airflow rate, and one eleventh of this value at about 30% airflow rate (equating to less than 0.1 K). Many standards require that the calibration of economiser temperature sensors be checked

annually because of the enormous influence their calibration has on overall energy efficiency. This is further exacerbated by damper seals wearing over time or their mechanisms seizing, especially if maintenance regimes are poor.

The table below shows a number of different *Ideal T.offset* values to achieve 0% return air damper 5 demand (so as not to waste fan energy) and 0% cooling demand from *heat exchanger 9* (so as not to waste cooling plant energy) for scenarios 1 to 6 presented in column 1, with column 2 describing temperature sensor error and drift for each scenario.

Scenario 1 applies directly following commissioning, with a -0.3 K calibration error assumed for each of *T.oa* and *T.s.ahu*. At 30% fan speed, the *Ideal T.offset* is -0.5 K to achieve neither a *return air damper 5* nor a cooling demand from *heat exchanger 9*.

Scenario 2 also applies directly following commissioning, but with a +0.3 K calibration error for each of *T.oa* and *T.s.ahu*. At 100% fan speed, the *Ideal T.offset* is +1.6 K to achieve neither a *return air damper 5* nor a cooling demand from *heat exchanger 9*.

Scenario 3 applies one year after commissioning scenario 1. It, additionally, includes -0.1 K drift for each of *T.oa* and *T.s.ahu*. At 30% fan speed, the *Ideal T.offset* is -0.7 K to achieve neither a *return air damper 5* demand nor a cooling demand from *heat exchanger 9*.

Scenario 4 applies to one year after commissioning scenario 2. It includes a -0.1 K drift for each of *T.oa* and *T.s.ahu*. At 100% fan speed, the *Ideal T.offset* is +1.8 K to achieve neither a *return air damper 5* demand nor a cooling demand from *heat exchanger 9*.

Scenario 5 applies several years after commissioning scenario 1. It includes 10% RA damper leakage plus -0.1 K drift for each of *T.oa* and *T.s.ahu* (based on each last being calibrated 1 year earlier). At 30% fan speed, the *Ideal T.offset* is +0.2 K to achieve neither a *return air damper 5* demand nor a cooling demand from *heat exchanger 9*.

Scenario 6 applies several years after commissioning scenario 2. It includes 10% RA damper leakage, plus +0.1 K drift for each of *T.oa* and *T.s.ahu* (based on each last being calibrated 1 year earlier). At 100% fan speed, the *Ideal T.offset* is +2.7 K to achieve neither a *return air damper 5* demand nor a cooling demand from *heat exchanger 9*.

S C E N A R I O	Case	T.oa Error [K]	T.s.ahu Error [K]	30% fan heat gain [K]	100% fan heat gain [K]	10% leakage RA damper 5 heat gain* [K]	Total heat gain @ 30% fan [K]	Total heat gain @ 100% fan [K]	Ideal minimum T.offset @ 30% fan [K]	Ideal maximum T.offset @ 100% fan [K]
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1	-0.3 K Tsens error	-0.3	-0.3	+0.1	+1.0		-0.5	+0.4	-0.5	+1.6
2	+0.3 K Tsens error	+0.3	+0.3	+0.1	+1.0		+0.7	+1.6		
3	Case 1 with -0.1 K drift	-0.4	-0.4	+0.1	+1.0		-0.7	+0.2	-0.7	+1.8
4	Case 2 with +0.1 K drift	+0.4	+0.4	+0.1	+1.0		+0.9	+1.8		
5	Case 3 with 10% RA leakage	-0.4	-0.4	+0.1	+1.0	+0.9	+0.2	+1.1	+0.2	+2.7
6	Case 4 with 10% RA leakage	+0.4	+0.4	+0.1	+1.0	+0.9	+1.8	+2.7		

* Heat gain [K] due to 10% return air leakage 23'' at 23°C temperature mixing with outdoor airflow 23' of 14°C.

It can be seen from the above that the typical *T.offset* value of +1 K is potentially not even sufficient to prevent a false load on the mechanical cooling plant at the time of commissioning, as the *ideal T.offset* values vary from -0.5 K to +1.6 K directly after commissioning.

The highest *Ideal T.offset* value needs to be selected in order to prevent a false load on the mechanical cooling plant. However, this will lead to significant fan energy being wasted, unnecessarily recirculating return air whenever the actual *T.offset* required is less than the maximum (eg at lower fan speeds, or if temperature sensor error or drift is less than the worst case that has to allowed for).

The *Ideal T.offset* values increase to a range +0.2 K to +2.7 K once 10% return air 5 damper leakage sets in over time, even if temperature sensors are recalibrated annually. Consequently, both significant fan energy and mechanical cooling plant energy are likely to be wasted (due to unnecessary recirculated return and unnecessary mechanical cooling), and the economiser system is unlikely to be

compliant with ASHRAE Standard 90.1-2019 Section 9.5.1.3, at least over time.

In figures 8b, scenario 6 is assumed to apply at 100% fan speed, resulting in a *total heat gain @ 100% fan speed* of +2.7 K. The *outdoor air 23* temperature is shown as 14°C with the 10% *return air leakage 23''* being at 23°C. Assuming that *T.offset* has been set to +1.0 K (as is typical), this will result in a +1.7 K false mechanical cooling load, causing *heat exchanger 9* to cool in order to treat this false load and bring *AHU supply air 9a* temperature down to the *S.AHU.Ts.sp* of 15°C (ie 14 °C + 1 K). This will drive *economiser PI curve 95* to move to the left until a steady state is achieved, shown at 95', providing 100% *stage 1 outdoor damper PI output* (circle 96a) plus sufficient *stage 2 mechanical cooling PI output* to provide 1.7 K (ie 2.7 K - 1 K) cooling, assumed here to be at 20% *mechanical cooling PI output* (circle 96b). If, however, scenario 5 were to apply, then the total heat gain at 30% fan speed would be only +0.2 K, resulting in a false "over-cooling" temperature differential of -0.8 K. This would result in the PI integral function driving *economiser PI curve 95* to position 95'' to produce zero *stage 2 mechanical cooling PI output* and a *stage 1 outdoor air damper PI output* of 91% (circle 96c). In other words, a return air damper 5 position of 9% will be called for (over and above the 10% leakage already taking place) in order to achieve the *S.AHU.Ts.sp* of 15°C. Fan energy will be wasted recirculating unnecessary return air.

20 Potentially Advantageous Features of the Embodiments Described Herein

A diffuser unit control system in accordance with embodiments of the invention may reduce temperature fluctuations in the zone or room, as the constant thermal capacity VAV damper operation of the diffuser unit ensures that the thermal capacity provided to the room remains substantially constant for a room in thermal equilibrium (steady state), even if system supply air temperature or supply air pressure change. By comparison, VAV diffuser control systems of the prior art do not maintain such constant thermal capacity if supply air temperature or pressure to the diffuser change. Instead, the room temperature is affected by such changes - even so-called "static pressure independent VAV modulation" only targets the provision of constant supply airflow rate control, but the thermal capacity provided is still affected by supply air temperature fluctuations - and a delayed response then takes place from the room temperature controller to correct the change, leading to unsteady temperature control with potential over-shoot or hunting as the system attempts to re-establish zone or room setpoint temperature.

35 A diffuser unit control system in accordance with embodiments of the invention allows lower supply air temperatures to be achieved at diffusers, which is particularly advantageous for low-temperature (also called cold-air) VAV systems, as low supply air temperatures allow fan energy savings

to be achieved in a broad range of climate zones. However, the precise supply air temperature at each diffuser is not known in diffuser control systems of the prior art; only that at a point – typically at the air handling unit or in the supply duct – is known, which is well removed from most of the diffusers. In order to ensure that Coanda-effect attachment of the supply air stream to the ceiling is not lost, resulting in supply air to dumping, causing draughts close by and stagnation further afield, a safety margin must be allowed for potential heat pickup to each diffuser. This applies in particular when the diffusers are turned down, which means that the dwell time of the supply air in the supply duct increases substantially, with corresponding increases in heat gain through the duct walls. By measuring supply air temperature at each diffuser, a lower supply air temperature can be supplied to the diffusers in cooling mode, as there is no need to apply a safety factor to mitigate the risk of dumping. This achieves increased fan energy savings as well as improved thermal comfort.

A diffuser unit control system in accordance with embodiments of the invention further allows lower operating pressure to be achieved by the fan than is possible with control systems of the prior art. This saves fan energy. The control system described in the invention determines static pressure at each diffuser unit, rather than well removed from the diffuser units - in ducts serving a plurality of diffuser units - as is done in the prior art. Consequently, system pressure can, under part load conditions, generally be reduced to the minimum permissible operating pressure of the diffuser units, and this applies even if loads change relative to one another causing the lowest pressure to change from one diffuser unit to another.

A diffuser unit control system in accordance with embodiments of the invention further allows the supply air pressure to be minimised by operating at least one diffuser unit damper fully open, unless doing so would result in a static supply air pressure to the diffuser unit that is less than the minimum permissible operating pressure of the diffuser. This ensures that fan pressure is minimised. Control systems of the prior art typically use trim-and-respond logic to iteratively reset supply air temperature, supply air pressure and diffuser damper positions over extended periods of time. The optimum operating points for each of these are seldom achieved (for example, system pressure requests are increased when diffuser units serving the highest heat load spaces almost fully open their VAV dampers, not only once they have fully opened their VAV dampers, which leads to higher system pressure than is required, and hence to higher fan energy usage).

A diffuser unit control system in accordance with embodiments of the invention further allows the absolute supply air temperature differential between room air and supply air to be minimised, not only saving mechanical cooling or heating energy by not having to cool to as low or as high a dry bulb supply air temperature, respectively, but also by wasting less energy on latent cooling for the former and in wasting less energy on stratification of heat due to a high level in rooms for the latter. This is achieved by, for each diffuser unit, adjusting the diffuser temperature offset request with an equivalent

increased airflow rate if excess static pressure is available to that diffuser unit, thereby allowing each such diffuser unit to provide the required sensible cooling or heating capacity at a reduced temperature differential but without requiring an increased supply air static pressure for the compensating increased airflow rate. This saves energy.

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A diffuser unit control system in accordance with embodiments of the invention further allows both the supply pressure and supply temperature to be optimised to operate the dampers of at least two diffuser units fully open whenever a sufficient absolute temperature differential excess exists at one diffuser unit to adjust that diffuser unit's pressure offset request to 0, and a sufficient pressure excess exists at another diffuser unit to adjust that other diffuser unit's temperature offset request to 0. This saves energy.

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A diffuser unit control system in accordance with embodiments of the invention reduces costs by using static pressure sensors at each diffuser unit to serve a dual function, viz to determine diffuser static pressure for system pressure and fan speed control, as well as to determine the airflow rate of each diffuser unit. Control systems of the prior art utilise either total pressure sensors or hot wire anemometers to determine diffuser unit airflow rate, and additionally use static pressure sensors in each duct run to determine system pressure and fan speed control. This costs more, as it involves more sensors and more site work installing and commissioning the sensors.

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A diffuser unit control system in accordance with embodiments of the invention increases reliability by increasing redundancy. Each diffuser unit is equipped with a static pressure sensor, and preferably with a supply air temperature sensor. If either one were to fail in a particular diffuser unit then system operation will be able to continue by substituting approximate temperature or pressure values from other diffuser units.

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A diffuser unit control system in accordance with embodiments of the invention provides a simple means, via increasing and decreasing factors, to raise or lower supply air temperature and simultaneously adjust diffuser airflow rate to compensate so that the thermal capacity provided by a lead diffuser unit remains constant in response to a requirement to, for example, increase dehumidification, prevent over-cooling, increase room ventilation for indoor air quality, etc. This ensures stable room air temperature in the room served by a lead diffuser unit (which typically operates with its variable damper fully open and therefore cannot adjust delivered capacity via damper modulation) even when system supply air temperature or pressure requirements change.

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A diffuser unit control system in accordance with embodiments of the invention provides, via the increasing and decreasing factors, a simple means to, for example, vary supply air temperature as a function of outdoor air temperature or fan load, or in response to fan energy and chiller energy meters, to vary supply air temperature so that overall energy consumption is minimised, and does so without
5 limiting design capacity.

In economiser mode, a diffuser unit control system in accordance with embodiments of the invention may provide two supply air temperature setpoints, viz an AHU supply air temperature setpoint for mechanical cooling and an economiser damper temperature setpoint, separated by a dynamic
10 deadband that maximises free-cooling potential by minimising the need for mechanical cooling (which wastes chiller energy) or recirculated air (which wastes fan energy)..

In economiser mode, a diffuser unit control system in accordance with embodiments of the invention, is not affected by AHU supply air temperature sensor calibration or drift, and economiser damper modulation or supplementary mechanical cooling operation is not additionally dependent upon
15 an outdoor air temperature sensor, thereby preventing the risk of a false cooling load on the mechanical plant from wasting mechanical plant energy or a false over-cooling load causing return air to be mixed into the supply air stream wasting fan energy. Full compliance with ASHRAE Standard 90.1-2019 Section 9.5.1.3 is achieved, as are reliable operation and simple maintenance. In contrast, the single supply air temperature setpoint used in economiser control of the prior art is inefficient and only achieves
20 questionable compliance with ASHRAE Standard 90.1-2019 Section 9.5.1.3 if free-cooling potential is to be maximised. It suffers from temperature sensor calibration errors and drift, is not able to automatically adapt to changing fan heat gain, and cannot accommodate return air damper leakage. Furthermore, since the prior art is based on controlling AHU supply air temperature to outdoor air temperature plus 1 K (typically) and may mix in at least a small percentage of return air during
25 economiser mode when supplementary mechanical cooling is not required, it cannot be assumed that the outdoor air damper is fully open for indoor air quality purposes. If this assumption cannot be made and if the actual outdoor air quantity is not known, then the minimum design outdoor air quantity from the outdoor air damper has to be assumed for indoor air quality purposes, which results in substantially increased diffuser minimum damper positions to ensure adequate indoor air quality in each room. This,
30 in turn, increases the risk of over-cooling rooms and results in a significant increase in overall energy consumption during economiser mode.

A diffuser unit control system in accordance with embodiments of the invention provides distributed control via a micro-processor per diffuser unit, and requires only minimal global controls
35 that only require a very global controller. The complex controls of determining diffuser airflow rate, diffuser damper position, diffuser increasing and decreasing as a function of and various diffuser offsets

(pressure, temperature, economiser damper) reside in each respective diffuser micro-controller, with only simplified global controls required in a global controller that the micro-processors communicate with. The global controller determines global setpoints (eg room temperature, relative humidity), the global factor, the economiser damper factor, and the offset controls (a supply pressure or fan speed PI controller with setpoint of zero; a supply temperature or thermal capacity PI controller with setpoint of zero; an over-pressure PI controller with setpoint of -5 Pa, for example; an economiser damper PI controller with setpoint of zero). The result is a highly scalable controls architecture that requires only a small amount of very simple global controls that can easily be performed by a BMCS (which most buildings suitable for a diffuser unit control system in accordance with this invention have anyway) without requiring any additional processors other than the diffuser micro-controllers. This saves substantial hardware costs. Additionally, since the global controls do not need to know which diffusers are lead diffusers, or to associate offsets with the diffuser units that they were generated by, controls programming and commissioning is extremely simple, and the global controls programming can hence be done by the BMCS supplier or contractor in accordance with simple instructions or standardised templates. This vastly reduces network traffic, increases reliability, and simplifies both design architecture and commissioning. A further advantage of the distributed controls is that if a diffuser unit loses communications it, nevertheless, is able to continue to provide constant thermal capacity VAV damper operation for zone temperature control, and may even be supplied to the market to operate in such "standalone" form, suitable for connection to a BMCS to provide AHU control at a future date (eg when a base building is converted to a tenancy fitout, or when tenancy fitout changes occur).

A further advantage is that a diffuser unit control system in accordance with embodiments of the invention is modular in nature, readily allowing diffuser units to be added or removed, thereby facilitating tenancy fitout changes.

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A diffuser unit control system in accordance with embodiments of the invention further provides a linear cooling or heating capacity response across the full PI output range of each room PI controller, thereby providing stable operation.

Claims

1. A method of controlling an HVAC system, the HVAC system comprising an air handling unit to produce a supply airflow, the air handling unit being connected via a ducting system to a plurality of VAV diffuser units, each VAV diffuser unit influencing air temperature in a corresponding zone by delivering supply air to the zone, wherein
- 5 the air handling unit comprises at least one supply air fan or motorised supply air damper; and
- each diffuser unit comprises a diffuser, a motorised variable damper to alter a diffuser airflow rate of a diffuser airflow delivered by the corresponding diffuser unit into the corresponding zone, and a diffuser pressure sensor for determining a pressure of the diffuser airflow; wherein
- 10 the HVAC system further comprises a plurality of zone temperature sensors;
- the HVAC system further comprising at least one control unit connected to the supply air fan or motorised supply air damper, each diffuser unit motorised variable damper, each zone temperature sensor and each diffuser pressure sensor;
- 15 the method comprising determining a target thermal capacity for a selected diffuser unit with reference to a target diffuser airflow rate and a target diffuser temperature differential and controlling the motorised variable damper of the selected diffuser unit with reference to the determined target thermal capacity for the selected diffuser unit.
- 20
2. The method of claim 1 wherein the control unit further comprises a plurality of micro-controllers that communicate with one another.
- 25
3. The method of claim 2 wherein the plurality of micro-controllers comprise micro-controllers provided on each of the diffuser units.
4. The method of any preceding claim in which the air handling unit further comprises at least one heat exchanger, and the at least one control unit is connected to the at least one heat exchanger.
- 30
5. The method of any preceding claim comprising, for the selected diffuser unit:
- determining a zone temperature setpoint;
- determining a zone air temperature with a corresponding zone temperature sensor;

determining a diffuser unit control output in dependence on the zone temperature setpoint and the determined zone air temperature;

determining the target diffuser airflow rate in dependence on a diffuser design airflow rate and the diffuser unit control output for the selected diffuser unit;

5 determining a diffuser design temperature differential;

determining the target diffuser air temperature differential in dependence on the diffuser design temperature differential and the diffuser unit control output for the selected diffuser unit; and

10 determining the diffuser target diffuser airflow rate in dependence on the diffuser target temperature differential, or determining the diffuser target temperature differential in dependence on the diffuser target diffuser airflow rate.

6. The method according to claim 5 wherein the design temperature differential is determined in relation to a diffuser supply air temperature.

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7. The method according to claim 6 wherein the diffuser supply air temperature is determined by a sensor incorporated into the selected diffuser unit.

8. The method according to claim 6 wherein the diffuser supply air temperature is determined by a sensor located remote from the selected diffuser unit.

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9. The method according to any of claims 6 to 8 wherein the target thermal capacity is calculated according to the following formula:

$$C\%D = D\%SdTR * D\%AR$$

25 where C%D is the target thermal capacity expressed as a percentage of maximum diffuser unit control output, D%SdTR is the target diffuser air temperature differential expressed as a percentage of the diffuser design temperature differential, and D%AR is the target diffuser airflow rate expressed as a percentage of the diffuser design airflow rate.

10. The method according to any preceding claim further comprising determining a plurality of sets of control variables, each set comprising a target factored diffuser airflow rate and a target factored diffuser temperature differential, in which each set satisfies the predetermined target thermal capacity for the selected diffuser unit.

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11. The method according to claim 10 wherein the HVAC system further comprises an economiser damper system for selectively mixing an air supply from outside the HVAC system with the supply air when the HVAC system is in an economiser mode, the method further comprising, when the HVAC system is in an economiser mode:

5 each set of control variables satisfies the predetermined target thermal capacity
for the selected diffuser unit additionally comprises
a target factored economiser deadband;
determining the target factored economiser deadband; and
controlling the economiser damper system with reference to said target
10 factored economiser deadband.

12. The method according to claim 10 or claim 11 further comprising selecting one of said plurality of sets of control variables according to one or more criteria.

13. The method according to claim 12 wherein the criteria include one or more of humidity, over-cooling, over-heating, outdoor air temperature, energy conservation, indoor air quality,
15 noise, excess pressure.

14. The method of any preceding claim wherein the control unit further controls one or more of: a supply air temperature, supply air pressure, and supply air fan speed or damper position of the motorised supply air damper; to alter or maintain a temperature of at least one zone.

15. The method according to claim 14 wherein the control unit controls the supply air
20 pressure to alter or maintain a temperature of at least one zone when the motorised variable damper of the diffuser unit serving the at least one zone is fully open.

16. The method according to claim 14 or claim 15 wherein the control unit controls the supply air temperature to alter or maintain a temperature of at least one zone when the motorised variable damper of the diffuser unit serving the at least one zone is fully open.

25 17. The method of any preceding claim further comprising, for the selected diffuser unit, adjusting the motorised variable damper to deliver an adjusted target factored damper airflow rate.

18. The method of claim 17 when dependent on claim 10 or claim 11 wherein the adjusted target factored damper airflow rate is determined with reference to an actual temperature differential

relative to a target factored diffuser air temperature differential for the diffuser unit, and an actual pressure at the diffuser unit relative to a target factored diffuser pressure.

19. The method of claim 18 further comprising, for a selected diffuser unit with fully
5 open motorised variable damper, adjusting the target factored diffuser pressure as a function of the target factored diffuser airflow rate.

20. The method according to any preceding claim further comprising, for each diffuser unit, determining a diffuser temperature offset request and a diffuser pressure offset request.

10 21. The method according to claim 20, wherein, for each diffuser unit, the diffuser temperature offset request is determined by subtracting the actual temperature differential from the target factored diffuser temperature differential, or if the actual pressure is greater than the target factored diffuser pressure then from the target factored diffuser temperature differential multiplied by the square root of the target factored diffuser pressure divided by the square root of the actual
15 pressure.

22. The method according to claim 21, wherein the diffuser pressure offset request is determined by subtracting the actual pressure from the target factored diffuser pressure, or if the actual temperature differential is greater than the target factored diffuser temperature differential then from the target factored diffuser pressure multiplied by the square of the target factored diffuser
20 temperature differential divided by the square of the actual temperature differential.

23. The method according to claim 21 or claim 22 realising, for each diffuser unit, a diffuser pressure offset request of zero when the target factored diffuser pressure is equal to the actual pressure and the target factored diffuser temperature differential is greater than or equal to the actual temperature differential.

25 24. The method according to claim 21 or claim 22 realising a diffuser temperature offset request of zero when the target factored diffuser temperature differential is equal to the actual temperature differential and the target factored diffuser pressure is greater than or equal to the actual pressure.

25. The method according to claims 21 to 24 realising, for each diffuser unit, a diffuser pressure offset request of zero when the target factored diffuser pressure is greater than the actual pressure and when the target factored diffuser pressure divided by the actual pressure is equal to the square of the actual temperature differential divided by the square of the target factored temperature differential.

26. The method according to claims 21 to 25 realising a diffuser temperature offset request of zero when the target factored diffuser temperature differential is greater than the actual temperature differential and the target factored diffuser temperature differential divided by the actual temperature differential is equal to the square root of the actual pressure divided by the square root of the target factored diffuser pressure.

27. The method according to claim 21, claim 22 or claim 23 further comprising ranking each diffuser temperature offset request in a temperature rank and ranking each diffuser pressure offset request in a pressure rank.

28. The method according to claim 27 wherein the HVAC system further comprises an air supply controller with proportional-integral (PI), proportional-integral-derivative (PID) or nudge-and-wait control, and with the setpoint set to zero, and with the process value designated as the highest diffuser pressure offset request from the pressure rank.

29. The method according to claim 27 and claim 28 wherein the HVAC system further comprises a thermal capacity controller with proportional-integral (PI), proportional-integral-derivative (PID) or nudge-and-wait control, and with the setpoint set to zero, and where the direction of the thermal capacity control is set by a requirement for system cooling or system heating, with the former designating the process value to be the lowest diffuser temperature offset request from the temperature rank and the latter designating the process value to be the highest diffuser temperature offset request from the temperature rank.

30. The method according to claim 29 wherein the requirement for system cooling or system heating is determined by a vote of the diffuser units.

31. The method according to claim 27 wherein the HVAC system further comprises an air supply target from the air supply controller, wherein the air supply target determines a supply air pressure

setpoint for the air handling unit in relation to an air handling unit supply air pressures sensor, or directly determines the fan speed of the at least one supply air fan.

32. The method according to claim 30 wherein the HVAC system further comprises a cooling or heating capacity target from the thermal capacity controller, wherein the cooling or heating capacity
5 target determines a supply air temperature setpoint for the air handling unit in relation to an air handling unit supply air temperature sensor, or directly determines the cooling or heating output of the at least one heat exchanger.

33. The method according to any of claims 12 to 32, when dependent on claim 11,
further comprising determining an economiser damper temperature setpoint with reference to a target
10 economiser deadband for each diffuser unit.

34. The method according to any of claims 7 to 33, when dependent on claim 6, further comprising, for each diffuser unit, subtracting the actual temperature from the economiser damper temperature setpoint to determine an economiser offset request.

35. The method according to claim 34, further comprising ranking each diffuser unit
15 economiser offset request in an economiser rank.

36. The method according to claim 35 wherein the HVAC system further comprises an economiser controller with proportional-integral (PI), proportional-integral-derivative (PID) or nudge-and-wait control, and with the setpoint set to zero, and with the process value designated as the lowest economiser offset request from the economiser rank.

20 37. The method according to claim 36 further comprising an economiser target from the economiser controller, wherein the economiser target determines an economiser supply air temperature setpoint for the economiser damper system in relation to the air handling unit supply air temperature sensor, or directly determines an economiser damper system output to directly modulate the economiser damper system.

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