

Control of Air Conditioning Cooling or Heating Coil

This invention relates to improvements in control of part load capacity of a fluid heat exchange device, especially a chilled water cooling coil used in air handling equipment and fan coil units for comfort cooling and industrial application. In the following the device is described for cooling application, but is usable for heating application as well.

- 10 With conventional throttle valve controlled cooling coils at part load the latent capacity is reduced much faster than the sensible capacity, resulting in an increase in space relative humidity and decrease in comfort. Sensible heat source up stream of the cooling coil other than in the conditioned space
- 15 does not contribute to effective sensible load from a latent removal standpoint. The coil needs to be selected at a high water side pressure drop at full load to ensure turbulent flow in circuits at partial load. The associated control valve represents equal or higher pressure drop than the coil,
- 20 resulting in high pump pressure head and considerable operating cost. Except for employing a reheat device, independent control of sensible versus latent capacity is not available. Effective treatment of high humidity outside air requires a dedicated air handler. Hot water heating coils exhibit the same negative
- 25 characteristics as far as high pressure head and part load controllability as cooling coils. To measure the energy used by an air handler requires a flow meter also entering - leaving water temperature differential measurement and accurate, repeatable flow metering is difficult and costly. Selecting a
- 30 cooling coil and control valve needs considerable experience despite sophisticated software selection tools to ensure low load performance and controllability. Water side balancing of a

chilled water system is time consuming and if not performed correctly reflects on system performance.

Preferred objects of the present invention include:

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Maintain latent capacity at least in proportion to the available sensible load during part load operation down to zero load.

- 10 Utilise sensible heat available in open return air plenum as an effective sensible load to enhance dehumidification of conditioned space.

- 15 Enable low pressure drop coil selection at full load and replace the high pressure drop control valve with a low differential pressure alternative.

Provide near independent means of part load sensible versus latent capacity control.

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Use the same air handler, the same cooling coil that serves the conditioned space to effectively treat high humidity outside air.

- 25 Permit low water side pressure drop heating hot water coil selection at full load, at the same time ensure controllability at low loads.

Provide an accurate water flow metering option.

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Provide optional water system balance indication and a degree of self balancing ability.

Reduce pumping power requirements for new system designs also for retrofit applications.

- 5 Ease of coil selection. Assuming the coil selected is large enough to meet full load, part load performance and controllability is ensured.

The principle of this invention is circuit by circuit control
10 of fluid flow. At full load all the circuits are active, thus there is fluid flowing through all the available circuits of the coil. At part load the flow of fluid is cut off to some of the circuits, while flow is maintained at or near full velocity in the active circuits. The number of active circuits at any
15 given time is determined by the prevailing air side load on the coil. The effective coil surface temperature around the active circuits remains constant, so dehumidification is maintained at part load, while around the inactive circuits no heat exchange to the air takes place.

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Other preferred objects will become apparent from the following description.

In a broad aspect the present invention resides in a control
25 method for chilled water cooling coils and hot water heating coils used in comfort and industrial air conditioning applications, including:

A movable piston located in the supply header of the coil. At
30 full load the piston is at it's upper most position and all the circuits are active, thus receiving full flow of chilled water. The position of this piston is dictated by the prevailing

sensible heat load on the coil. At partial load the piston is moved down, cutting off chilled water supply to the circuits above it's location.

- 5 The percent of active circuits being proportional to the sensible load and the effective coil surface temperature around the active circuits remaining constant ensures that the latent capacity of the coil is also proportional to the sensible load.
- 10 During part load operation around the upper inactive circuits the coil is at return air dry bulb temperature, no heat exchange takes place, thus any sensible heat source, be it up or down stream of the cooling coil is an effective source to enhance dehumidification of the conditioned space. This
- 15 includes heat generated by light fittings in open return air plenums.

The coil water side pressure drop at full load may be selected at a low value, as at part load there is no substantial change

20 in fluid flow velocity in the active circuits and the movable piston presents only minimal resistance.

Placing another movable piston, this time in the return pipe header of the cooling coil, will facilitate latent capacity

25 control. For full latent capacity this piston is at it's lower most position, below the exit of the lowest circuit. Elevating this piston the fluid flow is cut off to the circuits below it's position. For average space conditioning coil entering air conditions there is condensate on the higher active portion of

30 the coil. As this condensate runs down and reaches the low inactive area, it is partly or fully evaporated, resulting in rapid decrease in latent capacity and due to evaporative

cooling an increase in sensible cooling of the air stream.

Thus the piston in the supply pipe header controls sensible capacity by cutting fluid flow to upper circuits and the piston
5 in the return pipe header, near independently, controls latent capacity by cutting off fluid flow to the lower most circuits.

Ducting the outside air within the space serving air handling unit to the lower part of the cooling coil, the same air
10 handling unit may be used to effectively treat humid outside air as well as serve the conditioned space. The natural limit to this application is having sufficient sensible heat to perform the necessary dehumidification. Should there be insufficient sensible heat, some kind of reheat needs to be
15 applied, just as it would in case of a conventional air handling unit dedicated to treat outside air only.

For hot water heating coils the near constant flow in active circuits permits low coil differential pressure selection at
20 full load with assured low load performance and controllability.

One particular embodiment of this invention employs a weighted piston in the supply pipe header. The weight of the piston is
25 such as to impose the desired differential pressure across the coil thus ensure constant flow velocity in the active circuits. The weighted piston is acting as a pressure relief device, on rising pressure it moves up to expose more circuit entries, thus relieve the pressure and visa versa should the
30 differential pressure across the coil fall. In this instance there is a low pressure drop external control valve driven by the sensible load, for example a butterfly valve. The flow

velocity in the active circuits being constant at a fixed differential pressure across the coil, the number of active circuits thus the position of the piston is directly proportional to the water quantity flowing through the coil.

5 Thus monitoring the position of this weighted piston gives an accurate, repeatable option to monitor the fluid flow quantity. Addition of entering and leaving water temperature sensors provides energy monitoring capability.

10 Monitoring the water flow quantity via the position of this weighted free floating piston also facilitates water side balancing of the system. Keeping the external control valve full open and throttling the balance valve until the free floating piston just moves away from it's upper most position,
15 indicates that the coil is precisely at design water flow. All that remains is to lock the balance valve at this particular position.

An optional interlock between the weighted free floating piston
20 and external control valve will add self balancing capability. It is a limiting type interlock, when the free floating piston in the supply pipe header reaches it's upper most position, the external control valve is prevented from opening up further. Should the external control valve be wide open at start up, the
25 same interlock commands the valve to close until the piston drops just below it's uppermost position, thus restricting the coil to design chilled water quantity. During normal operation the external control valve is driven by the sensible load on the coil, however when the design water flow is exceeded the
30 limiting function takes preference. This self balancing ability is suitable for chilled and hot water distribution systems where the pressure change from full to minimum system load is

relatively small. For distribution systems where large pressure variations are expected, it is preferred to include manual balancing valves.

- 5 For a new installation the design can incorporate low pressure drop coils and control valves, resulting in substantial pumping power reduction. In a retrofit application where the original coil is retained, pumping power reduction is proportional to the pressure head reduction due to the removal of the original
10 high pressure drop control valve.

Selecting a coil, part load performance need not be considered as there is near constant flow velocity in the active circuits, thus transition from turbulent to laminar flow and subsequent
15 loss of heat transfer can no longer take place. A coil suitably sized to meet full load will perform and remain controllable at low partial loads.

To enable the invention to be fully understood, preferred
20 embodiments will now be described with reference to the accompanying drawings, in which:

FIG. 1 is a schematic view of the first embodiment highlighting the principle of this invention, operating as a
25 chilled water cooling coil.

FIG. 2 shows the application of this coil in an air handling unit for treating high humidity outside air and also serving a conditioned space.

FIG. 3 is a constant differential pressure, thus constant
30 circuit flow velocity embodiment of this invention, with weighted free floating piston in the supply pipe header of the coil.

FIG. 4 an alternative method to detect location of weighted free floating piston to facilitate water flow measurement.

FIG. 5 is a simplified way to detect upper most position of weighted free floating piston for water side balance indication and self balancing.

FIG. 6 shows an integral system powered control valve as an alternative to external control valve.

FIG. 7 illustrates a system powered method of controlling pressure differential across the coil to facilitate latent / sensible capacity ratio control.

FIG. 8 is a system pressure dependent low cost system powered alternative for general comfort cooling application.

FIG 9 shows a system pressure dependent motorised positioning of control piston in supply header. Optional latent / sensible capacity ratio control by additional piston placed in return pipe header is also illustrated.

FIG. 10 is a system powered alternative of positioning of control piston in return pipe header to facilitate latent / sensible capacity ratio control.

FIG. 11 illustrates a three stage solenoid valve controlled approach, where a number of circuits are controlled as a group.

FIG. 12 details of hydraulic actuated self propelled control piston.

FIG. 13 associated control elements of hydraulic actuated self propelled control piston.

FIG. 14 hybrid, hydraulic system and electric powered self propelled piston for pipe headers made of ferrous material.

FIG. 15 as in FIG. 14 but suitable for ferrous and non ferrous coil pipe headers.

FIG. 16 basic hydraulic actuated diaphragm circuit by circuit control without 100% positive shut off capability.

FIG. 17 hydraulic actuated diaphragm with near independent latent and sensible capacity control, also 100% shut off capability.

FIG. 18 basic system powered diaphragm control, no 100% shut off.

FIG. 19 system powered diaphragm control with modified pipe connection to supply pipe header. 100% shut off capability.

FIG. 20 hydraulic actuated diaphragm and external control valve, near independent sensible and latent control ability.

FIG. 21 hydraulic actuated diaphragms for circuit by circuit control and integral throttle valve, near independent sensible and latent control.

FIG. 22 mechanical actuated ball driven positioning of control piston with 100% shut off capability.

FIG. 23 hydraulic actuated ball driven positioning of control piston with 100% shut off capability.

FIG. 24 method of temperature based control piston position sensing.

FIG. 25 pneumatic powered diaphragm control with sliding bottom end clip used on diaphragm, shown under part load condition.

FIG. 26 same as in FIG. 25, however illustrated in the 100% shut off position.

FIG. 27 hydraulic actuated diaphragm with external, non system, hydraulic source. Utilising hydraulic fluid of less than 1 specific gravity.

FIG. 28 high pumping efficiency, low running cost, configuration with dedicated speed controlled pump.

FIG. 29 using a slotted cylinder to control coils with relatively low number of circuits, such as used in fan coil units.

Referring to FIG. 1, the chilled water coil 1, has a supply header 2, return header 3, and interconnecting circuits 4, between supply end return pipe headers. The plurality of
5 interconnecting circuits 4 are connected to each header 2, 3 by a corresponding plurality of connecting ports 202 at different locations lying in a row one above the other along the header 2, 3. Fluid flow is supplied to the supply header 2 through supply port 201 at the bottom end of header 2 and is
10 supplied from return header 3 through return port 203 at the top end of return header 3. Sliding piston 5, placed in supply pipe header 2, and equipped with water tight seals to prevent water flow from the lower to upper part of pipe header 2. In this illustration piston 5, cuts off the water flow to the
15 upper three circuits 6, the coil surface temperature in the region of circuits 6, is the same as of the entering air and no heat exchange takes place. The circuits 4, below the lower edge of piston 5, are active, receiving full supply of chilled water and the coil surface temperature around circuits 4, is at
20 design temperature. Air traversing this region is cooled and dehumidified. Moving piston 5, upwards increases both sensible and latent capacities of the coil. Moving piston 5, downwards reduces both sensible and latent capacities. The ratio of sensible versus latent capacity is defined at the coil
25 selection / design stage and this ratio remains constant at partial load across the whole operating range. The position of piston 5, is determined by the prevailing sensible load as sensed by a space or return air dry bulb temperature sensor. For most comfort cooling applications sensible heat control is
30 sufficient and piston 5, is the only required control element. For applications that need reduction of latent capacity, there is another piston, piston 7, located in the return pipe header.

For maximum latent capacity piston 7, is at it's lower most position. Elevating piston 7, thus cutting off water flow through the lowest circuits 8, results in a reduction of latent capacity. The condensate forming on the cold surface in the region of active circuits 4, reaching coil surface around inactive circuits 8, is partly or fully evaporated, thus reducing latent capacity and increasing the sensible by evaporative cooling the air passing through the lower portion of the coil. Should it be desired, by sufficiently elevating piston 7, in return header 3, a position is reached where the cooling coil is doing pure sensible cooling without effecting the total moisture content of the air stream.

Referring to FIG. 2, typical cooling only air handling unit 9, equipped with air filter 10, cooling coil 1, and supply air fan 11. In this illustration only the lower, shaded half of cooling coil 1, is active. Return air enters at location 12, outside / fresh air enters at 13, and the supply air leaves the air handling unit at location 14. The outside air entering at location 13, is guided by ducting / baffles 15, to the lower, active portion of cooling coil 1, where, depending on it's dry and wet bulb temperatures, it is cooled and dehumidified. Since the lower circuits of cooling coil 1, will remain active as long as there is sensible load, humid outside air can effectively be treated without the need for an air handling unit dedicated to outside air treatment alone.

Referring to FIG. 3, where one particular method of positioning of piston 5, in supply pipe header 2, is illustrated. External control valve 16, is of low pressure drop type when in the full open position, as for example a butterfly valve. The degree of opening of this motorised valve is determined by space or

return air temperature deviation from setpoint, thus by the prevailing sensible load on the coil. The weight of piston 5, is chosen to equal the design pressure difference between supply header 2, and return header 3. For this free floating
5 piston 5, to remain stationary, the supply header pressure acting on it's bottom must equal the return header pressure acting on it's top plus the weight of the piston. As the weight of piston 5, is fixed according to the design pressure drop of the coil, any deviation from design DIFFERENTIAL PRESSURE will
10 move this piston up or down until a new balance is reached and the coil DIFFERENTIAL PRESSURE is at design again. Evidently having the differential pressure constant will ensure constant water flow velocity in the active circuits, although the number of active circuits changes according to the heat load. Let us
15 assume that the differential pressure across the coil is at design with the illustrated position of piston 5, and current opening of control valve 16. Should the air side load on the coil increase, valve 16, is opened some more, thus the pressure drop at valve 16, is reduced. This pressure drop change at
20 valve 16, shows up as a pressure differential increase across the coil, piston 5, is no longer in balance at it's current position and starts to ride up, permitting water to flow through more circuits, thus reducing the differential pressure across the supply and return pipe headers. The upward progress
25 of piston 5, stops when balance is achieved, that is the differential pressure across the coil has dropped back to it's design value. Should the air side load decrease, the opposite happens and piston 5, moves down, closing off circuits until the new balance position is reached. From another perspective
30 piston 5, is acting as a gravity operated pressure relief valve.

The constant differential pressure maintained by piston 5, ensures constant velocity in the active circuits and the number of active circuits is dependent on the position of this piston, knowing the position of piston 5, provides an accurate means of measuring the quantity of water flowing through the coil. In this particular embodiment an ultrasonic transducer / receiver 17, is placed at the upper end of supply header 2. With it's associated electronic circuitry the ultrasonic transducer / receiver operates as an echo sounder and measures the distance of piston 5, relative to the piston's upper most position. The coil manufacturer's data can accurately relate the position of piston 5, to water flow rate. The addition of entering and leaving water temperature sensors will provide the necessary inputs to compute the energy used by the coil. Temperature sensors are not illustrated in FIG 3. Mechanical stop 18, is to prevent piston 5, from going all the way to the bottom of supply header 2 and cutting off the entering water supply connection.

Water side balancing of the system and providing self balancing facility, as explained earlier, requires that it is known when piston 5, is at or near it's upper most position. Evidently this echo sounder type device is more than capable of providing the required information and with some additional electronic circuitry to indicate water side balance and or to interlock control valve 16, to facilitate self balancing action.

Referring to FIG. 4, where an alternative method of indicating the position of piston 5, is illustrated. Multi turn potentiometer 19, is direct coupled to threaded rod 21, which in turn supported by bearings 20, and 22. Bearing 20, also

contains a water tight seal. Threaded rod 21, passing through piston 5, is meshed with female thread contained within piston 5, thus any up or down displacement of piston 5, from it's position causes threaded rod 21, to rotate. This in turn
5 rotates potentiometer 19, and a change in resistance indicates the location of piston 5. The pitch of the thread on threaded rod 21, is high, in the order of a few turns representing the full travel of piston 5. To prevent piston 5, from turning about threaded rod 21, there is a vertical protrusion on the
10 inner wall of supply header 2, and a matching groove in piston 5. This is not illustrated in FIG. 4, as there are numerous other ways to achieve this.

Referring to FIG. 5, for water side system balance indication
15 and or for self balancing only the near and at the uppermost position of piston 5, is of relevance. This may be accomplished without monitoring the full travel position of piston 5, thus in a much simplified manner. In this embodiment piston 5, contains a permanent magnet 23. Magnetic reed switches 24, and
20 25, are located on the outside of and near the top of supply pipe header 2. When piston 5, is at it's illustrated position both reed switches are in the off / normally open position. As piston 5, moves up and permanent magnet 23, is in line with reed switch 24, it closes and indicates full water flow at
25 correct differential pressure and velocity across the coil. For self balancing when magnetic reed switch closes it prevents control valve 16, from opening further. Should control valve 16, already be too far open, thus permitting excessive water flow, piston 5, rises higher and reed switch 25, is activated
30 by permanent magnet 23. When reed switch 25, is in the closed position it commands valve 16, to slowly close, thus reduce the water flow rate until piston 5, drops down sufficiently to

permit reed switch 25, to open. During normal operation, that is when piston 5, is away from it's upper position, control valve opening is set by space or return air deviation from setpoint. Reed switches 24, and 25, are only acting as an upper
5 limit to indicate and or to prevent excessive water flow across the coil, thus facilitating water side system balancing and or actual balance.

Referring to FIG. 6, where an integrated system powered control
10 valve is shown as an alternative to external control valve 16, of FIG. 3, and 4. This system powered valve consists of piston 26, cylindrical protrusion 27, and valve seat 28, housed in the upper enlarged portion of return pipe header 3. Small diameter pipe 29, originates in entering water supply pipe and via
15 filter 30, and solenoid valve 31, can supply high pressure water to space above piston 26. The water pressure available via solenoid valve 31, is greater than the pressure in return header 3, thus piston 26, is forced downwards, restricting the water flow rate and ultimately shutting it off when the lower
20 lip of protrusion 27, is in contact with valve seat 28. To move the control valve to a more open position solenoid valve 33, is opened, relieving the pressure above piston 26, and discharging the excess water via pipe 34, into the return water pipe. When solenoid valves 31, and 33, are closed piston 26, maintains
25 it's position. Pulsed opening of solenoid valves 31, and 33, moves piston 26, to the desired position, thus sets the required water flow rate. Protrusion 27, may be shaped other than cylindrical to facilitate linear valve characteristics. In normal operation position of piston 26, is set by the
30 prevailing sensible load, derived from temperature deviation of space or return air from setpoint for constant volume air handlers. For variable volume air handlers the supply air

temperature deviation from setpoint is the driver. If equipped with some form of position indication of weighted free floating piston 5, flow monitoring and water side system balance are handled as described in conjunction with FIG. 3, 4, and 5.

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Referring to FIG. 7, where the "weight" of piston 5 is variable, thus the differential pressure across the coil and consequently the chilled water flow velocity in the circuits is also variable. In this illustration piston 5 does not contain a weight. Vertical supply pipe header 2, is extended via a 90 degree bend into a horizontal section 42. Piston 43, is in this horizontal portion 42, of supply pipe header. Loose fitting balls 44, between piston 5, and piston 43, act as flexible "push rod" and transfer force between the two pistons. High pressure system water enters lower chamber of cylinder 36, via pipe 29, filter 30, fixed orifice 38, and pipe 39. The upper chamber of cylinder 36, is connected to the return pipe header 3, via pipe 40. There is a weighted free sliding piston 35, in cylinder 36, separating the lower and upper chambers. The weight of piston 35, is such that when the differential pressure between supply and return headers, 2, and 3, respectively is at design, piston 35, remains in the same relative vertical position to cylinder 36. Should the differential pressure increase, piston 35, moves upwards, further exposing pipe entry 41, which via pipe 34, relieves the excess pressure. The water quantity entering lower chamber of cylinder 36, is restricted by fixed orifice 38, and up and down movement of piston 35, governs the quantity of water leaving via pipe 41. In order for this balance to prevail, the differential pressure across the coil needs to remain constant and this is achieved by pipe 39, conveying the same pressure as in lower chamber of cylinder 36, to horizontal portion of

supply header 42. Thus the same pressure is acting on right hand surface of piston 43. When the pressure on the right hand side of piston 43, equals the pressure on lower side of piston 5, the system is in balance, thus operating at design differential pressure. Should the differential pressure decrease, due to downward movement of control piston 26, or a pressure reduction in the system distribution piping, pilot piston 35, drops down, closing off water entry to pipe 41, permitting more water to enter header pipe extension 42. Due to the reduced differential pressure across the coil pressure acting on lower surface of piston 5, is less than the pressure on the right side of piston 43, consequently, piston 43, acting via spacer balls 44, will force piston 5, to move down. Piston 5, in turn cuts off water flow to some more circuits, resulting in an increased differential pressure across the coil. The downward movement of piston 5, stops when the coil differential pressure is again at it's design value, thus the new balance point is reached and pilot piston 35, is permitting the same water quantity to escape via pipe 41, as the quantity entering via fixed orifice 38. Cylinder 36, is made of non ferrous material and pilot piston 35, is of ferrous material and solenoid 37, surrounds the upper portion of cylinder 36. When there is no current flow in solenoid 37, the weight of pilot piston 35, sets the magnitude of the coil differential pressure, acting as a gravity pressure relief valve. Introducing a current into solenoid 37, the magnetic force acting on pilot piston 35, is counter acting some of it's weight, thus the differential pressure setpoint for the coil is reduced. Thus different differential pressure setpoints may be maintained by changing the flow of current in solenoid 37. The surface temperature of the coil, which in turn depends on circuit flow velocity, effects the latent capacity of the coil

more than it effects it's sensible capacity. Therefore in this embodiment the space or return air relative humidity controls the differential pressure of the coil and the dry bulb temperature is in command of total flow by positioning piston 5 26, of the integral control valve. There is an interlock, not illustrated, when piston 26, is in it's upper most position, thus the control valve is full open, yet the coil is unable to meet the sensible load, the latent call for high circuit flow velocity is ignored and more circuits are made available for 10 sensible cooling. To achieve this the current flow in solenoid 37, is increased, reducing the coil differential pressure setpoint, thus more active coil surface area is presented to the air stream. The foregoing latent versus sensible capacity ratio control is and alternative to utilising piston 7, placed 15 in the return header 3, of the coil, as described in conjunction with FIG. 1.

Referring to FIG. 8, illustrating a simple low cost approach, without compensation for system pressure variations, utilising 20 a system powered control valve. Should the sensible load decrease, solenoid valve 31, opens, admitting more water into chamber on right side of piston 43, which in turn forces piston 5, downwards via ball shaped spacers 44, cutting off water flow in some more circuits. Keeping solenoid valve 31, open after 25 all the circuits are cut off, when lower surface of piston 5, reaches valve seat 45, all water flow through the coil is stopped. This would be the case when this particular air handling unit is not in service. Should the sensible load increase, solenoid valve 33, is opened permitting water to flow 30 out from header 42, on right side of piston 43. System pressure acting on piston 5, forcing piston 5, spacers 44, and piston 43, up and to the right and permitting water to flow from

supply header 2, into some of the circuits. When the coil cooling capacity is matching the air side cooling load, solenoid valve 33, is closed. Valves 31, and 33, remain in the closed position, the number of active circuits remain the same, until there is a change in the air side load. For constant volume air handlers solenoid valves 31, and 33, are controlled by deviation from space or return air temperature setpoint, for variable volume air handlers by deviation from supply air temperature setpoint.

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Referring to FIG. 9, an alternative method of positioning piston 5, is shown. Geared motor 45, drives worm screw 21, and piston 5, is coupled to worm screw 21, by a matching female thread. When geared motor 45, turns worm screw 21, in one direction, piston 5, is moved upwards, reverse rotation causes piston 5, to move downwards. When piston 5, is driven all the way down and it's lower surface contacts valve seat 46, all water flow through the coil is stopped. There is a vertical protrusion on the inner surface of supply header 2, with matching groove in piston 5, to prevent piston 5, to turn about it's axis when it is being repositioned by worm screw 21. The vertical protrusion and groove in piston 5, are not illustrated. Bearings 20, and 22, are to maintain axial and radial positions of worm screw 21, and bearing 20, also contains a water tight seal. Geared motor 45, is under the control of prevailing sensible air side load. The optional latent / sensible capacity ratio control is accomplished by piston 7. The mechanism to position piston 7, in return header 3, is identical to the one described above in conjunction with piston 5. Geared motor 47, positioning piston 7, is under the command of space or return air relative humidity deviation from setpoint.

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Referring to FIG. 10, where a system powered version of positioning piston 7, in return header 3, is illustrated.

System water may enter or escape from flexible bellows 48, via
5 small diameter pipe 32, and bellows 48, in turn moves piston 7,
to the desired position. Control piping arrangement, filter and
solenoid valves are the same as illustrated in FIG. 8. The
positioning signal for piston 7, originates from relative
humidity deviation from setpoint.

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Referring to FIG. 11, illustrating three stage control of
cooling coil by solenoid valves. Dividing plates 49, placed in
supply pipe header 2, create three separate chambers in supply
header 2. The water flow to each chamber is controlled by
15 individual solenoid valves 50, 51, and 52. In the illustration
valves 50, and 52, are closed and 51, is open, thus only
circuits 4, are active and there is no water flow in circuits
6, and 8. The current operation mode is at reduced sensible
capacity also at reduced latent / sensible capacity ratio. In
20 place of solenoid valves motorised on / off valves or
modulating control valves may be utilised and the groups of
circuits and valves are not limited to three.

Referring to FIG. 12, a self propelled hydraulic powered piston
25 assembly is illustrated. This piston assembly is placed in the
supply pipe header 2, of the cooling / heating coil. The three
chambers 53, 54, and 55, may be pressurised via connecting
pipes 56, 57, 58, respectively. Applying pressure to chamber
53, via pipe 56, expands flexible bellows 59, forcing split
30 friction ring 60, against the inner wall of pipe header 2, thus
clamping the upper part of the piston assembly in place.
Pressurising chamber 54, via pipe 57, will clamp the lower part

- of this piston assembly in position. Delivering pressurised fluid to chamber 55, via pipe 58, moves the upper and lower portions apart, as in illustrations A, & C. Permitting fluid to flow out from chamber 55, lets the upper and lower portions to move to close proximity as in illustration B. If chamber 53, is pressurised and chamber 54, is not under pressure, the lower portion of the piston assembly will move down when fluid is delivered to chamber 55, and move up when fluid is permitted to flow out from chamber 55. The upper part of the assembly may move up or down when the lower part is clamped in place in a similar manner. Thus alternate application of fluid pressure to chambers 53, 54, and 55, enables the piston assembly to "climb" up or down in pipe header 2.
- Referring to FIG. 13, the pressurised fluid is derived from the system, flows through replaceable filter 30, fixed orifices 61, flexible pipes 56, 57, 58, to chambers 53, 54, and 55. When all three solenoid valves 62, are closed, the three chambers 53, 54, and 55, are pressurised. Opening one or more of the solenoid valves relieves the pressure in the respective chamber / chambers. An electronic controller, not illustrated, generates the sequential signal to drive the piston assembly up or down according to the prevailing load on the coil. Alternative to utilising system power, a dedicated hydraulic or pneumatic pressure source may be used. Another alternative is to incorporate solenoid valves 62, in the piston assembly and bring the connecting wires out from header 2, in place of flexible hydraulic pipes 56, 57, 58.
- Referring to FIG. 14, where the clamping of upper / lower portions of piston assembly is accomplished by magnetic force. This is for mild steel pipe headers only. Solenoid 63, is wound

around ferrous bobbin 64. Split rings 65, are made of plastic and ferrite mixture. When solenoid 63, is energised the magnetic circuit is completed via ferrous bobbin 64, split rings 65, and steel pipe header. Split rings 65, expand and clamp the powered end of piston assembly in place. Fluid at system pressure is admitted to central chamber 55, via orifice 61, and when solenoid valve 62, is closed, expands bellows 59, as shown in illustrations A & C. Opening solenoid valve 62, relieves the pressure in central chamber 55, thus reducing the distance between upper and lower portions of the piston assembly, as per illustration B. Flexible wiring loom, not illustrated, connects the three solenoids to an external electronic sequencer.

Referring to FIG. 15, a magnetic clamping arrangement is shown suitable for non ferrous also for ferrous pipe headers. Split ring 65, of plastic and ferrite mix is loaded by springs 66, to expand and clamp against pipe header 2. In this configuration solenoid 63, is energised to enable free movement of piston assembly portion within the pipe header. Closed and open positions of solenoid valve illustrated in A & B. Split ring 65, and springs 66, details in illustration C.

Referring to FIG. 16, a long tubular flexible diaphragm 67, is fitted inside pipe header 2. The upper expanded circular end of diaphragm 67, is connected to pipe 71, and the lower collapsed semi circular end is fastened to pipe header 2, at location 70. System fluid via filter 30, forced by pump 68, via non return valve 69, and connecting pipe 71, into below 67, extends the circular portion of diaphragm 67, downwards, closing off additional circuits. Turning pump 68, off and opening solenoid valve 62, fluid is permitted to flow out from diaphragm 67,

collapsing more of the circular section into semi circular and permitting water flow through more circuits of the coil.

Illustrations A. & B. show expanded circular and collapsed semi circular sections of diaphragm 67, respectively. This

5 particular configuration is suitable for cooling coils in humid tropical climate, where there is always some sensible and latent load on the coil and positive shut off of the water flow is not critical.

10 Referring to FIG. 17, where an additional tubular diaphragm 75, is placed inside return pipe header 3. Fully inflating diaphragms 67, and 75, results in positive shut off of water flow through the coil. As diaphragm 75, shuts off flow through the lower circuits, during normal operation it is under
15 relative humidity control, while diaphragm 67, is under sensible heat control. Running pump 68, and opening solenoid valves 72, 74, will reduce sensible and latent capacity respectively and near independently. Opening solenoid valves 62, 73, will result in near independent respective increase in
20 sensible and latent capacity. Upright pipe extension 76, is to ensure removal of air trapped in diaphragm 75. The length of diaphragm 75, may be 1/2 - 1/3 or less of pipe header 3.

Referring to FIG. 18, there is a heavy ball 77, inside
25 diaphragm 67, thus eliminating the need for controls pump 68, as shown in FIG. 16, & 17. Opening solenoid valve 72, admits more fluid into tubular diaphragm 67, and cuts off flow to additional circuits. Opening solenoid valve 62, permits fluid to scape out of diaphragm 67, resulting in more circuits made
30 open to flow.

Referring to FIG. 19, where tubular diaphragm 67, is turned 90

deg. clockwise, looking down supply header 2, end extended all the way to the bottom of same. Entering pipe connection 78, to supply header 2, is turned 90 deg. anti - clockwise and split into two connections to header 2. In this configuration full
5 shut off of the chilled water flow through the coil is possible, besides circuit by circuit control of sensible coil capacity.

Referring to FIG. 20, the pressure inside tubular diaphragm 67,
10 is maintained at a constant differential above the pressure in return pipe header 3. Control fluid pump 68, takes system fluid via filter 30, and delivers this fluid to diaphragm 67, via fixed orifice 61. Pressure relief valve 36, maintains this pressure at a constant level relative to the pressure in return
15 header 3. The pressure differential maintained is equal to the design pressure drop of the coil. The action of diaphragm 67, in this application is same as of free sliding piston 5, described in association with FIG. 3. Butterfly valve 16, sets the quantity of chilled water flowing through the coil and
20 controlled by the sensible load and diaphragm 67, maintains constant differential pressure across the coil by exposing more or less circuits to chilled water flow. Piston 35, of pressure relief valve 36, is made of ferrous material. Increasing / decreasing current flow through solenoid 37, the differential
25 pressure setting of relief valve 36, is changed. The differential pressure maintained across the coil sets the sensible to latent capacity ratio of the coil, thus it is under relative humidity control. When there is water flow in all the circuits, ie. diaphragm 67, is at minimum inflation, permanent
30 magnet 23, comes to close proximity to reed switch 24. Contact closure of reed switch 24, indicates design flowrate across the coil when operating at design pressure drop. This may be used

as an indication to facilitate water side system balancing or used as an interlock to prevent butterfly valve to open more, thus limiting the coil to it's design water quantity.

- 5 Referring to FIG. 21, the function and operation of diaphragm 67, is the same as described in conjunction with FIG. 20. The difference is in replacing the butterfly valve with diaphragm 75, and modification of return pipe connection 79. Solenoid valves 72, and 62, when open, will permit fluid flow to or from
10 diaphragm 75, respectively. The solenoid valves are under sensible load control and diaphragm 75, is acting as a conventional throttle valve.

- Referring to FIG. 22, free sliding piston 80, of positive
15 buoyancy is located in supply pipe header 2. Rotating toothed wheel 81, clockwise transfers balls 82, from reservoir 83, to pipe header 2. The balls 82, force piston 80, downwards. Since piston 80, has a bore through it the pressure above and below the piston are the same, thus only a little force is required
20 to overcome the the buoyancy of piston 80. The balls 82, are of slight positive buoyancy and if any circuit entrance above piston 80, is open to water flow, this flow tends to move the nearest ball to block the open circuit entry. Rotating toothed wheel 81, anti - clockwise transfers balls 82, from supply
25 header 2, to reservoir 83, piston 80, now free to move up and more circuits become open to water flow. Toothed wheel 80, is driven by a gear motor, which is not illustrated, and is under the control of prevailing sensible load.

- 30 Referring to FIG. 23, moving balls 82, from reservoir 83, to supply pipe header 2, is accomplished by pump 68, supplying pressurised fluid via non return valve 69, to the space in

reservoir 83, below piston 84. Opening solenoid valve 62, relieves the pressure under piston 84, in reservoir 83, and permits the balls 82, to move from header 2, to reservoir 83.

- 5 Referring to FIG. 24, illustrating a temperature based method of determining % of circuits in operation. Air handling unit 9, consisting of filter 10, cooling coil 1, and supply air fan 11. Return air enters the air handler 9, at location 12, and supply air leaves at location 14. Temperature sensor T1, is placed
- 10 near the top of cooling coil 1, and temp. sensor T2, is near the bottom of same. There is also a temp. sensor T3, located in the leaving air stream. Where temperature sensed by T3, falls between temperatures sensed by T1, and T2, is proportional to % of coil circuits in operation, except when 100% or 0% of the
- 15 circuits are active. At full and zero load all three temperatures sensed T1, T2, and T3, are the same, however the full or zero load condition is easily determined from the actual value of sensed temperatures.
- 20 Referring to FIG. 25, a long tubular elastic diaphragm 67, is fitted inside pipe header 2. The upper expanded circular end of diaphragm 67, is connected to pipe 71, and the lower collapsed semi circular end is fastened to and sealed at sliding guide 70. Compressed air from air source 85, enters diaphragm 67,
- 25 when solenoid valve 72, is open, via connecting pipe 71, extends the circular portion of diaphragm 67, downwards, closing off additional circuits. Solenoid valve 72, closed and solenoid valve 62, open air is permitted to exhaust out of diaphragm 67, collapsing more of the circular section into semi
- 30 circular and permitting water flow through more circuits of the coil. Illustrations A. & B. show expanded circular and collapsed semi circular sections of diaphragm 67, respectively.

Illustration C shows detail of sliding guide 70. The construction of tubular elastic diaphragm 67, is such, that it exhibits more elasticity along it's length than around it's circumference.

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Referring to FIG. 26, where the advantage of non uniform elasticity of tubular elastic diaphragm 67, is illustrated. Once diaphragm 67, is fully inflated with air of equal pressure to prevailing pressure in supply pipe header 2, further increase in air pressure expands diaphragm 67, downwards.

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Ultimately, reaching the bottom of supply header 2, and shutting off water flow through the coil 100%. This full shut off condition is illustrated here, all other functions are the same as described in conjunction with prior FIG. 25.

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Referring to FIG. 27, a hydraulic fluid of lower than 1 in specific gravity is used to inflate the non uniform elasticity tubular diaphragm 67. The buoyant fluid is to ensure that the uppermost portion of diaphragm 67, takes up circular shape

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first, thus cuts off chilled water flow to the uppermost circuits first. When more hydraulic fluid is admitted, this circular shape extends downwards, cutting off water flow to more circuits progressively. Elevating the hydraulic fluid pressure inside diaphragm 67, above the prevailing pressure in

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supply pipe header 2, expands diaphragm 67, downwards and provides 100% water flow shut off. To increase hydraulic fluid volume and or pressure in diaphragm 67, pump 68, is started and fluid is pumped from reservoir 86, via non return valve 69, and connecting pipe 71, into diaphragm 67. To reduce the fluid

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volume and or pressure in same, solenoid valve 62, is opened and the hydraulic fluid is free to flow back to reservoir 86.

Although not illustrated, reservoir 86, may be directly

pressurised from the chilled water supply pipe in order to minimise the load placed on pump 68.

Referring to FIG. 28, where the same method of positioning of
5 piston 5, in supply pipe header 2, is illustrated, as described
in conjunction with FIG. 3, except that the butterfly valve is
replaced with a variable speed pump 87. The pump speed is set
by speed controller 88, which in turn is derived from space air
temp. deviation from setpoint, thus from the prevailing
10 sensible load on the coil. The weight of piston 5, is chosen to
equal the design pressure difference between supply header 2,
and return header 3. For this free floating piston 5, to
remain stationary, the supply header pressure acting on it's
bottom must equal the return header pressure acting on it's top
15 plus the weight of the piston. As the weight of piston 5, is
fixed according to the design pressure drop of the coil, any
deviation from design DIFFERENTIAL PRESSURE will move this
piston up or down until a new balance is reached and the coil
DIFFERENTIAL PRESSURE is at design again. Evidently having the
20 differential pressure constant will ensure constant water flow
velocity in the active circuits, although the number of active
circuits changes according to the heat load. Let us assume that
the differential pressure across the coil is at design with the
illustrated position of piston 5, and current speed of pump 87.
25 Should the air side load on the coil increase, controller 88,
ramps up the speed of pump 87, which attempts to force more
chilled water through the currently active coil circuits,
resulting in a pressure differential increase across the coil.
Piston 5, is no longer in balance at it's current position and
30 starts to ride up, permitting water to flow through more
circuits, thus reducing the differential pressure across the
supply and return pipe headers. The upward progress of piston

5, stops when balance is achieved, that is the differential pressure across the coil has dropped back to it's design value. Should the air side load decrease, the opposite happens and piston 5, moves down, closing off circuits until the new
5 balance position is reached. From another perspective piston 5, is acting as a gravity operated pressure relief valve.

The constant differential pressure maintained by piston 5, ensures constant velocity in the active circuits and the number
10 of active circuits is dependent on the position of this piston, knowing the position of piston 5, provides an accurate means of measuring the quantity of water flowing through the coil. In this particular embodiment an ultrasonic transducer / receiver
15 17, is placed at the upper end of supply header 2. With it's associated electronic circuitry the ultrasonic transducer / receiver operates as an echo sounder and measures the distance of piston 5, relative to the piston's upper most position. The coil manufacturer's data can accurately relate the position of
20 piston 5, to water flow rate. The addition of entering and leaving water temperature sensors will provide the necessary inputs to compute the energy used by the coil. Temperature sensors are not illustrated in FIG. 28. Mechanical stop 18, is to prevent piston 5, from going all the way to the bottom of supply header 2 and cutting off the entering water supply
25 connection.

Water side balancing of the system and providing self balancing facility, as explained earlier, requires that it is known when piston 5, is at or near it's upper most position. Evidently
30 this echo sounder type device is more than capable of providing the required information and with some additional electronic circuitry to indicate water side balance and or to

interlock pump speed controller 88, to facilitate self balancing action. While the embodiment illustrated in FIG. 28, is not a low first cost solution, however from a standpoint of pumping power, thus operating cost, it represents the most efficient approach. Also incorporates optional features ranging from self balancing to accurate energy measurements of the coil's energy usage.

Referring to FIG. 29, where a slotted cylinder 89, is placed inside supply pipe header 2. The slots 91, on cylinder 89, are progressively longer going from top towards the bottom. This is to ensure that the upper circuits are cut off from chilled water flow prior to progressing sequentially downwards. The opened up and flattened mantle 90, of control cylinder 89, also shows progression of length of slots 91. For progressive and sequential circuit by circuit control, cylinder 89, is rotated through 180 degrees by modulating motor 92. Cylinder 89, is open at the bottom permitting supply chilled water to enter the cylinder. The supply water is admitted from inside cylinder 89, to coil circuits via slots 91. Reference point being at the coil circuit entry, when cylinder 89, is at 0 degree position all the circuits are receiving chilled water, at 180 degree position all the circuits are shut off. This particular embodiment shows two circuits being switched in or out of circulation in single steps. Evidently this concept is also valid for individual circuit by circuit control, also for switching of multiple circuits in single step by arranging slots 91, in an appropriate manner. The embodiment illustrated in FIG. 29, is most suitable for controlling chilled water coils with low number of circuits, such as used in fan coil units and small air handling units. The modulating motor 92, is a conventional one used in standard control applications.

Positioning signal for modulating motor 91, originates from conditioned space temperature deviation from setpoint for constant air volume systems and from supply air temperature deviation from setpoint for variable volume air distribution systems.

In general, it would be evident to a skilled engineer that logical combinations of above described embodiments would also provide functional solutions, exhibiting some or all of the attributes described in the foregoing. Utilising different embodiments the principles of this invention remain the same. Circuit by circuit control, shutting off the upper circuits first and progressing downwards provides sensible capacity control. Circuit by circuit control, shutting off lower circuit first and progressing upwards facilitates latent / sensible load ratio control. Maintaining different water side differential pressure across the coil effects circuit flow velocity, thus temperature rise of chilled water consequently effective coil surface temperature, is another way of effecting latent / sensible load ratio control. At fixed differential pressure across the coil the water flow velocity in the circuits is constant, thus the number of active circuits is directly proportional to the water quantity through the coil, offering an accurate means for measuring water flow rate.

What is claimed is:

1. A fluid heat exchange device, comprising a header and a plurality of interconnecting circuits between an supply port
5 and a return port, the interconnecting circuits being connected to the header by a corresponding plurality of connection ports at different locations along the header wherein the header includes a blocking control element inside the header, the blocking control element being positionally adjustable along
10 the header to selectively block fluid flow from the supply port through the connection ports of the plurality of interconnecting circuits, thereby selectively controlling those interconnecting circuits of the plurality of interconnecting circuits which are subjected to fluid flow therethrough in
15 dependency on the position of the blocking control element.

2. A fluid heat exchange device according to claim 1, wherein the header is an supply header interconnected between the supply port and the plurality of interconnecting circuits.

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3. A fluid heat exchange device according to claim 1, wherein the header is a return header interconnected between the plurality of interconnecting circuits and the return port.

25 4. A fluid heat exchange device, comprising an supply header connected to an supply port, a return header connected to a return port, and a plurality of interconnecting circuits which are connected to each of the supply header and to the return header by a corresponding plurality of connection ports at
30 different locations along each of the supply header and return header, wherein the supply header and the return header each includes a blocking control element inside the header, the

blocking control element being positionally adjustable along the header to selectively block fluid flow through the connection ports of the plurality of interconnecting circuits, thereby selectively controlling those interconnecting circuits
5 of the plurality of interconnecting circuits which are subjected to fluid flow therethrough in dependency on the positions of the blocking control elements of the supply header and return header.

10 5. A fluid heat exchange device according to any of claims 1 or 4, wherein the blocking control element is a piston assembly which is movable along the header, thereby separating the header into two chambers extending along a part of the header dependent on the position of the piston assembly.

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6. A fluid heat exchange device according to claim 5, wherein at the return port a valve is arranged for controlling the fluid flow through the return port, wherein the position of the movable piston assembly in the header is controlled by fluid
20 flow through the return port. (Fig. 3)

7. A fluid heat exchange device according to claim 5, wherein the position of the piston assembly in the header is measured by means of a ultrasonic transducer placed at one end of the
25 header for measuring the distance between the piston assembly and said end of the header. (Fig. 3)

8. A fluid heat exchange device according to claim 5, wherein the position of the piston assembly in the header is measured
30 by means of a multi turn potentiometer located outside of the header, wherein the potentiometer is coupled to a threaded rod coupled to the piston assembly. (Fig. 4)

9. A fluid heat exchange device according to claim 5, wherein the position of the piston assembly near an end of the header is measured by means of a system comprising a permanent magnet at the piston assembly and a cooperating respective magnetic reed switch at a housing of the header, wherein the magnetic reed switch is closed if the permanent magnet is in close proximity to the switch near said end of the header. (Fig. 5)

10. A fluid heat exchange device according to claim 6, wherein at the return port a return control element is arranged for controlling the fluid flow through the return port, wherein the return control element is controlled by a fluid pressure difference between fluid return pressure at the return port and a reduced pressure which is reduced by a reduction valve in proportion to fluid pressure prevailing at the supply port. (Fig. 6)

11. A fluid heat exchange device according to claim 5, wherein the supply port is arranged at one end of the supply header, and the supply header includes an supply header portion at the other end thereof, the supply header portion being extended by a tubular extension portion which is bent at an angle relative to the supply header portion, and the piston assembly is comprised of a first piston and a second piston with a plurality of neutral buoyancy spacer balls therebetween to be movably arranged in the supply header portion and in the extension portion of the header with the first piston located in the supply header portion and the second piston located in the extension portion, wherein the position of the piston assembly is controlled by the fluid pressure difference between fluid pressure acting on the first piston in the supply header

and a fluid pressure acting on the second piston in the extension portion, wherein the fluid pressure is controlled to maintain the desired differential pressure between the supply port and return pipe header. (Fig. 7, 8)

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12. A fluid heat exchange device according to claim 5, wherein a motor is arranged outside that header comprising the piston assembly and is coupled to a threaded rod which is coupled to the piston assembly for driving the piston assembly along the
10 header. (Fig. 9)

13. A fluid heat exchange device according to claim 5, wherein the piston assembly in the header is drivingly supported by a flexible bellows which is fixed at an end of the header,
15 wherein the bellows is connected to a fluid supply to be filled with fluid or be released from fluid, thereby extending and retracting the length of the bellows, respectively and controlling the position of the piston assembly inside the header in dependency on the length of the bellows. (Fig. 10)

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14. A fluid heat exchange device according to claim 5, wherein the piston assembly is comprised of a flexible bellows and a first piston and a second piston, which are arranged at opposite ends of the flexible bellows, wherein each piston
25 includes a radially expandable chamber and a friction ring formed at the circumference of the chamber which friction ring is adapted to be pressed at the inner wall of the header for fixing the respective piston, wherein the flexible bellows and the chambers each are controllably connected to a fluid supply,
30 from which the chambers or the flexible bellows may be separately supplied with fluid pressure or released from fluid pressure, so that, for moving the position of the piston

assembly, the first piston is adapted to be fixed at the wall of the header by supplying pressure into the chamber of the first piston, and the second piston is then displaceable along the header direction by supplying pressure into the flexible bellows, and in the displaced position the second piston is adapted to be fixed at the wall of the header by supplying pressure into the chamber of the second piston, whereafter the first piston is removable from the wall of the header by releasing of fluid pressure from the first piston and is displaceable towards the second piston by releasing of fluid pressure from the flexible bellows. (Fig. 12, 13)

15. A fluid heat exchange device according to claim 5, wherein the header is comprised of a magnetizable material, and the piston assembly is comprised of a flexible bellows and a first piston and a second piston, wherein the first piston and the second piston are arranged at opposite ends of the flexible bellows, wherein each piston includes an electromagnet and a radially expandable clamp ring formed at the circumference of each piston, wherein the clamp ring is adapted to be pressed at the inner wall of the header for fixing the respective piston, wherein the flexible bellows is controllably connected to a fluid supply from which the flexible bellows may be supplied with fluid pressure or released from fluid pressure, so that, for moving the position of the piston assembly, the first piston is adapted to be fixed by the clamp ring thereof to the wall of the header by energizing the electromagnet of the first piston, and the second piston is then displaceable along the header direction by supplying pressure into the flexible bellows, and in the displaced position the second piston is adapted to be fixed by the clamp ring thereof to the wall of the header by energizing the electromagnet of the second

piston, whereafter the first piston is removable from the wall of the header by de-energizing the electromagnet of the first piston and is displaceable towards the second piston by releasing of fluid pressure from the flexible bellows. (Fig.

5 14)

16. A fluid heat exchange device according to claim 5, wherein the blocking control element is a diaphragm extending along the header from the supply port at one end of the header to the
10 other end of the header along the plurality of connection ports, wherein the diaphragm is adapted to be filled with fluid, so that the connection ports located along the diaphragm are closed if the diaphragm is filled, and the connection ports are subsequently opened one after another in proportion to a
15 fluid pressure reduction in the diaphragm beginning from the one end of the header at the supply port to the other end of the header. (Fig. 16)

17. A fluid heat exchange device according to claim 5, wherein
20 the supply port is arranged at one end of the supply header, wherein the piston assembly is comprised of a piston and a plurality of neutral buoyancy spacer balls, wherein the neutral buoyancy spacer balls are located in the chamber opposite to the supply port and said chamber is connected with a neutral
25 buoyancy spacer ball reservoir, wherein a transfer means is provided which transfers neutral buoyancy spacer balls from the reservoir to said chamber or from said chamber to the reservoir, thereby the position of the piston is controlled dependent on the number of neutral buoyancy spacer balls in
30 said chamber. (Fig. 22)

18. A fluid heat exchange device according to claim 5, wherein

at the supply port a pump is arranged for controlling the fluid flow through the supply port, wherein the position of the movable piston assembly in the header is controlled by fluid flow through the supply port. (Fig. 28)

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19. A fluid heat exchange device according to claim 5, wherein the blocking control element is a sleeve which is rotatable in the header and comprising slots at its circumference at locations corresponding to those of the connection ports,

10 wherein the slots have different circumferential lengths so that different numbers of connection ports are in dependence of the rotational position of the sleeve. (Fig. 29)

20. A fluid heat exchange device according to any of claims 1
15 to 19 and connected as a cooling fluid device.

21. A fluid heat exchange device according to any of claims 1
to 19 and connected as a heating fluid device.

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