United States Patent

[11] 3,596,474

[72] Inventors **Alden T. Bloxham**
Bridgeville, Pa.;
Harry C. Fischer, Royal Oak, Md.

[21] Appl. No. **784,597**
[22] Filed **Dec. 18, 1968**
[45] Patented **Aug. 3, 1971**

[73] Assignee **Kellogg American, Inc.**
Oakmont, Pa.
by said Harry C. Fischer

---

[54] **GAS-HANDLING APPARATUS AND METHOD**
16 Claims, 7 Drawing Figs.

[52] **U.S. Cl.** 62/93, 62/113, 62/85, 62/513, 62/272
[51] **Int. Cl.** **F25d 17/06**
[50] **Field of Search** 62/85, 93, 513, 113, 272

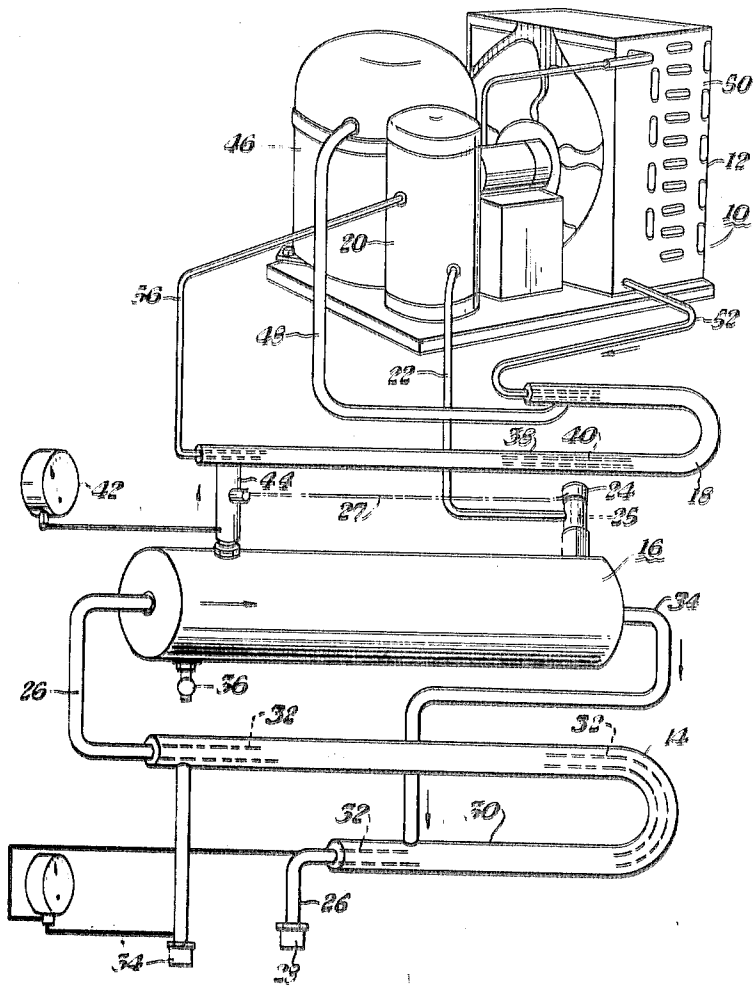
[56] **References Cited**

UNITED STATES PATENTS

1,853,236	4/1932	Shadle	62/93
1,909,227	5/1933	Shadle	62/272
2,051,971	8/1936	Swart	62/513
2,120,764	6/1938	Newton	62/573
2,477,772	8/1949	Simpson	62/93
2,745,254	5/1956	Malkoff	62/513
2,841,965	7/1958	Etherington	62/113
3,041,842	7/1962	Heinecke	62/93
3,050,954	8/1962	Royse	62/93
3,359,753	12/1967	Fiedler	62/93

Primary Examiner—William J. Wye
Attorney—Don J. Smith

ABSTRACT: We disclose gas-handling apparatus comprising a primary evaporator, first conduit connections for establishing a flow path for a compressed gas and the like through said evaporator, second conduit connections for establishing a flow path of a refrigerant fluid through said primary evaporator in heat exchanging relation with said gas path, a refrigerant unit coupled to said second conduit connections, and an auxiliary evaporator coupled between said refrigerating unit and said primary evaporator.



PATENTED AUG 3 1971

3,596,474

SHEET 1 OF 3

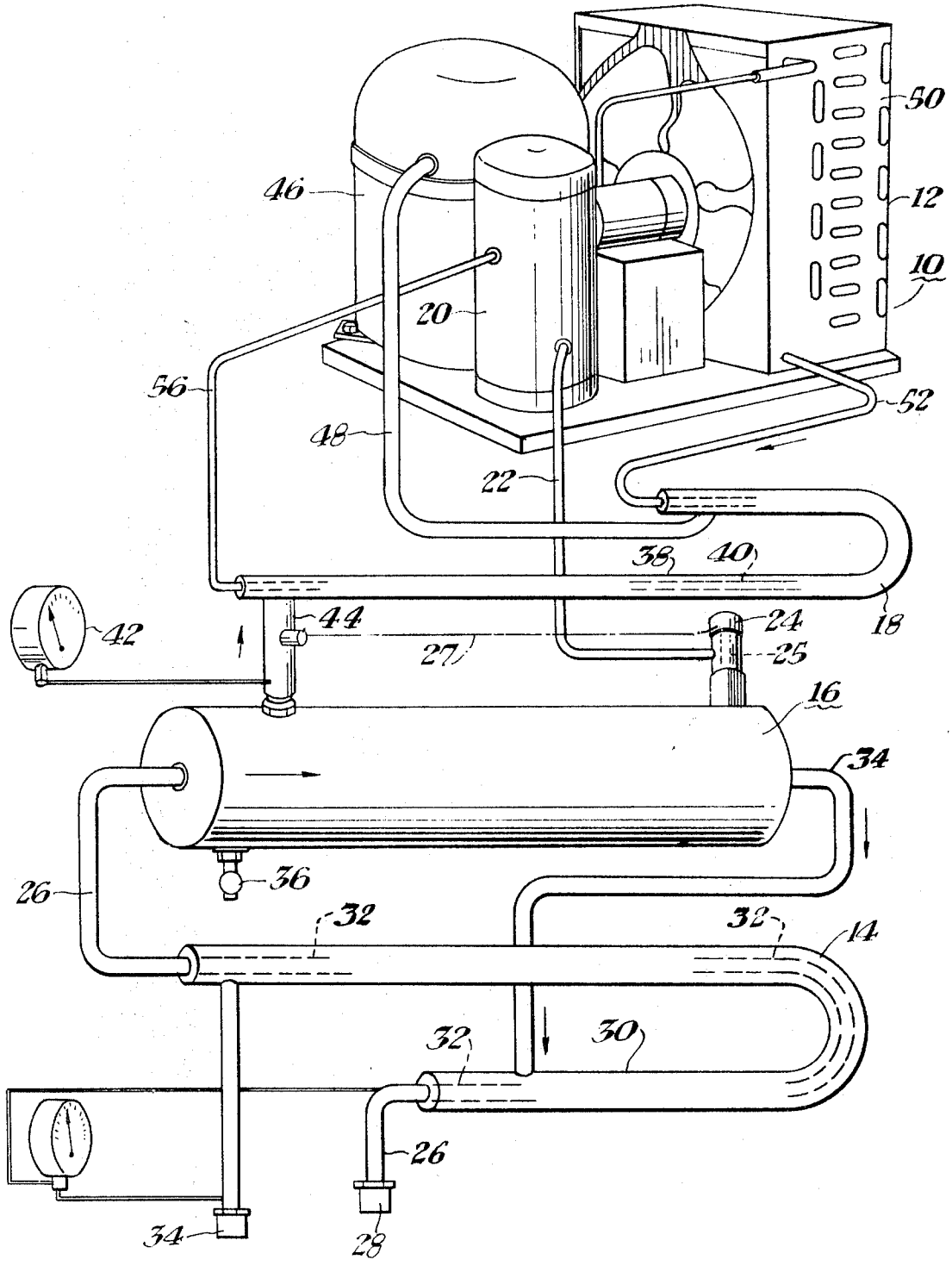


Fig. 1.

INVENTORS
Alden T. Bloxham &
Harry C. Fischer.
BY
Buell, Blenko & Ziesenheim
THEIR ATTORNEYS

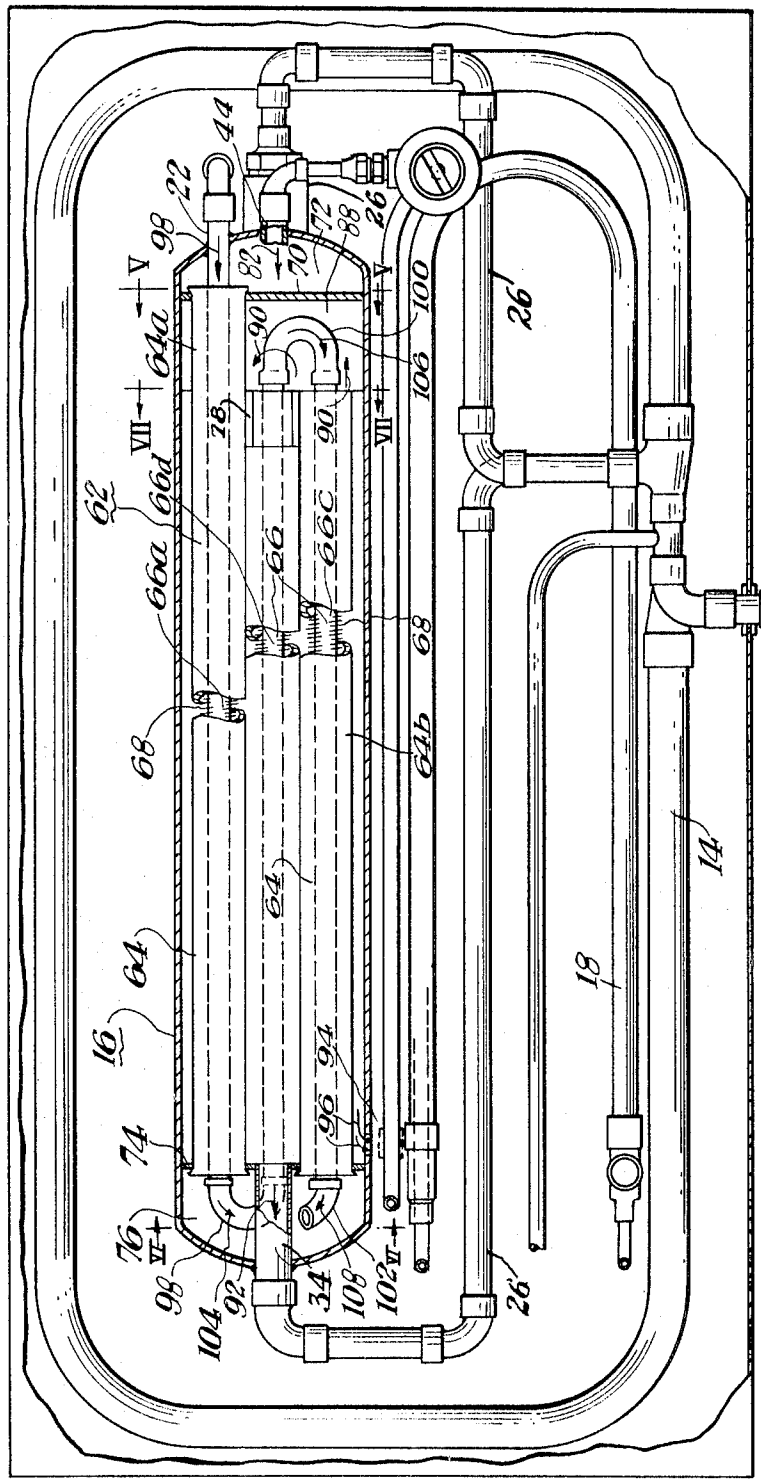


Fig. 2.

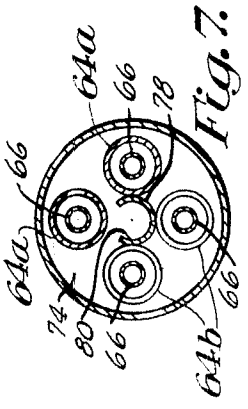


Fig. 7.

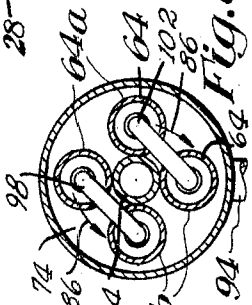


Fig. 6.

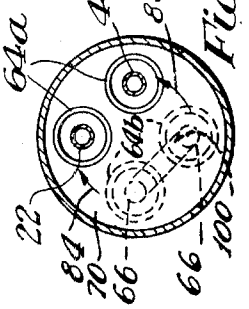


Fig. 5.

INVENTORS
Alden T. Bloxham &
Harry C. Fischer
BY
Buell, Blenko & Ziesenheim
THEIR ATTORNEYS

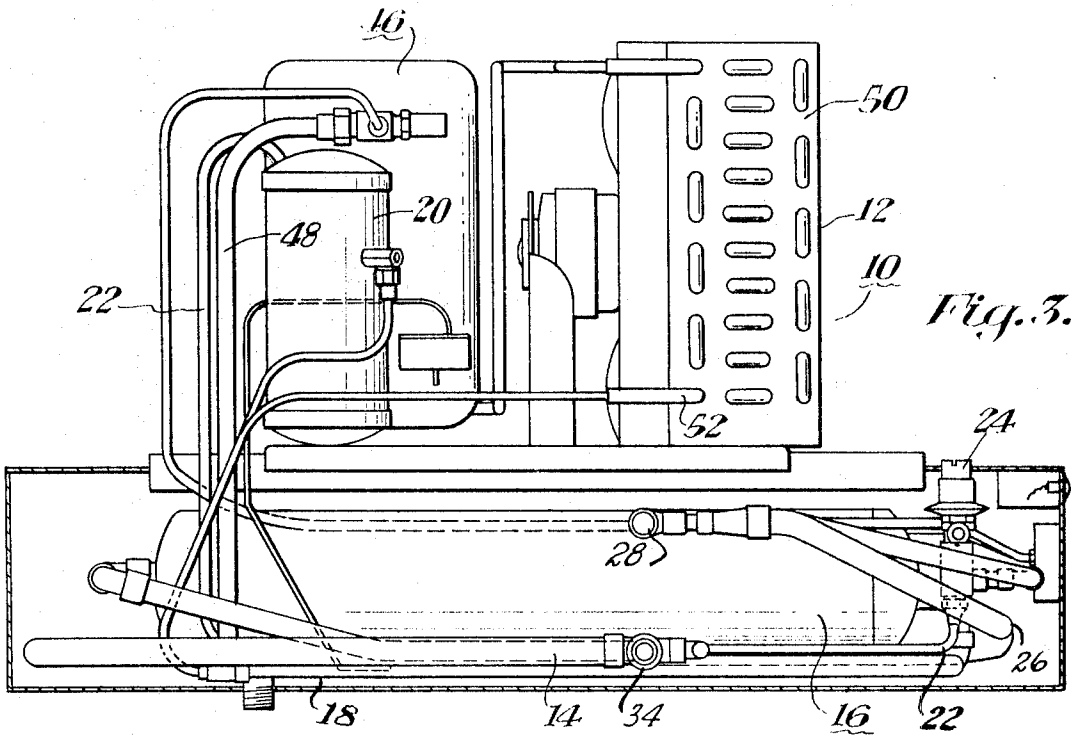


Fig. 3.

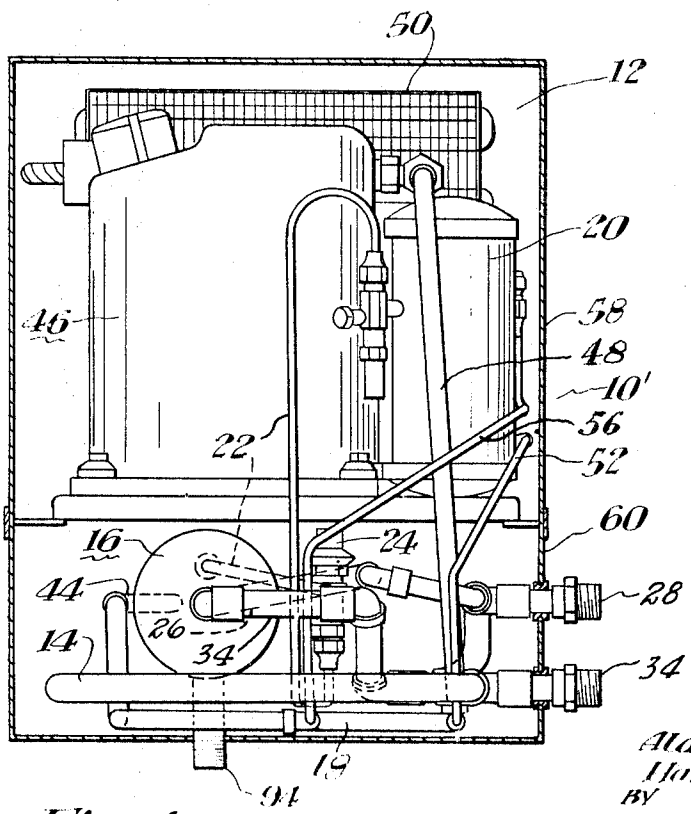


Fig. 4.

INVENTORS
Alden T. Bloxham &
Harry C. Fischer
BY
Buell, Blenko & Zeuthein
THEIR ATTORNEYS

GAS-HANDLING APPARATUS AND METHOD

The present invention relates to means and methods of handling compressed gases and more particularly to dryers of the refrigeration type for compressed air and other gases.

Although our invention is described herein primarily with reference to compressed air handling equipment, it will be obvious as this description proceeds that our invention can be used with other gases or combinations of gases.

A variety of equipment for cleaning and/or drying compressed air and other gases is known. Widely ranging operating conditions have made such equipment difficult to control with the desired stability. For example, in known refrigeration drying systems, stable control is difficult where the drying function is variable, i.e., when the compressed air or other gas is supplied in differing degrees of wetness. Simple pressure and temperature control means usually are not adequate for the required degree of control. The separated moisture may freeze in parts of the system and render it inoperative.

The operational difficulties of conventional equipment for handling compressed air and other gases are further aggravated where the compressed gas must be supplied at widely varying conditions of pressure and temperature. Conventional equipment has been unable to supply, without complex controls and other complicated components, a consistent dewpoint dryness under these varying conditions, which are further complicated by variations in ambient air temperatures. For example, in industrial applications, compressed air handling equipment may be used for hot wet incoming air varying from 10 p.s.i.g. to 1,000 p.s.i.g. at an inlet temperature of 130° F. or higher. Ambient temperatures may vary between 40° F. and 110° F. Under these conditions rather complicated conventional equipment is required to provide cooled, compressed gas with an acceptable and consistent dewpoint dryness. Insofar as we are aware there is no available equipment which is capable of continuous operation and wherein the compressed air drying function can be varied at will between zero load and full load while maintaining a consistent dewpoint dryness.

In the past, a number of air handling systems have been proposed, typified by the U.S. patents to Coblentz, U.S. Pat. No. 2,632,315; Ritter, U.S. Pat. No. 2,513,679; Meckler, U.S. Pat. No. 3,102,399; Shipman, U.S. Pat. No. 2,257,983; Cook et al., U.S. Pat. No. 2,119,201; Schweller, U.S. Pat. No. 2,692,481; Newton, U.S. Pat. No. 2,367,305; Carrier, U.S. Pat. No. 2,154,263; Kohut, U.S. Pat. No. 2,835,476; and Lund, U.S. Pat. No. 2,451,682, all of which suffer from the aforementioned defects. Most of these prior systems utilize various arrangements of evaporators and regenerative heat exchangers for the incoming air or other gas which is being cooled, cleaned and/or dehumidified. However, one of these systems is capable of an output of compressed air having consistent dewpoint dryness over a wide range of operating or ambient conditions. In particular, none of these references employs primary and auxiliary evaporators together with other novel features of our invention, which are described below. As mentioned above, freezing is likely to occur under certain load conditions.

Freezing of moisture in such equipment has been prevented in the past by brute force techniques involving for example a capillary expansion valve coupled with a hot-gas bypass valve. The hot-gas bypass valve is particularly subject to erosion of the bypassed gases as a result of scouring and cavitation. The bypass valve is complicated in construction and difficult to manufacture and repair. The capillary expansion valve cannot be regulated properly, save by replacing with a differently sized valve.

Thermostatic expansion valves also are commonly used for controlling gas handling and drying equipment. These valves are rather complex and are subject to hunting. Moreover, the control function provided by thermostatic expansion valves is confined to a very narrow range of operating conditions.

When the design limits of these valves are exceeded, even slightly, the control function is largely lost.

We overcome these deficiencies of the prior art by providing a continuously operable refrigeration drying system capable of consistent dryness in output air irrespective of variable operating and ambient conditions. Our invention engenders stability and controllability through the use of a unique arrangement of primary and auxiliary evaporators. These characteristics are further enhanced by use of an automatic expansion valve or the like, in conjunction with our evaporative system. The refrigeration system of our invention is capable of continuous noncycling operation, and the actual compressed air drying function thereof can be selectively varied from zero to full load, while producing a consistent dewpoint dryness of exit air. Although we describe our novel gas-handling equipment primarily in terms of removing or controlling the moisture content of compressed air, the compressed air or other gas is further cleaned by removal of oil and other entrained foreign matter along with condensed moisture. Compressed gas can be dried or otherwise cleaned by removal of other condensibles or entrained matter with our novel equipment.

Our invention also contemplates the use of an "automatic" expansion valve, which in this example is pressure sensitive. This type of expansion valve is of known construction but is provided with an internal pressure transmitting aperture where the valve is sensitized to line pressure. Alternatively, the valve can be provided instead with a bleed line where it is desired to sensitize he valve to pressure changes in an external conduit.

In conventional refrigeration drying systems, a pressure-sensitive expansion valve exhibits relatively poor responses to load changes in the system. We have found, however, that a pressure-sensitive expansion valve exhibits an admirable control characteristic and readily adjusts itself to the compressor load, when the compressor is run constantly and at the same speed. It is also desirable that the refrigerant compressor be provided with adequate size so that it is capable of forcing the "automatic" expansion valve further open under certain conditions. As a result the expansion valve responds quickly to changes in load demand upon the equipment. In this respect, the expansion valve of the invention adjusts readily to load changes at the refrigerant evaporator and compressor. The pressure-sensitive means of the expansion valve in conjunction with the continuously operated compressor controls the expansion valve such that the flow of refrigerant varies through main and auxiliary evaporators of the invention with variation in load on the load or air side of the main evaporator. This diverts the evaporation or condensation from a conventional condenser of the refrigeration unit to the aforementioned auxiliary evaporator and/or diverts evaporation from the main evaporator to the auxiliary evaporator, depending upon the amount and direction of load change. For this purpose, the pressure-sensitive means is coupled to the refrigerant path adjacent its outlet from the main evaporator.

With this arrangement of our invention, freezing of the removed moisture is obviated. Expansion of the refrigerant fluid is closely controlled with respect to load changes, irrespective of variable input and ambient conditions. The arrangement is not sensitive to narrow design limitations nor is it subject to hunting.

We accomplish these desirable results by providing gas-handling apparatus comprising a primary evaporator, first conduit connections for establishing a flow path for a compressed gas or the like through said evaporator, second conduit connections for establishing a flow path of a refrigerant fluid through said primary evaporator in heat-exchanging relation with said gas path, a refrigerating unit coupled to said second conduit connections, and an auxiliary evaporator coupled between said refrigerating unit and said primary evaporator.

We also desirably provide similar gas-handling apparatus wherein said auxiliary evaporator is coupled regeneratively to said second conduit connections.

We also desirably provide similar gas-handling apparatus wherein a second heat exchanger for said gas is coupled regeneratively to said first-mentioned conduit connections.

We also desirably provide similar gas-handling apparatus wherein an expansion valve and a third conduit connection are coupled between said primary evaporator and said refrigerating unit in bypassing relation to said auxiliary evaporator.

We also desirably provide a heat-exchanging structure for an evaporator and the like, said structure comprising a plurality of sheath tubes, a plurality of inner tubes inserted respectively and spacedly within said sheath tubes, and flow means for connecting one group of the group of sheath tubes and the group of coaxial tubes in series and for connecting at least some of the other group of tubes in parallel.

We also desirably provide a method for operating refrigerating gas-handling apparatus having a compressor, condenser, evaporator, and expansion valve, said valve being mounted for expansion of a refrigerant fluid into said evaporator, said method comprising the steps of operating said compressor continuously, preventing complete closure of said valve, sensitizing opening and closing movements of said valve to pressure fluctuations at a given point in said apparatus, and sizing said compressor for operation at constant speed and for forcing said valve further open in response to load changes in a given direction.

During the foregoing discussion, various objects, features and advantages of the invention have been set forth. These and other objects, features and advantages of the invention together with structural details thereof will be elaborated upon during the forthcoming description of presently preferred embodiments of the invention and presently preferred methods of practicing the same.

In the accompanying drawings we have shown certain presently preferred embodiments of the invention and have illustrated presently preferred methods of practicing the same, wherein:

FIG. 1 is a schematic fluid circuit of one arrangement of our compressed gas handling equipment;

FIG. 2 is top plan view, with parts broken away and other parts removed, of a packaged equipment assembly arranged in accordance with FIG. 1;

FIG. 3 is a front elevational view of the apparatus as shown in FIG. 2;

FIG. 4 is a left side elevational view of the apparatus as shown in FIG. 3;

FIG. 5 is a cross-sectional view of the primary evaporator shown in FIG. 2 and taken along reference line V–V thereof;

FIG. 6 is another cross-sectional view of the evaporator of FIG. 2 and taken along reference line VI–VI thereof; and

FIG. 7 is still another cross-sectional view of the evaporator of FIG. 2 taken along reference line VII–VII thereof.

Referring now more particularly to the drawings, an exemplary gas-handling equipment **10** of our invention comprises a refrigeration and condensing unit **12** of known construction, regenerative heat exchanger **14**, primary evaporator **16** and auxiliary evaporator **18**. The receiver **20** of the refrigeration and condensing unit **12** is connected through conduit **22** to expansion valve **24** and thence to the main evaporator **16** where the refrigerant fluid is vaporized. In this example, the expansion valve **24** is controlled automatically by pressure which is the function of degree of vaporization of the refrigerant fluid within the main evaporator **16**.

The refrigerant is further vaporized within the evaporator **16** by heat transfer from hot wet compressed air supplied to the main evaporator **16** through conduit **26**. Between the end of the conduit **26** connected to the evaporator **16** and the inlet end **28** of the conduit **26**, the regenerative heat exchanger **14** is coupled. In this example, conduit **26** is effectively extended through the heat exchanger **14**, which is formed in this example from a pair of coaxial tubes **30**, **32**. The outer heat exchanger tube **30** is coupled to conduit **34** forming the outlet of the main evaporator **16**. Before reaching the main evaporator **16**, the pressurized and dried outgoing air or other gas is therefore warmed regeneratively by heat-exchanging association with the hot wet compressed air passing through inner coaxial tube **32**.

Within the main evaporator **16**, water, lubricating oil, dirt and other foreign matter are condensed or leached out of the pressurized gas passing through the evaporator **16** by heat-exchanging association with the expanding refrigerant. Water or other foreign material is periodically drained from the evaporator **16** through drain valve **36**.

The refrigerant, which is partially or completely vaporized in the evaporator **16** flows thence to the auxiliary evaporator **18**, in this example, consisting of outer and inner coaxial tubes **38**, **40**. Suitable instrumentation **42** can be coupled to outlet refrigerant conduit **44**, by which the primary evaporator **16** is connected to the outer coaxial tube **38** of the auxiliary evaporator **18**. In passing through the auxiliary evaporator **18**, the output refrigerant from the primary evaporator **16** is expanded further and returns to compressor **46** through conduit **48**. The expansion of refrigerant in the auxiliary evaporator **18** in turn removes heat from refrigerant flowing from condenser **50** through conduit **52** which is coupled to the inner tube **40** of the auxiliary evaporator **18** from which it flows to receiver **20** through connecting conduit **56**. The refrigerant is then recycled from the receiver **20** to expansion valve **24** and primary evaporator **16** through conduit **22**.

In effect, then, both the incoming hot wet compressed air and the refrigerant supplied to the main evaporator **16** are cooled regeneratively before entering the primary evaporator **16**. This arrangement permits the gas-handling equipment **10** to be controlled in the stable fashion under a wide range of operation and ambient conditions. In particular, refrigerant is supplied to the condensing unit **50** (conduit **52**) and to the primary evaporator **16** (conduit **22**) at correspondingly lower temperatures, whereby a marked increase in efficiency is obtained.

The valve **24** itself is of conventional construction but is provided with an internal pressure-transmitting aperture **25** to render the valve **24** sensitive to fluctuations in line pressure. Alternatively as denoted by conduit outline **27** the valve **24** can be sensitized through conduit **27** to pressures elsewhere in the apparatus, for example the evaporator outlet conduit **44**. A suitable valve is available from Alco Controls Corp., St. Louis, Mo., for example catalog No. ACP–2 (internal pressure transmitting), ACPE–5 (external pressure transmitting) or CP–300. Other sizes are available depending on particular equipment capacities. Additionally, the valve **24** is provided with a stop member **29** to avoid complete closure of the valve **24**.

In conventional gas-handling apparatus of this type, it is expected that a pressure-sensitive valve, such as the valve **24**, will not closely follow load changes in the system, i.e., both the stability and the control in the system will be poor. However, we attain an unexpectedly stable system having an excellent control characteristic by employing the pressure-sensitive valve **24** in conjunction with a continuously operated compressor **46**. The compressor **46** is not operated intermittently or cyclically as in conventional systems. This eliminates one source of the phenomenon known as hunting in the system.

The use of a continuously operated compressor, such as the compressor **46** in conjunction with the pressure-sensitive valve **24** lays a basis for the vastly improved control characteristic exhibited by our novel apparatus. As the valve **24** is prevented from complete closure, the compressor **46** is sized so that it is capable of forcing the valve **24** more nearly closed when required for load changes in the increasing direction. The valve **24** moves toward its open position upon decreasing load, which increases load on the auxiliary evaporator **18**. While the main evaporator **16** loads, the auxiliary evaporator **18** unloads. The compressor **46** also is selected on the basis of known criteria, for constant-speed operation regardless of the load upon the system, within system capability. The use of a pressure-sensitive valve, and a continuously operated con-

stant-speed compressor, coupled with the ability of the compressor to adjust the expansion valve opening endows the apparatus with considerable stability and with a control characteristic which is capable of compensating wide variations in incoming gas, pressure, temperature, and moisture content, as well as widely varying changes in ambient conditions. Our apparatus, therefore, is capable of supplying cool and dry compressed air or other gas at a consistent dewpoint dryness. At the same time, the cooperative aspects of the pressure-sensitive valve **24** and the continuously operated compressor **46** positively prevent freezing of condensed moisture in any part of the equipment.

As better shown in Figures **2—4** the package equipment unit **10'** is arranged such that the refrigeration and condensing unit **12** is mounted within an appropriately shaped individual casing **58**, while the primary evaporator **16**, auxiliary evaporator **18** and the regenerative eat exchanger **14** are supported within a separate casing **60**. Inlet and outlet conduits **28**, **34** respectively, are supported upon a wall portion of the casing **60**.

Figure **2** of the drawings further illustrates a unique form of our primary evaporator **16** which is useful in the operation of our novel gas handling apparatus **10** or **10'**. The evaporator **16** is arranged for obtaining an unusually high coefficient of heat transfer between air or other pressurized gas entering and leaving the evaporator via conduits **26**, **34** respectively and the refrigerant or other heat exchange fluid entering and leaving the evaporator **16** via conduits **22** and **44** respectively (Figure **4**).

In this arrangement of the invention, the evaporator **16** is provided with a heat-exchanging structure denoted generally by reference character **62**. The heat exchanger **62** in this example includes a plurality of sheath tubes **64** each of which contains a spined inner tube **66** supported substantially coaxially within the respective sheath tubes **64**. The convecting spines **68** of each tube **66** are made by lancing and erecting narrow strips of the wall material of the tubes **66**, and the outer ends of the spines **68** are closely fitted within the associated sheath or tube **64**. The spines **68** are substantially less in thickness than that of the tube walls to avoid any possibility of leakage. Because bases of the spines are rooted integrally in the tube walls, the heat transfer characteristics are excellent.

In the illustrated arrangement the heat exchanger **62** includes four such sheath tubes **64** and a like number of associated or inner spined tubes **66**, although a different number of sheaths and tubes obviously can be utilized as required.

Two of the sheath tubes **64**, for example **64*a***, and the associated coaxial tubes as better shown in Figure **5** are extended beyond the ends of the remaining sheath tubes **64*b*** at the right end of the evaporator **16** as viewed in Figure **2** and are secured to baffle plate **70** where they communicate with plenum chamber **72** which in turn communicates with air inlet conduit **26**.

The other ends of the sheath tubes **64*a*** are secured to baffle plate **74** adjacent the other end of the evaporator **16** and thus communicate with opposite end plenum chamber **76**. Desirably, the remaining sheath tubes, such as the tubes **64*b***, likewise are secured to the right-hand baffle plate **74** for communication with the adjacent plenum chamber **76** and with the first group of sheath tubes **64*a***. The tubes **64*b*** however terminate short of the right hand baffle plate **70** at the opposite end of the evaporator **16** and are supported relative to one of the longer sheath tubes **64*a*** by semicircular ring or bracket **78** (Figure **7**) which desirably is welded or otherwise secured to each of the shorter sheath tubes **64*b*** and to one of the longer sheath tubes **64*a***. Use of the bracket **78** does not interfere with the flow of fluid between and around the outer surfaces of the sheath tubes **64*a***, **64*b*** the purpose of which flow is described below. The flow baffle **74** at the opposite end of the evaporator **16** is provided with a central opening **80** as better shown in Figures **6** and **7** whereat the gas exit conduit **34** is joined and sealed (as by welding or the like—Figure **2**) for communication with fluid flowing between and around the sheath tubes **64**.

With the arrangement just described, it will be seen that compressed air or other gas is caused to flow in parallel-series first through the longer sheath tubes **64*a*** and then through the shorter sheath tubes **64*b***. In addition, the compressed air or other fluid is induced to flow transversely of the spines **68** to enhance heat transfer between the respective fluids within tubes **64**, **66**. More specifically, compressed air enters the left-hand plenum chamber **72**, as viewed in FIG. **2**, from inlet conduit **26** as denoted by flow arrow **82**. From the chamber **72** compressed air flows into both of the longer sheath tubes **64*a*** as denoted by flow arrows **84** in FIG. **5**. After following parallel annular paths through the sheath tubes **64*a*** the compressed air enters plenum chamber **76** at the opposite end of the evaporator **16**. In the plenum chamber **76** the compressed gas flows from the adjacent ends of the sheath tubes **64*a*** to the ends of the sheath tubes **64*b*** as denoted by flow arrows **86** in FIG. **6**. Thence, the compressed gas again follows parallel paths through the spines **68** of the shorter sheath tubes **64*b***.

At the opposite ends of the shorter sheath tubes **64*b*** the compressed gas enters a third plenum chamber **88** which is segregated from the adjacent end plenum chamber **72** by baffle **70**. From the plenum chamber **88**, as denoted by flow arrows **90**, the compressed gas flows between and around the outer surfaces of the sheath tubes **64** to the central opening **80** of the baffle **74** at the opposite end of the evaporator **16**. From the baffle opening **80**, the compressed gas exits from the evaporator **16** through conduit **34** as denoted by flow arrow **92**. Alternatively, the exit conduit **34** can be closed or the conduit **34** and the baffle opening **80** can be omitted, and the compressed gas can be withdrawn via chain-outlined exit conduit **94** which communicates with the plenum **88** through the spaces between and around the sheath tube **64** as denoted by flow arrows **96**.

From the foregoing paragraph, it will be seen that pressure drops are reduced by causing the compressed gas to flow through successive pairs of the sheath tubes **64** in series-parallel fashion. The pressure drop if desired can be further minimized by flowing the gas through all of the sheath tubes in simple parallel fashion. On the other hand, the refrigerant fluid, because of lesser pressure drops, is flowed in series through the refrigerant tubes **66**. Thus, the refrigerant fluid enters conduit **22** as denoted by arrow **98** and flows through refrigerant tube **66*a***. At the other end of the refrigerant tube **66*a***, the fluid is conducted to refrigerant tube **66*b*** through a reverse bend fitting or hose section **98**. The refrigerant tube **66*b*** at its opposite end is similarly coupled by a second hose section **100** to refrigerant tube **66*c***, which in turn is coupled to refrigerant tube **66*d*** through a third hose fitting **102**. The refrigerant tubes **66*a*** and **66*d*** are disposed in sheath tubes **64*a*** while refrigerant tubes **66*b*** and **66*c*** are in the shorter sheath tubes **64*b***. The fourth refrigerant tube **66*d*** communicates with exit refrigerant conduit **44** as shown in FIG. **5**. For convenience, the reverse fittings **98**, **102**, are disposed in plenum **76** while the reverse fitting **100** is in the third plenum **88**, of the evaporator **16**. The flow of refrigerant successively through the refrigerant tubes **66*a***, **66*b***, **66*c*** and **66*d*** and the respectively associated fittings **98**, **100**, **102** is denoted respectively by flow arrows **104**, **106**, **108**.

It will be understood, of course, that other flow paths for the compressed gas and the refrigerant fluid respectively can be provided through our novel heat exchanger structure **62**. We have found however that the aforedescribed flow paths are very effective in the efficient transfer of heat between compressed gas and a refrigerant fluid. It will be further understood that other heat-exchanging fluids can be utilized in place of those noted above.

From the foregoing it will be apparent that novel and efficient forms of compressed air cleaners and/or dryers have been disclosed herein. While we have shown and described certain presently preferred embodiments of the invention and have illustrated certain presently preferred methods of practicing the same, it is o be distinctly understood that the invention is not limited thereto but may be variously embodied and practiced within the scope of he following claims.

We claim:

1. Gas-handling apparatus comprising a primary evaporator, first conduit connections for establishing a flow path for a compressed gas and the like through said evaporator, second conduit connections for establishing a flow path of a refrigerant fluid through said primary evaporator in heat exchanging relation with said gas path, a pressure-sensitive expansion valve in said refrigerant path, a refrigerating unit coupled to said second conduit connections, an auxiliary evaporator coupled between said refrigerating unit and said primary evaporator, and pressure-sensitive means coupled to said refrigerant path adjacent its low-pressure side of the evaporator for controlling said expansion valve so that a variation in said compressed gas flow varies the flow of said refrigerant so as to divert condensation thereof from said refrigerating unit to said auxiliary evaporator and/or to divert evaporation thereof from said main evaporator to said auxiliary evaporator.

2. The combination according to claim 1 wherein said auxiliary evaporator is coupled regeneratively to said second conduit connections.

3. The combination according to claim 1 wherein said refrigerating unit includes a condensing unit therefor, some of said second conduit connections couple said condensing unit to one side of said auxiliary evaporator and others of said second conduit connections couple the other side of said auxiliary evaporator to said refrigerating unit and to said primary evaporator.

4. The combination according to claim 1 wherein a heat exchanger for said gas is coupled regeneratively to said first-mentioned conduit connections.

5. The combination according to claim 1 wherein said expansion valve and a third conduit connection are coupled between said primary evaporator and said refrigerating unit in bypassing relation to said auxiliary evaporator.

6. The combination according to claim 5 wherein said expansion valve is provided with means for preventing its complete closure.

7. The combination according to claim 1 wherein said primary evaporator includes a heat exchanger structure having flow means associated with the structure for inducing serpentine heat exchanging paths of said gas and of said refrigerant fluid therethrough.

8. Gas-handling apparatus comprising a primary evaporator, first conduit connections for establishing a flow path for a compressed gas and the like through said evaporator, second conduit connections for establishing a flow path of a refrigerant fluid through said primary evaporator in heat-exchanging relation with said gas path, a refrigerating unit coupled to said second conduit connections, and an auxiliary evaporator coupled between said refrigerating unit and said primary evaporator, said primary evaporator including a heat exchanger structure having flow means associated with the structure for inducing serpentine heat-exchanging paths of said gas and of said refrigerant fluid therethrough, said heat-exchanging means including a plurality of sheath tubes and inner tubes mounted spacedly within said sheath tubes, and flow means for conducting one of said gas and said fluid sequentially through one group of the group of sheath tubes and the group of inner tubes and for conducting the other of said gas and said fluid in parallel through at least some of the remainder of said tubes.

9. The combination according to claim 8 wherein said inner tubes are each provided with a plurality of heat transfer spines extending transversely between each coaxial tube and the associated sheath tube.

10. The combination according to claim 8 wherein said flow-conducting means are arranged to couple said inner tubes in series and said sheath tubes in series-parallel.

11. A method for operating refrigerating gas-handling apparatus having a compressor, condenser, main evaporator, auxiliary evaporator and expansion valve, said valve being mounted for expansion of a refrigerant fluid into said main evaporator, said method comprising the steps of operating said compressor continuously, preventing complete closure of said valve, sensitizing opening and closing movements of said valve to pressure fluctuations in the path of said refrigerant fluid between the refrigerant outlet of said main evaporator and said auxiliary evaporator, and operating said compressor at constant speed for forcing said valve further open in response to load changes.

12. The combination according to claim 1 wherein said expansion valve is coupled in an inlet portion of said refrigerant flow path relative to said primary evaporator, and said pressure-sensitive means are coupled in an outlet portion of said refrigerant flow path relative to said primary evaporator.

13. The combination according to claim 12 wherein said pressure-sensitive means are coupled between said primary evaporator and said auxiliary evaporator.

14. The combination according to claim 1 wherein said primary evaporator includes a heat exchanger structure having flow means therein for inducing a serpentine heat-exchanging path in the flow of at least one of said gas and of said refrigerant therethrough.

15. The combination according to claim 7 wherein said serpentine heat-exchanging paths are coaxial.

16. The combination according to claim 4 wherein auxiliary evaporator and said heat exchanger are each shaped to define coaxial heat-exchanging paths therethrough.