HVAC ELECTRIFICATION
DE-CARBONIZATION AND
ENERGY EFFICIENCY TECHNOLOGY

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Redefining Industrial HVAC Systems for Sustainable Decarbonization Leveraging Atmospheric Air and Facility Based Waste Heat.

DISCLAIMER:

1. Performing HVAC using Turbo-Expander already exists. The following embodiment does not claim that however this embodiment integrates the same to achieve energy conservation.
2. The embodied analysis is validated digitally only, the field testing or piloting is not performed so it does not guarantee materialization of the same on implementation.

Abstract Of The Disclosure

The chemicals processing and heavy industries are facing significant pressure to reduce their carbon emissions. One effective way to achieve this is by eliminating the use of fossil fuels for heating and cooling buildings. This innovative method utilizes ambient air as an operating fluid. The chilled ambient air gains waste heat to produce hot air during winter months for heating purposes, whereas, compressed and subsequently expanded air during summer months for cooling purposes.

The process involves compressing and expanding ambient air through a turbo-expander to recover compression power and produce chilled air for cooling buildings. During winter, atmospheric chilled air is readily available in cold geographical locations and can be used to dissipate heat gained by closed-loop cooling water systems or other industrial processes, producing warm air for heating buildings.

This sustainable approach to industrial HVAC systems leverages facility-based resources to utilize atmospheric air effectively.

Field Of The Invention

As global warming becomes more evident, extreme weather conditions are becoming more common. This means that Heating, Vacuum, and Air Conditioning (HVAC) systems need to work harder to cope with these changes, leading to a larger environmental footprint. To address this issue, it is imperative to implement innovative technologies and processes to develop environmentally sustainable HVAC systems. Optimizing and redefining HVAC systems is crucial in the processing industry, as they are energy-intensive operations.

The invention aims to redefine industrial HVAC systems to benefit the environment. This involves stopping the use of fossil fuels and introducing modern technologies to achieve HVAC in a unique way. These innovative technologies include efficient mechanical systems powered by electricity from green resources. Achieving HVAC uniquely means using readily available resources from existing operations and the surrounding atmosphere.

To lessen the environmental impact of industrial HVAC systems, it is essential to assess, integrate, or reform conventional operations to ensure minimal resource consumption and promote sustainable usage. Energy efficiency, waste heat recovery, and electrification are key tools for making HVAC systems sustainable. This embodiment consists of heating and cooling requirements. The term HVAC is used collectively to describe both aspects.

Background Of The Invention

The invention focuses on electrifying existing processes and recovering waste heat to decarbonize industrial HVAC systems. It involves recovering low-grade thermal energy or waste heat from site-based resources, turbo-expander for compression power recovery, and utilizing the thermal capacity of the expanded air product to provide a cost-effective and energy-efficient HVAC solution.

During summer, the invention uses a Combined Heat and Power (CHP) cycle with atmospheric air as the working fluid. It employs one or two stages of compression and subsequent expansion through a turbo-expander. The compressed air's useful heat is recovered through inter-stage and aftercoolers, whereas the expanded chilled air produced by turboexpander is utilized for building cooling. The turboexpander also recovers partial compression power leading to system energy conservation. This process can be implemented cost-effectively by evaluating existing air compression capacity and installing additional capacity as needed. The idle time (time between on-off cycles) of the existing plant air compression system can be utilized effectively, and additional storage vessels may help optimize the use of installed air compression capacity.

During winter, the invention utilizes rejected heat from existing processes to perform building heating. Waste heat is recovered using a heat exchanger, the ambient air is heated by dissipating waste heat from the cooling water return stream or closed-loop cooling water. The hot air production capacity, temperature, and heat exchanger device specifications are determined based on building heating load estimations. Heat pumps or industrial air compressors will be required further to boost the temperature of the heated ambient air to meet HVAC hot air specifications. Utilizing heat rejected from the cooling water return stream or closed-loop cooling water can eliminate partially or completely the use of cooling towers based on cooling/heating capacity requirements.

Another method of producing hot air involves heating ambient air using waste heat emitted from process heaters or furnaces. An air header/piping laid out surrounding the outer shell of the process heaters in close contact recovers energy for warm air production. The process heaters, vessels, and heat exchangers emit significant energy into the atmosphere and this energy can be recovered to produce the warm air for building heating purposes. Since the HVAC air supply header operates pressurized, the chances of cross contamination are remote.

These integrated processes enhance energy efficiency, waste heat utilization, and the use of under-appreciated resources like ambient cold air. They improve operational economics and significantly reduce the emission footprint of HVAC systems. Conventional processes employ compressors, turbo-expanders, and closed-loop cooling systems but do not integrate them to realize the combined benefits for reducing carbon emissions. Implementing these processes offers system energy conservation, reduced maintenance, increased production capacity, stabilized process operations, decarbonized HVAC systems, and improved ecological footprints.

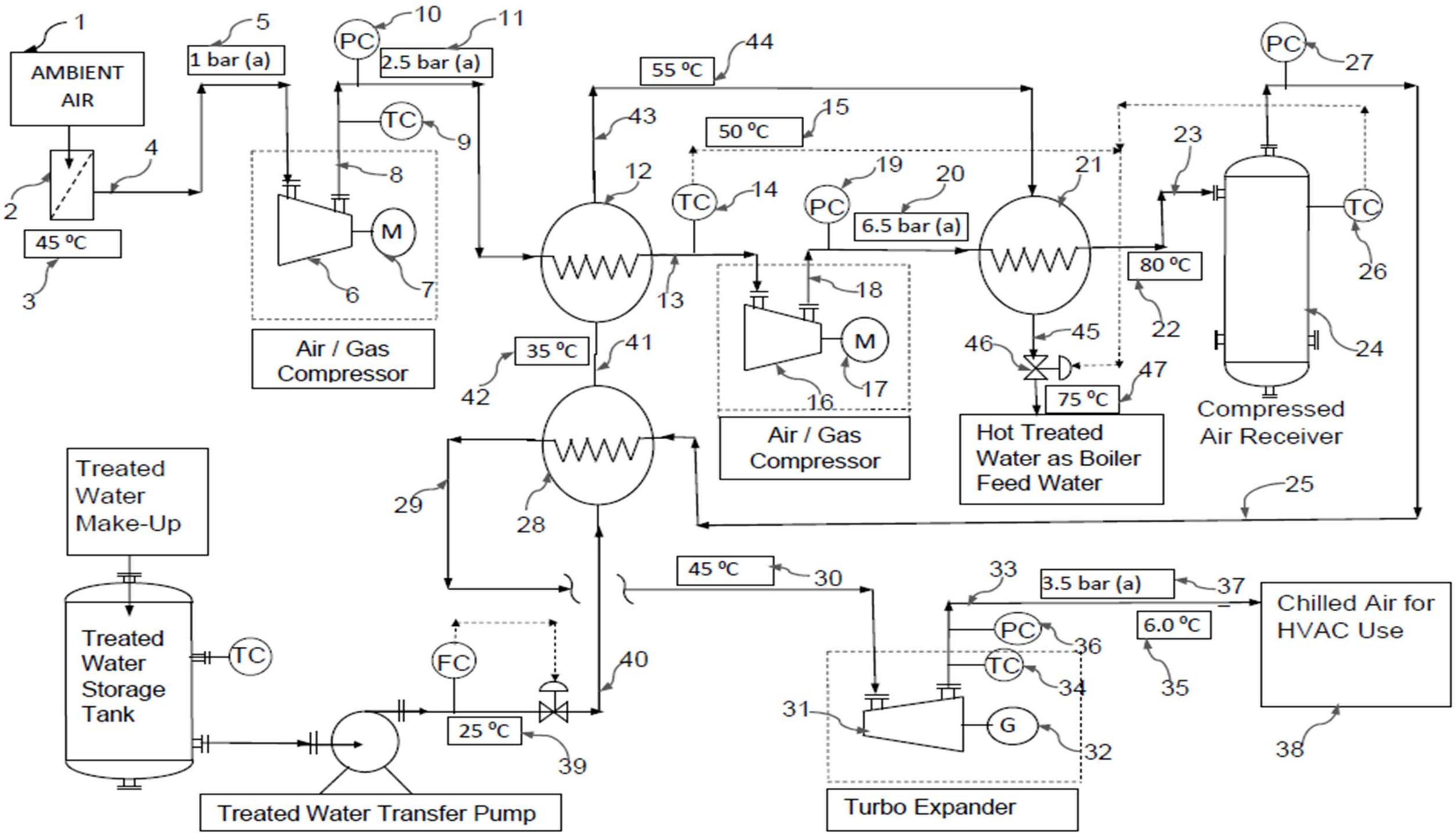

Figure 1 (Process -1) Production of chilled air for HVAC purposes

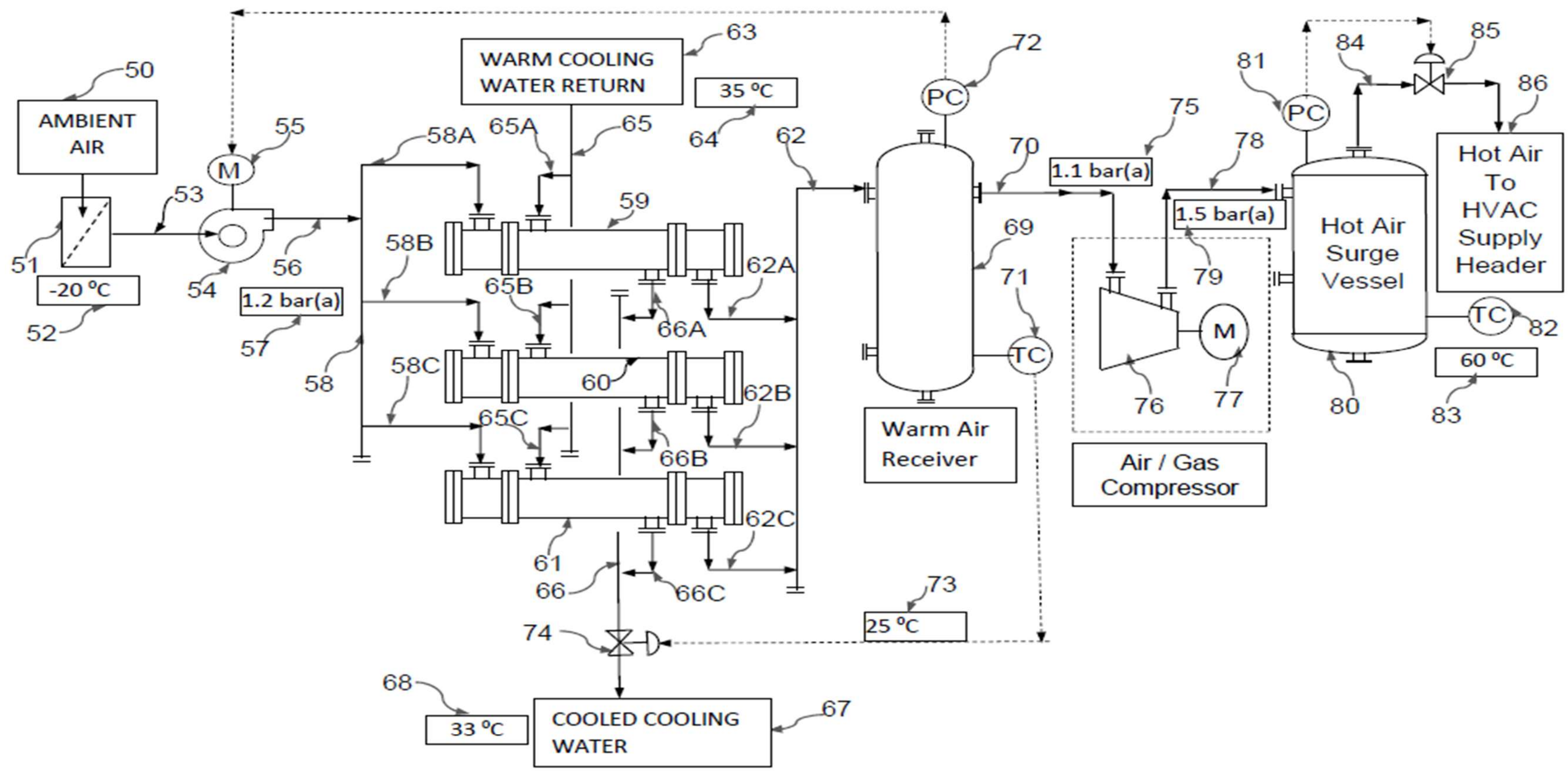

Figure 2 (Process – 2) Hot air production process utilizing low-grade waste heat for HVAC purposes

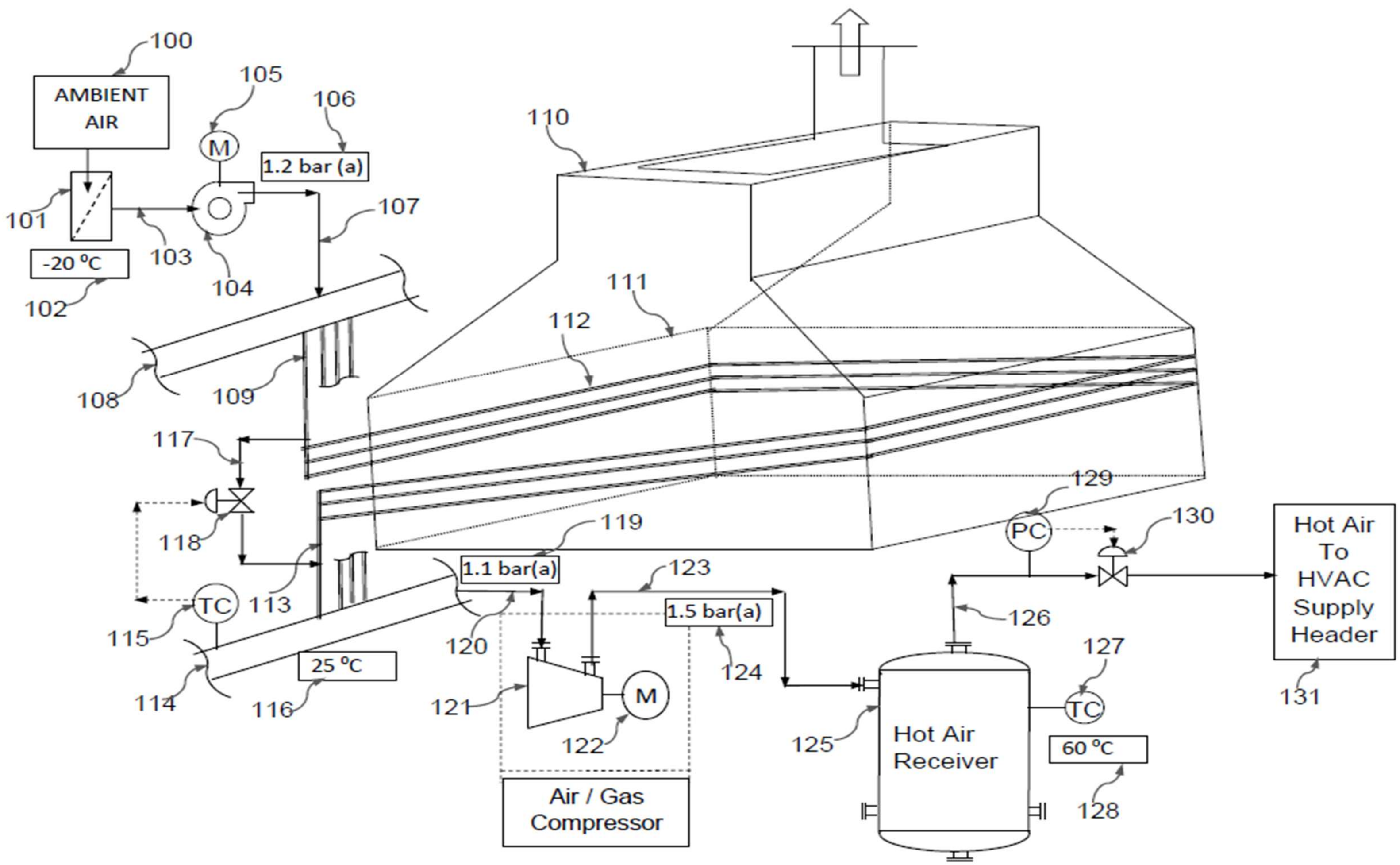

Figure 3 (Process – 3) Hot air production process utilizing heat radiated from process heaters for HVAC purposes

CHILLED AIR PRODUCTION PARAMETERS

Working Fluid	Air Mass Flowrate (kg/h)	1st Stage Compressor Inlet Pressure (kPa)	1st Stage Compressor Inlet Air Temperature (°C)	1st Stage Compressor Discharge Pressure (kPa)	2nd Stage Compressor Discharge Pressure (kPa)	Turbo-Expander Inlet Temperature (°C)	Turbo-Expander Discharge Chilled Air Temperature (°C)	Inter-Aftercooler Water Mass Flowrate (kg/h)	Hot Water Production Temperature (°C)
Air	10,000	101	45	250	650	45	6	12,500	75

POWER CONSUMPTION AND RECOVERY ANALYSIS

Working Fluid	1st Stage Compression Power (kW)	2nd Stage Compression Power (kW)	Total Power Consumption (kW)	Turbo-Expander Power Recovery (kW) (1)	Thermal Heat Recovery at Interstage Coolers (kW) (2)	Total Energy Recovery (kW) (1) +(2)	% Total Power Recovery (kW)
Air	355	387	742	105	743	848	114%

Figure 4 (Table - 1) Summer Season Chilled Air Production and Energy Balance Estimates

WARM AIR PRODUCTION PARAMETERS

Working Fluid	Ambient Air Mass Flowrate (kg/h)	Ambient Air Temperature (ºC)	Air Blower Discharge Pressure / Temperature (kPa)/(ºC)	Warm Cooling Water Return Temperature (ºC)	Warm Air Temperature at outlet Heat Exchangers (ºC)	Air Compressor Discharge Pressure (kPa)	Air Compressor Discharge Air Temperature (ºC)
Air	10,000	-20	120 / -2	35	25	150	60

POWER CONSUMPTION AND RECOVERY ANALYSIS

Working Fluid	Air Blower Power Consumption (kW)	Heat Recovery from Cooling Water Return (kW)	Air Compressor Power Consumption (kW)	Total Power Consumption (kW)	Net Power Consumption (kW)	% Total Power Recovery (kW)
Air	50	75	100	150	75	50%

Figure 5 (Table - 2) Winter Season Hot Air Production and Energy Balance Estimates Recovering Low-Grade Heat

WARM AIR PRODUCTION PARAMETERS

Working Fluid	Ambient Air Mass Flowrate (kg/h)	Ambient Air Temperature (ºC)	Air Blower Discharge Pressure / Temperature (kPa)/(ºC)	Approximate Furnace Outer Shell Temperature (ºC)	Warm Air Temperature outlet of Tubing/Coil (ºC)	Air Compressor Discharge Pressure (kPa)	Air Compressor Discharge Temperature (ºC)
Air	10,000	-20	120 / -2	85	25	150	60

POWER CONSUMPTION AND RECOVERY ANALYSIS

Working Fluid	Air Blower Power Consumption (kW)	Heat Recovery from Process Heaters (kW)	Air Compressor Power Consumption (kW)	Total Power Consumption (kW)	Net Power Consumption (kW)	% Total Power Recovery (kW)
Air	50	75	100	150	75	50%

Figure 6 (Table - 3) Winter Season Warm Air Production and Energy Balance Estimates Recovering Emitted Heat

Brief Description Of The Drawings

Preferred embodiments of the invention will now be described in conjunction with the accompanying drawings in which:

Figure 1 is a process flow diagram of chilled air production process for HVAC use according to a first embodiment of the invention that incorporates two stages of Air Compression / Heat Pump with Inter and Aftercoolers, which enable compression heat recovery by circulating treated water. Additionally, Turbo Expander applications are used for compression power recovery and producing expanded chilled air.

Figure 2 is a process flow diagram of hot air production process for HVAC use according to a second embodiment of the present invention. This process dissipates low-grade waste heat from the Process Cooling Water Return stream to heat ambient cold air, to produce warm air. The warm air is then further compressed to increase its temperature, and pressure, compensating for transport hydraulics losses and meeting HVAC hot air supply specifications for pressure and temperature.

Figure 3 is a process flow diagram illustrating a variant of the second embodiment for producing hot air for the HVAC use according to a third embodiment of the present invention that comprised of tubing/piping/ducting laid out in close contact around the outer perimeter of the heat-emitting sources such as process heaters, furnaces, vessel or heat exchanger. The warm air is then compressed further to boost its temperature and pressure, compensating for transport hydraulics losses and meeting the pressure and temperature specifications for the HVAC hot air supply.

Figure 4 (Table – 1) presents the estimation of compression heat recovery, energy consumption and recovery parameters for the process depicted in Figure 1, which includes air compression and compressed air expansion using a turbo-expander.

Figure 5 (Table – 2) presents the estimation of low-grade waste heat recovery and energy consumption parameters for the process depicted in Figure 2. This includes ambient air heating with a process stream containing waste heat, followed by compression.

Figure 6 (Table – 3) demonstrates the estimation of waste heat recovery and energy consumption parameters for the process depicted in Figure 3. This process consists of ambient air heating utilizing the waste heat emitted by the process equipment, followed by compression.

Detailed Description Of The Invention

This embodiment of the invention relates to unique mechanical arrangements formed by integration of existing process operations for achieving objectives of electrification and enhanced energy efficiency. Consequently, this leads to a reduced environmental footprint of the existing operations, improved process performance, and cost savings. This invention employs the fundamental principles of conservation of energy and accomplishes the objectives of enhanced energy efficiency and reduced environmental emission footprint. It integrates available thermal resources for waste heat recovery and utilizes them to fullest potential such that the system's total thermal energy is conserved. While current industrial practices end up discarding low-grade thermal resources as they are not cost-effective to recover. However, this invention makes it possible to recover and use this low-grade heat efficiently and reliably.

First Embodiment (Process - 1): Production of chilled air for HVAC purpose using ambient air as an operating fluid

Referring to Fig. 1, in the first embodiment (Process - 1) shown therein, Ambient Air 1 undergoes a cycle of two stage compression and expansion for transforming to chilled air.

Ambient air 1 at an operating conditions of pressure 1 bar(a) 5 and temperature 45 ^{0}C 3 is filtered through Air Filter 2. The filtered air is provided as a suction to the first stage Air Compressor 6 through air duct 4. The suction air temperature 45 ^{0}C represents the extreme summer season conditions. The first stage air compressor 6 is driven by the electric motor 7. The first stage compressor 6 takes suction through line 4 and discharges the compressed air through discharge line 8 at pressure 2.5 bar(a) 11 and temperature 172 ^{0}C 9. The first stage compressor discharge is cooled to 50 ^{0}C temperature 15 through the Heat Exchanger (intercooler) 12 before being fed to the second stage compressor 16 via suction line 13.

The second stage air compressor 16 is driven by electric motor 17. The second stage compressor 16 takes suction through line 13 and discharges the compressed air through discharge line 18 at pressure 19 of 6.5 bar (a) 20 and temperature 188 ^{0}C. The second stage compressor 16 discharge is cooled to 80 ^{0}C temperature 22 through the aftercooler 21 before being fed to the Compressed Air Receiver 24 via line 23. The second stage compressor 16 suction temperature and the air receiver 24 operating temperatures are controlled by the temperature controllers 14 and 26 which provides a control signal to the final control element 46 to regulate the Treated Water flowrate through the Heat Exchanger 12 and 21. A temperature controller 14 can be integrated with a flow controller located at the discharge of the Treated Water Transfer Pump, to regulate circulation of treated water through the inter-cooler 12 and the aftercooler 21 and 28. The temperature and pressure of the compressed air stored within the Compressed Air Receiver are recorded by the temperature controller 26 and pressure controller 27. Air Receiver 24 acts as a knock-out drum eliminates pressure waves.

The compressed air stored within the Air Receiver 24 has significant thermal energy which can be recovered effectively either by deployment of the third heat exchanger 28 or by feeding directly to the Turbo-Expander 31. When fed directly to the Turbo-Expander, the chilled air produced will have significantly higher temperature 35 specifications at 3.5 bar (a) 37 pressure.

An additional aftercooler 28 is included within the embodiment which provides an optimized process design. It helps in achieving all the product specifications such as hot water production at 75 ^{0}C 47 and Chilled Air production at 6 ^{0}C 35 at a product pressure 3.5 bar (a) 37. The aftercooler 28 permits efficient use of heat transfer area within all the heat exchangers 12 and 21 in loop and allows integration for maximizing heat transfer efficiency for achieving all products specifications. However, the installation of the aftercooler 28 is contingent, it is function of process design criteria, demand and supply capacity parameters, and product specifications. Analysis indicates that it would be a trade-off between product capacity and specifications. However, within this context, the analysis has demonstrated that including aftercooler 28 would result in optimal product capacities, meeting product specifications, and efficient utilization of heat transfer surface area for both the intercooler 12 and aftercooler 21. Additionally, it helps in reducing the Turbo-Expander inlet temperature, thereby allowing for the optimization of compression ratios at each compressor stage, which ultimately leads to decreased power consumption.

The Compressed Air Receiver 24 acts as a suction surge drum for the Turbo Expander 31. The Turbo-Expander is a critical component of the embodied process which converts the high temperature air to chilled air and generates power as compressed air expands through it. Deployment of the turbo-expander helps in recovering the extent of energy lost during the air compression stages. The compressed air is fed to the Turbo-Expander via suction line 29 at a temperature 45 ^{0}C 30; undergoes isentropic expansion through the Turbo-Expander 31 which produces work to drive power generator 32. The expander 31 shaft is close coupled with the power generator motor 32 such that when the expander shaft rotates due to the conversion of the compressed air potential energy into the kinetic energy, it drives the coupled power generator motor 32.

The Compressed Air pressure let-down from 6.5 bar (a) 20 to 3.5 bar (a) 37 across the Turbo Expander 31. It has further potential to generate chilled air at a much lower temperature than targeted 6 ^{0}C 35 and by producing chilled air at lower pressure than targeted 3.5 bar (a) 37 within the embodied process. The only adverse consequences of producing the chilled air at a temperature lower than 4 ^{0}C is moisture solidifying which leads to building up ice formation. However, this problem can be overcome by designing spare capacity and by designing periodic de-frost cycle. Blending of chilled air with recycled or fresh air during the defrost cycle can be an on-line solution to eliminate the condensation issue and maintain continuity of the process of producing chilled air at lower temperatures.

The pressure losses across the Intercooler 12 and Aftercoolers 21, 28 may need to be accounted based on the capacity. The expanded chilled air exits the Turbo-Expander 31 at reduced pressure of 3.5 bar (a) 37 and at a temperature of 6.0 ^{0}C 35 via a discharge line 33. The temperature and pressure of the chilled air from the Turbo-Expander 31 is recorded by the temperature controller 34 and pressure controller 36, which provides a control signal to the HVAC control system for the manipulation of the final control element based on the temperature and pressure at the turbo-expander discharge. The exhaust from the Turbo Expander is provided to HVAC distribution 38 via line 33.

The closed loop or once-through cooling water system is designed to achieve system energy conservation, extracts the compressed air heat energy after each stage of the air compression to produce the Hot Process Water (HPW) or Boiler Feed Water (BFW) at a temperature specification of 75 - 80 ^{0}C 47 or greater. A treated water storage tank receives soft/filtered water as make-up water. Treated Water Transfer Pump takes suction from the Treated Water Storage Tank and circulates treated water through the intercooler 12 and aftercoolers 21, 28. A flow controller with a final control element is provided at the Treated Water Transfer Pump discharge to regulate the flow.

Aftercooler 28 is a heat exchanger which receives Treated Water as a cold stream at a temperature of 25 ^{0}C 39 through line 40 and exits the heat exchanger gaining heat at a temperature of 35 ^{0}C 42 through line 41. The compressed air enters the Aftercooler 28 as a hot stream at a temperature of 80 ^{0}C 22 through line 25 and exits dissipating heat at a temperature of 45 ^{0}C 30 through line 29.

Intercooler 12 is a heat exchanger which receives the Treated Warm Water from aftercooler 28 through line 41 as cold inlet stream and exits gaining heat at temperature of 55 ^{0}C 44 through line 43. The intercooler 12 receives the first stage compressor 6 discharge as a hot inlet stream through line 8 and exits at a temperature of 50 ^{0}C 15, losing heat to the Treated Water through outlet line 13.

Aftercooler 21 is a heat exchanger which receives the Treated Warm Water, discharged from heat exchanger 12, as inlet cold stream and exits at a temperature of 75 – 80 ^{0}C 47 gaining heat through line 45. The aftercooler 21 receives the second stage compressor 16 discharge as a hot inlet stream through line 18 and exits at a temperature of 80 ^{0}C 22, losing heat to the Treated Hot Water through outlet line 23.

The Treated Hot Water Stream exiting the aftercooler 21 is heated to a temperature specification of circa 75 - 80 ^{0}C 47 which is supplied as Hot Process Water to the process / real-estate infrastructure or as a Boiler Feed Water to the Boiler for steam production. In the real estate environment, the thermal energy recovered from the process would suffice the hot water supply requirements.

Fig. 4 (Table - 1) illustrates the technical analysis performed for evaluating the parameters cited within the first embodiment (Process -1). Ambient air at flowrate of 10,000 kg/h and 45 ^{0}C 3 is compressed from 1 bar(a) 5 to 2.5 bar(a) 11 in 1st stage and from 2.5 bar(a) 11 to 6.5 bar (a) 20 in 2nd stage of compression. The intercooler 12 and aftercoolers 21, 28 extract the heat of compression and delivers two product streams at the discharge of the aftercooler: the cooled compressed air at 45 ^{0}C 30 and hot water at 75 ^{0}C 47. Turbo-Expander expands the compressed air from 6.5 bar (a) 20 and 45 ^{0}C 30 to 3.5 bar (a) 37 and 6 ^{0}C 35 isentropically while delivering work to generate power.

The 1st stage and 2nd stage of compressor consume 355 kW and 387 kW respectively. The power recovered from the turbo-expander is approximately 105 kW and power recovery in the form of thermal energy from the inter and after coolers combined is around 743 kW. Thus, the combined total power recovered is estimated at 114% of the total power consumed.

This embodied process allows use of the atmospheric air as an operating fluid and thereby eliminating use of refrigerant chemicals which are considered hazardous and toxic to the ecosystem.

Since demand for chilled air in the HVAC system peaks during the summer months and be minimal during the winter months, the air compressor should be selected and sized to ensure maximum capacity delivered at elevated ambient air temperatures and humidity levels. This approach will prevent any compromise in the HVAC chilled air supply capacity. During the winter months, the surplus chilled air capacity can be leveraged to treat or cool the warm cooling water return stream from the existing process or the air compressors can be used for process 2 or 3 (fig. 2 or 3) for production of hot air.

The outputs shown in Fig. 4 (Table -1) are on dry air basis. For moisturized or saturated air, a higher compression ratio may be necessary to achieve the specified chilled air temperature and Hot Process Water or Boiler Feed Water temperature. Variations in humidity and suction temperatures are key variables that affect the volumetric delivery of air compressors. Therefore, selection and sizing of the air compressor should be carried out to ensure it meets design requirements and delivers optimal performance.

TURBO-EXPANDER

When an aftercooler 28 is excluded from the process design, compressed air at 80°C will be fed directly to the Turbo-Expander. Under these conditions, a higher let-down ratio is required across the Turbo-Expander to achieve the chilled air temperature specifications of 6 ^{0}C. A higher pressure let-down ratio across the Turbo-Expander results in increased power generation and reduced product stream pressure at discharge. Given that the chilled air produced by the Turbo-Expander is utilized directly for HVAC distribution, the product stream pressure must be at sufficient pressure to offset the hydraulic, fitting, and control losses incurred during transportation to end users. Consequently, the inclusion of aftercooler 28 involves a trade-off: either feed the Turbo-Expander at a lower temperature of 45 ^{0}C to produce chilled air at 6 ^{0}C with a higher product pressure of 3.5 bar (a) or feed it at a higher temperature of 80 ^{0}C to produce chilled air at 6 ^{0}C with a lower product pressure of 2 bar (a).

Under identical pressure reduction ratios across the Turbo-Expander, meaning the product stream pressure remains constant, a feed stream at 40 ^{0}C results in chilled air at significantly lower temperatures and marginally reduced power output. In contrast, a feed stream at 80 ^{0}C produces chilled air at significantly higher temperatures and slightly increased power output.

Second Embodiment (Process - 2): Production of hot air for HVAC purpose using ambient air as an operating fluid and by extracting waste heat from cooling water return stream

Referring to Fig. 2, in the second embodiment (Process - 2) shown therein, Ambient chilled air 50 gains waste heat dissipating from the cooling water return stream 63 through heat exchanger 59/60/61, followed by Air Compression 76 to transform into hot air 86.

Ambient air 50 at a pressure of 1 bar(a) and -20 ^{0}C 52 is filtered through Air Filter 51 before provided as suction to Air Blower 54 through air duct 53. The ambient air temperature represents the northern hemisphere winter season extreme conditions. The Air Blower 54 is driven by an electric motor 55. The Air Blower 54 takes suction through line 53 and discharges the compressed air through discharge line 56 at pressure 1.2 bar(a) 57 and temperature -2 ^{0}C. The air temperature rises from -20 ^{0}C to -2 ^{0}C due to gain in compression thermal energy as it undergoes compression through the Air Blower. The Air Blower provides the driving force required to compensate for hydraulic losses across the heat exchangers 59/60/61 and to provide positive suction head requirements for the Air Compressor 76. The pressurized air from the Air Blower 54 discharge is fed to the common header 58 which further distributes the air through line 58 A/B/C to Heat Exchangers 59/60/61 respectively. The Heat Exchanger 59/60/61 are shell and tube type or plate and frame type, which facilitates heat exchange between warm cooling water stream returning from the process and the air from Air Blower. The warm air exits from Heat Exchangers 59/60/61; is collected within the warm air header main 62 via line 62 A/B/C.

The warm air header main 62 collects and transports the warm air to Warm Air Receiver vessel 69. The temperature controller 71 is provided at Warm Air Receiver 69 to control the warm air temperature at 25 ^{0}C 73. The temperature controller 71 generates a control signal to manipulate a final control element 74. The final control element 74 regulates the warm cooling water flow rate through the Heat Exchangers 59/60/61. The pressure controller 72 is provided at Warm Air Receiver 69 generates a control signal which regulates the speed of the Air Blower Motor 55 to maintain the constant pressure of 1.1 bar (a) 75 at Warm Air Receiver 69.

To meet the HVAC air temperature 82 specifications of 60 ^{0}C 83, the warm air stored within the Warm Air Receiver 69 would need to be further heated. Therefore, a Heat Pump/Air Compressor 76 is provided to boost the temperature 73 and pressure 75. The produced warm air is further heated to elevate the stream temperature up to 60 ^{0}C across the air compressor 76 which also boosts the pressure of the warm air stream for making up the hydraulic losses incurred due to transportation and HVAC distribution. The Air Compressor 76 is driven by electric motor 77, which takes suction through line 70 at pressure 1.1 bar(a) 75 and discharges the compressed air through line 78 at pressure 1.5 bar(a) 79 and temperature 82 of 60 ^{0}C 83.

The Hot Air Surge Vessel 80 is provided at the downstream of Air Compressor 76 acts as a surge vessel which provides buffer capacity to control load/unload function of the Air Compressor 76. Pressure controller 81 is provided to record and control the pressure 79 by manipulating the final control element 85. A temperature controller 82 records the temperature of the hot air stored within Hot Air Surge Vessel 80 for the

HVAC distribution. The hot air leaves the Hot Air Surge Vessel 80 through outlet line 84 for the further HVAC distribution 86. HVAC header pressure and distribution shall be controlled by integrated HVAC logic control system which receives the control commands from the local thermostats and input signal from the pressure controller 81 and temperature controller 82 provided at the Hot Air Surge Vessel 80. The HVAC logic control may generate control command to manipulate the final control element 85 to maintain the HVAC distribution header 86 pressure. The hot air leaves the Hot Air Surge Vessel 80 at a temperature specification of 60 ^{0}C 83 and pressure of 1.5 bar (a) 79.

The Warm Cooling Water Return 63 from the existing process is transported within cooling water return header main 65 at an operating temperature 64 of 35 ^{0}C, which is branched off to supply warm cooling water to Heat Exchangers, 59/60/61 via line 65 A/B/C respectively in parallel. The cooling water discharge from Heat exchangers 59/60/61 is collected within Cooled Cooling Water Header main 66 via lines 66 A/B/C. A final control element 74 is provided at line 66 to regulate the Cooled Cooling Water 67 discharge flow to achieve controlled heat transfer within the heat exchangers. Cooled Cooling Water 67 is returned to the cooling water storage at 33 ^{0}C 68 temperature or lower.

Fig. 5 (Table - 2) illustrates the technical analysis performed for evaluating the parameters cited within the Second Embodiment (Process - 2). Ambient air 50 at 1 bar(a) and -20 ^{0}C 52 with flowrate of 10,000 kg/h is provided to Air blower 54 as suction. The ambient air temperature rises from -20 ^{0}C 52 to -2 ^{0}C across the Air Blower as pressure increases to 1.2 bar(a) 57, and subsequently up to 25 ^{0}C across the heat exchanger 59/60/61. The warm air is further compressed through air compressor 76 for boosting up the air temperature up to 60 ^{0}C 83 and pressure 1.5 bar (a) 79.

Air Blower 54 consumes 50 kW and Air compressor 76 consumes 100 kW whereas the energy recovered from the cooling water return stream in terms of thermal duty is 75 kW. Thus, the total power recovery is 50% of the total power consumed. The direct benefits in the term of power saving are estimated at 50%, However, there are additional indirect benefits associated with the embodied process. Warm cooling water returns from the processing facility, after losing heat to cold air becomes cold water which can be provided straight back to the facility, resulting in eliminated use of Cooling Tower and losses associate with that.

Third Embodiment (Process - 3): Production of hot air for HVAC purpose using ambient air as an operating fluid and by recovering waste heat radiated from the industrial equipment

Referring to Fig. 3, in the third embodiment (Process - 3) illustrated therein, the ambient chilled air 100 is circulated through the conduit 112 laid out at an outer periphery of the heat emitting sources such as process heaters / furnaces 110 for extracting heat radiated, followed by air compression 121 for transforming to hot air 131.

Ambient air 100 at a pressure of 1 bar(a) and temperature of -20 ^{0}C 102 is filtered through Air Filter 101 before fed as an inlet to Air Blower 104 through air duct 103. The Air Blower 104 is driven by an electric motor 105. It takes suction through line 103 and discharges the compressed air through discharge line 107 at a pressure 1.2 bar(a) 106 and temperature of -2 ^{0}C. The Air Blower 104 discharge is fed to a common air header 108 which further distributes the cold air to the tubing or piping or a duct 112 (conduit) through sub header 109 for recovering heat radiated from the process equipment, which otherwise lost to the atmosphere.

The conduit 112 is laid out at the outer perimeter of the process heater or furnace 110 within the radiation section 111. It is arranged to establish metal-to-metal contact, which is crucial for heat recovery through conduction and convection as well radiation. Warm air at the discharge of conduent is collected within the warm air collection header 114 through sub-header 113. A temperature control element 115 is provided at the warm air collection header 114 for recording and maintaining header temperature 115 at 25

^{0}C 116. The temperature controller 115 generates a control signal to manipulate the final control element 118 for regulating flow. The control valve 118 functions as a bypass valve which diverts the air flow directly from inlet header 109 to outlet header 113, bypassing the conduit 112 for controlling the warm air collection header 114 temperatures 115 at 25 ^{0}C 116 and pressure 1.1 bar (a).

The warm air collected within the Warm Air Header 114 would need to be heated further to meet the HVAC supply air temperature 127 specifications. Therefore, a Heat Pump or Air Compressor 121 is provided for boosting up the temperature 116 and pressure 119. The Air Compressor 121 is driven by electric motor 122 which takes suction through line 120 at pressure around 1.1 bar(a) 119 and discharges the compressed air through discharge line 123 at pressure 1.5 bar(a) 124 and temperature 127 at 60 ^{0}C 128.

Hot Air Receiver 125 is provided at the downstream of Air Compressor 121 acts as a surge vessel which provides buffer capacity to control load/unload function of the Air Compressor 121. Pressure controller 129 is provided to record the operating pressure 124 and control the hot air supply header 131 pressure by manipulating the final control element 130. The hot air leaves the Hot Air Receiver 125 through outlet line 126 at a temperature specification of 60 ^{0}C 128 and pressure of 1.5 bar (a) 124 for the further HVAC distribution 131. HVAC header pressure and distribution shall be controlled by integrated HVAC logic control system which receives control signals from the local thermostats and input signal from the pressure controller 129 and temperature controller 127 provided at the Hot Air Surge Vessel 125. The HVAC logic control may generate control command to manipulate the final control element 130 to maintain the HVAC distribution header 131 pressure.

Fig. 6 (Table - 3) illustrates the technical analysis performed for evaluating the parameters cited within the Third Embodiment (Process - 3). Ambient air 100 at 1 bar(a) and -20 ^{0}C 102 with flowrate of 10,000 kg/h is provided to Air blower 104 as suction which circulates air through the conduit laid out peripheral to the process heater or equipment. The ambient air temperature rises from -20 ^{0}C 52 to -2 ^{0}C across the Air Blower 104 as pressure increases to 1.2 bar(a) 57, and subsequently up to 25 ^{0}C 116 across the process heater 111. The warm air is further compressed through air compressor 121 for boosting up the air temperature up to 60 ^{0}C 128 and pressure at 1.5 bar (a) 124.

The Air Blower consumes 50 kW, and the Air Compressor consumes 100 kW whereas the power recovered from the process heater/furnace in terms of thermal duty is 75 kW. The system's recovery of the total power is 50% of the total power consumed.

Heat and material balance calculations, specifically heat transfer balance calculations, consider only heat transfer via conduction. These calculations do not account for heat gain or transfer through radiation. The estimation process takes into consideration the temperature at the outer shell of the equipment and the heat transfer to the surface of the conduit solely by conduction. The number of tube or coil passes surrounding the heater is determined by the radiating surface temperature, perimeter area, and the desired hot air temperature based on the air flow rate.