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(54) METHOD, SYSTEM AND APPARATUS FOR COOLING HIGH POWER DENSITY DEVICES

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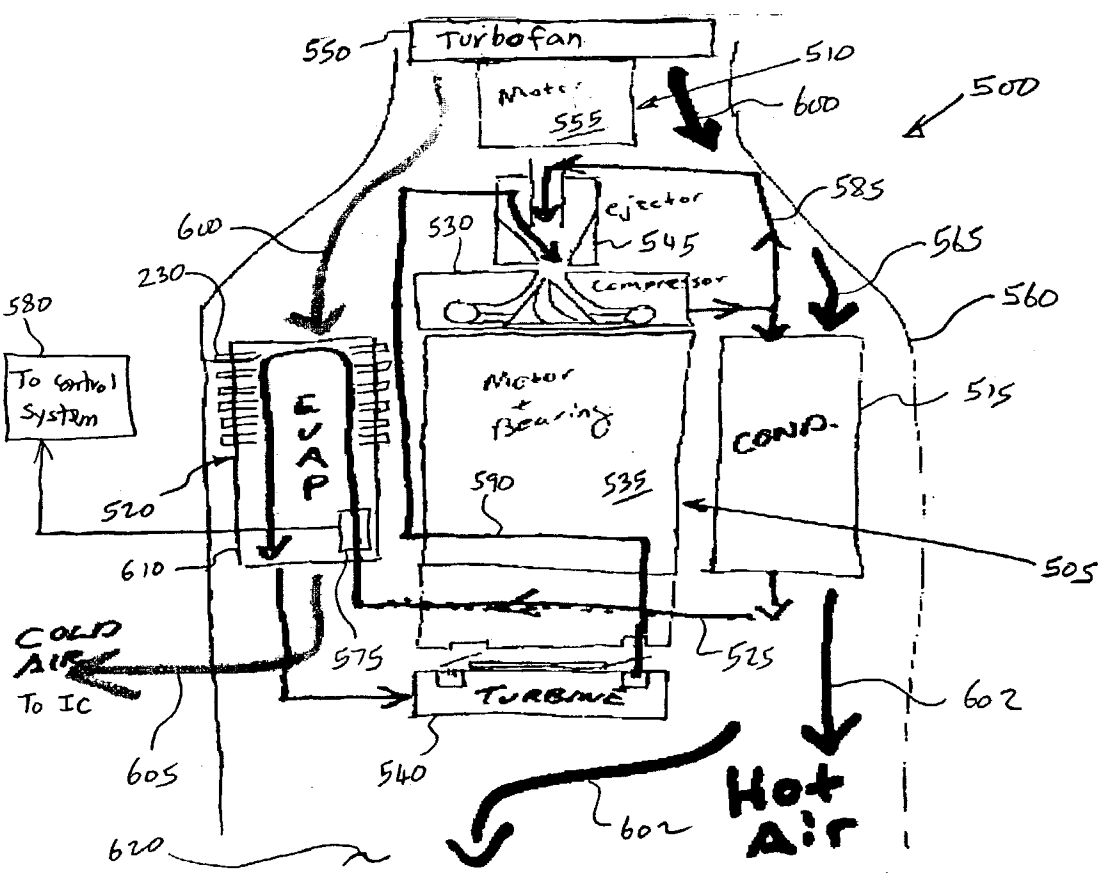
(57) ABSTRACT

A turbomachinery system for cooling a high power density device includes a turbomachine configured to deliver a high flux cooling medium and a high power density device arranged in fluid communication with the turbomachine, the turbomachine having a motor and a compressor driven by the motor.



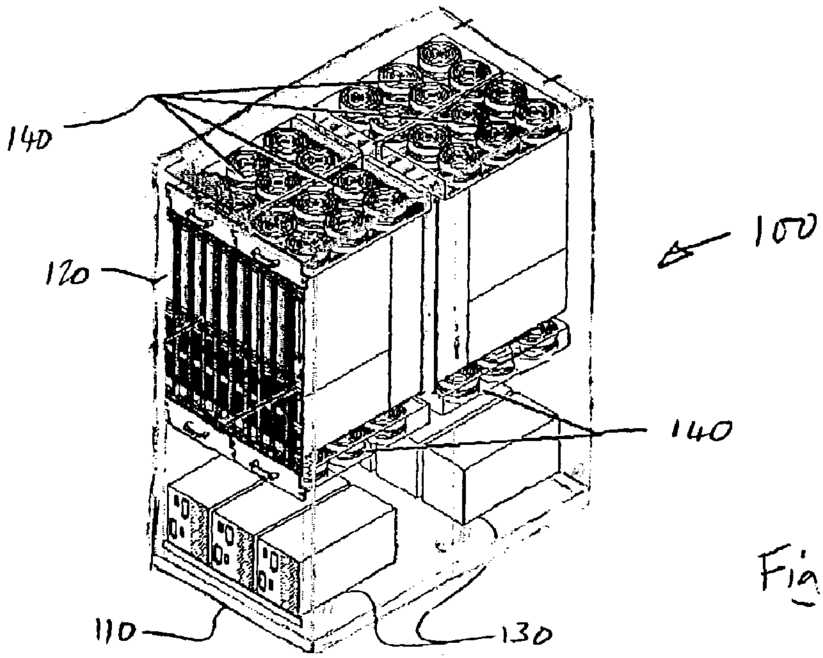


Fig. 1

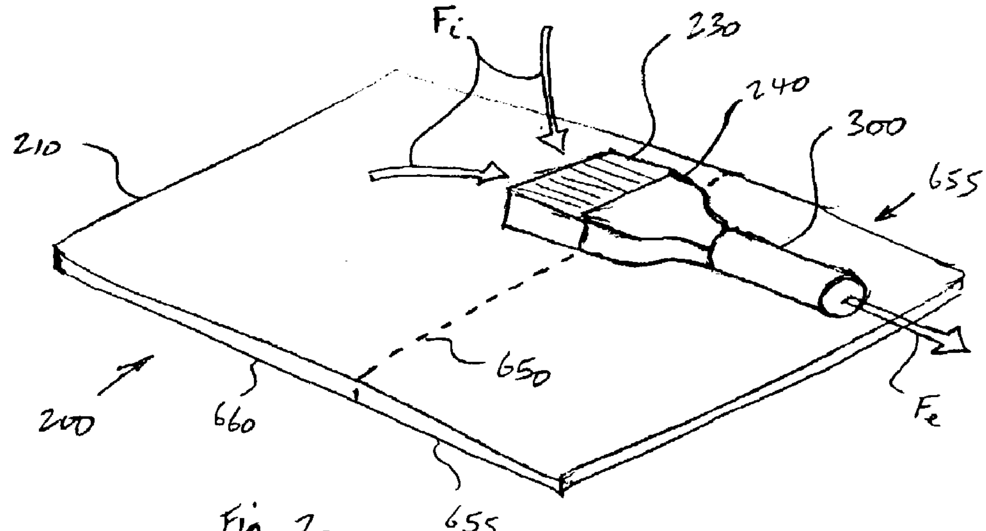


Fig. 2

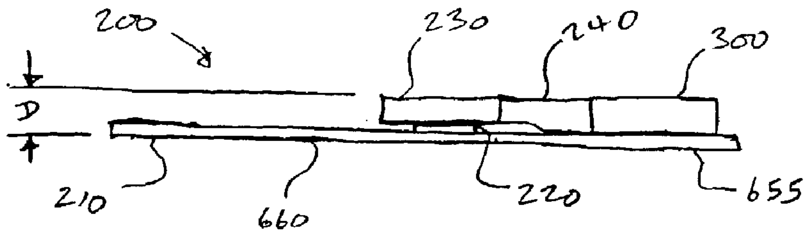


Fig. 3

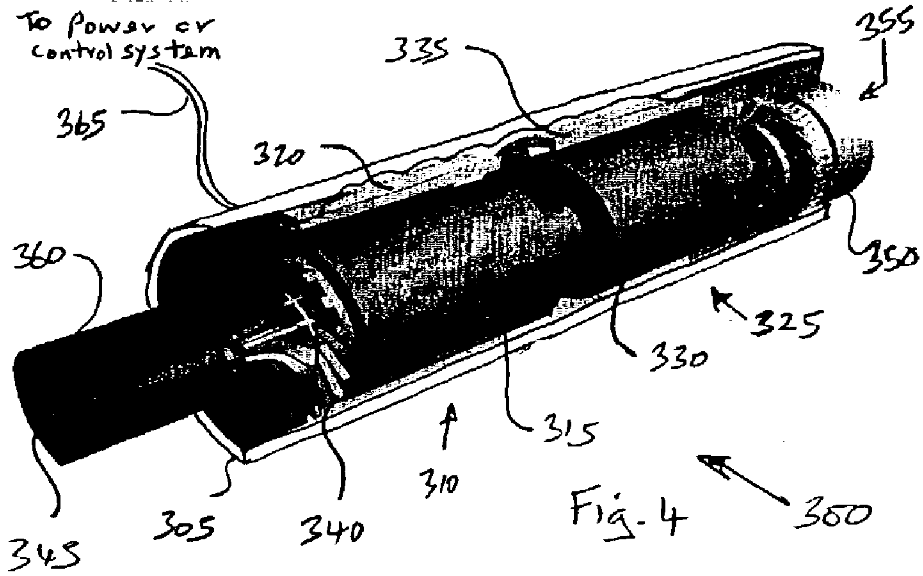


Fig. 4

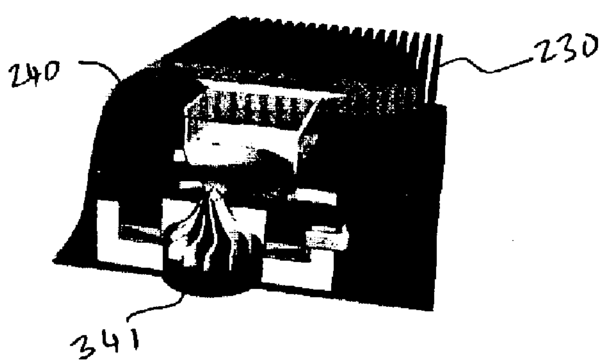


Fig. 5

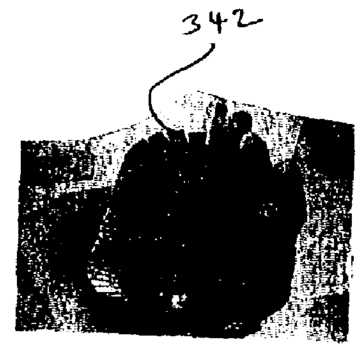


Fig. 6

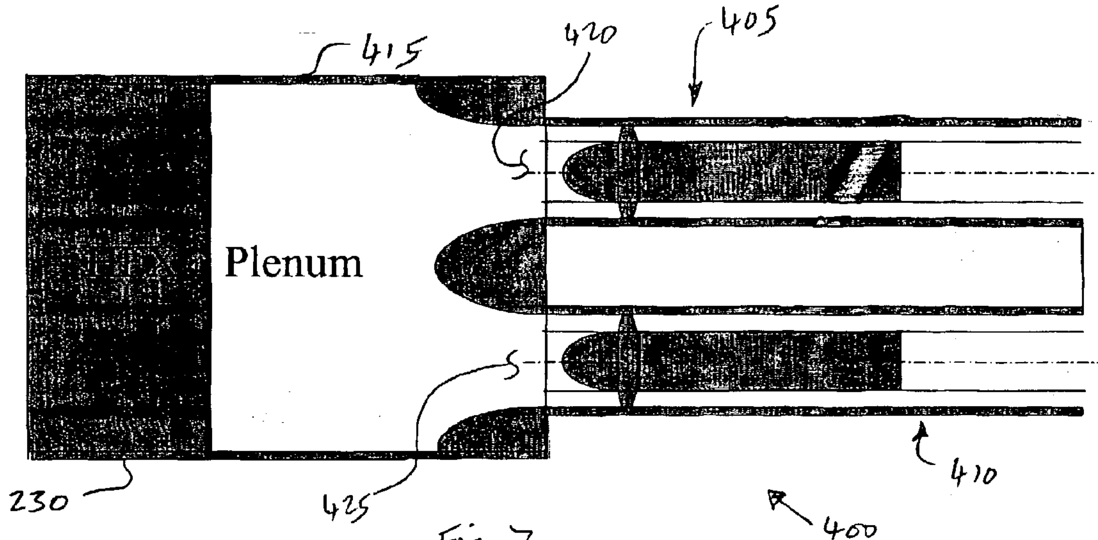

415
420
405
Plenum
410
230
425
400

Fig. 7

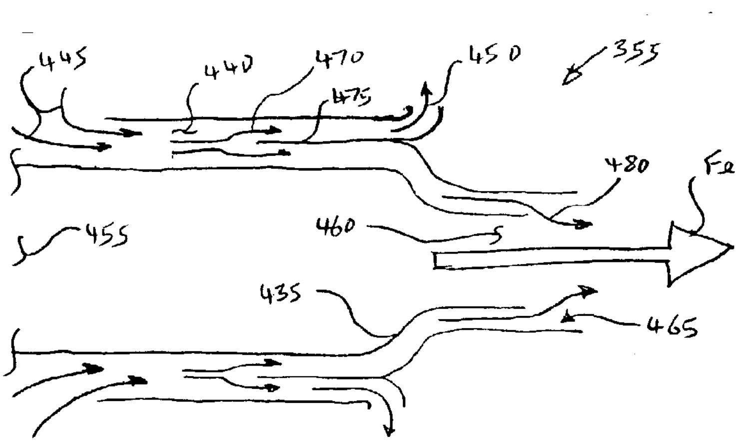

445
440
470
475
450
355
480
Fe
455
460
435
465

Fig. 8

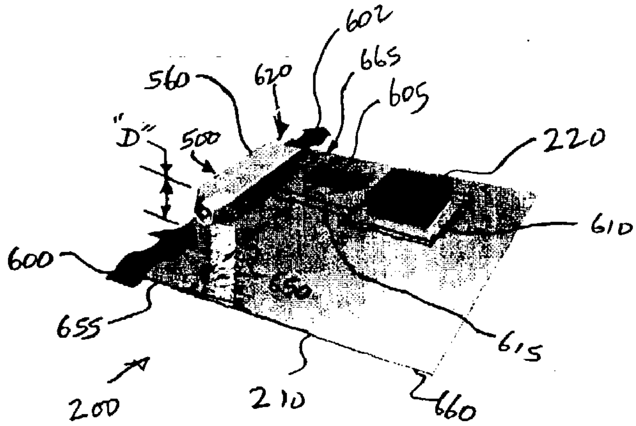

602
560
620
665
605
"D"
500
220
610
600
650
655
615
200
210
660

Fig. 9

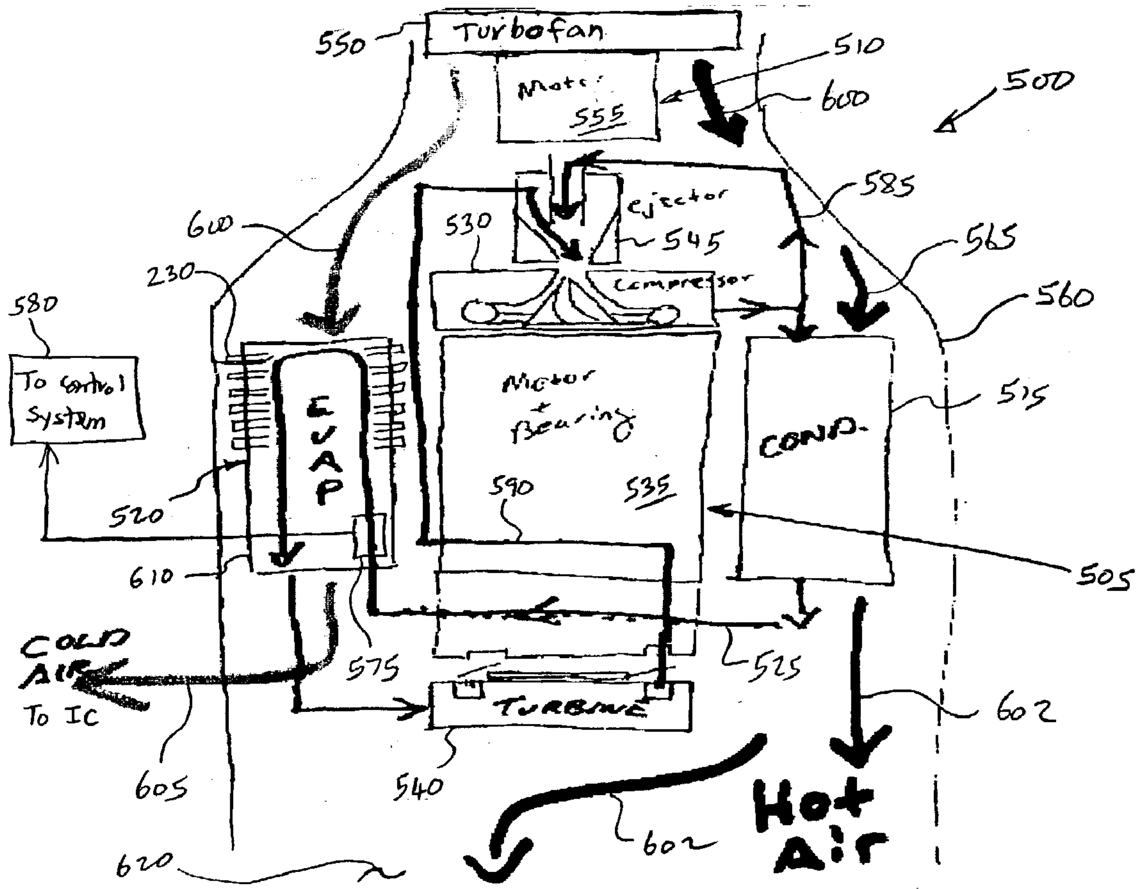

Turbofan
550
510
Motor
555
600
500
ejector
585
600
530
545
565
Compressor
230
560
580
To control system
EVAP
Motor + Bearing
COND.
515
520
590
535
505
610
575
525
COLD AIR
To IC
TURBINE
602
605
540
602
Hot Air
620

Fig. 10

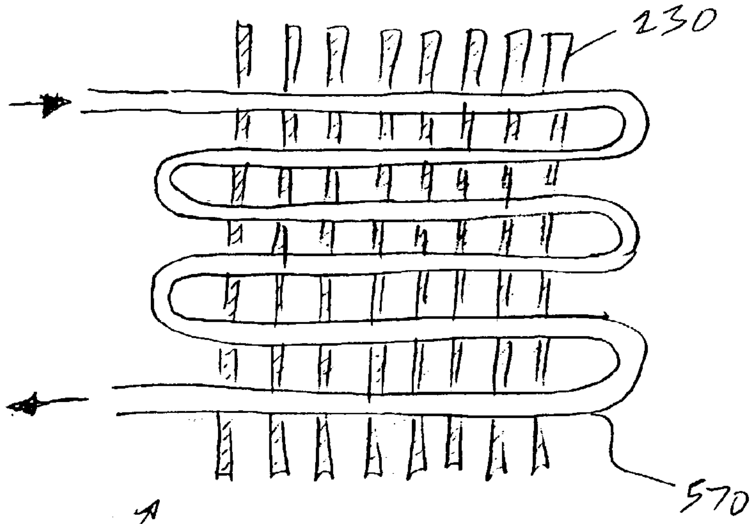


Fig. 11

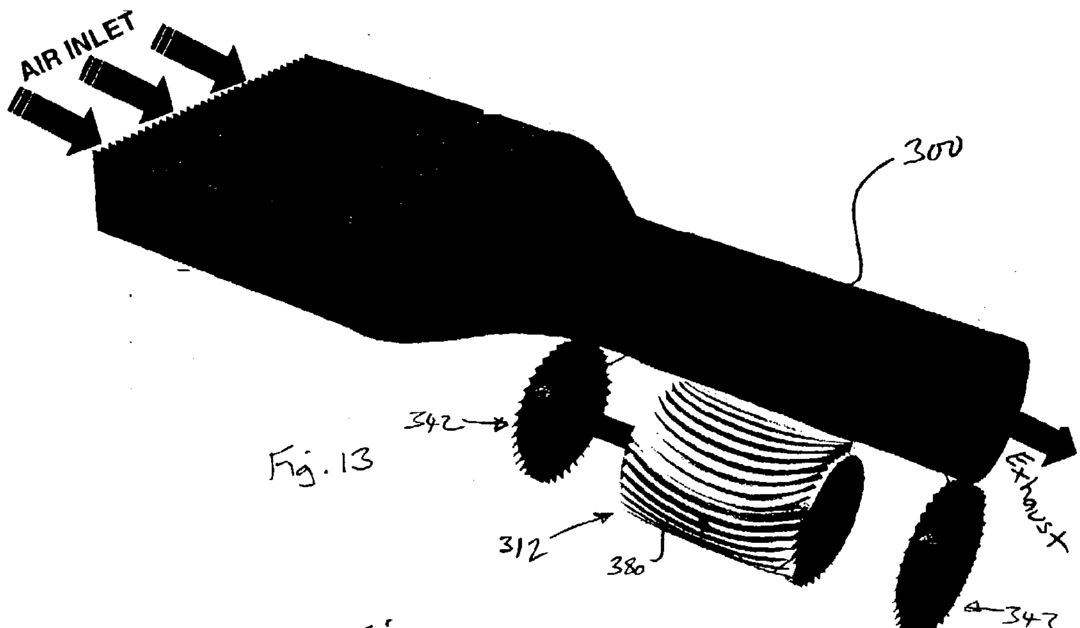
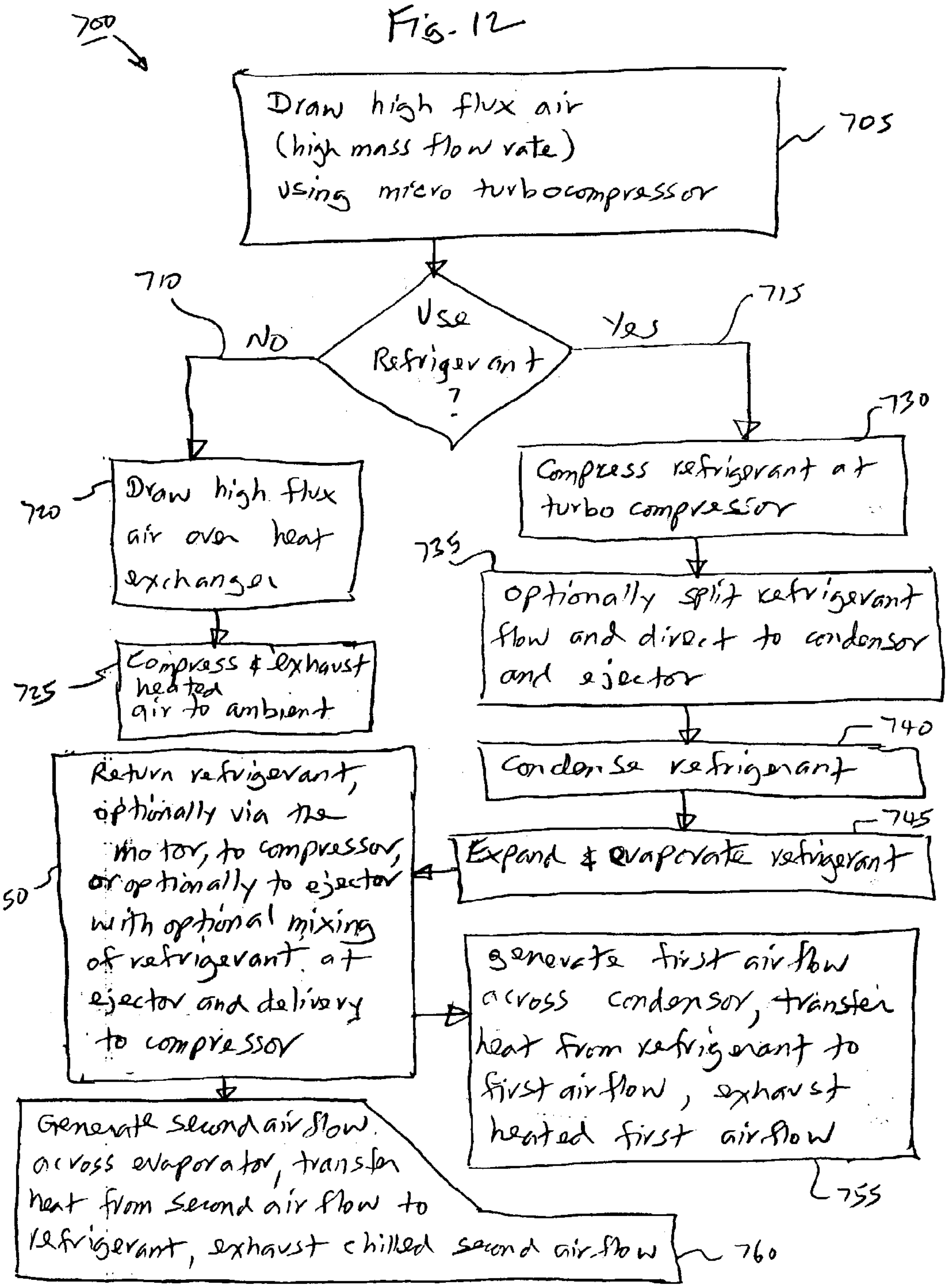


Fig. 13

Fig. 12


700
Draw high flux air (high mass flow rate) using micro turbocompressor
705
710
Use Refrigerant ?
No
Yes
715
Draw high flux air over heat exchanger
720
Compress & exhaust heated air to ambient
725
Compress refrigerant at turbo compressor
730
optionally split refrigerant flow and direct to condensor and ejector
735
Condense refrigerant
740
Expand & evaporate refrigerant
745
Return refrigerant, optionally via the motor, to compressor, or optionally to ejector with optional mixing of refrigerant at ejector and delivery to compressor
750
generate first air flow across condensor, transfer heat from refrigerant to first air flow, exhaust heated first air flow
755
Generate second air flow across evaporator, transfer heat from second air flow to refrigerant, exhaust chilled second air flow
760

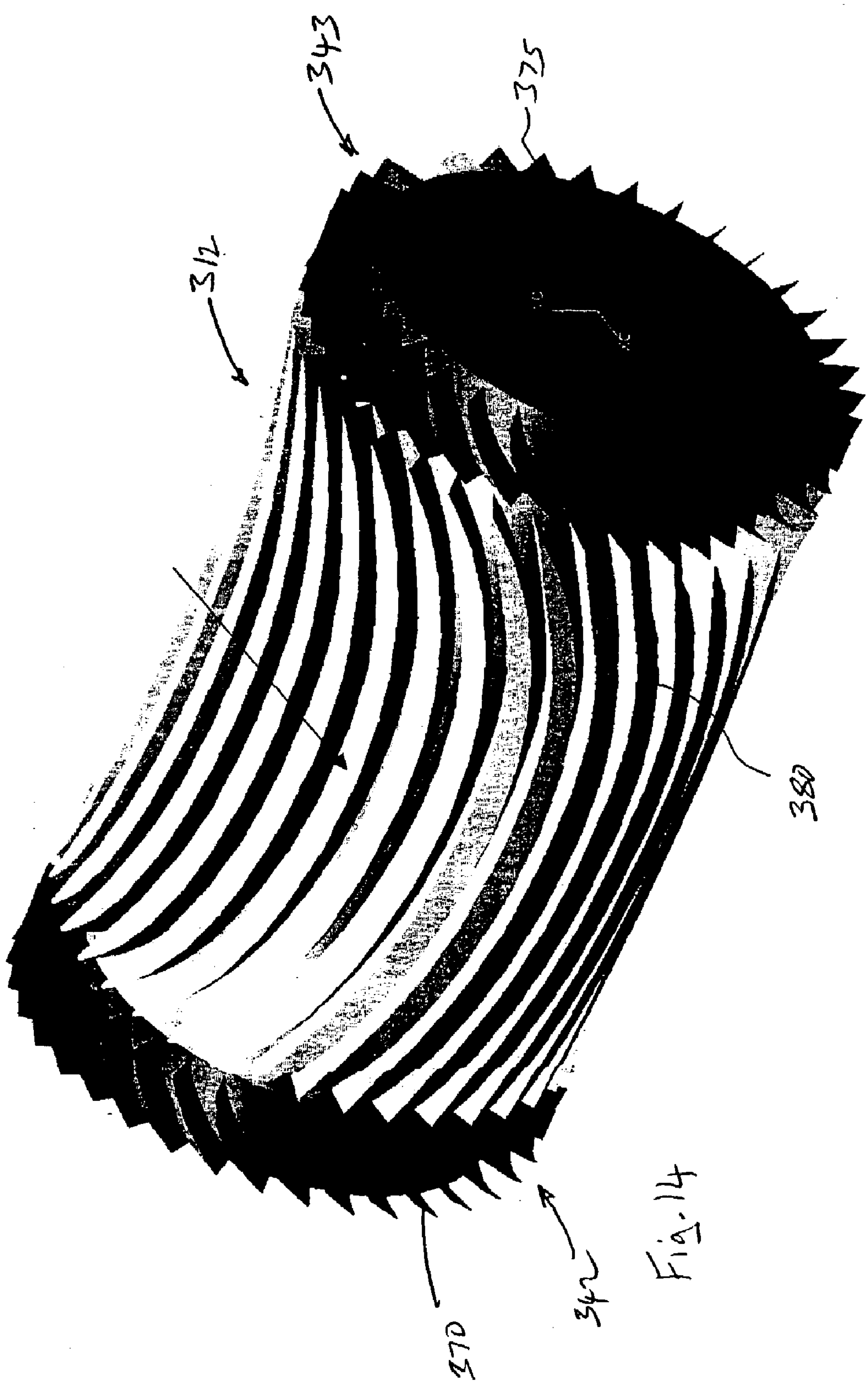


Fig. 14

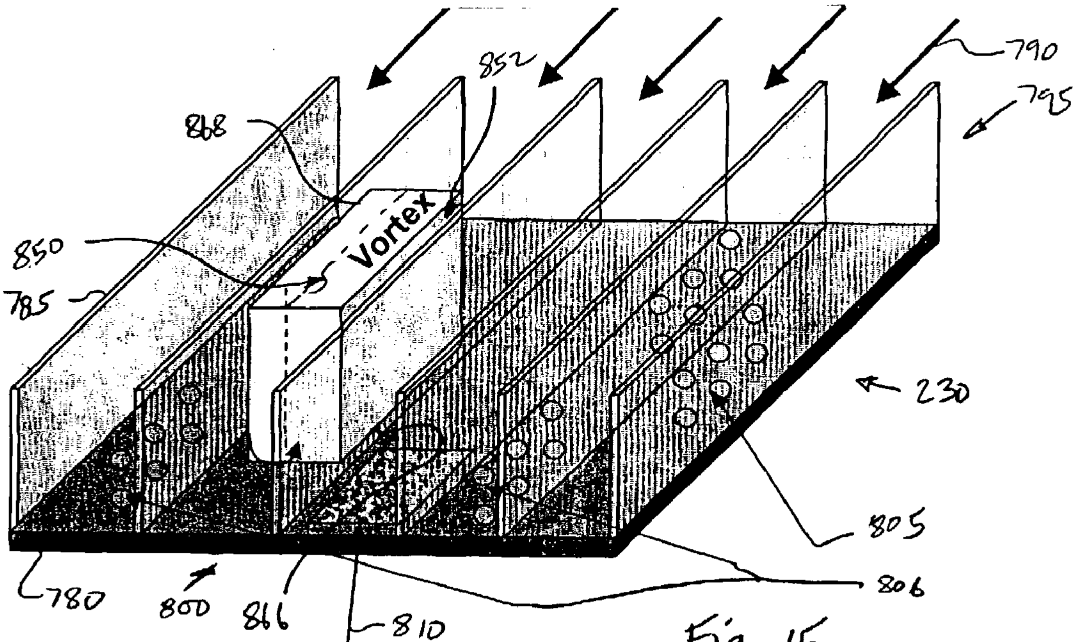

852
790
795
868
Vortex
850
785
230
805
780
800
866
810
806

Fig. 15

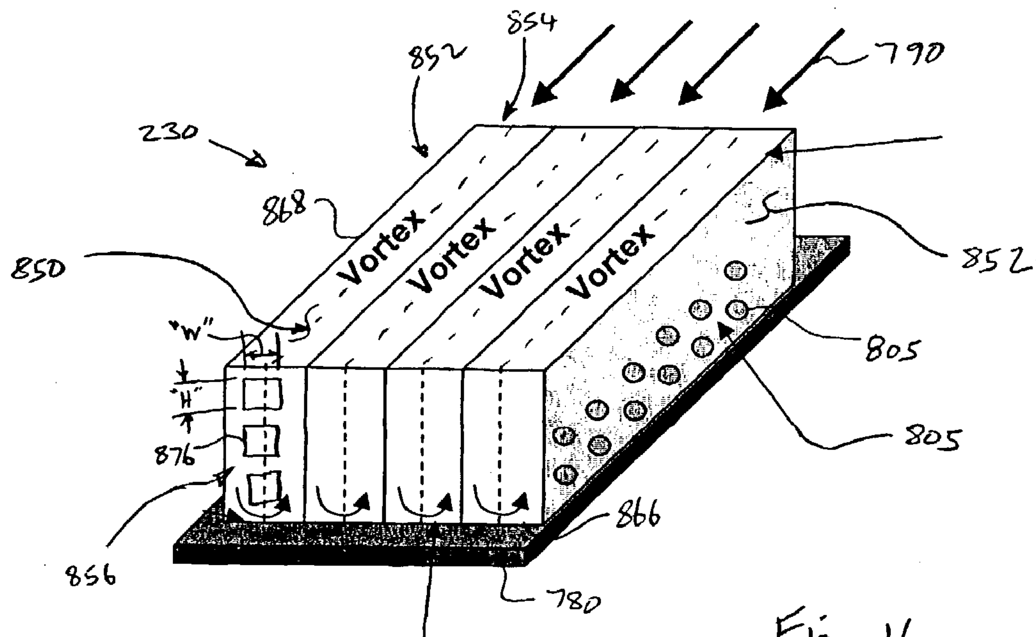

854
852
790
230
868
Vortex
Vortex
Vortex
Vortex
850
852
"W"
"H"
805
876
805
856
866
780

Fig. 16

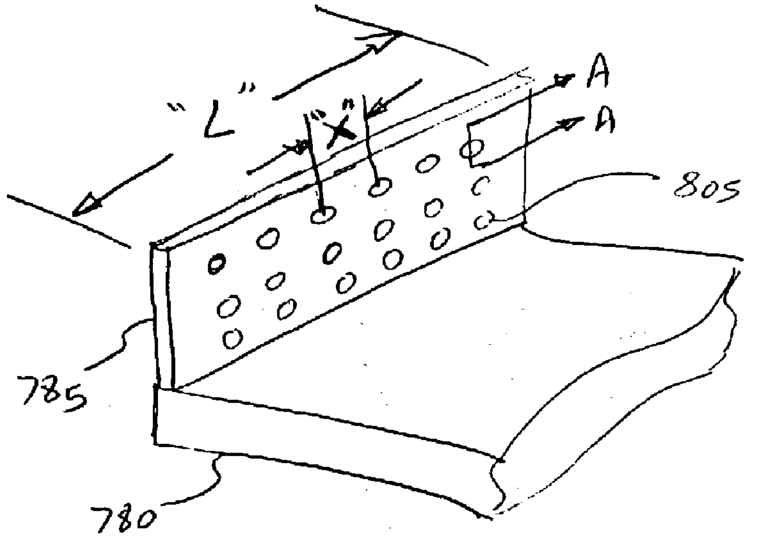


Fig. 17

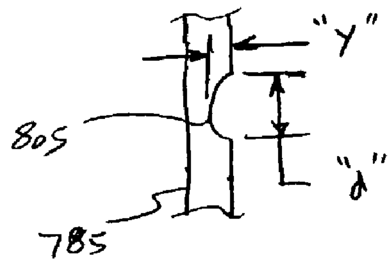


Fig. 18

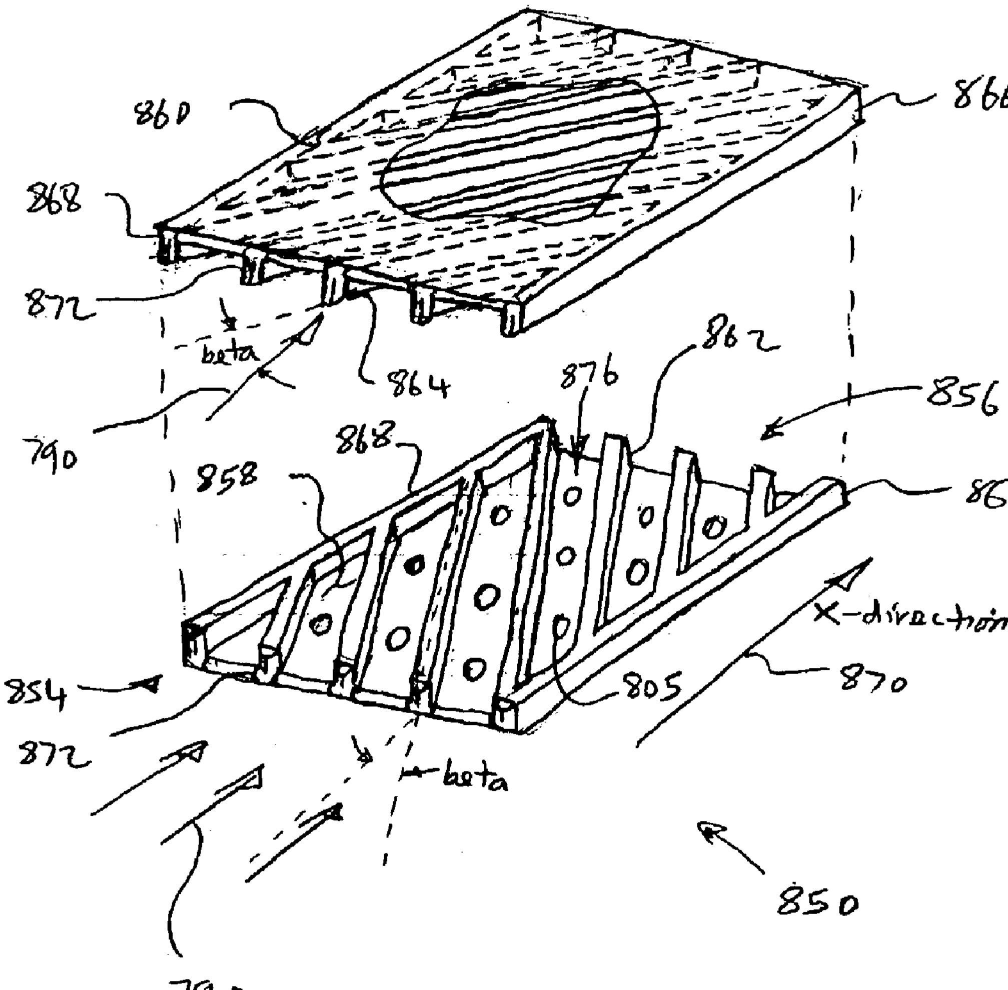

860
866
868
872
beta
864
790
876
862
856
868
858
866
X-direction
854
805
870
872
beta
850
790

Fig. 19

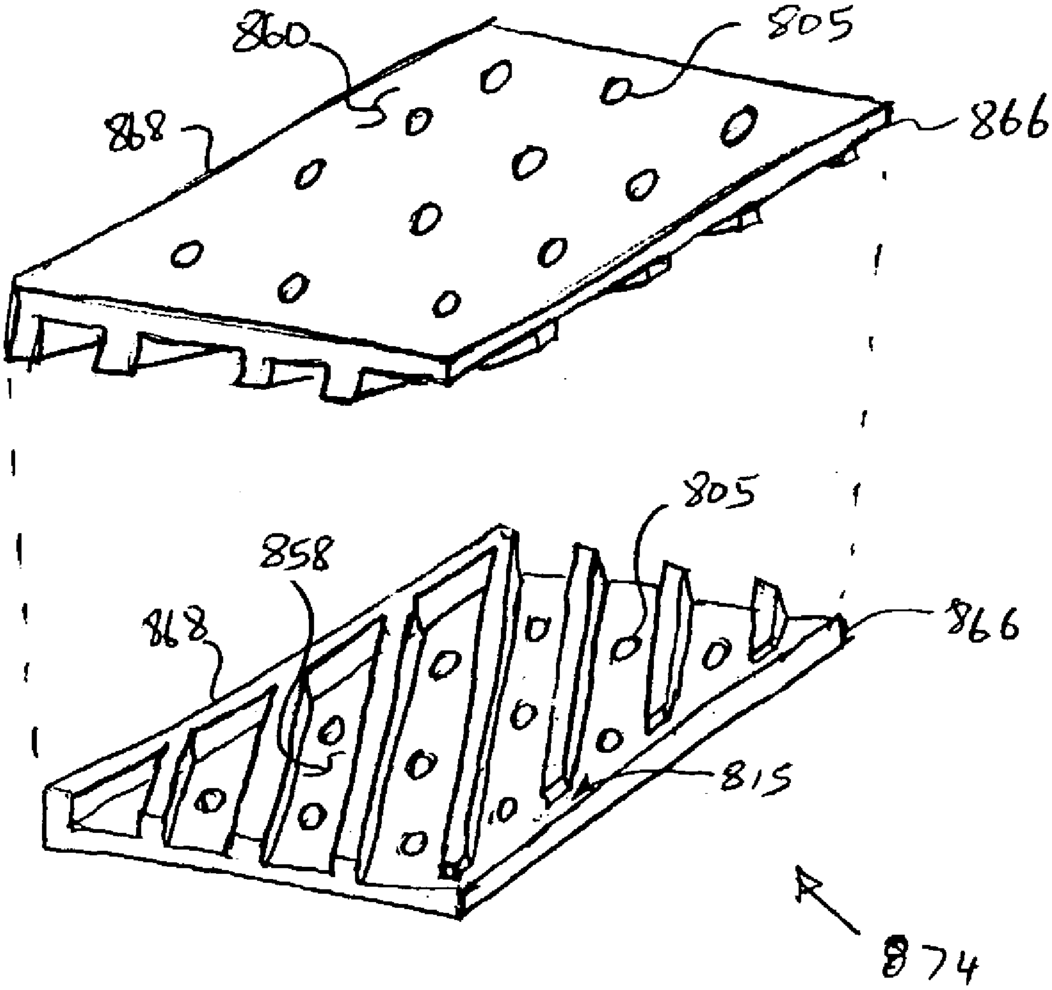


Fig. 20

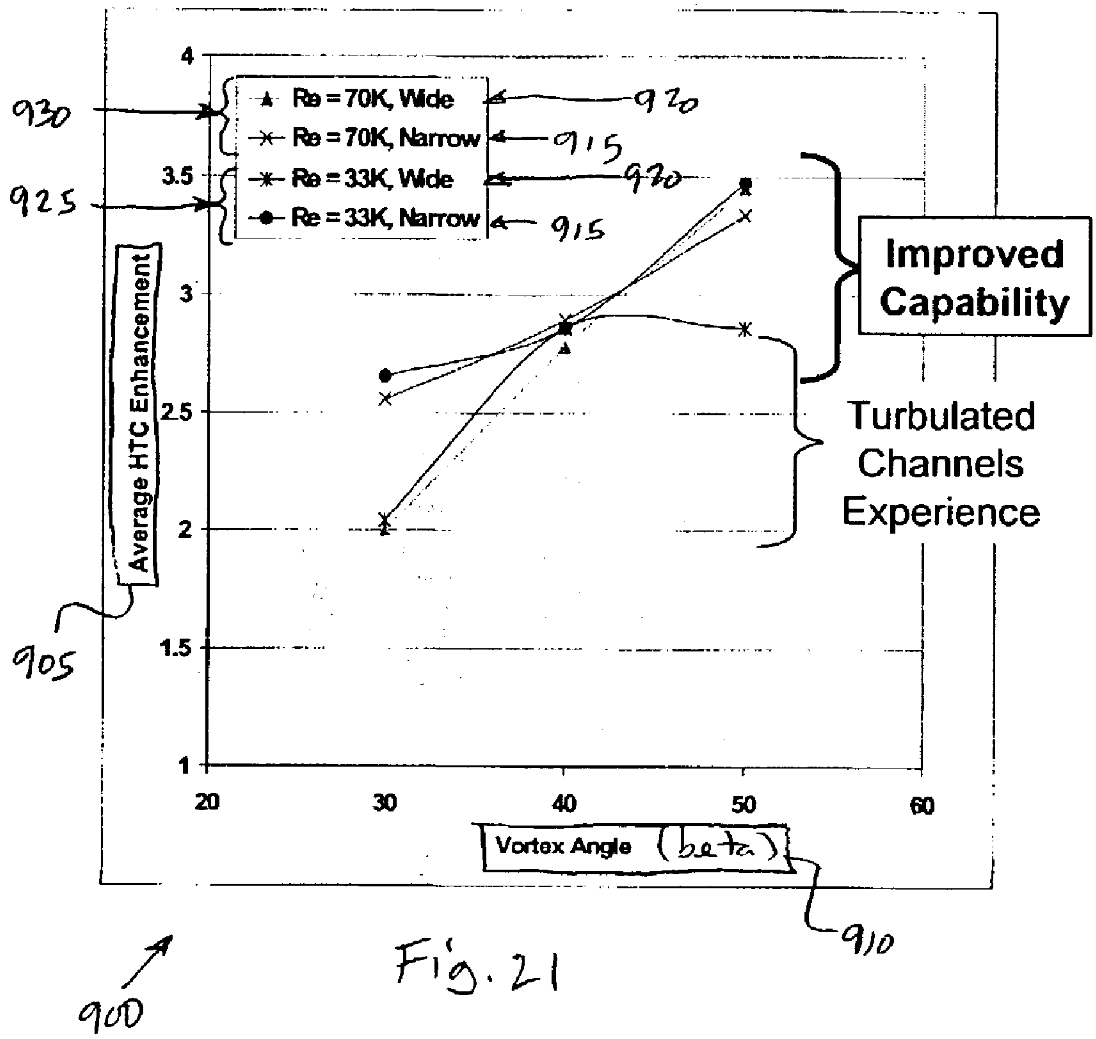
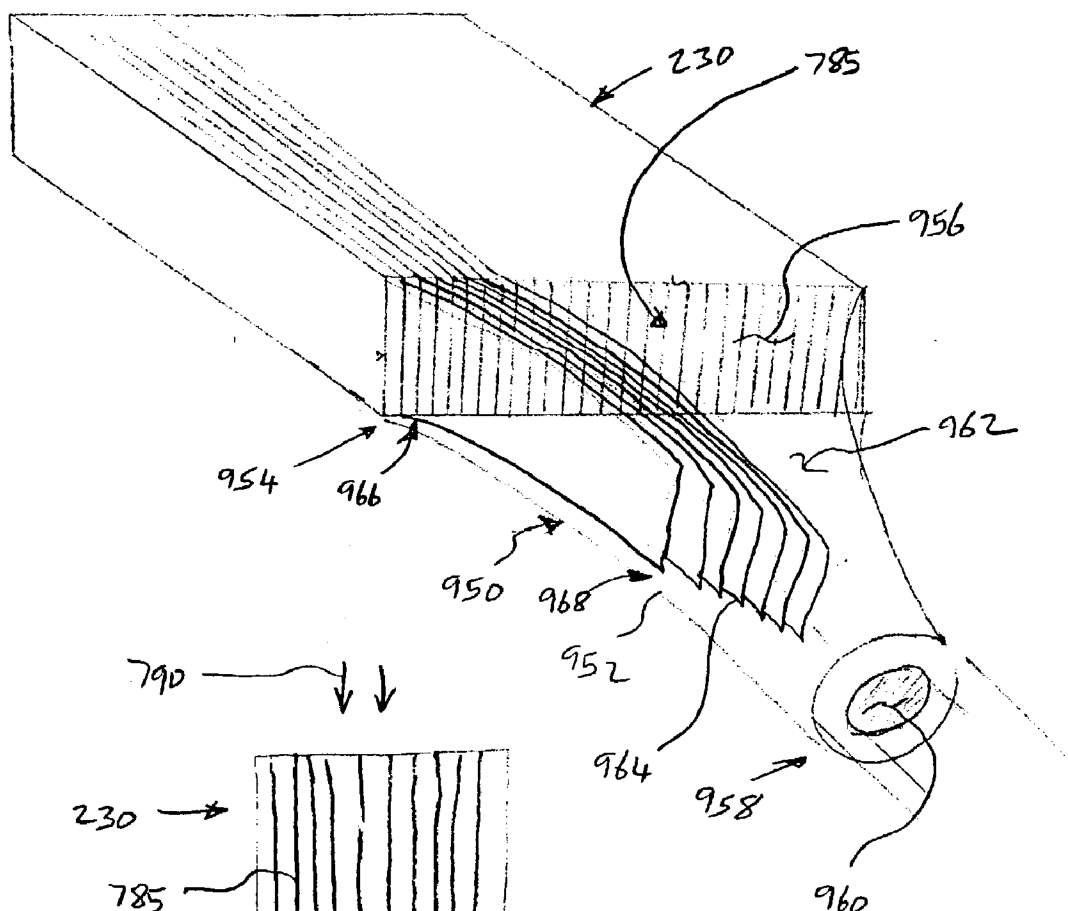

930
925
905
910
900
Re = 70K, Wide
Re = 70K, Narrow
Re = 33K, Wide
Re = 33K, Narrow
920
915
920
915
Improved Capability
Turbulated Channels Experience
Average HTC Enhancement
Vortex Angle (beta)
4
3.5
3
2.5
2
1.5
1
20
30
40
50
60

Fig. 21

Fig. 22


230
785
956
962
954
966
950
968
952
964
958
960

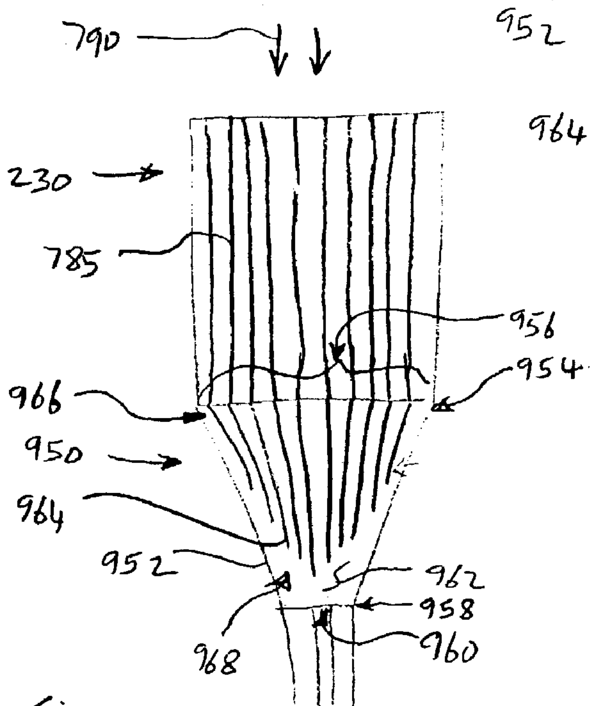
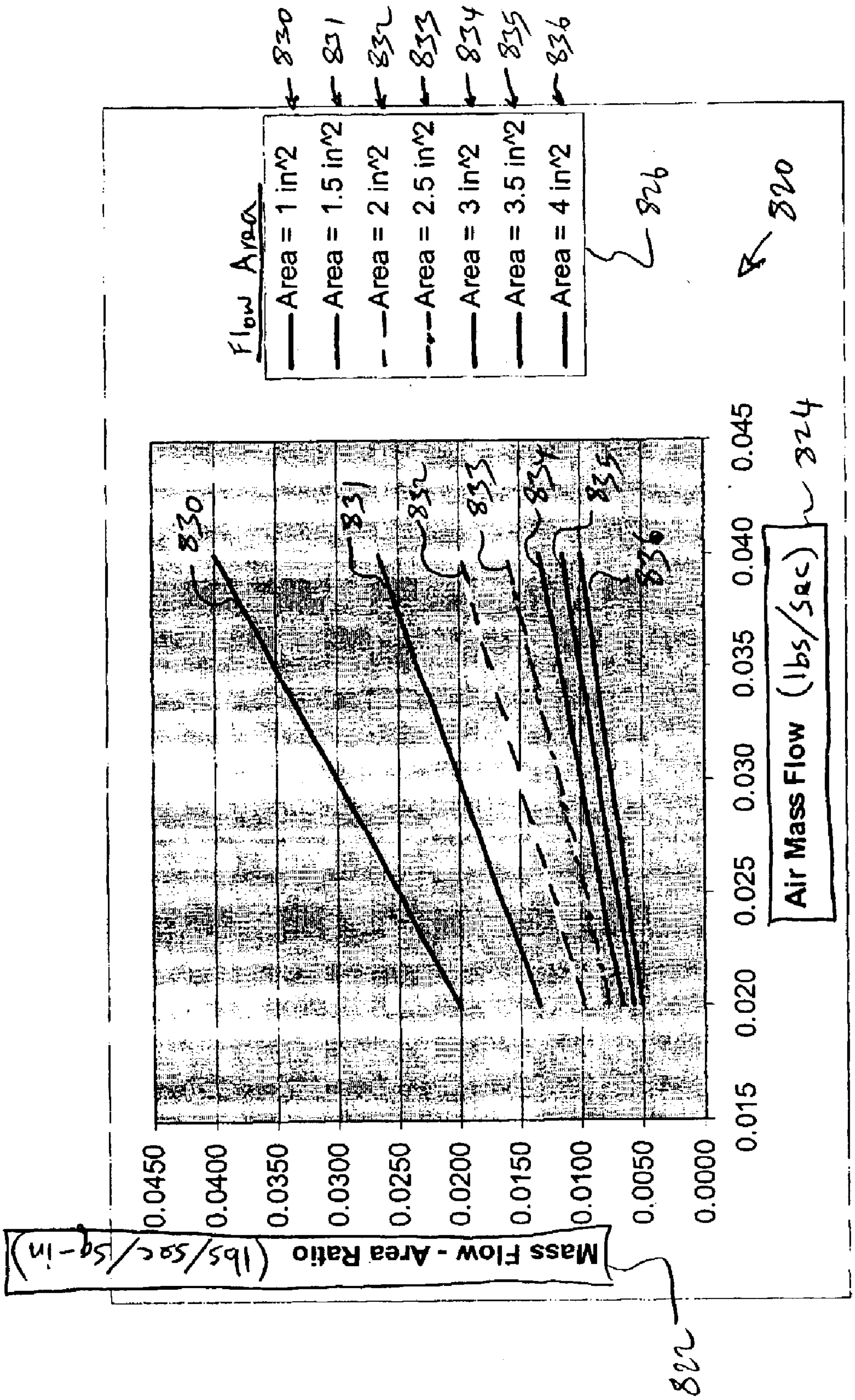

790
230
785
956
966
954
950
964
952
962
958
968
960

Fig. 23



Flow Area
Area = 1 in^2
Area = 1.5 in^2
Area = 2 in^2
Area = 2.5 in^2
Area = 3 in^2
Area = 3.5 in^2
Area = 4 in^2
830
831
832
833
834
835
836
826
820
Mass Flow - Area Ratio (lbs/sec/sq-in)
822
0.0450
0.0400
0.0350
0.0300
0.0250
0.0200
0.0150
0.0100
0.0050
0.0000
0.015
0.020
0.025
0.030
0.035
0.040
0.045
Air Mass Flow (lbs/sec)
824

Fig. 24

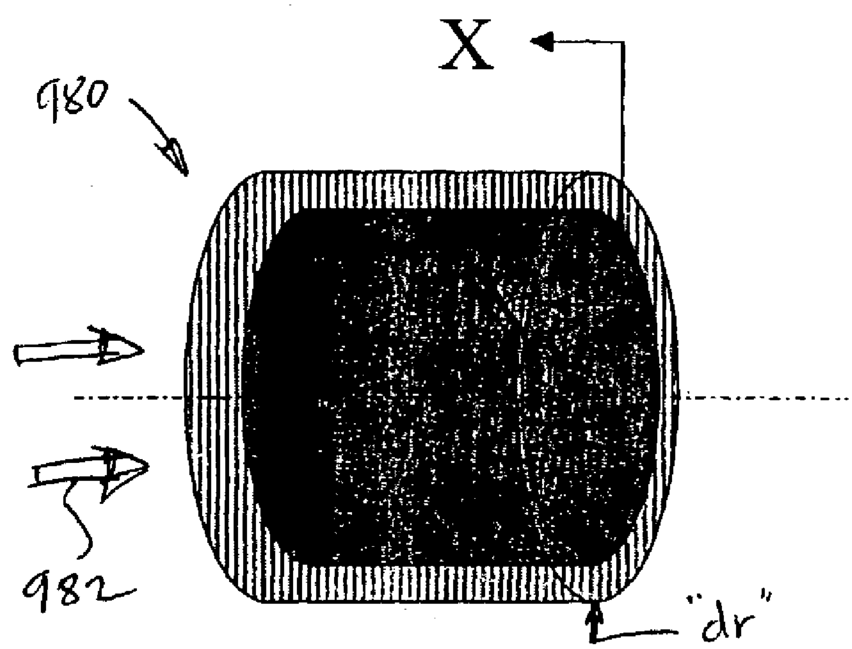


Fig. 25

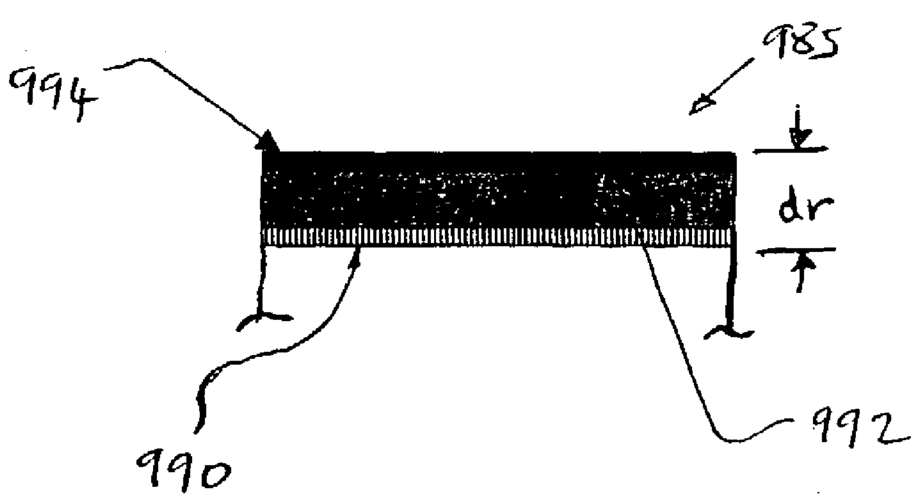


Fig. 26 (Section X-X)

METHOD, SYSTEM AND APPARATUS FOR COOLING HIGH POWER DENSITY DEVICES

BACKGROUND OF THE INVENTION

[0001] The present disclosure relates generally to the cooling of high power density devices, and particularly to the use of turbomachinery for cooling high power density devices.

[0002] The cooling of high power density devices (HPDD's), such as high power density integrated circuits (IC's) and central processing units (CPU's) for example, is a significant consideration in the design of computer servers, military avionic equipment, medical imaging equipment, and other systems employing high power density electronic devices. The term HPDD used herein refers to heat generating devices having a heat flux in excess of 100 Watts-per-square-centimeter. Today's trend is not only to design electronic systems with greater and greater computational speed and power, but also to design them with smaller and smaller footprints, the end result being a HPDD that generates a lot of heat in a small area that needs to be dissipated in order to avoid IC and CPU degradation. While the power density of today's electronic systems may be as high as 200 watts per square centimeter (W/sq-cm), the trend is toward 800 W/sq-cm or beyond over the next several years. In addition to heat generation, enclosure size constraints must also be taken into consideration. For example, today's computer servers typically employ circuit boards that are housed in enclosures with a height restriction of 1.75 inches, referred to as a 1U application, with multiple circuit boards being stacked adjacent one another in a rack chassis. With a typical electronic component having an ambient use temperature of no greater than 120 degree-Celsius (deg-C.) and a junction temperature restriction of 90 deg-C., cooling systems are employed to transfer the heat of the HPDD to the surrounding ambient. Typical cooling systems employed today include fans, blowers, heat sinks, and refrigeration systems, which tend to increase in size as the heat transfer demands increase. This size increase, however, is contrary to the design objective of a 1U application.

SUMMARY OF THE INVENTION

[0003] In one embodiment, a turbomachinery system for cooling a high power density device includes a turbomachine configured to deliver a high flux cooling medium and a high power density device arranged in fluid communication with the turbomachine, the turbomachine having a motor and a compressor driven by the motor.

[0004] In another embodiment, a turbomachine for delivering a high flux cooling medium to a device includes a motor, a compressor driven by the motor for compressing the high flux cooling medium, a housing containing the motor, the compressor, or both, wherein the housing provides a passage for the high flux cooling medium. An inlet at a first end of the housing accepts the high flux cooling medium, and an outlet at a second end of the housing discharges the high flux cooling medium.

[0005] In a further embodiment, a turbomachinery module for attaching to a surface and used for cooling high power density devices includes a module surface, a turbomachine for delivering high flux air, a heat exchanger for thermally coupling to and cooling a high power density device, and a transition duct arranged intermediate the heat exchanger and the turbomachine for funneling the air flow from the heat exchanger to the turbomachine, the turbomachine being downstream from the heat exchanger. The turbomachine includes a motor, a compressor driven by the motor for compressing the high flux air, and a housing. The housing contains the motor, the compressor, or both and provides a passage for air flow. The housing includes an inlet at a first end for accepting air flow and an outlet at a second end for discharging air flow.

[0006] In yet another embodiment, a method for cooling a high power density device includes drawing air over a heat exchanger at a high mass flow rate, compressing the air at a turbocompressor, and exhausting the heated air to ambient. The heat exchanger is thermally coupled to a high power density device that results in an increase in air temperature as the air passes over the heat exchanger. The turbocompressor has an overall dimension of no greater than 1.75 inches.

[0007] In yet a further embodiment, a method for delivering a high flux cooling medium to a high power density device includes compressing a refrigerant at a turbocompressor having an overall dimension of no greater than 1.75 inches, condensing the refrigerant by removing heat from the refrigerant, expanding and evaporating the refrigerant to create a cold surface for cooling the high power density device, and returning the expanded refrigerant to the turbocompressor to repeat the closed-loop cycle.

[0008] In another embodiment, a micro turbocompressor for delivering a high flux cooling medium to a device includes a motor, a first stage micro compressor disposed at one end of the motor, and a second stage micro compressor disposed at the other end of the motor.

[0009] In a further embodiment, a micro turbocompressor for delivering a high flux cooling medium to a device includes one or more motors and a plurality of micro compressors disposed at opposite ends of the motors, the plurality of micro compressors arranged to drive air across the outer surface of the stators of the motors.

[0010] In yet another embodiment, a heat exchanger for cooling a high power density device includes a base for thermally coupling the heat exchanger to the high power density device, and a plurality of parallel cooling fins arranged perpendicular to the base for receiving driven air at one end and discharging the driven air at the opposite end, the cooling fins having a plurality of concavities.

[0011] In yet a further embodiment, a heat exchanger for cooling a high power density device includes a base for thermally coupling the heat exchanger to the high power density device, a plurality of parallel cooling fins arranged perpendicular to the base for receiving driven air at one end and discharging the driven air at the opposite end, and a localized cooling region at the base and between the cooling fins for providing a local region of increased surface area for enhanced heat transfer.

[0012] In another embodiment, a heat exchanger for cooling a high power density device includes a base for thermally coupling the heat exchanger to the high power density device, a plurality of parallel cooling fins arranged perpendicular to the base for receiving driven air at one end and discharging the driven air at the opposite end, and a vortex chamber arranged between the cooling fins for creating a vortex like air flow as the air is driven between the cooling fins and through the vortex chamber.

[0013] In a further embodiment, a heat exchanger for cooling a high power density device includes a base for thermally coupling the heat exchanger to the high power density device, a plurality of parallel cooling fins arranged perpendicular to the base for receiving driven air at one end and discharging the driven air at the opposite end, a plurality of vortex chambers arranged between the plurality of parallel cooling fins for receiving the driven air at one end and discharging the driven air at the opposite end, the plurality of vortex chambers for creating a vortex like air flow as the air is driven from the one end to the opposite end. The plurality of vortex chambers include a plurality of first sides that have internal ribs arranged oblique to the air flow with a portion of the ribs being at an angle of positive-beta with respect to the direction of air flow, and a plurality of second sides that have internal ribs arranged oblique to the air flow with a portion of the ribs being at an angle of negative-beta with respect to the direction of air flow. The plurality of first and second sides further include a plurality of first and second edges, the plurality of first edges being open and the plurality of second edges being closed. The plurality of first edges are arranged proximate to the base. A localized cooling region at the base proximate the plurality of first edges provides a local region of increased surface area for enhanced heat transfer.

[0014] In yet another embodiment, a method for enhancing the heat transfer characteristic of a heat exchanger for cooling a high power density device includes receiving driven air at a first end of a plurality of cooling fins of the heat exchanger, driving the air across the plurality of cooling fins for transferring heat from the high power density device to ambient, disturbing the air flow as it is driven across the plurality of cooling fins by employing at least one of a plurality of concavities in the plurality of cooling fins, a plurality of localized cooling regions at the base of the heat exchanger and between the plurality of cooling fins, or a plurality of vortex chambers between the plurality of cooling fins for creating a vortex flow between the plurality of cooling fins, and discharging the heated air at a second end of the plurality of cooling fins.

[0015] In yet a further embodiment, a transition duct for a turbomachinery system includes a duct housing having a first end with a first flow area for receiving driven air from a heat exchanger having a plurality of cooling fins and a second end with a second flow area for discharging driven air to a turbomachine, the duct housing defining an internal cavity that transitions from the first flow area to the second flow area, and a plurality of flow control fins within the internal cavity for managing the change in flow area from the first flow area to the second flow area.

[0016] In another embodiment, a method for transitioning high flux air from a region having a first flow area to a region having a second flow area wherein the first flow area is larger than the second flow area includes receiving the high flux air at a first end of a transition duct having the first flow area, segmenting the high flux air flow into separate air flow channels between a plurality of flow control fins, funneling the high flux air between the flow control fins from the first flow area toward the second flow area, and discharging the high flux air at a second end of the transition duct having the second flow area.

BRIEF DESCRIPTION OF THE DRAWINGS

[0017] Referring to the exemplary drawings wherein like elements are numbered alike in the accompanying FIGS.:

[0018] **FIG. 1** depicts a computer server system for use with an embodiment of the invention;

[0019] **FIG. 2** depicts an isometric view of a turbomachinery system in accordance with an embodiment of the invention;

[0020] **FIG. 3** depicts a side view of the system of **FIG. 2**;

[0021] **FIG. 4** depicts a turbomachine in accordance with an embodiment of the invention;

[0022] **FIG. 5** depicts a centrifugal compressor for use in an embodiment of the invention;

[0023] **FIG. 6** depicts a multi-wheel axial compressor for use in an embodiment of the invention;

[0024] **FIG. 7** depicts an alternative embodiment to the turbomachinery system of **FIGS. 2 and 3**;

[0025] **FIG. 8** depicts an alternative embodiment to the turbomachine of **FIG. 4**;

[0026] **FIG. 9** depicts an isometric view of an alternative embodiment to the turbomachinery system of **FIG. 2**;

[0027] **FIG. 10** depicts an alternative embodiment to the turbomachine of **FIG. 4**;

[0028] **FIG. 11** depicts generally a sectional view of a condensor for use in an embodiment of the invention;

[0029] **FIG. 12** depicts a process for employing an embodiment of the invention;

[0030] **FIGS. 13 and 14** depict an embodiment of the invention having multi-stage micro compressors;

[0031] **FIGS. 15 and 16** depict heat exchangers for use in an embodiment of the invention;

[0032] **FIG. 17** depicts a detailed view of a portion of the heat sink of **FIG. 15**;

[0033] **FIG. 18** depicts a further detailed view of the detailed view of **FIG. 17**;

[0034] **FIGS. 19 and 20** depict exploded isometric views of a vortex chamber for use in an embodiment of the invention;

[0035] **FIG. 21** depicts a graphical illustration of the average heat transfer coefficient enhancement as a function of vortex angle in accordance with an embodiment of the invention;

[0036] **FIG. 22** depicts an isometric view of a heat exchanger and transition duct for use in an embodiment of the invention;

[0037] **FIG. 23** depicts a plan view of the heat exchanger and transition duct of **FIG. 22**;

[0038] **FIG. 24** depicts a graphical illustration of mass-flow-rate-to-flow-area ratio as a function of air mass flow rate for more than one embodiment of the invention;

[0039] **FIG. 25** depicts a portion of a flow duct for use in an embodiment of the invention; and

[0040] **FIG. 26** depicts a cross-sectional view of the wall thickness of the flow duct of **FIG. 25**.

DETAILED DESCRIPTION OF THE INVENTION

[0041] An embodiment of the present invention provides an apparatus and method for cooling a high power density device such as a high-end integrated circuit (IC) for use in a server computer system using small turbomachinery, such as a micro turbocompressor, and a high flux heat exchanger. The turbomachinery is sized for applications having a dimensional restriction of 1.75 inches ("1U" applications). While the embodiment described herein depicts an integrated circuit as an exemplary high power density device, it will be appreciated that the disclosed invention is also applicable to other high power density devices, such as military avionics and medical imaging componentry and equipment for example.

[0042] **FIG. 1** is an exemplary embodiment of a computer server system **100** utilizing an embodiment of the invention, the system **100** including a mounting fixture **110** and a plurality of system circuit boards **120**, with each circuit board **120** having a turbomachinery system **200**, best seen by now referring to **FIG. 2**.

[0043] An embodiment of turbomachinery system **200** is depicted in **FIG. 2** having a mounting surface **210**, which may be a section of circuit board **120** or a separate modular support structure, a high power density device (HPDD) (e.g., an integrated circuit) **220** (seen in **FIG. 3**) coupled to surface **210**, a turbomachine **300** coupled to surface **210** for delivering high flux air (a first type of cooling medium), a heat exchanger **230** thermally coupled to HPDD **220**, and a transition duct **240** arranged intermediate heat exchanger **230** and turbomachine **300** for funneling the air flow from heat exchanger **230** to turbomachine **300**. The term high flux air used herein refers to air flow on the order of 150 pounds-per-hour (lbs/hr) or greater, as will be discussed in more detail below. Turbomachinery system **200** has an overall dimension "D" not greater than 1.75 inches, thereby making it suitable for 1U applications. An alternative embodiment of transition duct **240** is discussed below in reference to **FIGS. 22 and 23**.

[0044] Heat exchanger **230** may be any heat exchanger suitable for the cooling requirements of the application, but preferably is a heat exchanger that is not susceptible to fouling, thereby avoiding stalling of turbomachine **300** from excessive pressure drop across heat exchanger **230**, as will be discussed in more detail below in reference to FIGS. **15-19**. In an alternative embodiment, heat exchanger **230** has acoustical damping characteristics, which may be achieved through material selection, material deposition, or geometric design, which will be discussed in more detail below in reference to **FIG. 25**.

[0045] Turbomachine **300**, as best seen by now referring to **FIG. 4**, includes a housing **305**, a first motor **310** having a first rotor **315** and a first stator **320**, an optional second motor **325** having a second rotor **330** and a second stator **335**, a compressor **340** driven by motor **310**, **325** for compressing the high flux air as it enters turbomachine **300** at an inlet **345**, and an optional expansion turbine **350** driven by motor **310**, **325** for expanding the high flux air as it is discharged from turbomachine **300** at an outlet **355**. An optional ejector **360** at the inlet end of turbomachine **300** may be included for controlling the air flow at inlet **345**.

[0046] Motors **310**, **325** preferably have magnetic bearings or other non-contact bearing solutions, thereby reducing frictional effects for improved overall efficiency, reduced heat generation, reduced wear, and prolonged life.

[0047] Compressor **340** may be either an axial compressor or a centrifugal compressor, and may be arranged as a single wheel compressor or a multi-wheel compressor. **FIG. 4** depicts compressor **340** as a single wheel axial compressor, while **FIGS. 5 and 6** depict a centrifugal compressor **341** and a multi-wheel axial compressor **342**, respectively.

[0048] Under operating conditions, turbomachine **300** receives power from wires **365** to energize motors **310**, **325**. Wires **365** are connected to power supplies **130** in system **100** using appropriate wire connection means (not shown). Motors **310**, **325** drive compressor **340** and optional expansion turbine **350** at a high rate of speed, on the order of 50,000 revolutions-per-minute (RPM) for example, thereby drawing air into inlet **345**, compressing the air at compressor **340**, driving the air along the length of housing **305**, and optionally expanding and exhausting the air at expansion turbine **350** and outlet **355**. As the high flux air is drawn through turbomachine **300** it is also drawn through transition duct **240** and across heat exchanger **230** (see **FIG. 2**), thereby providing for the transfer of heat from HPDD **220** to ambient through heat exchanger **230** (see **FIG. 3**). Inlet and exhaust flow arrows Fi, Fe, shown in **FIG. 2**, depict the direction of air flow, indicating that turbomachine **300** is downstream from HPDD **220** and heat exchanger **230**, which avoids the heat of motors **310**, **325** from negatively influencing the cooling performance of heat exchanger **230**. However, under appropriate circumstances, turbomachine **300** may be arranged upstream from HPDD **220** and heat exchanger **230**, which may be achieved by reversing the flow of air through turbomachine **300**. Housing **305** may be arranged to contain compressor **340**, motor **310**, **325**, and optional expansion turbine **350**, or alternatively, may be arranged to contain only compressor **340**, as will be discussed in more detail below. Surface **210** may include more than one HPDD **220**, and typically includes many circuit components, including both high and low power density devices.

[0049] The air drawn across heat exchanger **230** and exhausted through outlet **355** by turbomachine **300** may be drawn from the ambient internal to or external of mounting fixture **110**, and may be exhausted to the ambient internal to or external of mounting fixture **110**, depending on the system cooling needs. Duct work to accomplish the alternative air flows may be accomplished by incorporating air ducts into the structure of the optional fan trays **140**, or alternatively replacing the optional fan trays **140** with appropriate air ducts.

[0050] In an alternative embodiment that employs a parallel system **400**, as depicted in **FIG. 7**, turbomachine **300** is replaced by two turbomachines **405**, **410** arranged in parallel, and transition duct **240** is replaced by transition duct **415** having dual outlet ports **420**, **425** for delivering high flux air flow to turbomachines **405**, **410**. See the discussion below in reference to **FIGS. 22 and 23** for an alternative embodiment of transition duct **415**. High flux air flow across heat exchanger **230** is similar to that described above. The operation of turbomachines **405**, **410** and the geometry of outlet ports **420**, **425** provide equivalent air flow through each turbomachine **405**, **410**. Regarding the single turbomachine configuration of **FIG. 2**, it has been estimated that a 320 Watt motor in turbomachine **300** having a diameter of 1.5 inches and a length of 1 inch would be required in order to deliver 0.1 pounds-per-second (lbs/sec) of air at a motor speed of 50,000 revolutions-per-minute (RPM), a pressure ratio of 1.04, a compressor efficiency of 0.6, and a motor efficiency of 0.8, while in the dual turbomachine configuration of **FIG. 7**, the same total air flow characteristics would be achieved by employing 160 Watt motors in each of the two turbomachines **405**, **410**. While the total power requirements of the two configurations are the same, the power requirement for a given turbomachine in the two configurations differs by a factor of 2:1, which is substantial for a turbomachine having an overall dimension limited to no greater than 1.75 inches.

[**0051**] In another alternative embodiment that employs either a single turbomachine **300**, as in turbomachinery system **200** of **FIG. 2**, or parallel turbomachines **405**, **410**, as in the parallel system **400** of **FIG. 7**, an air particle ionizer arranged upstream of heat exchanger **230** and an air particle deionizer arranged downstream of the outlets of turbomachines **300**, **405**, **410** serve to ionize and deionize the air particles to prevent particulate fouling and improve system life. Prior to entering heat exchanger **230** and turbomachines **300**, **405**, **410**, the air particles are ionized in order to prevent particulate fouling. Upon discharge from turbomachines **300**, **405**, **410**, the air particles are deionized in order to neutralize the electrical charge of the air particles to prevent static buildup on electrical components within system **100**.

[**0052**] In an alternative embodiment that employs turbomachinery system **200**, outlet **355** of housing **305** includes an entrainment nozzle **435** having a secondary air duct **440** for entraining ambient air **445** for providing a secondary cooling air flow **450** as depicted in **FIG. 8**. Referring to **FIG. 8**, which depicts a cross section view of entrainment nozzle **435**, the entry **455** of entrainment nozzle **435** is attached to outlet **355** of turbomachine **300** such that air flow through turbomachine **300** passes through outlet **355**, into entry **455** of entrainment nozzle **435**, and out the exhaust port **460** as indicated by exhaust flow arrow Fe. The exhausting air flow Fe creates a suction at entrainment port **465** that draws ambient air **445** into secondary air duct **440**. The secondary air flow **470** is bifurcated at internal wall section **475** resulting in a secondary exhaust air flow **480** that mixes with and exhausts with exhaust flow Fe, and a secondary cooling air flow **450** that is directed as needed for additional cooling. Entrainment nozzle **435** may be integral with outlet **355** of turbomachine **300** or separately attached using known means such as bolts, screws, or welding for example.

[**0053**] Another embodiment of turbomachinery system **200** is depicted in **FIG. 9** having a mounting surface **210**, which may be a section of circuit board **120** or a separate modular support structure, a high power density device (HPDD) (e.g., an integrated circuit) **220** coupled to surface **210**, and a turbomachine **500** that replaces turbomachine **300**, heat exchanger **230** and transition duct **240** of **FIG. 2**. Turbomachine **500** is coupled to surface **210** and delivers high flux cooled air (a first type of cooling medium) and high flux refrigerant (a second type of cooling medium) for cooling HPDD **220**. Alternatively, turbomachine **500** may deliver only high flux cooled air or high flux refrigerant, as will be discussed below in more detail. Turbomachinery system **200** has an overall dimension "D" not greater than 1.75 inches, thereby making it suitable for 1U applications. In an embodiment, turbomachinery system **200** is integral with circuit board **120**, which fits inside mounting fixture **110** of system **100** depicted in **FIG. 1**.

[**0054**] Turbomachine **500**, best seen by now referring to **FIG. 10**, includes a first turbomachine **505**, a second turbomachine **510**, a condenser **515**, and an expander/evaporator **520**. First turbomachine **505**, condensor **515** and expander/evaporator **520** are part of a closed-loop refrigeration cycle (CLRC) **525**. First turbomachine **505** includes a compressor **530**, a motor **535**, preferably with magnetic bearings, a turbine **540**, and an optional ejector **545**. Motor **535** drives both compressor **530** and turbine **540**. Expander/evaporator **520** may consist of a cold plate, a heat exchanger, or both. Second turbomachine **510** includes a turbofan **550** and a motor **555**, which preferably has magnetic bearings. Surrounding turbomachine **500** is a housing **560** to provide air flow passages for directing high flux air flow across condenser **515** and expander/evaporator **520**, as will be discussed in more detail below.

[**0055**] Under operating conditions, first turbomachine **505** operates to drive a refrigerant through CLRC **525**, and second turbomachine **510** operates to drive high flux air across condenser **515** and evaporator/expander **520**.

[**0056**] In the CLRC **525**, compressor **530** compresses refrigerant gas to raise the refrigerant's temperature and pressure. Compressor **530** is coupled via supply lines, depicted generally by numeral **525**, to condensor **515**, which includes coils **570** (seen in **FIG. 11**) that allow the hot refrigerant gas to dissipate heat. The coils **570** are cooled by high flux air, generally depicted by numeral **565**, generated by turbofan **550** at second turbomachine **510** flowing across condensor **515**. Condensor **515** may also include a heat exchanger **230** coupled to coils **570** for improved heat transfer. Ambient air **600** following air flow path **565** passes over heat exchanger **230** of condensor **515**, whereby heat is transferred to the air and the refrigerant gas is cooled. The heated air **602** is exhausted through outlet **620** of housing **560**. As the refrigerant gas cools, it condenses into a refrigerant liquid at high pressure and flows into expander/evaporator **520**. Expander/evaporator **520** includes an expansion device, which is typically an expansion valve **575** such as a needle valve. Expansion valve may be controlled by a control system **580** for controlling the pressure drop of the refrigerant across the expansion valve, thereby providing controlled pressure drops within the CLRC **525** to prevent stalling of first turbomachine **505** on startup. As the refrigerant liquid flows through the expansion valve **575**, the refrigerant liquid moves from a high pressure zone to a low pressure zone, which allows the refrigerant to expand and evaporate within the expander/evaporator **520**, resulting in a drop in refrigerant temperature. The reduced refrigerant temperature is used to cool HPDD **220**, which is discussed below in more detail. The refrigerant gas is then returned to compressor **530** via turbine **540** and optional ejector **545**, where the CLRC **525** repeats. An alternative flow path for the low pressure/low temperature refrigerant gas from turbine **540** to ejector **545** is via motor **535**, depicted generally by numeral **590**, which enables cooling of motor **535**.

[**0057**] Optional ejector **545** is used to mix refrigerant at high pressure coming from compressor **530** with refrigerant at low pressure coming from turbine **540**, thereby increasing the mass flow through compressor **530** to prevent compressor stall. The high pressure flow path from compressor **530** to ejector **545** is depicted generally by numeral **585**. An alternative arrangement to prevent compressor stall includes a variable speed compressor and a control system (see wires **365** in **FIG. 4**) for controlling the variable speed compressor.

[**0058**] In one embodiment, expander/evaporator **520** includes a heat exchanger **230** (see **FIGS. 2 and 11**), which is cooled by the low temperature refrigerant. Ambient air **600** driven by second turbomachine **510** passing over expander/evaporator **520** is cooled by heat exchanger **230**. The chilled air **605** on the downstream side of expander/evaporator **520** is directed out of turbomachine **500** via passages in housing **560** and directed towards HPDD **220**, as shown in **FIGS. 9 and 10**.

[**0059**] In another embodiment, expander/evaporator **520** includes a cold plate **610**, which is cooled by the low temperature refrigerant. Cold plate **610**, which is thermally coupled to HPDD **220**, cools HPDD **220** as the high temperature HPDD **220** passes heat to cold plate **610**. Cold plate **610** may be located within housing **560** of turbomachine **500** as shown in **FIG. 10**, or alternatively located on surface **210** of turbomachinery system **200** at a distance from turbomachine **500** via supply lines **615** as depicted in **FIG. 9**.

[**0060**] In yet another embodiment, expander/evaporator **520** includes both a heat exchanger **230** and a cold plate **610**, where heat exchanger **230** is located within housing **560** of turbomachine **500** and cold plate **610** is located external of housing **560** at a distance from turbomachine **500**. By segmenting expander/evaporator **520** into two discrete components, one having a cold plate **610** component and a second having a heat exchanger **230** component, and sizing turbomachine **500** appropriately, cooling of HPDD **220** may be achieved by cold plate **610** and cooling of secondary components on circuit board **120** may be achieved by chilled air **605** from heat exchanger **230** being directed for that purpose. Since cooling of HPDD **220** is desired, it is preferable that CLRC **525** first delivers cold refrigerant to cold plate **610** and then to heat exchanger **230**, however, this arrangement is not limiting and alternative refrigerant flow paths may be employed. It will also be appreciated that the relative location of cold plate **610** and heat exchanger **230** described herein is not meant to be limiting and that any arrangement suitable for a particular cooling task will still adhere to the teaching of the present invention.

[**0061**] A modular embodiment of turbomachinery system **200** is seen by now referring to both **FIGS. 2 and 9**, where surface **210** is segmented at line **650** into a first surface **655** and a second surface **660**. In a first modular embodiment, first surface **655** provides a turbomachinery module **665** that includes a turbomachine **300**, a heat exchanger **230**, and a transition duct **240**, as discussed above in reference to **FIG. 2**. In a second modular embodiment, first surface **655** provides a turbomachinery module **665** that includes a turbomachine **500**, a cold plate **610**, and supply lines **615**, as discussed above in reference to **FIG. 9**. Second surface **660** represents circuit board **120**, in whole or in part, that includes an HPDD **220** to be cooled. In first turbomachinery module **665** of **FIG. 2**, heat exchanger **230** overhangs edge **650** of first surface **655**, thereby enabling heat exchanger **230** to be thermally coupled to HPDD **220** on second surface **660**. Alternatively, heat exchanger **230** may be a standalone component that is first thermally coupled to HPDD **220**, with first surface **655** then being fastened to second surface **660** such that transition duct **240** abuts heat exchanger **230** to function in the manner described above. In second turbomachinery module **665** of **FIG. 9**, cold plate **610** extends beyond edge **650** of first surface **655** via flexible supply lines **615**, thereby enabling cold plate **610** to be thermally coupled to HPDD **220** on second surface **660**. Alternatively, cold plate **610** may be first incorporated on or within second surface **660** and thermally coupled to HPDD **220**, with first surface **655** then being fastened to second surface **660** and supply lines **615** being connected together with suitable connectors, such as plugs or soldered connections, at segmentation line **650**. First and second surfaces **655**, **660** are fastened together by any suitable fastening means, such as but not limited to snap-fit connectors, plug-in connectors, bridging straps fastened to each surface **655**, **660** using screws or bolts, or adhesive means. Turbomachinery module **665** of either **FIG. 2** or **FIG. 9** is also suitable for 1U applications where dimension "D" is no greater than 1.75 inches.

[**0062**] Turbomachinery system **200** may include various control electronics (see wires **365** in **FIG. 4** and control system **580** in **FIG. 10**) for controlling turbomachines **300**, **500** that include such systems as, but not limited to, motors **310**, **325**, variable speed compressor **530**, and expansion valve **575**. The control electronics may be coupled to surface **210** by any suitable means, including but not limited to, surface mount technologies, adhesives, and thermal bonding (welding, soldering, hot gluing, thermoplastic melt). The control electronics may also include a soft starter for controlling the start up of turbomachinery system **200**, thereby preventing power surges or stalling during start up.

[**0063**] An embodiment of the cooling method employed by turbomachinery system **200** is depicted in **FIG. 12**, the cooling method being used to cool a HPDD **220**, such as an integrated circuit. Referring now to **FIG. 12**, method **700** begins at step **705** where high flux air (high mass flow rate) is drawn into a turbomachine **300**, **500** that uses a micro turbocompressor **340**, **530**. If the cooling method **700** uses air as the cooling medium, then path **710** is followed. If the cooling method **700** uses refrigerant as the cooling medium, then path **715** is followed.

[**0064**] Referring now to process path **710**, the high flux air is drawn **720** over a heat exchanger **230**, which is thermally coupled to HPDD **220**, thereby resulting in an increase in air temperature as the air passes over heat exchanger **230** and heat is transferred from HPDD **220** to heat exchanger **230** and then to the high flux air. The high flux air from heat exchanger **230** is then compressed **725** at compressor **340** and exhausted **725** at outlet **355** of turbomachine **300**.

[**0065**] Referring now to process path **715**, a refrigerant is first compressed **730** at turbocompressor **530**. Upon leaving compressor **530**, the refrigerant either flows **735** to condenser **515** or is optionally split **735** into two flow paths, one path flowing to condensor **515** and the other path flowing back to ejector **545**. The decision to include an optional flow path back to ejector **545** is determined by the system design and whether stalling of turbomachine **500** is a design consideration or not, the additional high pressure refrigerant from compressor **530** providing increased mass flow to compressor **530** via ejector **545**. At condenser **515** the refrigerant is condensed **740** by removing heat from the refrigerant, the heat being removed by a high flux air flow discussed below. After the refrigerant condenses, it flows through expander/evaporator **520** where it expands and evaporates **745** into a low temperature/low pressure gas, the low temperature/low pressure refrigerant gas being used to cool HPDD **220** via cold plate **610**, discussed above, or via high flux air flow discussed below. The low pressure refrigerant gas then returns **750** to compressor **530** to start the cycle over. The return path of the low pressure refrigerant gas to compressor **530** may be via motor **535** of turbomachine **505**, whereby motor **535** is cooled. If ejector **545** is employed, the low pressure refrigerant gas returns **750** to ejector **545** where it is mixed with high pressure refrigerant gas from compressor **530** before starting the cycle over. In parallel to the refrigeration cycle, second turbomachine **510** generates **755**, **760** a first high flux airflow across condenser **515**, and a second high flux airflow across expander/evaporator **520**. At condensor **515**, which may include a heat exchanger similar to heat exchanger **230**, heat is transferred **755** from the refrigerant to the first high flux airflow via the heat exchanger. The heated air is then exhausted **755** through outlet **620** of housing **560**. At expander/evaporator **520**, which includes coils **570** and may include heat exchanger **230**, heat is transferred **760** from the second high flux air flow to the refrigerant via coils **570** and heat exchanger **230**. The chilled air is then exhausted **760** to cool HPDD **220**, or other low power density devices on circuit board **120**, as depicted in **FIG. 9**.

[**0066**] Referring now to **FIGS. 13 and 14**, turbomachine **300** is depicted as a micro turbocompressor, which includes a motor **312**, a first stage micro compressor **342** having first stage compressor vanes **370** and disposed at one end of motor **312**, and a second micro compressor **343** having second stage compressor vanes **375** and disposed at the other end of motor **312**. First and second stage micro compressors **342**, **343** may be constructed integral with motor **312**, thereby providing a compact micro turbocompressor system. To enhance the cooling of motor **312**, cooling fins **380** may be machined on the outer surface of the stator of motor **312**, the cooling fins **380** extending from one end of motor **312** to the other end. First and second stage micro compressors **342**, **343** are arranged with appropriate diameters and air flow profiles to drive and draw, respectively, air between cooling fins **380**. Cooling fins **380** are typically nonlinear in geometry, and generally have a curvilinear profile for enhanced cooling. While **FIGS. 13 and 14** depict only one motor **312** and two micro compressor **342**, **343**, alternative arrangements may consist of a plurality of motors, as depicted in **FIG. 4**, and a plurality of micro compressors.

[**0067**] Referring now to FIGS. **15-17**, a high flux/low pressure drop heat exchanger **230** is depicted having various heat exchange enhancing features. Heat exchanger **230** includes a base **780** for thermally coupling heat exchanger **230** to HPDD **220**, as depicted in **FIG. 3**, and a plurality of parallel cooling fins **785** arranged perpendicular to base **780** for receiving air, depicted by arrows **790** and driven by turbomachine **300**, at one end **795** of heat exchanger **230**, the air being discharged at the opposite end **800** of heat exchanger **230**.

[**0068**] A first heat exchange enhancing feature is provided by a plurality of concavities **805** arranged on cooling fins **785**, or concavities **806** on base **780**, to enhance the effectiveness of cooling fins **785**, the term concavities **805**, **806** referring to depressions, indentations, dimples, pits or the like, which serve to produce flow vortices enhancing the thermal mixing and heat transfer. While concavities may in an embodiment consist of through holes, the effective heat transfer compared to dimples, for example, will be diminished. The shape of the concavities **805** is typically hemispherical or an inverted and truncated conical shape. In another embodiment, the shape of the concavities **805** is any sector of a full hemisphere. In some embodiments, the concavities **805** are disposed on an entirety or a portion of the abovementioned cooling fins **785**. The concavities **805** are formed on the abovementioned surfaces in a pattern that serves to enhance the heat transfer from cooling fins **785**. The manner of operation of concavities **805**, **806** is described generally as an increase in the interaction between the air stream and the cooling fins **785**, thereby resulting in improved heat transfer as compared to cooling fins **785** without concavities **805**. In addition, the thermal interaction between the air stream and each respective concavity is increased due to an increase in surface area with respect to a surface without concavities, the increased surface area being a result of the shape of each respective cavity. As such, the air flow interacts with the increased surface area thereby enhancing the removal of heat energy from the cooling fins **785**.

[**0069**] The depth "Y" (see **FIGS. 17, 18**) for a given one of the concavities **805** typically, but not necessarily, remains constant through the length "L" of a cooling fin **785**. The depth "Y" (see **FIGS. 17, 18**) is generally in the range between about 0.10 to about 0.50 times the concavity surface diameter "d". In addition, the depth "Y" of the concavities **805** is in the range between about 0.002 inches to about 0.125 inches. The center-to-center spacing "X" (see **FIGS. 17, 18**) of the concavities **805** is generally in the range between about 1.1 to about 2 times the surface diameter "d" of the concavities **805**. In one embodiment, the concavities **805** are typically formed by using a pulse electrochemical machining (PECM) process. In an alternative embodiment, the concavities **805** are typically formed by using an electro-discharge machining (EDM) process.

[**0070**] A second heat exchange enhancing feature is provided by localized cooling region **810** disposed on base **780**, which may be provided by discrete roughness elements adhered (using braze or adhesion for example) to base **780** that locally increase the surface area or may be provided using a mechanical process (etching, machining, pulsed electro-discharge machining, or masked electrodes for example) that increases the surface texture (i.e., roughness), for enhanced surface area heat transfer. Localized cooling regions **810** may be used at predefined locations on base **780** where base **780** abuts discrete circuit components that generate excessive heat and develop hot spots on base **780**. While only a few concavities **805** are depicted on cooling fins **785** in **FIG. 15**, and only one localized cooling region

810 is depicted on base **780** in **FIG. 15**, it will be appreciated that the illustration depicted in **FIG. 15** is exemplary only and that any number of concavities **805** in any arrangement or any number of localized cooling regions **810** at any location on base **780** fall within the scope of this invention.

[**0071**] A third heat exchange enhancing feature depicted in FIGS. **15-17** is a vortex air flow through and across heat exchanger **230**, which utilizes swirl flow techniques to more effectively transfer heat from base and cooling fins **780, 785** to ambient, the base being thermally coupled to HPDD **220**. **FIG. 15** depicts a vortex chamber **850**, to be discussed in more detail below in reference to **FIG. 19**, disposed between cooling fins **785** and thermally coupled to base **780**. **FIG. 16** depicts a plurality of vortex chambers **850** arranged parallel to each other and each thermally coupled to base **780**. In **FIG. 16**, the sides **852** of vortex chambers **850** and cooling fins **785** are integral with one another, the sides **852** acting as cooling fins **785**. The sides **852** of vortex chamber **850** may include concavities **805** to enhance heat transfer as discussed above. Air flow **790** enters vortex chamber **850** at a first end **854** and is discharged at a second end **856** of vortex chamber **850**. Vortex chamber **850** may be machined, cast or molded out of any material having suitable heat transfer properties, such as for example aluminum, zinc alloy, copper alloy or molded plastic having heat transfer additives.

[**0072**] Referring now to **FIG. 19**, an exploded isometric view of an embodiment of vortex chamber **850** is depicted having first and second ends **854, 856**, first and second sides **858, 860**, internal ribs **862, 864** on first and second sides **858, 860**, respectively, first and second edges **866, 868**, and concavities **805**, which are shown only on first side **858**, but it will be appreciated that concavities **805** may also be on second side **860**. Air flow **790** is received at first end **854** and discharged at second end **856**. Internal ribs **862** are arranged oblique, that is, diverging from a given straight line, to the impinging air flow **790** (x-direction **870**), and are depicted at an angle of positive-beta with respect to x-direction **870**. Internal ribs **864** are also arranged oblique to the x-direction **870**, but at an angle of negative-beta with respect to x-direction **870**. The included angle between ribs **862** and **864** is then two-times-beta (2 beta). It will be appreciated that while internal ribs **862, 864** are typically straight ribs at the positive and negative beta angles, respectively, curved ribs may also be employed where only a portion of internal ribs **862, 864** are arranged at the positive and negative beta angles, respectively. It will also be appreciated that internal ribs **862, 864** may not necessarily be at the same positive and negative beta angles, respectively. The primary function of ribs **862, 864** is to redirect the air flow at the turning regions **815** from one set of channels (between ribs **862, 864**) to the other, and in so doing produces enhanced heat transfer. To accomplish this, the geometry is arranged such that the channels between ribs **862, 864** overlap with open areas in the turning regions **815** (see **FIG. 20**).

[**0073**] Turning regions **815** serve to locally enhance the heat transfer from base **780** of heat exchanger **230** compared to conventional heat exchangers having no turning regions. It will be appreciated that the number of turning regions **815** for enhancing heat transfer is left to the artisan to determine based upon predetermined design requirements, for example, heat transfer rate and thermal gradient uniformity dependent on location of HPDD's **220** thermally coupled to base **780**. Also left to the artisan is the overall width and length of turning regions **815** at first edges **866** (and at second edges **868**, although not depicted) of first and second sides **858, 860**, as well as the channel dimensions and also the rib **862, 864** dimensions and shaping.

[**0074**] In the assembled state, first and second sides **858, 860** abut each other such that internal ribs **862, 864** are proximate each other. First and second sides **858, 860** may be coupled using known means, such as fasteners, mechanical or snap fit, or thermal or chemical bonding for example. Alternatively, positioning of first and second sides **858, 860** in heat exchanger **230** may be retained by offsets (not shown) on base **780**, cooling fins **785**, or both. First end **854** of first and second sides **858, 860** may include rounded edges **872** to reduce entry pressure losses, which reduces noise and improves air flow for less pressure drop. The ends of internal ribs **862, 864** may or may not be aligned, depending on the desired cooling characteristics of heat exchanger **230**.

[**0075**] As the air flow **790** impinges first end **854**, it is confronted with internal ribs **862, 864**, having opposite oblique angles. The air flow influenced by internal ribs **862** is directed in a positive-beta direction, while the air flow influenced by internal ribs **864** is directed in a negative-beta direction. The transition of air from first side **858** to second side **860**, and vice versa, causes the air flow within vortex chamber **850** to change direction, thereby creating a vortex flow.

[**0076**] Referring now to **FIG. 20**, an alternative embodiment of vortex chamber **874** is depicted having first edge **866** open while second edge **868** remains closed. In the assembled state, first edge **866** is arranged proximate base **780**, while second edge **868** faces away from base **780**. In the arrangement of **FIG. 20**, vortex air flow through vortex chamber **874** is permitted to impinge base **780**, thereby enhancing the heat transfer from base **780** to ambient. Vortex chamber **874** also depicts concavities **805** on second side **860**, as discussed above.

[**0077**] Vortex chambers **850, 874** depicted in **FIGS. 17 and 18** include internal ribs **862, 864** having positive and negative beta angles, respectively, which may vary from a value equal to or greater than about 15 degrees to a value equal to or less than about 60 degrees, as best seen by now referring to the graph **900** depicted in **FIG. 21**. A beta angle that ranges from about 15 degrees to about 60 degrees will result in first side internal ribs **862** being disposed at an angle in the range between about 30 degrees and about 120 degrees with respect to the second side internal ribs **864**. The beta angle may vary somewhat from the stated dimensions to accommodate tolerances without detracting from the scope of the invention. Graph **900** illustrates the relationship between the average heat transfer coefficient (HTC) enhancement **905** and the vortex angle (beta) **910**, for narrow **915** and wide **920** vortex channels and for different Reynolds numbers (Re equal to 33,000 (numeral **925**) and 77,000 (numeral **930**)). As can be seen, an increase in the beta angle **910** generally results in an increase in average HTC enhancement **905**. However, in a wide **920** vortex chamber at Re=33,000 (**925**), turbulent flow was experienced. In narrow **915** vortex chambers or where **Re=77,000 (930)**, improved average HTC enhancement **905** was consistently experienced as the beta angle **910** increased. Narrow **915** vortex chambers are preferred over wide **920** vortex chambers since the narrow chambers adapt well to heat sink geometry and size where many cooling fins are employed. The use of vortex chambers (narrow and wide) having a vortex beta angle of equal to or greater than about 30 degrees and equal to or less than about 50 degrees was found to enhance the average HTC between 2-3.5 times as compared to the same heat exchanger without a vortex chamber. A preferred beta angle is about 45 degrees. A beta angle of greater than about 45 degrees, such as about 50 degrees for example, may improve the average heat transfer coefficient, as seen in **FIG. 21**, but the vortex chamber **850** will experience a greater pressure loss, thereby establishing a design tradeoff for the artisan. For example, at large air flow rates on the order of 720 lbs/hr, an embodiment of heat exchanger **230** with a vortex chamber **850** may produce a 10% pressure drop. Thus, in order to offset the pressure drop due to a flow rate of 720 lbs/hr, it may be necessary to reduce the beta angle in vortex chamber **850**.

[**0078**] An embodiment of vortex chamber **850** provides each side **858, 860** with at least three individual angled flow passages **876** between ribs **862, 864**, but could have as many as ten passages, all of which will turn at the edges **866, 868** to redirect the flow to the other wall. It will be appreciated that while turning regions **815** depicted in **FIG. 20** are shown only at first edge **866**, turning regions **815** may also be applied at second edge **868**. The preferred height-to-width ("H" and "W" in **FIG. 16**) aspect ratio of each individual passage **876** is 1.0, but the range of such aspect ratios can be from 0.5 to 2. The size of each passage **876** is therefore commensurate with the number of channels in vortex chamber **850**, the number of cooling fins **785** (separating walls of the vortex chambers), and the overall dimension of the entire heat exchanger **230**. By way of example and not limitation, a heat exchanger **230** having 5 cooling plates **785** could house 4 vortex chambers **850**, each chamber being composed of two opposing side walls **858, 860** having flow passages **876**. If the distance between cooling fins **785** is 1-inch, then each flow passage **876** would be about 0.5 inches in height. If the overall heat exchanger **230** is 1.75 inches high and the number of flow passages **876** is 4, then each passage width would be about 0.4 inches at its inlet plane. If the passage angle (beta) is about 30 degrees, then the passage width seen by the flow would be about 0.35 inches, making the passage aspect ratio about 1.4.

[**0079**] Referring now to **FIGS. 22 and 23**, an alternative embodiment to transition ducts **240, 415** is depicted. While the alternative embodiment transition duct **950** more closely resembles transition duct **240**, it will be appreciated that the same principles hereinafter disclosed may also apply to transition duct **415**.

[**0080**] Transition duct **950** includes a duct housing **952** having a first end **954** with a first flow area **956** for receiving driven air from heat exchanger **230** having a plurality of cooling fins **785**, and a second end **958** with a second flow area **960** for discharging the driven air to turbomachine **300** (shown in FIGS. **2-4**). An example of a first flow area **956** is 2.75 square-inches, the first flow area **956** being less than the projected area of 4 square-inches to account for the thickness of cooling fins **785**. An example of a second flow area **960** is 1 square-inch. Other flow areas may be employed while still being suitable for 1U applications. Internal to duct housing **952** is a cavity **962** that transitions from a geometry having first flow area **956** to a geometry having second flow area **960**. Within cavity **962** is a plurality of flow control fins **964** for managing the change in flow area from first flow area **956** to second flow area **960**. Flow control fins **964** have first ends **966** arranged proximate cooling fins **956** of heat exchanger **230**, and second ends **968** extending toward second end **958** of transition duct **950**. While some flow control fins **964** may extend all the way to second end **958**, it is preferable that at least some flow control fins **964** do not extend all the way to second end **958**, which produces less flow constriction and less pressure drop across transition duct **950**. The number of flow control fins **964** in transition duct **950** may be equal to or less than the number of cooling fins **785** on heat exchanger **230**, the actual number of flow control fins **964** being determined by the desired heat transfer characteristics and the desired pressure drop characteristics of turbomachinery system **200**.

[**0081**] As the high flux air flow **790** is received at first end **954** of transition duct **950** it is segmented into separate flow channels by first ends **966** of flow control fins **964**. The segmented air flow is then funneled from first flow area **956** toward second flow area **960** where it is joined by adjacent air flows in adjacent flow channels at second ends **968** of flow control fins **964**. The recombined air flow is then discharged at second end **958** of transition duct **950** for entry into turbomachine **300**.

[**0082**] In an exemplary embodiment, an air mass flow rate driven by turbomachine **300** through heat exchanger **230** is on the order of 150-360 pounds-per-hour (lbs/hr), and a flow area through heat exchanger **230** is 2.75 square inches (sq-in) (discussed above), thereby resulting in a mass-flow-to-flow-area ratio of between 54.5 lbs/hr/sq-in and 130.9 lbs/hr/sq-in. In another exemplary embodiment, the mass-flow-to-area ratio is between 54.5 lbs/hr/sq-in (as discussed above) and 90.9 lbs/hr/sq-in (250 lbs/hr air mass flow rate through 2.75 sq-in flow area). The exemplary mass-flow-to-flow-area ratios may be applicable with any of the aforementioned transition ducts **240, 415, 950**. It will be appreciated that the flow area of 2.75 sq-in discussed above is for example only and is not intended to be limiting in any manner, as is best seen by now referring to **FIG. 24**. Graph **820** in **FIG. 24** illustrates the mass-flow-to-flow-area ratio (mass flow-area ratio) **822**, in units of lbs/sec/sq-in, as a function of air mass flow **824**, in units of lbs/sec, for different flow areas **826**, in units of sq-in (in^2), ranging from 1 sq-in to 4 sq-in **830-836**. As shown, the mass-flow-to-flow-area ratio ranges from 0.005 lbs/sec/sq-in (18 lbs/hr/sq-in) at a flow area of 4 sq-in to 0.04 lbs/sec/sq-in (144 lbs/hr/sq-in) at a flow area of 1 sq-in. Thus, the range of mass-flow-to-flow-area ratios applicable to an embodiment of the invention is illustrated to be from 18-144 lbs/hr/sq-in.

[**0083**] In an embodiment of the invention, the inner surfaces of transition ducts **240, 415, 950** and exhaust ducts, such as outlet **355** and entrainment nozzle **435**, or the exposed surfaces of cooling fins **785**, may be treated with an acoustic absorbing material to provide an acoustical damping characteristic, as depicted in **FIG. 25**.

[**0084**] While **FIG. 25** depicts a portion of a flow duct **980**, it will be appreciated that the teachings relating to **FIG. 25** are equally applicable to other surfaces, such as the cooling fins of a heat exchanger for example, where acoustical damping may also be desired. Flow duct **980** is depicted having a wall thickness "dr", which is shown as a wall section **985** in the cross-sectional view of **FIG. 26**. Wall section **985** is composed of an inner wall **990**, a bulk absorber **992**, and an outer wall **994**, bulk absorber **992** being sandwiched between inner and outer walls **990**, **994**. Outer wall **994** is structural in function and is typically composed of a solid material such as metal, inner wall **990** is a porous material such as a ceramic or a sintered metal having a porosity of about 20% to about 60% for example, and bulk absorber **992** is composed of an acoustic damping material such as poly-paraphenylene terephthalamide (Kevlar® from DuPont) or fiberglass for example. Alternatively, bulk absorber **992** may be composed of a honeycomb structure having a cell size tuned for absorbing specific frequencies. As the air flow **982**, which may include a turbulent component, passes through flow duct **980** and across the surface of inner wall **990**, it generates acoustical waves that impinge upon the porous surface of inner wall **990**. While a portion of these acoustical waves may reflected back, a portion of them pass through the porous surface of inner wall **990** and are absorbed by bulk absorber **992**, thereby resulting in reduced operational noise of turbomachinery system **200**.

[**0085**] While the invention has been described with reference to exemplary embodiments, it will be understood by those skilled in the art that various changes may be made and equivalents may be substituted for elements thereof without departing from the scope of the invention. In addition, many modifications may be made to adapt a particular situation or material to the teachings of the invention without departing from the essential scope thereof. Therefore, it is intended that the invention not be limited to the particular embodiment disclosed as the best mode contemplated for carrying out this invention, but that the invention will include all embodiments falling within the scope of the appended claims. Moreover, the use of the terms first, second, etc. do not denote any order or importance, but rather the terms first, second, etc. are used to distinguish one element from another.

1. A turbomachinery system for cooling a high power density device, comprising:

a turbomachine configured to deliver a high flux cooling medium, said turbomachine having a motor and a compressor driven by said motor; and

at least one high power density device arranged in fluid communication with said turbomachine.

2. The turbomachinery system of claim 1, wherein the high flux cooling medium is air.

3. The turbomachinery system of claim 2, wherein said turbomachine has an air mass-flow-to-flow-area ratio of equal to or greater than about 18 lbs/hr/sq-in and equal to or less than about 144 lbs/hr/sq-in.

4. The turbomachinery system of claim 3, wherein said turbomachine has an air mass-flow-to-flow-area ratio of equal to or greater than about 54.5 lbs/hr/sq-in and equal to or less than about 90.9 lbs/hr/sq-in.

5. The turbomachinery system of claim 1, wherein the high flux cooling medium is a refrigerant.

6. The turbomachinery system of claim 2, wherein said turbomachine further comprises:

a housing containing at least one of said motor and said compressor, said housing providing a passage for air flow;

an inlet at a first end of said housing for accepting air flow; and

an outlet at a second end of said housing for discharging air flow.

7. The turbomachinery system of claim 6, further comprising:

a heat exchanger thermally coupled to said at least one high power density device; and

said turbomachine being downstream from said heat exchanger.

8. The turbomachinery system of claim 7, wherein said heat exchanger comprises:

a base for thermally coupling the heat exchanger to the high power density device;

a plurality of parallel cooling fins arranged perpendicular to said base for receiving driven air at one end and discharging the driven air at the opposite end; and

a plurality of vortex chambers arranged between said cooling fins for creating a vortex like air flow as the air is driven between said cooling fins and through said vortex chamber, said plurality of vortex chambers having side walls integral with said plurality of parallel cooling fins.

9. The turbomachinery system of claim 7, further comprising:

a transition duct arranged intermediate said heat exchanger and said inlet for funneling the air flow from said heat exchanger to said turbomachine.

10. The turbomachinery system of claim 9, wherein said transition duct comprises:

a duct housing having a first end with a first flow area for receiving driven air from said heat exchanger having a plurality of cooling fins and a second end with a second flow area for discharging driven air to said turbomachine, said duct housing defining an internal cavity that transitions from said first flow area to said second flow area; and

a plurality of flow control fins within said internal cavity for managing the change in flow area from said first flow area to said second flow area.

11. The turbomachinery system of claim 9, further comprising:

an expansion turbine driven by said motor and arranged proximate said outlet for expanding the high flux air flow as it is discharged from said turbomachine.

12. The turbomachinery system of claim 9, further comprising a second turbomachine arranged for parallel air flow with said turbomachine, wherein said transition duct is arranged for funneling the air flow from said heat exchanger to said turbomachine and said second turbomachine in parallel.

13. The turbomachinery system of claim 12, wherein the air flow from said heat exchanger to said turbomachine and said second turbomachine provides equivalent air flows through said turbomachine and said second turbomachine.

14. The turbomachinery system of claim 9, wherein said at least one high power density device includes an integrated circuit.

15. The turbomachinery system of claim 6, wherein said motor comprises magnetic bearings.

16. The turbomachinery system of claim 1, wherein said compressor comprises at least one of an axial compressor and a centrifugal compressor.

17. The turbomachinery system of claim 16, wherein said compressor comprises at least one of a single wheel compressor and a multi-wheel compressor.

18. The turbomachinery system of claim 1, wherein said turbomachine has an overall dimension of no greater than 1.75 inches.

19. The turbomachinery system of claim 1, wherein said turbomachine has an overall dimension suitable for 1U applications.

20. The turbomachinery system of claim 9, wherein at least one of said heat exchanger and said transition duct has an acoustic damping structure comprising at least one material having an acoustic damping characteristic.

21. The turbomachinery system of claim 20, wherein said acoustic damping structure comprises at least one of a porous surface and an acoustic damping bulk absorber.

22. The turbomachinery system of claim 21, wherein said porous surface comprises at least one of a ceramic and a sintered metal, and said acoustic damping bulk absorber comprises at least one of a polymeric and a fiberglass material.

23. The turbomachinery system of claim 6, wherein said outlet further comprises:

a nozzle outlet having secondary air ducts for entraining ambient air for providing a secondary cooling air flow.

24. The turbomachinery system of claim 1, further comprising:

a soft starter for controlling start up of said turbomachine for preventing power surges at said turbomachine during start up.

25. The turbomachinery system of claim 7, further comprising:

an air particle ionizer arranged upstream of said heat exchanger for ionizing the air particles prior to entering said heat exchanger and said turbomachine for preventing particulate fouling of the turbomachinery system; and

an air particle de-ionizer arranged downstream of said outlet for neutralizing the electrical charge of the discharging air particles.

26. The turbomachinery system of claim 5, further comprising:

a closed-loop refrigeration circuit configured to cool said at least one high power density device, the refrigerant flowing in said closed-loop refrigeration circuit, said closed-loop refrigeration circuit comprising:

said compressor;

a condensor;

an expander;

an evaporator coupled to said at least one high power density device; and

a turbine driven by said motor.

27. The turbomachinery system of claim 26, further comprising:

an ejector arranged for receiving and mixing the refrigerant at high pressure from said compressor and the refrigerant at low pressure from said turbine, the mixed refrigerant being delivered to said compressor for increasing the mass flow through said compressor.

28. The turbomachinery system of claim 27, further comprising:

a second turbomachine configured to deliver high flux air across said evaporator to deliver cooled air to said at least one high power density device and to deliver high flux air across said condenser to transport heated air to ambient, said second turbomachine including a second motor, a fan driven by said second motor and a housing for directing the flow path of the high flux air.

29. The turbomachinery system of claim 26, wherein:

the refrigerant flows from said evaporator to said turbine, then to said motor, and then to said compressor.

30. The turbomachinery system of claim 26, wherein said expander comprises and expansion valve.

31. The turbomachinery system of claim 30, wherein said expansion valve comprises and active expansion valve and said turbomachinery system further comprises:

an expansion valve control system for controlling the pressure drop of the refrigerant across said active expansion valve.

32. The turbomachinery system of claim 29, wherein said compressor is a variable speed compressor.

33. A turbomachine for delivering a high flux cooling medium to a device, comprising:

a motor;

a compressor driven by said motor for compressing the high flux cooling medium;

a housing containing at least one of said motor and said compressor, said housing providing a passage for the high flux cooling medium;

an inlet at a first end of said housing for accepting the high flux cooling medium; and

an outlet at a second end of said housing for the discharging high flux cooling medium.

34. The turbomachine of claim 33, wherein the cooling medium is at least one of air and a refrigerant.

35. The turbomachine of claim 33, wherein said turbomachine has an overall dimension of no greater than 1.75 inches.

36. The turbomachine of claim 33, wherein said turbomachine has an overall dimension suitable for 1U applications.

37. The turbomachine of claim 33, wherein said motor comprises magnetic bearings.

38. The turbomachine of claim 33, wherein said compressor comprises at least one of an axial compressor and a centrifugal compressor.

39. The turbomachine of claim 38, wherein said compressor comprises at least one of a single wheel compressor and a multi-wheel compressor.

40. An assembly comprising:

a mounting fixture; and

a plurality of turbomachinery systems of claim 9 mounted adjacent one another within said mounting fixture.

41. An assembly comprising:

a mounting fixture; and

a plurality of turbomachinery systems of claim 26 mounted adjacent one another within said mounting fixture.

42. A turbomachinery module for attaching to a surface and used for cooling high power density devices, comprising:

a module surface;

a turbomachine configured to deliver high flux air, said turbomachine including a motor, a compressor driven by said motor for compressing the high flux air, and a housing;

said housing containing at least one of said motor and said compressor and providing a passage for air flow;

said housing including an inlet at a first end for accepting air flow and an outlet at a second end for discharging air flow;

a heat exchanger for thermally coupling to and cooling a high power density device, said turbomachine being downstream from said heat exchanger; and

a transition duct arranged intermediate said heat exchanger and said turbomachine for funneling the air flow from said heat exchanger to said turbomachine.

43. The turbomachinery module of claim 42, wherein said turbomachinery module has an overall dimension of no greater than 1.75 inches.

44. The turbomachinery module of claim 42, wherein said turbomachinery module has an overall dimension suitable for 1U applications.

45. The turbomachinery module of claim 42, wherein said motor comprises magnetic bearings.

46. The turbomachinery module of claim 42, wherein said compressor comprises at least one of an axial compressor and a centrifugal compressor.

47. The turbomachinery module of claim 46, wherein said compressor comprises at least one of a single wheel compressor and a multi-wheel compressor.

48. A method for cooling a high power density device, comprising:

drawing air over a heat exchanger using a turbomachine, said heat exchanger being thermally coupled to a high power density device resulting in an increase in air temperature as the air passes over the heat exchanger;

compressing the air at a turbocompressor of the turbomachine having an overall dimension of no greater than 1.75 inches; and

exhausting the heated air to ambient.

49. The method of claim 48, wherein:

said drawing air comprises drawing air from at least one of an external ambient external to the structure housing the high power density device and an internal ambient internal to the structure housing the high power density device; and

said exhausting the heated air comprises exhausting the heated air to at least one of an external ambient external to the structure housing the high power density device and an internal ambient internal to the structure housing the high power density device.

50. A method for delivering a high flux cooling medium to a high power density device, said method comprising:

compressing a refrigerant at a turbocompressor having an overall dimension of no greater than 1.75 inches;

condensing the refrigerant by removing heat from the refrigerant;

expanding and evaporating the refrigerant to create a cold surface for cooling the high power density device; and

returning the expanded refrigerant to the turbocompressor to repeat the closed-loop cycle.

51. The method of claim 50, further comprising a method for delivering low temperature air to cool a high temperature device, said method comprising:

generating a first airflow across the condensor using a turbofan;

transferring heat from the refrigerant to the first airflow using a heat exchanger;

exhausting the heated first airflow to ambient;

generating a second airflow across the cold surface of the evaporator using the turbofan;

transferring heat from the second airflow to the evaporator using a heat exchanger; and

exhausting the chilled second airflow to cool the high temperature device.

52. The method of claim 51, further comprising:

directing a portion of the refrigerant from the compressor to the condensor and another portion from the compressor to an ejector located upstream to the compressor;

mixing at the ejector high pressure refrigerant from the compressor with low pressure refrigerant from the evaporator; and

delivering the mixed refrigerant to the compressor.

53. The method of claim 50, wherein said returning the expanded refrigerant to the turbocompressor further comprises:

returning the expanded refrigerant via a motor for cooling the motor.

54. A micro turbocompressor for delivering a high flux cooling medium to a device, comprising:

a motor;

a first stage micro compressor disposed at one end of said motor; and

a second stage micro compressor disposed at the other end of said motor.

55. The micro turbocompressor of claim 54, wherein said first and second stage micro compressors are integral with said motor.

56. The micro turbocompressor of claim 54, further comprising:

cooling fins disposed on the outer surface of the stator of said motor, said cooling fins extending from one end of said motor to the other end of said motor, said first stage micro compressor arranged to drive air between said cooling fins, and said second stage micro compressor arranged to draw air between said cooling fins.

57. The micro turbocompressor of claim 56, wherein said cooling fins extend from one end of said motor to the other end of said motor in a nonlinear arrangement.

58. A micro turbocompressor for delivering a high flux cooling medium to a device, comprising:

at least one motor; and

a plurality of micro compressors disposed at opposite ends of said at least one motor, said plurality of micro compressors arranged to drive air across the outer surface of at least one stator of said at least one motor.

59. The micro turbocompressor of claim 58, wherein said at least one motor further comprises:

cooling fins disposed on the outer surface of at least one stator of said at least one motor, said cooling fins extending from one end of said at least one motor to the other end of said at least one motor in a nonlinear arrangement.

60. A heat exchanger for cooling a high power density device, comprising:

a base for thermally coupling the heat exchanger to the high power density device; and

a plurality of parallel cooling fins arranged perpendicular to said base for receiving driven air at one end and discharging the driven air at the opposite end, said cooling fins having a plurality of concavities.

61. A heat exchanger for cooling a high power density device, comprising:

a base for thermally coupling the heat exchanger to the high power density device;

a plurality of parallel cooling fins arranged perpendicular to said base for receiving driven air at one end and discharging the driven air at the opposite end; and

a localized cooling region at said base and between said cooling fins for providing a local region of increased surface area for enhanced heat transfer.

62. A heat exchanger for cooling a high power density device, comprising:

a base for thermally coupling the heat exchanger to the high power density device;

a plurality of parallel cooling fins arranged perpendicular to said base for receiving driven air at one end and discharging the driven air at the opposite end; and

a vortex chamber arranged between said cooling fins for creating a vortex like air flow as the air is driven between said cooling fins and through said vortex chamber.

63. The heat exchanger of claim 62, wherein said vortex chamber comprises:

first and second sides having first and second ends for receiving and discharging air flow;

said first side having internal ribs arranged oblique to the air flow, a portion of said first side internal ribs being at an angle of positive-beta with respect to the direction of air flow; and

said second side having internal ribs arranged oblique to the air flow, a portion of said second side internal ribs being at an angle of negative-beta with respect to the direction of air flow.

64. The heat exchanger of claim 63, wherein said first side internal ribs and said second side internal ribs are arranged diagonal to the air flow.

65. The heat exchanger of claim 63, wherein the first ends of said first and second sides have rounded edges to reduce entry pressure losses at said vortex chamber.

66. The heat exchanger of claim 63, wherein said first and second sides further comprise first and second edges, said first and second edges being closed and said first edge being thermally coupled to said base.

67. The heat exchanger of claim 63, wherein said first and second sides further comprise first and second edges, said first edges being open, second edges being closed, and said first edges arranged proximate to said base.

68. The heat exchanger of claim 63, wherein said first and second sides are integral with said plurality of parallel cooling fins and further comprise concavities at their side walls.

69. The heat exchanger of claim 63, wherein said first side internal ribs are disposed at an angle of equal to or greater than about 30 degrees and equal to or less than about 120 degrees with respect to said second side internal ribs.

70. The heat exchanger of claim 63, wherein said beta angle is equal to or greater than about 20 and equal to or less than about 60 degrees.

71. The heat exchanger of claim 70, wherein said beta angle is equal to or greater than about 40 and equal to or less than about 60 degrees.

72. The heat exchanger of claim 71, wherein said beta angle is about 45 degrees.

73. A heat exchanger for cooling a high power density device, comprising:

a base for thermally coupling the heat exchanger to the high power density device;

a plurality of parallel cooling fins arranged perpendicular to said base for receiving driven air at one end and discharging the driven air at the opposite end;

a plurality of vortex chambers arranged between said plurality of parallel cooling fins for receiving the driven air at one end and discharging the driven air at the opposite end, said plurality of vortex chambers configured to create a vortex like air flow as the air is driven from said one end to said opposite end;

said plurality of vortex chambers having a plurality of first sides, said plurality of first sides having internal ribs arranged oblique to the air flow, a portion of the internal ribs of said plurality of first sides being at an angle of positive-beta with respect to the direction of air flow;

said plurality of vortex chambers having a plurality of second sides, said plurality of second sides having internal ribs arranged oblique to the air flow, a portion of the internal ribs of said plurality of second sides being at an angle of negative-beta with respect to the direction of air flow;

said plurality of first and second sides further comprising a plurality of first and second edges, said plurality of first edges being open, said plurality of second edges being closed, and said plurality of first edges arranged proximate to said base; and

a localized cooling region at said base proximate said plurality of first edges for providing a local region of increased surface area for enhanced heat transfer.

74. The heat exchanger of claim 73, wherein:

said plurality of parallel cooling fins are integral with said plurality of first and second sides of said plurality of vortex chambers.

75. A method for enhancing the heat transfer characteristic of a heat exchanger for cooling a high power density device, comprising:

receiving driven air at a first end of a plurality of cooling fins of the heat exchanger;

driving the air across the plurality of cooling fins for transferring heat from the high power density device to ambient;

disturbing the air flow as it is driven across the plurality of cooling fins by employing at least one of a plurality of concavities in the plurality of cooling fins, a plurality of localized cooling regions at the base of the heat exchanger and between the plurality of cooling fins, and a plurality of vortex chambers between the plurality of cooling fins for creating a vortex flow between the plurality of cooling fins; and

discharging the heated air at a second end of the plurality of cooling fins.

76. The method of claim 75, wherein said creating a vortex flow further comprises:

creating a vortex flow between the plurality of cooling fins wherein the plurality of cooling fins are integral with the plurality of vortex chambers.

77. The method of claim 75, further comprising:

enhancing the heat transfer characteristic of the heat exchanger by at least two-times in comparison to the same heat exchanger absent the plurality of vortex chambers between the plurality of cooling fins for creating a vortex flow between the plurality of cooling fins.

78. A transition duct for a turbomachinery system, comprising:

a duct housing having a first end with a first flow area for receiving driven air from a heat exchanger having a plurality of cooling fins and a second end with a second flow area for discharging driven air to a turbomachine, said duct housing defining an internal cavity that transitions from said first flow area to said second flow area; and

a plurality of flow control fins within said internal cavity configured to manage the change in flow area from said first flow area to said second flow area.

79. The transition duct of claim 78, wherein said plurality of flow control fins further comprises:

a plurality of first ends arranged proximate the plurality of cooling fins of the heat exchanger; and

a plurality of second ends extending toward said second end of said transition duct, wherein at least one of said plurality of second ends of said flow control fins does not extend to said second end of said transition duct.

80. The transition duct of claim 79, wherein said plurality of flow control fins are equal in number to the number of cooling fins of the heat exchanger.

81. A method for transitioning high flux air from a region having a first flow area to a region having a second flow area wherein said first flow area is larger than said second flow area, comprising:

receiving the high flux air at a first end of a transition duct having the first flow area;

segmenting the high flux air flow into separate air flow channels between a plurality of flow control fins;

funneling the high flux air between the flow control fins from the first flow area toward the second flow area; and

discharging the high flux air at a second end of the transition duct having the second flow area.

82. A turbomachinery system for cooling a high power density device, comprising:

a surface connectible to a high power density device; and

a turbomachine in fluid communication with said surface and configured to deliver a high flux cooling medium, said turbomachine having a motor and a compressor driven by said motor.

83. The turbomachinery system of claim 82, wherein said surface comprises at least one of a heat exchanger and a cold plate.

84. The turbomachinery system of claim 83, further comprising a transition duct in fluid communication with said turbomachine and said heat exchanger.

85. The turbomachinery system of claim 84, further wherein said turbomachine is configured to deliver at least one of air and refrigerant.

86. The turbomachinery system of claim 85, wherein said turbomachine has an overall dimension of no greater than 1.75 inches.

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