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(54) **COMPRESSOR COOLING FOR TURBINE ENGINES**

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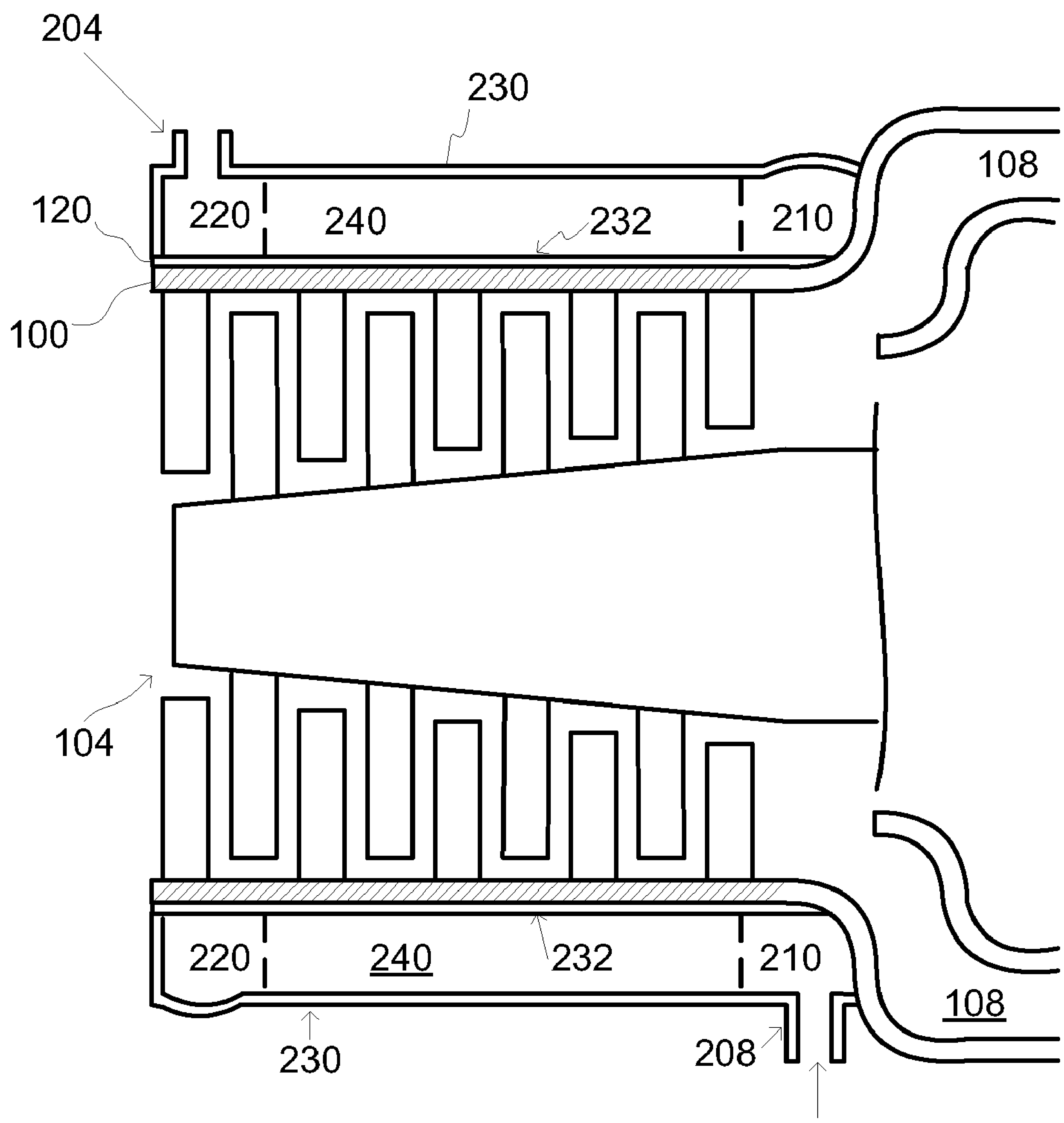
(57) **ABSTRACT**

Systems, apparatuses and methods (“utilities”) for use in “internally” cooling the compressor of a gas turbine engine so as to approximate isothermal compression and thereby increase the power and/or efficiency of the engine. In one arrangement, a “cooling jacket” or heat exchanger having coolant circulating or passing therethrough may be mounted around an outer surface of the compressor to absorb heat or thermal energy generated from the compressor. In another arrangement, the stator blades of the compressor may include passages through which a coolant may be circulated or passed to absorb heat from air passing through the compressor.



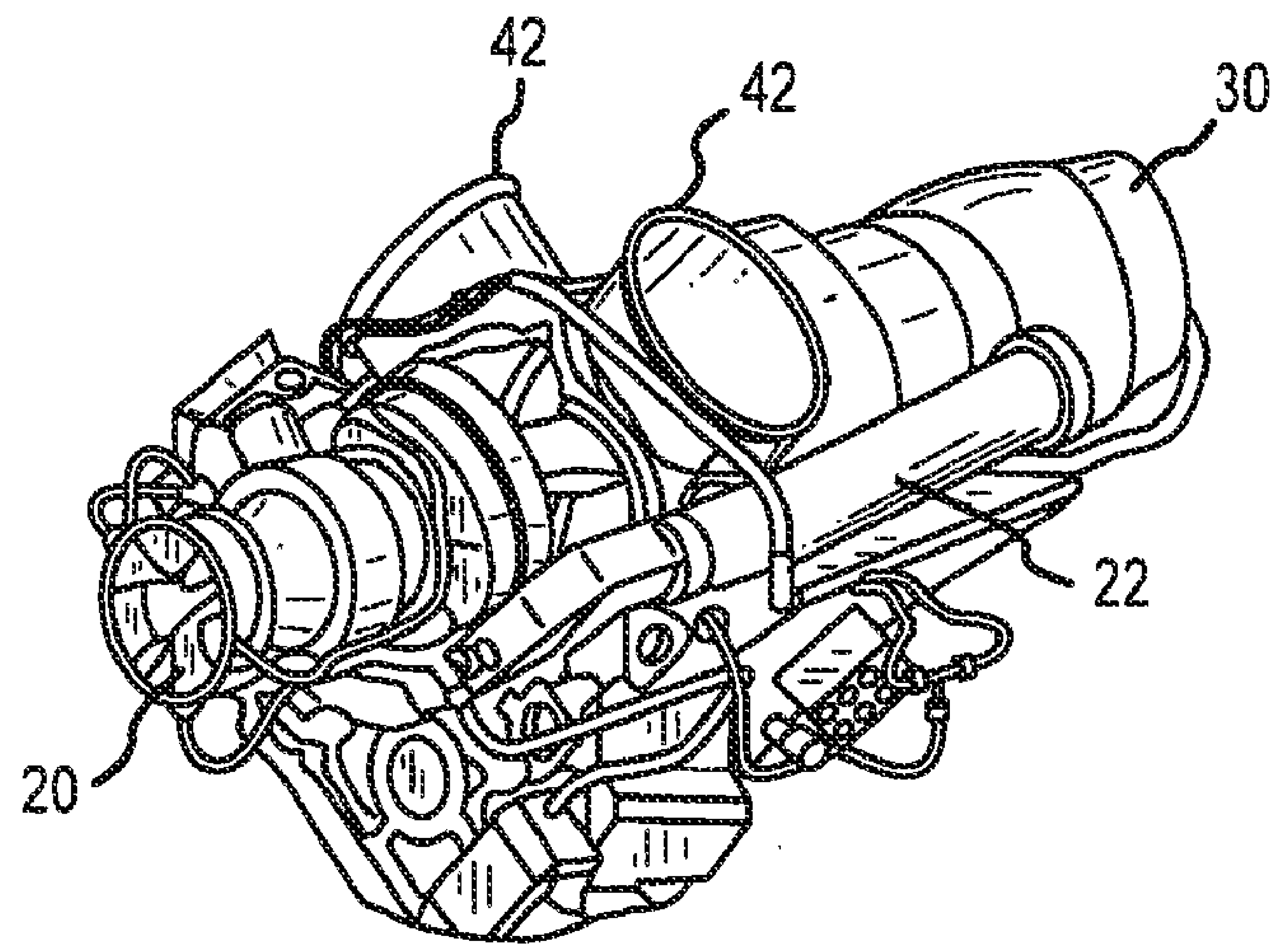


FIG.1

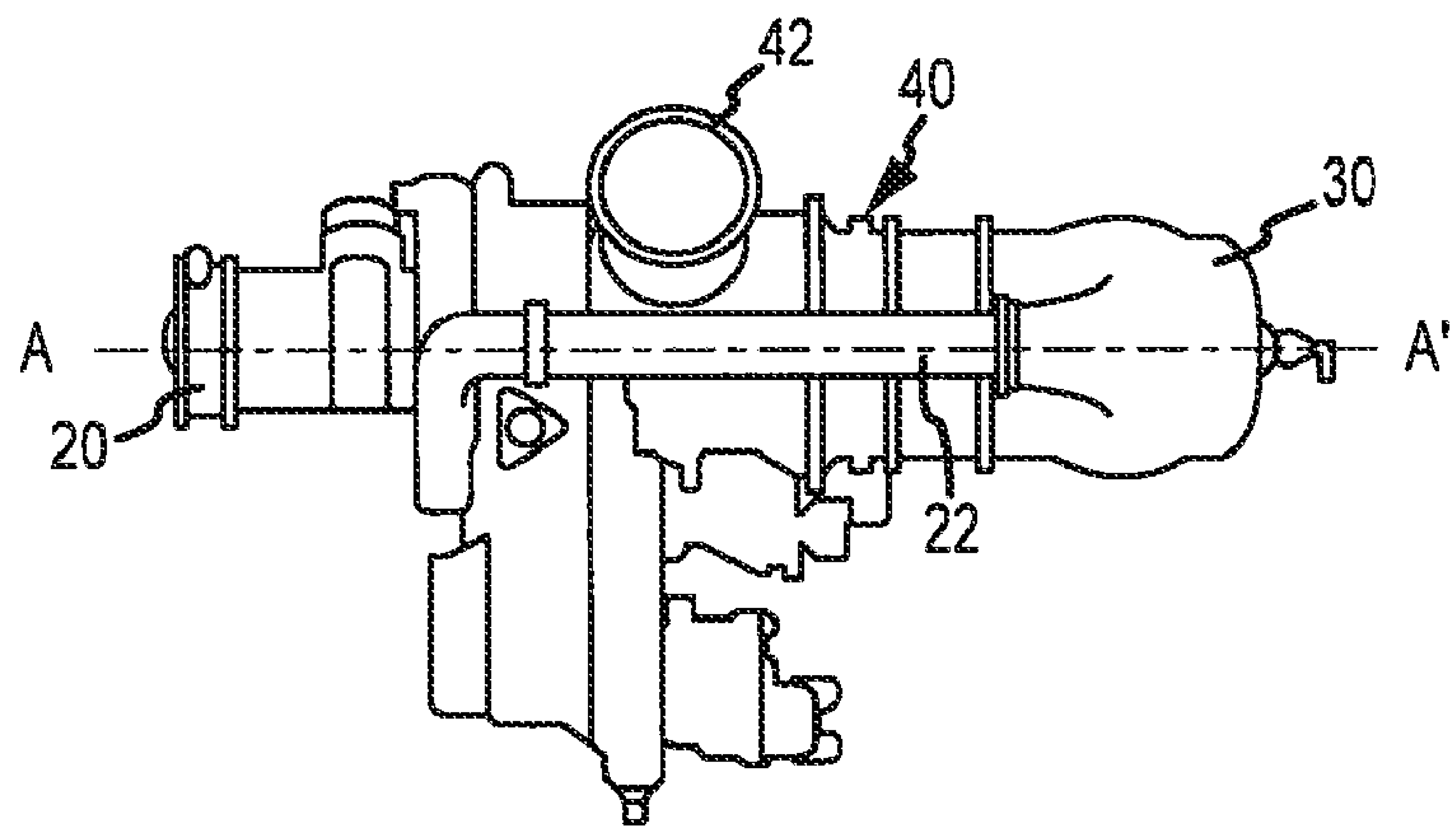


FIG.2

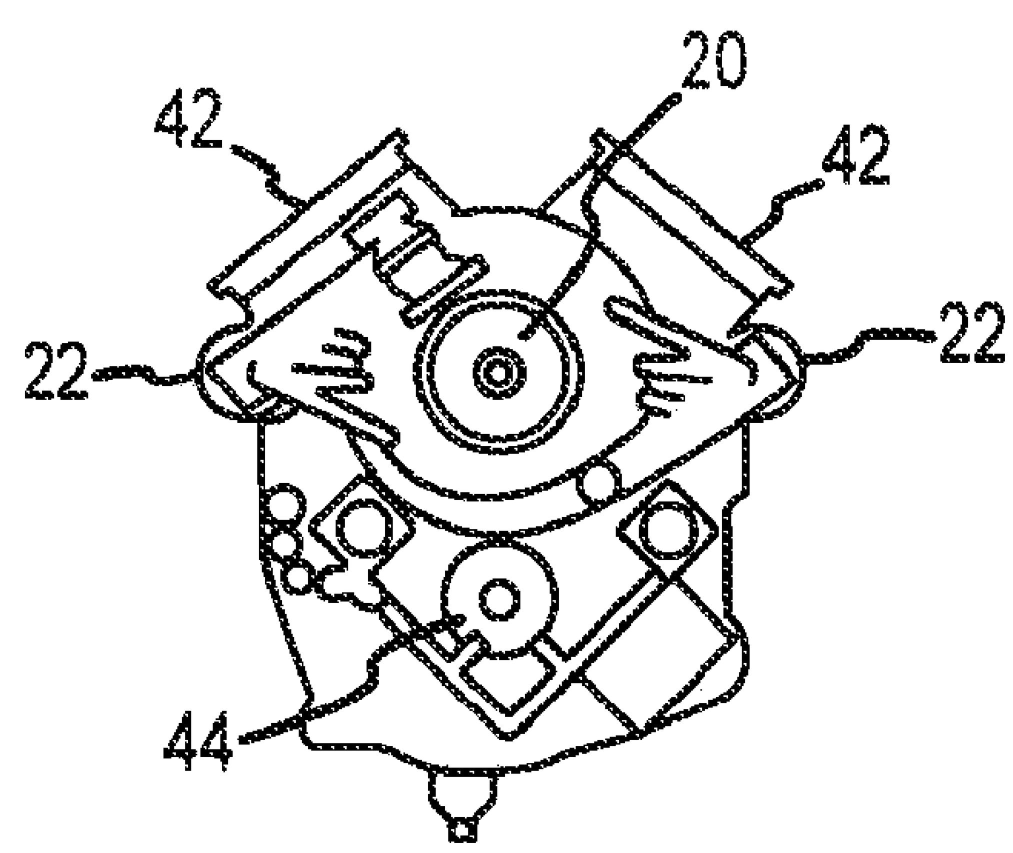
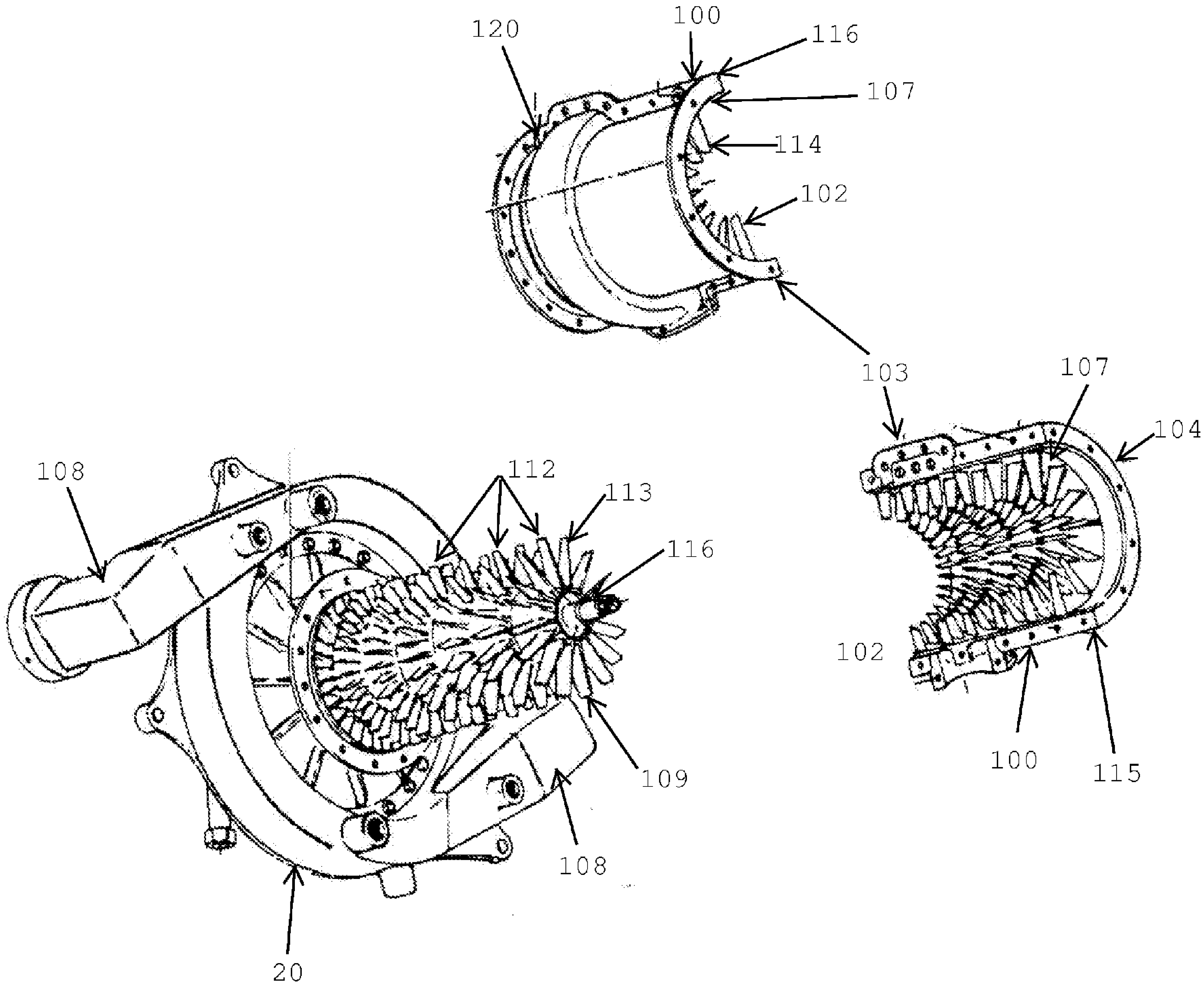


FIG.3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115
108
20

Figure 4

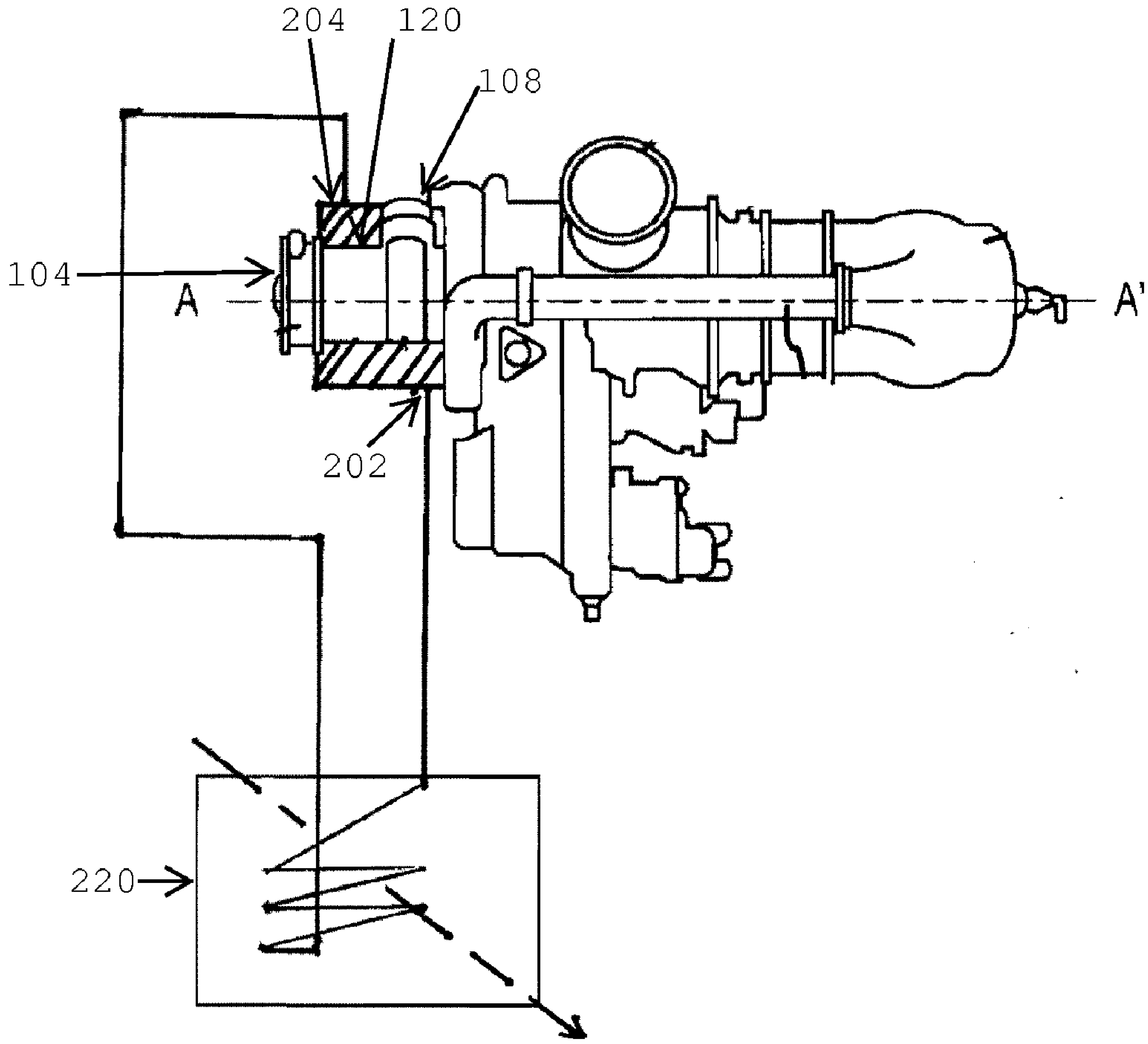
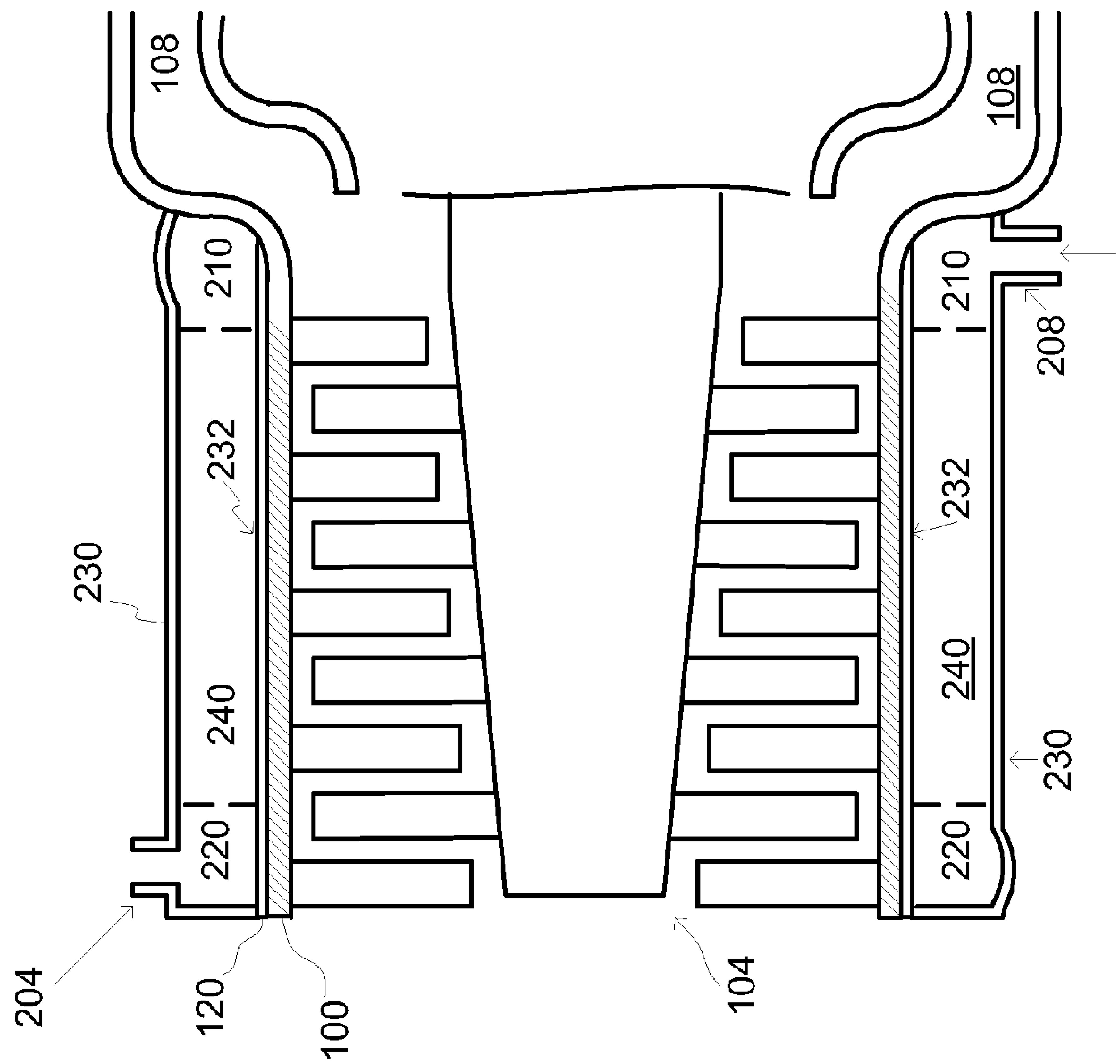


Figure 5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208
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230
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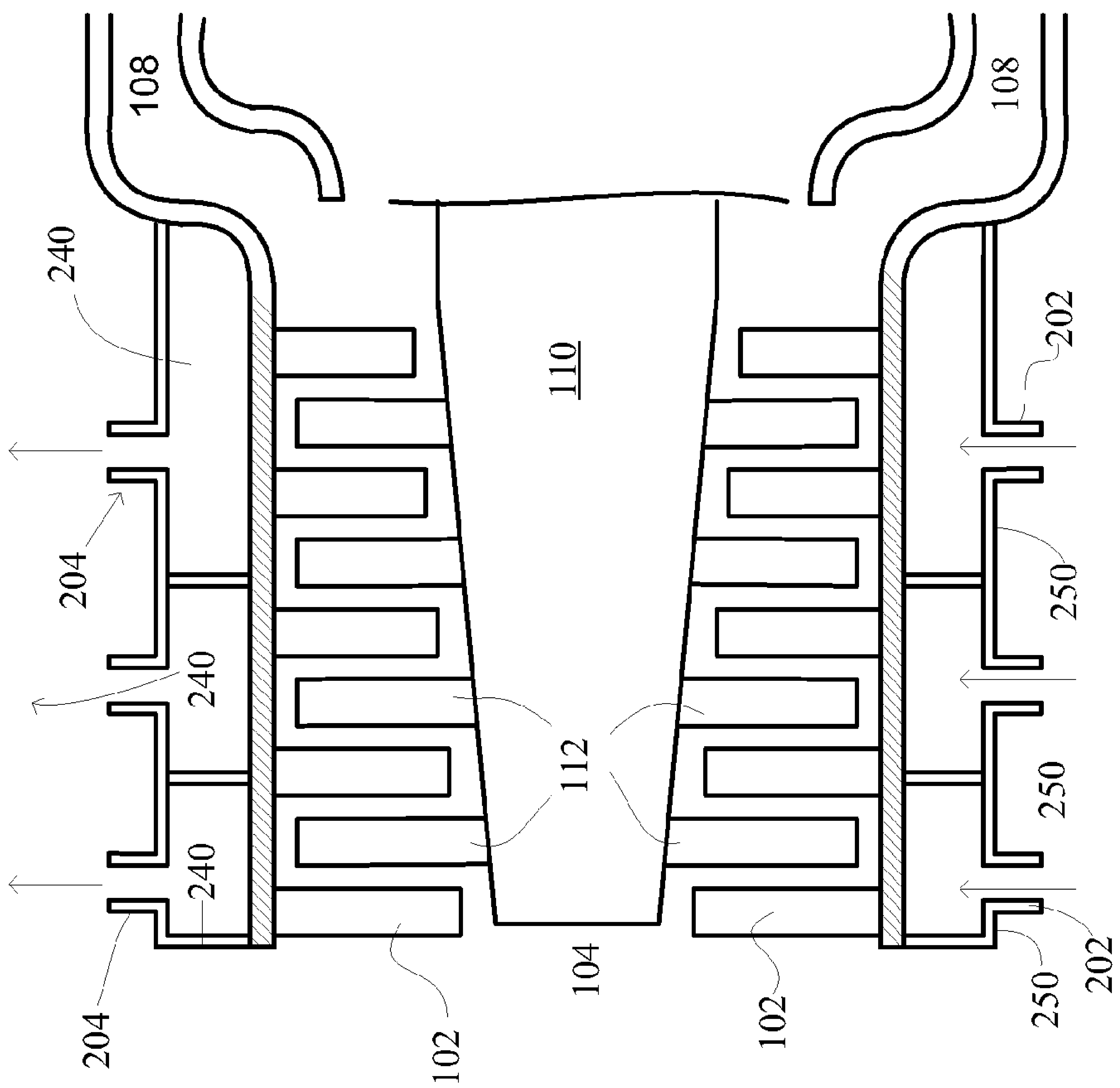
204
120
100
104

Fig. 6A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250
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104
102
102

Fig. 6B

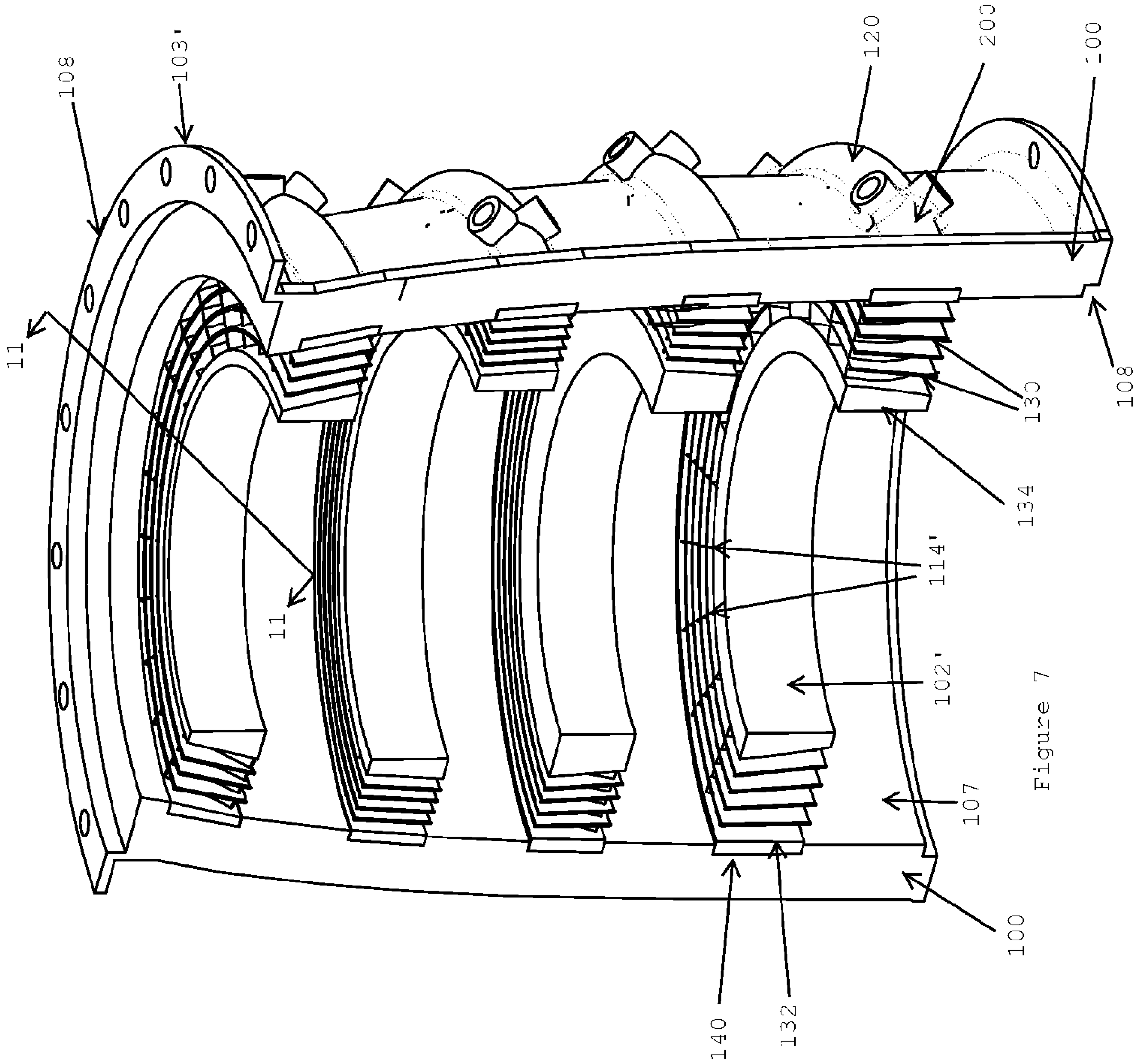


Figure 7

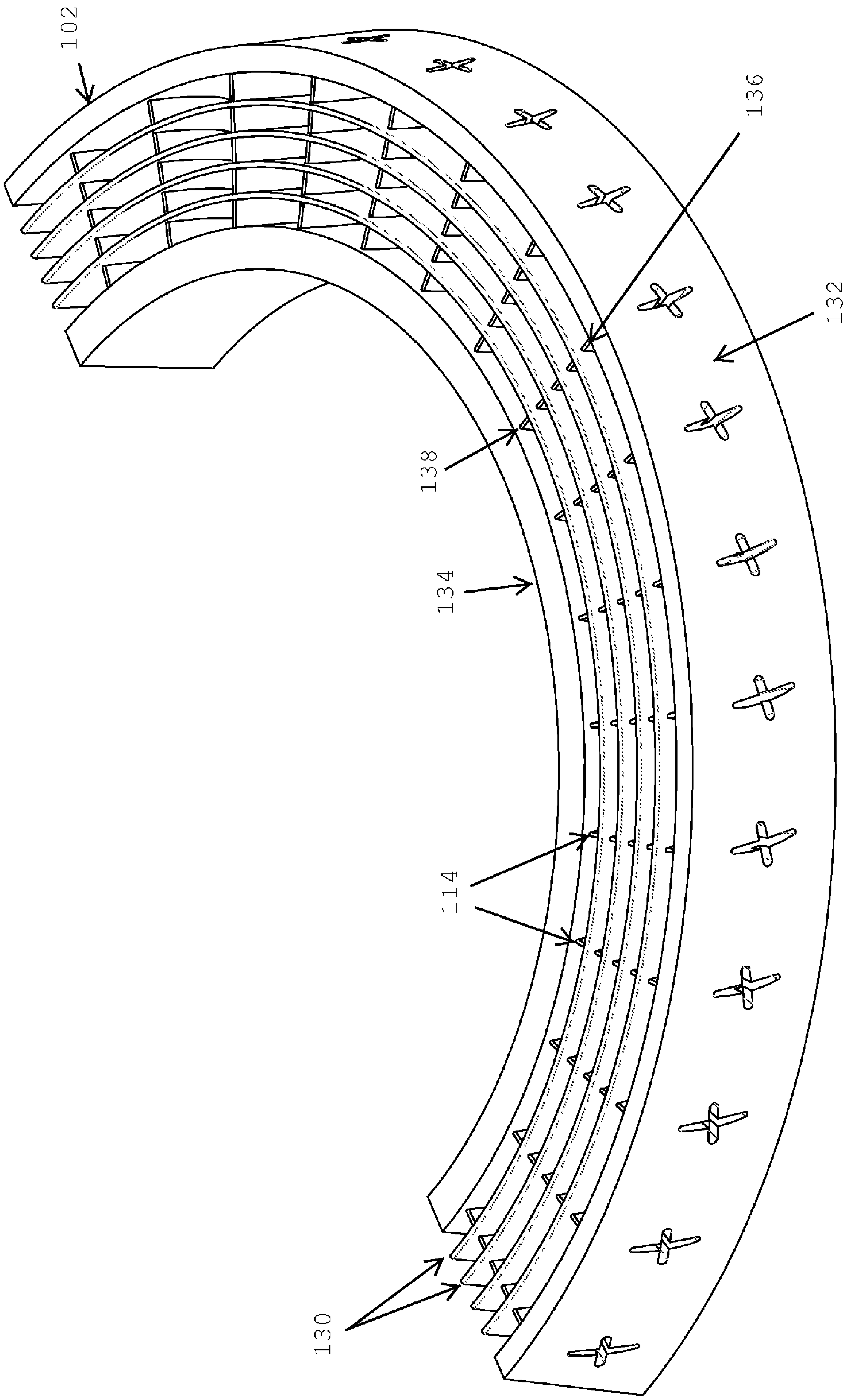


Figure 8

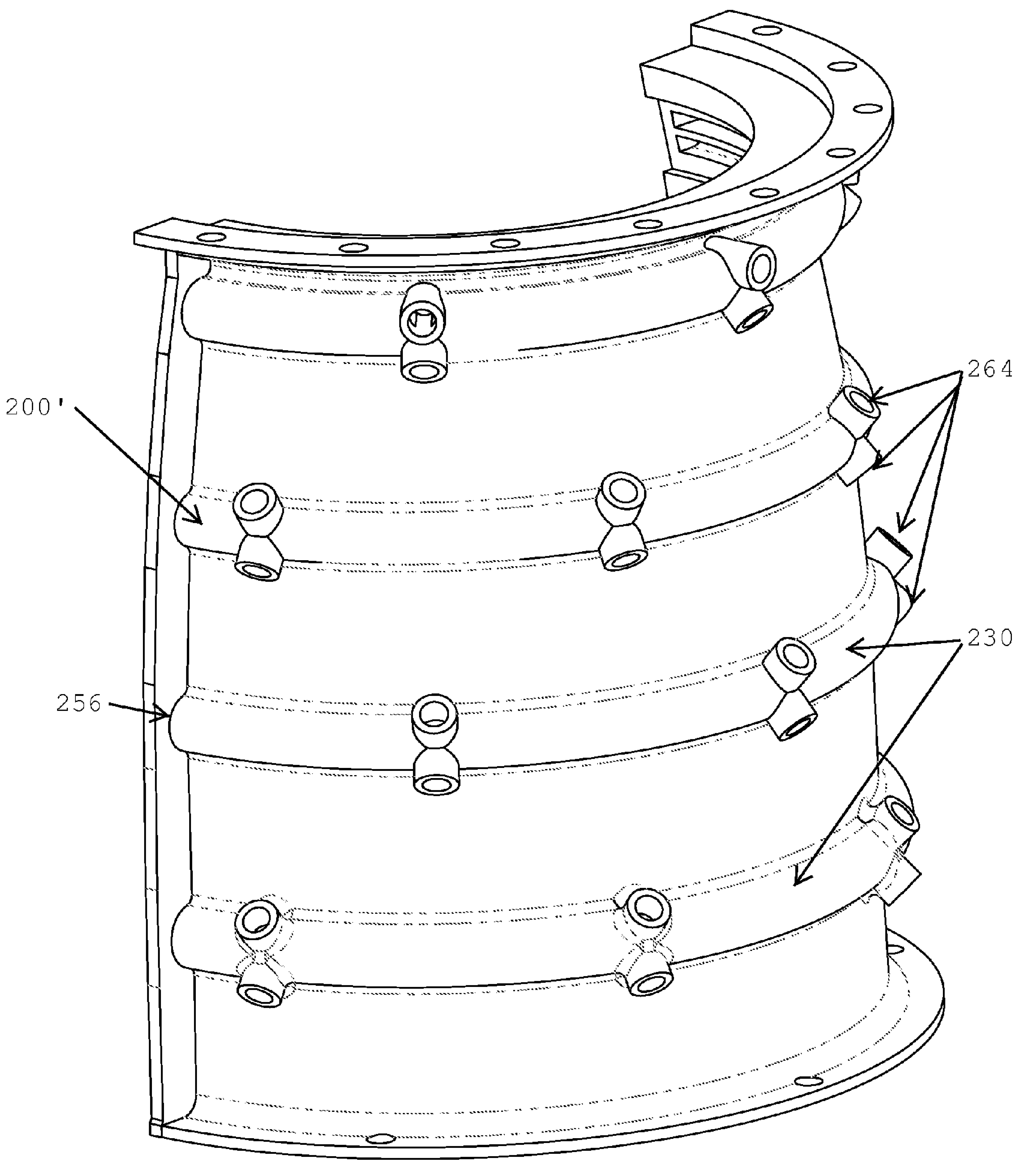


Figure 9

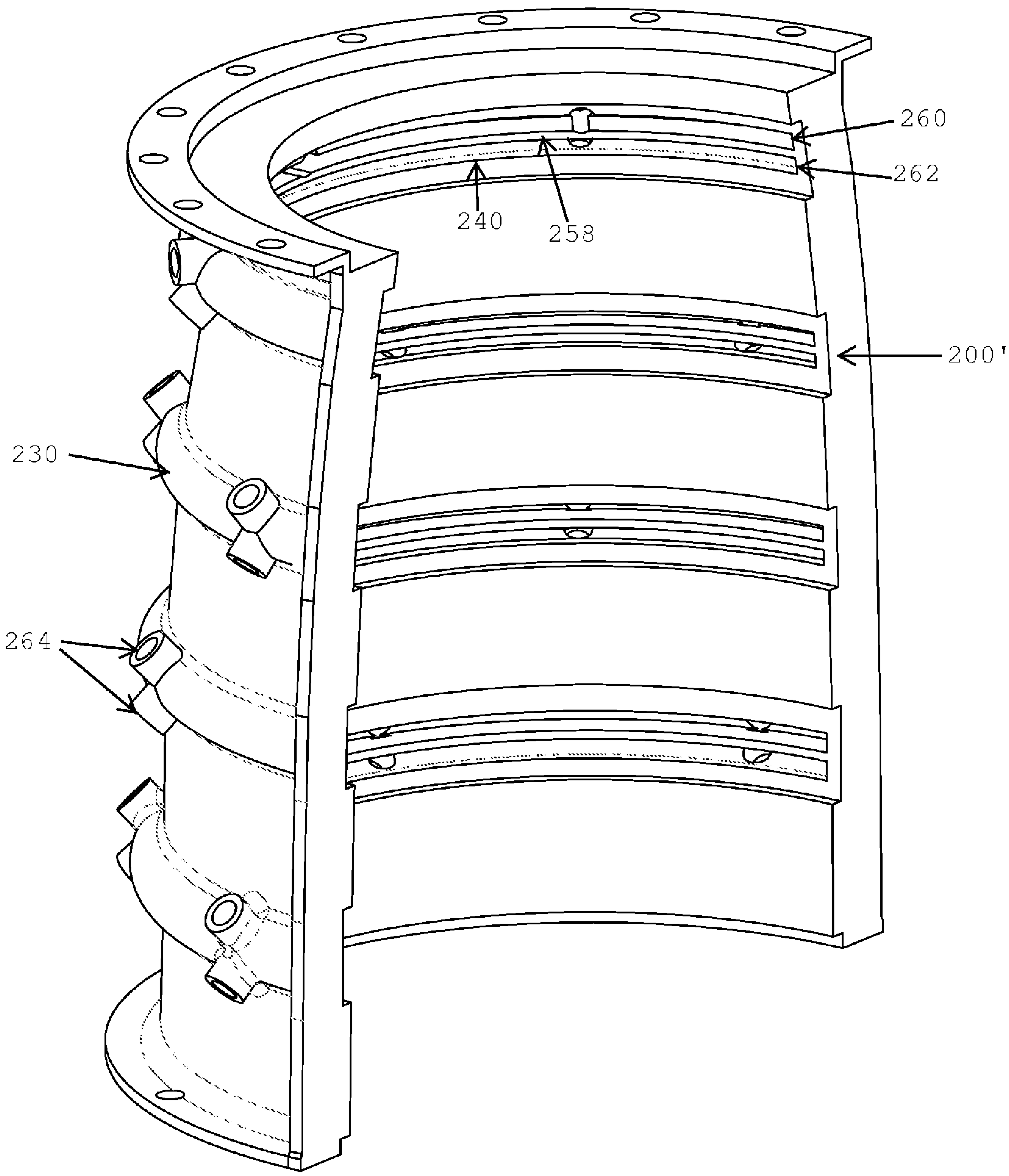


Figure 10

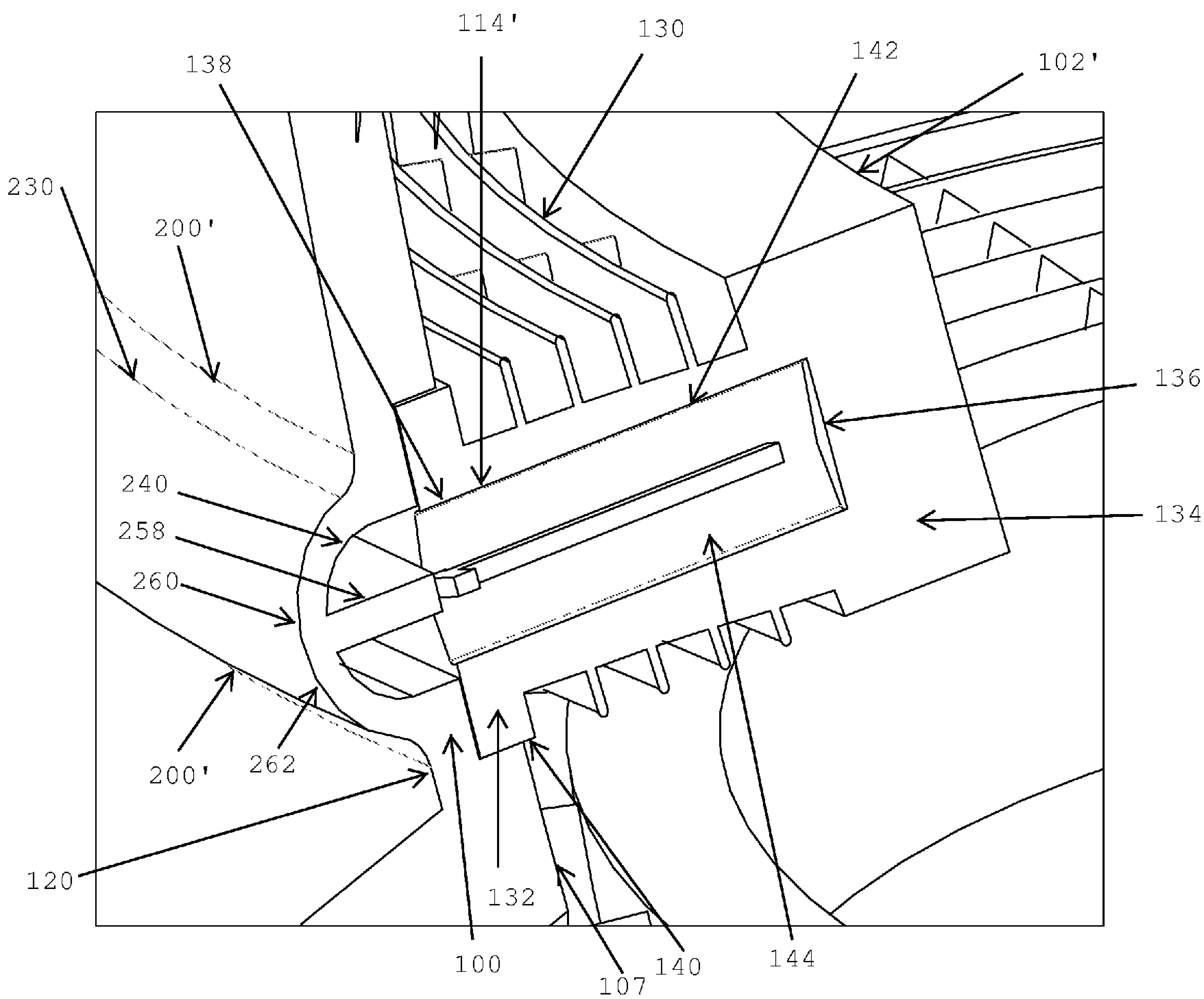

114'
130
138
142
102'
230
200'
136
240
134
258
260
200'
262
120
132
100
140
144
107

Figure 11

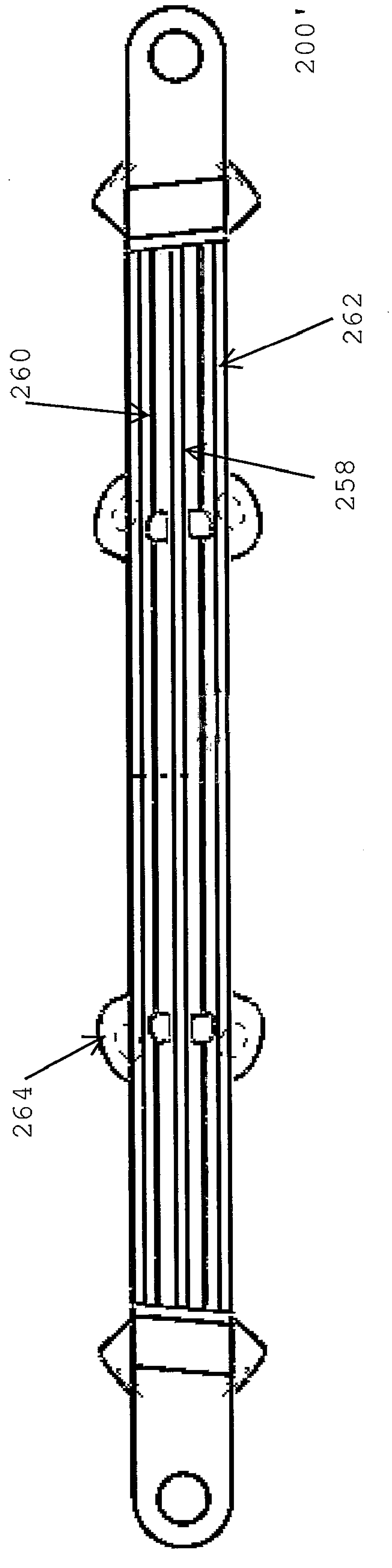


Figure 12

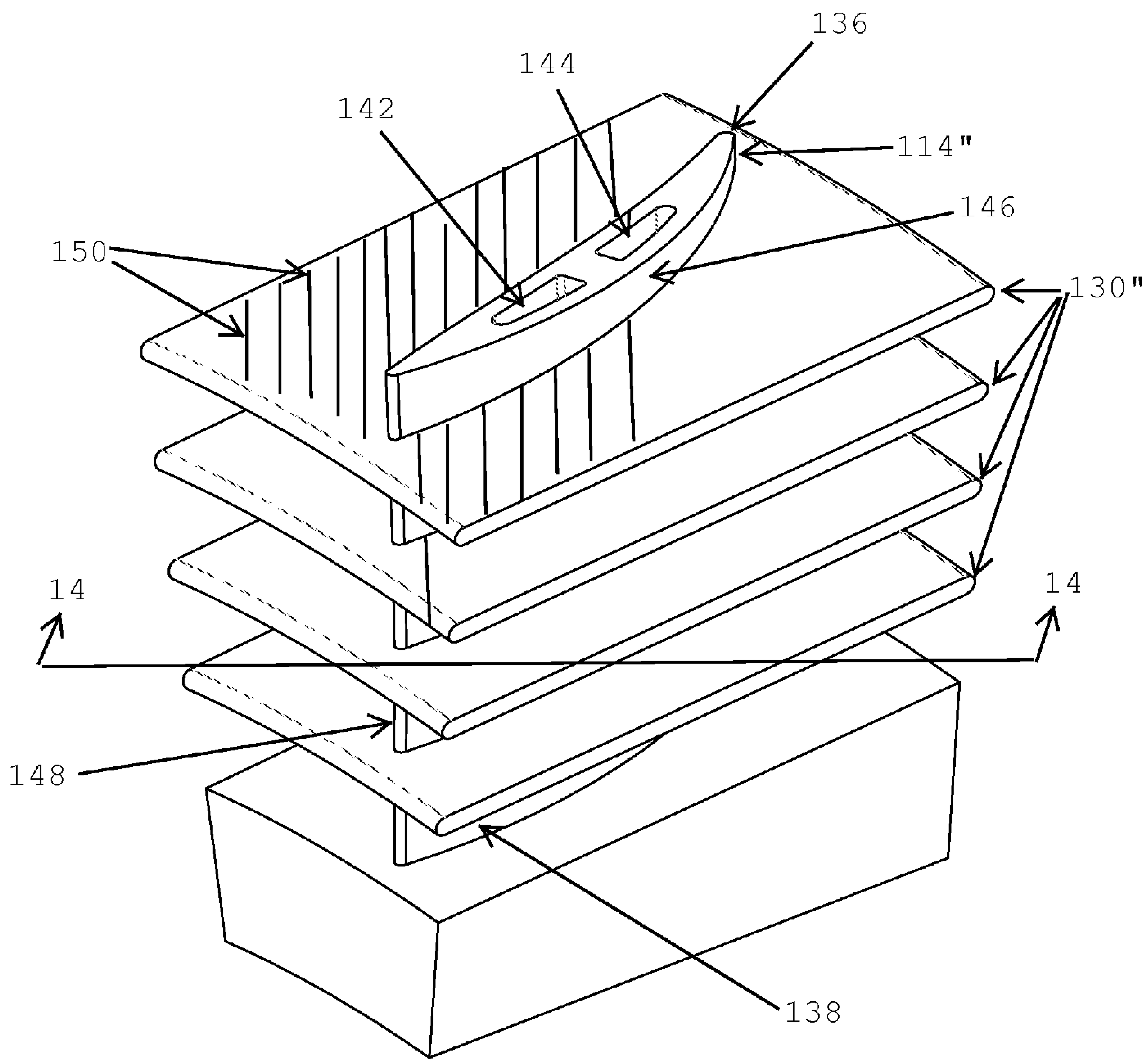


Figure 13

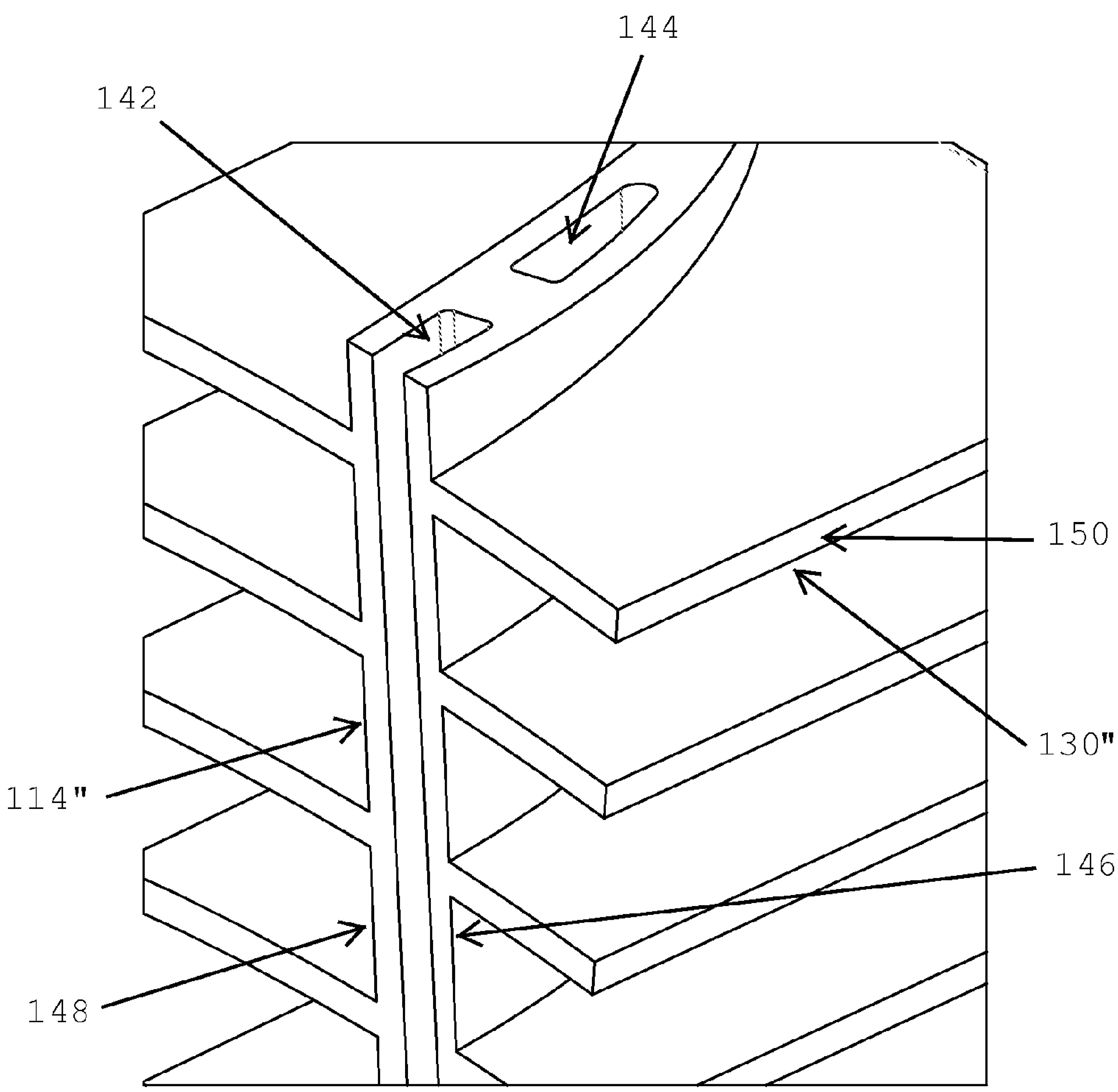


Figure 14

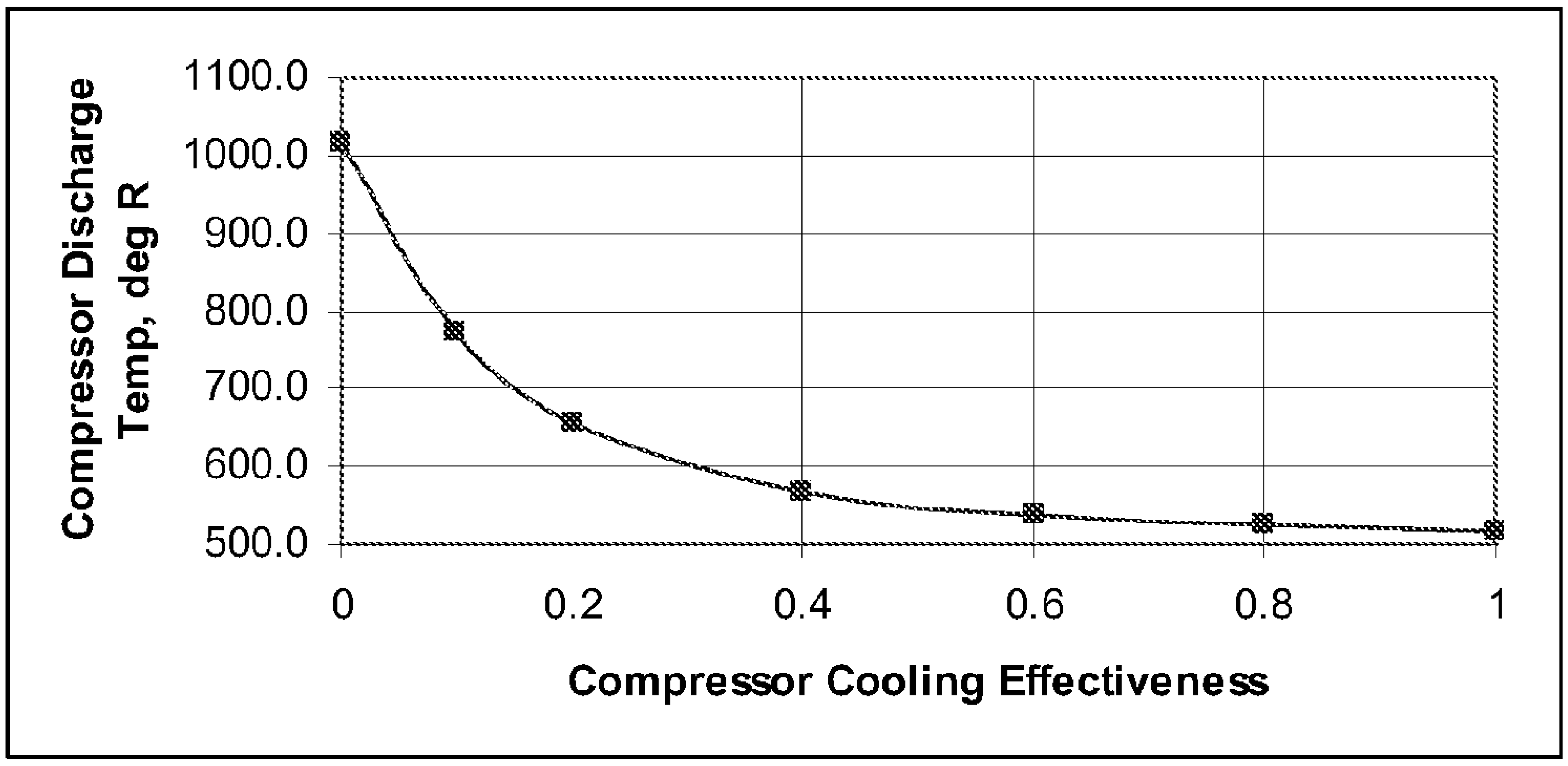


Figure 15 A

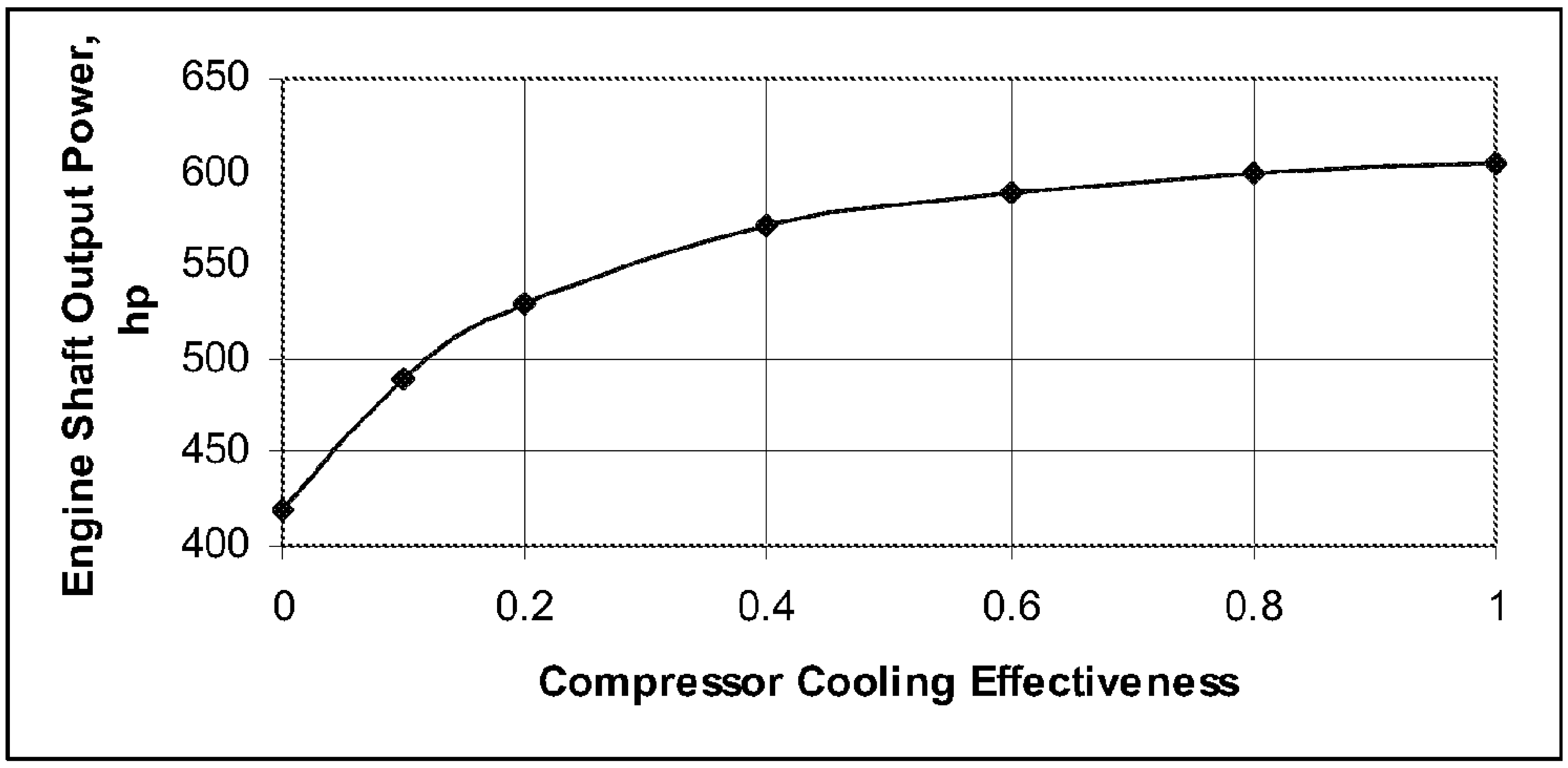


Figure 15B

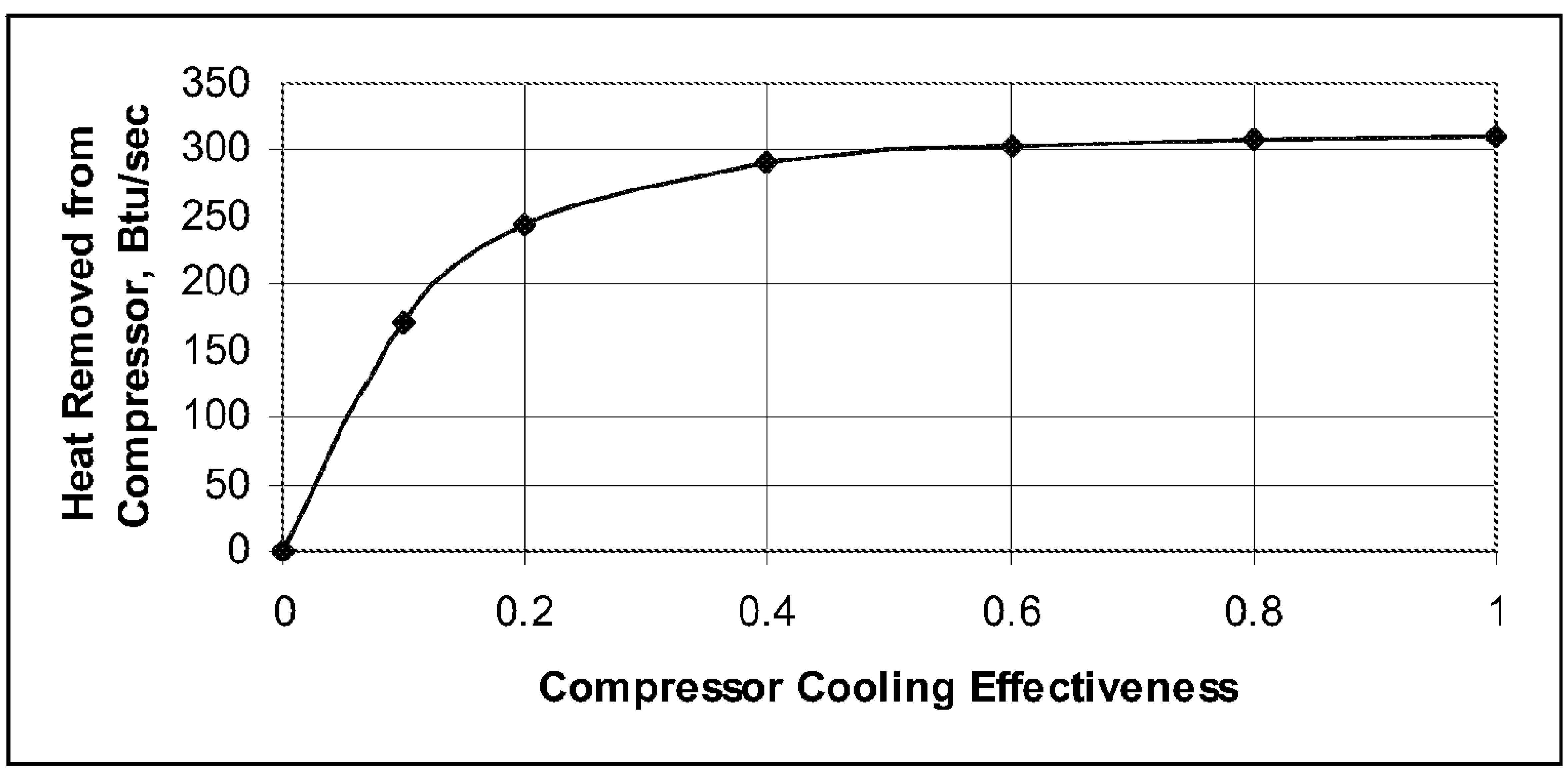
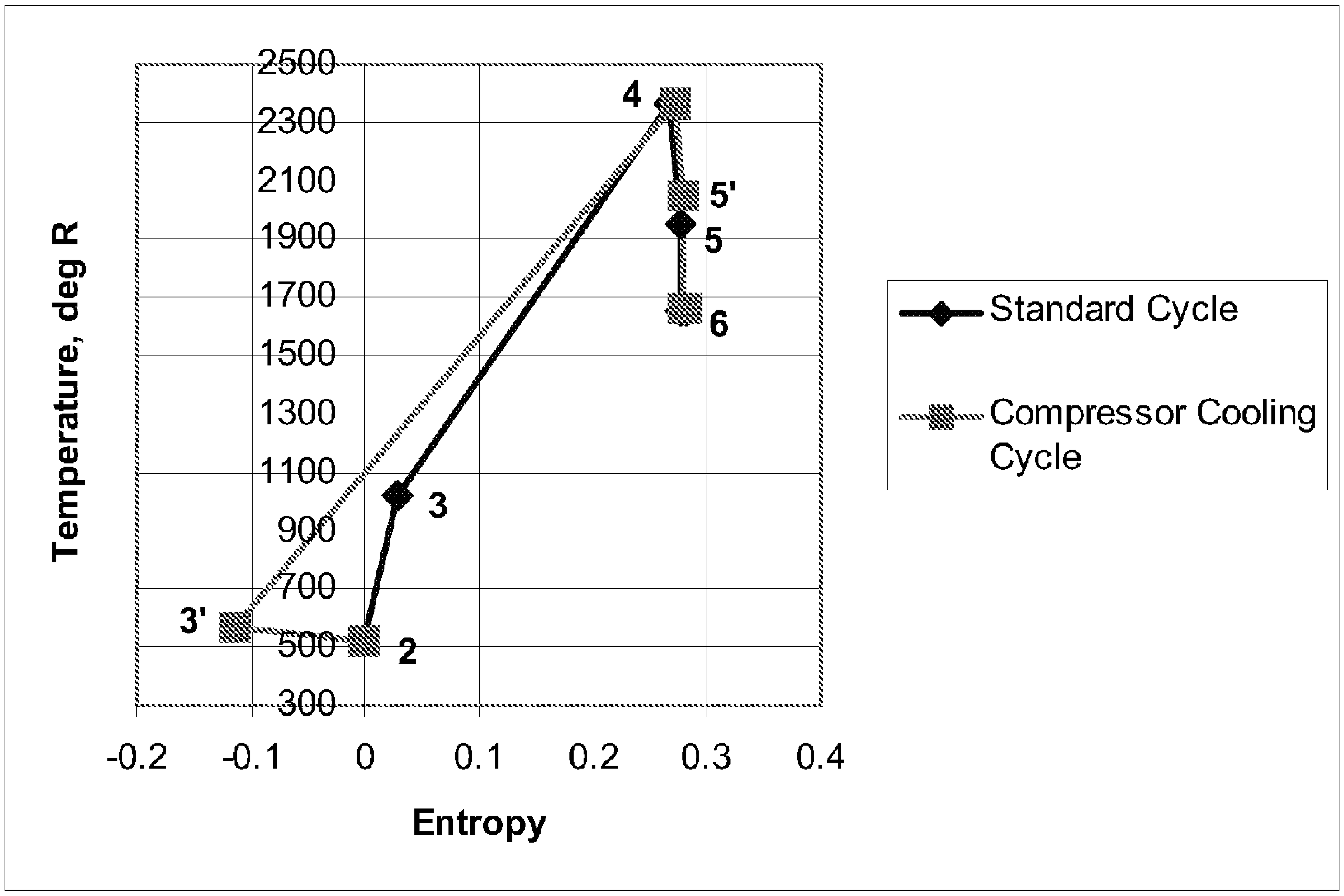


Figure 15C



Temperature, deg R
2500
2300
2100
1900
1700
1500
1300
1100
900
700
500
300
-0.2
-0.1
0
0.1
0.2
0.3
0.4
Entropy
4
5'
5
6
3
3'
2
Standard Cycle
Compressor Cooling Cycle

Figure 16

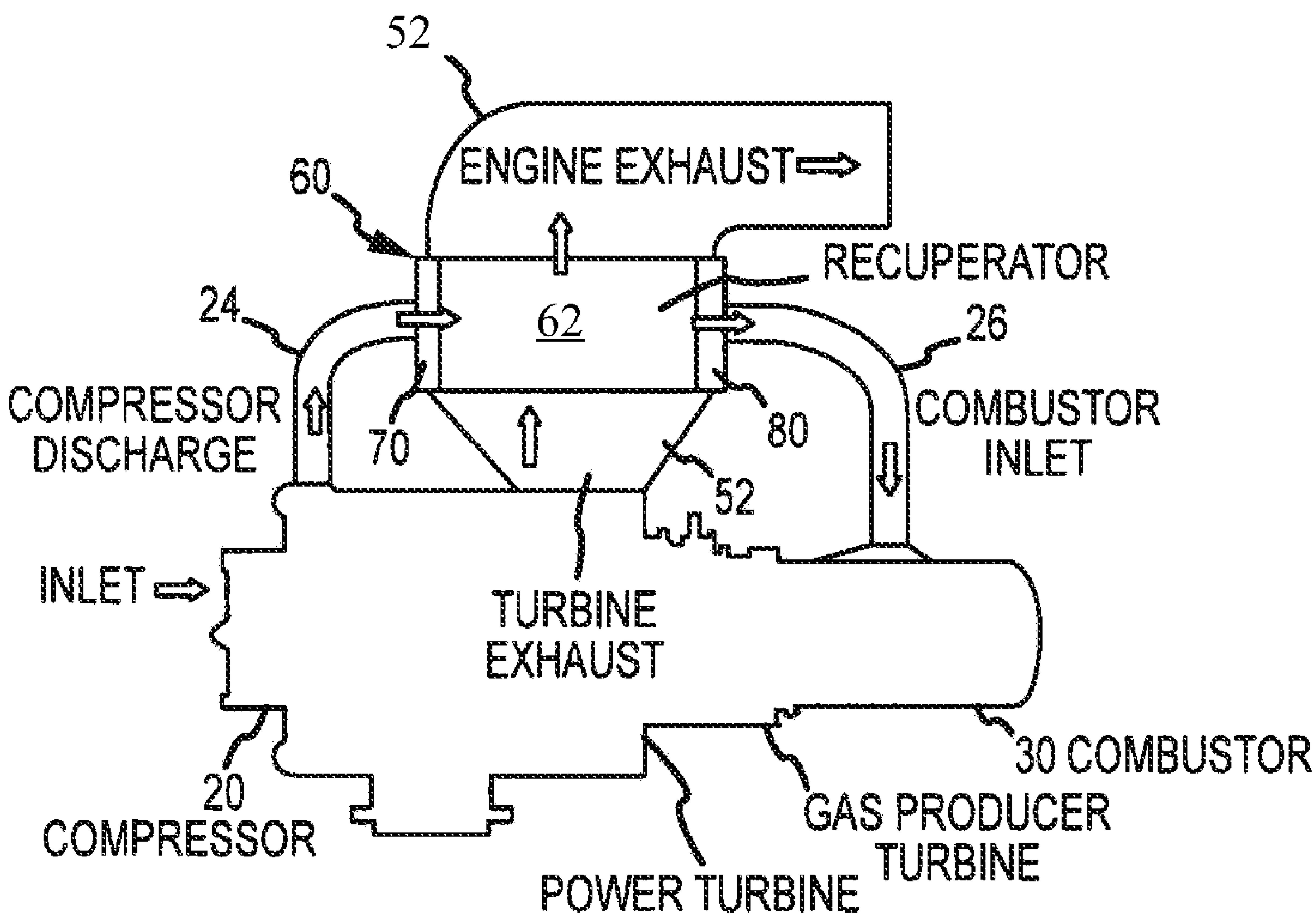

52
60
ENGINE EXHAUST
RECUPERATOR
62
24
26
COMPRESSOR DISCHARGE
70
80
COMBUSTOR INLET
52
INLET
TURBINE EXHAUST
20
COMPRESSOR
30 COMBUSTOR
GAS PRODUCER TURBINE
POWER TURBINE

FIG. 17

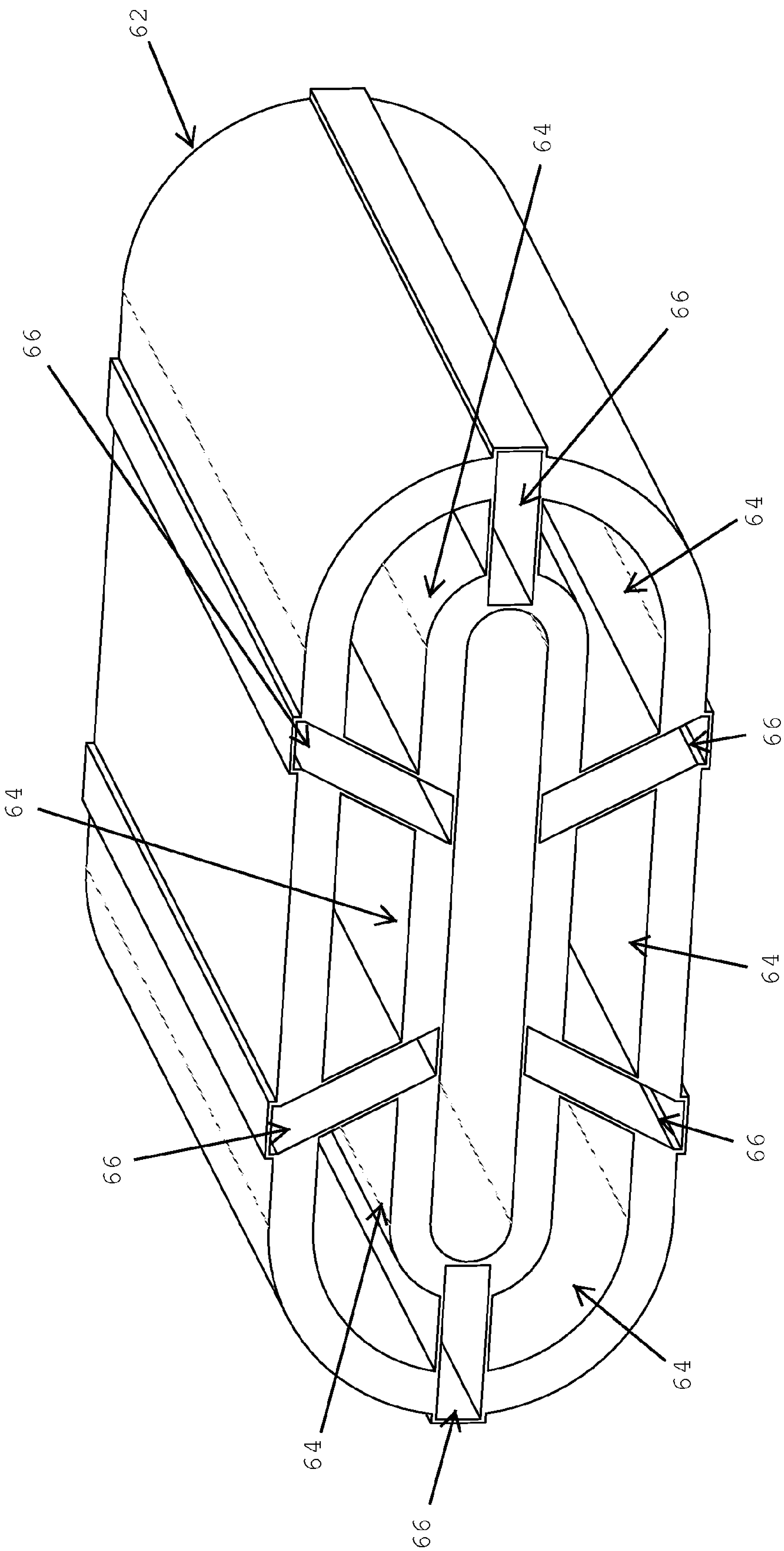
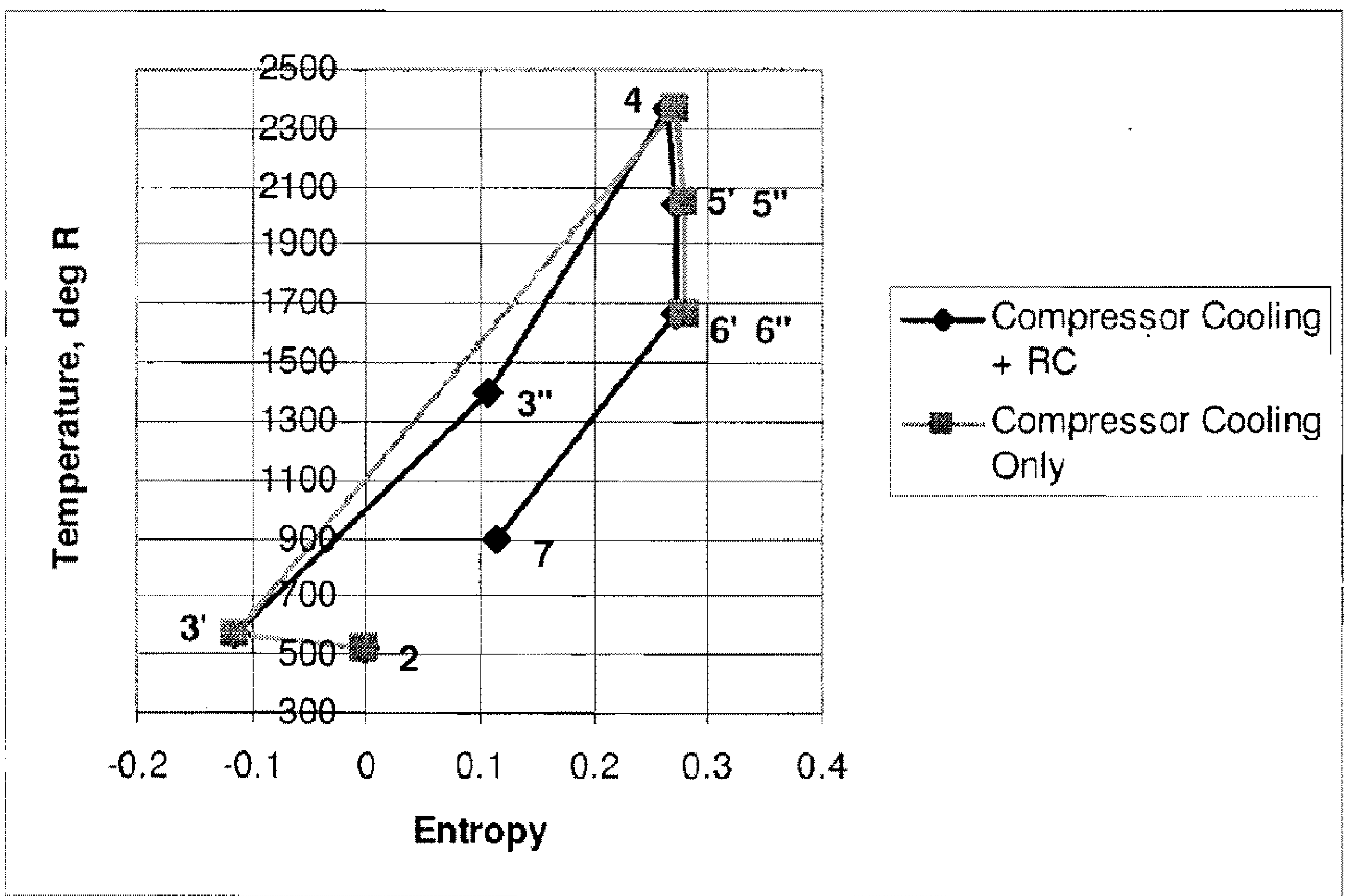


Figure 18


2500
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1900
1700
1500
1300
1100
900
700
500
300
Temperature, deg R
-0.2
-0.1
0
0.1
0.2
0.3
0.4
Entropy
4
5' 5"
6' 6"
3"
7
3'
2
Compressor Cooling + RC
Compressor Cooling Only

Figure 19

COMPRESSOR COOLING FOR TURBINE ENGINES

CROSS-REFERENCE TO RELATED APPLICATIONS

[0001] This application claims the benefit of U.S. Provisional Application No. 61/224,393, entitled "COMPRESSOR COOLING," filed on Jul. 9, 2009. The disclosure of this related application is hereby incorporated into the present application.

FIELD

[0002] The present disclosure is directed toward turbine engines. More specifically, the present disclosure is directed towards a system and method for cooling a compressor of a turbine engine to reduce the temperature rise of air passing through the compressor and thereby reduce the required compressor power.

BACKGROUND

[0003] A gas turbine engine extracts energy from a flow of hot gas that is produced by the combustion of gaseous or liquid fuel with compressed air. In its basic form, a gas turbine engine employs a rotary air compressor driven by a turbine with a combustion chamber disposed between the compressor and the turbine.

[0004] As used herein, the terms "regeneration" (or regenerator), "recuperation" (or recuperator)", and "external intercooling" (external intercooler) will be given the following meanings:

[0005] a) Regenerator—any heat exchanger that transfers heat from an engine exhaust stream to the compressor discharge air;

[0006] b) Recuperator (sometimes called regenerator in technical literature)—a heat exchanger by which heat is transferred by convention and solid conduction across the walls separating exhaust gas from compressor discharge air flowing in adjacent channels (as opposed to rotating solid matrix devices which are sometimes called recuperators and sometimes called regenerators).

[0007] c) External Intercooler—General term for any device which cools air stage-by-stage as the air is being compressed in a multistage compressor wherein the air is ducted out of the compressor at each stage into an external heat exchanger, and then ducted back into the inlet of the next stage.

[0008] Principles of thermodynamics teach that when the temperature of the gases entering the turbine exceeds that entering the compressor, the turbine can deliver more power than the compressor consumes. In this regard, the engine can produce a net power output contingent upon other criteria being met. The efficiency with which the engine converts thermal energy into mechanical energy depends on many factors including compressor and turbine efficiencies, temperature and pressure levels, and the presence or absence of enhancements such as regeneration and compressor air stream cooling (intercooling). The power produced is proportional to the efficiency as well as the mass flow rates of air and fuel. Turboshaft engines deliver mechanical power through a rotating output shaft. Turbojet or turbofan engines require only enough turbine power to operate the compressor (with or without a fan) and the excess fluid power is available in the form of jet thrust.

[0009] Conventional gas turbine engines operate approximately according to the ideal "Gas Turbine" or "Brayton" cycle which, by definition, embodies reversible adiabatic (without heat transfer) compression of atmospheric air, addition of heat at constant pressure, reversible adiabatic expansion through a turbine back to atmospheric pressure, and finally exhausting to the atmosphere. Deviations from the ideal cycle (e.g., irreversibilities) arise due fluid friction and turbulence, inefficiencies in compressors and turbines, combustion heat loss, and the like. When regeneration is present the cycle is designated a "Regenerated Brayton Cycle".

[0010] The Ericsson Cycle patented in 1830 embodies constant pressure regeneration, isothermal compression, and isothermal expansion (reheat), but proposes no means of accomplishing either isothermal compression or expansion. The ideal Ericsson Cycle has "Carnot" efficiency (classical thermodynamics proves that no ideal heat engine operating between given source and sink temperatures can exceed Carnot Cycle efficiency). While the visionary scientists of the nineteenth century, Nicolas Carnot, James Joule, Lord Kelvin, Rudolf Clausius, and Ludwig Boltzman who developed the new branch of science (i.e., Thermodynamics) as well as modern engineers have recognized the benefits of isothermal compression and turbine reheat, no known practical method of achieving approximate isothermal compression (or expansion) has been perfected.

[0011] One attempt to remove compression heat from the engine ("external intercooling") diverts air out of each stage of the compressor, passes the air through a separate heat exchanger/radiator, and re-injects the cooled air into the inlet of the next compressor stage. However, the circuitous piping and multiple changes in flow direction could defeat much, or all of any thermodynamic advantage of external intercooling.

[0012] Another disadvantage of external-intercooling is how the increased complexity of such systems significantly increases the weight of a turbine engine. This is especially relevant to aircraft applications where turbine engines are often utilized due to their high power to weight ratio. That is, in most cases, gas turbine engines are considerably smaller and lighter than reciprocating engines of the same power rating. For this reason, turboshaft engines are used to power almost all modern helicopters. However, incorporation of external intercoolers into turbine engines would result in a significant addition of weight which would more than offset any power gain benefits for such applications.

SUMMARY

[0013] To address the aforementioned problems and inefficiencies of prior attempts to cool compressed air in a turbine engine, disclosed herein are various apparatuses, systems and methods (i.e., utilities) to achieve what will be referred to herein as "internal intercooling". More specifically, internal intercooling is the cooling of the compressor airstream within the compressor (e.g., multistage compressors) without disrupting the normal flow path of the airstream through the compressor. Doing so can expel much of the compression heat to approximate isothermal compression and thereby reduce the consumption of power by the compressor.

[0014] That is, various aspects of the present inventionare directed to internal-compressor-cooling (ICC), which is a practical and effective means of expelling much of the compression heat in order to reduce the consumption of power by the compressor. The most important distinction between the proposed method of approximating isothermal compression,

and previous attempts, is that the transfer of heat to the coolant takes place entirely inside the compressor. Only the coolant needs to be led out of the compressor and circulated through an external heat exchanger (radiator) and the airflow paths through the compressor are essentially unchanged from those of uncooled compressors. By contrast, the previous methods of repeatedly (once for each stage) leading the very large, high velocity airflow of a gas turbine engine out of the engine through massive stage intercoolers then back into the compressor through circuitous flow ducts means adding unacceptable pressure drop and bulk to the engine. Since air compression typically consumes nearly half of the turbine power, the power and efficiency penalty due to loss of pressure can easily outweigh any gain from intercooling.

[0015] While cooling of the compressor and lowering of the compressor discharge temperature for non-recuperated engines can cause an increase in the fuel flow rate needed to maintain the turbine inlet temperature at its set value, the incremental increase in the required combustion heat is the same as the incremental decrease in compressor specific work. Thus, the turbine net specific work (i.e., total turbine specific work minus compressor specific work) increases by that same amount (i.e., the output power increases by exactly the same amount as the increase in combustion heat rate). As efficiency is given by net-power/combustion-heat-rate, efficiency actually increases because the same increment is added to the numerator and denominator of a fraction less than 1.0 (i.e., this causes an increase in the value of the fraction). When recuperation is present, the efficiency increase may be much more pronounced because less of the heat rate is supplied by combustion resulting in a larger heat transfer driving potential for the recuperator.

[0016] One of the utilities disclosed herein includes a "fluid jacket" or heat exchanger that may be designed to be mounted around a portion or the entirety of an outer surface of the compressor housing to absorb heat or thermal energy generated by the compressor and thereby approximate isothermal compression. A coolant may circulate through the cooling jacket to absorb thermal energy from the compressor and then through another heat exchanger (e.g., radiator) to release the thermal energy, before returning to the cooling jacket. Existing compressors may be efficiently retrofitted using such cooling jackets.

[0017] Another of the utilities disclosed herein includes specially designed stator blades that may be operable to absorb thermal energy which can then be transferred away from the stator blades (and the compressor as a whole). As one example, the stator vanes or blades (or possibly rotor blades or centrifugal compressor vanes) may include passages through which a fluid (e.g., coolant) may be circulated to absorb heat from the airstream traveling through the compressor. As can be appreciated, only the coolant needs to be led out of the compressor and circulated through an external heat exchanger (e.g., radiator). Stated otherwise, the airflow paths through the compressor may be unchanged from previous compressors. Again, existing compressors may be retrofitted by removing an existing compressor or stator housing and replacing the existing housing with a housing including the specially designed and inventive stator blades.

[0018] In one aspect, an apparatus for use with a gas turbine engine is disclosed. The apparatus includes an annular compressor housing including inside and outside surfaces, and inlet and outlet ends, such that air generally moves in an air flow direction from the inlet end towards the outlet end. A plurality of spaced sets of stators extend from the inside surface of the compressor housing each of which includes a plurality of stator blades (e.g., vanes). Extending around at least a portion of the outside surface of the compressor housing is a first heat exchanger (e.g., one or more fluid or cooling jackets) that is operable to absorb thermal energy from the compressor housing and transfer thermal energy away from the stator blades. This apparatus advantageously provides for "internal intercooling" of the compressor airstream within the compressor to approximate isothermal compression and thereby reduce the consumption of power by the compressor.

[0019] The first heat exchanger may include at least one coolant fluid path that extends over at least a portion of the outer surface of the compressor housing to absorb thermal energy from the compressor. In one arrangement, a single coolant fluid path may extend over substantially the entire outer surface of the compressor housing. The coolant fluid path may be defined by an outer wall spaced from the outer surface of the compressor housing to form a fluid tight cavity between the outer wall and the outer surface of the compressor housing, an inlet port for introducing coolant into the cavity, and an outlet port for removing coolant from the cavity, whereby coolant is operable to flow through the cavity between the inlet and outlet ports. In some variations, the first heat exchanger may also include an inner wall spaced from the outer wall that forms the cavity collectively with the outer wall. In this case, the inner wall may be placed in appropriate thermal or conductive contact with the outer surface of the compressor housing. In another variation, the first heat exchanger may be in the form of first and second portions (e.g., identical portions) that may extend around or about a respective portion of the compressor housing for securement together (e.g., via bolts or other fastening arrangements).

[0020] A second heat exchanger (e.g., a radiator) may be appropriately fluidly interconnected (e.g., tubing, passages) to the first heat exchanger for removing or releasing heat or thermal energy from the coolant existing the first heat exchanger and then circulating the coolant back to the first heat exchanger. Other equipment and componentry (e.g., pumps) may also be appropriately interconnected to the first and second heat exchangers as needed.

[0021] In other arrangements, the first heat exchanger may include multiple coolant paths that may extend over separate portions of the outer surface of the compressor housing. This arrangement may advantageously allow more of the coolant circulating over the outer surface of the compressor housing to be "fresh" coolant or in other words coolant of a higher heat absorbing capacity than if the same coolant traveled over the entire outer surface of the compressor housing via a single coolant fluid path. As an example, a different one of the coolant fluid paths of the first heat exchanger may be aligned with a different one of the sets of stators as most of the thermal energy generated by the compressor may be concentrated in the stator blades. For instance, the first heat exchanger may include a dividing wall extending between the outer wall and the outer surface of the compressor housing to form first and second cavities, each of which includes an inlet port for receiving coolant and passing the coolant into the respective first or second cavity and an outlet port for removing coolant from the respective first or second cavity.

[0022] The stator blades of the apparatus may be appropriately designed to increase the thermal energy or heat absorbing capacity of the stator blades, and such thermal energy may be subsequently removed from the stator blades via, for instance, conduction through the base of the stator blades and/or via a circulated coolant. In one arrangement, one or more of the stator blades may be designed to accept a circulated coolant therethrough which may be operable to absorb and remove heat that has been absorbed by the stator blades (e.g., transfer thermal energy away from the stator blade(s)). For instance, each stator blade may include an intake passage extending along a portion of a length of the stator blade for receiving a flow of coolant into the stator blade, and a return passage, fluidly interconnected to the intake passage, and extending along a portion of the length of the stator blade for passing the flow of coolant out of the stator blade. In a further arrangement, external surface features may be added to the stator blades and/or the shape of the stator blades (e.g., turning radius, angles, height, width) may be changed to increase the heat absorbance of the stator blades and thereby increase the quantity of thermal energy that may be removed via the coolant circulating about or passing through the stator/compressor housing and/or about or through the passages in the stator blades. For instance, a number of fins (e.g., outwardly extending plates may be added to at least one of first and second generally opposing surfaces of the stator blades to increase such heat acceptance. In one arrangement, one or more fins may interconnect two or more adjacent stator blades. In another arrangement, ridges may be added to the fins and may be oriented at least partially transverse to the air flow direction through the compressor housing. Additionally, it should be appreciated that the stator blades, compressor housing, first heat exchanger, etc. may be constructed of any appropriate materials that further the disclosure herein. For instance, the stator blades may be constructed of any appropriate metallic material to absorb and then conduct heat to the compressor housing (which may also be constructed of metal (s)).

[0023] In another aspect, a stator structure for a compressor is disclosed including a stator casing having inside and outside surfaces, and inlet and outlet ends, such that air generally moves in an air flow direction from the inlet towards the outlet end, and a plurality of spaced stator sections extending from the inside surface of the stator casing, each of which includes a plurality of stator blades. In this aspect, each stator blade includes intake and return passages in the manner as discussed previously.

[0024] In one arrangement, each stator section includes a stator ring that is appropriately secured to the inside surface of the stator casing (e.g., via complementary dovetail or other features) and the plurality of stator blades extend away from the stator ring. In another arrangement, a first end of the stator blades are interconnected to the inside surface of the stator casing (or the stator ring) and a second end of the stator blades in each respective stator section may be interconnected by a shroud cover. One or more cooling fins may extend from the stator blades as discussed above.

[0025] To allow fluid access to the passages of the stator blades through the compressor or stator casing, the inside surface of the stator casing may include a plurality of pairs of ports, each pair including a first port that is fluidly interconnected to the intake passage of a stator blade and a second port that is fluidly interconnected to the return passage of the stator blade such that the first port is operable to pass coolant from a heat exchanger into the intake passage and the second port is operable to pass coolant from the return passage towards the heat exchanger. Any appropriate manifold(s) or cooling jacket(s) may be attached to the outside surface of the stator casing to provide coolant to the first port and receive coolant from the second port of each of the pair of ports, and such manifold(s) or cooling jacket(s) may be appropriate fluidly interconnected to a radiator and/or a pump.

[0026] Another aspect is directed to a compressor including a stator structure and a rotor structure. The stator structure includes a stator casing with inside and outside surfaces, inlet and outlet ends, and a central axis running through a center of the stator casing, such that air generally moves in an air flow direction from the inlet towards the outlet end. The stator structure also includes a plurality of spaced stator sections extending from the inside surface of the stator casing each of which has a plurality of stator blades, and a plurality of coolant passages extending through the stator sections that receive a circulated coolant operable to absorb thermal energy from the stator sections and transfer the thermal energy away from the stator sections. The rotor structure includes a rotatable shaft having a longitudinal axis that is coincident with the central axis of the stator casing, and a plurality of rotor segments attached to and extending from the rotatable shaft that are disposable between the spaced stator sections. One or more heat exchangers (e.g., cooling jackets) may extend around at least a portion of the outside surface of the stator casing to absorb thermal energy from the compressor and transfer thermal energy away from the compressor.

[0027] In another aspect, a gas turbine engine is disclosed including a turbine section, a combustion chamber that is fluidly interconnected to the turbine section, a compressor that is fluidly interconnected to the combustion chamber and that has a housing with inside and outside surfaces, and a fluid jacket (e.g., cooling jacket as discussed above) extending around at least a portion of the outside surface of the compressor that is operable to circulate coolant over the outside surface of the compressor to absorb thermal energy from the compressor and transfer thermal energy away from the compressor.

[0028] In a gas turbine engine utilizing any of the above aspects or arrangements, a regenerator (i.e., a device or system that uses exhaust gases from the turbine section to heat compressed air exiting the compressor prior to entry into the combustion chamber; e.g., a recuperator) may be appropriately incorporated into the gas turbine engine to further increase the efficiency and power output of the engine. That is, as the internal intercooling systems disclosed herein reduce the compressor specific work requirements by reducing temperature and thermal energy increases within the compressor, a recuperator would at least partially make up for decreases from turbine inlet set temperature values owing to such internal intercooling apparatuses and thereby at least partially reduce increases in the fuel flow rate within the combustion chamber to maintain such turbine inlet set temperature values.

[0029] For instance, the recuperator may be fluidly interconnected to an outlet of the compressor and an inlet to the combustion chamber so as to use exhaust gases from the turbine section to heat compressed air exiting the compressor prior to entry into the combustion chamber. The recuperator may be in the form of a counter-flow recuperator, cross-flow recuperator, and the like. As will be appreciated more fully below, use of a recuperator along with the various cooling systems and arrangements disclosed herein may improve fuel economy by reducing the fuel rate needed to attain a desired turbine inlet temperature of the gases flowing therethrough.

[0030] Some embodiments provide various methodologies for use in cooling a compressor in a gas turbine engine to increase engine power and efficiency. One method includes establishing a coolant fluid path that extends at least partially over an outside surface of a compressor housing of the compressor, first passing a coolant along the coolant fluid path over the outside surface of the compressor housing, absorbing, using the coolant, thermal energy from the compressor, and second passing the coolant along the coolant fluid path away from the outside surface of the compressor housing.

[0031] In one variation, the coolant may be a non-circulated fluid (e.g., water) that absorbs thermal energy from the compressor and is then exhausted. In another variation, thermal energy may be released from the coolant in any appropriate manner. For instance, the coolant may be circulated to an external heat exchanger for release of thermal energy. Thereafter, the coolant may be circulated through the coolant fluid path back over the outside surface of the compressor housing. In this regard, the coolant fluid path may be in the form of a circuit through which the coolant may continuously travel or flow. In one arrangement, the establishing step may include establishing the coolant fluid path through passages within a plurality of stator blades of the compressor such that, for instance, the coolant may be first circulated along the coolant fluid path over the outside surface of the compressor housing, and then through the passages in the plurality of stator blades. In another arrangement and after the coolant has flowed or circulated through the passages in the plurality of stator blades, the method may include circulating the coolant along the coolant path back over the outside surface of the compressor housing, and then along the coolant path to an external heat exchanger for release of any thermal energy absorbed. In one variation, the fluid may be circulated through a cooling jacket (e.g., as discussed previously) disposed about an outer housing of the compressor.

[0032] In another aspect, a thermodynamic cycle for use in a gas turbine engine is disclosed. The cycle includes, using a compressor having a housing, compressing air at a substantially constant temperature; in conjunction with the compressing step, circulating a cooling fluid over an outside surface of the compressor housing or through stator blades in the compressor, where the cooling fluid removes heat from the air to allow the compressing step to operate at the substantially constant temperature. The cycle also includes, using a combustion chamber, heating the compressed air at a substantially constant pressure; expanding the heated, compressed air through a turbine stage to drive the turbine stage and the compressor; and exhausting the air to the atmosphere. As can be appreciated, this thermodynamic cycle approximates isothermal compression which advantageously reduces power consumption by the compressor and correspondingly increases the efficiency of the gas turbine engine. In one arrangement, the cycle also may include passing the compressed air through a recuperator disposed within the exhaust flow of the engine to increase the temperature of air exiting the compressor. In this regard, less fuel energy may be required to obtain a desired or target temperature of the gases entering the turbine stage of the gas turbine engine.

BRIEF DESCRIPTION OF THE DRAWINGS

[0033] FIG. **1** illustrates a perspective view of a gas turbine engine.

[0034] FIG. **2** shows a side view of the engine of FIG. **1**.

[0035] FIG. **3** shows an end view of the engine of FIG. **1**.

[0036] FIG. **4** shows an exploded view of a compressor assembly.

[0037] FIG. **5** illustrates the side view of the engine of FIG. **2**, but with a cross-sectional view of a cooling jacket being mounted about a compressor of the engine, along with a schematic view of associated componentry (e.g., tubing, radiator).

[0038] FIG. **6**A illustrates a more detailed cross-sectional view of the cooling jacket of FIG. **5**.

[0039] FIG. **6**B illustrates a cross-sectional view of another embodiment of the cooling jacket of FIGS. **5** and **6**A.

[0040] FIG. **7** illustrates a perspective view of a stator structure of a compressor according to another embodiment.

[0041] FIG. **8** illustrates a perspective view of a stator section of the stator structure of FIG. **7**.

[0042] FIG. **9** illustrates a perspective view of a cooling jacket of the stator structure of FIG. **7**.

[0043] FIG. **10** illustrates another perspective view of the cooling jacket of FIG. **9**.

[0044] FIG. **11** illustrates a cross-sectional view through the line **11-11** of FIG. **7**.

[0045] FIG. **12** illustrates a top, inside view of the cooling jacket of FIG. **9**.

[0046] FIG. **13** illustrates a perspective view of a stator blade according to one embodiment.

[0047] FIG. **14** illustrates a cross-sectional view through the line **14-14** of FIG. **13**.

[0048] FIG. **15**A presents a graphical representation illustrating compressor cooling effectiveness versus compressor discharge temperature in deg R.

[0049] FIG. **15**B presents a graphical representation illustrating compressor cooling effectiveness versus engine shaft output power in HP.

[0050] FIG. **15**C presents a graphical representation illustrating compressor cooling effectiveness versus heat removed from the compressor in BTU/sec.

[0051] FIG. **16** presents a graphical representation illustrating entropy versus temperature in deg R for a standard gas turbine engine cycle and for a compressor cooled gas turbine engine cycle.

[0052] FIG. **17** illustrates a perspective view of a recuperator usable together with the cooling jackets and stator sections in a turbine engine.

[0053] FIG. **18** illustrates a perspective view of a core of the recuperator of FIG. **17**.

[0054] FIG. **19** presents a graphical representation illustrating entropy versus temperature in deg R for a compressor cooled gas turbine engine cycle and for a compressor cooled gas turbine engine cycle with recuperation.

DETAILED DESCRIPTION

[0055] Reference will now be made to the accompanying drawings, which assist in illustrating the various pertinent features of the various novel aspects of the present disclosure. Although described primarily with respect to compressor cooling systems, apparatuses and methods (i.e., "utilities") that may be combined with recuperation and used with a turbine engine, aspects of the utilities are applicable to axial compressors that may be utilized for gas compression applications such as gas pipeline compressors. In this regard, the following description is presented for purposes of illustration and description. Furthermore, the description is not intended to limit the inventive aspects to the forms disclosed herein. Consequently, variations and modifications commensurate with the following teachings, and skill and knowledge of the relevant art, are within the scope of the present inventive aspects.

[0056] As noted, the compressor cooling utility discussed herein may be utilized with a variety of different gas turbine engines. The present description describes the compressor cooling utility in relation to the Rolls-Royce Model 250 family of engines (US military designation T63). This family of engines has a number of different sizes and varying configurations. The engine was originally designed by a General Motors offshoot, the Allison Engine Company, in the early 1960's. A program of continuous development has resulted in a range of engine models that power many of the world's most popular small aircraft and helicopters. For instance, a small non-inclusive list includes the Bell 206B/TH-67, MDH MD500/520N and Eurocopter AS.355/BO 105.

[0057] The Model 250 engine **10**, as schematically shown in the perspective, side and front views of FIGS. **1-3**, utilizes what is sometimes referred to as a "trombone" engine configuration whereby air enters the intake of the compressor **20** in a conventional fashion, but whereby compressed air leaving the compressor **20** is ducted rearwards around the turbine system via external air ducts **22**. That is, unlike most other turboshaft engines, the compressor **20**, combustion chamber or combustor **30** and turbine section or stage **40** are not provided in an inline configuration with the compressor at the front and the turbine at the rear where compressed air flows axially through the engine. Rather, in the Model 250 engines, the engine air from the forward compressor **20** is channeled through the external compressed air ducts **22** on each side of the engine **10** to the combustor **30** located at the rear of the engine. The exhaust gases from the combustor **30** then pass into a turbine stage **40** located intermediate the combustor **30** and the compressor **20**. The exhaust gases are exhausted mid-engine in a radial direction from the turbine axis A-A of the engine, through two exhaust ducts **42**. A power take-off shaft **44** connects the power turbine of the turbine stage to a compact reduction gearbox (not shown) located inboard between the compressor and the exhaust/power turbine system.

[0058] Gas turbine engines are described thermodynamically by the idealized Brayton cycle, in which air is compressed isentropically, combustion occurs at constant pressure, and expansion over the turbine occurs isentropically back to the starting pressure. In practice, friction and turbulence cause non-isentropic compression. Specifically, the compressor tends to deliver compressed air at a temperature is higher than ideal. Further, pressure losses in the air intake, combustor and exhaust reduce the expansion available to provide useful work. By some estimates, up to half of the power produced by the engine goes to powering the compressor.

[0059] FIG. **4** illustrates an exploded view of the compressor **20**. Broadly, the compressor **20** may include a rotor structure **109** and a stator structure **103**. The rotor structure **109** may include a rotating shaft **110** that extends through the engine **10** to the turbine stage **40**. The rotating shaft **110** may include multiple rotor sections or rows **112** spaced along the length of the rotating shaft **110**, each of which include a series or set of rotor blades **113** extending away from the rotating shaft **110**. The stator structure **103** may include a stator housing or casing **100** (this may also be referred to herein as the "compressor housing" or "compressor casing") of any appropriate shape (e.g., annular shape) having inside and outside surfaces **107**, **120**, inlet and outlet ends **104**, **108** (note that either or both of the inlet and outlet ends **104**, **108** may include multiple inlets or outlets), and a central axis (not shown) running through the center of the stator casing **100**. As seen, the stator casing **100** may be divided into first and second halves **115**, **116**. A plurality of stator rows or sections **102** may be disposed on the inside surface **107**, each of which may include a series or set of stator vanes or blades **114**.

[0060] In assembly, the first and second halves **115**, **116** of the housing may be interconnected together (e.g., via bolts and apertures, not labeled) such that the stator casing **100** surrounds the shaft **110** and rotor sections **112** and a longitudinal axis (not shown) of the rotating shaft **110** is coincident with the central axis of the stator casing **100**. At this point, the stator sections **102** and rotor sections **112** may alternate and the rotor sections **112** may be operable to rotate in the spaces between the stator sections **102**. As will be appreciated, the angles of each of the stator and rotor sections **102**, **112** may also alternate. Further, the various stator and rotor sections **102**, **112** may have different spacing (e.g., blade density) as well as different angles from the previous rows of blades.

[0061] When the rotor blades **113** turn relative to the stator blades **114**, air advances from the inlet end **104** of the stator casing **100** through the multiple rows of stator and rotor blades **114**, **113** and discharges through the compressor outlet end **108**. As the air advances through the compressor **20**, the air may be compressed from ambient pressure to over 100 psi. However, the compression pressure may vary between different engines. In addition to being compressed, the friction of the blades rotating and air passing over the blades applies significant heat to the air. For instance, air entering at ambient temperature of approximately 518.67° R may be heated to a temperature over 1000° R. Again, the temperature increase may vary between different engines.

[0062] The increase in the temperature of the air as it passes through the compressor **20** results in the air expanding and thus working against its compression. Stated otherwise, the addition of heat to the compressed air requires that the engine supply more compression power to achieve the desired output pressure. Accordingly, utilities disclosed herein are directed to reducing the temperature gain of air flowing through the compressor to reduce compression power requirements and thereby increase the available shaft output power of the engine.

[0063] The stator casing **100** on this particular engine is substantially exposed and thereby provides a surface for extracting heat from the compressor and thus lowering the compressor discharge temperature of compressed air exiting the compressor. That is, in a first embodiment, a heat exchange system is provided for cooling the outside surface **120** of the stator casing **100** without significantly altering the existing engine. In this arrangement, the stator blades **114**, which are affixed to the stator casing **100** (i.e., are in thermal contact with the stator casing **100**) may function as "cooling fins". That is, each stator blade **114** may absorb heat which may be extracted through the root of the stator blade **114** to the stator casing **100**, which is cooled. This effectively forms a surface intercooling arrangement where heat is removed from the compressed air without having to divert the air out of the compressor **20** and through a heat exchanger between compression stages and while limiting the aforementioned problems associated therewith.

[0064] With reference now to FIG. **5**, a cooling or fluid jacket **200** (e.g., first heat exchanger) may be attached to the outside surface **120** of the stator casing **100** to provide surface intercooling to the stator casing **100** (i.e., the cooling jacket is operable to circulate a coolant over the outside surface **120** of the stator casing **100** to absorb thermal energy from the compressor **20** and transfer thermal energy away from the compressor **20**). The cooling jacket **200** may be a counter flow heat exchanger and may include an inlet port or fluid inlet **202** near the base of the stator casing **100** (e.g., substantially adjacent to the compressor outlet end **108**) and an outlet port or fluid outlet **204** near the inlet end **104** of the compressor **20**. A heat exchange medium or working fluid, such as water, glycol and/or other appropriate fluids (e.g., coolant), may flow or be passed or moved axially through the cooling jacket **200** along at least one coolant fluid path from the fluid inlet **202** to the fluid outlet **204**. The fluid inlet **202** and fluid outlet **204** may be fluidly connected to a second heat exchanger (e.g., radiator **220**) for removing heat or thermal energy from the coolant upon the coolant exiting the first heat exchanger. It will be appreciated that various pumps and/or compressors may be included with the cooling jacket and heat exchangers to circulate the working fluid through the heat exchanger **200** and the radiator **220** where the heat extracted from the compressor **20** may be rejected into the atmosphere. In aircraft applications, such a radiator **220** may be advantageously positioned in the airflow of the aircraft to provide improved heat rejection. Furthermore, the size and weight of this radiator may be significantly reduced in comparison to a standard intercooling radiator which typically must be sized to accommodate the mass fluid flow of the compressor. Stated otherwise, the mass flow of the cooling fluid is typically significantly less than the mass flow rate of the compressor **20**.

[0065] FIG. **6**A illustrates a more detailed cross-sectional view of the cooling jacket of FIG. **5**. As can be seen, the cooling jacket **200** may include an inlet header **210** including one or more inlet ports or fluid inlets **202**, an outlet header **220** with one or more outlet ports or fluid outlets **204**, and a core or cavity **240** that extends between the inlet and outlet headers **210**, **220** and that may serve as at least one coolant fluid path that is fluidly interconnected to the fluid inlet and outlets **202**, **204**. It will be appreciated that the cooling jacket **200** may include at least two substantially identical sections (i.e., similar to the first and second halves **115**, **116** of the stator casing **100**) that bolt or otherwise attach together around the stator casing **100** (or compressor housing). Such separate sections may be in fluid communication, or each section may form a separate cooling jacket or heat exchanger having its own inlet, outlet and core.

[0066] With continued reference to FIG. **6**A, cooled fluid from the radiator **220** (the radiator **220** is shown in FIG. **5**) is introduced into the core **240** via the inlet **202** of the inlet header **210**, passes through the core **240** and into the outlet header **220**, and is removed from the core **240** or otherwise passes out of the core **240** via the fluid outlet **204**. In other arrangements, multiple fluid inlets and outlets may be disposed about the periphery of these headers. The core **240** of the cooling jacket **200** may include an outer wall **230** and an inner wall **232** that is in direct contact with the housing **100** of the compressor. A thermally conductive paste or adhesive may be disposed between the inner wall **232** and the outer surface **220** of the stator casing **100** to improve thermal contact between these components. In other embodiments, the inner wall **232** may be absent such that the working fluid is in direct contact with the outside surface of the stator casing **100**. That is, the outer surface **120** of the stator casing **100** may form the inside wall of the cooling jacket **100** so that the core **240** is in the form of a fluid tight cavity that is formed by the outer surface **120** and the outer wall **230**.

[0067] The core **240** may be devoid of any structure to form a simple fluid jacket and/or may include various channels and/or structures for directing fluid flow between the headers and/or for conducting heat from the stator casing **100** or inner wall **232** into the core. This typically improves the transfer of heat from the stator casing **100** to the working fluid. In a further embodiment illustrated in FIG. **6**B, the cooling jacket **200** may be formed as a multistage cooling jacket or heat exchanger. That is, rather than the working fluid being introduced at the base of the stator casing **100** and passing over the entire length of the stator casing, multiple separate stacked cooling jackets **250** may extend over the length of the stator casing **100**. Each cooling jacket **250** may include at least one fluid inlet **202**, at least one fluid output **204** and a core **240** which may serve as a coolant or fluid path. In such an arrangement, the working fluid at the inlet of each cooling jacket **250** stage may be cool fluid received from the radiator **220**. Furthermore, as the working fluid in the stacked heat exchangers passes shorter distance over the stator casing **100**, the temperature rise of the working fluid may be reduced. In this regard, the temperature differential between the working fluid and the stator casing **100** may be increased, and the heat transfer between the working fluid and the stator casing **100** may be improved. Furthermore, by reducing the outlet temperature of the working fluid, the radiator **220** may provide better heat rejection. In one arrangement, a different one of the cores **240** or fluid paths may be aligned with a different one of the stator sections or sets **102**.

[0068] As discussed previously, the stator blades **114** effectively function as "fins" that can absorb thermal energy generated while the airstream is being compressed and that can be subsequently absorbed by a cooling jacket **200** or other heat exchanging device to cool the stator structure **103** and thus the compressor as a whole. However, the stator blades **114** themselves may function poorly as cooling fins by sometimes contributing only 13% of the effective surface area of the inside of the compressor housing whereas the actual blade surface may be 55% of the combined area of the inside surface of the compressor housing and blade surfaces. For instance, when the stator blades **114** are only about 1 mm thick on average and are made of stainless steel, such stator blades **114** may have a collective "fin efficiency" of about 11%. Thus, improving the thermal performance of the stator blades **114** may lead to improved effectiveness of the surface intercooling systems and methods.

[0069] For instance, where the stator blades **114** are made of aluminum and are 3 mm thick at their base (which increases the effective transfer area of the stator blades **114**), the fin efficiency could be increased to approximately 65%. By improving the fin efficiency, more heat may be removed from the stator or compressor housing thereby further improving system performance. In the above example, the corresponding efficiency gain is 1.6% and the power increases by 9.5% in a surface intercooling system without regeneration. To further increase the cooling efficiency, the stator blades may be constructed of a material having a higher thermal conductivity, by thickening the blades, by actively cooling the blades, by modifying the turning radius, height, angles, width, etc. of the stator blades **114**, and/or the like. For instance, the thickness of the stator blades **114** may be increased to improve the conduction of each blade and thereby increase engine power. In one arrangement, the stator blades **114** may be formed of a copper alloy that may include, for example, a tough/wear resistant coating (e.g., chrome, etc.).

[0070] FIGS. **7-12** illustrate a perspective view of a stator structure **103'** and various components thereof for a compressor according to another embodiment that serves to increase the heat absorbing capacity of the stator blades and the compressor as a whole (e.g., whereby such absorbed heat may be removed by one or more cooling jackets or heat exchangers disclosed herein). Corresponding components between the embodiments of FIGS. **4-6**B are identified by common reference numerals. Those corresponding components that differ in at least some respect from the embodiments of FIGS. **4-6**B are identified by a "single prime" designation in FIGS. **7-12**. As with the stator structure **103**, the one or more components of the stator structure **103'** may be of any appropriate size, shape, configuration and/or type. While only a first half, portion or section of the stator structure **103'** (and various components thereof) has been illustrated in FIGS. **7-12**, it should be appreciated that another substantially identical half or portion may be appropriately interconnected with the first half or portion (e.g., via bolts, fasteners) to form a complete stator structure **103'** that may be placed around a rotor structure to form a compressor for a turbine engine.

[0071] As seen in FIGS. **7-8**, a series of fins **130** may interconnect adjacent stator sections **102'** so as to extend away from at least one of first and second opposing surfaces (not labeled) of the stator blades **114'** to further increase the thermal absorbance of the stator sections **102'**. As can be appreciated, the fins **130** increase such thermal absorbance by increasing the overall surface area of each stator blade **114'** that is exposed to thermal energy generated by the compression process. The additional heat absorbed by the fins **130** may be removed or absorbed by a cooling jacket **200'** (discussed in more detail below). Removing thermal energy from a compressor in this manner may allow the approximation of isothermal compression which may provide for a more efficient turbine engine.

[0072] As shown, each of the fins **130** may be generally parallel to an overall direction of airflow from the inlet end **104** towards the outlet end **108** of the stator structure **103'**. Additionally, each of the fins **130** is generally devoid of an airfoil shape (i.e., even though the fins **130** may curve around the inside surface **107** of the stator casing **100**, the fins **130** may have opposing first and second surfaces (not labeled) that are generally parallel to each other). It has been found that this arrangement reduces pressure losses that may be caused by the stator blades **114'**. Other arrangements of fins **130** are also envisioned to increase the thermal absorbance of the stator blades **114'**. For instance, each of the stator blades **114'** may include its own fins **130** extending away from one or more sides of the stator blades **114'** (as opposed to a single fin interconnecting multiple stator blades **114'** as shown in FIGS. **7-8**). In further arrangements, the fins **130** themselves may include additional features (e.g., fins, ridges) extending therefrom to further increase the thermal absorbance of the stator blades **114'**.

[0073] With continued reference to FIGS. **7-8**, each of the stator sections **102'** may also include a stator ring **132** and/or a shroud cover **134** that are respectively appropriately interconnected (e.g., via welding, fasteners, openings) to first and second ends **136**, **138** of the stator blades **114'**. The stator ring **132** and shroud cover **134** may at least partially form a "frame" for each stator section **102'** and allow each stator section **102'** to be conveniently interconnected to the inside surface **107** of the stator casing **100**. Additionally, the stator ring **132** and shroud cover **134** may increase the thermal absorbency of the stator sections **102'**. It should also be appreciated that the stator sections **102'** may be thermally interconnected to the stator casing **100** in any appropriate manner. For instance, the inside surface **107** of the stator casing **100** may include a series of spaced grooves **140**, and each groove **140** may be sized to receive the stator ring **132** of a respective stator section **102'**. In one arrangement, the stator rings **132** and grooves **140** may include corresponding dovetail or other appropriate features that allow the stator rings **132** to be slid or otherwise inserted into the grooves **140** and thereby secured to the inside surface **107** of the stator casing **100**.

[0074] Once the stator ring **132** of a stator section **102'** has been received in a groove **140**, the stator ring **132** (and thus the entire stator section **102'**) may be in thermal contact with the stator casing **100**. See. FIG. **11**. As will be discussed in more detail below, this arrangement allows a cooling jacket **200'** that is placed in thermal contact with the outside surface **120** of the stator casing **100** to more efficiently absorb thermal energy from the stator structure **103'** and thereby cool a compressor utilizing the stator structure **103'**.

[0075] With reference now to FIGS. **7** and **9-12**, one or more cooling jackets **200'** may be disposed about the stator casing **100** and in at least partial thermal contact with the outside surface **120** of stator casing **100** to remove or otherwise absorb thermal energy from the stator sections **102'** and thereby reduce the temperature of the airstream within a compressor utilizing the stator structure **103'**. As shown, a different one of the cooling jackets **200'** may be aligned with a different one of the stator sections **102'**. In other arrangements, the cooling jackets **200'** may not necessarily be aligned with a stator section **102'**, a single cooling jacket **200'** may exist for the entire stator casing **100**, and the like. In any event, one or more of the cooling jackets **200'** may be collectively considered a "heat exchanger," "cooling jacket" or "fluid jacket".

[0076] Each cooling jacket **200'** may be in the form of first and second portions or halves which may be appropriately interconnected to essentially form a ring about the outside surface **120** of the stator casing **100**. In this regard, each portion of the cooling jacket **200'** may generally be in the form of a half-circle (or other shape to conform to the outside surface **120** of the stator casing **100**) with first and second ends **252**, **254**, each having at least one bore **256** (or other fastening or attaching feature). The bores **256** (or other fastening features) on the first and second ends **252**, **254** of one portion of the cooling jacket **200'** may be aligned and secured together with the bores **256** of another portion of the cooling jacket **200'** via any appropriate fasteners (e.g., via bolts). In other arrangements, the cooling jacket **200'** may be a single piece designed to extend substantially around the entire outside surface **120** of the stator casing **103'** or only around a portion of the outside surface **120**.

[0077] In any arrangement, the cooling jacket **200'** may broadly include the outer wall **230** and a core or cavity **240** formed by the outer wall **230** and the outer surface **120** of the stator casing **100** (see FIG. **11**) to form at least one coolant or fluid path through the cooling jacket **200'**. Additionally, ports (e.g., fluid inlets and outlets) may be fluidly interconnected to the cavity **240** (discussed below) for receiving and passing a circulated coolant through the cavity **240** which may be operable to remove or absorb thermal energy generated by the airstream via the outer surface **120** of the stator casing **100**. As shown, the cooling jacket **200'** may include at least one dividing wall **258** extending from the outer wall **230** that divides the cavity **240** into first and second cavities **260**, **262**. Each of the first and second cavities **260**, **262** may include at least two ports **264** (e.g., a fluid inlet and a fluid outlet) for receiving and passing a circulated coolant through each of the first and second cavities **260**, **262** (i.e., the fluid inlet port **264** may receive coolant and pass the coolant into one of the cavities and the fluid outlet port **264** may remove or pass coolant from one of the cavities). As the cavity **240** has been divided up into at least first and second cavities **260**, **262**, a greater portion of the outside surface **120** of the stator casing will be in contact with low temperature coolant (e.g., near ambient temperature) or in other words coolant of a higher thermal energy absorbing capacity than if the same coolant traveled over the entire outside surface **120** via a single cavity having first and second ports **264**.

[0078] Of course, additional dividing walls **258** and/or fluid paths may be appropriately incorporated into the cooling jacket **200'** to form additional cavities each having fluid inlets and fluid outlets. For instance and with reference to FIG. **10**, each of the first and second cavities **260**, **262** may be divided into first and second sub-cavities (not shown or labeled) via including a dividing wall that extends between the dividing wall **258** and the outer wall **230** (e.g, it extends generally perpendicularly to the dividing wall **258** and outer wall **230**) of each of the first and second cavities **260**, **262**. As seen in FIG. **10**, additional ports may be included for such additional cavities. In some embodiments, the cooling jacket **200'** may include an inner wall (not shown) to enclose the cavities and such inner wall may be disposed over and in contact with the outside surface **120** of the stator casing **100**.

[0079] In assembly, first and second portions of a cooling jacket **200'** may be aligned with a respective stator section **112'** or portion of a stator section **112'** (see FIGS. **7** and **11**) and secured together around the outside surface **120** of the stator casing **100**. The ports **264** may be fluidly interconnected (e.g., via tubing and appropriate connectors) to another (e.g., external) heat exchanger (e.g., radiator, see FIG. **5**), pump(s) and the like to form a circuit through which the coolant may be circulated. After absorbing heat or thermal energy from the outside surface **120** of the stator casing **100** via the various cavities, the coolant may travel to the radiator whereby such thermal energy may be released before the coolant travels returns to the various cavities to again absorb thermal energy.

[0080] It should be appreciated that other manners of improving or increasing the ability of the compressor to absorb thermal energy which may be removed or otherwise cooling the compressor are disclosed herein. For instance, stator blades may be actively cooled via either solid state conduction and/or forced circulation of liquid cooling within each stator blade. With reference again to FIG. **11**, one or more stator blades **114'** may include an intake or inlet passage **142** in fluid interconnection with a return or outlet passage **144**. The intake and return passages **142**, **144** may be designed to receive a circulated coolant therethrough that absorbs thermal energy received by the outer surfaces of the stator blade **114'** and removes such thermal energy from the stator blade **114'**.

[0081] For instance, the intake and return passages **142**, **144** of each stator blade **114'** may be respectively fluidly interconnected to first and second ports **143**, **145** that extend between the inside and outside surfaces **107**, **120** of the stator casing **100**, and such first and second ports **143**, **145** may be respectively fluidly interconnected to the first and second cavities **260**, **262** of the cooling jacket **200'** via various pairs of ports **204** (e.g., see FIG. **9**). Of course, multiple pairs or sets of first and second ports **143**, **145** may be included through the stator casing **100** which may be respectively fluidly interconnected to multiple stator blades **114'**. In one arrangement, each of the ports **264** that are fluidly interconnected to the first cavity **260** may be considered a "fluid inlet" and each of the ports **264** that are fluidly interconnected to the second cavity **262** may be considered a "fluid outlet" (again, also see FIG. **9**). In this regard, coolant may be pumped or otherwise travel into the first cavity **260** via its respective ports **264** and then into the intake passage(s) **142** of the stator blade(s) **114'** via the port(s) **143**. In this regard, the cooling jacket(s) **200'** may function as manifolds. Thereafter, the coolant may enter the return passage(s) **144** and then the second cavity **262** via the port(s) **145**. Coolant may then exit the ports **264** of the second cavity **262** whereby it may travel through tubing or passages to an external heat exchanger or radiator **220** (see FIG. **5**). The coolant may release heat via the radiator **220** and then return back to the port(s) **264** of the first cavity **260** of the cooling jacket(s) **200'**. As can be appreciated, this arrangement may allow coolant to be circulated through the cooling jacket(s) **200'** and the stator blade(s) **114'** and eventually through one or more external heat exchangers to cool the compressor and the airstream traveling therethrough by facilitating the transfer of thermal energy away from the stator blades **114'** and the compressor as a whole.

[0082] Other arrangements are also envisioned and encompassed by the disclosure herein. In one arrangement, the stator blades **114'** may include only a single passage, and the passages of respective stator blades **114'** may be fluidly interconnected, for instance, via passages in the stator ring **132** and/or shroud cover **134**. Alternatively, the stator blades **114'** may each include multiple intake and return passages **142**, **144**. In another arrangement, each of the first and second cavities **260**, **262** may be divided into multiple cavities, each being "served" by a respective port **264**. In this regard, the port **264** of one of the multiple cavities of the first cavity **260** may provide "fresh coolant" to only a few stator blades **114'** (or even a single stator blade **114'**) such that the coolant entering the stator blades **114'** has a higher heat or thermal energy absorbing capacity. A similar arrangement may be embodied in the second cavity **262**. Although the inlet and return passages **142**, **144** are illustrated as generally extending along a length of the stator blade **114'**, other arrangements of the passages and additional passages are also envisioned. In other embodiments, some stator blades **114'** need not include such passages at all. In a further arrangement, the stator structure **103'** need not include the cooling jackets **200'** and instead only includes intake and return passages in at least some of the stator blades **114'**. In this regard, the coolant supplied by the external heat exchanger/radiator may be directed directly into the stator blades **114'** (e.g., via a manifold) instead of first through the cooling jackets **200'**.

[0083] With reference now to FIGS. **13-14**, a stator blade **114"** according to another embodiment is illustrated that may be used with any of stator structures or compressors disclosed herein. Corresponding components between previous embodiments are identified by common reference numerals. Those corresponding components that differ in at least some respect from previous embodiments are identified by a "double prime" designation in FIGS. **13**-**14**. As shown, the stator blade **114**" may include a series of cooling fins **130**" that extend away from opposing first and second sides **146**, **148** of the stator blade **114**" to further increase the ability of the stator blade **114**" to absorb thermal energy from the air-stream in the compressor. That is, instead of cooling fins that interconnect multiple stator blades (e.g., as in FIG. **8**), each stator blade **114**" may include its own cooling fins **130**". As shown, the cooling fins **130**" may extend from one or more of the first and second opposing sides **146**, **148**. It has been found that orienting the cooling fins **130**" to be parallel to the air stream improves the heat transfer from the air to the stator blade **114**". For instance and while not limiting, the stator blade **114**" may include four cooling fins **130**" on each of the first and second opposing sides **146**, **148** that are each a passive plate substantially devoid of an airfoil shape.

[0084] As also shown in FIGS. **13**-**14**, one or more surfaces of the cooling fins **130**" may include ridges **150** (e.g., micro ridges) or other structures that are at least partially transversely disposed in relation to the free stream flow **152** of the compressed air through the compressor. Such ridges **150** may disrupt the free stream flow and thereby improve the suction side aerodynamic efficiency and heat transfer of the stator blade **114**". In one embodiment, the stator blade **114**" may produce 2½ times the heat transfer from the compressed air to the stator blade **114**" as compared to a stator blade devoid of such ridges **150**, cooling fins **130**" and/or cooling passages **142**, **144**.

[0085] Moreover, the cooling fins **130**" on the first side **146** may be offset from those on the second side **148** (see FIG. **13**). This arrangement allows the cooling fins **130**" of adjacent stator blades **114**" to overlap, and thus allows the inclusion of a greater number of stator blades **114**" in a stator section **102**.

[0086] The ability to provide cooling to the compressor housing can significantly reduce the compressor air outlet temperature. That is, compressed air temperature rise may be significantly reduced in comparison to the temperature rise in a conventional turbine engine. This reduced compressor output temperature is a modification of the basic gas turbine Brayton cycle. In a theoretical limit, compression may be done at constant temperature or 'isothermal' compression with the remainder of the cycle being the same as the Brayton cycle—constant pressure combustion and isentropic expansion. This modified cycle is referred to herein as the 'Iso-Compression' cycle, which utilizes isothermal or reduced temperature rise compression. Necessarily, the Iso-Compression cycle requires the removal of heat from the compressor during the compression process.

[0087] For purposes of this disclosure, a discussion of the Iso-Compression cycle is provided where reduced temperature or isothermal compression is achieved using a large number (e.g., 12) of idealized counter flow heat exchangers to remove heat from the compressor housing temperature after each small compression step. These heat exchangers are idealized in that the coolant temperature may remain constant for each compression step. For this analysis, the coolant inlet temperature may be maintained at ambient air temperature. In the real system where coolant fluid flows parallel to the compressor air flow, there may be a temperature rise of the coolant. In this case, the first step in the compression process would see an ambient coolant temperature, and this temperature would be increased as compression proceeds to the compressor discharge.

[0088] However, the following discussion is pertinent to the real system as the stator casing cooling effectiveness necessary to achieve the desired improvement of engine power is well within the range of effectiveness that may be achieved by the heat exchangers discussed above. The engine cycle used in the analysis is that of the Rolls Royce Model 250-C20B turboshaft engine. Results are shown for the levels of the compressor cooling effectiveness ranging from zero (the basic engine without compressor cooling) to 1.0 (the theoretical upper limit used to model isothermal compression). The actual effectiveness of the real system lies between these extremes. All performance data in the sections below are for sea level static, standard day conditions at maximum rated power for the engine. As noted above, compressor cooling has been modeled by assuming a series of 12 heat exchanges where the inlet coolant temperature for each step was maintained at the same temperature—standard day, sea level ambient temperature (518.67° R).

[0089] FIG. **15**A shows compressor air discharge temperature as a function of the assumed cooling effectiveness. Effectiveness is defined in the usual way for a heat exchanger, and it is measure of the exit temperature of the hot side flow (the compressor flow) compared to the cold side (coolant) inlet temperature. With an effectiveness of zero, the resulting data are that for the basic Model 250 engine. With an effectiveness of 1.0, the data represents isothermal compression where the compressor discharge temperature for this case is 518.67° R. The non-linearity of the curve shows that a significant effect in compressor discharge temperature reduction is achieved even with a cooling effectiveness as low as 0.2.

[0090] The cooling effectiveness impact on output shaft power is shown in FIG. **15**B where the same non-linearity is demonstrated. As shown, a cooling effectiveness of 0.4 can be considered a reasonable upper design limit since power gains above this level are diminishing. Note in FIG. **15**B that the power at a compressor cooling effectiveness of zero is 420 hp, which is the rated power of the Model 250-C20B engine.

[0091] The heat to be removed from the compressor housing per unit time is shown in FIG. **15**C. This data was computed assuming an airflow rate of 3.454 lb/sec through the compressor—the rated value for the Model 250-C20B engine. As with all of the previous curves, this one is highly non-linear becoming asymptotic to the maximum value of 311 Btu/sec as the cooling effectiveness goes to 1.0. As shown in FIGS. **15**A-C, a cooling system that provides an effectiveness of 0.4 provides significant improvement in shaft output power of the engine and that increasing the effectiveness above this level provides marginal additional benefits.

[0092] A T-S diagram for the standard Brayton gas turbine cycle and the compressor-cooling Iso-Compression cycle is shown in FIG. **16**. In FIG. **16**, the statement point designations are defined as follows:

[0093] **2** Compressor Face/Inlet

[0094] **3** Compressor Discharge—Standard Cycle

[0095] **3'** Compressor Discharge—Iso-Compression Cycle

[0096] **4** Combustor Exit—Turbine Inlet

[0097] **5** Turbine Discharge—Standard Cycle

[0098] **5'** Turbine Discharge—Iso-Compression Cycle

[0099] **6** Power Turbine Discharge

[0100] Entropy is a relative parameter, and it is set at zero at the ambient condition (i.e., sea level, standard day). The cooling effectiveness for Iso-Compression Cycle has been set to 0.4, so there is a slight rise in temperature along with the reduction in entropy after compression. The reduced work required for compression with Iso-Compression Cycle 1 is reflected in the increased temperature at state point **5'** over that at state point **5**. This allows more work to be developed in the power turbine (**5'** to **6**). This is the shaft output power. Increased fuel energy input is shown in the change in temperature from state point **3'** to state point **4**. This demonstrates why the thermal efficiency of an engine that utilizes the various systems, apparatuses and methods disclosed herein for cooling a compressor and/or the airstream flowing therethrough (e.g., cooling jackets, stator blade designs, etc) remains approximately constant because the increase in power output is matched by the increase in fuel input. That is, the various systems, apparatuses and methods disclosed herein for cooling a compressor and/or the airstream flowing therethrough generally have little or no effect on engine efficiency. The advantage of such systems, apparatuses and methods is that the required compression power is reduced thereby increasing output shaft power. The reason for the non-effect on efficiency is that as the compressor discharge temperature has been reduced, more fuel energy must be added to the compressed air to raise the gas temperature up to the required turbine inlet temperature.

[0101] To improve engine efficiency and power output, any appropriate manner of achieving regeneration may be included along with one or more of the various systems, apparatuses and methods disclosed herein for cooling a compressor and/or the airstream flowing therethrough (e.g., cooling jackets, stator blade designs, etc.). As discussed previously, regeneration is the use of a heat exchanger to transfer heat from an engine exhaust stream to the compressor discharge air (thus preheating the compressor discharge air) in a turbine engine such that less fuel energy is required to achieve the required turbine inlet temperature for the compressed air. By recovering some of the energy usually lost as waste heat, a regenerator can make a gas turbine engine significantly more efficient.

[0102] For instance, the compressed air ducts **22** of the gas turbine engine **10** of FIGS. **1-3** (which may incorporate any of the cooling jackets, stator blades, etc. disclosed herein) may be readily tapped, replaced and/or rerouted through a recuperator **60** that is appropriately incorporated into ducting **52** that is connected to the exhaust ducts of the engine (i.e., from the turbine section **40**). See FIG. **17**. As discussed previously, a recuperator is a heat exchanger that allows regeneration by transferring heat via convention and solid conduction across walls that separate turbine stage exhaust gas from compressor discharge air. In any event and once rerouted, air that is drawn into the compressor **20** (not labeled in FIG. **17**) and compressed may be discharged into a pair of compressor outlet ducts **24** (or other number of outlet ducts **24**) that extend between the compressor discharge and the inlet end of the recuperator **60**.

[0103] After the compressed air exits the outlet ducts **24**, it may enter one or more passages in a core of the recuperator **60** whereby the exhaust gases may be directed through separate but adjacent passages (e.g., the passages directing the compressed air may share common walls or with the passages directing the exhaust gases). After the compressed air has been heated by the exhaust gases in the core, the heated compressed air may then pass from the recuperator **60** through outlet ducts **26** and eventually into an inlet end of the combustor **30**. It should be appreciated that the recuperator may be constructed so as to be, for instance, a "counter-flow", "cross-flow", or other type of recuperator. U.S. patent application Ser. No. 12/650,857, entitled "Recuperator for Gas Turbine Engines," illustrates one non-limiting exemplary embodiment of a recuperator for use with a gas turbine engine.

[0104] Turning to FIG. **18**, one embodiment of a core **62** for a recuperator (e.g., recuperator **60**) is illustrated that is operable to carry fluid flow (e.g., compressed air) between inlet and outlet headers (not shown) of the recuperator **60** while allowing heating of the compressed air. As seen, the core **62** may include multiple gas channels **64** and multiple compressed air channels **66**, which may laterally connect. The exhaust gas and compressed air may be appropriately routed through the at least one exhaust gas and compressed air channels **64**, **66** in a counter-flow arrangement (e.g., by routing the compressed air through one end of the core **62** and the exhaust gas through an opposite end of the core **62**). The core **62** may be formed of any appropriate materials, such as, but not limited to, copper/moly, stainless steels, aluminums, nickel alloys and/or the like.

[0105] The overall length of core **62** (or other types of cores) may be selected as a function of the effectiveness, mass flow and pressure drop of the heat exchanger/recuperator. In relation to the effectiveness of the recuperator, it is noted that the effectiveness of a counter-flow recuperator/heat exchanger is defined by the temperature differential of the exhaust gases (i.e., Ex) across the recuperator divided by the temperature differential of the compressed air (i.e., CA) across the recuperator. Specifically:

$$\text{Effectiveness} = \frac{(TEx_{in} - TEx_{out})}{(TCA_{out} - TCA_{in})}. \qquad \text{Eq. 1}$$

[0106] Simply stated, the effectiveness is a fraction of the total temperature difference of the flows into the hot side and cold side of the heat exchanger. When the effectiveness is 1.0, the hot side out temperature of the compressed air would equal the exhaust gas inlet temperature. However, this can never happen as an infinite heat exchange surface would be required. However, while a 1.0 effectiveness is not achievable, use of the core **62** may allow for achieving 0.6, 0.7, 0.8 or greater effectiveness while maintaining a compact and light weight recuperator. It should be appreciated that by having an effectiveness of over at least 0.6, engine efficiency may be increased significantly (e.g., 10-40%).

[0107] The combination of recuperation with the various systems, apparatuses and methods disclosed herein for cooling a compressor and/or the airstream flowing therethrough (e.g., cooling jackets, stator blade designs, etc) may result in a significant increase in both power and efficiency. This is illustrated in FIG. **19** in conjunction with the following analysis where the Temperature-Entropy (TS) diagram of compressor cooling and recuperation (i.e., Iso-Compression Cycle 2) is compared with compressor cooling without recuperation (Iso-Compression Cycle). The recuperator for Iso-Compression Cycle 2 has the following design characteristics:

[0108] Recuperator Effectiveness 0.7

[0109] Cold Side (compressor discharge air) pressure drop fraction 0.02

[0110] Hot Side (power turbine discharge gas) pressure drop fraction 0.02

[0111] The impact on performance by incorporating a recuperator into the system may be seen by comparing the TS diagram for compressor cooling only with that for compressor cooling coupled with recuperation (see FIG. **19**). Again, the cooling effectiveness is 0.4. The statement point designations for FIG. **19** may be defined as follows:

[0112] **2** Compressor Face/Inlet

[0113] **3'** Compressor Discharge—Iso-Compression Cycle & Iso-Compression Cycle 2

[0114] **3"** Recuperator Discharge (cold side)—Iso-Compression Cycle 2

[0115] **4** Combustor Exit—Turbine Inlet

[0116] **5'** Turbine Discharge—Iso-Compression Cycle

[0117] **5"** Turbine Discharge—Iso-Compression Cycle 2

[0118] **6'** Power Turbine Discharge—Iso-Compression Cycle

[0119] **6"** Power Turbine Discharge—Iso-Compression Cycle 2

[0120] **7** Recuperator Discharge (hot side)—Iso-Compression Cycle 2

[0121] The reduced compressor discharge temperature shown in FIG. **19** for compressor cooling does not change, and thus state point **3'** is identical for both Iso-Compression Cycle and Iso-Compression Cycle 2. For Iso-Compression Cycle 2, the change in state point from **3'** to **3"** represents the heat added to the compressor discharge air from the recuperator. One can see the significant change in fuel energy by comparing the heat added in Iso-Compression Cycle (state point **3'** to state point **4**) and the heat added in Iso-Compression Cycle 2 (state point **3"** to state point **4**). The work in the high pressure turbine from state point **4** to either state point **5'** or **5"** is identical for the two cycles because the compressor work has not changed. Thus, state points **5'** and **5"** are identical.

[0122] The work in the power turbine from state point **5'** to **6'** for Iso-Compression Cycle and from state point **5"** to **6"** for Iso-Compression Cycle 2 is almost identical. However, for Iso-Compression Cycle 2 there is a back pressure to the power turbine as a result of the pressure drop in the recuperator, so state point **6"** is at a slightly higher temperature than that for state point **6'**. Thus there is a reduction in the shaft power for Iso-Compression Cycle 2. With a compressor cooling effectiveness of 0.4, there is a 6.4% loss in power when comparing the shaft output power of Iso-Compression Cycle to that of Iso-Compression Cycle 2, though the Iso-Compression Cycle 2 still provides a power increase over an engine that does not incorporate compressor cooling. As can be appreciated, recuperation provides an improvement in fuel economy. With a compressor cooling effectiveness of 0.4, the thermal efficiency increases from 0.2210 for Iso-Compression Cycle 1 to 0.3667 for Iso-Compression Cycle 2.

[0123] Ideally all of the compression heat would be removed and the temperature would remain constant throughout compression (i.e., there would be isothermal compression). The ideal, of course, cannot be actually achieved as it would require an infinite heat transfer surface area or an infinite heat transfer coefficient. Computational fluid dynamics (CFD) analyses have shown that the compressor airflow friction pressure drop, air turning angles, and flow paths are essentially unaffected by the heat transfer blading. For a preliminary design of finned stator heat transfer blades for a Rolls-Royce C250B engine, CFD analysis predicted 47% cooling effectiveness (47% of heat removal required for isothermal compression). Without recuperation, internal-intercooling at 47% effectiveness raised the engine efficiency from 21.4% for the adiabatic compressor to 23.9%. With 70% effective recuperation, internal-intercooling at 47% raised engine efficiency from 30.8% to 35.5% and horsepower increased from 400 to 471 with the same air mass flow rate. These figures illustrate that the combined effect of recuperation and internal-intercooling of the compressor creates a larger heat transfer-driving temperature difference across the recuperator.

[0124] In summary, the various systems, apparatuses and methods disclosed herein for cooling a compressor and/or the airstream flowing therethrough (e.g., cooling jackets, stator blade designs, etc) with an effectiveness of 0.4 provides a small improvement in engine efficiency of approximately 1.0% and a modest increase in power of approximately 6.1%. The addition of regeneration (e.g., via a recuperator) may provide additional engine efficiency and an increase in power. Stated otherwise, use of the various systems, apparatuses and methods disclosed herein for cooling a compressor and/or the airstream flowing therethrough with or without regeneration provides significant benefits.

[0125] It should also be appreciated that the various embodiments and arrangements disclosed herein may be used in conjunction with each other in various manners and should not be construed in isolation. For instance, the stator blades **114'** of FIGS. **7-8** may include the cooling fins **130"** of FIGS. **13-14** instead of the cooling fins **130** that interconnect multiple stator blades **114'**. As another example, the cooling jackets **200'** of FIGS. **7** and **9-12** may be used in conjunction with the stator casing **100** of FIGS. **5-6**B instead of the cooling jacket **200**. Numerous other arrangements are also envisioned and encompassed herein.

What is claimed is:

1. A stator structure for a compressor, comprising:

a stator casing comprising inside and outside surfaces, and inlet and outlet ends, wherein air generally moves in an air flow direction from the inlet towards the outlet end;

a plurality of spaced stator sections extending from the inside surface of the stator casing, wherein each stator section comprises a plurality of stator blades, and wherein each stator blade comprises:

an intake passage extending along a length of the stator blade for receiving a flow of coolant into the stator blade; and

a return passage, fluidly interconnected to the intake passage, and extending along the length of the stator blade for passing the flow of coolant out of the stator blade, wherein the coolant is operable to absorb thermal energy from the stator blade and transfer thermal energy away from the stator blade.

2. The stator structure of claim **1**, wherein each stator section comprises a stator ring that is secured to the inside surface of the stator casing, wherein the plurality of stator blades extend away from the stator ring.

3. The stator structure of claim **1**, wherein each stator blade comprises first and second opposing ends, wherein the first end is interconnected to the inside surface of the stator casing, and wherein the second ends of the stator blades in a stator section are interconnected by a shroud cover.

4. The stator structure of claim **1**, wherein each stator blade comprises first and second generally opposing blade surfaces, wherein at least one of the first and second generally opposing blade surfaces comprises at least one fin extending therefrom.

5. The stator structure of claim **4**, wherein the at least one fin extends between and interconnects at least first and second adjacent stator blades.

6. The stator structure of claim 1, wherein the inside surface of the stator casing comprises a plurality of pairs of ports, each pair comprising a first port that is fluidly interconnected to the intake passage of a stator blade and a second port that is fluidly interconnected to the return passage of the stator blade.

7. The stator structure of claim 6, further comprising a fluid jacket attached to the outside surface of the stator casing that provides coolant to the first port and receives coolant from the second port of each of said pair of ports of the stator casing.

8. The stator structure of claim 7, wherein the fluid jacket comprises a first cavity that provides coolant to the first ports and a second cavity that receives coolant from the second ports.

9. The stator structure of claim 8, wherein the first cavity includes at least one port that receives coolant from a heat exchanger, and wherein the second cavity includes at least one port that passes coolant to the heat exchanger.

10. An apparatus for use with a gas turbine engine, comprising:

an annular housing including inside and outside surfaces, and inlet and outlet ends, wherein air generally moves in an air flow direction from the inlet end towards the outlet end; and

a plurality of spaced sets of stators extending from the inside surface of the housing, wherein each set of stators comprises a plurality of stator blades; and

a first heat exchanger extending around at least a portion of the outside surface of the housing, wherein the first heat exchanger is operable to absorb thermal energy from the housing and transfer thermal energy away from the stator blades.

11. The apparatus of claim 10, wherein the first heat exchanger comprises:

at least one coolant fluid path.

12. The apparatus of claim 11, wherein the coolant fluid path comprises:

an outer wall spaced from the outer surface of the housing to form a fluid tight cavity between the outer wall and the outer surface of the housing;

an inlet port for introducing coolant into the cavity; and

an outlet port for removing coolant from the cavity, wherein coolant is operable to flow through the cavity between the inlet and outlet ports.

13. The apparatus of claim 12, wherein the first heat exchanger comprises:

a plurality of fluid paths, wherein a different one of the fluid paths is aligned with a different one of the sets of stators.

14. The apparatus of claim 13, wherein the first heat exchanger comprises:

an inlet header including a plurality of inlet ports interconnected to at least one of the plurality of fluid paths; and

an outlet header including a plurality of outlet ports interconnected to at least one of the plurality of fluid paths.

15. The apparatus of claim 12, further comprising:

a second heat exchanger interconnected to said inlet and outlet ports for removing heat from coolant upon said coolant exiting said first heat exchanger.

16. The apparatus of claim 15, wherein said second heat exchanger comprises a radiator.

17. The apparatus of claim 11, wherein the first heat exchanger comprises:

first and second sections, wherein the first and second sections are adapted to extend about a respective half of the annular housing and be secured together.

18. The apparatus of claim 11, wherein the first heat exchanger comprises:

a dividing wall extending between the outer wall of the first heat exchanger and the outer surface of the housing to form first and second cavities, wherein each of the first and second cavities comprises:

an inlet port for receiving coolant and passing the coolant into the respective first or second cavity; and

an outlet port for removing coolant from the respective first or second cavity

19. The apparatus of claim 10, wherein at least a portion of the stator blades comprise:

an intake passage extending along a portion of a length of the stator blade for receiving a flow of coolant into the stator blade; and

a return passage, fluidly interconnected to the intake passage, and extending along a portion of the length of the stator blade for passing the flow of coolant out of the stator blade.

20. The apparatus of claim 10, wherein each stator blade comprises first and second generally opposing blade surfaces, wherein at least one of the first and second generally opposing blade surfaces comprises at least one fin extending therefrom.

21. The apparatus of claim 20, wherein the at least one fin extends between and interconnects with an adjacent stator blade.

22. The apparatus of claim 20, wherein said at least one fin further comprises at least one ridge on its surface, where the at least one ridge is oriented at least partially transverse to the air flow direction.

23. A gas turbine engine, comprising:

a turbine section;

a combustion chamber that is fluidly interconnected to the turbine section;

a compressor that is fluidly interconnected to the combustion chamber, wherein the compressor comprises a housing having an inside surface and an outside surface; and

a fluid jacket extending around at least a portion of the outside surface of the housing, wherein the first fluid jacket is operable to circulate coolant over the outside surface of the housing to absorb thermal energy from the compressor and transfer thermal energy away from the compressor.

24. The engine of claim 23, wherein the fluid jacket comprises:

an outer wall spaced from the outside surface of the housing to form a cavity between the outer wall and the outer housing;

an inlet port for receiving coolant and passing the coolant into the cavity; and

an outlet port for receiving coolant from the cavity, wherein coolant is operable to flow through the cavity as the coolant flows between the inlet and outlet ports.

25. The engine of claim 24, wherein the fluid jacket comprises a dividing wall extending between the outer wall of the fluid jacket and the outer housing to form first and second cavities, wherein each of the first and second cavities comprises:

an inlet port for passing the coolant into the respective first or second cavity; and an outlet port for removing coolant from the respective first or second cavity, wherein coolant is operable to flow through the respective first or second cavity as the coolant flows between the inlet and outlet ports.

26. The engine of claim **24**, wherein the fluid jacket comprises first and second sections, wherein the first and second sections are adapted to extend about the outside surface of the compressor and be secured together.

27. The engine of claim **24**, wherein the fluid jacket comprises:

an inlet header including the inlet port, and an outlet header including the outlet port.

28. The engine of claim **27**, wherein the inlet header comprises:

a plurality of inlet ports and the outlet header comprises a plurality of outlet ports.

29. The engine of claim **23**, wherein the fluid jacket further comprises:

a plurality of fluid paths extending around separate portions of the outside surface of the housing.

30. The engine of claim **29**, wherein a different one of the fluid paths is aligned with a different stator section connected to the inside surface of the housing.

31. The engine of claim **23**, further comprising a recuperator that is fluidly interconnected to an outlet of the compressor and an inlet to the combustion chamber, wherein the recuperator uses exhaust gases from the turbine section to heat compressed air exiting the compressor prior to entry into the combustion chamber.

32. A compressor, comprising:

a stator structure, comprising:

a stator casing including inside and outside surfaces, inlet and outlet ends, and a central axis running through a center of the stator casing, wherein air generally moves in an air flow direction from the inlet towards the outlet end;

a plurality of spaced stator sections extending from the inside surface of the stator casing, wherein each stator section comprises a plurality of stator blades; and

a plurality of coolant passages extending through the stator sections, wherein the coolant passages receive coolant that is operable to absorb thermal energy from the stator sections and transfer the thermal energy away from the stator sections; and

a rotor structure, comprising:

a rotatable shaft having a longitudinal axis that is coincident with the central axis of the stator casing; and

a plurality of rotor sections attached to and extending from the rotatable shaft, wherein each rotor section includes a plurality of rotor blades, wherein the rotor sections are disposable between the spaced stator sections.

33. The compressor of claim **32**, further comprising a heat exchanger extending around the outside surface of the stator casing, wherein the first heat exchanger is operable to absorb thermal energy from the compressor and transfer thermal energy away from the compressor.

34. A method for use in cooling a compressor in a gas turbine engine, comprising:

establishing a coolant fluid path that extends at least partially over an outside surface of a housing of the compressor;

first passing a coolant along the coolant fluid path over the outside surface of the housing;

absorbing, using the coolant, thermal energy from the compressor; and

second passing the coolant along the coolant fluid path away from the outside surface of the compressor housing.

35. The method of claim **34**, wherein the establishing step comprises establishing the coolant fluid path through passages within a plurality of stator blades of the compressor.

36. The method of claim **35**, wherein the first passing step comprises, after passing the coolant over the outside surface of the compressor housing, passing the coolant along the coolant path through the passages in the plurality of stator blades.

37. The method of claim **36**, wherein the second passing step comprises, after passing the coolant through the passages in the plurality of stator blades, passing the coolant to an external heat exchanger.

38. The method of claim **34**, wherein the releasing step comprises circulating the fluid through a heat exchanger.

39. The method of claim **34**, further comprising passing the coolant through a cooling jacket disposed about an outside surface of the compressor.

40. A process for use in a gas turbine engine, comprising:

using a compressor having a housing, compressing air at a substantially constant temperature;

in conjunction with the compressing step, circulating a cooling fluid over an outside surface of the compressor housing or through stator blades in the compressor, wherein the cooling fluid removes heat from the air to allow the compressing step to operate at the substantially constant temperature;

using a combustion chamber, heating the compressed air at a substantially constant pressure;

expanding the heated, compressed air through a turbine stage to drive the turbine stage and the compressor; and

exhausting the air to the atmosphere.

41. The thermodynamic cycle of claim **40**, further comprising:

passing the compressed air through a recuperator disposed within the exhaust flow of the engine to increase the temperature of air exiting the compressor.

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