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(54) **APPARATUS FOR ACOUSTIC DAMPING AND OPERATIONAL CONTROL OF DAMPING, COOLING, AND EMISSIONS IN A GAS TURBINE ENGINE**

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USPC **60/722**

(57) **ABSTRACT**

Acoustic damping resonators (**60**, **62**, **76**, **78**) formed in pockets (**32**) between reinforcing ribs (**30**) in a grid of ribs on the outer surface of a wall (**74**) surrounding a combustion gas flow path (**18**). Each resonator has a perforated cover plate (**64**A-C) spanning between sides formed by the ribs (**30**). Film cooling exit holes (**44**) are provided in the wall (**74**) under each resonator. Resonating chambers (**48**A-C) of different volumes may be provided on the wall to damp different unwanted frequencies. Different sets of resonators with different volumes may be separately controlled by respective airflow control manifolds (**66**) via throttle valves (**68**, T**1**-T**3**). Control logic (**70**) may control the valves based on frequency/airflow response functions (**80**, **82**) for each size of resonator to optimize damping and cooling and to lower emissions over varying engine operating conditions.

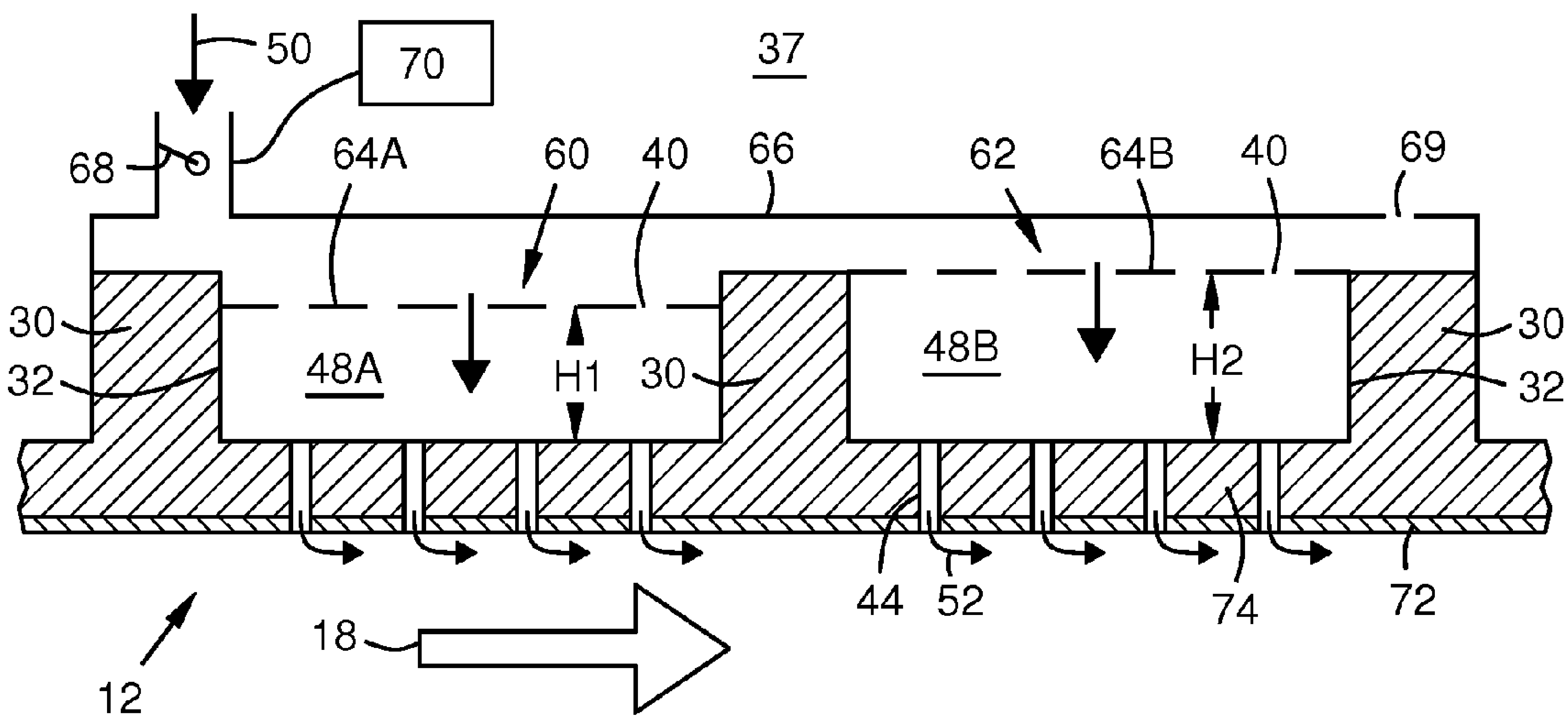


FIG 1
PRIOR ART

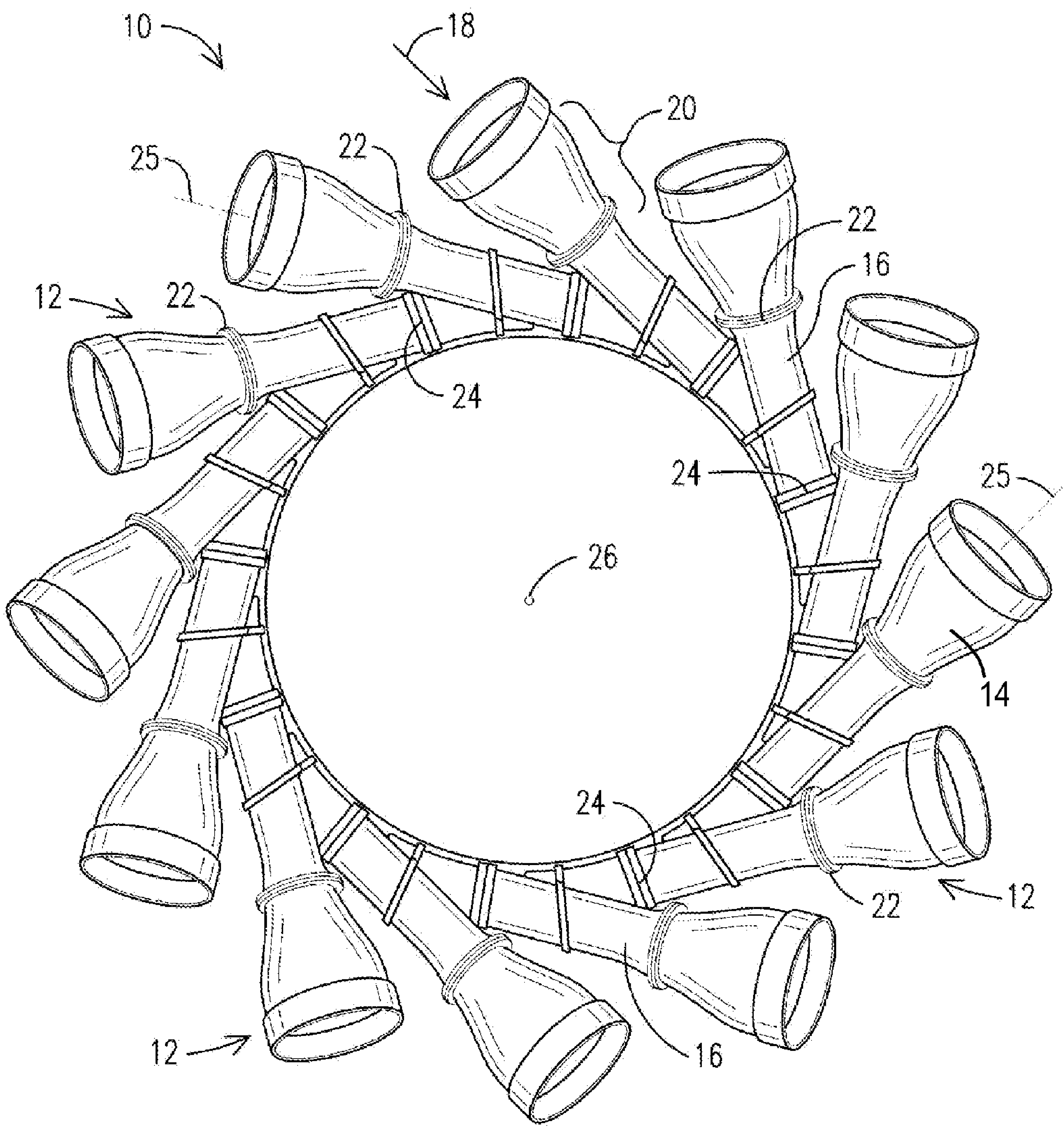


FIG 2
PRIOR ART

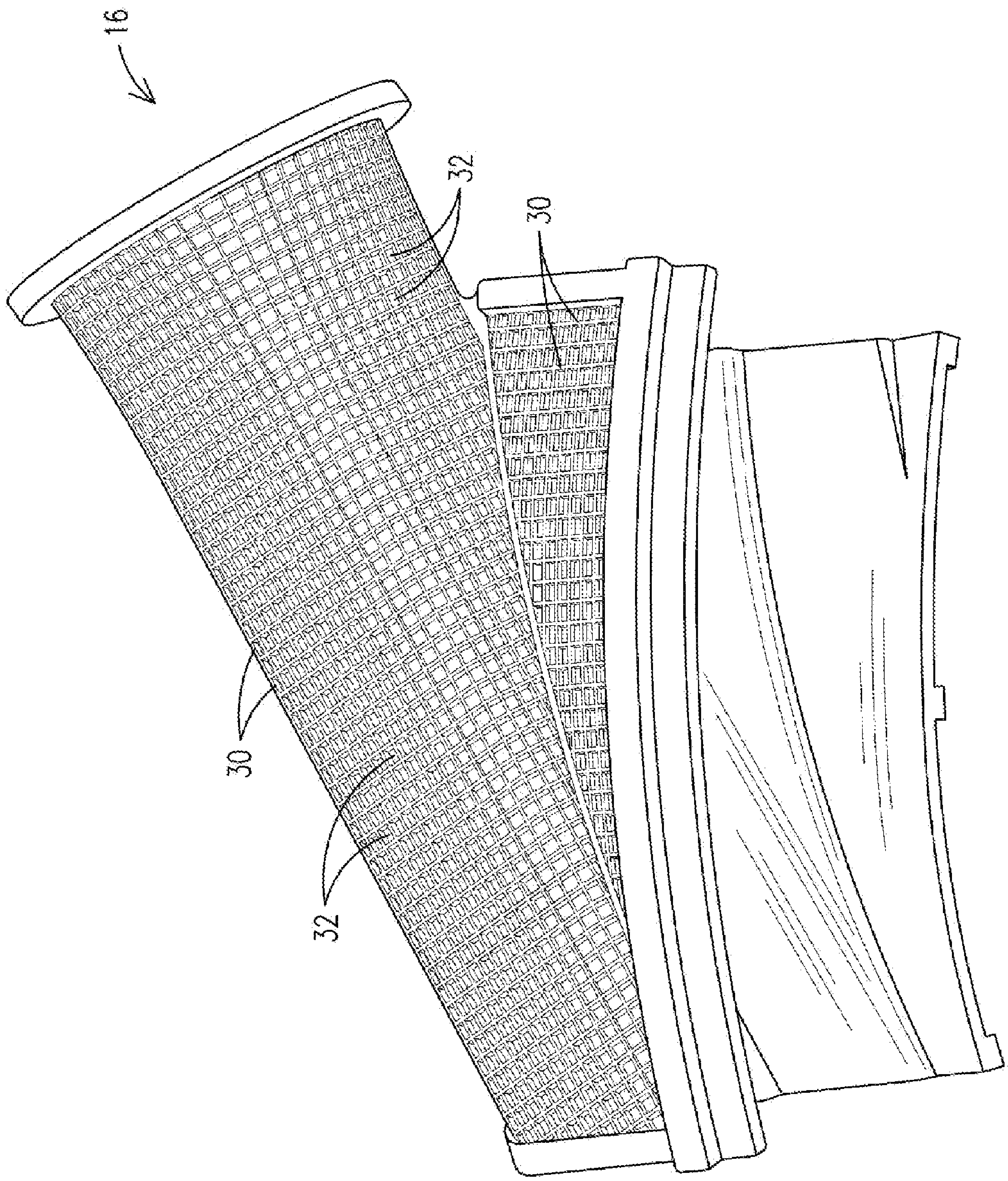
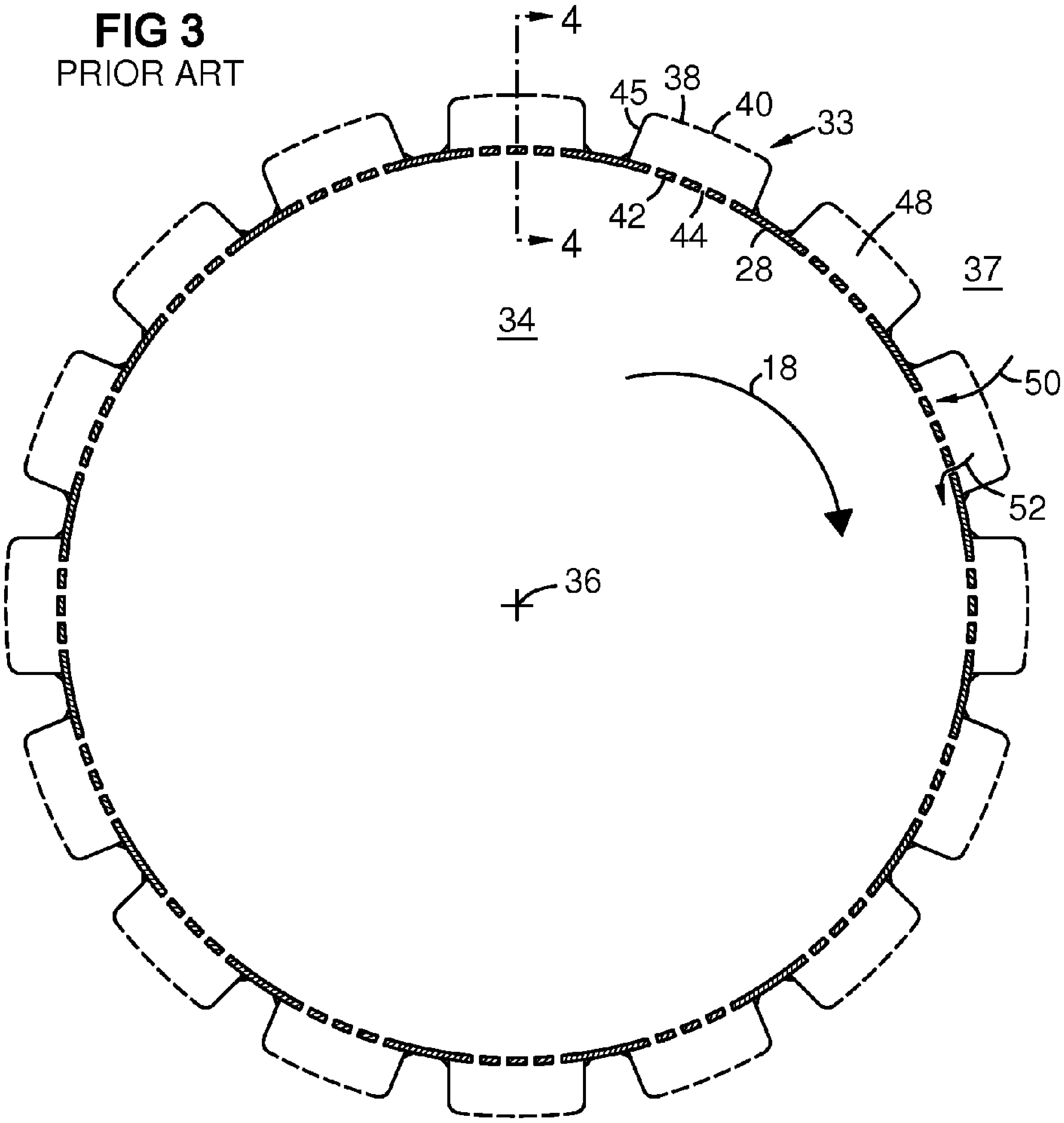


FIG 3
PRIOR ART



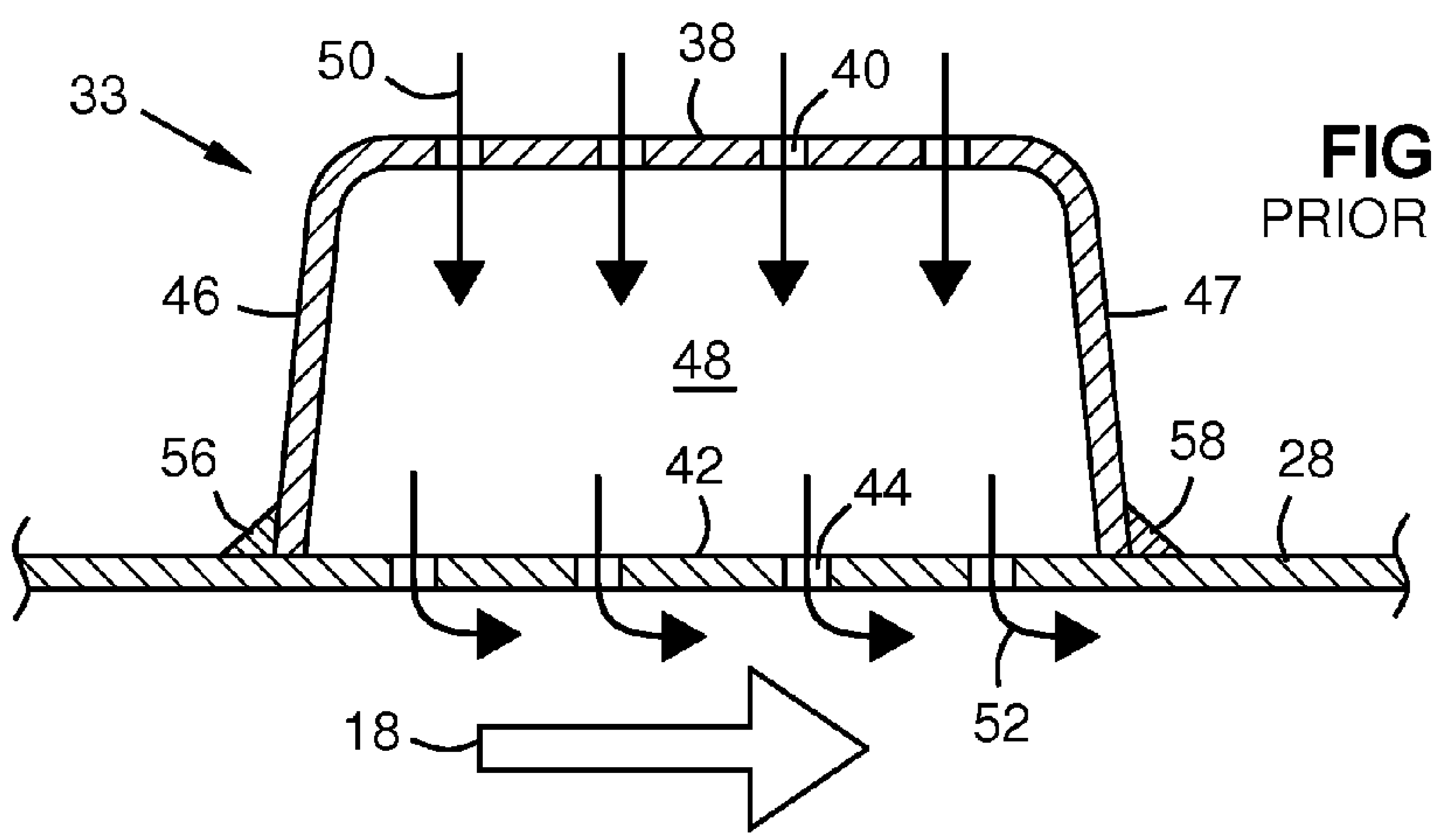
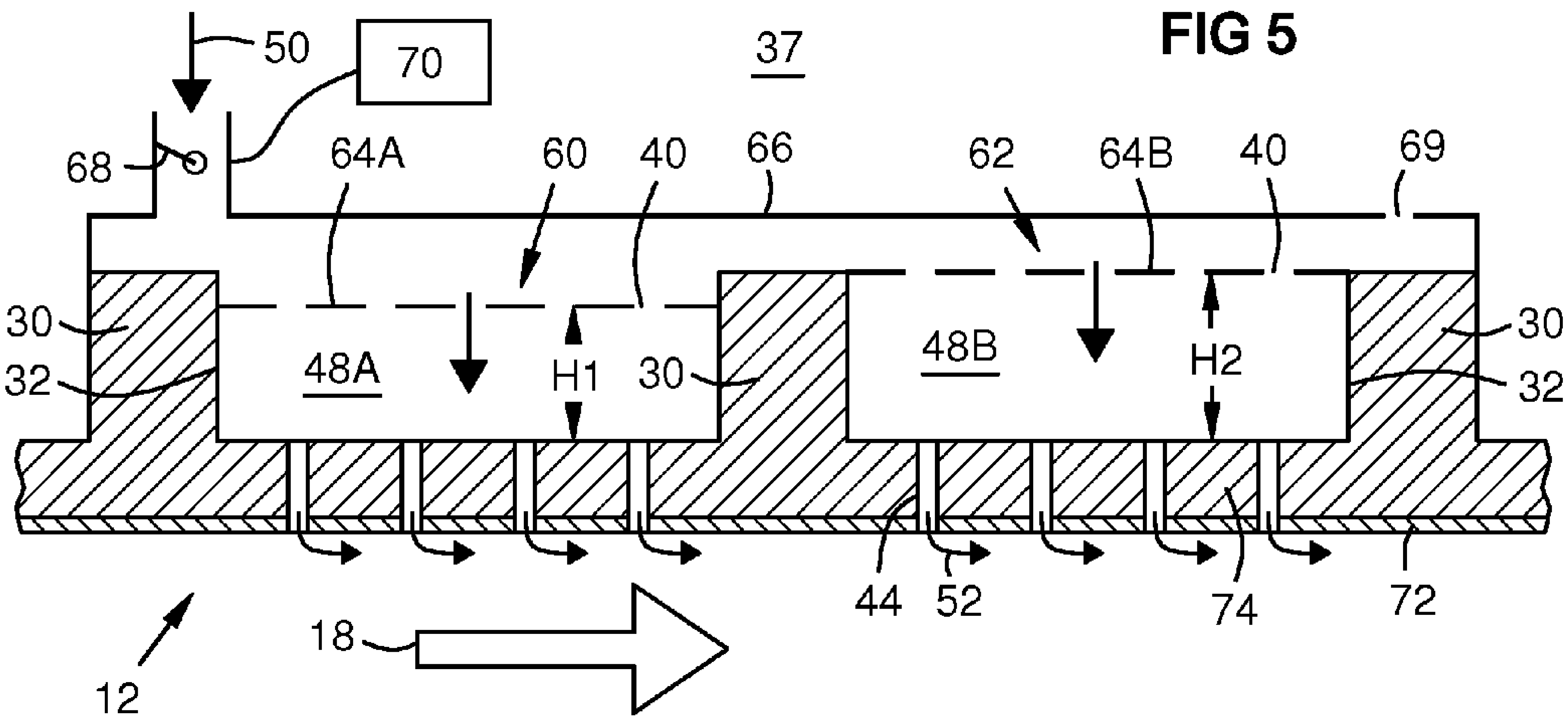


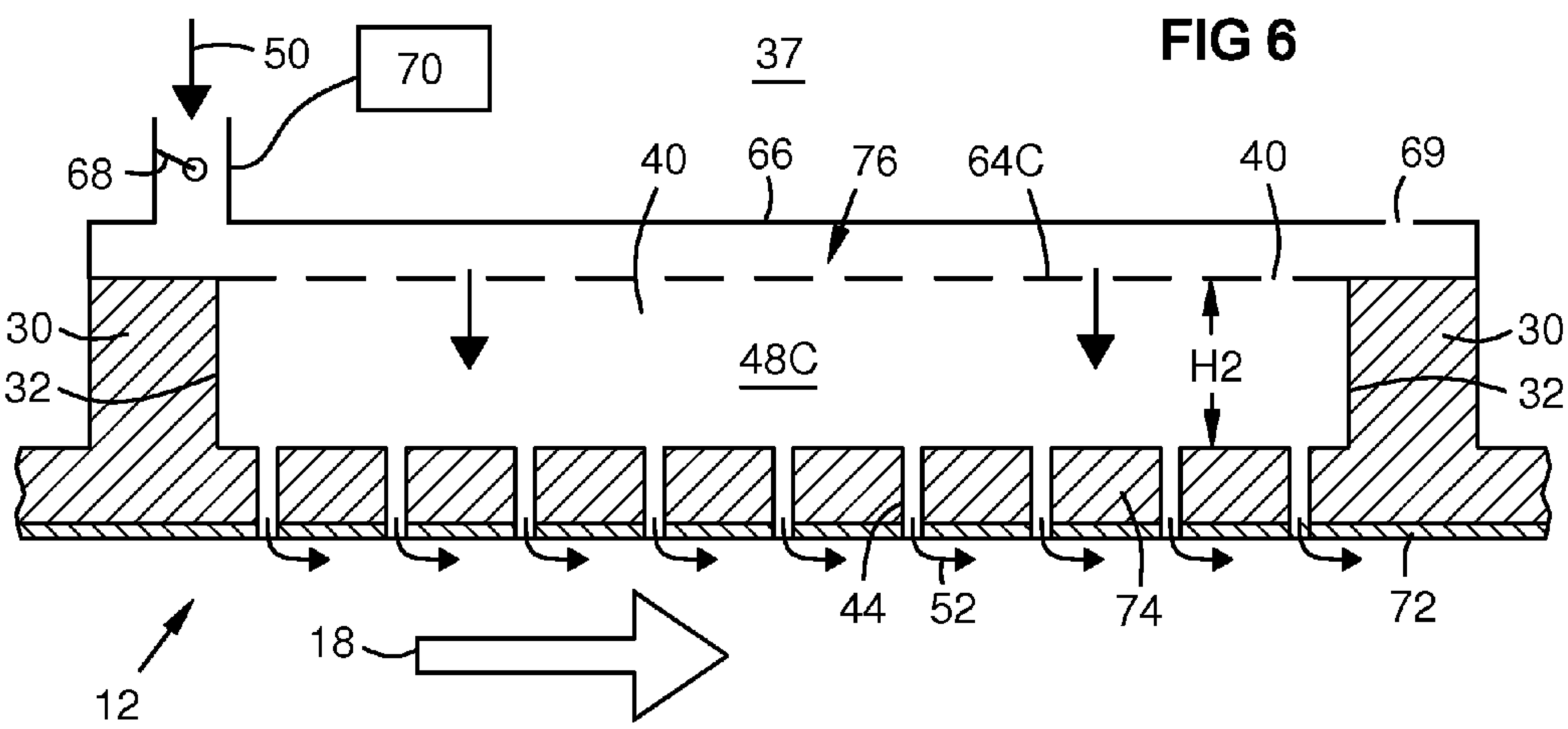
FIG 4
PRIOR ART

FIG 5



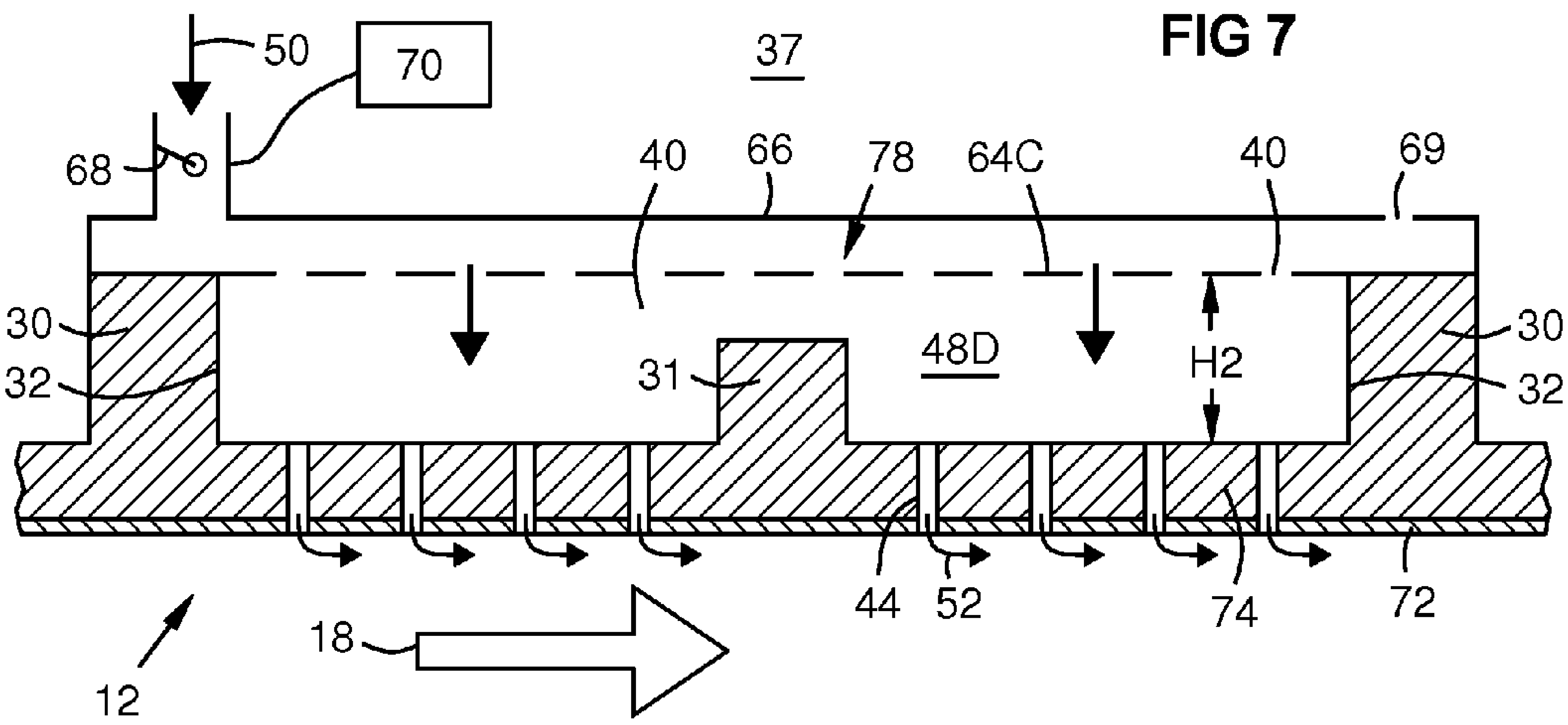
50
70
37
68
64A
60
40
66
62
64B
40
69
30
32
48A
H1
30
48B
H2
30
32
44
52
74
72
12
18

FIG 6



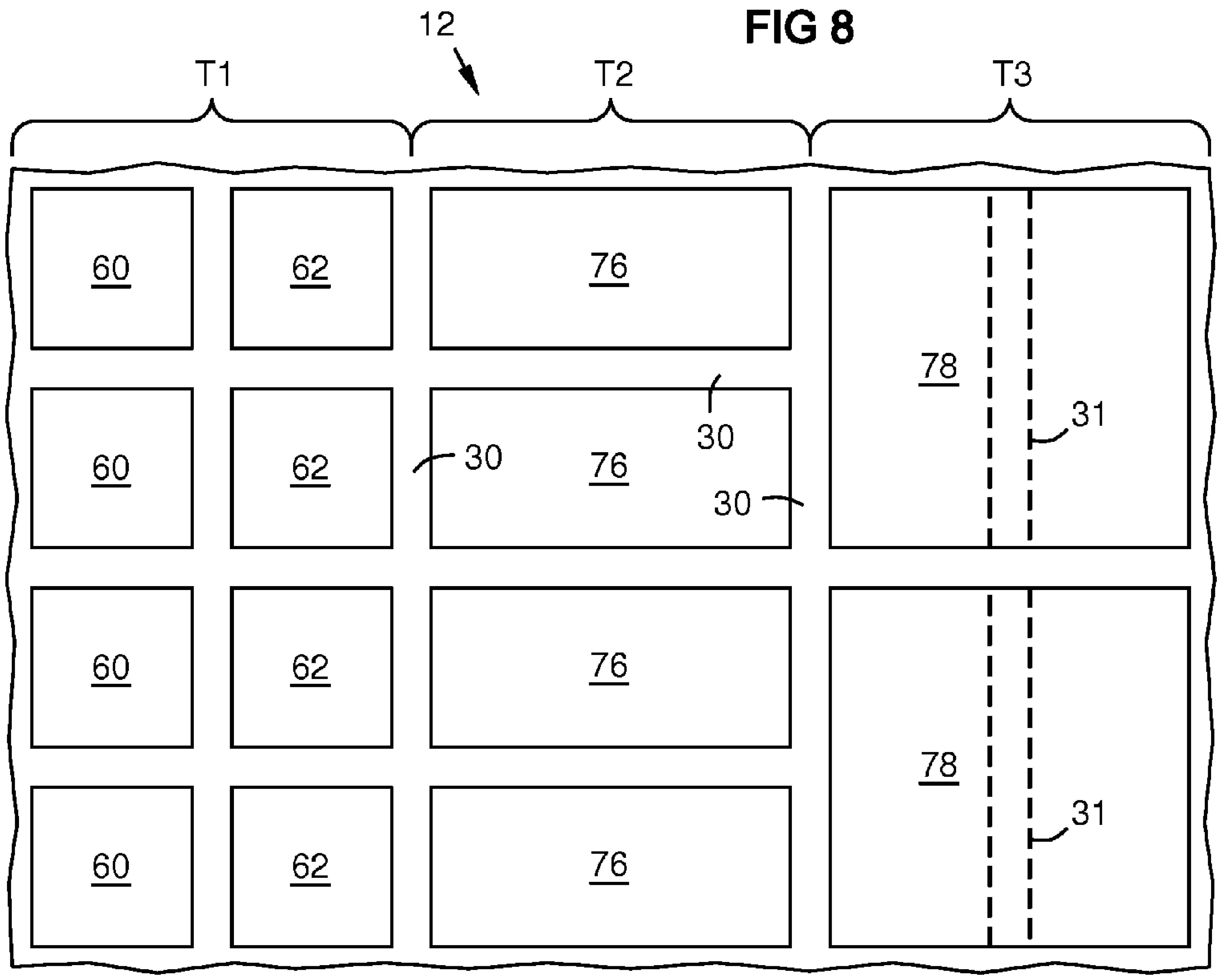
50
70
37
68
40
66
76
64C
40
69
30
32
48C
H2
30
32
44
52
74
72
12
18

FIG 7



50
70
37
68
40
66
78
64C
40
69
30
30
32
31
48D
H2
32
44
52
74
72
18
12

FIG 8



12
T1
T2
T3
60
62
76
60
62
76
30
30
30
78
31
60
62
76
60
62
76
78
31

FIG 9

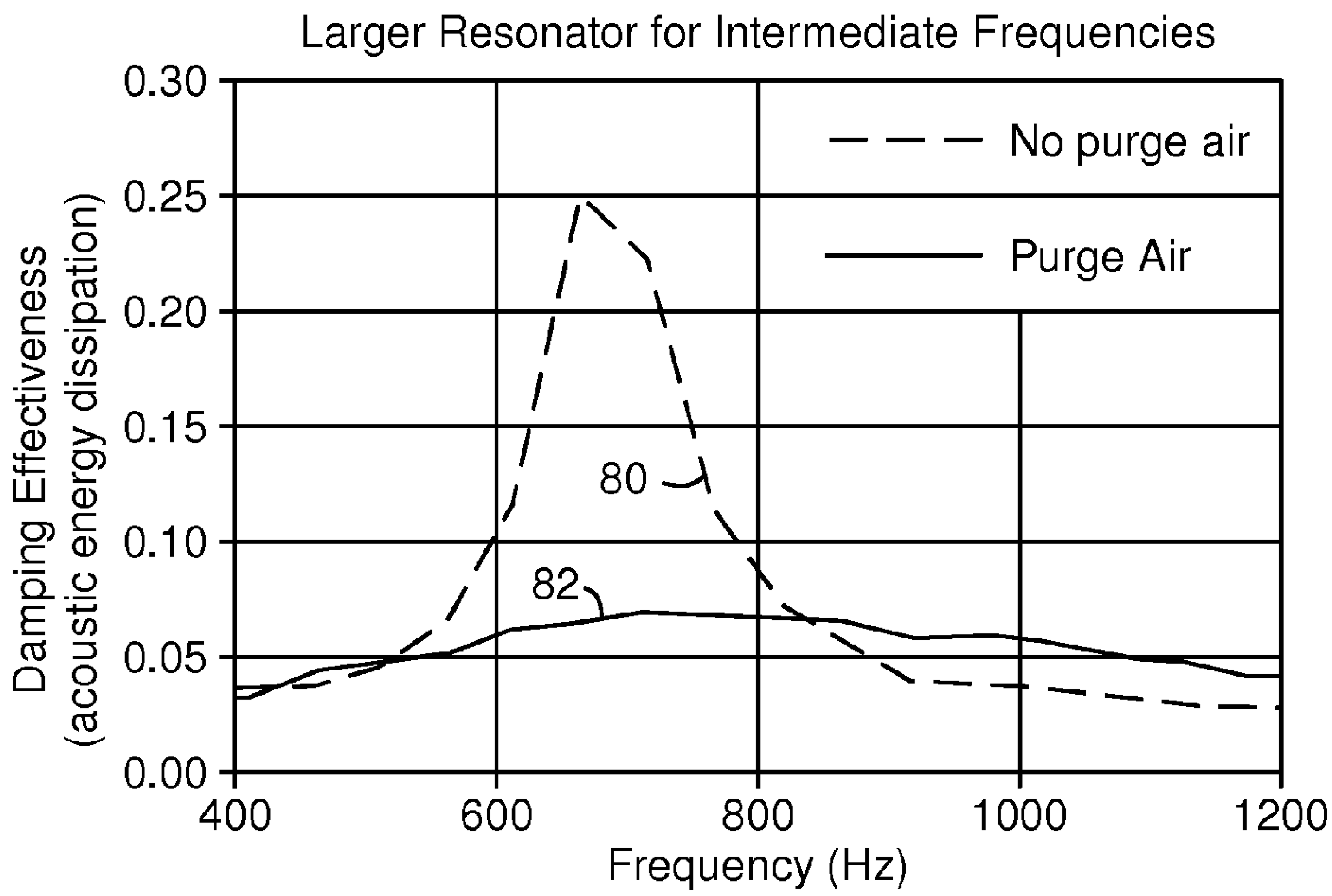
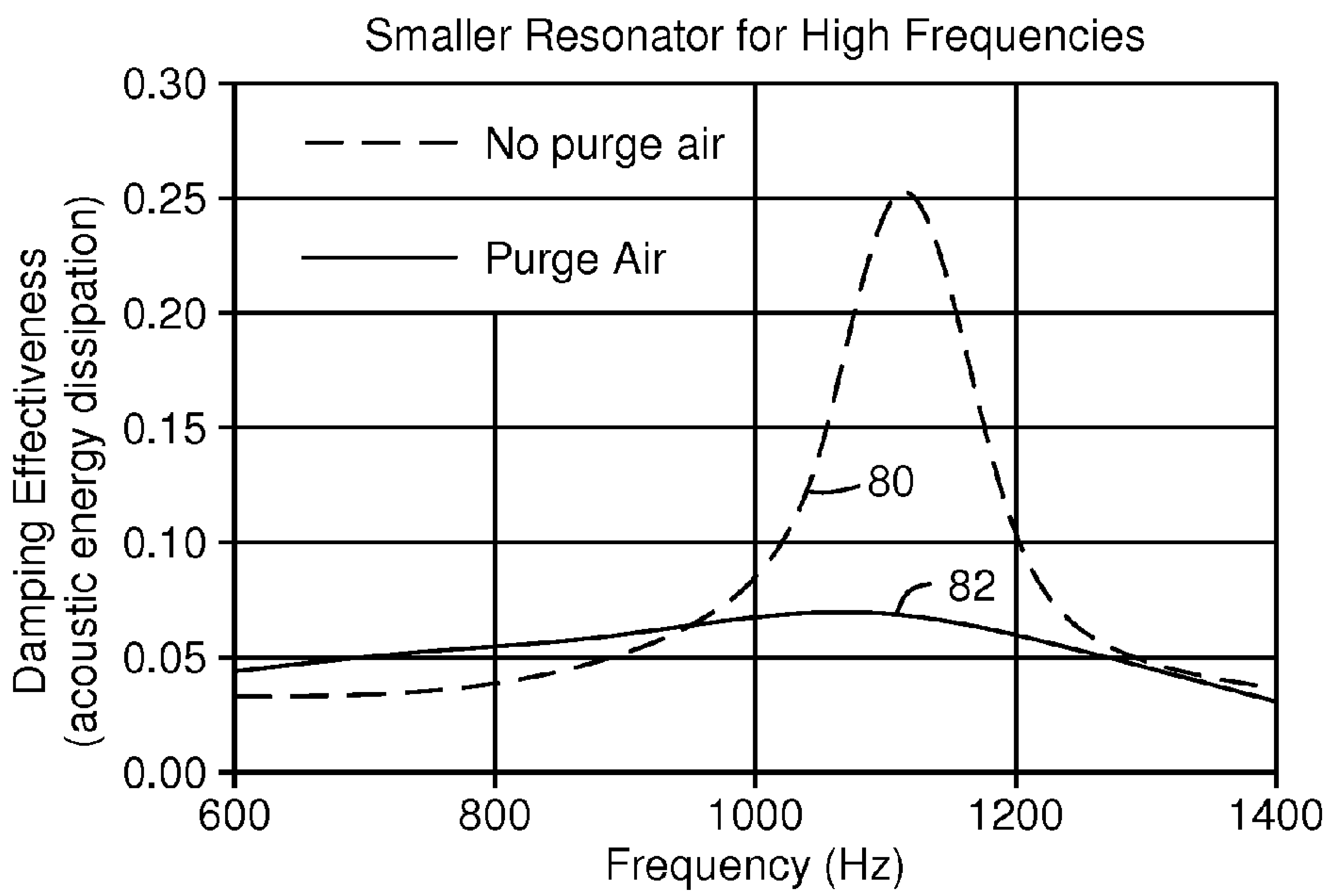

Larger Resonator for Intermediate Frequencies
Damping Effectiveness (acoustic energy dissipation)
0.30
0.25
0.20
0.15
0.10
0.05
0.00
400
600
800
1000
1200
Frequency (Hz)
No purge air
Purge Air
80
82

FIG 10


Smaller Resonator for High Frequencies
Damping Effectiveness (acoustic energy dissipation)
0.30
0.25
0.20
0.15
0.10
0.05
0.00
600
800
1000
1200
1400
Frequency (Hz)
No purge air
Purge Air
80
82

APPARATUS FOR ACOUSTIC DAMPING AND OPERATIONAL CONTROL OF DAMPING, COOLING, AND EMISSIONS IN A GAS TURBINE ENGINE

FIELD OF THE INVENTION

[0001] The invention relates to an apparatus for acoustic damping of a range of acoustic frequencies, and for operational control that optimizes acoustic damping, cooling, and emission control in a combustion section of a gas turbine engine.

BACKGROUND OF THE INVENTION

[0002] Gas turbine engines often use a portion of air from the compressor for cooling and emissions control. Combustion gas temperatures can approach or exceed limits for structures in the working gas flow path. Therefore, cooling of surfaces adjacent the combustion gas ("hot surface") may be implemented. Film/effusion cooling holes are provided through walls of flow-directing structures lining the working gas flow path so that a portion of the compressed air bypasses the combustor inlets and flows through these walls. This approach is used on such structures as the combustion chamber liner, transition ducts, transition exit pieces, and other components. The holes provide film cooling and effusion cooling of these components.

[0003] In conventional gas turbines a transition duct directs a flow of combustion gas traveling at about mach 0.1 to 0.3 between each combustor and the first row of turbine blades. Compressed air in a plenum surrounding this duct has higher static pressure than the combustion gas within the transition duct. This drives compressed air from the plenum through cooling holes in the duct walls. An emerging technology for can annular gas turbine engines provides transition duct structures that direct gas from the combustor to the first row of turbine blades along a mainly tangential and partly axial flow path at proper speed and orientation to drive the first row of rotating blades without an intervening row of stationary vanes. An assembly of such transition ducts is disclosed in U.S. Pat. No. 7,721,547 to Bancalari et al. issued May 25, 2010, incorporated in its entirety herein by reference.

[0004] In the emerging technology the transition ducts accelerate the combustion gas above mach 0.3 to about mach 0.8. This increased speed provides a decrease in static pressure in the newer transition duct design, so a greater pressure difference exists between the compressed air in the plenum and the combustion gas in the duct. This, pressure difference can provide more air than is needed for film cooling. It is so great that film cooling can overshoot and separate from the hot inner surface of the duct, reducing cooling effectiveness, unless the cooling holes are kept smaller than in prior designs and smaller than is ideal. Smaller holes clog with particles more quickly.

[0005] Acoustic damping resonators have been used in gas turbine engines to damp vibrations during operation. They may be called Helmholtz resonators or High Frequency Dynamics (HFD) damping resonators. Examples are disclosed in U.S. Pat. No. 6,530,221. Each resonator includes a chamber in an enclosure welded to a wall lining the working gas flow path. They are used on structures such as a combustion chamber liners and transition ducts. The resonator enclosure may have holes that admit cooling air to purge the resonator chamber. This prevents contamination from entering the chamber from the working gas, and cools the resonator walls and flow path wall. The cooling air passes through the resonator walls, impinges on the flow path wall, and then passes through effusion/film cooling holes in the flow path wall to form a cooling film on the hot inner surface of the flow path wall. These film cooling holes also act as Helmholtz resonation ports energized by the working gas flow.

[0006] The volume of a resonator is the main determinant of its resonant frequency. Space limitations can limit the size of damping resonators, thus limiting them to damping high frequencies only, such as over 1000 Hz. But unwanted intermediate frequencies between 50-1000 Hz are generated in gas turbine engines under some conditions. Damping resonators are often needed most at areas of high heat release. This exposes their enclosure attachment welds to high temperatures via heat conduction through the flow path wall, which can be a design-limiting factor.

BRIEF DESCRIPTION OF THE DRAWINGS

[0007] The invention is explained in the following description in view of the drawings that show:

[0008] FIG. **1** is a front view of a prior art assembly of combustion gas flow directing structures called transition ducts and exit pieces in an emerging technology thereof.

[0009] FIG. **2** is a back perspective detailed view of a prior art exit piece of one of the flow directing structures in the emerging technology of FIG. **1**.

[0010] FIG. **3** is a back sectional view of a prior art combustion chamber liner with a circular array of acoustic damping resonators on the outer surface thereof.

[0011] FIG. **4** is a side sectional view of a damping resonator taken on line **4-4** of FIG. **3**.

[0012] FIG. **5** is a side sectional view of two damping resonators with different chamber volumes showing aspects of an embodiment of the invention.

[0013] FIG. **6** is a side sectional view of a larger damping resonator embodiment.

[0014] FIG. **7** is a side sectional view of a larger damping resonator embodiment with an intermediate reinforcing rib not extending to the cover plate.

[0015] FIG. **8** is a schematic view of four sets of damping resonators with four respectively different chamber volumes controlled in three groups by three air inflow throttle valves.

[0016] FIG. **9** is a frequency/airflow response chart for a resonator sized to damp a frequency or band of frequencies at about 700 Hz.

[0017] FIG. **10** is a frequency/airflow response chart for a resonator sized to damp a frequency or band of frequencies at about 1100 Hz.

DETAILED DESCRIPTION OF THE INVENTION

[0018] FIG. **1** shows an assembly **10** with a plurality of flow directing structures **12**. Each flow directing structure **12** may include a cone **14** and an associated exit piece **16**. Each cone **14** receives combustion gas **18** from a respective combustor (not shown), and begins accelerating the combustion gas **18** to a speed appropriate for delivery onto the first row of turbine blades (not shown). Acceleration of the combustion gas **18** is accomplished by an acceleration geometry **20** using flow cross section constriction based on the Bernoulli principle and the Venturi effect. In the example shown, this is embodied as a cone-shaped duct **14** with a centerline **25** oriented mainly tangentially and partly axially relative to the turbine axis **26**.

The cone **14** may abut the exit piece **16** at a joint **22**. Adjacent exit pieces abut each other at joints **24**. The exit pieces form an annular chamber immediately upstream of the first row of turbine blades (not shown). Combustion gas **18** enters each cone **14** and travels along a flow path within the cone **14**. The acceleration geometry **20** accelerates the combustion gas **18** to more than mach 0.3, and especially to between 0.3-0.8 as appropriate for direct delivery onto the first row of turbine blades. Upon entering the exit piece **16** the combustion gas may continue to accelerate to the final speed and may morph from a circular cross section to a non circular cross section. The combustion gas then enters an annular chamber formed by the plurality of exit pieces and forms a helical flow about the turbine longitudinal axis **26** for a short time prior to reaching the first row of turbine blades. Other embodiments may vary the specific shape of the flow directing structure **12** and the acceleration geometry **20**, and these various configurations are considered within the scope of the disclosure.

[0019] FIG. **2** shows a grid of ribs **30** on an outer surface of an exit piece **16** of the flow directing structures **12** of FIG. **1**. The ribs structurally reinforce the exit piece to withstand the pressure difference created by the acceleration geometry **20** (FIG. **1**). Pockets **32** are defined between the ribs on the outer surface of the exit piece **16**. These ribs and pockets may be present on the combustor, the cone **14**, the exit piece **16**, or anywhere needed.

[0020] FIG. **3** is a sectional view of a prior art combustion chamber liner **28** surrounding a combustion chamber **34**, which may be cylindrical about an axis **36**. This view is taken on a section plane normal to the axis **36** through a circular array of acoustic damping resonators **33**, looking upstream relative to the combustion gas flow **18**. Each resonator **33** has a top wall **38** with coolant inlet holes **40**, a bottom wall **42** with holes **44** for coolant exit from the resonating chamber **48**, and side walls **45** between the top and bottom walls. The bottom wall **42** is formed by the combustor liner **28** bounding the combustion gas flow **18**, which flows generally axially, although it is shown here in a swirl for clarity. The coolant exit holes **42** may serve three functions: 1) as Helmholtz resonation ports that energize resonant vibrations in the chamber **48**; 2) as coolant exits; and 3) for effusion/film cooling of the liner **28**. The air plenum **37** receives compressed air from the turbine compressor (not shown). Some of this air **50** enters the coolant inlet holes **40** in the top wall **38** of each resonator, and then escapes **52** into the combustion chamber **34**, providing effusion/film cooling of the inner surface of the liner **28**.

[0021] FIG. **4** is a side sectional view of a resonator **33** taken along line **4-4** of FIG. **3**. Acoustic vibrations occur in each chamber **48** when the working gas **18** flows past the holes **44** in the liner **28**. These vibrations are excited by fluid dynamics mechanisms such as Helmholtz resonance. Such resonators have been used on the combustion chamber liner and the transition piece in prior art. However, the welds **56**, **58** can reach thermal limits during operation—especially the upstream welds **56**, which are not cooled by film cooling **52** exiting the resonator. This can be a design-limiting factor.

[0022] FIG. **5** is a side sectional view of two acoustic dampers **60**, **62** exemplifying an embodiment of the invention formed in respective pockets **32** between reinforcing ribs **30** on the outer surface of a flow directing structure **12** such as the exit piece of FIG. **2**. Two resonating chambers **48**A, **48**B are covered by two respective perforated resonator plates **64**A, **64**B enclosing the two respective pockets **32** by spanning between the ribs **30** surrounding each pocket. The two plates are attached at two different heights H1, H2, thus providing two different damping frequencies. An airflow control manifold **66** may cover and enclose the resonators. Compressed air **50** from the air plenum **37** surrounding the flow directing structure **12** may be metered into the manifold **66** by fixed inlet hole(s) **69** and/or by an air inlet throttle valve **68**. The throttle may be a type that never fully closes, as shown, and/or additional inlet holes **69** may be provided on the manifold **66** to assure a minimum flow. The throttle may be controlled by control logic **70** integrated into or connected to an engine control/sensor system that determines and sets optimum throttle settings based on engine speed, sensed vibrations, and/or other operating conditions including ambient conditions. This optimizes acoustic damping in conjunction with cooling of the flow directing structure **12** for different stages of engine operation as later described. An array of such resonators **60**, **62** may be provided around the flow directing structure **12**. This may be a circular array where the flow directing structure has a circular cross section. However, a variety of configurations of different sized resonators is possible. A thermal barrier coating **72** such as a ceramic layer may be provided on the hot side of the wall **74** lining the working gas path **18**.

[0023] FIG. **6** shows a resonator embodiment **76** with a larger chamber **48**C formed by a larger pocket **32** between ribs **30** covered by a resonator plate **64**C. The casting of the flow directing structure **12** may provide a plurality of pocket sizes for a range of damping frequencies. Wider rib spacing providing wider pockets **32** may be cast on wall portions of the flow directing structure **12** that need less reinforcement than other portions. Cylindrical or conic portions of the wall **74** may need less reinforcement that flat portions of the wall. Different resonator chambers may have different volumes within the same chamber height by using different spacings between the ribs **30** rather than different chamber depths. This is illustrated by chamber **48**B of FIG. **5** compared with **48**C of FIG. **6**, which have the same height H2 but different volumes. Such a height limitation prevents the larger resonators from extending outward into the air plenum **37** and impeding the flow therein.

[0024] FIG. **7** shows a resonator embodiment **78** with a relatively large chamber **48**D formed by a perforated resonator plate **64**C covering a large pocket **32** between ribs **30** of a given height H2, and further providing an intermediate rib **31** with less height across the pocket **32**, wherein the intermediate rib **31** does not reach the height of the resonator plate. This provides a damping frequency close to that of FIG. **6**, based on the total volume of the chamber **48**D, while providing more structural reinforcement compared to the resonator **76** of FIG. **6**.

[0025] FIG. **8** schematically illustrates a plurality of resonators of different sizes arranged on an outer surface of a wall of a flow directing structure **12**. A plurality of resonator sizes may be provided to damp a respective plurality of unwanted acoustic frequencies.

[0026] This example shows a row of each type of resonator previously illustrated herein—**60**, **62**, **76**, and **78**. The rows of resonators may encircle the flow directing structure or may be oriented along the working gas flow path or in any other direction. The shapes, sizes, number, arrangement, and orientation of resonators may be designed for each engine to optimize damping and rib reinforcement. The resonators do not need to be rectangular. They can be any shapes needed to fit around a flow directing structure that may be conic or irregular. Thus, the chamber shapes may include, but are not limited to, trapezoidal, triangular, hexagonal, and irregular. One or more airflow control manifolds may cover all or some of the resonators **60**, **62**, **76**, **78**. In this non-limiting example, independent throttling T**1**, T**2**, and T**3** is provided by three manifolds **66** as shown in FIGS. **5**, **6**, and **7** respectively (not shown here for clarity). Each manifold/throttle T**1**, T**2**, T**3** variably meters compressed air inflow to a subset of the resonators, where each subset damps a different frequency or a different range of frequencies from the other subsets. In another embodiment, the manifolds may provide fixed metering, without an active throttle valve but providing different fixed metering for each subset of resonators. The different chamber volumes of the different sets of resonators may be sized to provide damping over a wide range, for example with peak frequency responses distributed over a range of 300-4000 Hz.

[0027] FIG. **9** shows frequency/airflow damping response curves **80**, **82** for an exemplary resonator that is large enough to damp intermediate frequencies (50-1000 Hz). It is most effective at about 700 Hz with minimal purge air. FIG. **10** shows estimated corresponding damping curves **80**, **82** for a smaller resonator with peak effectiveness at about 1100 Hz with minimal purge air. The control logic **70** may control the throttle valves T**1**, T**2**, T**3** to vary each inflow based on such damping frequency/airflow response curves to optimize acoustic damping, cooling, and combustion temperature in the engine under varying operating conditions.

[0028] At low engine loads, more compressed air **50** is available than is needed for combustion. If this excess air enters the combustor, it cools the combustion zone **34**, increasing carbon monoxide (CO) emissions. Thus, at low engine loads, one or more of the resonator throttles **68**, T**1**, T**2**, T**3** may be opened, causing more compressed air **50** to bypass the combustor inlet, thus increasing combustion temperature to an optimum range, and reducing CO emissions. At higher loads including full rated power, maximum air is needed for combustion to avoid excessive temperatures in the combustion zone that increase nitrogen oxide emissions (NOx). Thus, under high loads, the resonator throttles may be closed, thus providing more compressed air to the combustion zone, which reduces its temperature to optimum range, and reducing emissions of NOx. This also maximizes damping effectiveness at high engine power when it is most needed, as exemplified by the sets of function curves in FIGS. **9** and **10**.

[0029] Specific unwanted frequencies that occur at partial engine loads can be damped by minimizing the flow to certain subset(s) of resonators while not minimizing the flow to other resonators to avoid over-cooling the combustion zone. For example, for prolonged operation at ¾ load, the control logic may set throttles T**1** and T**2** half closed, and throttle T**3** fully closed to maximize damping of a specific intermediate frequency by larger resonators **78**. However, at ½ load, the control logic may fully open all throttles to minimize CO emissions, especially if ½ load is a short-term transitional stage. The apparatus herein provides a wide range of such options that can be selected by an operator and/or by predetermined control logic **70** to optimize the combination of noise reduction, emission reduction, and cooling over a wide range of operating conditions.

[0030] Some resonators may be specialized to damp specific frequencies that occur only during specific operating conditions. Under other conditions, the peak resonance (the peak of trace **80** in FIGS. **9** and **10**) in these specialized resonators may be partially or largely quenched by increasing the inflow (trace **82** in FIGS. **9** and **10**) to reduce resonance that could otherwise be audible and unneeded in such other operating conditions.

[0031] One benefit of this resonator design is elimination of welds on the hot wall **74**. These welds are a limiting factor in the prior design of FIG. **4** (welds **56**, **58** on wall **28**).

[0032] The attachment points of the resonator cover plates **64**A, **64**B, and the manifold **66** are separated by a distance (exemplified by H1 and/or H2 herein) from the hot wall by the ribs **30**. The acceleration geometry **20** provides more pressure differential than is minimally needed to purge and cool the resonators **60**, **62**. Reducing the airflow **50** entering in the manifold **66** under some operating conditions improves engine efficiency, cooling, and damping, and reduces emissions. Reducing the pressure of the compressed air in the manifolds under all conditions allows the coolant/purge inlet and exit holes **40**, **44** to be larger, thus less susceptible to particulate clogging, and causes the film cooling **52** to flow more slowly and thus adhere better to the hot inner surface of the component wall **74** without overshoot.

[0033] Separate fabrication and attachment of side walls for the resonators is not needed when reinforcing ribs **30** are already provided on the casting of the flow directing structure to oppose the pressure differential previously described. In that case, these resonators take advantage of existing pockets between the ribs of the castings of the transition piece and exit piece.

[0034] While various embodiments of the present invention have been shown and described herein, it will be obvious that such embodiments are provided by way of example only. Numerous variations, changes and substitutions may be made without departing from the invention herein. Accordingly, it is intended that the invention be limited only by the spirit and scope of the appended claims.

The invention claimed is:

1. Apparatus for acoustic damping in a gas turbine engine, comprising:

a combustion gas flow directing structure comprising a wall that lines a flow path for a combustion gas, the wall being surrounded by compressed air at higher pressure than a pressure of the combustion gas;

a plurality of pockets formed by a grid of ribs on an outer surface of the wall;

a perforated resonator plate covering one of the pockets forming a first resonator comprising a first resonating chamber having a first volume in the pocket;

a plurality of air exit holes in the wall under the resonating chamber, the holes acting as Helmholtz resonator ports for the first resonating chamber and as film cooling holes for cooling an inner hot surface of the wall; and

a first airflow manifold covering the first resonator and metering a first inflow of the compressed air to the first resonator.

2. The apparatus of claim **1**, wherein the first inflow metering is controlled by a first throttle valve connected to a control logic that varies a position of the first throttle valve based on varying operating conditions of the engine.

3. The apparatus of claim **2**, wherein the control logic controls the first throttle valve to meter the first inflow in inverse proportion to engine load, wherein the first throttle valve reduces the first inflow as an engine load increases, and increases the first inflow as the engine load decreases.

4. The apparatus of claim **1**, wherein the flow directing structure comprises a flow accelerating geometry that accelerates the combustion gas in the flow path and reduces static pressure of the combustion gas in the flow path by constricting a sectional flow area of the combustion gas flow path.

5. The apparatus of claim **4**, wherein the flow directing structure comprises a transition duct comprising an upstream conic portion and a downstream exit piece.

6. The apparatus of claim **1**, wherein the ribs are reinforcing ribs that are cast into the flow directing structure, wherein each rib comprises a rib height above the outer surface of the wall, wherein the resonator plate is attached to the reinforcing ribs at a first height that separates the resonator plate from the outer surface of the wall, and wherein the resonator plate is not attached directly to the outer surface of the wall.

7. The apparatus of claim **6**, further comprising a second resonator plate covering a second one of the pockets at a second height above the outer surface of the wall, forming a second resonator with a second resonating chamber with a different volume than the first volume.

8. The apparatus of claim **7**, wherein the first airflow manifold covers both the first and second resonators, and the first inflow metering is controlled by a first throttle valve connected to a control logic that varies a position of the first throttle valve based on an operating condition of the engine;

wherein the control logic controls the first throttle valve to meter the first inflow in inverse proportion to engine load, wherein the first throttle valve reduces the first inflow as an engine load increases, and increases the first inflow as the engine load decreases.

9. The apparatus of claim **6**, further comprising:

a second resonator formed by a second resonator plate covering a second one of the pockets, wherein the second resonator comprises a second resonating chamber having a different volume than the first volume; and

a second airflow manifold covering the second resonator and metering a second inflow of the compressed air to the second resonator at a different flow rate than a metered flow rate of the first inflow during at least some operating conditions of the engine.

10. The apparatus of claim **9**, wherein the first and second inflows are controlled by respective first and second throttle valves connected to a control logic that varies a position of the each throttle valve based on an operating condition of the engine.

11. The apparatus of claim **9** wherein the first and second resonating chambers have the same height above the outer surface of the wall.

12. Apparatus for acoustic damping in a gas turbine engine, comprising:

a wall surrounding a flow path for a combustion gas;

a plenum for compressed air around an outer surface of the wall;

a first and second plurality of acoustic damping resonators on the outer surface of the wall, the resonators of the first plurality each comprising a resonating chamber with a first volume, and the resonators of the second plurality each comprising a resonating chamber with a second volume that is different from the first volume; film cooling exit holes in the wall under each chamber;

a first airflow control manifold that meters a first inflow of the compressed air to the first plurality of acoustic damping resonators; and

a second airflow control manifold that meters a second inflow of the compressed air to the second plurality of acoustic damping resonators;

wherein the first and second manifolds meter the first and second inflows by different amounts from each other during at least some operating conditions of the engine, and both manifolds reduce a pressure of the compressed air provided to the resonators compared to a pressure of the compressed air in the plenum.

13. The apparatus of claim **12**, wherein at least some of the first resonating chambers have the same height as at least some of the second resonating chambers.

14. The apparatus of claim **12**, wherein the first and second airflow control manifolds comprise respective first and second throttle valves that meter the respective first and second inflows, wherein the throttle valves are connected to a control logic that controls the throttle valves to vary each of the first and second inflows based on a set of damping frequency/airflow response curves of the resonators to optimize acoustic damping, cooling, and combustion temperature in the engine under varying operating conditions of the engine.

15. The apparatus of claim **12**, wherein the wall comprises a flow accelerating geometry that accelerates the combustion gas in the flow path to more than mach 0.3, and reduces the static pressure of the combustion gas in the flow path by constricting a sectional flow area of the combustion gas flow path; wherein the resonating chambers are formed in pockets between reinforcing ribs cast on the outer surface thereof.

16. The apparatus of claim **12**, further comprising a third plurality of acoustic damping resonators on the outer surface of the wall, each resonator of the third plurality comprising a resonating chamber of different volume than the chamber volumes of the first and second pluralities; wherein the first second and third pluralities of damping resonators in combination damp at least some frequencies over range of 300-4000 Hz.

17. The apparatus of claim **12**, wherein the first and second inflows are metered by respective valves controlled by control logic that controls acoustic damping of a range of acoustic frequencies both above and below 1000 Hz, and controls a combination of acoustic damping, cooling, and emission control in a combustion section of the gas turbine engine over a range of operating conditions.

18. The apparatus of claim **12**, wherein the respective inflows of the compressed air to the first and second pluralities of resonators are variably metered by respective first and second throttles to control CO and NOx emissions and to control damping of frequencies both above and below 1000 Hz over a range of engine operating conditions under control of a control logic based on a set of frequency/airflow response functions for each of the first and second pluralities of resonators.

* * * * *