HIGH EFFICIENCY RADIANT BURNER

Inventors: D. Redwood Stephens, Seattle, WA (US); John Porensky, Seattle, WA (US)

Correspondence Address:
GRAYBEAL, JACKSON, HALEY LLP
155 - 108TH AVENUE NE, SUITE 350
BELLEVUE, WA 98004-5973 (US)

Assignee: Cascade Designs, Inc., Seattle, WA (US)

Appl. No.: 12/069,020

Filed: Feb. 5, 2008

Publication Classification

Int. Cl. F23D 14/28 (2006.01)

U.S. Cl. 431/344

ABSTRACT

A naturally aspirated, fully aerated radiant burner and optional heat exchanger arrangement where the radiant burner has a generally enclosed cavity defined, at least in part, by fuel gas impermeable surroundings and a lower surface of fuel gas permeable burner element, wherein cavity preferably has two opening exposed to an oxidizer source. Sealingly coupled to openings are mix tubes, each having respective first ends and second ends, wherein first ends occupy openings and second ends extend into and are exposed to cavity. Fuel gas injectors, which during use are in fluid communication with fuel gas, are positioned to introduce fuel gas into mix tubes and entrain only slightly more air than needed for stoichiometric combustion. Pre-combustion gasses migrate to upper surface where stable stoichiometric combustion occurs, resulting in low CO and NOx emissions, increased wind resistance and elevated combustion gas temperatures. Connecting the heat exchanger directly to the burner further increases its wind resistance and prevents dilution of the combustion gases by wind or free convection.
HIGH EFFICIENCY RADIANT BURNER

FIELD OF THE INVENTION

[0001] The present invention relates to controlled, low emissions, combustion and more particularly to pressurized hydrocarbon gas burners and most particularly to a liquid pressurized gas (LPG) burner/heat exchanger system that includes a high efficiency heat exchanger working in conjunction with a fully aerated radiant burner.

DESCRIPTION OF THE PRIOR ART

[0002] Conventional gas external combustion apparatus traditionally use partially aerated fuel-air mixtures and require a production of relatively large quantities of secondary air for complete combustion to occur. This dilution of the post combustion gases reduces heat transfer efficiencies into a heat transfer surface, such as a fluid container in a cooking or water heating system, e.g., a pot, or a commercial or residential hot water tank. Additionally, the volume of introduced secondary air is dependent on the apparatus' natural convection and diffusion properties, which limit the driving pressure of the pre-combustion gases and excess air to pressures that can be attained only by the buoyancy effect of the hot rising gases—this mode of heat transfer is know as free convection. Thus, while partially aerated, free convection combustion apparatus are generally simple in construction and operation, they suffer from a loss of heat transfer potential through thermal dilution by secondary air and lack the ability, particularly in smaller scale implementations, to create significant heating through convection.

[0003] In addition to the foregoing, apparatus relying on free convection heat transfer are further limited by the need to allow adequate space between the apparatus burner and target surface (container surface and/or heat exchanger) so that sufficient secondary air is available for complete combustion and so that the flame does not impinge upon, and is quenched by, the cooler target surface. Therefore, while minimizing the distance between the burner flame and the target surface increases heat transfer efficiencies, both inadequate air and flame quenching lead to elevated CO production, which is particularly undesirable in small and substantially enclosed environments.

[0004] One approach used in the prior art to increase heat transfer efficiencies involves the use of a heat exchanger. A number of companies (for example, Cascade Designs Inc. and JetBoil, Inc.) offer portable gas combustion apparatus with heat exchangers that boost efficiency from conventional portable stove designs (35%-55%) to (45%-65%). However, because these apparatus are limited by free convection heat transfer coefficients and dilution of the combustion gases with secondary air, higher efficiency opportunities for apparatus of these designs are limited. Another approach used in the prior art to increase heat transfer efficiencies is to exploit the advantages of forced convection. Significantly higher heat transfer rates can be obtained if the combustion gases can be driven at elevated pressures (forced convection). U.S. Pat. Nos. 4,773,390 and 5,749,356 describe hot water heaters, for example, that combine forced convection and a heat exchanger to boost heat transfer efficiencies. In order to achieve this, however, the invention described in U.S. Pat. No. 4,773,390 requires a blower to introduce adequate air for combustion and the invention described in U.S. Pat. No. 5,749,356 demands that fuel and air be mixed via a gas-air feed circuit. Thus, both systems show increased complexity and/or infrastructure requirements over a naturally aspirated burner.

[0005] While manufacturers of combustion-based heat transfer apparatus continually strive for increased combustion and heat transfer efficiencies, they must also address environmental concerns relating to combustion by-products. One such class of combustion by-products, nitrous oxides (NOx), is of particular concern with respect to domestic gas water heaters. Initial combustion of gases in most partially aerated burners occurs at high temperatures, which are conducive to nitrous oxide formation. U.S. Pat. Nos. 5,645,413; 6,446,581 and 6,508,207 claim naturally aspirated burners with reduced NOx emissions. However, these designs require the introduction of excess secondary air (between 20-100% excess air) and are thus subjected to the heat transfer penalties imposed by its diluting effects, as described above. Finally, staged-air combustion chambers, as described in U.S. Pat. No. 5,645,413, present a trade-off between NOx and CO emissions—"the lower the NOx . . . the more difficult it becomes to burn out CO." (column 7).

[0006] In view of the foregoing, a need exists for an external combustion, naturally aspirated heating apparatus that provides both inherent and exploitable heating abilities while mitigating NOx and CO emissions.

SUMMARY OF THE INVENTION

[0007] The present invention is directed to a naturally aspirated, fully aerated burner, an optional heat exchanger arrangement optimized for fluid containers, and systems incorporating the combination thereof. Burner embodiments of the invention use premixed fuel-air in conjunction with relatively low burner surface temperatures and incandescent surface combustion to efficiently create heat with minimal CO and NOx combustion by-products. High heat transfer values to containers exposed to the burner are achieved through forced convection of hotter, undiluted, combustion gases to optimized heat exchangers associated with the containers, which increase overall efficiency of system embodiments (70% to 85% in the embodiment described in detail herein), without adding excessive heat exchanger surface area or materially impeding the convective gas flow. An additional benefit realized by system embodiments of the invention is markedly increased resistance to the deleterious effects of wind, particularly in exposed conditions.

[0008] Unlike free convection prior art burners that rely upon the introduction of secondary air to provide sufficient oxidizer for proper combustion, burner embodiments of the invention exploit forced convection principles. Thus, as heat output is increased, the driving pressure for forced convection is also increased, and heat transfer efficiency is generally constant over a wide range of heat load outputs. Because complete combustion is achieved at the burner element outer surface without the addition of secondary air, the optional heat exchanger can mate directly with the burner, eliminating the cooling effects of convecting air and making the burner essentially impervious to wind. An optional thermally activated fuel flow interrupt increases the safety of the burner by stopping fuel flow in the case of an overheat scenario. The result of this arrangement provides for a radiant burner that is has increased resistance to the deleterious effects of wind on the burner, that greatly increases the safety of operation of the radiant burner, and that significantly reduces the output of NOx and CO.
Burner embodiments of the invention comprise a generally enclosed cavity defined, at least in part, by a fuel gas impermeable surrounding and an inner surface of a fuel gas permeable burner element, wherein the cavity has at least one opening exposed to an oxidizer source, preferably oxygen present in the ambient environment. Sealingly coupled to the at least one opening is fuel-air mixing element or mixing means having a first end and a second end, wherein the first end is exposed to and/or is fluidly coupled with the at least one opening of the cavity and the second end extends into and is exposed to and/or is fluidly coupled with the cavity. In a preferred embodiment, the mixing element or mixing means is a mix tube, which maximizes both the air and fuel interaction and results in momentum transfer as well as thorough mixing of the air with the fuel gas. As those persons skilled in the art will appreciate, any structure capable of mixing a gaseous fuel with a gaseous oxidizer, preferably air from the ambient environment, can be used as, or in place of, a mix tube, and therefore such structures are considered equivalent thereto.

A fuel gas injector, which during use of the burner embodiments is in fluid communication with a source of fuel gas, is positioned to introduce fuel gas into each mix tube, preferably at or proximate to the first end, thereby encouraging momentum transfer from the fuel gas to the air when the air is also introduced at or proximate to this location. Variables such as fuel gas pressure, location of gas introduction, mixing element volume, orifice size and related parameters are assessed and established to ensure proper fuel-air ratios in support of proper combustion. For example, in a preferred embodiment, the fuel gas injectors, mix tubes, burner permeability and port area are designed such that the injectors entrain approximately 1-6% more oxidizer than necessary for stoichiometric combustion. In addition, the port area is sized in a way to optimize port loading and consequent combustion surface temperatures such that NOx and CO production is minimized. As a consequence, the resulting pre-combustion gas (fuel-air mixture) requires no additional oxidizer in order to achieve proper combustion, which would normally be introduced at the site of combustion, thereby decreasing the temperature of the resulting convective flow.

Because of the porosity of the burner element and the momentum transfer of the fuel gas to the fuel-air mixture, a pressure gradient exists between the cavity and an outer surface of the burner element. Consequently, pre-combustion gasses diffuse from an inner surface of the burner element, which is exposed to the cavity, to the burner element outer surface. Fully aerated, pre-combustion gasses at the outer surface of the burner element may then be ignited, such as by an igniter that is associated with the burner, whereupon combustion takes place.

A feature of selected burner embodiments of the invention is the incorporation of a thermal fuse disposed between the fuel gas source and the gas injector(s). This fuse may be constructed from any material that will be predictably responsive to heat such that when exposed to heat higher than a certain temperature for an established period of time, the material changes form, which operates to interrupt fuel flow to the gas injector(s). In one series of embodiments, the fuse comprises a cut-out metal, such as a cadmium-lead-tin alloy, which is formed into a washer that operatively keeps a check valve between the gas source and the burner in the open position. Thus, in the event of a light-back or thermally derived malfunction, the increased temperature will cause the washer to undergo a phase change, such as from solid to glass or liquid, and thereby permit the check valve to close and terminate fuel gas delivery from the fuel gas source.

In order to increase the efficiency of burner embodiments of the invention, containment vessels, such as pots, can be specially adapted to exploit the quantity and quality of heat output of such burners, as previously intimated. A primary mode of efficiency enhancement comprises the use of integrated or removable heat exchanging structure at or near the bottom of containment vessels. Such structure preferably comprises a plurality of fins, either as fin elements integral with the vessel or as fin bodies attachable to the vessel, arranged to maximize radiant and convective heat transfer of combustion gasses from the burner. Alternatively, efficiency enhancement comprises the use of heat exchanging structure at or near the outer surface of the burner element, which may or may not be removable. Efficiency can be further increased by maximizing the thermal absorptivity of the vessel surface to optimize radiant heat transfer. Each relevant containment vessel will have a bottom surface and a lower side surface that is linked to the bottom surface by a shoulder.

The burners described and illustrated below provide a user with exceptional efficiency and significantly decreased undesirable combustion byproducts. For example, CO emissions are about 8 times less than a comparably sized conventional portable stove. Similarly, nitrogen oxides are significantly reduced (approximately 90-93%) when compared to commercially available competing portable stoves.

**BRIEF DESCRIPTION OF THE DRAWINGS**

FIG. 1 is an elevation view of an assembled burner and heat exchanger equipped pot system;

FIG. 2 is a cross section elevation view of a burner;

FIG. 2A is a detailed cross section of a thermal fuse/strip that can be used in the embodiment shown in FIG. 2;

FIG. 3 is a cross section plan view of the burner of FIG. 2;

FIG. 4 is a cross section elevation view of a first heat exchanger equipped pot;

FIG. 5A is a perspective view of the first heat exchanger equipped pot wherein post pot manufacture fin elements are attached to the bottom of the pot and external covers and rings are removed for clarity;

FIG. 5B is a perspective view of the first heat exchanger equipped pot wherein fin bodies are integrated into the bottom of the pot during manufacture of the pot and external covers and rings are removed for clarity;

FIG. 6 is a cross section elevation view of second heat exchanger equipped pot wherein a peripheral heat exchanger ring is employed to increase the surface area available for heat transfer; and

FIG. 7 is a perspective view of a peripheral heat exchanger ring segment for use with the embodiment of FIG. 6.

**DESCRIPTION OF THE PREFERRED EMBODIMENTS**

The following discussion is presented to enable a person skilled in the art to make and use the invention. Various modifications to the preferred embodiments will be readily apparent to those skilled in the art, and the generic principles herein may be applied to other embodiments and applications without departing from the spirit and scope of the present invention as defined by the appended claims. Thus, the
present invention is not intended to be limited to the embodiments shown, but is to be accorded the widest scope consistent with the principles and features disclosed herein.

Unless otherwise noted herein, most parts of burner 10 and heat exchanger 90 are constructed from metal. Depending upon the part's application, the metal may be aluminum, steel, copper, brass or similar conventional metal. The selection of metal is primarily driven by thermal transfer considerations, although resistances to corrosion and high temperatures, as well as weight considerations are also valid criteria for material selection. In a preferred embodiment, burner element 60 comprises a porous metal foam material sold under the trademark METPORE by Porvair Advanced Materials, Inc. of Hendersonville, N.C. However, those persons skilled in the art will appreciate that other gas porous, heat resistant materials can be used, such as ceramics and metal-ceramic composites.

Turning then to FIGS. 2 and 3, a burner embodiment of the invention is shown in cross section elevation and plan views, respectively. Burner 10 comprises metallic base 12, which provides fuel delivery infrastructure 30 (discussed below) and which partially defines cavity 24. Cavity 24 is further defined by metal surround 14 and burner element 60. As will be described in more detail below, cavity 24 is generally sealed from the environment with two major exceptions. First, mix tubes 50a and 50b are sealingly attached to surround 14 and are exposed to the environment via proximal ends 52a and 52b (see FIG. 3). Second, burner element 60 is porous to gasses (see FIG. 2). As a result of this arrangement, gasses introduced at proximal ends 52a and 52b of mix tubes 50a and 50b may travel the length of the mix tubes and into cavity 24 through distal ends 54a and 54b. Because of the momentum transfer of the fuel gas to the fuel-air mixture, a gas pressure gradient exists between cavity 24 and the environment at outer surface 64 such that gasses present in cavity 24 will diffuse through burner element 60 towards outer surface 64.

Fuel gas, such as Liquid Pressurized Gas (LPG), is delivered to burner element 60 in the following manner. An LPG bottle (not shown) is rotationally coupled to fuel delivery infrastructure 30, as is best shown in FIGS. 2 and 2A. To permit such coupling, fuel delivery infrastructure 30 includes inlet housing 27 having threaded portion 32, preferably conforming to the B-188 standards, as described in EN 521—Specifications for Dedicated Liquefied Petroleum Gas Appliances—Portable Vapour Pressure Liquefied Petroleum Gas Appliances, to ensure wide compatibility with gas bottle suppliers. Once securely coupled and referring to FIG. 2A, probe 36 opens a valve in the LPG bottle and pressurized gas travels through probe 36 and into chamber 26. Chamber 26 is generally defined by inlet housing 27 and seat 28. Within chamber 26 are sealing plug 29 and compression spring 25. Compression spring 25 provides an outward bias to sealing plug 29, which is prevented from translational movement by seat 28 reacting against outlet housing 31 via thermal fuse body 38. LPG occupies both chamber 26 and area 26', which is in fluid communication with outlet conduit 40 via port 39 and prevented from escape to the environment by O-ring 34. Outlet conduit 40 then permits LPG to discharge into pressure regulator 42 (see FIG. 2).

A feature of the disclosed arrangement is directed towards a thermal LPG interrupt that functions to autonomously stop the flow of gas from the container to the burner. As briefly described above and as best shown in FIG. 2A, seat 28 functions to prevent sealing plug 29 from extending into contact with sealing surface 41. In turn, seat 28, which is in a compression mode through the bias imparted by spring 25 to sealing plug 29, reacts against outlet housing 31 via thermal fuse body 38. But for the presence of fuse body 38, seat 28 would be urged to translate away from compression spring 25, thereby permitting sealing plug 29 to come in sealing contact with sealing surface 41, and thereby occlude further gas passage into outlet conduit 40. Therefore, fuse body 38 is intentionally constructed to lose structural cohesion at or above a general temperature to prevent potentially dangerous conditions such as might be encountered during a “light back” or reverse ignition propagation event. While the ultimate determination of the appropriate temperature is a matter of design consideration, the disclosed embodiment contemplates thermal conditions of between about 145° C. to 200° C. as being candidate temperatures for a thermal trip.

While those persons skilled in the art will appreciate the broad selection of candidate materials, particularly satisfying results have been observed when eutectic alloys are chosen. A benefit of using eutectic alloys concerns both the precise nature of their phase conversion and the very sharp transition provided by them. This second characteristic is of importance to the operational life of the burner, because the thermal fuse is in an axial compression mode, mechanical creep can occur, particularly at higher temperatures, thereby potentially decreasing the performance of the system during normal conditions. Creep is further limited by keeping thermal fuse body 38 well contained. By doing this, as the material creeps, it is forced to “flow” through small gaps between seat 28 and outlet 31. This would require large shear stresses when fuse body 38 is solid but very low stresses once the fuse has melted. One alloy that has yielded favorable results comprises cadmium –18.2% wt.; lead –30.6% wt.; tin –51.2% wt. This alloy has a melting point of about 145° C. ±1.5° C.

Upon passing into outlet conduit 40, the compressed gas is directed towards regulator 42 and valve assembly 44 for pressure and flow regulation. Regulated gas is then directed to both gas jets 48a and 48b via distribution manifold 46, which in turn direct fuel gas into mix tubes 50a and 50b. Entrainment of an oxidizer, in this case oxygen bearing air, occurs at the injector and throughout the length of the mix tube by drawing air into the mix tube at openings 16a and 16b to create pre-combustion gasses. Those persons skilled in the art will appreciate that other forms of oxidizer introduction could take place via the same or different structure. However, the present embodiment represents an efficient and cost-effective approach to the production of a combustible gas. Because the described method and related structure rely upon momentum transfer (a venturi effect is established at opening 16a and 16b, which creates a localized area of low pressure, thereby drawing in ambient air to aid in combustion), mixing of the fuel gas with an oxidizer is accomplished efficiently and inexpensively. Moreover, because there are no moving parts, reliability and longevity are also increased.

To optimize the introduction of air as an oxidizer and minimize the effects of the environment (primarily wind for portable burner operations), surrounding 14 is coaxially surrounded by perforated housing 18. Consequently, a generally annular space is created between surrounding 14 and housing 18, from which air is drawn into openings 16a and 16b. In this manner, any wind impacting perforated housing 18 is diffused prior to entering 16a and 16b.
The fuel gas and air combination (pre-combustion gases) exit from ends 54a and 54b of mixing tubes 50a and 50b, and enters cavity 24, where upon it impinges heat transfer posts 56. Because posts 56 are thermally coupled to base 12, heat generated by burner 10 and transferred to base 12 by radiation, conduction and/or convection is partially removed by incoming cool pre-combustion gases contacting posts 56. Beneficially, this drawing of heat from base 12 not only decreases the handling temperature of base 12, but also increases the heat content of pre-combustion gas, which promotes more efficient combustion thereof.

As noted earlier, during operation of burner 10, a pressure gradient exists between upper surface 64 of burner element 60, which is exposed to ambient conditions, and lower surface 62 of burner element 60, which is exposed to slightly pressurized pre-combustion gases. After transport of pre-combustion gases from cavity 24 to upper surface 64, a piezoelectric igniter (not shown) may be operated to initiate combustion of pre-combustion gases, in a manner well known in the art. Upon ignition, combustion migrates to just below upper surface 64 of burner element 60, and is prevented from further propagation by the low bulk thermal conductivity and small pore size of burner element 60. At this point, burner 10 becomes a radiant burner with virtually no perceptible freely convective frame. The incandescent filaments at the burner surface help to sustain and catalyze combustion, even in the presence of wind.

Screen 20 is provided as a protective feature to prevent unintentional physical contact with burner element 60 and to serve as an interface with cookware employing a heat exchanger as described in detail below. Both screen 20 and perforated housing 18 are secured to burner 10 by way of screen retainer ring 22.

As mentioned earlier, the fuel gas injectors, mix tubes, burner permeability and port area are designed in this embodiment such that the injectors entrain approximately 16% more oxidizer than necessary for stoichiometric combustion. This limits the energy diluting effects of excess air while ensuring sufficient oxidant for complete combustion. In addition, the port area is sized in a way to optimize port loading and consequent combustion surface temperatures such that NOx and CO production is minimized. When run at maximum output (approximately 9,000 BTU/hr in this embodiment), NOx concentrations average 11 ppm corrected to 3% oxygen, well below the industry accepted value of 55 ppm corrected to 3% oxygen and below Southern California’s SCAGMD Rule 1121, which limits NOx emissions of residential gas-fired water heaters to 15 ppm at 3% oxygen as of Jan. 1, 2008. When run in conjunction with the optional heat exchanger at 5,750 BTU/hr, air free CO emissions average 78 ppm. Testing shows that CO emissions drop as the stove power increases beyond this output. European standard EN 521 (Specifications for Dedicated Liquefied Petroleum Gas Appliances-Portable Vapour Pressure Liquefied Petroleum Gas Appliances) and CSA 11.2-2000 (American National Standard/CSA Standard for Portable Type Gas Camp Stoves) allow air free CO values of 2000 ppm and 765 ppm, respectively.

While radiant burner 10 represents a significant advance in heating technology with respect to efficiency, emissions, wind resistance, safety and reliability, further advances have been achieved when this technology is used in conjunction with a heat exchanger purposefully adapted to extract the maximum amount of heat from burner 10. As best shown in FIGS. 1 and 4-7, heat exchanger 90 can be integrated into a fluid vessel, and more particularly vessel or pot 70. The purpose of heat exchanger 90 is to efficiently extract heat generated by burner 10 by taking advantage of its combustion mode. In this respect, the mass flow and temperature attributes of heat generated by burner 10 are considered in the design of heat exchanger 90.

As shown in the several drawings, the constitution of heat exchanger 90 can take many forms. The ultimate selection of one form over another may be driven by design considerations such as the volume of vessel 70, the nature of the liquid to be heated, the fluid dynamic properties of the post-combustion gases, and similar factors. Thus, the presently illustrated embodiments are intended to show several variations, but are by no means representative of an exhaustive inventory of available heat exchangers adapted to exploit the combustion mode of the burner. However, the presently illustrated embodiments all attempt to maximize the surface area exposed to the radiant heat and combustion gases from burner 10 without significantly increasing the pressure drop through the system, and consequently reducing air entrainment below stoichiometric levels. Thus, the illustrated embodiments employ a plurality of channels having relatively unobstructed exit paths where the channels maximize the distance the combustion gases must travel from burner element 60 to the ambient environment. Additionally, constructing heat exchanger 90 out of high absorptivity material, or using high absorptivity coatings (e.g., hard anodizing) increases radiative heat absorption to maximize heat transfer efficiency.

Turning first to FIG. 5A, a weld-on heat exchanger arrangement is shown. Here, a plurality of fin elements 80 are formed separately from pot 70, and subsequently attached to pot 70 such as by spot welding, brazing, laser welding or similar heating techniques to create a plurality of channels 86 through which combustion gases may travel. Fin elements 80 are preferably constructed from aluminum by stamping or similar high volume creation means. Fin elements 80 are preferably formed for placement on bottom surface 78 of pot 70 in a spiral or involute pattern to maximize exposure time of the combustion gases with the elements. The curved fin shape also aids in minimizing thermal boundary layers of the flowing combustion gases, further increasing heat transfer. FIG. 5B shows a similar pattern of fin bodies 82 formed on bottom surface 78 of pot 70; however, fin bodies 82 are integral with bottom surface 78. In this embodiment, fin bodies 82 may be formed by machining the desired pattern in bottom surface 78 or during casting of bottom surface. While the thermal transfer rates from fin bodies 82 to pot 70 and overall durability are greater than the thermal transfer rates from fin elements 80 to pot 70 due to the more robust association of the former with the pot, manufacturing costs are higher.

In addition to machining or casting methods for creating suitable fin bodies, a preferred means of manufacturing integral fin bodies is by impact extrusion processes. These processes provide the benefits of exceptional thermal conductivity (superior to that of casting), desirous surface finish for the cooking surface (superior to that of casting or machining), low weight (superior to that of casting and machining, which also generates avoidable waste) and low cost (superior to that of machining and welding). While there are size limitations using these processes, they are not material to the form factors commonly used in portable cookware.
The embodiment of FIG. 6 illustrates a perimeter heat exchanger arrangement that can be used in conjunction with the heat exchangers of FIGS. 5A and 5B, or with other arrangements. By linking a plurality of perimeter elements 84 as shown in FIG. 7, for example, and surrounding the perimeter of pot 70 with such elements, waste heat exiting from channels 86, for example, impinges upon perimeter elements 84 and is redirected along reduced diameter portion 74 of pot 70. In this manner, additional surface area for heat exchange is created at both perimeter elements 84, which are thermally linked to heat exchanger 90, as well as directly to pot 70. To prevent the unintentional migration of fluid in pot 70 from entering heat exchanger 90, drip ring 76 is provided above reduced diameter portion 74.

Heat transfer and wind resistance can be further improved by mating the heat exchanger 90 to retainer ring 22, eliminating the cooling effects of blowing air and making the burner essentially impervious to wind. As mentioned above, this is possible because complete combustion is able to take place without the addition of secondary air. In such embodiments, bottom surface 78 is not planar or flat. Again depending upon design parameters, bottom surface 78 can be conical or frusto-conical like, with the apex at the center of the vessel. Such a geometry will not only beneficially modify the residency of any combustion gasses during operation of a burner, but when used in conjunction with a burner such as burner 10 having screen 20, will restrict the selection of containment vessels to those that properly mate with the burner. Alternatively, a plurality of surface features, such as convex or concave features, can be established in or on bottom surface 78 to alter the egress of post-combustion gasses to the environment.

What is claimed:

1. A fuel gas burner, for use with a source of fuel gas, comprising:
   a cavity defined, at least in part, by a fuel gas impermeable surrounding and a fuel gas permeable burner element having an interior surface exposed to the cavity and an exterior surface exposed to the environment, wherein the fuel gas impermeable surrounding has at least one opening;
   a mixing element having a first end and a second end, wherein the first end is fluidly coupled to the at least one opening and the second end is fluidly coupled to the cavity; and
   a fuel gas injector, operatively coupled to the source of fuel gas, at or proximate to the first end of the mixing element for injecting fuel gas there into, whereby introduction of pressurized fuel gas by the injector into the mixing element and entrainment of an oxidizer from the first end of the mixing element creates a volume of pressurized pre-combustion gas within the cavity, which diffuses from the interior surface of the burner element to the exterior surface of the burner element.

2. The burner of claim 1 wherein the source of fuel gas is a portable bottle of fuel gas and the burner is characterized as portable by a single person.

3. The burner of claim 1 further comprising at least one heat transfer post thermally linked to at least part of the fuel gas impermeable surrounding.

4. The burner of claim 3 wherein the at least one heat transfer post is disposed in the path of the pre-combustion gas exiting from the mixing element.

5. The burner of claim 1 further comprising a non-planar shielding element proximate to the burner element

6. The burner of claim 1 further comprising a thermal fuse for occluding a fuel gas passage between the source of fuel gas and the fuel gas injector.

7. The burner of claim 6 wherein the thermal fuse comprises a eutectic alloy.

8. The burner of claim 6 wherein the thermal fuse is radially constrained.

9. The burner of claim 6 further comprising a check valve having a seat and a plug wherein the thermal fuse is located between the seat and a reacting structure.

10. A system according to any of the previous claims and further comprising a heat exchanger adapted to mate with a housing portion of the burner.

11. A heating system comprising:
   a cavity, defined at least in part, by a fuel gas impermeable surrounding and a fuel gas permeable burner element having an interior surface exposed to the cavity and an exterior surface exposed to the environment, wherein the fuel gas impermeable surrounding has at least one opening;
   a mixing element having a first end and a second end, wherein the first end is fluidly coupled to the at least one opening and the second end is fluidly coupled to the cavity;
   a fuel gas injector, operatively coupled to the source of fuel gas, at or proximate to the first end of the mixing element for injecting fuel gas there into, whereby introduction of pressurized fuel gas by the injector into the mixing element and entrainment of an oxidizer from the first end of the mixing element creates a volume of pressurized pre-combustion gas within the cavity, which diffuses from the interior surface of the burner element to the exterior surface of the burner element; and
   a fluid container comprising an integral heat-exchanger adapted to engage with a housing portion of the burner.

* * * * *