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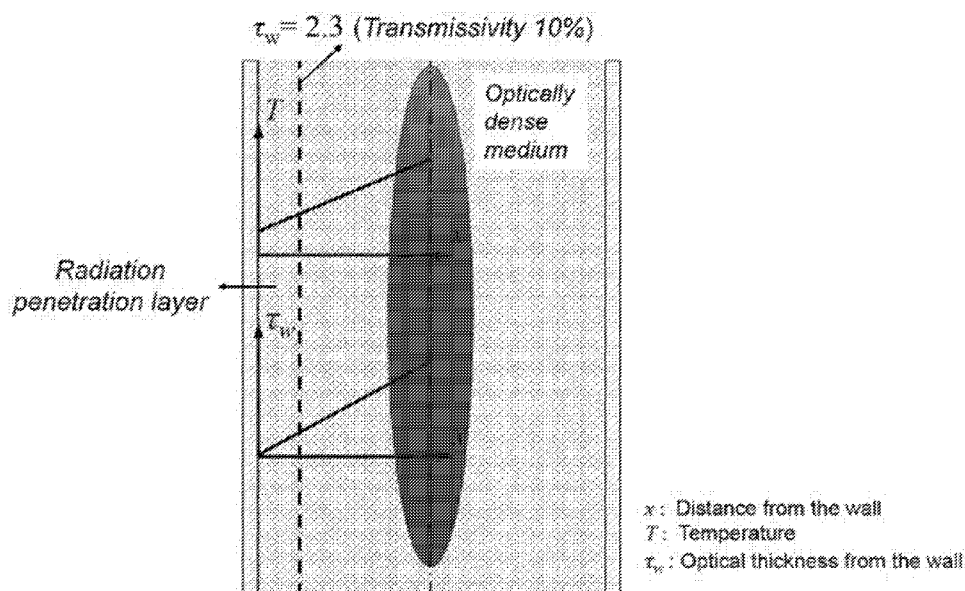


FIG. 1

(57) Abstract: The present disclosure is generally directed to a burner and boiler/furnace for pressurized oxy-combustion boilers and furnaces. The disclosure includes a design of a burner and boiler for a staged, pressurized oxy-combustion (SPOC) technology process and designs that affect wall heat flux. The disclosure further includes the introduction of wall rings to increase, for example, advection.

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BURNER AND BOILER/FURNACE FOR PRESSURIZED OXY-COMBUSTION BOILERS AND FURNACES

CROSS-REFERENCE TO RELATED APPLICATION

[0001] This application claims the benefit of U.S. Provisional Patent Application Serial No. 62/464,159, filed February 27, 2017, the entire contents of which are incorporated herein by reference.

Government Support

[0002] This invention was made with government support under Grant #DE-FE0009702, awarded by the Department of Energy. The Government has certain rights in the invention.

BACKGROUND OF THE DISCLOSURE

[0003] Fossil fuels produce highly reliable and low-cost energy. Nevertheless, increasing concerns over global warming and the role of anthropogenic carbon dioxide emissions from fossil fuel combustion have increased the demand for low-carbon technologies. Carbon capture, utilization, and sequestration (CCUS) is an important strategy to minimize these emissions by capturing the carbon dioxide produced. CCUS is a relatively low cost option, and it could also have a wide impact on supplying reliable low-cost electricity worldwide. Oxy-fuel combustion is one of the three major strategies for capturing carbon dioxide (CO₂) from stationary combustion furnaces and power plants. The most common oxy-fuel concept is to combust a pulverized fuel (coal) in a stream of oxygen that has been diluted with a large amount of recycled flue gas (~ 70% of the flue gas is recycled). The flue gas, after moisture removal, consists of mainly CO₂, which, after purification to the required standard, can be either utilized or sequestered. The recycled flue gas acts as an inert to reduce the temperature in the boiler and thereby control the heat flux to within the constraints of the boiler tube materials. A challenge of existing “first-generation” oxy-combustion technologies is that the plant efficiencies are rather low and the cost of electricity is rather high, relative to power generation without capture. The two system parameters that have the greatest potential for increasing

efficiencies and reducing costs are the gas pressure in the boiler and the amount of flue gas recycled.

[0004] The inherent requirement of high pressure carbon dioxide for either enhanced oil recovery (EOR) or sequestration makes it possible to pressurize the oxy-combustion process with no intrinsic added cost, since for coal combustion compressing oxygen before combustion requires comparable energy to compressing carbon dioxide after combustion. The dew point of the flue gas moisture increases with pressure, so that at higher pressure, condensation occurs at a higher temperature than at atmospheric pressure. Thus, pressurized oxy-combustion is a well-posed method to increase the efficiency of carbon capture. A significant portion of the latent heat of condensation, extracted at higher temperature, can be utilized effectively in the steam cycle, instead of being wasted, as it is in atmospheric pressure oxy-combustion. A number of approaches to pressurized oxy-combustion have been proposed, and their efficiencies have been independently compared. In some embodiments, staged, pressurized oxy-combustion (SPOC) technology reduces the efficiency penalty for carbon capture from about 10 percentage points for a 1st generation technology to less than about 3 percentage points.

[0005] The reduction of flue gas recycle has traditionally been challenging because flue gas recycle has been used in oxy-combustion boilers to create an environment similar to air-fired combustion, i.e., to control the temperature and residence times in the oxy-combustion boiler so that they are similar to those in air-fired combustion. Though the primary need is to control wall heat flux and ensure that it is always within the material constraints of the boiler tubes, most first-generation approaches do this by controlling temperature, and flue gas recycle is the most common approach to controlling temperature, with recycling being typically 70% of the flue gas. Still, there has been interest in trying to reduce the amount of flue gas recycle to increase efficiency and reduce cost. Furthermore, reducing flue gas recycle increases the proportion of heat transfer on the furnace wall that is due to radiative heat transfer over convective heat transfer. Since radiative heat flux from a flame is much higher than convective heat flux in power plants, reducing flue gas recycle could reduce the amount of boiler tube materials required, thereby reducing the capital cost. A few combustor

designs have been proposed for reduced recycle and have provided a summary of various efforts towards reduced flue gas recycle designs and demonstrations in. These noted that even though many of the demonstrated designs have shown performance improvements, they were demonstrated for either industrial furnaces or for boilers with low thermal input and low temperature and pressure steam, where much higher heat fluxes can be handled than in a typical utility-scale boiler.

[0006] None of these designs, however, have been demonstrated for utility-scale boilers, due to their inability to control wall heat flux within manageable levels. The SPOC process uses very unique approaches of fuel staging combined with radiative trapping to reduce flue gas recycle without increasing the radiative heat flux to the wall. Radiative trapping is a method to control radiative wall heat flux even with very high flame temperatures. In the SPOC process, separate boilers or stages are connected in series, with the fuel and oxidizer being distributed through the stages. The oxidizer is fed to the first stage at a much higher stoichiometry than required for the fuel in that stage. After combustion of the fuel fed to the first stage and subsequent heat transfer to the boiler tubes, the partially cooled flue gas and the excess oxygen are transferred to the next stage, where more fuel and some oxidizer is fed, and this process is repeated in subsequent stages until all the fuel and nearly all the oxygen is consumed. Conceptually, this method of fuel-staging could be used to reduce the amount of flue gas recycle to a very small amount. In some embodiments, the use of fuel staging provides the conditions necessary to successfully incorporate radiative trapping to control heat flux. Using computational fluid dynamics (CFD) simulations, shows a design of a SPOC boiler system where, even at very low flue gas recycle, the wall heat fluxes for all stages were controlled to manageable levels for utility scale applications.

[0007] Due to combustion in a high oxygen concentration environment, the first stage in the SPOC process is the most challenging in terms of controlling wall heat flux. In some embodiments of the present disclosure, a potential practical design of the burner and boiler for a SPOC process is disclosed, which provides additional flexibility in terms of operation, and the opportunity to control the oxygen concentration near the boiler tubes. In some embodiments, computational fluid dynamics (CFD) simulations are used,

and the effect of various design parameters on the wall heat flux show how these are utilized as a toolbox for the design of a low recycle, pressurized oxy-combustion burner/boiler where wall heat flux is controlled via radiative trapping rather than large amounts of flue gas recycle.

SUMMARY OF THE DISCLOSURE

[0008] In one aspect, the present disclosure is directed to a staged, pressurized oxy-combustion system, the system comprising a unique design including at least one burner and at least one boiler.

[0009] In another aspect, the present disclosure is directed to a method of capturing carbon dioxide, the method comprising using a staged, pressurized oxy-combustion system including a unique design, where the design includes at least one burner and at least one boiler.

[0010] In yet another aspect, the present disclosure is directed to a staged, pressurized oxy-combustion system, the system comprising a unique design including at least one burner and at least one boiler, wherein the boiler includes at least one wall ring.

[0011] In still another aspect, the present disclosure is directed to a method of capturing carbon dioxide, the method comprising using a staged, pressurized oxy-combustion system including a unique design, where the design includes at least one burner and at least one boiler, wherein the boiler includes at least one wall ring.

BRIEF DESCRIPTION OF THE DRAWINGS

[0012] FIG. 1 is an exemplary embodiment of an illustration of radiative trapping in accordance with the present disclosure.

[0013] FIG. 2 is an exemplary embodiment of the geometry of a burner and a boiler in accordance with the present disclosure.

[0014] FIG. 3 is an exemplary embodiment of a first 20-meter axial profile of the mass-averaged mean velocity in a boiler in accordance with the present disclosure.

[0015] FIG. 4 is an exemplary embodiment of the first 30-meters of a volatile reaction rate and temperature contours for a base case in accordance with the present disclosure.

[0016] FIG. 5 is an exemplary embodiment of the radiative and total heat flux profile for a base case in accordance with the present disclosure.

[0017] FIG. 6A is an exemplary embodiment of the first 30 meters of the axial velocity for different conical frustum lengths in accordance with the present disclosure. FIG. 6B is an exemplary embodiment of the first 30 meters of the temperature contours for different conical frustum lengths in accordance with the present disclosure.

[0018] FIGs. 7A and 7B are exemplary embodiments of near-wall radial temperature and optical thickness profiles at 19 meters in accordance with the present disclosure.

[0019] FIG. 8 is an exemplary embodiment of wall heat flux profiles for different conical frustum lengths in accordance with the present disclosure.

[0020] FIG. 9 is an exemplary embodiment of temperature contours with optical thickness of 2.3 meters from the wall overlaid for different burner configurations in accordance with the present disclosure. The arrows indicate the flame lengths measured by volatile reaction rates.

[0021] FIGs. 10A and 10B are exemplary embodiments of radiative and total heat flux profiles for different burner configurations in accordance with the present disclosure.

[0022] FIGs. 11A and 11B are exemplary embodiments of radiative and total heat flux profiles for different SR_{centOx} configurations in accordance with the present disclosure.

[0023] FIG. 12 is an exemplary embodiment of temperature contours for different SR_{centOx} in accordance with the present disclosure. The arrows indicate the flame lengths measured by volatile reaction rates.

[0024] FIG. 13 is an exemplary embodiment of temperature contours for 385 MW and 700 MW in accordance with the present disclosure. The arrows indicate the flame lengths measured by volatile reaction rates.

[0025] FIG. 14 is an exemplary embodiment of heat flux profiles for 385 MW and 700 MW boilers in accordance with the present disclosure.

[0026] FIG. 15 is an exemplary embodiment of a simplified process flow diagram for a three-stage SPOC power plant in accordance with the present disclosure.

[0027] FIG. 16 is an exemplary embodiment of the radiative and total heat flux for a base case (100% thermal input) and a turndown case (40% thermal input) in accordance with the present disclosure.

[0028] FIG. 17 is an exemplary embodiment of the first 30 meters of the temperature contours for a base case (100% thermal input) and a turndown case (40% thermal input) with $\tau_w = 2.3$ curve overlaid in accordance with the present disclosure.

[0029] FIG. 18 is an exemplary embodiment of the first 30 meters of the axial velocity contours for a base case (100% thermal input) and a turndown case (40% thermal input) in accordance with the present disclosure. The buoyancy induced internal recirculation is shown using iso-contour for zero axial velocity.

[0030] FIG. 19 is an exemplary embodiment of a schematic of a modified boiler design with wall rings added to increase advection in accordance with the present disclosure.

[0031] FIG. 20 is an exemplary embodiment of the first 30 meters of temperature and axial velocity contours for a base case and a turndown case with rings in accordance with the present disclosure.

[0032] FIGs. 21A-21D are exemplary embodiments of comparisons of wall heat flux for different cases in accordance with the present disclosure. FIGs. 21A and 21B are the radiative and total heat flux for a 100% thermal input case. FIGs. 21C and 21D are the radiative and total heat flux for a 40% thermal input case.

[0033] FIGs. 22A and 22B are exemplary embodiments of radial temperature and optical thickness profiles at 7.5 meters and 15.5 meters for 100% thermal input cases with and without rings in accordance with the present disclosure.

[0034] FIG. 23 is an exemplary embodiment of the first 40 meters of temperature and axial velocity contours for a 100% thermal input case with wall rings and longer expansion in accordance with the present disclosure.

[0035] FIGs. 24A and 24B are exemplary embodiments of radiative and total heat flux profiles for a 100% thermal input case with wall rings and longer expansion in accordance with the present disclosure.

[0036] FIG. 25 is an exemplary embodiment of the first 40 meters of temperature and axial velocity contours for a 40% thermal input case with wall rings and longer expansion in accordance with the present disclosure.

[0037] FIGs. 26A and 26B are exemplary embodiments of radiative and total heat flux profiles for a 40% thermal input case with wall rings and longer expansion in accordance with the present disclosure.

DETAILED DESCRIPTION OF CERTAIN EMBODIMENTS

[0038] The present disclosure is generally directed to a burner and boiler/furnace for pressurized oxy-combustion boilers and furnaces. The disclosure includes a design of a burner and boiler for a staged, pressurized oxy-combustion (SPOC) technology process and designs that affect wall heat flux. The disclosure further includes the introduction of wall rings to increase, for example, advection.

Burner and Boiler Design Concepts

[0039] To understand the burner and boiler design for SPOC, it is important to understand the concept of radiative trapping.

Radiative Trapping

[0040] Radiative trapping, in the present context, is a means to control the radiative heat flux to the boiler tubes. In some boilers and furnaces, the local radiative

emission in the central core could be very high due to the high particle number concentrations and high temperatures there. But, with radiative trapping, in some embodiments, the radiation leaving this region is “trapped” by the “optically dense medium” between the high temperature region and the wall. Depending on the effectiveness of the implementation of radiative trapping, the peak wall heat flux is reduced to a value comparable to or even lower than those in traditional boilers.

[0041] An exemplary embodiment of radiative trapping is shown in FIG. 1, which shows a boiler with high temperature in the central core and low temperature near the wall. The optical thickness from the wall, $\tau_w(x)$, in the figure is a measure of how opaque the medium between any location, x distance from the wall, and the wall is to radiation passing through it. It is calculated by integrating the local extinction coefficient of the medium, $k_e(x)$ over the radiation path length,

$$\tau_w(x) = \int_0^x k_e(x) dx.$$

[0042] The extinction coefficient is defined as the inverse of the distance that radiation travels in a medium before it is scattered and/or absorbed. τ_w increases from zero at the wall to a higher thickness when approaching the axis of the boiler. A medium is considered optically dense when τ_w is much larger than 1, because the radiation penetration depth is much smaller than the physical distance to the wall from this location.

[0043] Based on Beer’s law, the transmissivity of a medium with an optical thickness, τ_w , is given by $\exp(-\tau_w(x))$. A Higher optical thickness leads to a lower transmission of radiation through the medium. For example, if τ_w is 2.3 at a certain location, the transmissivity between this location and the wall is about 10% (about $=e^{-2.3}$), meaning that of all the incident radiation reaching this location, only 10% passes through to the wall. The rest is “trapped” in the form of back- scattering and absorption within the medium. In such a situation, even if the local emission in the system is very high in some regions, especially in regions where both the temperature and the number of emitters (particles) are very high, the incident radiation at the wall is independent of the high local emissions. Instead, it is mainly dependent on the temperature and medium

properties between the wall and the location where τ_w is 2.3. This region is termed as the “radiation penetration layer” (see, e.g., FIG. 1). Thus, when a burner and a boiler are designed so that the medium inside the boiler is optically dense and the temperature in the radiation penetration layer is low, the wall heat flux is also low, irrespective of the temperature in the core of the boiler. An optically dense medium is created by increasing the extinction coefficient of the medium. In a particle-laden medium, the local extinction coefficient of the medium is given by

$$k_e(x) = \sum_i N_{p,i}(x) Q_{e,i}(x) \frac{\pi}{4} d_{p,i}^2(x)$$

[0044] where N_p and d_p are the particle number concentration and diameter respectively, and Q_e is the extinction efficiency factor of each particle. With the same particle sizes and extinction efficiency factors, higher particle number concentration leads to larger extinction coefficient. But, for most standard boilers, where the optical thickness is low enough that the medium is optically thin, increasing the particle concentration or the extinction coefficient of the medium by a small amount increases the heat flux to the wall. This is because of the increase in the number of strong emitters (particles), without the trapping offered by an optically dense medium. Thus, from a conventional radiation perspective, a high particle number concentration leads to high flame emissions and thus high wall heat flux.

[0045] However, in a low recycle, pressurized oxy-combustion system, the particle number concentration is increased significantly due to the combination of high pressure and low dilution gas flow rate, that the medium becomes optically dense. Such a medium then effectively traps a significant portion of the radiation emanating from the core of the boiler. With a proper burner and boiler design to achieve low temperature in the radiation penetration layer, low-recycle, pressurized oxy-combustion boilers utilize radiative trapping to control wall heat flux, instead of flue gas recycle.

[0046] Note that, in some embodiments, radiation is also trapped in other pressurized systems, such as gasifiers, where the heat transfer in regions of high extinction coefficients is effectively modeled as a conduction problem rather than radiation.

Burner and Boiler Design

[0047] In some embodiments, parametric analysis of process efficiency showed that for the SPOC process, the boilers should operate at about 16 bar. Since the boiler must be a pressure vessel, a cylindrical vessel was used as the base design. To minimize the total heat transfer surface area required in this boiler, a high, yet controlled and relatively uniform heat flux is desired over a long length of the boiler. Thus, an axial-flow burner is preferred over a swirl-stabilized burner, because the latter relies on high mixing and rapid heat release near the burner, while the former has slower mixing and axially-distributed heat release. While swirling flow is required for flame stabilization in air-fired and first-generation oxy-combustion systems, here high oxygen concentration in the oxidizer flows is relied upon, as well as on specially designed flame holders. An axisymmetric axial-flow burner also helps avoid flame impingement and particle deposition on the boiler tubes (walls).

[0048] A down-fired boiler was chosen over an up-fired for a number of reasons, the most obvious being that an up-fired burner would be prone to bottom ash hitting the burner. Nonetheless, buoyancy-induced recirculation in the flame region of a down-fired boiler would push the flame radially outwards, and could possibly lead to high temperatures in the radiation penetration layer and even flame impingement on the walls. To minimize the impact of buoyancy, the initial section of the boiler was designed as the frustum of a cone, as shown in FIG. 2, so that the Richardson number (i.e., the importance of natural convection relative to forced convection) was low throughout the flame region. Such a cone-shaped boiler is effective in avoiding buoyancy-induced recirculation. The Richardson number is low at the burner head because the small cross-sectional area of the conical design yields a high velocity. As the flow moves downstream, the cross-sectional area increases, but the volumetric flow rate of the gas also increases because the coal devolatilizes and combusts, increasing the moles of gas and the gas temperature. The cone angle of the boiler is designed by matching the increasing volumetric flow rate of the gas with the increasing cross-sectional area. The first 20-meter axial profile of the mass-averaged mean velocity for the base case is shown in FIG. 3 to demonstrate the effectiveness of the conical design in maintaining a relatively uniform, high velocity throughout the flame region.

[0049] The design incorporates a central pure oxygen stream surrounded by fuel in an annulus, which is further surrounded by a secondary oxidizer (SO) with an oxygen concentration that can be varied, and in some embodiments is about 35 vol.%. The use of the central pure oxygen stream helps reduce the flue gas recycle, for a given SO oxygen concentration and overall stoichiometric ratio. Flame holders were designed as circular rings on both the surrounding oxidizer side and the central oxygen side of the fuel annulus. These rings trip both the oxidizer flows at the burner exit and cause small localized recirculation zones at the burner to enhance flame stability and attachment for a wide range of flows, including all the flow conditions studied in this disclosure. Furthermore, in some embodiments, the first 2 m of the boiler wall is refractory to further aid flame stabilization and attachment even during minor fluctuations. The design of the flame holder rings allows for uniform flow in the bulk while locally allowing for mixing, which allows the flame to sit stably on the burner.

[0050] With the use of two oxidizer streams, two flames are formed: a high temperature inner flame where the fuel and pure oxygen stream react, and a lower temperature outer flame where the fuel reacts with the surrounding oxidizer. The high number density of fuel particles between the high temperature flame and the wall effectively traps the radiation from this flame, avoiding high wall heat flux. Further downstream, with the dispersion of char and ash particles, part of the radiation from the surrounding oxidizer flame is also trapped. It is important to note here that this design with central oxygen is quite different in both intent and implementation from the use of a central oxygen lance by some atmospheric pressure systems. In the atmospheric pressure systems, an oxygen lance stabilizes the flame at the burner tip and improves boiler efficiency. In such systems, the amount of oxygen used is less than 5% of the total oxygen required for the fuel. On the other hand, in SPOC, the amount of oxygen supplied from the central tube can even be more than the oxygen required by the fuel in that stage. Furthermore, as discussed above, the placement of this large stream of pure oxygen in the center, with the fuel in the annulus, reduces the amount of flue gas recycle required while still maintaining a manageable wall heat flux because of radiative trapping by the coal particles in the annulus.

[0051] In some embodiments, the burner has a tapered shape. The tapered shape minimizes the effects of buoyancy so that no particle deposition on the walls occurs and so that the heat flux is controlled. The tapered shape keeps the Richardson number small throughout the burner, which is needed to minimize buoyance. Also, the flows between the various ports are controlled this way to avoid circulation. These two concepts keep a uniform flow through the combustor section.

[0052] Furthermore, the use of fuel-staging, with the boilers connected in series, provides a large amount of gas to the stage as compared to that required for complete combustion of the fuel in the stage. This gas creates a “buffer layer” between the flame and the wall, thus maintaining a low temperature inside the radiation penetration layer for a long distance, enhancing the effectiveness of radiative trapping.

[0053] In some embodiments, the first stage of a three-stage SPOC system was analyzed because this is the most challenging stage in terms of managing the heat flux. There are several ways in which the flows of fuel and oxidizer in the three stages are distributed. An exemplary distribution provides an understanding of the 3-stage SPOC system, with all the stages connected in series. In some embodiments, fuel is evenly distributed between each of the three stages ($385 \text{ MW}_{\text{th}}$ each). When the surrounding oxidizer’s oxygen concentration is about 35 vol.%, at least some flue gas recycle is required in the first stage. For the later stages, the flue gas from the previous stage is used to control the surrounding oxidizer concentration. Approximately 33% of the total flue gas from the exhaust of the third stage will be recycled to the first stage in this disclosure. This is less than half the amount of recycle incorporated in first-generation approaches, which is typically about 70%. The exact amount of recycle is dependent on the oxygen concentration in the secondary oxidizer and the distribution of the fuel among the various stages, as well as the coal proximate and ultimate analyses.

[0054] In some embodiments, PRB coal is used, with the analyses provided in Table 1. The coal is carried by a 1.2 kg/s stream of CO_2 . The central oxidizer was pure oxygen with a flow rate of 21.7 kg/s for the base case, corresponding to 78% of the total oxygen required for the fuel in this stage. The flow rate of the surrounding oxidizer for the base case was 69.44 kg/s.

Table 1 - Coal properties (Powder River Basin)

Proximate analysis (%wet)				Ultimate analysis (%daf)				HHV(MJ/kg)
Moisture	VM	FC	Ash	CH	O	N	S	
27.42	31.65	36.43	4.5	73.81	5.01	19.91	0.95	20.47

[0055] In some embodiments, the impacts of several different design parameters and system characteristics related to the burner and boiler on the wall heat flux are:

[0056] (1) The length of the conical frustum: By varying the length of the conical section for a given cone angle, the onset of buoyancy-induced internal recirculation in the flame region is controlled.

[0057] (2) The geometry of the burner: The sizes of the different affect the velocities of the different streams, and thus the gas mixing and the flame shape. Additionally, they also affect the distance between the flame and the wall, thus the temperature inside the radiation penetration layer. For example, a tapered shape for the burner minimalizes the effects of buoyancy and keeps the Richardson number small throughout the burner.

[0058] (3) The amount of central oxygen supplied: For a fixed surrounding oxidizer composition and flow rate, the amount of central oxygen changes both the near-field mixing and the stoichiometric ratio of the stage.

[0059] (4) Burner and boiler scaling: Different scaling methods are suitable for different systems. The optimal scaling method for SPOC boilers have been determined.

CFD Methods

[0060] In some embodiments, ANSYS FLUENT version 13.0 was used to simulate different burner designs. The flow field was modeled using the Reynolds-Averaged Navier-Stokes (RANS) equations with the Semi-Implicit Method for Pressure Linked Equations (SIMPLE). Particle trajectories were computed in the Lagrangian frame and were coupled to the gas phase. The sub-models used in this study are presented in Table 2, together with the particle and wall properties. Further details on the selection criteria for the sub-models and the sensitivity studies of the particle and wall

properties are discussed throughout this disclosure. Mesh insensitivity studies, including comparison with a 3D mesh consisting of 10 million cells, showed that a 2D axisymmetric mesh with 150,000 cells was sufficient to model the system.

Table 2 - CFD sub-models and some input parameters

	Sub-models and parameters
Turbulence	shear stress transport k- ω
Turbulence-chemistry	Finite rate/eddy dissipation model
Particles turbulent	Discrete random walk model
Coal devolatilization	Chemical percolation devolatilization model
Char oxidation	Kinetics/diffusion-limited model
Radiation	Discrete ordinates model
Gas absorption coefficient	Weighted sum of gray gases model
Coal size distribution	Rosin-rammler distribution, $d_{\text{mean}} = 65 \mu\text{m}$, $d_{\text{min}} = d_{\text{min}} = 10 \mu\text{m}$, $d_{\text{max}} = 200 \mu\text{m}$, $s = 3.5$
Particle emissivity	0.6
Particle scattering factor	0.6
Wall emissivity	0.8
Wall temperature	700 K

Results and Discussion

Low Recycle SPOC Boiler – Base Design

[0061] The use of two oxidizer streams produces a flame structure with two flame fronts, as can be seen from the volatile reaction rate contour in FIG. 4. The inner flame front (closer to the axis) results from the reaction between the fuel and pure oxygen (central oxygen), and its flame temperature is very high. The outer flame front results from the reaction between the fuel and the secondary oxidizer (SO), and its flame temperature is lower. Under these operating conditions, the two flames interact with each other and their tips merge to form a “halo” type flame structure. The temperature contour is also presented in FIG. 4. The location where the radial optical thickness from the wall is 2.3 (i.e., transmissivity is 10%) is also shown FIG. 4. The region between the wall and this curve is the radiation penetration layer. Due to the high particle number density caused by the elevated pressure and low flue gas recycle, the radiation penetration layer is very close to the wall. Furthermore, the temperature inside the radiation penetration layer is much lower than in the core of the boiler, especially in the flame region. This is

caused by a cold “buffer layer” created by secondary oxidizer stream, and the cooling effect of the water tubes. This is shown in more detail by example of a local zoomed-in view of the temperature contour in the radiation penetration layer. The heat flux profiles are shown in FIG. 5. It is clear that in the initial region, the radiative heat flux is much lower than what would be expected from such a high temperature flame (the temperature of the inner flame is as high as 2800 K) if radiative trapping was not effectively implemented. Further downstream, as the buffer layer gradually dissipates due to increased mixing, the average temperature in the radiation penetration layer increases, and so does the heat flux. The burner and boiler design still ensures that even far downstream the radiative heat flux is not very large due to the relatively lower temperature maintained in the radiation penetration layer throughout the boiler. The peak radiative heat flux is comparable to that observed in air-fired boilers, even though the temperature in the core of the boiler is much higher.

[0062] Note that for such an optically dense, axial flow boiler, especially in the radiation penetration layer, the axial gradients are much smaller than the radial gradients. And, the radiation penetration layer effectively acts like a boundary layer for radiation. Similar to the concept of other boundary layers, the radiation penetration layer is analyzed as a one-dimensional problem. Thus, the radial temperature profiles in this region are sufficient for a qualitative discussion on the impact of various design parameters and system characteristics on the wall heat flux in the following sections.

Parametric study of important design parameters and system characteristics

Effect of the length of the conical frustum

[0063] As described above, the first part of the boiler was a conical frustum. The cone angle and the frustum length were designed to avoid buoyancy-induced internal recirculation in the flame region. The cone angle was designed to match the increase in area with the increased volumetric flow rate of the gases.

[0064] For a given cone angle, a larger frustum length leads a larger boiler diameter, thus a larger surface area per unit length and a shorter boiler. Although a larger conical frustum may be desired from this perspective, it may also lead to buoyancy-

induced internal recirculation. A 10 m conical frustum length was designed as an optimal length (base design), and the impact of increasing the conical frustum length on wall heat flux is demonstrated in this section. FIGs. 6A and 6B show the first 30 m of axial velocity and temperature contours for the base design and another design with the conical frustum length increased to 20 m. An internal recirculation zone is formed at about 18 m in latter case. This recirculation makes the high temperature region shorter and “bushier”, pushing it closer to the wall and increasing the temperature in the radiation penetration layer. FIG. 7A shows a comparison of the near-wall radial temperature and optical thickness profiles at 18 m as an example of how the buoyancy-induced internal recirculation changes the temperature in the radiation penetration layer. Since the effectiveness of radiative trapping in controlling wall heat flux is dependent on ensuring a low temperature near the wall, the increased temperature in the radiation penetration layer caused by internal recirculation leads to a high heat flux. FIG. 7B compares the radial temperatures as a function of optical thickness. This profile can be used to estimate the incident radiation to the wall using a 1-D equation,

$$q_w = 2 \int_0^{\tau_w'} \sigma T^4(\tau_w') \int_0^1 -e^{-\tau_w'/\mu} d\mu d\tau_w' \quad .$$

[0065] where τ_w' is the optical thickness from the wall to the location of interest inside the medium, and τ_w is the optical thickness of the radiation penetration layer, which, in this case, is 2.3. T is the local temperature, and σ is the Stefan–Boltzmann constant. Calculations based on this equation, as shown from FIGs. 7A and 7B, show that the case with 20 m conical section has a higher heat flux in this region. Note that the figure only shows the near-wall region due to its importance in determining the heat flux.

[0066] FIG. 8 compares the heat flux profiles for the two cases. Starting from approximately 14 m from the burner, for the case with recirculation, the heat flux is higher than the case without recirculation. Further downstream, after approximately 22 m from the burner, the heat flux in the case with a longer frustum length is lower due to a larger surface area per unit length. Note that, even with equal heat fluxes, the 20 m frustum length case has a higher heat transfer rate due to a larger diameter.

Effect of burner geometry

[0067] Three different cases were simulated to understand the impact of the burner geometry, i.e., the sizes of the different tubes, on the wall heat flux. The size of the central oxygen tube was progressively increased while keeping the overall size of the burner and the hydraulic diameter of the fuel tube fixed. The hydraulic diameter of the surrounding oxidizer (SO) was thus reduced. The sizes chosen were such that in one case the inlet velocity of the central oxygen was higher than that of the SO (Case G1), in the second case they were nearly equal (Case G2), and in the third case the inlet velocity of the SO was higher than that of the central oxygen (Case G3).

[0068] The details of the three cases are listed in Table 3. Since the momentum ratios for all cases were such that external recirculation due to flow entrainment was never present, the main impact of changing the size of the burner was on the relative velocities of different streams and the distance between the flame and the wall. The difference in the velocities affects the near field mixing and combustion intensity, while the proximity of the flame to the wall increases the temperature in the radiation penetration layer.

Table 3 - Velocities and geometries of the burner

Hydraulic diameter (m)		Velocity (m/s)	
Central	Surrounding	Central	Surrounding
0.508	0.625	4.74	4.72
0.635	0.513	7.40	4.10
0.889	0.292	2.42	7.48

[0069] FIG. 9 shows the temperature contours and radiation penetration layers for the first 30 m for the three cases. The temperatures for the G2 and G3 cases are similar, whereas G1 has a bushier high temperature core in the initial region (0 - 10 m). In the radiation penetration layer, in this initial region, the temperatures for G1 and G3 are higher than for G2, as seen from the zoomed-in view of the temperature contours. Far downstream (> 22 m), the temperatures for the three cases in the radiation penetration layer are similar. This is reflected in the heat flux profiles, as shown in FIGs. 10A and 10B. In the initial region, 2 m – 10 m, the radiative heat flux of G1 seems to be the

highest, followed by that of G3 and then G2. The peak heat fluxes for the first two cases are similar, whereas that for Case G3 is higher. The higher peak heat flux for G3 is primarily due to the increased proximity of the high temperature core to the wall, and all along the flame region the temperature in the radiation penetration layer is higher than that for Case G2. Downstream of the flame, due to loss of cold buffer gas and the radial mixing of higher temperature gases for all cases the temperatures are similar in the radiation penetration layer, and hence the heat flux.

[0070] Though the peak heat flux for G1 is similar to G2, the heat flux profile of G1 has multiple peaks. Due to the high central oxygen velocity, the mixing between the central oxygen and the fuel stream increases. This leads to increased combustion intensity and a shorter volatile flame (~ 9 m for G1 compared to ~ 12 m for G2). The higher combustion intensity combined with the high radial mixing results in a higher temperature on average at short distances from the burner. Hence the temperature in the radiation penetration layer and the heat flux are higher in the initial region for G1. The first peak in the heat flux profile is related to this high mixing, short volatile flame.

[0071] In the post-flame region, the heat flux for G1 first reduces and then increases to a second peak, and then finally tails off. The first reduction in the heat flux is due to the end of the volatile flame, which results in a reduction of energy generation in the core. At these distances from the burner, the high velocity of the central oxygen stream dissipates and the mixing is similar to G2. The surrounding gas, on the other hand still has significant momentum to maintain a low temperature radiation penetration layer, especially with the reduced energy generation in the core and reduced mixing. Further downstream, the mixing of the cooler buffer gas near the wall with the hot combustion gases increases and the effective radiation temperature in the radiation penetration layer increases. The second peak in the heat flux profile occurs where the mixing between the hot combustion gases and the buffer gas is almost complete. Even further downstream, the mixing of the hot combustion gases with the colder SO gas for all cases is high enough that the temperature in the radiation penetration layer is similar and the heat flux profiles are similar, but due to lack of substantial energy generation in the core, the heat flux profiles tail off.

[0072] Thus, from the geometry cases, it is seen that with radiative trapping the peak heat flux is not strongly dependent on the relative velocities, till the flame is moved very close to the wall. Increased velocity difference between the different inlet streams increases turbulence and thus the reaction rates. The high mixing and combustion rates lead to a flatter radial temperature profile in the near burner region, and hence a higher heat flux in the initial region, followed by a decrease in heat flux due to continued trapping beyond the flame region. Further downstream a second peak in heat flux is also obtained due to buffer layer momentum loss with distance. This type of multiple peaks results in challenges in terms of steam integration.

[0073] It is clear from these cases that, to obtain a smooth heat flux profile, it is important to create a long flame away from the wall. Furthermore, a cool buffer gas flow is important to extract the most benefits of radiative trapping. For the purposes of this disclosure, case G2, which has the central oxygen inlet velocity comparable to the other inlet velocities, is used as the base case.

Effect of Central Oxygen Flow

[0074] The effect of central oxygen flow was evaluated on the basis of the central stoichiometric ratio (SR_{CentOx}), which is defined as the ratio of the oxygen supplied by only the central oxidizer to the stoichiometric amount of oxygen required by the fuel. Four different central oxygen flow rates were considered, corresponding to central stoichiometric ratios of 0.35, 0.78 (base case), 1.28 and 1.78. With a fixed SO oxygen concentration, the amount of oxygen supplied by the SO affects the flue gas recycle ratio and the buffer gas flow next to the wall. To keep the same flue gas recycle ratio and the buffer gas momentum, the SO flow rate was kept constant. The central oxygen flow, on the other hand, can be varied in the first stage by simultaneously changing the direct oxygen supply to the other stages, without affecting the overall oxygen supply to the three-stage process. In all cases, it was ensured that the total oxygen supplied to the first stage was more than required by the fuel in the stage.

[0075] In some embodiments, the central oxygen flow rates have central stoichiometric ratios of from about 0.1 to about 10.0, from about 0.2 to about 7.5, from about 0.3 to about 5.0, or from about 0.35 to about 2.0.

[0076] FIGs. 11A and 11B show the heat flux profiles for the four cases with varying central stoichiometric ratios. The peak heat flux for all the cases are very similar. But, the profiles of the high central oxygen cases are not smooth. As in burner geometry case G1, a high central oxygen flow rate increases the mixing between the fuel and oxygen and produces a shorter flame, as shown in FIG. 12, leading to “double peak” heat flux profiles for SR_{CentOx} of 1.78 and 1.28. As expected from the previous discussion, a SR_{CentOx} of 1.78 has a shorter flame, leading to an earlier peak than a SR_{CentOx} of 1.28. In the post-flame region, due to the difference in the total heat transferred till the buffer gas momentum dissipates, the SR_{CentOx} 1.78 case has a higher peak heat flux for the second peak. For the high SR_{CentOx} cases, the heat flux profile is not smooth and could pose some difficulty in terms of steam integration.

[0077] The heat flux for SR_{CentOx} 0.35, on the other hand, closely matches that for SR_{CentOx} 0.78 in the initial region. But, SR_{CentOx} 0.35 has a slightly higher heat flux that continues for a much longer distance because of the much longer flame. The higher heat flux for a long distance is due to the higher temperature in the radiation penetration layer caused by a lower stoichiometric ratio and the increased reliance of the volatile flame on the surrounding oxidizer of the SR_{CentOx} 0.35 case. With small differences in the heat flux profiles between the two lower SR_{CentOx} cases, the case with SR_{CentOx} of 0.78 was chosen as more appropriate due to the higher reliance of the fuel on the inner pure oxygen flame, thereby keeping the volatile flame away from the wall.

Reactor Scaling

[0078] With a burner and boiler designed at one utility scale, it is important to be able to easily scale the size of the boiler as per the requirement. Understanding the best scaling method to scale between various utility scales without having to go through all the burner and boiler design considerations highlighted is important. There are three major types of scaling for industrial furnaces and boilers, viz., constant combustion

intensity scaling, constant residence time scaling, and constant velocity scaling. The combustion intensity scaling itself can be divided into two methods – constant heat release per unit volume (Q/V scaling), and constant heat release per unit wall surface area (Q/S scaling). For the boiler under consideration, the former is equivalent to the constant residence time scaling, whereas the latter is equivalent to the constant velocity scaling. Thus, in some embodiments of the present disclosure, it is sufficient to consider only two different types of scaling, viz., constant velocity (or Q/S) and constant residence time (or Q/V) scaling.

[0079] To understand which is a better scaling method for SPOC, scaling the thermal input by a factor of ' f ' was considered. For a co-axial flame, as in SPOC, with a Q/S scaling, the flame length scales similar to the boiler length, both by a factor of $f^{0.5}$. With Q/V scaling, the flame length scales as f , while the boiler length scales by a factor of $f^{0.5}$. The difference in the scaling of flame length and boiler length for the different scaling methods leads to differences in the location of the peak heat flux. On a normalized length scale, the peak heat flux location for the Q/S scaling doesn't change, whereas for the Q/V scaling method, the peak heat flux location shifts downstream when scaling up and upstream when scaling down.

[0080] To show this quantitatively, the constant velocity (Q/S) scaling with a constant residence time (Q/V) scaling was done when scaling between two utility scales – 385 MW_{th} and 770 MW_{th} (per SPOC stage). These represent a three-stage SPOC power plant with a nominal power output of 450 MW_{el} and 900 MW_{el}, respectively. The purpose of this comparison is to study the sensitivity of varying the size of the boiler unit on the heat flux profile.

[0081] FIG. 13 compares the temperature and axial velocity contours of the base case (385 MW_{th}) with the scaled-up cases, using the different scaling methods. The axial distance for the two scaled cases is normalized by their respective diameters. No recirculation is observed in either of the cases, but the temperature profile of the constant velocity (Q/S) scaling case is clearly more similar to the base case (385 MW_{th}). The flue gas outlet temperature for the constant velocity scaling is similar to the base case, whereas for the constant residence time scaling, the flue gas outlet temperature is much

higher. FIG. 14 shows the heat flux profile for the base case and the two scaling cases. The constant velocity scaling results in the same peak heat flux location as the base case, whereas, as expected, the peak heat flux for the constant residence time (Q/V) scaling case is shifted downstream.

[0082] With the constant Q/S scaling method, not just the location of the peak heat flux, but also the value of the heat flux everywhere matches the base case almost exactly. This is because for the SPOC boilers, which are optically dense, heat transfer to the wall is mainly dependent on the temperature in the radiation penetration layer, which is very close to the boiler wall, a larger flame does not lead to a higher emission, as in atmospheric pressure boilers.

[0083] This shows that a constant velocity based scaling, which also ensures a relatively constant surface area per unit thermal input, results in similar heat fluxes for such axial-flow, optically dense systems. It further shows that the size of the SPOC boilers is able to be modified based on the final requirement, without having to optimize the burner and boiler design again at the new scale.

[0084] The present disclosure thus provides a practical and flexible design for the first stage boiler of a 3-stage SPOC power plant. Because the total oxygen supplied to this stage is high, potentially high wall heat fluxes are expected. The burner and boiler designs are based on the philosophy of ensuring a relatively flat and high heat flux profile, below the material constraint limitations of the boiler tubes. For additional flexibility in operation, in some embodiments an axial flow burner with pure oxygen in the center, surrounded by a fuel annulus (coal carried by CO_2), which is in turn surrounded by a secondary oxidizer is designed. The oxygen concentration is varied in the surrounding oxidizer, and in some embodiments is about 35% (by vol.). In this design, because of combustion of fuel with pure oxygen in some regions, locally the temperature is very high. Unlike a traditional approach of controlling heat flux by controlling the temperature, SPOC utilizes radiative trapping. Due to the large number concentration of particles from the fuel tube, the use of pure oxygen in the center, surrounded by a fuel annulus, ensured a high optical thickness of the medium between the pure oxygen-volatile flame and the wall. The high optical thickness, combined with a

low temperature “buffer gas” near the wall in the region from which radiation penetrates to the wall (radiation penetration layer), is used to control the heat flux. The conical design of the initial section of the boiler ensures a low Richardson number over its length and avoids buoyancy-induced recirculation, which causes a high temperature in the radiation penetration layer or even flame impingement on the wall, and hence a high heat flux. The frustum is sized to avoid such recirculation.

[0085] Other burner and boiler parameters are disclosed to understand their impact on the heat flux. The effect of both burner geometry and central oxygen flow rate affects the peak heat flux only slightly. But, with increasing velocity difference due to either reduced central oxygen tube size or increased flow, the combustion intensity and radial mixing both increased. This causes differences in the heat flux profiles in some local regions close to the burner. These differences are explained using temperature contours, and radial profiles of temperature and optical thickness. In cases with increased central oxygen velocity, the heat flux profile has a double peak structure, due to competing effects of high combustion intensity and mixing, and radiative trapping. This poses some challenge to steam integration. To avoid this, conditions with similar velocities of the different streams are used as the base design.

[0086] Finally, in some embodiments, using the results from these analyses and the principles for the design of such a burner and boiler, a constant velocity based scaling shows the effectiveness of the principles at various scales of utility boiler units.

[0087] Two scales were chosen which correspond to a 420 MW_{el} and 900 MW_{el} unit. Scaling by this method ensures the heat flux profile remains the same at different scales. This is because such a scaling ensures the heat transfer surface area scaled with the thermal input, and hence the length of the radiating region. It also ensures similar near-field aerodynamics, avoiding any non-smoothness in the heat flux profile. A comparison with a constant residence time scaling, further proves this. Thus, the present disclosure provides a new, flexible, staged, pressurized oxy-combustion boiler design. And by the systematic study of several burner and boiler parameters, its resilience to the different firing conditions is shown. The present disclosure further discloses the most ideal conditions for operation to ensure a high and relatively flat heat flux profile.

Furthermore, the present disclosure provides a toolbox which is utilized to control the heat flux profile in a pressurized boiler with localized high temperatures. Using scaling analysis, the present disclosure demonstrates the effectiveness of this design when scaled using the constant velocity scaling method.

[0088] In some embodiments, ANSYS FLUENT version 13.0 was used to simulate different burner designs.

[0089] The flow field was modeled using the Reynolds-Averaged Navier-Stokes (RANS) equations with the Semi-Implicit Method for Pressure Linked Equations (SIMPLE) algorithm to address pressure-velocity coupling. RANS has been used extensively to study the flame behaviors of pulverized fuel. Previously, none of the solid fuel combustion submodels have been validated for pressurized oxy-combustion environment. In the present disclosure, sub-models that have been extensively validated under atmospheric pressure oxy-coal combustion environment are used. It has been shown that RANS is able to provide reasonable agreement between experiments and CFD simulations under atmospheric pressure oxy-combustion conditions. The shear stress transport (SST) $k-\omega$ model was used in this simulation, which has been shown to be the combination of the best aspects of the two standard models, $k-\epsilon$ and $k-\omega$. The finite rate/eddy dissipation model was used for turbulence-chemistry interactions.

[0090] A two-step global reaction mechanism was used to capture the gaseous volatile reaction. The first step was volatile oxidation to carbon monoxide, and the second step was oxidation of carbon monoxide to carbon dioxide. More detailed reaction mechanisms together with a more advanced reaction model, i.e., eddy dissipation concept (EDC) model have also been tested. The EDC model takes account of the CO_2 and H_2O dissociation and thus gives a lower flame temperature and slightly lower peak wall heat flux than the finite rate/eddy dissipation model. In the present disclosure, the more conservative results from the finite rate/eddy dissipation model are presented. Particle trajectories were computed in the Lagrangian frame and were coupled to the gas phase. All discrete particles were assumed to be spherical. The pulverized coal particle size was assumed to follow the Rosin-rammler distribution, with an average diameter of $65\text{ }\mu\text{m}$. The minimum and maximum particle sizes were $10\text{ }\mu\text{m}$ and $200\text{ }\mu\text{m}$, respectively,

and the spread parameter of the distribution was 3.5. The effect of turbulence on the particle trajectories was accounted for with the Discrete Random Walk (DRW) model.

[0091] The devolatilization rate of the coal particles was calculated using the Chemical Percolation Devolatilization (CPD) model. NMR chemical structure parameters were calculated according to known correlations. The char surface oxidation rate with oxygen was modeled by a kinetics/diffusion-limited model. A multiple-surface-reaction char model, which includes the reactions between char particles and oxygen, carbon dioxide, and water vapor, was also considered for comparison. The simulation results for these two models were essentially the same, indicating that char oxidation is dominant in this boiler condition.

[0092] The Discrete Ordinates (DO) radiation model was used to solve the radiative transfer equation (RTE). Each octant of the entire angular space 4π at any spatial location was discretized into 5×5 solid angles (i.e. control angles), leading to a total of 200 directions. Further refined discretization did not show any noticeable improvement in terms of the distributions of temperature and incident radiation. Gaseous emission and absorption were considered using the domain-based Weighted Sum of Gray Gases Model (WSGGM). A particle emissivity of 0.6 was assumed for this study. The emissivity of coal particles is approximately 1, while that of fly ash is 0.6. Based on a sensitivity analysis, it was found that, counterintuitively, a lower emissivity leads to a higher wall heat flux due to radiative trapping.

[0093] Thus, a conservative value of 0.6 was chosen so that the results reported here will yield the maximum incident radiation at the wall. The particle scattering factor was set to be 0.6. The surface temperature of the boiler wall was assumed to be 700 K. A sensitivity analysis of wall temperature was conducted and showed that a variation from 100 K to 1000 K produced only about an 8% difference in wall heat flux. For wall emissivity, it was assumed that during operation the walls would accumulate a small amount of ash. Depending on the properties of the ash, and whether the deposit is loose, sintered or fused, values between 0.37 to 0.93 have been reported in the literature.

[0094] In the present embodiment, a relatively conservative value of 0.8 was chosen for the emissivity of the wall. A high emissivity implies a less reflective wall, and hence a higher net heat flux to the wall. A 2-D axisymmetric model was built based on the burner and boiler geometry, taking advantage of the axisymmetric nature of the boiler. Simulation results for cases with 30,000, 150,000 and 300,000 cells were compared. Negligible differences in velocity, temperature and heat transfer were observed for the last two cases. Moreover, a 3-D model with 10 million cells was constructed, and the difference between the results of the 3-D model and those of the 2-D model with 150,000 cells was negligible. Thus, only results for the 2-D model with 150,000 cells are reported.

SPOC Boiler with Low Flue Gas Recycle – Operational Flexibility

[0095] Oxy-fuel combustion is one of the leading strategies for capturing carbon dioxide (CO₂) from stationary combustion furnaces and power plants. Compared to existing “first-generation” oxy-combustion technologies, which operate at atmospheric pressure, pressurized oxy-combustion offers a unique advantage. Elevating the gas pressure leads to an increased dew point of flue gas moisture. Thus, the flue gas moisture can be condensed at a higher temperature than in atmospheric pressure systems, and the latent heat of condensation can be utilized in the steam cycle. With a proper design, pressurized oxy-combustion can significantly increase process efficiency. The impact of pressure on the efficiency of various configurations of pressurized oxy-combustion has been studied, and it has been shown that, depending on the actual configuration, the pressure for optimal plant performance ranges from about 10 bar to 16 bar. Also, since the flue gas, which is mainly CO₂ after moisture condensation, is usually required at very high pressures (~ 150 bar) for transportation, utilization, and storage, increasing the pressure of the combustion process does not impose efficiency or economic costs. On the other hand, due to the higher pressure and hence the smaller volumetric flow rates, the gas handling and cleanup equipment can be smaller. In addition to pressurizing the combustion process, reducing flue gas recycle can also improve plant performance and cost.

[0096] In order to maintain boiler tube temperatures within material constraints, the heat flux to the boiler tubes must be controlled. In a typical oxy-combustion design, this is achieved by controlling the flame and flue gas temperature by mixing the inlet oxygen with large amounts (~ 70% by volume) of recycled flue gas. Such a high flue gas recycle is costly: There are operating and capital costs for the recycling equipment and an associated performance loss from the fan's power consumption. Use of additional water or steam to control the gas temperature and hence the heat flux has also been shown to worsen the overall performance of the process because not all the latent heat of condensation can be effectively recovered. Even if all the latent heat were to be recovered, performance would suffer from additional water injection into the boiler, due to the difference in exergy between the moisture vaporizing in the boiler (heat transfer with high temperature difference) and the condensation in the recovery unit (heat transfer with lower temperature difference). A comparison of various power plant configurations – air-fired, atmospheric oxy-combustion, and two different pressurized oxy-combustion processes – using energy and exergy analysis shows that the plant configuration can have a significant impact on efficiency, and a low-recycle, dry-coal feeding plant configuration, as in SPOC, would have very high efficiency.

[0097] In some embodiments, the SPOC process is a high efficiency, low recycle pressurized oxy-combustion process, with the boilers optimized for operation at from about 10 bar to about 20 bar, from about 12 bar to about 18 bar, or about 16 bar. In some embodiments, it comprises multiple boilers (or stages) connected in series, with the fuel and oxidizer distributed to the different stages. An exemplified process flow diagram is shown in FIG. 15 for a three stage SPOC process. The oxygen supplied to the first stage is in excess of what is required stoichiometrically in that stage. The flue gas from this stage, along with the excess oxygen, after transferring heat to the steam, is fed to the next stage, where more fuel and oxygen are added. This process continues until all the fuel and oxygen is consumed. These boilers do not employ high flue gas recycle or feed the fuel as a slurry to control flame temperature. Instead, they operate with a very high core temperature, and the wall heat flux is controlled using radiative trapping achieved by a strategic design of the boiler and burner.

[0098] Apart from cost-effectively capturing carbon dioxide, another major challenge faced by power generation systems today is an increased variability in demand due to the high market penetration of renewable sources. Coal and combined cycle gas power plants, which were historically considered as baseload generators requiring minimal cycling, are now being subjected to significant cycling. New plant designs must focus on being flexible to operate at various reduced plant loads, i.e., turndowns. The impact of flexible operation on carbon capture processes, has been studied from both process and boiler operation perspectives.

[0099] For the SPOC process, many plant units such as the air separation unit and CO₂ purification unit, require turndown considerations similar to those for other oxy-combustion plants. Some other aspects of the SPOC process, such as low temperature NO_x removal (instead of a high temperature SCR / SNCR based process), are advantageous for flexible operation. In the SPOC process, with a proper distribution of oxygen and fuel between the stages, the last two stages can operate under favorable conditions during turndown. For example, if a turndown to 66% thermal input were desired, a single stage system would undergo a big change in the heat flux profile. On the other hand, due to the similarity of the exhaust gas characteristics in stages 2 and 3, the SPOC process could operate by only shutting-down stage 3, without affecting the heat flux profiles in the first two stages and hence the steam integration.

[00100] As discussed above, the SPOC system design provides additional flexibility in plant operation and the opportunity to control the oxygen concentration near the boiler tubes. In some embodiments, alternative designs are present compared to those discussed above and provide better control heat flux and boiler exhaust temperatures under varying thermal loads.

Alternative Boiler Design

[00101] The main philosophy of the design discussed above is based on creating a long flame to evenly distribute the wall heat flux, while utilizing radiative trapping to control the heat flux below material limitations, thereby enabling reduced flue gas recycling. Radiative trapping occurs in an optically dense system where the temperature

is high in the core of the system but relatively low near the wall. In this case, the radiation emitted from the high temperature region is effectively trapped in the core region. The incident radiation on the wall is mainly determined by the temperature in the radiation penetration layer, which is defined as the region from the wall up to an optical thickness (τ_w) of 2.3. In some embodiments, a burner is designed to keep the temperature in the radiation penetration layer low, and the medium inside the boiler is optically dense, the wall heat flux is low, irrespective of the temperature in the core of the boiler. This is analogous to the sun, where the core temperature is about 15.7 million K but the radiation temperature of the sun is only about 6000 K. The radiation from the core is trapped because of a high optical thickness, and only the radiation from the photosphere is able to penetrate through and leave the sun.

[00102] In the SPOC system, due to high pressure and low recycle, the particle number concentration is very high, leading to a large optical thickness. The low recycle leads to a high flame temperature in the core of the boiler, but with a strategic design of the burner and boiler, the temperature in the radiation penetration layer is kept low, thereby keeping the heat flux to the boiler tubes within material constraints. Furthermore, the burner and boiler design results in a long flame, with radiative trapping controlling the heat flux for a long distance and maintaining a relatively constant and high heat flux, within material constraints. This high heat flux reduces the total heat transfer surface area required and hence the cost.

[00103] The SPOC boiler system has many flexibilities in terms of the number of stages, and fuel and oxidizer distributions among different stages. In some embodiments of the present disclosure, a three stage SPOC system is disclosed with an even distribution of fuel between the stages ($385 \text{ MW}_{\text{th}}$ each). Constraints and requirements lead to a design with a down-fired, axisymmetric, cylindrical boiler, with the initial section designed as the frustum of a cone. The schematic of the burner and boiler from is shown in FIG. 2. The cone angle is set to maintain a high Richardson number throughout. At the burner head, the small cross-sectional area of the conical design yields a high velocity. As the flow moves downstream, the cross-sectional area increases, but the volumetric flow rate of the gas also increases because the coal

devolatilizes and combusts, increasing the volumetric flow rate of gas. Matching the cone's expansion with the increase in volumetric flow rate maintains a relatively constant and high velocity throughout the flame region, avoiding buoyancy-induced internal recirculation. Such internal recirculation in the flame region is detrimental because it pushes the flame towards the wall, increasing the average temperature in the radiation penetration layer and hence the wall heat flux. Due to this phenomenon, the cone angle and cone length were constrained compared to alternative embodiments, meaning the maximum boiler diameter is constrained. This diameter leads to a long boiler to maintain sufficient surface area for heat transfer. To make the boiler more compact, in some embodiments, a second expansion is designed in the post flame region.

[00104] In some embodiments, an axial-flow burner is chosen to reduce mixing between the fuel and the oxidizer and thus creates a long flame. It also avoids flame impingement and minimizes particle impaction on walls. The burner is made up of a central oxygen tube, surrounded by a fuel annulus, which is itself surrounded by a flow of secondary oxidizer (SO). In the present disclosure, the fuel is carried by a small flow of pure CO₂, and the secondary oxidizer composition is about 35% O₂ and about 65% CO₂ by volume. Flame holders on both SO and central oxygen side are incorporated for flame attachment at the burner. The first 2 m of the conical section is designed as a refractory wall to ensure flame attachment and stabilization, even during minor, unexpected burps or fuel clogs. The benefits are similar to those observed in industrial furnaces which have refractory walls.

[00105] With the use of two oxidizer streams, two flames are formed: a high temperature, fuel and pure oxygen inner flame, and a lower temperature, fuel and SO outer flame. The high number density of fuel particles between the high temperature flame and the wall effectively traps the radiation from this high temperature flame before it reaches the wall, avoiding high wall heat flux. Further downstream, with the dispersion of char and ash particles, part of the radiation from the lower temperature flame is also trapped. The central oxidizer (pure oxygen) delivers about 80% of the total oxygen required for the fuel in the first stage. The total amount of oxygen supplied to the first stage is more than required for the fuel in the stage. Since most of the oxygen for the first

stage is added as central oxygen, and flue gas recycle is used only in stage 1 to control surrounding oxidizer oxygen concentration, the total flue gas recycle ratio is very low. The exact amount depends on the surrounding oxidizer oxygen concentration, the configuration of the three stages, and the fuel properties. The fuel and oxidizer composition, and flow rates through the various tubes used in this disclosure for the base or designed condition (100% thermal input) are listed in Table 4.

Table 4 - Coal Properties and Operating Conditions

Coal properties (Powder River Basin)									
Proximate analysis (%wet)				Ultimate analysis (%daf)					HHV
Moisture	VM	FC	Ash	C	H	O	N	S	(MJ/kg)
27.42	31.65	36.43	4.5	73.81	5.01	19.91	0.95	0.32	20.47
Operating conditions									
		Mass flow rate (kg/s)			O ₂ (vol.%)		CO ₂ (vol.%)		Temp. (K)
Coal		18.8							
Coal carrier gas		1.2			0		100		458
Central oxygen		21.7			100		0		458
Secondary oxidizer		69.4			35		65		458

[00106] The design and flow conditions described above are considered as the baseline for analyzing the performance of the SPOC boiler under turndown conditions. A number of different turndowns were tested, including a turndown to about 40% thermal load. Other turndowns include, but are not limited to, about 90% thermal load, about 80% thermal load, about 70% thermal load, about 60% thermal load, about 50% thermal load, about 40% thermal load, about 30% thermal load, about 20% thermal load, about 10% thermal load, about 5% thermal load and about 1% thermal load.

CFD Methods

[00107] ANSYS FLUENT version 13.0 was used to simulate different burner designs. The flow field was modeled using the Reynolds-Averaged Navier-Stokes (RANS) equations with the Semi-Implicit Method for Pressure Linked Equations (SIMPLE). Particle trajectories were computed in the Lagrangian frame and were coupled to the gas phase. The sub-models used in this study are presented in Table 5, together with the particle and wall properties. Further details on the selection criteria for

the sub-models and the sensitivity studies of the particle and wall properties are discussed throughout the disclosure. In some embodiments, mesh insensitivity studies, including comparison with a 3D mesh consisting of 10 million cells, showed that a 2D axisymmetric mesh with 150,000 cells was sufficient to model the system.

Table 5 - CFD sub-models and some input parameters

	Sub-models and parameters
Turbulence	Shear stress transport k- ω
Turbulence-chemistry interaction	Finite rate/eddy dissipation model
Particles turbulent dispersion	Discrete random walk model
Coal devolatilization	Chemical percolation devolatilization model
Char oxidation	Kinetics/diffusion-limited model
Radiation	Discrete ordinates model
Gas absorption coefficient	Weighted sum of gray gases model
Coal size distribution	Rosin-rammler distribution, $d_{\text{mean}} = 60 \mu\text{m}$, $d_{\text{min}} = 10 \mu\text{m}$, $d_{\text{max}} = 200 \mu\text{m}$, $s = 3.5$
Particle emissivity	0.6
Particle scattering factor	0.6
Wall emissivity	0.8
Wall temperature	700 K

Results and Discussion

Effect of turndown on the previous design

[00108] The base case, or 100% thermal input case (T100), was chosen as an exemplary design case. The radiation and total wall heat flux for the base case T100 and the turndown case, T40 (40% thermal input), are shown in FIG. 16. As the thermal input was reduced, the peak heat flux rose, which is counter-intuitive. Also, the peak heat fluxes were higher than what is considered acceptable for standard boiler tube materials, and advanced alloys may need to be used at the peak heat flux location.

[00109] The first 30 m of the temperature contours for the two cases are shown in FIG. 17. In both cases the flames were attached, and with turndown, the flame gets shorter. Under turndown, the flame moved closer to the wall and the temperature in the radiation penetration layer increased. This is clearly shown in FIG. 17 by the zoomed-in view of the radiation penetration layer (i.e., the region between the wall and the $\tau_w = 2.3$

curve) between 8 and 10 m. The axial velocity contours for the two cases are shown in FIG. 18. During turndown, the fuel and oxidizer inputs to the boiler were reduced, leading to lower velocities in the boiler, while the flame temperature was independent of thermal input. This reduced advection lead to a higher Richardson number and a buoyancy induced recirculation (FIG. 18), further shortening the flame and pushing it closer to the wall. Furthermore, the secondary oxidizer flow acted as a high momentum cold buffer layer, leading to a relatively low near-wall temperature in the flame region. During turndown, due to lower flow rates, the buffer layer next to the wall dissipated at shorter distances, reducing the benefits of radiative trapping downstream of the volatile flame, the energy in the core reduced and the heat flux dropped.

[00110] Even though properly designed for 100% thermal input, two main aspects of the design need to be adapted for reduced thermal input: avoiding buoyancy induced adverse pressure gradients, and having a cold buffer gas flow over a long distance from the burner. One way to achieve both with the same tool is discussed below.

Alternative boiler design for operational flexibility

[00111] In some embodiments, to enhance the effect of radiative trapping for reduced thermal input, the high temperature gases must be “focused” to the center (axis) of the reactor, and not allowed to spread very quickly to the walls. Also, advection should be kept high, even at the reduced flow rates corresponding to the turndown conditions. To achieve this, in some embodiments, small obstructions in the form of rings, are added to the wall so that small, localized re-circulations are formed in their wake next to the wall, which effectively pushes the remaining flow away from the wall, keeping the near wall region relatively cold for a longer distance. The displaced flow essentially increases the advection in the main flow, reducing the effect of buoyancy forces, and consequently the bulging of the flame towards the wall the added advantage of such a modification is that the size (diameter) of the boiler is increased further without causing buoyancy-induced internal recirculation in the flame region, as was observed upon increasing the boiler diameter for the base design discussed in alternative embodiments of the present disclosure.

Effect of Wall Rings

[00112] In some embodiments of the present disclosure, the boiler includes rings with a rectangular cross-section on the wall of the boiler, starting at about 2 m downstream of the burner and repeating about every 1 m up to 31 m from the burner. In some embodiments, the wall rings are present from about 0.1 m, about 0.5 m, about 1 m, about 2 m, about 3 m about, 5 m, about 7 m or about 10 m downstream of the burner. In some embodiments, the wall rings repeat from about 0.5 m to about 50 m from the burner, including various lengths in between.

[00113] In some embodiments, each wall ring includes a radial projection from about 0.05 m to about 5 m, from about 0.1 m to about 3 m. In some embodiments, each wall ring has a thickness of from about 0.01 to about 0.1 m, from about 0.02 to about 0.05 m. In some embodiments, each wall ring has a radial projection of about 0.1 m and a thickness of about 0.0254 m. In some embodiments, the rings are made of refractory material or water-cooled tubes. In some embodiments, the rings are modeled as adiabatic walls. Since the surface area of the rings is negligible compared with the total surface area of the wall, the boundary conditions of the rings have little impact on the overall heat flux profile. An exemplary schematic of a boiler with these wall rings is shown in FIG. 19. The axial velocity and temperature contours for the 100% thermal input (T100) and 40% thermal input (T40) case with wall rings are shown in FIG. 20. As desired, the use of these wall rings increases advection and eliminates the buoyancy induced internal recirculation in the T40 case. The temperature profile also shows a longer and thinner high temperature zone, rather than the short and bushy high temperature zone observed for the turndown case without rings (FIG. 17).

[00114] The radiative and total heat flux profiles for the 100% and 40% thermal input cases with and without the wall rings are shown in FIGs. 21A-21D. Before discussing the effectiveness of the rings to control the peak heat flux under turndown, it is important to discuss the key features of the heat flux profile with rings. The heat flux profile with rings has periodic dips for both radiative and total heat flux. These dips are located on the rings and at the corners joining the wall and the rings. Since these rings are modeled as adiabatic surfaces, the total heat fluxes on these rings are zero. The low

heat fluxes at the corner regions are related to the localized recirculation zones created in the wake of the rings, which cause effectively stagnant regions at the corner. These stagnant regions, due to negligible mixing with the combustion gases, are at low temperatures, thus the heat fluxes to the wall at these locations are very small. Thus, in some embodiments, about every 1 m, dips in heat flux are observed for the cases with rings.

[00115] For the turndown (T40) case with rings, the total heat flux peak is reduced by more than 100 kW/m^2 compared to without rings. The location of the peak heat flux shifts downstream as well, due to the reduced impact of buoyancy. Similarly, the use of rings reduces the radiative heat flux for the T100 case in the flame region. But with increased convection, the total heat fluxes in the T100 cases are similar with and without rings. However, for the T40 case, since the peak heat flux was caused primarily by high temperatures in the radiation penetration layer, the reduction in radiative heat flux surpasses the modest increase in convective heat flux, and the total is lower with rings than without. In the post-flame region, the heat flux is slightly higher, especially for the T40 case, because of different flue gas temperatures caused by different upstream heat transfer rates to the boiler tubes.

[00116] The effect of the rings on the heat flux profile is understood by analyzing the way rings affect the local and global fluid dynamics. Locally, the rings create small recirculation zones, which enhance local mixing. Globally, these small recirculation zones push the main flow towards the core (axis) of the boiler since the near-wall gases start at a lower temperature and there is limited spread of energy from the core of the boiler to these near-wall gases, the enhanced mixing creates a thicker buffer layer of “cold” gases compared with the case without rings, leading to a lower heat flux. Radial temperature and optical thickness profiles provide a more quantitative picture of the effect of wall rings on heat flux.

[00117] FIGs. 22A and 22B show the radial temperature and optical thickness profiles at an axial distance of 7.5 m (flame zone) from the burner for the 100% thermal input cases with and without the rings. Note that the figure only shows the near-wall region due to its importance in determining the heat flux. FIG. 22A shows the

temperature and optical thickness profile against the radial distance from the wall, while FIG. 22B shows the temperature as a function of optical thickness from the wall. Clearly, the use of rings creates a flatter profile near the wall due to enhancement in local mixing. In radiation penetration layer (i.e., the region where τ_w is less than 2.3), the temperature and optical thickness gradients in the axial direction are negligible compared with those in the radial direction. Thus this layer can be reasonably considered as a 1-D medium, from which the radiation power can be estimated. Qualitatively, it can be seen that in most of the region, the temperature is higher for the case without rings than with rings. The difference in heat flux can also be estimated more quantitatively by the following simple 1-D equation,

$$q_w = 2 \int_0^{\tau_w'} \sigma T^4(\tau_w') \int_0^1 -e^{-\tau_w'/\mu} d\mu d\tau_w'$$

[00118] where τ_w' is the optical thickness from the wall to the location of interest inside the medium, and τ_w is the optical thickness of the radiation penetration layer. T is the local temperature, and σ is the Stefan–Boltzmann constant. From calculations, the radiative power for the case with rings is more than 200 kW/m² lower than for the case without rings at 7.5 m. A similar analysis for T40 case produces a similar result, and hence for concision is not shown.

Effect of longer expansion with wall rings

[00119] The boiler diameter in some embodiments is constrained to 2.83 m by the need to maintain a high Richardson number in order to avoid buoyancy- induced internal recirculation in the flame region. With the addition of wall rings, and the associated increase in advection, the cone length is increased without causing internal recirculation. In some embodiments, the boiler was extended up to about 4 m in diameter by extending the conical frustum length to about 25 m from the burner. This diameter was chosen such that off- site manufacturing and shipping of the boilers on railroads would be possible. In some embodiments, the diameter is changed with evolving capabilities for shipping the boilers. FIG. 23 shows the first 40 m of temperature and axial velocity contours for the base case with this design. Unlike the case without rings, the increased advection with rings for the 100% thermal input case was high enough to

avoid any recirculation. The resulting heat flux profile is shown in FIGs. 24A and 24B. The peak heat flux for the T100 case is more than 150 kW/m^2 lower for the longer conical frustum boiler (i.e., larger diameter) than for the shorter conical frustum boiler (i.e., smaller diameter). The peak heat flux for the T100 case is less than 500 kW/m^2 , and hence standard boiler tube materials can be utilized everywhere in the boiler. Furthermore, with a larger boiler diameter, more heat transfer surface area is available per unit length of the boiler, and hence a shorter boiler can be designed with a larger diameter.

[00120] FIG. 25 shows the temperature and axial velocity contours for the first 40 m of the 40% thermal input case with larger boiler diameter. In this case, recirculation starts at approximately 15 m. Since the volatile flame length is approximately 8.5 m, the recirculation does not affect the volatile flame. Thus, the heat flux profiles for the two diameters firing at 40% of the designed thermal input (shown in FIGs. 26A and 26B), shows negligible difference in the flame region. Downstream, the heat flux drops faster for the larger diameter case due to a larger heat transfer surface area per unit length of the boiler be designed with a larger diameter.

[00121] Because the exhaust flue gas temperature must be reduced to an acceptable value before convective tubes can be added, the length of the radiant boiler is dependent on the total heat transferred per unit length. In some embodiments, increasing the diameter of the boiler helps reduce its overall length. In some embodiments, this is achieved by creating a second expansion downstream of the flame and high temperature region, where the gases are sufficiently mixed. The design with rings enables a continued expansion of the conical frustum and hence a larger diameter boiler is designed with less loss of surface area and a simpler construction. The rings, especially in the flame region, also improve the performance of the boiler under varying thermal loads. The number, size and shape of these rings are not limited by this disclosure. That is, the number, size and shape of these rings vary depending upon the desires and objectives of the user.

[00122] The present disclosure discusses the impact of low-thermal load operation of SPOC boilers on wall heat flux. The first stage of the three-stage process was analyzed because it was considered as the most challenging one in terms of

controlling the heat flux. The results from the analysis of this stage are utilized in the design of the other stages as well. Since SPOC boilers use radiative trapping to control heat flux, importance was placed on the effect of turndown on the overall fluid dynamics, especially in relation to the impact on the temperature in the radiation penetration layer. The turndown considered in this disclosure was 40% of the designed thermal load, and the flame characteristics and heat flux profiles were studied. By shutting down the last stage boiler, the multi-stage approach of SPOC effectively achieved equivalent turndowns with lower impact on heat flux profiles, and hence reduced difficulties in steam integration, compared to a single stage process.

[00123] In terms of boiler design, the restriction on the length of the conical section due to buoyancy-induced recirculation, and the excessive peak heat flux during turndown operations were addressed with the use of wall rings. The wall rings enhanced the effectiveness of radiative trapping in the flame region, and also enhanced advection. The increased advection allowed increasing the size of the conical frustum section from about 10 m to about 25 m, so that the boiler diameter was about 4 m, without causing an increase in heat flux due to buoyancy-induced recirculation. The increased boiler diameter, on the other hand, reduced the peak heat flux under the designed firing conditions. With a 40% turndown, even though the buoyancy-induced internal recirculation occurs, it was far downstream of the volatile flame, and had no impact on the shape of the high temperature core. Thus, the temperature in the radiation penetration layer did not increase and the heat flux in the flame region was similar with and without expansion. With the final boiler design, incorporating wall rings and a longer conical frustum section, the resulting heat flux profile under both thermal inputs presented here was less than 500 kW/m^2 , thereby enabling the use of standard boiler tube material everywhere.

Equivalents and Scope

[0124] In view of the above, it will be seen that the several advantages of the disclosure are achieved and other advantageous results attained. As various changes could be made in the above processes and composites without departing from the scope of the disclosure, it is intended that all matter contained in the above description and

shown in the accompanying drawings shall be interpreted as illustrative and not in a limiting sense.

[0125] When introducing elements of the present disclosure or the various versions, embodiment(s) or aspects thereof, the articles “a”, “an”, “the” and “said” are intended to mean that there are one or more of the elements. It is also noted that the terms “comprising”, “including”, “having” or “containing” are intended to be open and permits the inclusion of additional elements or steps.

WHAT IS CLAIMED IS:

1. A staged, pressurized oxy-combustion system, the system comprising:
a unique design including at least one burner and at least one boiler.
2. The system of claim 1, wherein the design further includes a central oxygen stream.
3. The system of claim 2, wherein the design further includes an annulus comprising a fuel.
4. The system of claim 3, wherein the central oxygen stream is surrounded by the fuel.
5. The system of claim 3, wherein the design further includes a secondary oxidizer stream.
6. The system of claim 5, wherein the secondary oxidizer surrounds the central oxygen stream.
7. A method of capturing carbon dioxide, the method comprising:
using a staged, pressurized oxy-combustion system including a unique design, where the design includes at least one burner and at least one boiler.
8. The method of claim 7, wherein the design further includes a central oxygen stream.
9. The method of claim 8, wherein the design further includes an annulus comprising a fuel.
10. The method of claim 9, wherein the central oxygen stream is surrounded by the fuel.
11. The method of claim 9, wherein the design further includes a secondary oxidizer stream.

12. The method of claim 11, wherein the secondary oxidizer surrounds the central oxygen stream.
13. The method of claim 7, further including reducing a flue gas cycle.
14. The method of claim 13, wherein the flue gas cycle is reduced by radiative trapping.
15. The method of claim 13, wherein the flue gas cycle is reduced by creating a long flame to evenly distribute a wall heat flux.
16. A staged, pressurized oxy-combustion system, the system comprising:
a unique design including at least one burner and at least one boiler,
wherein the boiler includes at least one wall ring.
17. The system of claim 16, wherein the design further includes a central oxygen stream.
18. The system of claim 17, wherein the design further includes an annulus comprising a fuel.
19. The system of claim 18, wherein the central oxygen stream is surrounded by the fuel.
20. The system of claim 18, wherein the design further includes a secondary oxidizer stream.
21. The system of claim 20, wherein the secondary oxidizer surrounds the central oxygen stream.
22. A method of capturing carbon dioxide, the method comprising:
using a staged, pressurized oxy-combustion system including a unique design, where the design includes at least one burner and at least one boiler, wherein the boiler includes at least one wall ring.

23. The method of claim 22, wherein the design further includes a central oxygen stream.
24. The method of claim 23, wherein the design further includes an annulus comprising a fuel.
25. The method of claim 24, wherein the central oxygen stream is surrounded by the fuel.
26. The method of claim 24, wherein the design further includes a secondary oxidizer stream.
27. The method of claim 26, wherein the secondary oxidizer surrounds the central oxygen stream.
28. The method of claim 22, further including reducing a flue gas cycle.
29. The method of claim 28, wherein the flue gas cycle is reduced by radiative trapping.
30. The method of claim 28, wherein the flue gas cycle is reduced by creating a long flame to evenly distribute a wall heat flux.
31. The system of any preceding claim, wherein the burner has a tapered shape.
32. The method of any preceding claim, wherein the burner has a tapered shape.
33. The system of any preceding claim, wherein the burner includes at least one flame holder ring.
34. The method of any preceding claim, wherein the burner includes at least one flame holder ring.

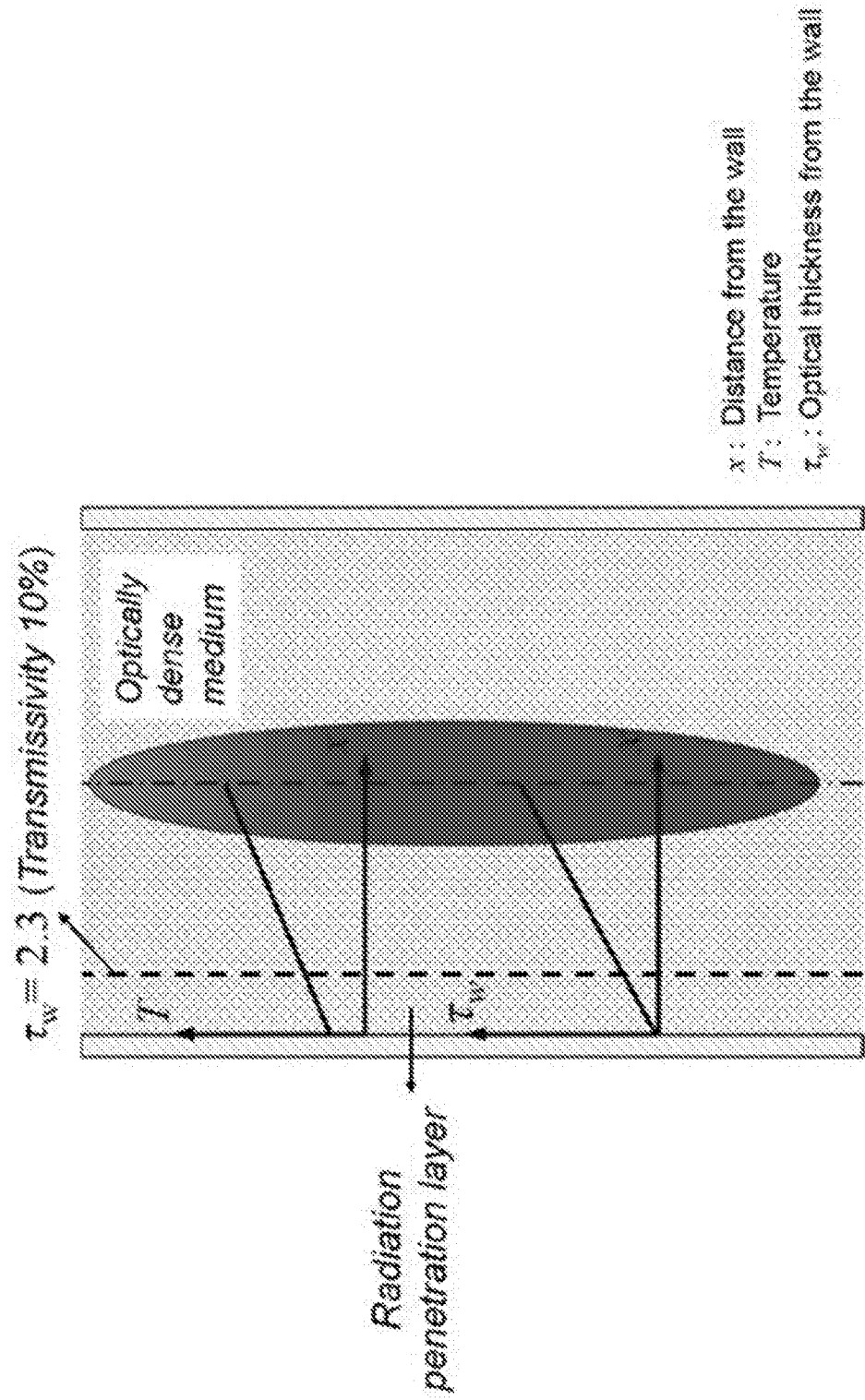


FIG. 1

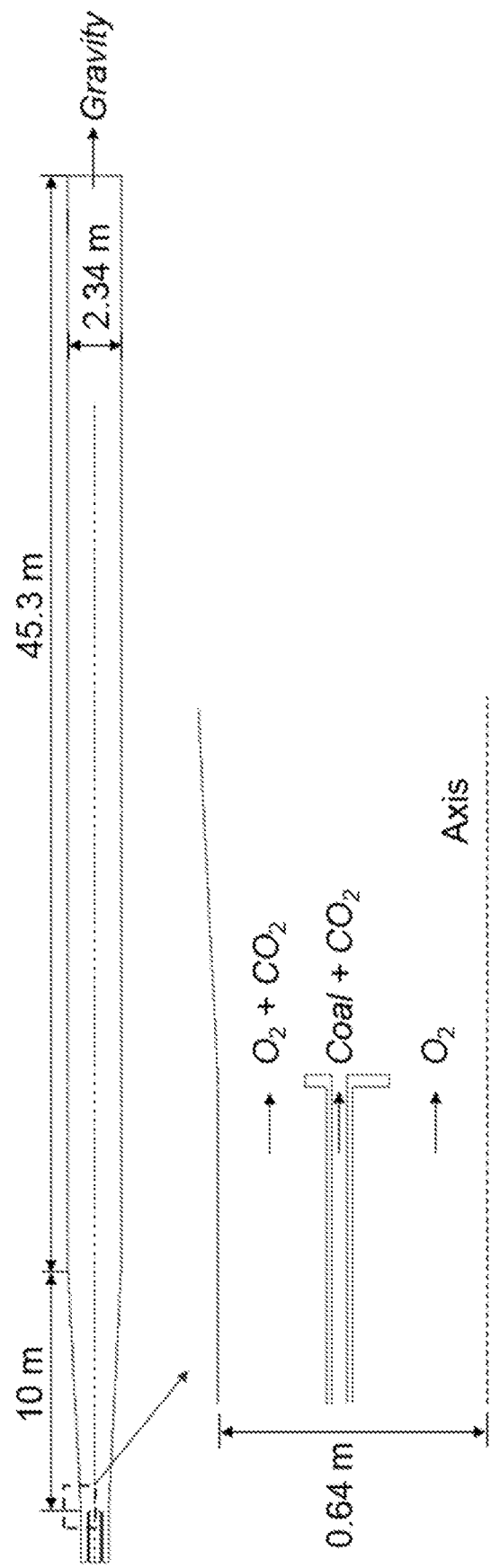


FIG. 2

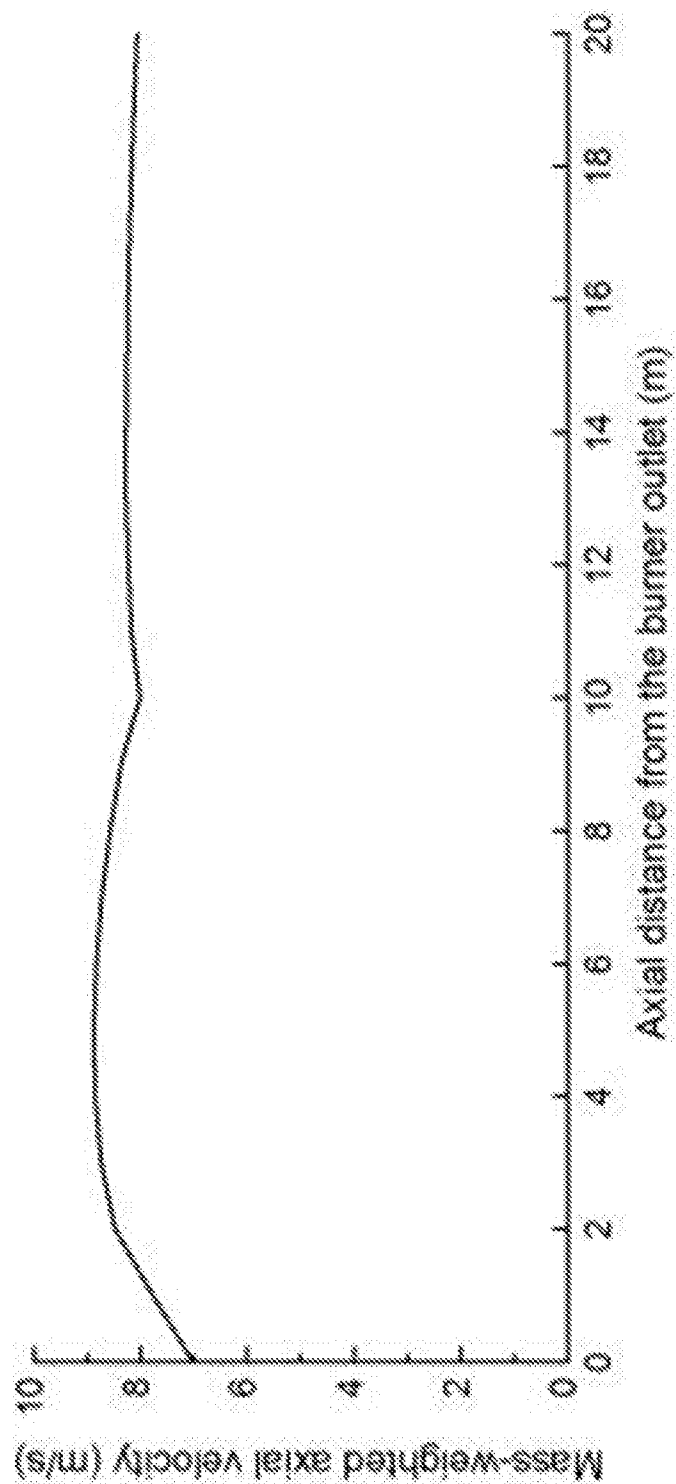


FIG. 3

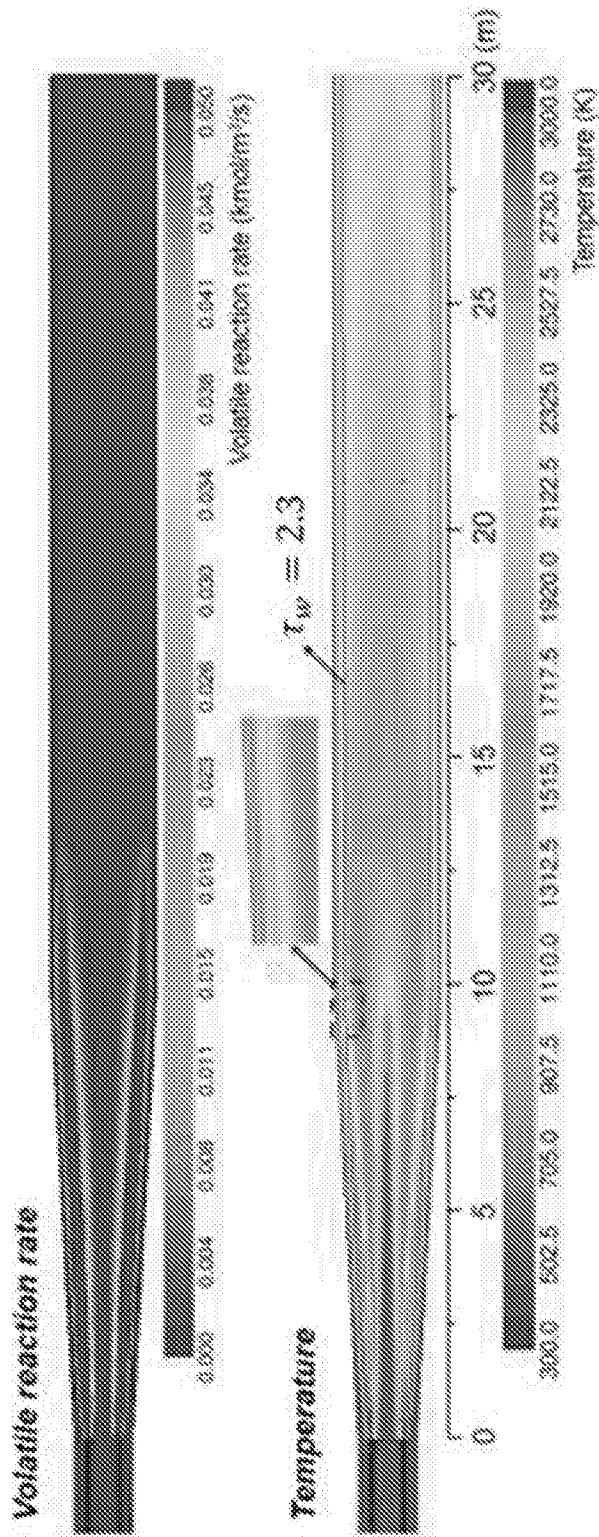


FIG. 4

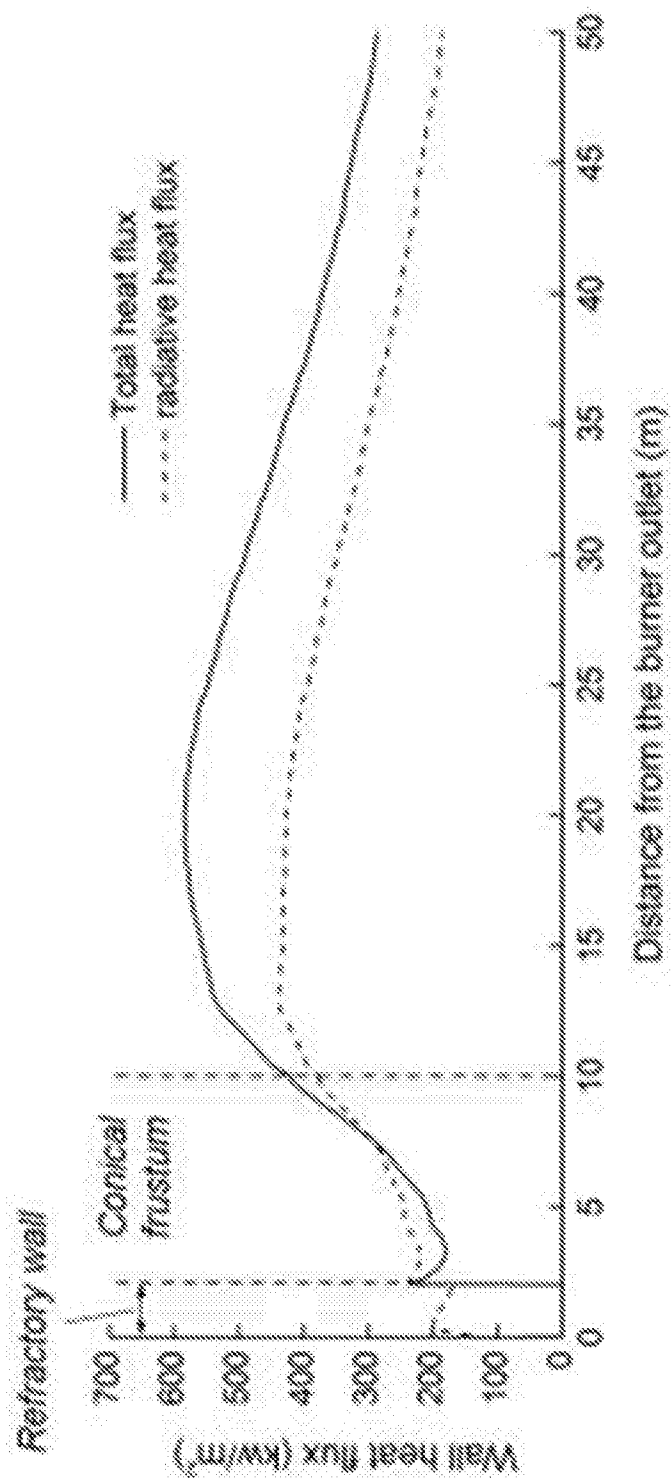


FIG. 5

FIG. 6A

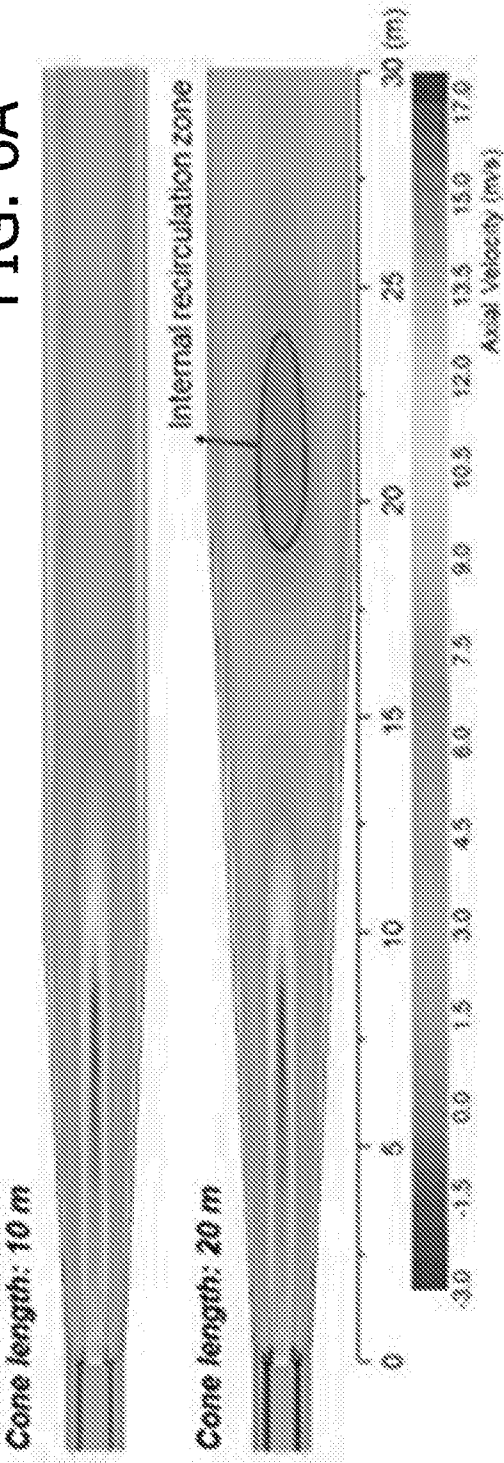
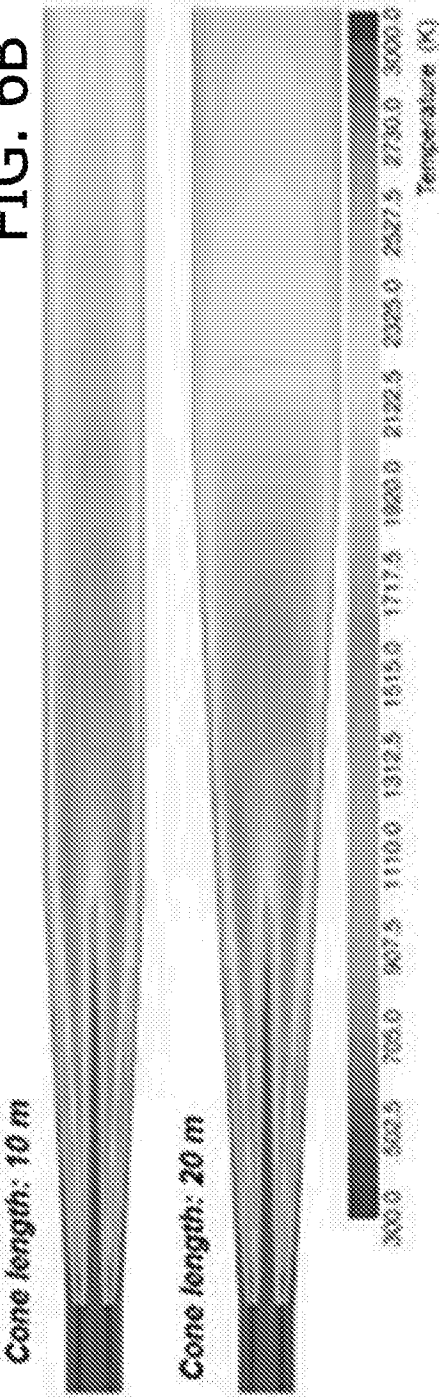


FIG. 6B



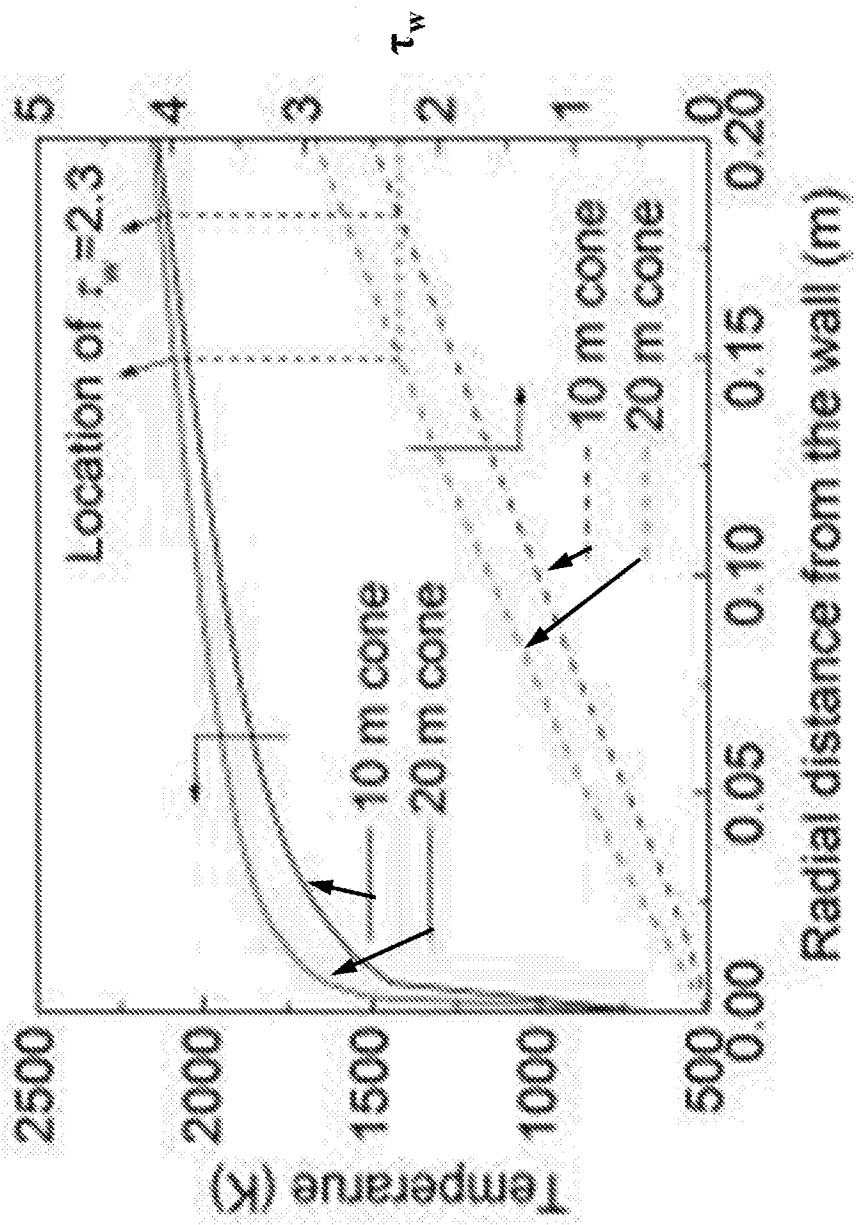


FIG. 7A

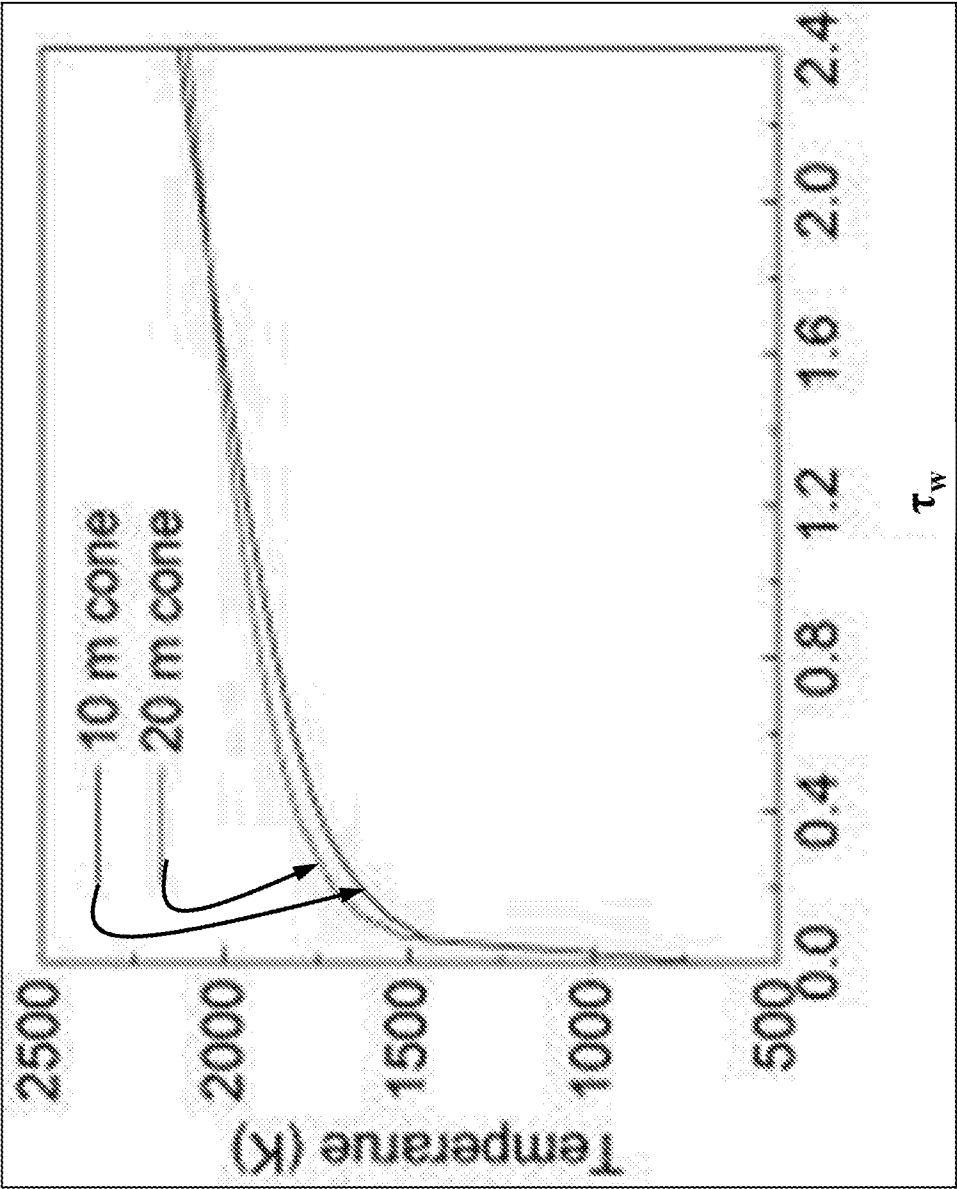


FIG. 7B

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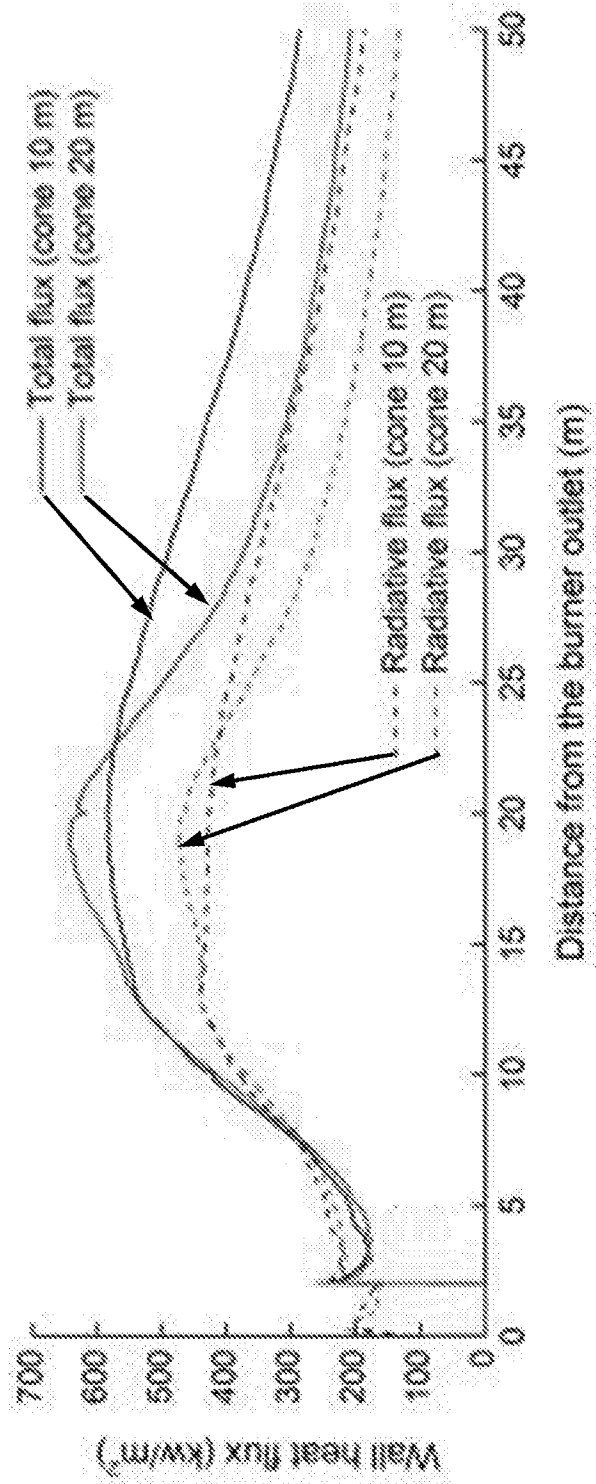


FIG. 8

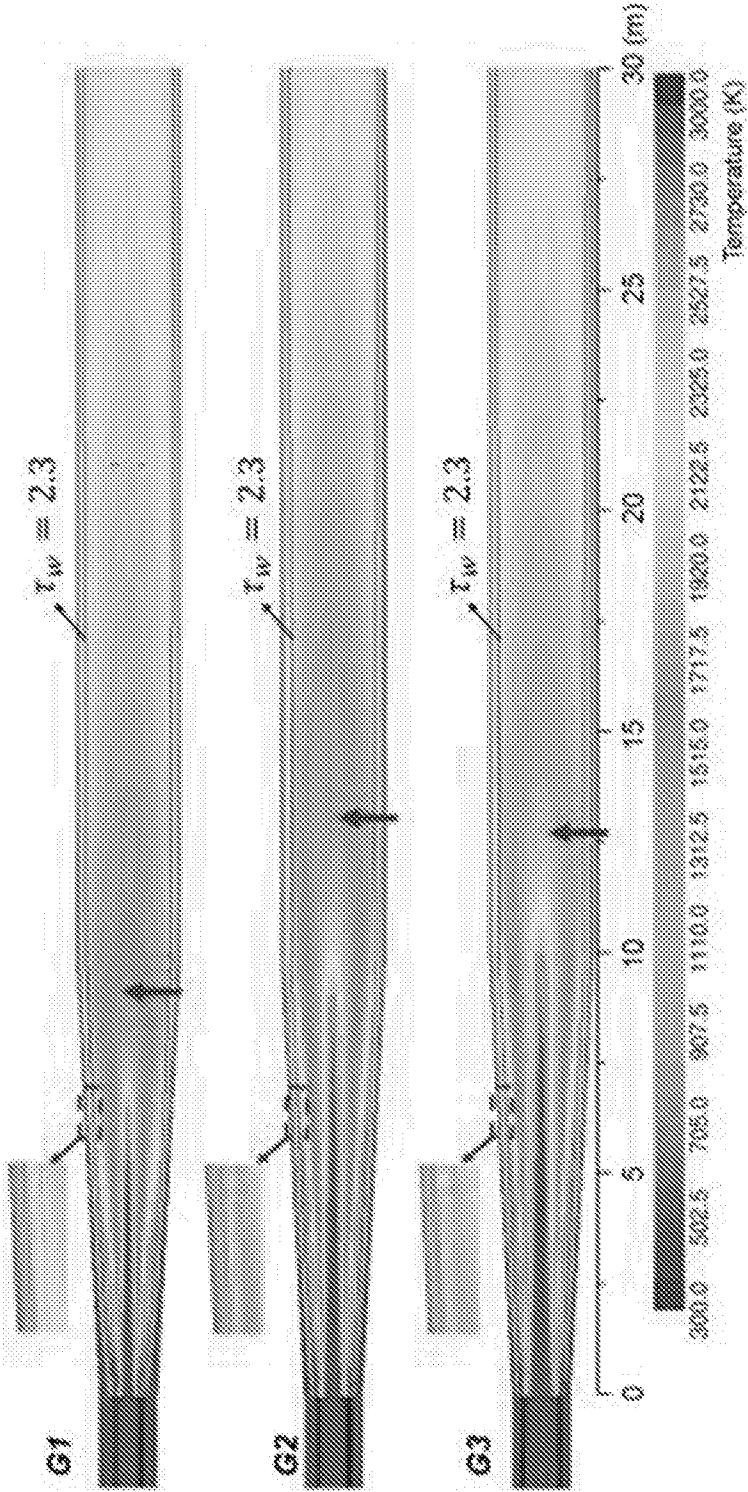


FIG. 9

FIG. 10A

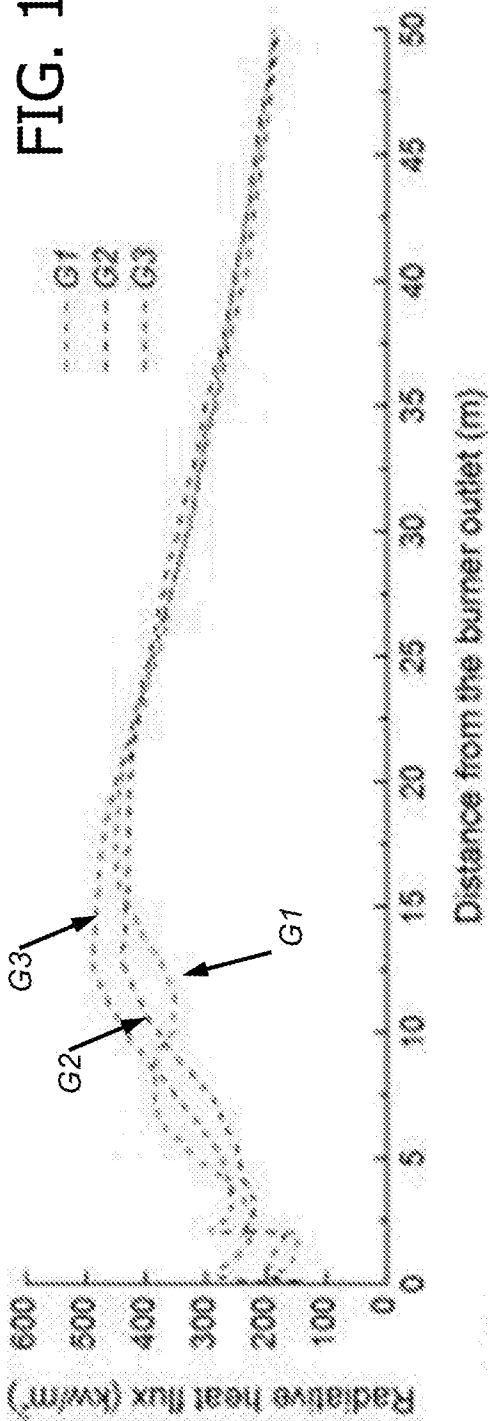
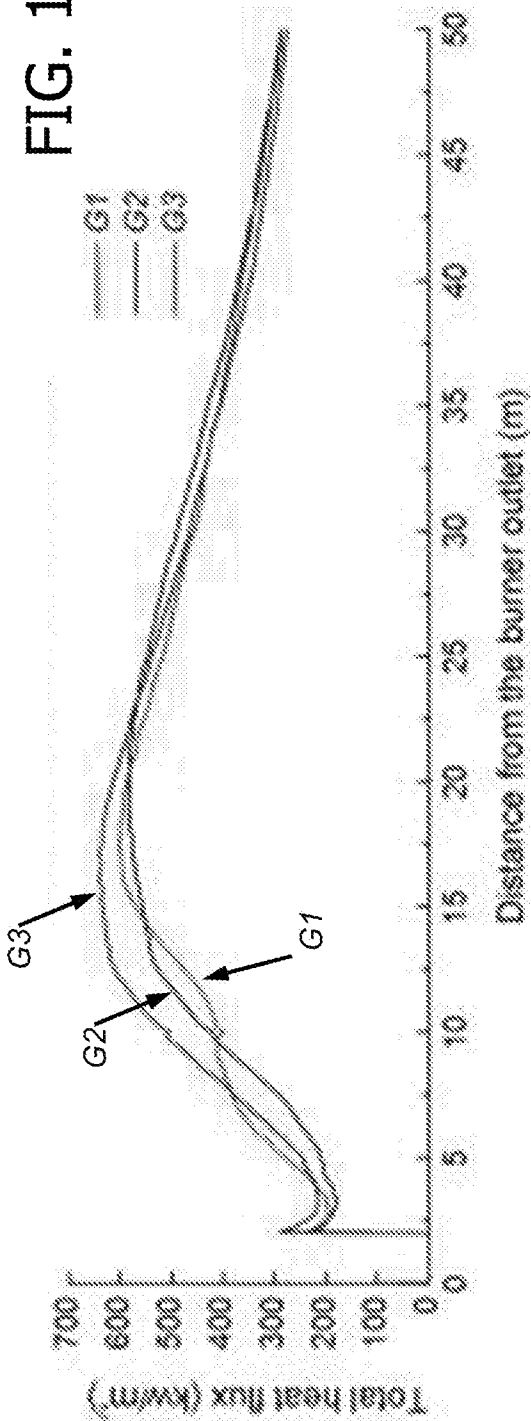
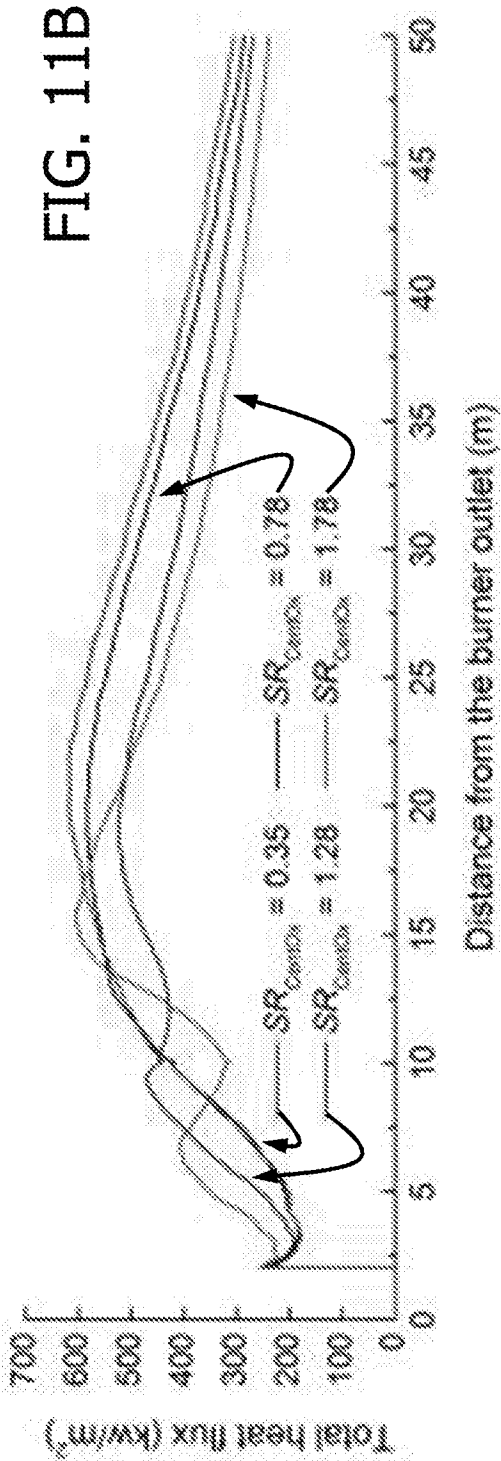
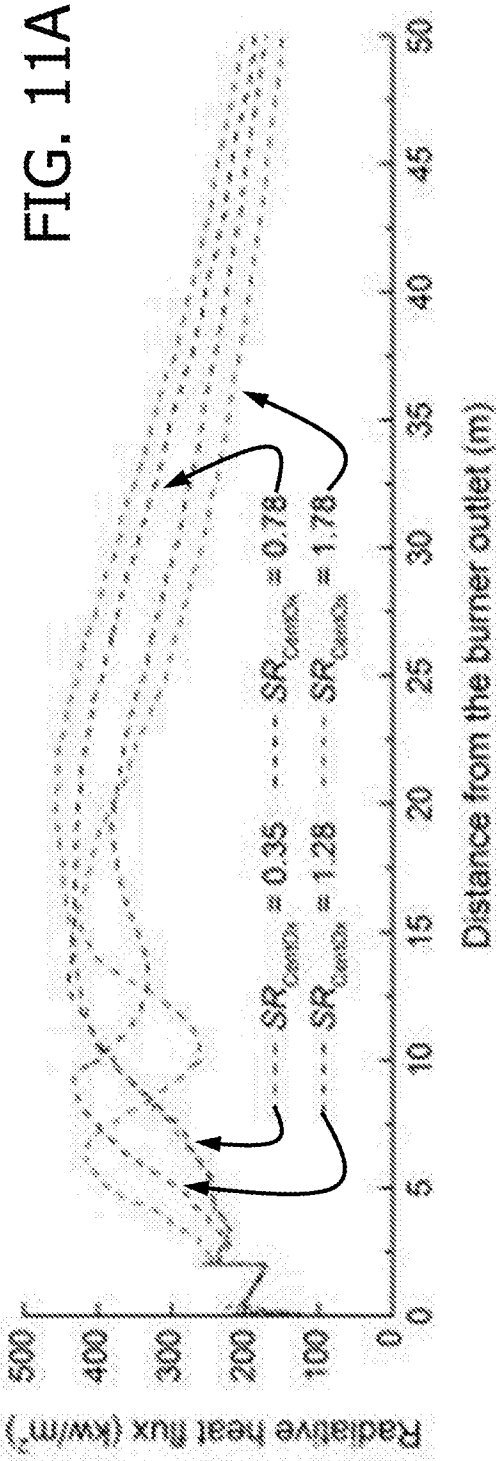


FIG. 10B





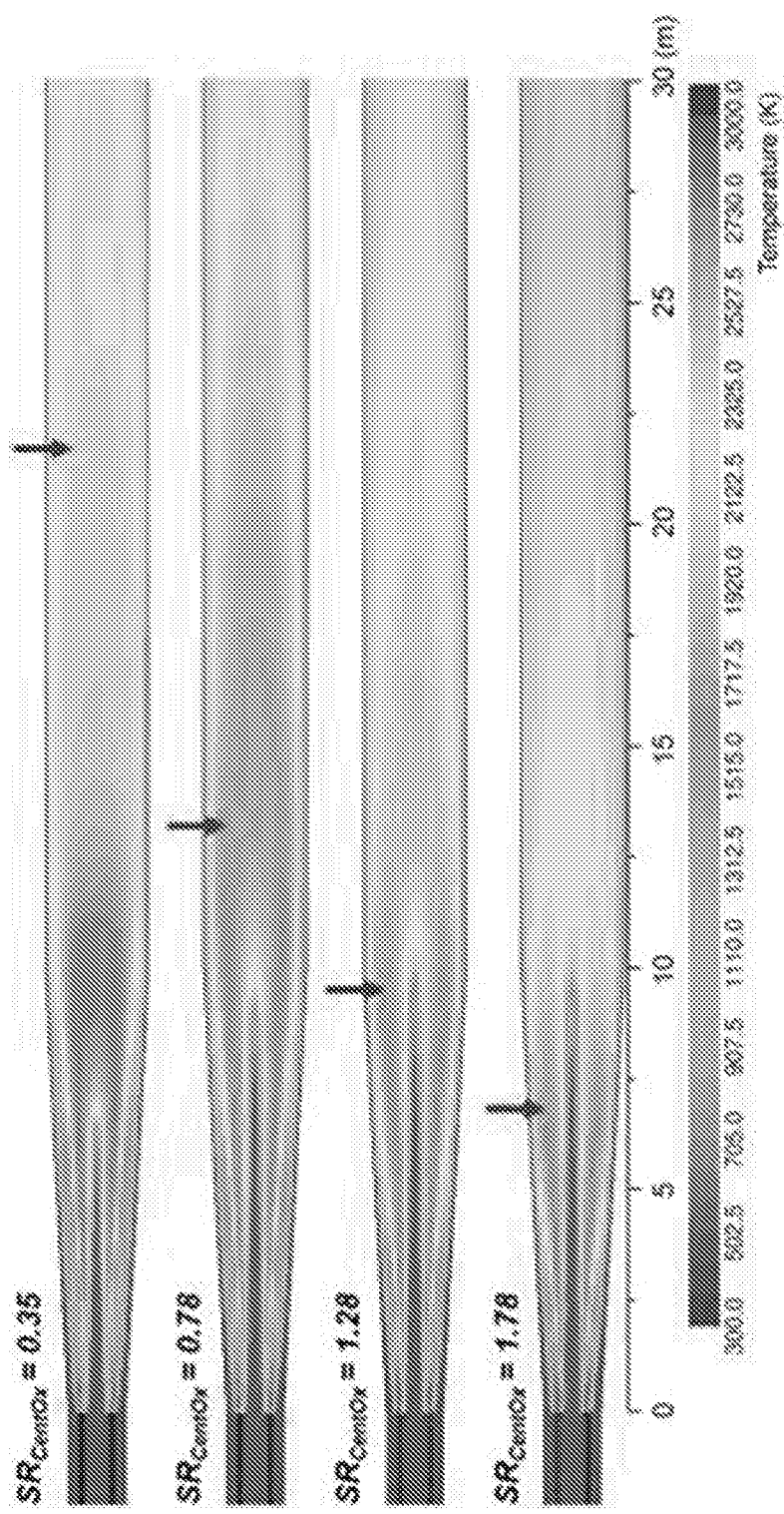


FIG. 12

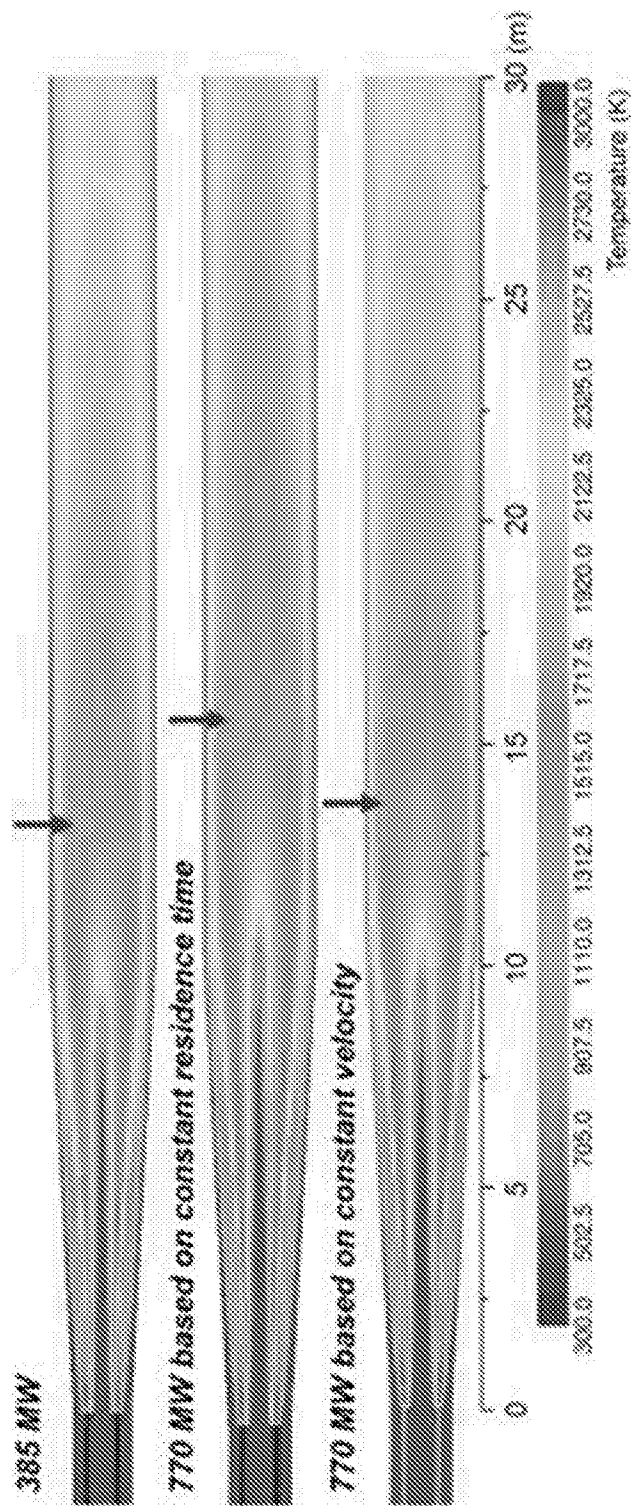


FIG. 13

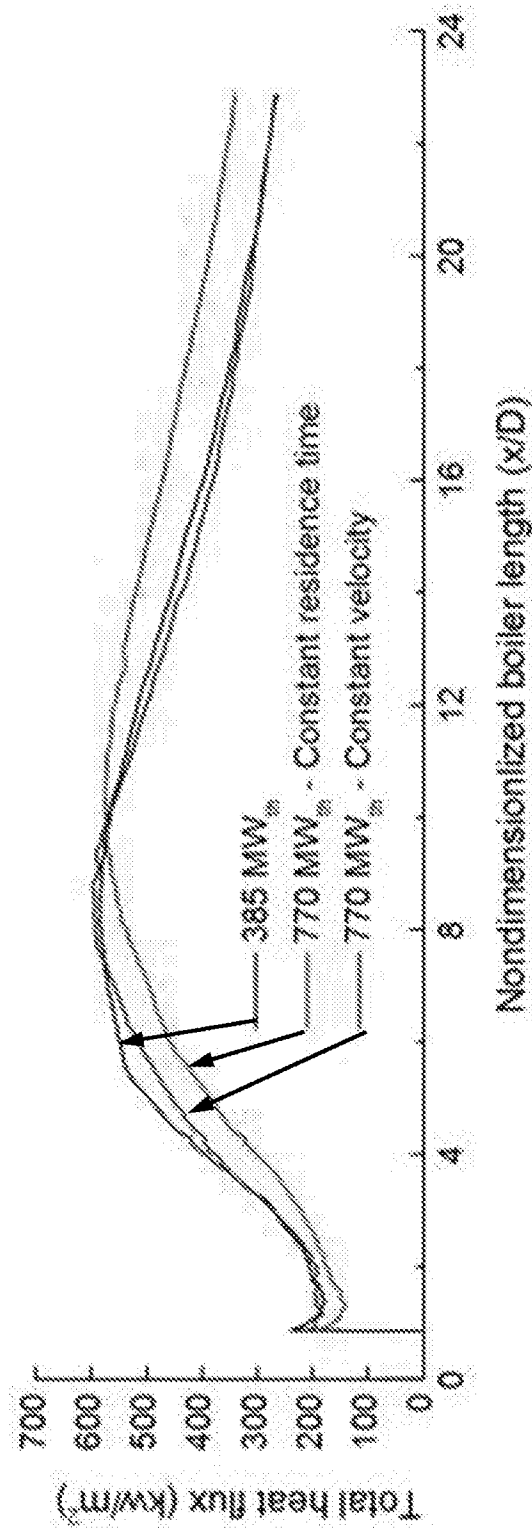


FIG. 14

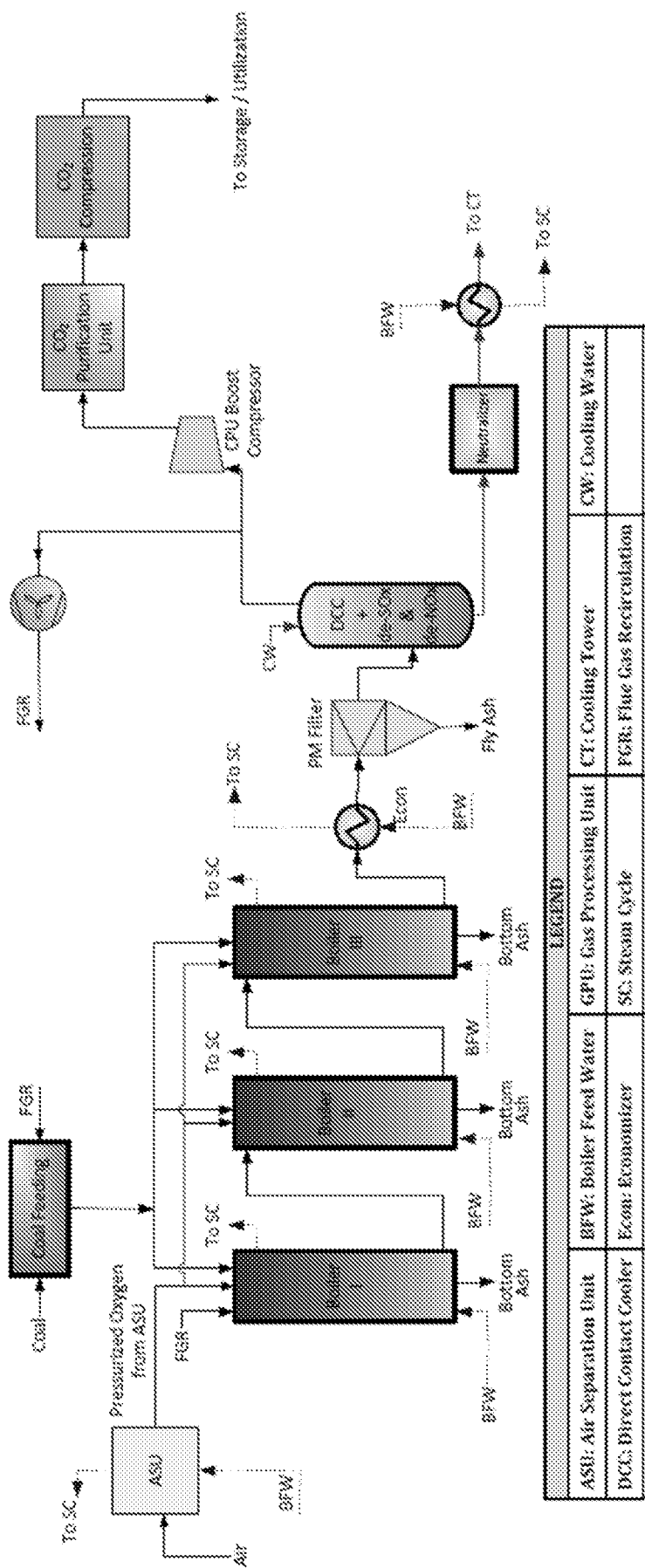


FIG. 15

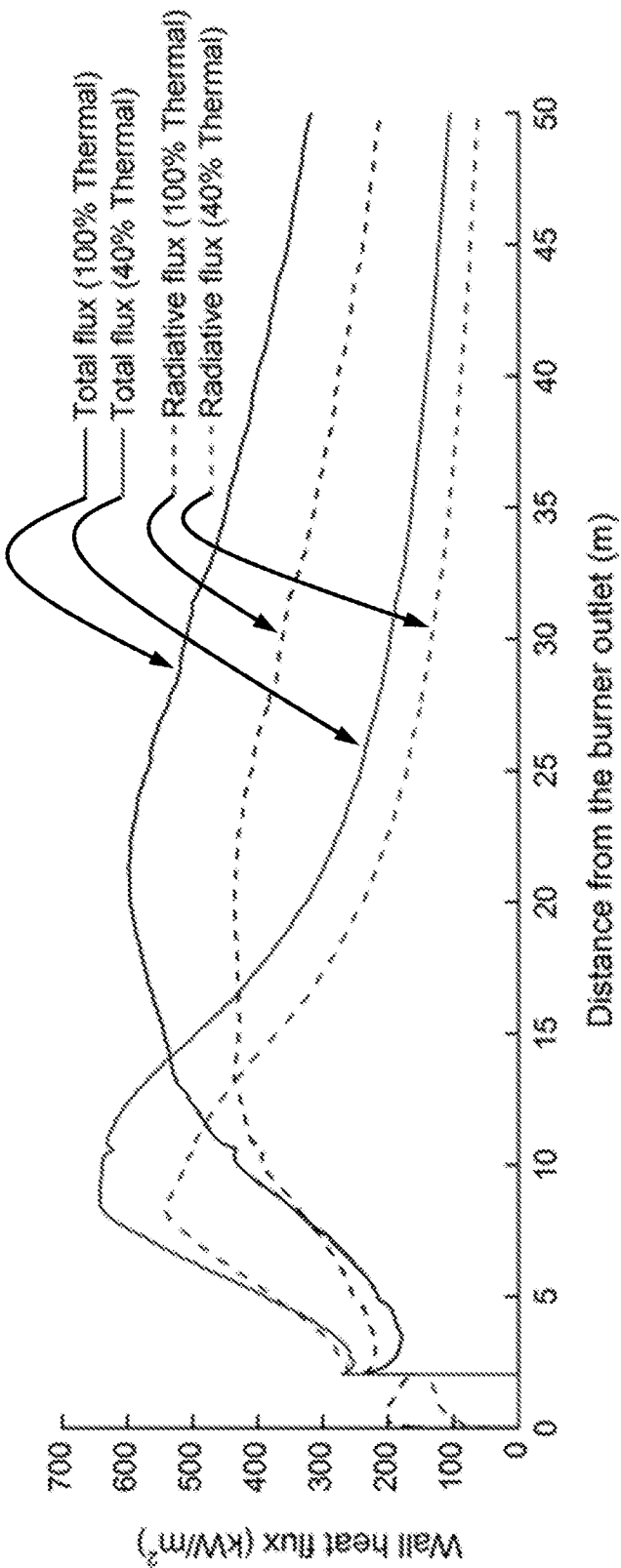


FIG. 16

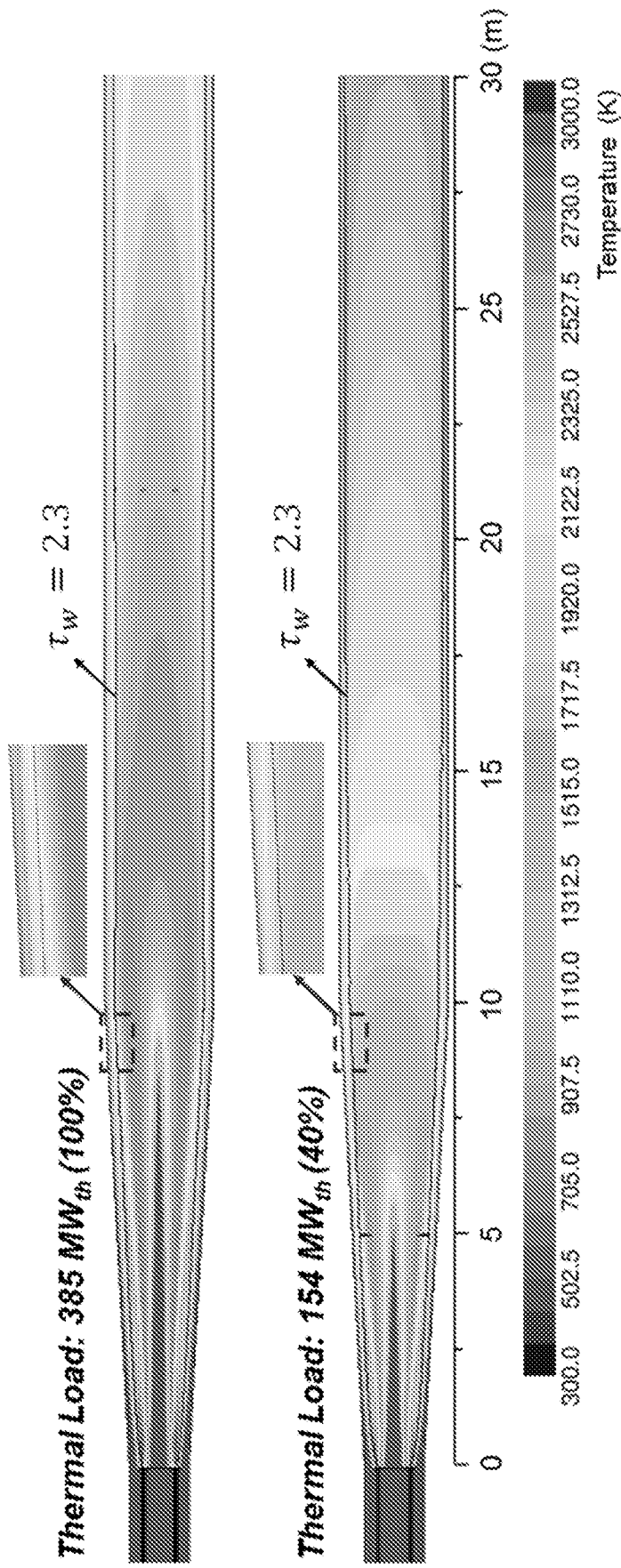


FIG. 17

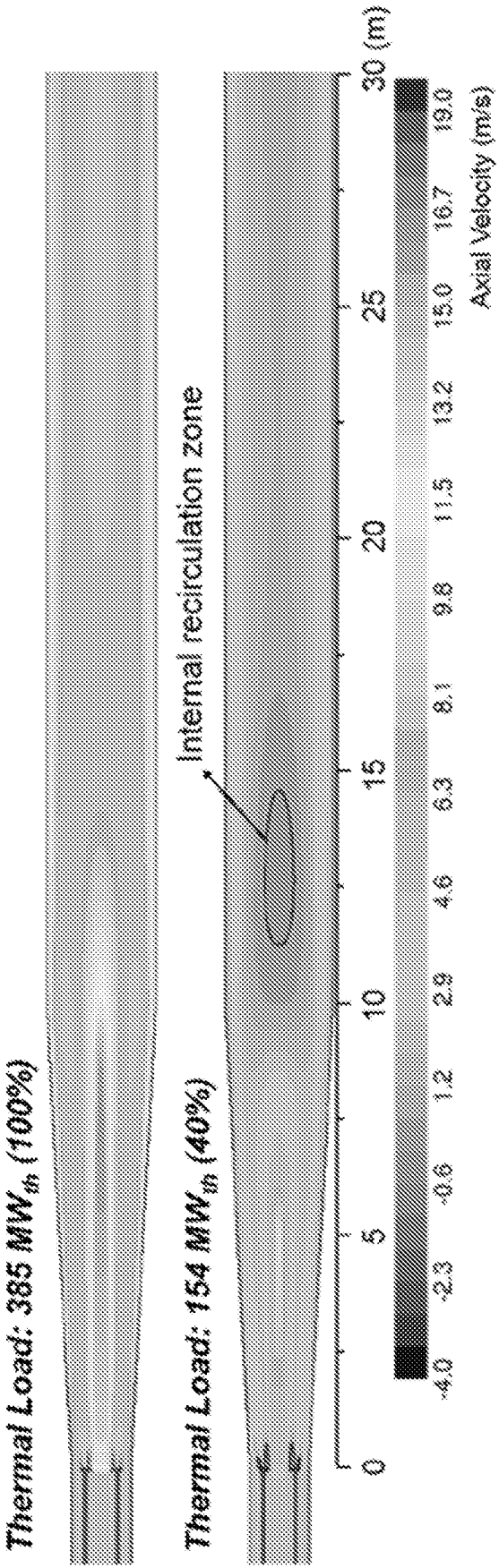


FIG. 18

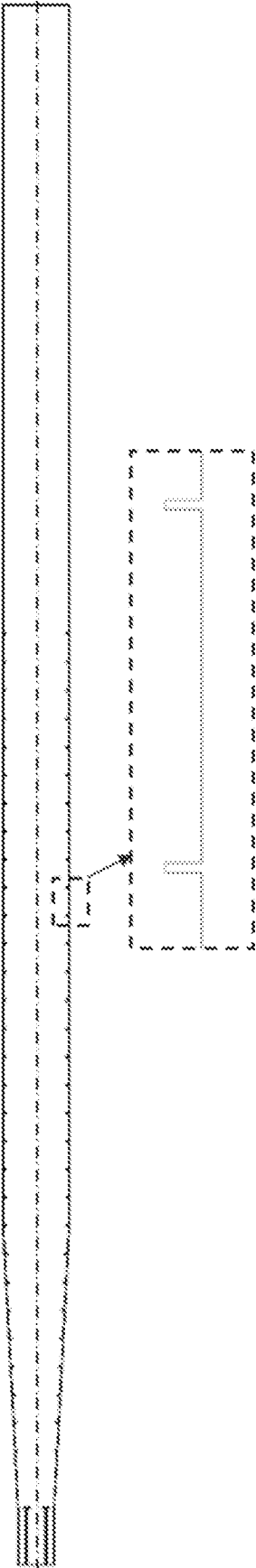


FIG. 19

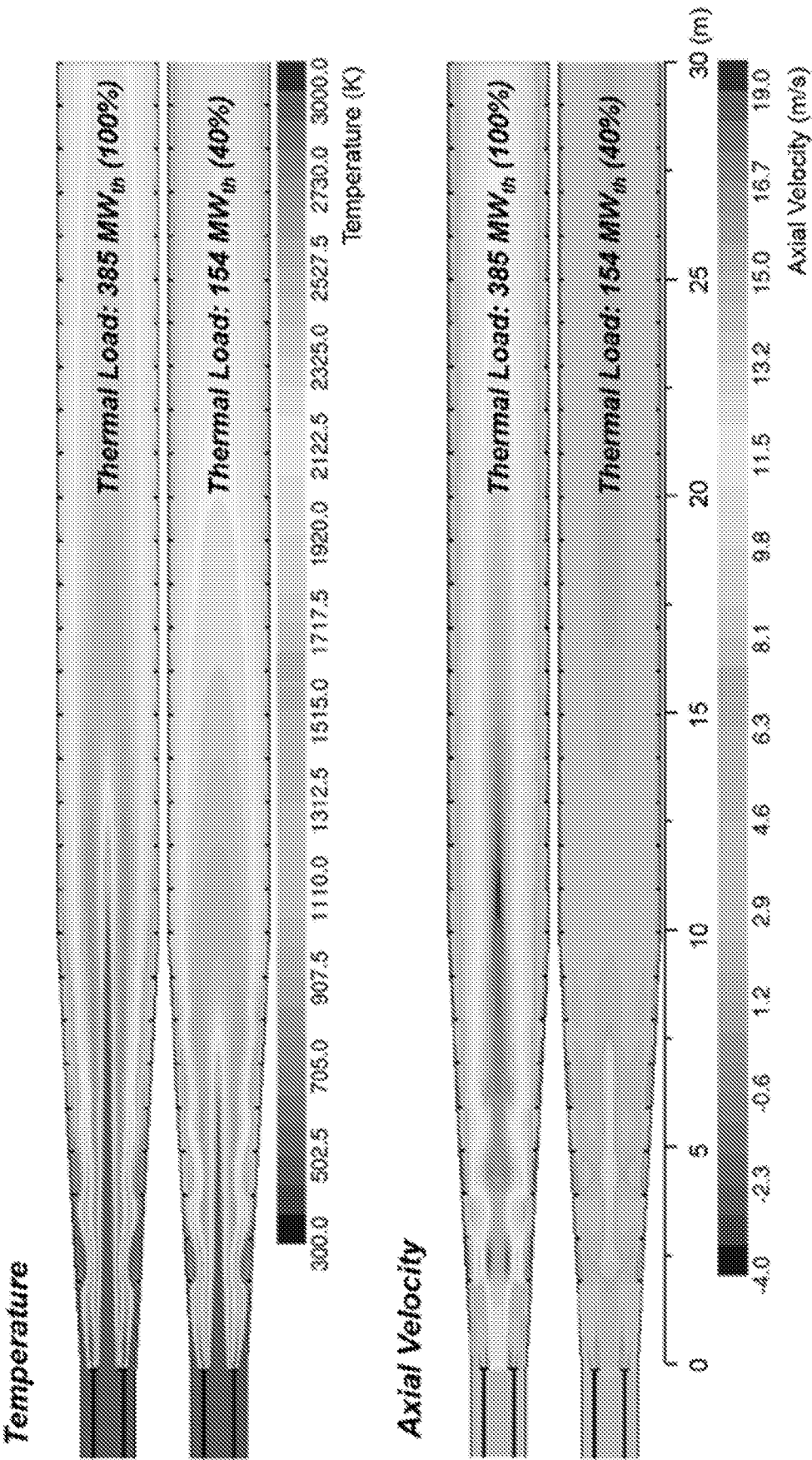


FIG. 20

FIG. 21A

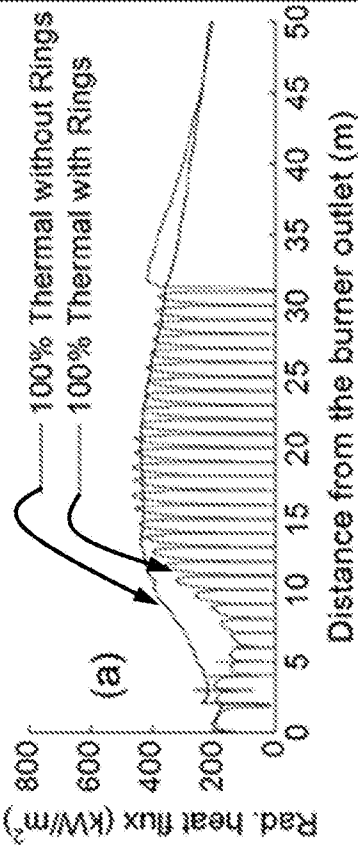


FIG. 21B

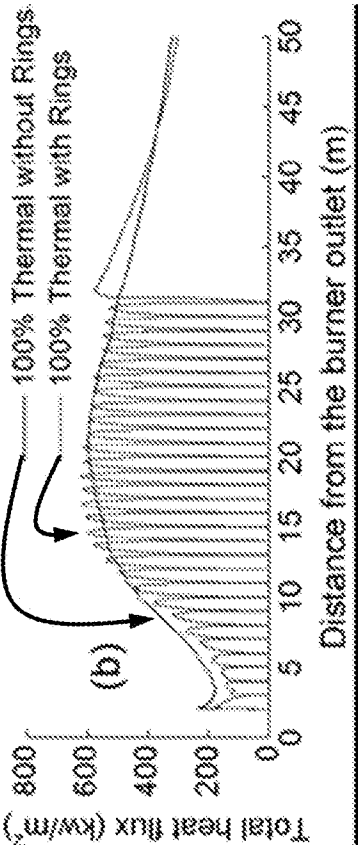


FIG. 21C

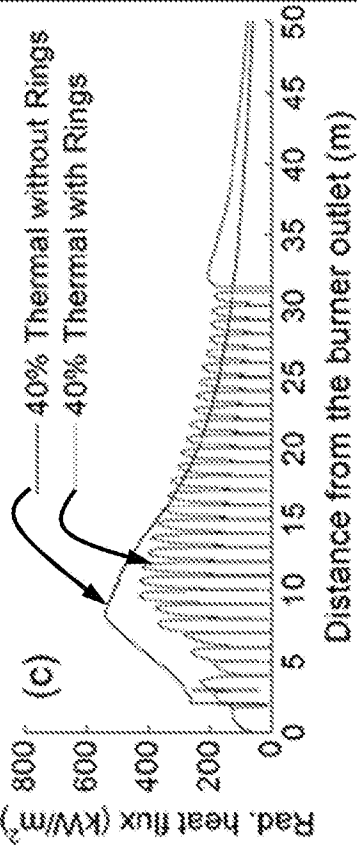


FIG. 21D

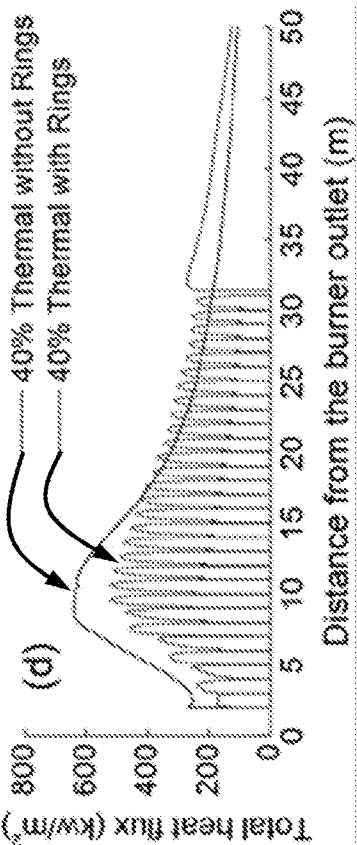


FIG. 22A

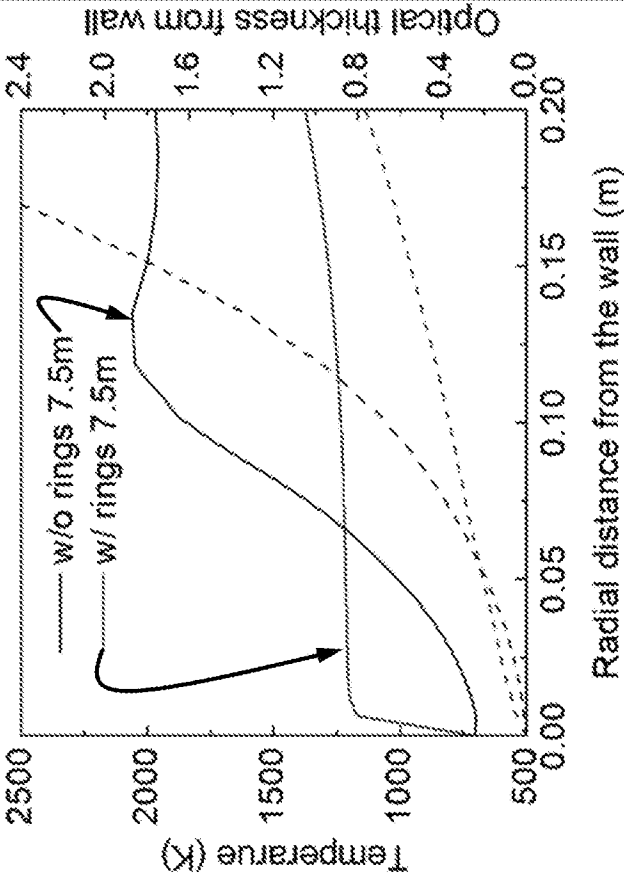
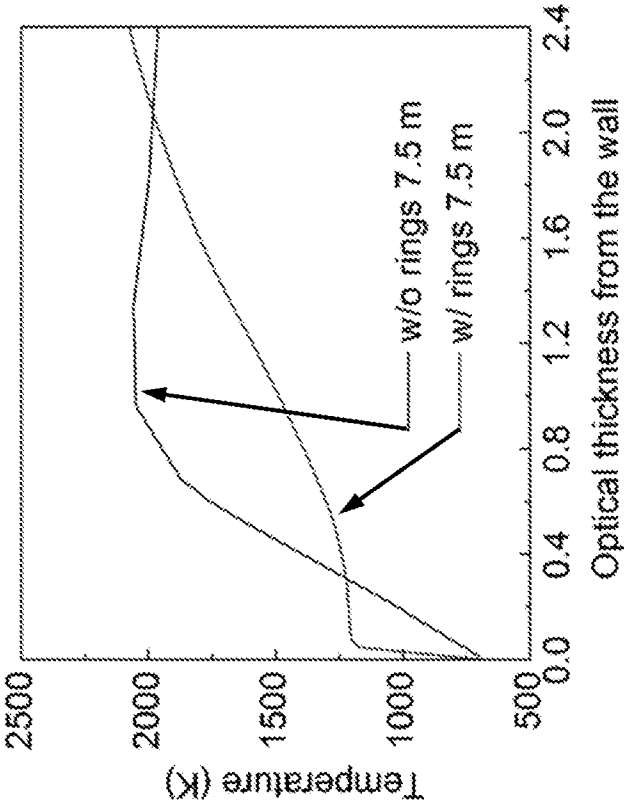


FIG. 22B



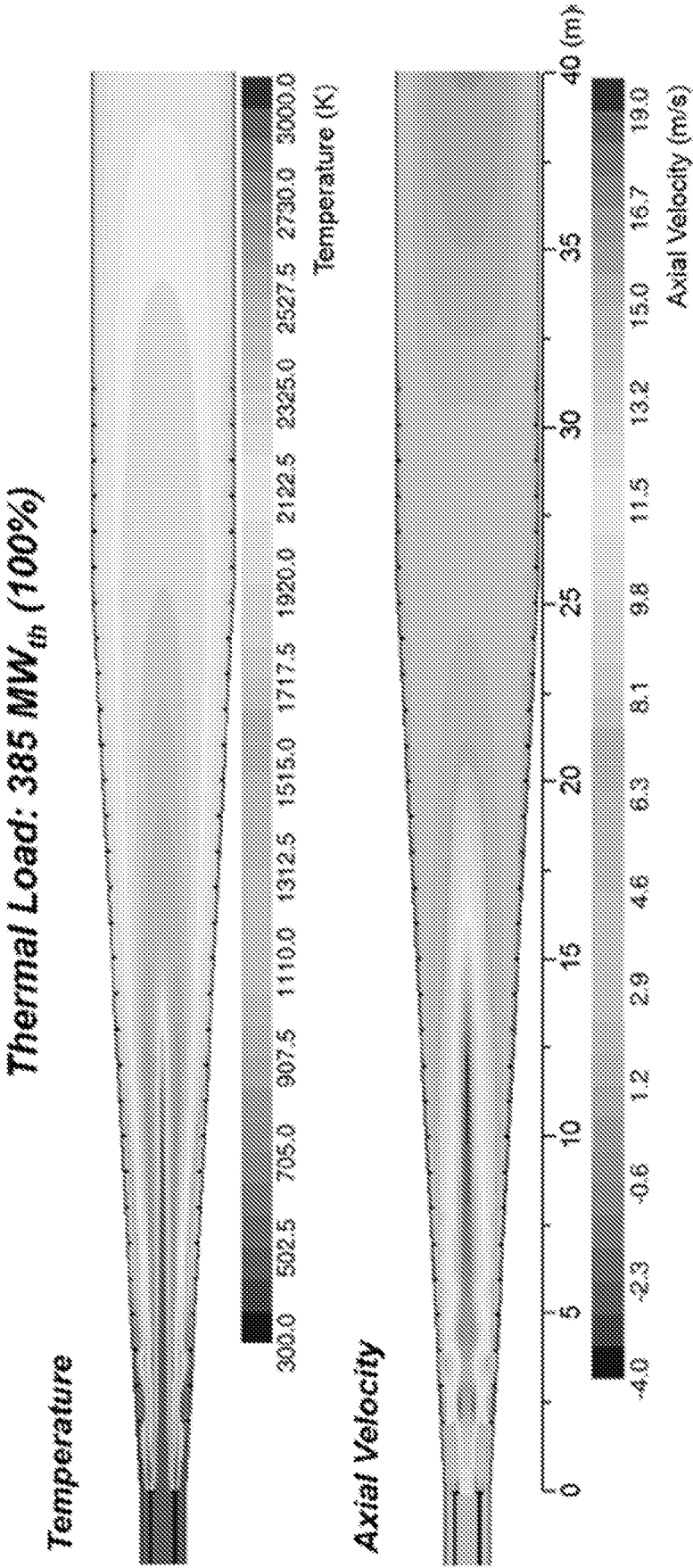
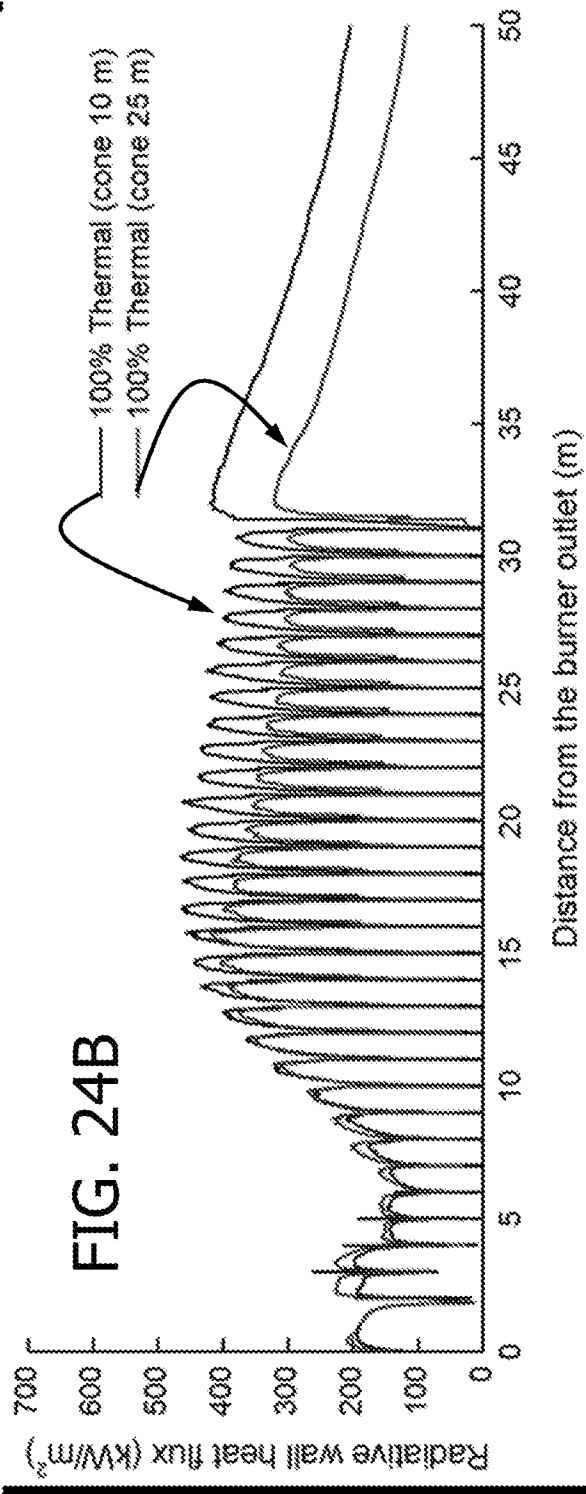
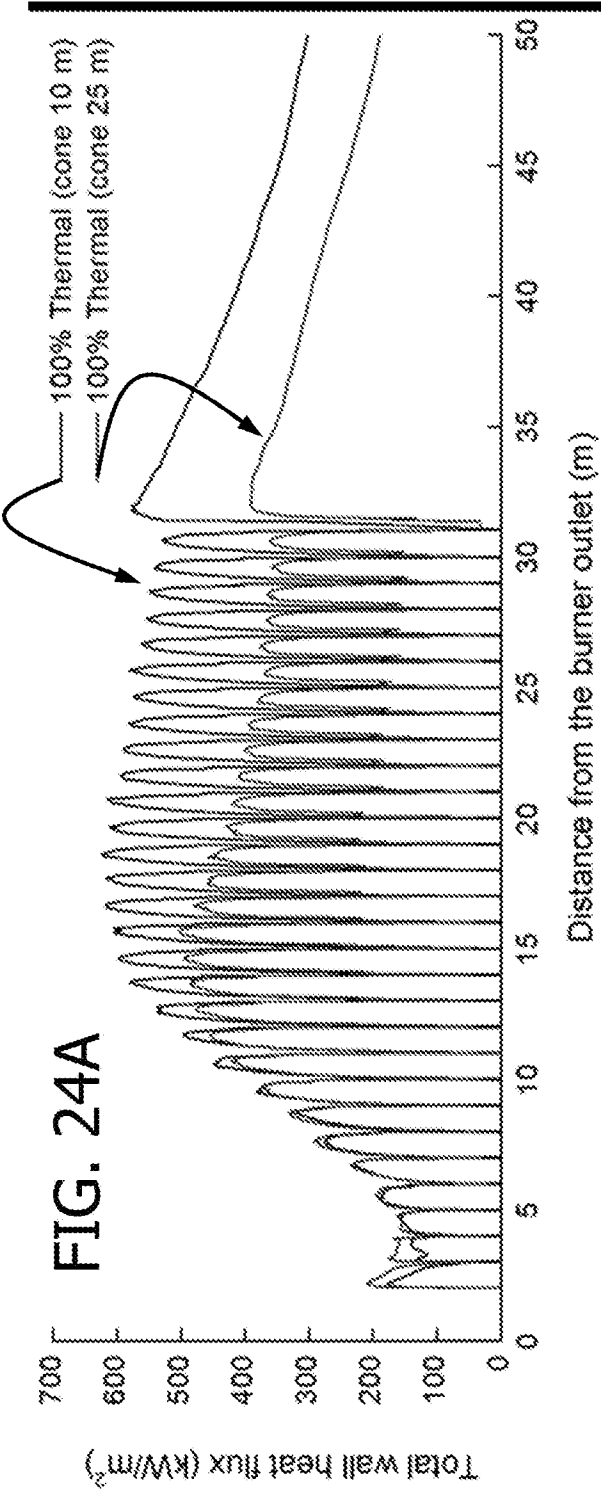


FIG. 23



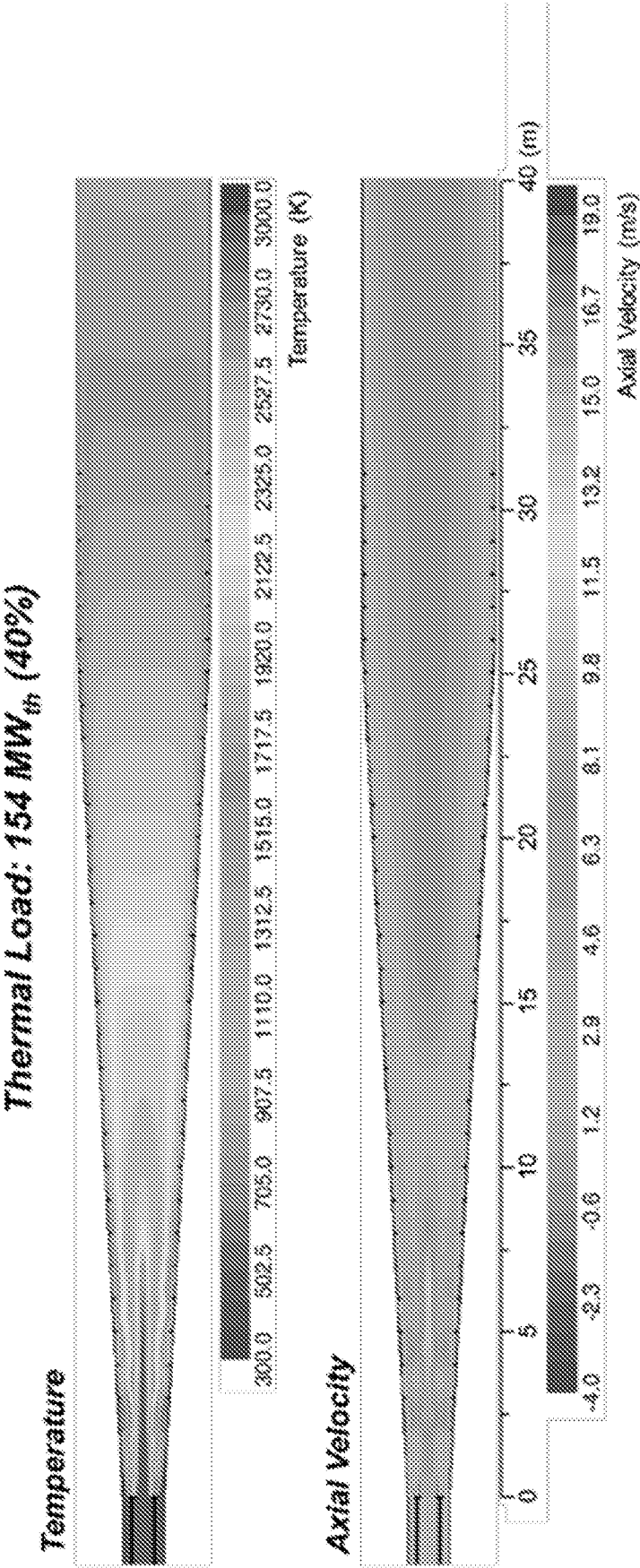
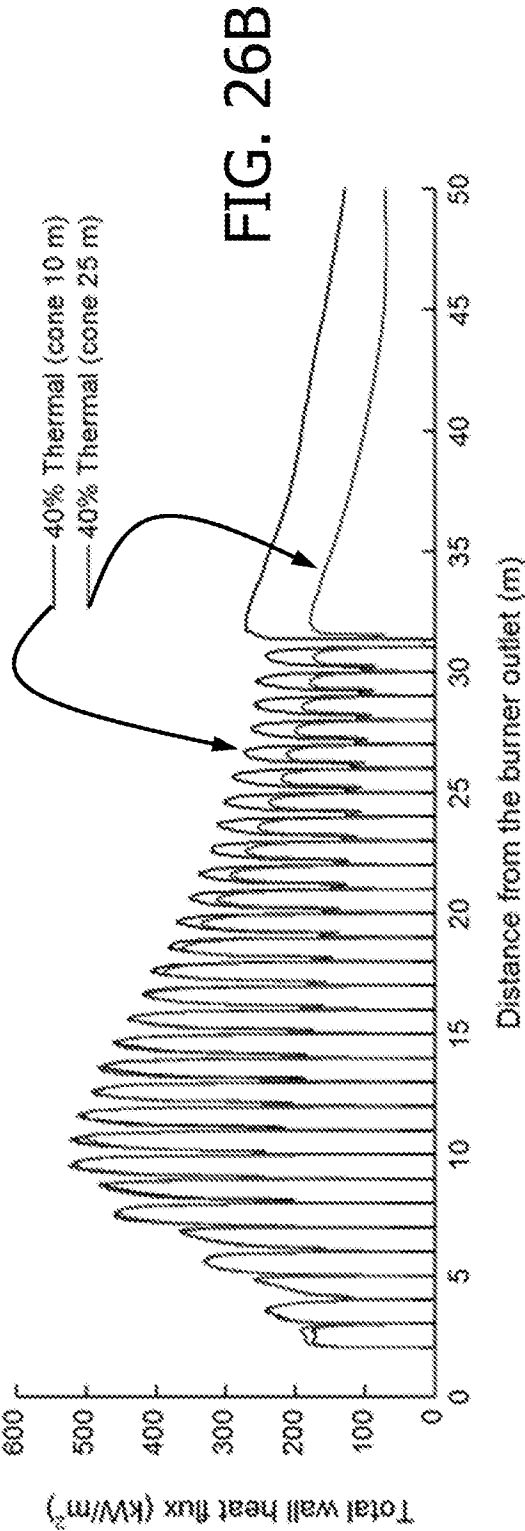
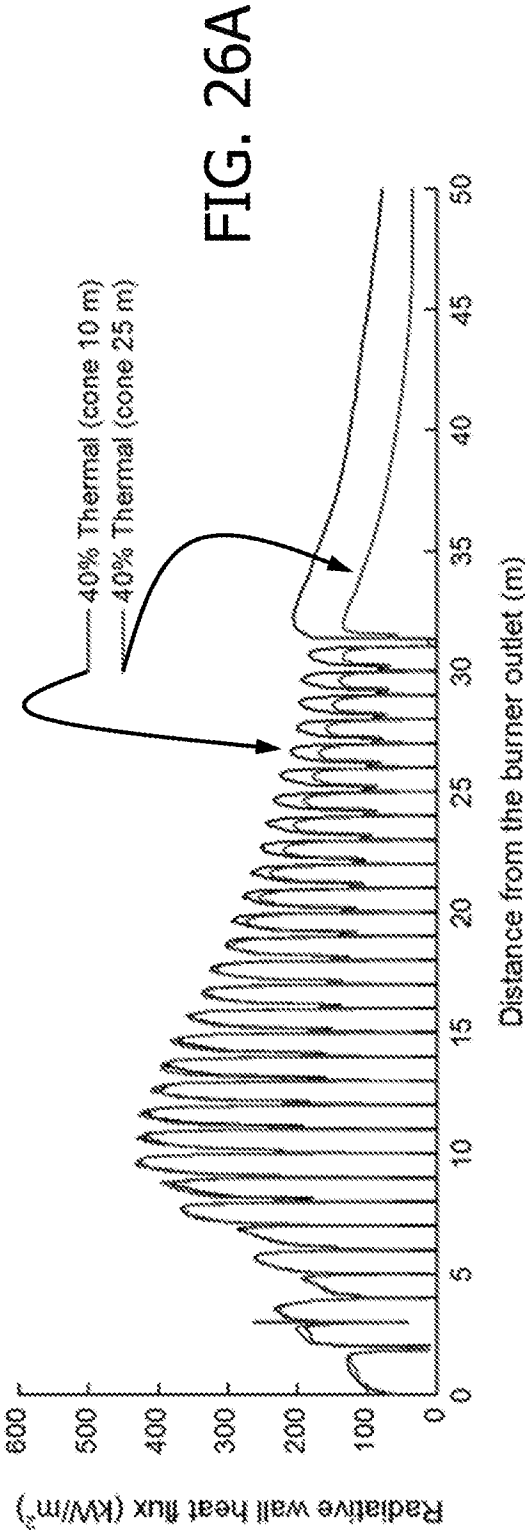


FIG. 25



INTERNATIONAL SEARCH REPORT

International application No.

PCT/US 2018/019942

Box No. II Observations where certain claims were found unsearchable (Continuation of item 2 of first sheet)

This international search report has not been established in respect of certain claims under Article 17(2)(a) for the following reasons:

1. ☐ Claims Nos.:
because they relate to subject matter not required to be searched by this Authority, namely:
2. ☐ Claims Nos.:
because they relate to parts of the international application that do not comply with the prescribed requirements to such an extent that no meaningful international search can be carried out, specifically:
3. ☒ Claims Nos.: 33-34
because they are dependent claims and are not drafted in accordance with the second and third sentences of Rule 6.4(a).

Box No. III Observations where unity of invention is lacking (Continuation of item 3 of first sheet)

This International Searching Authority found multiple inventions in this international application, as follows:

1. ☐ As all required additional search fees were timely paid by the applicant, this international search report covers all searchable claims.
2. ☐ As all searchable claims could be searched without effort justifying additional fees, this Authority did not invite payment of additional fees.
3. ☐ As only some of the required additional search fees were timely paid by the applicant, this international search report covers only those claims for which fees were paid, specifically claims Nos.:
4. ☐ No required additional search fees were timely paid by the applicant. Consequently, this international search report is restricted to the invention first mentioned in the claims; it is covered by claims Nos.:

Remark on Protest

- ☐ The additional search fees were accompanied by the applicant's protest and, where applicable, the payment of a protest fee.
- ☐ The additional search fees were accompanied by the applicant's protest but the applicable protest fee was not paid within the time limit specified in the invitation.
- ☐ No protest accompanied the payment of additional search fees.

INTERNATIONAL SEARCH REPORT

International application No.

PCT/US 2018/019942

A. CLASSIFICATION OF SUBJECT MATTER <i>F23C 1/00 (2006)</i> <i>F23L 7/00 (2006)</i> According to International Patent Classification (IPC) or to both national classification and IPC		
B. FIELDS SEARCHED Minimum documentation searched (classification system followed by classification symbols) F23K 3/00, F23M3/04, F23D1/00, C21C5/48, F23C 1/00, F23L 7/00 Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched Electronic data base consulted during the international search (name of data base and, where practicable, search terms used) PatSearch (RUPTO internal), USPTO, PAJ, Esp@cenet, DWPI, EAPATIS, PATENTSCOPE		
C. DOCUMENTS CONSIDERED TO BE RELEVANT		
Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	US 2006/0160040 A1 (L'AIR LIQUIDE SOCIETE ANONYME A DIRECTOIRE ET CONSEIL DE SURVEILLANCE POUR L'ETUDE ET L'EXPLOITATION DES PROCEDES GEORGES CLAUDE) 20.07.2006, [0002]-[0003], [0007], [0028]-[0029], [0039]-[0042], fig. 1-2	1-6, 16-32
Y A	WO2008/050289 A2 (FOSTER WHEELER ENERGY CORP) 02.05.2008, fig.1	1-13, 16-32 14-15
Y	SU 501073 A (VSESOUZNIJ NAUCHNO-ISSLEDOVATELSKY INSTITUT METALLURGICHESKOY TEPLOTEKHNI) 30.01.1976, fig. 1-2	2-13, 17-21, 23-27
Y	GB 2129918 A (ALUMINIUM PECHINEY) 23.05.1984, p.1 col. 2, lines 93-102, fig. 3	16-32
☒ Further documents are listed in the continuation of Box C. ☐ See patent family annex.		
*	Special categories of cited documents:	"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention "X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone "Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art "&" document member of the same patent family
"A"	document defining the general state of the art which is not considered to be of particular relevance	
"E"	earlier document but published on or after the international filing date	
"L"	document which may throw doubts on priority claim(s) or which is cited to establish the publication date of another citation or other special reason (as specified)	
"O"	document referring to an oral disclosure, use, exhibition or other means	
"P"	document published prior to the international filing date but later than the priority date claimed	
Date of the actual completion of the international search 31 May 2018 (31.05.2018)		Date of mailing of the international search report 14 June 2018 (14.06.2018)
Name and mailing address of the ISA/RU: Federal Institute of Industrial Property, Berezhkovskaya nab., 30-1, Moscow, G-59, GSP-3, Russia, 125993 Facsimile No: (8-495) 531-63-18, (8-499) 243-33-37		Authorized officer V. Krasina Telephone No. 8 499 240 25 91

INTERNATIONAL SEARCH REPORT

International application No.

PCT/US 2018/019942

C (Continuation). DOCUMENTS CONSIDERED TO BE RELEVANT

Category*	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
Y	SU 672216 A (CHAJKIN B.S. et al.) 10.07.1979, (col.2-4, fig.1)	31-32
A	SU 1561831 A (ALUMINIUM PECHINEY) 30.04.1990	1-32