

[54] APPARATUS FOR ADMIXING FUEL AND COMBUSTION AIR

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[58] Field of Search 431/9, 183, 184, 353; 239/402.5, 405, 406

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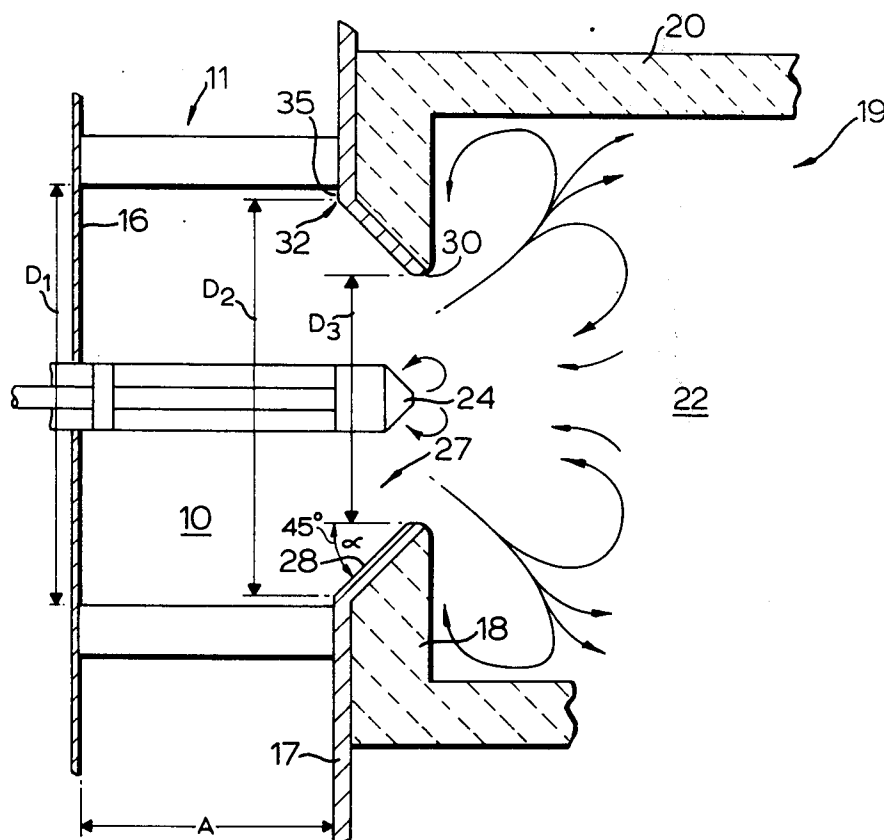
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[57] ABSTRACT

This invention provides apparatus for admixing combustion air, particularly air supplied at low pressure, with fluid or finely divided solid fuel for combustion, preferably in substantially stoichiometric proportions, in which air is passed from a windbox spirally inwardly between multiple overlapping vanes into a vortex chamber, thence through a coaxially aligned frusto-conical nozzle into a coaxially aligned cylindrical combustion chamber with the fuel being injected into the air passing through the nozzle, the critical dimensions of the air passage being related so that the flow of the air is constrained by its passage between the vanes, through the vortex chamber, through the nozzle and into the combustion chamber to generate therein a tangential velocity (swirl) which is sufficiently strong to establish a critical recirculating centre core of combustion gases in the combustion chamber and ensure complete combustion of the fuel.

4 Claims, 3 Drawing Figures



APPARATUS FOR ADMIXING FUEL AND COMBUSTION AIR

This application is a continuation-in-part of application Ser. No. 444,719 filed February 22, 1974, now Pat. No. 3,852,020, which in turn is a continuation-in-part of application Serial No. 264,635 filed June 20, 1972 and now abandoned.

This invention relates generally to admixing of air and fuel, particularly in oil burners and powdered coal burners in which combustion air is introduced into the combustion chamber in a swirling or vortex motion in the centre of which is coaxially located a fuel injector, e.g. an atomizer for liquid fuel or a feed tube for powdered coal.

An object of this invention is to provide an apparatus for admixing air and fuel in a burner including a combustion chamber configuration in which the design of the vortex chamber is such that the burner requires a relatively low air pressure differential, while providing the advantages of a high turndown ratio and a low excess air requirement.

A preferred embodiment of apparatus of this invention is shown on the accompanying drawings, in which like numerals denote like parts throughout the several views, and in which:

FIG. 1 is an axial sectional view through an oil burner and combustion chamber;

FIG. 2 is an enlarged axial sectional view of a portion of the combination shown in FIG. 1; and

FIG. 3 is a cross-sectional view of the inlet blades utilized in the preferred embodiment.

FIG. 1 shows a vortex chamber 10 which is defined by a plurality of arcuate inlet vanes 11 the configuration of which is clearly seen in FIG. 3. In the embodiment shown, there are twelve identical blades equally spaced at 30° intervals around the exterior of the vortex chamber 10. In essence, there is no specific cylindrical surface defining the outer periphery of the vortex chamber 10. Rather, the vortex chamber 10 can be considered to be defined by the inner edges 13 of the vanes 11 (see FIG. 2). A windbox 14 is defined between an outer wall 16 and an inner wall 17. A combustor 19 includes a circumferential wall 20, and an end wall 18. The walls 18 and 20 define a combustion chamber 22. As in conventional vortex burners, a suitable fan is utilized to provide air pressure within the windbox 14, and this pressure causes air to move inwardly between the vanes 11 and into the vortex chamber 10, where the air spirals inwardly. A fuel injector 24 is supported on the end of a tube 26 which contains fuel lines, for example in accordance with U.S. Pat. No. 3,510,061, issued May 5, 1970 to C. F. Peczel et al., and entitled "Two-Stage Sonic Atomizing Device".

The right-hand or discharge end of the vortex chamber 10, as seen in FIG. 2, comprises a nozzle 27 which is defined in part by a substantially 45° conical frustum 28. The upstream extremity 32 of the conical frustum 28 is in the same plane as the right-hand edges of the vanes 11, and thus also the inner wall 17, but defines a circle generally of slightly less diameter than the hypothetical circle which touches the inner edges 13 of all of the vanes 11. In FIG. 2, the diameter of the upstream end of the conical frustum 28 marked as D_2 , whereas the hypothetical circle touching the inner edges of the vanes 11 is indicated as having diameter D_1 . The difference between D_1 and D_2 represents an annular wall 35

which of course extends further to become the inner wall 17 of the windbox 14. The minimum diameter of the conical frustum 28, i.e. the diameter of the downstream or rightward end of the conical frustum 28, is marked as D_3 on the accompanying Figures.

Essentially, this invention resides in the disclosure of an apparatus for admixing air and fuel in a burner utilizing the vortex principle (swirl burner) and having particular optimum relative values for the different critical dimensions marked on the drawings. The invention thus consists of an apparatus for admixing combustion air and fluid or finely divided solid fuel in substantially stoichiometric proportions for combustion preferably at normal heat generation rates of 4×10^6 Btu/hr. and up, said apparatus comprising:

1. a windbox for maintaining a supply of combustion air under pressure, said windbox having an outer wall and a parallel inner wall contiguous to
2. a cylindrical combustion chamber,
3. a fuel injector,

4. a plurality of overlapping vanes spaced around a hypothetical cylindrical surface coaxial with said fuel injector and extending between said outer and inner walls and perpendicular thereto to form a vortex chamber, the diameter of said surface in inches being designated as a dimension D_1 and the length thereof, defined by the distance between the said parallel outer and inner walls, being designated as a dimension A in inches which is substantially $0.64 D_1$, the total minimum inlet area I between said vanes in square inches being substantially $(0.43 D_1^2 - 7.63)$, said fuel injector extending through said outer wall and hypothetical cylinder with its discharge end terminating in

5. a substantially 45° frusto-conical nozzle extending through said inner wall between the vortex chamber and the combustion chamber coaxially with said fuel injector, the largest circular diameter of said nozzle being adjacent the vortex chamber and having a diameter D_2 in inches which is substantially $0.95 D_1$, the smallest circular diameter of the nozzle having a dimension D_3 of substantially $(0.34 D_1^2 - 6)^{1/2}$ inches, and
6. the forward end wall of a combustor, adjacent said nozzle and normal to the axis thereof, with a contiguous cylindrical side wall forming said cylindrical combustion chamber coaxial with said nozzle.

Although it has long been known to use a windbox, overlapping vanes, a vortex chamber, a fuel injector, a conical combustion air nozzle, and/or a cylindrical combustion chamber in various combinations in fuel burners, it has not previously been disclosed how dimensional criteria of the combustion air passage through the burner can be proportioned to achieve complete combustion of a fuel, with substantially stoichiometric proportions of air and fuel, at a relatively low air pressure differential and over a wide range of loads (e.g. 10% to 175% of rated load). In order to explain why the dimensional criteria are of such critical importance, a brief digression into theory is necessary.

One of the primary requirements for good combustion of a liquid fuel in a combustion chamber of the type illustrated in the accompanying drawings is that some of the heat from the downstream end of the combustion zone be transferred back to the upstream end. This process is an essential aid to evaporation of liquid fuels and, as all liquid fuel must be vaporized before it can be burned, this transfer of heat becomes of critical importance. The process is also an even more essential

aid to the combustion of powdered solid fuels, as the recirculation of heat is necessary to ignite and burn the solid fuel. Although gaseous fuel does not require vaporization before its combustion, the present invention is fully applicable also for complete combustion of such fuel at optimum efficiency.

There are essentially three processes which can take place in a combustion chamber and which provide a heat transfer from the downstream end of a combustion zone to the upstream end. One of these processes is the establishment of a recirculating centre core, as illustrated by the centre arrows in FIG. 2. When the air-fuel mixture is ejected from the vortex chamber 10 into the combustion chamber 22 with a relatively high tangential velocity (i.e. a relatively high swirl), then the centrifugal force in the swirling mixture creates a low pressure core in the centre. At the downstream end of the swirling mass the centrifugal force is appreciably less, hence some of the hot gases are drawn into the centre core and are carried in a continuous stream toward the upstream end of the combustion zone.

In addition to the recirculating centre core, a recirculating eddy pattern is established to increase heat transfer. The eddy currents take place in the upstream corner area of the combustion chamber 22, and in FIG. 2 the back-curving arrows in the corner area show the direction of eddy flow. Essentially, a swirling mixture discharging conically into the combustion chamber creates a low pressure area in the corner zone, and this low pressure draws in hot gases from its downstream side, creating the eddy currents.

Although the latter mechanism can be established without swirl, it is more effective with a swirl.

Another mechanism by which heat from the combustion area is used to help vaporize the liquid fuel has to do with radiation. If the hot gases heat up the refractory walls of the combustion chamber, part of the heat will be radiated back to and absorbed by the finely-divided fuel particles in the central zone.

While the principles described above are simple and generally known, the details of the various processes are very complex. Because of this complexity, it has hitherto been difficult to produce a high performance burner capable of optimizing the different characteristics that must be taken into consideration.

Through a process of repeated testing while altering incrementally first one dimensional parameter and then another, we have arrived at a dimensional formulation for a finely divided solids and fluid fuel burner combination, which formulation is capable of admixing combustion air and fuel to provide a high intensity burner with extremely low excess air requirement, and simultaneously a low static pressure differential requirement across the vanes.

Certain remarks in regard to the swirl itself are called for. A swirl strong enough to create an efficient recirculation in the combustion zone has a very pronounced tendency to extend the recirculating patterns to the discharge plane (the small diameter of the conical frustum 28), and possibly within the vortex chamber 10. Not only is a recirculation pattern inside the vortex chamber merely a waste of energy, but any recirculation across the discharge plane will carry unburnt and partially burned liquid or solid fuel particles back into the vortex chamber. In the relatively cool interior of the vortex chamber these particles will tend to form an ever-increasing burden of deposits.

Since the build-up of deposits cannot be tolerated, many burners avoid deposits by the use of a weak swirl only, which assists mixing but does not generate an effective recirculation pattern. Other burners employ a complex baffle arrangement to prevent back flow.

Using the dimensional formulations disclosed herein and a combustion air supply pressure which can vary from low values, e.g. below six inches of water column pressure, to the higher values commonly used with other swirl burners, e.g. 18 to 24 inches of water column pressure or more, the burner combination described herein is capable of generating a swirl sufficiently strong to assure virtually complete combustion of all common finely divided solid or fluid (i.e. liquid and gaseous) fuels, while using a single air supply only. At the same time, back flow and the back flow deposits are eliminated. Only a relatively low static pressure is required, although higher pressures can be used.

We now wish to discuss in greater detail the generation of the swirl pattern within the vortex chamber and the nozzle 27. To begin with, the rate of air flow is set by the requirements of the selected fuel rate and air:fuel ratio. The air is delivered into the windbox at a pressure which may be, for example, in the range of about 6 inches water column (0.0152 kg/cm²). Under normal conditions, the energy of the air in the windbox would be primarily in the form of pressure, and only fractionally in the form of velocity. The vanes 11 guide the air into the vortex chamber 10, and as the air passes between the vanes it is accelerated in the tangential direction. At the same time, the radial component remains at a low value. Both the radial and the tangential velocity components increase as the air is forced inwardly from D₁ to D₂ between the parallel walls due to the pressure gradient. From D₂ to D₃ the axial dimension of the vortex chamber increases gradually due to the conical shape of the nozzle 27. As the air is forced inwardly from D₁ to D₂, its tangential velocity component continues to increase, but more importantly its velocity component in planes intersecting the axis changes smoothly from being purely radial to being partly axial-partly radial. In other words, as the air moves inwardly from D₂, it gradually begins to move toward the combustion chamber due to the fact that the vortex chamber increases in axial dimension from D₂ to D₃.

The flow pattern generated from D₁ to D₂ has a strong tendency to create back flow, because from D₁ to D₂ no velocity component parallel to the axis of the vortex chamber is created. Thus, most of the available energy goes into the creation of straight swirl which produces a high centrifugal force, in turn producing a low pressure at the central core and inducing a back flow recirculation pattern along the centre core. For this reason, the step from D₁ to D₂ is kept as small as manufacturing considerations will allow.

Since the swirl component added between D₁ and D₂ is minimized, it follows that the total swirl at D₃ (the entrance to the combustion chamber) is essentially the sum of that imparted by the vanes themselves, plus the increase from D₂ to D₃. Practical experience indicates that the tendency for centre core back flow is reduced if the swirl imparted by the vanes is small and the increase from D₂ to D₃ is relatively large.

Practical consideration, however, limits the extent of the increase from D₂ to D₃. It must firstly be appreciated that the value for D₃ is set by the requirement for an appropriate mean axial velocity component of the

air-fuel mixture as it enters the combustion chamber when operating at the rated load. Thus, the tangential velocity gained between D_2 and D_3 can be increased only by enlarging D_2 and thus increasing the diameter of the vortex chamber. However, as the vortex chamber becomes larger, its cost increases, and its external dimensions can reach a practical limit particularly where several burners are spaced closely together in multiple installations.

Measurement of the axial component of velocity along a radius from the center to the circumference of the nozzle at D_3 shows that the axial component has a maximum value close to the circumference and that it drops sharply towards the centre. The rate of this change increases with increasing values of the tangential component. This tangential component in turn is influenced by the size of D_3 , as mentioned above.

With the critical dimensions of the air passage between the vanes, through the vortex chamber, and through the nozzle related in accordance with the present specification, the air pressure differential required to maintain the necessary flow of air is low and the axial velocity component in the center of the nozzle does not fall below zero, hence backflow of material from the combustion chamber into the nozzle and vortex chamber is precluded; likewise the mass of combustion mixture ejected from the nozzle at D_3 has sufficient tangential velocity component to fill the combustion chamber and to generate the required recirculating patterns. When a burner utilizing this invention is operated at loads higher or lower than the normal rated load of the burner, the ratio of tangential to axial velocity components remains substantially constant. Higher loads require higher air pressure differential; this provides more energy for recirculation and mixing, hence more fuel can be burned in the same combustion space. Conversely, at lower loads the air pressure differential (and correspondingly the energy), are reduced, hence recirculation and mixing are less vigorous but, with reference to the combustion chamber, the mass flow rate therethrough is lower therefore the residence time is higher and hence the quality of the combustion is maintained despite less vigorous mixing. Any substantial alteration in the interrelationships between these critical dimensions set out hereinafter will adversely affect either the low air pressure differential requirement or the quality of combustion or both.

To illustrate the criticality of the relationship disclosed in this invention, it can be pointed out that reduction of the dimension D_3 alone in any specific embodiment would increase the axial velocity component near the periphery at D_3 , not only because of the reduced cross-sectional area having to pass the same mass of air but also because such reduction of D_3 would increase the tangential velocity component, thereby increasing the axial velocity differential between periphery and center at D_3 . Consequently there would be a sharp rise in the air pressure differential required to maintain the flow of the necessary mass of air. In spite of the increased energy that would thus be supplied, the quality of combustion would deteriorate as the high velocities would hinder the formation of the proper recirculation patterns in the combustion chamber.

Likewise, an increase of the dimension D_3 alone in any specific embodiment of the invention would cut the energy (pressure differential) requirement for maintaining necessary air flow, but the slower moving air would not form strong enough recirculation currents in

the combustion chamber and the desired combustion would not be achieved. Also, the gain in the tangential velocity component between D_2 and D_3 would be reduced, and with the reduction of the mean axial velocity component at D_3 combined with the altered recirculation currents in the combustion chamber, there would be increased likelihood of backflow through the plane of the nozzle at D_3 .

The dimensional relationships set out hereinafter represent a practical compromise for the foregoing problem of achieving the desired admixture of air and fuel with relatively low air pressure differential, most commonly for complete combustion with little or no excess air. Actual experience indicates that there are no undesirable deposits and no centre core back flow through the conical nozzle with the dimensional formulations set out below, and further that the diameter of the vortex chamber remains smaller than that of the combustion chamber, and hence the spacing of the burners in a multiple installation is not affected.

Because burners are of many different sizes due to the many different heat outputs required, it has been found convenient to correlate the various dimensions against the "rating". The rating is defined as the gross heat input with essentially stoichiometric mixtures and with air supplied arbitrarily under pressure of 6 inches water column (0.0152 kg/cm²) and 80°F (27°C) temperature.

It will be understood that the burner is perfectly capable of loads higher or lower than the rating defined above, provided that the other conditions are appropriately changed, e.g. the air supply pressures varied both above and below 6 inches water column. Further, the burner is fully capable of providing good combustion with fuel-rich mixtures (to give highly reducing conditions in the combustion products) or lean mixtures containing considerably excess air and giving strongly oxidizing conditions in the combustion products.

The burner to which this application is directed is primarily intended for industrial applications, and thus the design formulae given below relate more particularly to burners with rated loads of 4×10^6 Btu/hr (1×10^6 kilogram calorie/hr.) and up, but smaller loads also can be accommodated.

Firstly, the angle of the nozzle 27 defines the direction of the flow in axial planes within the combustion chamber immediately adjacent the nozzle. It has been found that a change of 5° or more from the preferred angle of 45° in either direction will markedly change the amount of back flow along the centre core.

The following parameters are used in defining the principal dimensions of the apparatus:

Parameter	Definition	Units (f.p.s.)	Units (c.g.s.)
R	=Rating	10 ⁶ Btu/hr.	10 ⁶ kilogram cal/hr.
D_1	=Hypothetical inside diameter of vanes	inches	millimeters
D_2	=Large diameter of conical frustum	inches	millimeters
D_3	=Small diameter of conical frustum	inches	millimeters
α	=Angle of conical frustum	degrees	degrees
A	=Axial Separation between walls 16 and 17 (see FIG. 2)	inches	millimeters

-continued

Para- meter	Definition	Units (f.p.s.)	Units (c.g.s.)
n	=Number of vanes		
h	=Minimum separation between two adjacent vanes (see FIG. 3)	inches	millimeters
I	=Inlet area = n × h × A	inches ²	cm ²

In view of the foregoing considerations, the dimensional relationships defining a burner claimed in this application in terms of rating using air supplied at 6 inches of water column pressure and at 80°F are as follows, in both the f.p.s. and c.g.s. systems:

f.p.s. System	c.g.s. System
I=7.63 R	I=196R
D ₂ =4.03 (R+1) ^{1/2}	D ₂ =204 (R+0.25) ^{1/2}
D ₁ =1.05 D ₂ =4.23 (R+1) ^{1/2}	D ₁ =1.05 D ₂
D ₃ =2.45 R	D ₃ =124 R ^{1/2}
A=2.69 (R+1) ^{1/2}	A=136 (R+0.25) ^{1/2}
α=45°	α=45°

It has been found that any substantial deviation from these values, e.g. even plus or minus 10%, seriously affects the performance of the burner.

From the foregoing relationships which correlate burner rating and the corresponding critical combustion air passage parameters of a burner, simple algebraic substitution permits expression of these latter parameters in terms of one another; hence corresponding values for A, I, D₂ and D₃ can be expressed in terms of D₁, independently of rating, by substitution of the value of D₁ in the formulae for the corresponding value of R for any burner. Thus for a burner in accordance with this invention, the foregoing parameters are related by formulae (independently of the air supply pressure), for example in the f.p.s. system:

$$A=0.64 D_1;$$

$$D_2=0.95 D_1;$$

$$I=0.43 D_1^2-7.63;$$

$$D_3=(0.34 D_1^2-6)^{1/2};$$

Hence by application of these formulae there is defined an assembly of the critical elements of a burner which form the passageway for combustion air through the burner to the zone of combustion of fuel, and the assembly of the elements so defined inherently delivers combustion air, supplied at any pressure, to the combustion zone in a swirling motion that creates an efficient recirculation pattern in the combustion zone to achieve substantially complete combustion of a stoichiometric equivalent flow of fuel without recirculating any combustion air or fuel back into the air passageway leading to the combustion zone.

From the foregoing relationships, it also follows that at D₁ the ratio of the tangential and radial velocity components is

$$4.69 \frac{R+I}{R}$$

in the f.p.s. system and

$$4.69 \frac{R+0.25}{R}$$

in the c.g.s. system.

- Also, but only under rated conditions (stoichiometric air supply at 6 inches water column pressure and 80°F): Tangential velocity at D₁ is 3140 feet per minute (f.p.s.) or 16m/sec (c.g.s.), equivalent to 0.62 inches of water column (f.p.s.) of 0.0016 kg/cm² (c.g.s.). Radial velocity in the same location = 660 feet per minute (f.p.s.) or 3.35 meters per second (c.g.s.) and thus velocity pressure is negligible. Nominal axial velocity at D₃ is 5080 feet per minute (1.63 inches water column) or 25.8 meters per second (0.00413 kg/cm²). In practice, most of the air is discharged through the nozzle in the plane of its least dimension with an axial velocity of approximately 6750 feet per minute (2.88 inches water column) or 34.3 meters per second (0.0073 kg/cm²).

- The effect of changing the principal dimensions to values outside of the relationships given above is illustrated in the following comparison of the performance of two burners, one constructed with a preferred configuration (i.e. not falling within the relationships defined), both burning the same grade of fuel oil (Bunker 6C). Dimensions not given below, including combustion chamber dimensions, are identical in the two burners.

f.p.s. System	R	I	D ₁	D ₂	D ₃	Air Pressure
Inferior	48	328	28.5	27	19	6 inches water column
Preferred	60	458	33	31.5	19	6 inches water column
Preferred, run at same load as Inferior	48	458	33	31.5	19	3.8 inches water column
Inferior	12	2120	725	686	483	0.0152 kg/cm ²
Preferred	15	2955	840	800	483	0.0152 kg/cm ²
Preferred, run at same load as Inferior	12	2955	840	800	483	0.0096 kg/cm ²

- The burner with the inferior configuration (the one not falling within the relationships) gained more swirl between the vanes and less between D₂ and D₃. As a consequence, its rated load was only 80% of the burner whose dimensions fall within the relationships. Further, the preferred burner performed well even at the highest load (96 × 10⁶ Btu/hr or 24 × 10⁶ kilogram-calorie/hr) permitted by the supporting equipment (windbox air pressure of 15 inches water column), while the inferior configuration showed a noticeable deterioration at loads above the rated load. It will be noted in the table that the burner with the preferred configuration requires less windbox air pressure than does the burner with the inferior construction at the same load.

It is believed that the performance of the vanes, within reason, does not depend upon the number of the

vanes. It can be said, however, that if the number is less than 6, then the individual layers of the air entering will be too thick (h will be too large), which leads to unnecessary turbulence. Furthermore, a minimum ratio of vane overlap to radial vane spacing at the inner edges is required to prevent "short-circuiting" of the air. This ratio is around 3:1.

The outside diameter of the set of vanes can be reduced by increasing the number of vanes, without decreasing D_1 , but manufacturing economy will set a limit here.

Preferably, the vanes are gently curved as shown in FIG. 3, and ideally the direction of the air as it enters the vortex chamber should correspond to the tangential/radial ratio defined above.

We claim:

1. An apparatus for admixing combustion air and fluid or finely divided solid fuel in substantially stoichiometric proportions for combustion, comprising:

1. a windbox for maintaining a supply of combustion air under pressure, said windbox having an outer wall and a parallel inner wall contiguous to
2. a cylindrical combustion chamber,
3. a fuel injector,
4. a plurality of overlapping vanes spaced around a hypothetical cylindrical surface coaxial with said fuel injector and extending between said outer and inner walls and perpendicular thereto to form a vortex chamber, the diameter of said surface in inches being designated as a dimension D_1 and the length thereof, defined by the distance between the said parallel outer and inner walls, being designated as a dimension A in inches which is substantially $0.64 D_1$, the total minimum inlet area I between said vanes in square inches being substan-

tially $(0.43 D_1^2 - 7.63)$, said fuel injector extending through said outer wall and hypothetical cylinder with its discharge end terminating in

5. a substantially 45° frusto-conical nozzle extending through said inner wall between the vortex chamber and the combustion chamber coaxially with said fuel injector, the largest circular diameter of said nozzle being adjacent the vortex chamber and having a diameter D_2 in inches which is substantially $0.95 D_1$, the smallest circular diameter of the nozzle having a dimension D_3 of substantially $(0.34 D_1^2 - 6)^{1/2}$ inches, and
6. the forward end wall of a combustor, adjacent said nozzle and normal to the axis thereof, with a contiguous cylindrical side wall forming said cylindrical combustion chamber coaxial with said nozzle.
2. An apparatus as claimed in claim 1 for a normal heat generation rate designated as its rating R in 10^6 Btu/hr. with air supplied to the windbox under a pressure of 6 inches of water column at 80°F in which the angle of the frusto-conical nozzle is 45° and the other critical dimensions of the apparatus are related to said rating R by the following formulae:

$$I = 7.63R$$

$$D_2 = 4.03 (R + 1)^{1/2}$$

$$D_1 = 4.23 (R + 1)^{1/2}$$

$$D_3 = 2.45 R^{1/2}$$

$$A = 2.69 (R + 1)^{1/2}.$$
3. An apparatus as claimed in claim 1 in which the fuel injector is an oil atomizer.
4. An apparatus as claimed in claim 1 in which there are at least six vanes and the ratio of the overlapping to the minimum distance between adjacent vanes is at least about 3:1.

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