United States Patent

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[54] **FLUID-DYNAMIC ENGINE**
9 Claims, 4 Drawing Figs.

[52] **U.S. Cl.** **60/59,** 60/39.02, 60/264
[51] **Int. Cl.** **F01k 3/18,** F02c 9/00, F02k 1/00
[50] **Field of Search** ..... 60/59, 59 T, 39, 39.02, 39.75, 39.14, 39.53, 264, 270

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ABSTRACT: This invention relates to a fluid-dynamic engine wherein a gas is accelerated through the engine at the speed of sound at the sonic speed of the gas and imparting energy to the gas while maintaining it at the sonic speed. The engine may comprise a duct having a sonic duct section interposed between convergent and divergent sections so that it is successively accelerated to the sonic speed through the convergent section at the sonic speed. The engine includes means for deriving power from the fluid stream by coupling a fluid-responsive element to the stream and recompressing the fluid of the fluid stream while moving at the sonic speed.

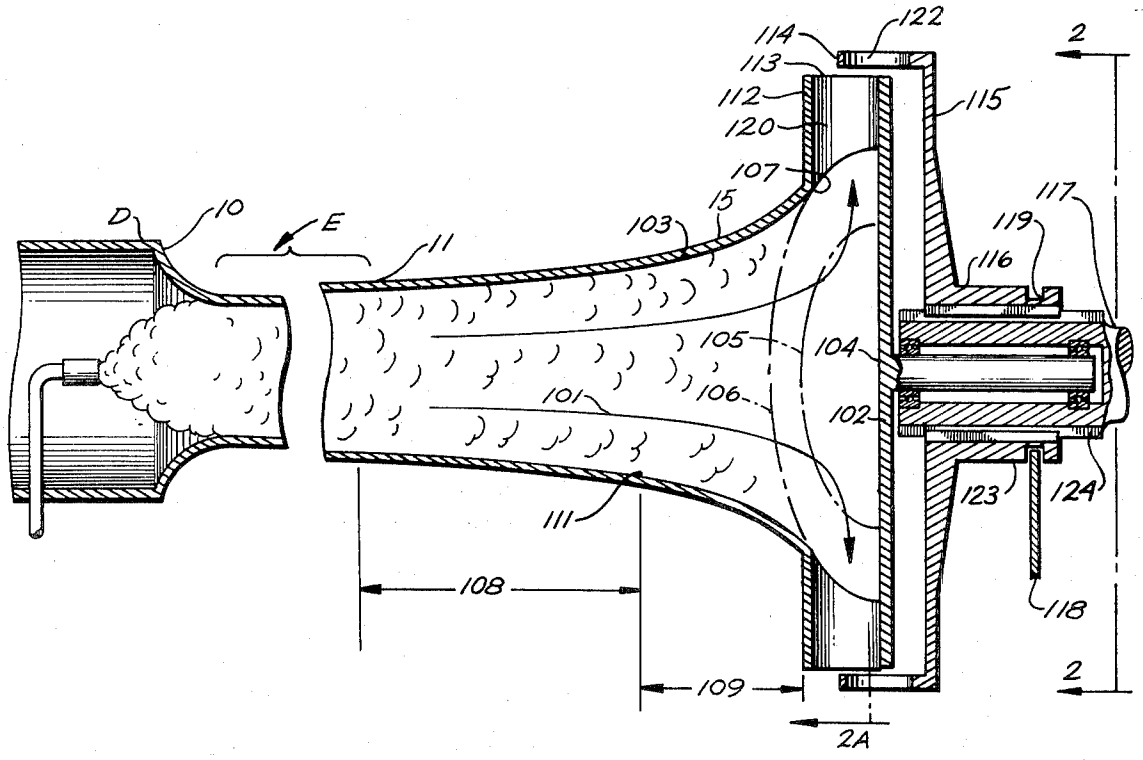


FIG. 1

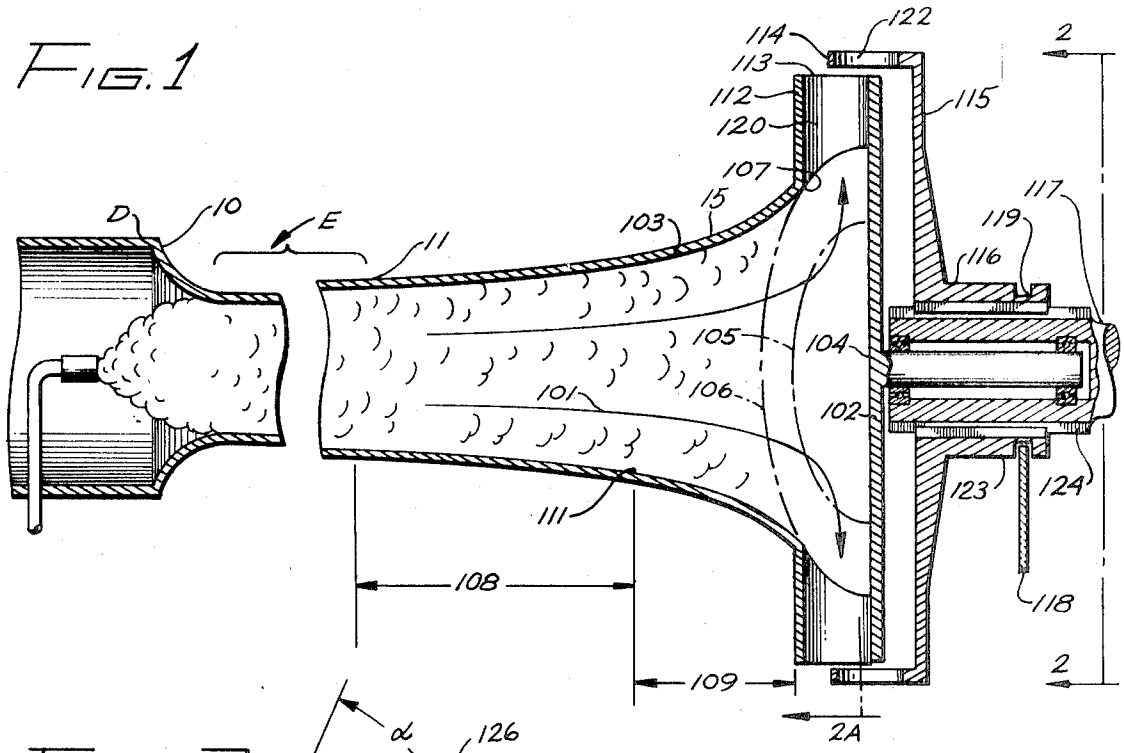


FIG. 2

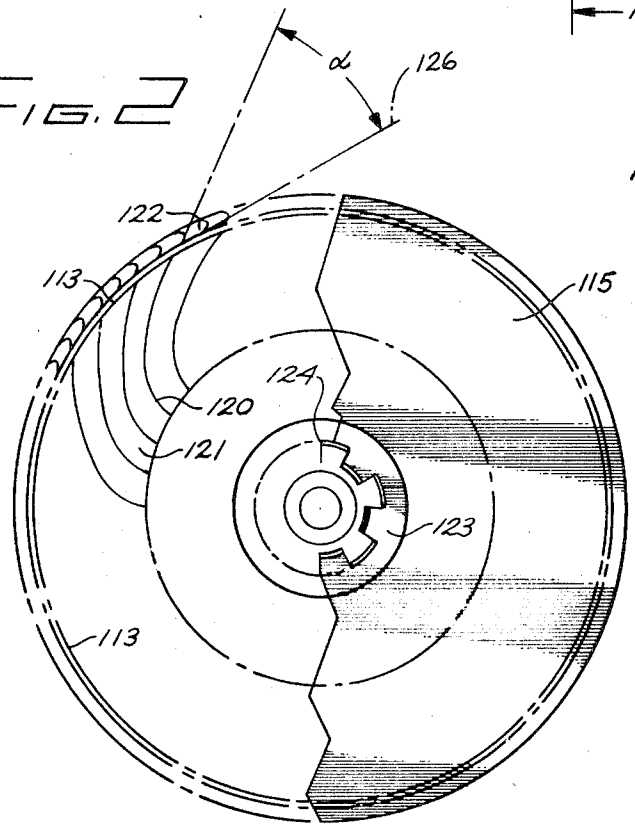


FIG. 3

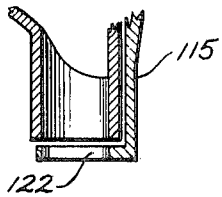
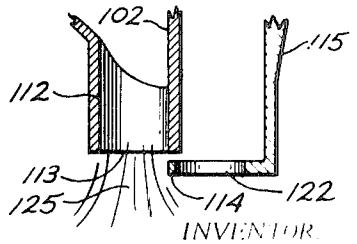


FIG. 4



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FLUID-DYNAMIC ENGINE

This application is an improvement over the teachings of my earlier filed copending application bearing Ser. No. 798,367 and entitled "FLUID-DYNAMIC ENGINE." Briefly, the device therein described is characterized by the process of heating a gas which is moving at the speed of sound in a duct of suitable shape, and by the utilization of the resulting increase of kinetic energy of the gas. The shape of the duct is related to the rate of heat delivery to the gas, so as to maintain the velocity of the gas equal to the local speed of sound over substantially the entire length of duct wherein the heat delivery process takes place. Conversely, this means that, once a duct is built with a certain profile of cross sections matched to a given profile of heat delivery rate, the heat delivery rate cannot be changed much from the design profile without causing the flow to deviate from the desired condition of sonic velocity. As a consequence an engine of this type can be designed to operate very efficiently for a fixed value of rate of heat absorption and power output, but does not lend itself easily to changes of power output.

There are two kinds of variation of power output that are of great practical interest. One is the adjustment of operating conditions to an optimum set of values resulting in the most efficient operation for relatively long periods of time: an example of this is the adjustment of operating conditions of an airplane engine for most efficient cruise at a given altitude with a given gross airplane weight. The other is the quick variation of power setting required for maneuvers, e.g., takeoffs and landings. Here responsiveness is more important than optimum utilization of thermal energy in the engine. Similarly, a ship requires responsive variations of propeller power for docking maneuvers independently from the need for efficient propulsion during cruise. These two distinct requirements can be separately met by two different contrivances, which can, however, be both simultaneously applied to the same engine. Accordingly, this invention refers to means for quickly and responsively varying the power output of a fluid-dynamic engine of the type described in the aforementioned copending application, in the case in which the power output is extracted from the engine in the form of mechanical power delivered to a rotating shaft.

These and other features of the invention can be more fully appreciated through reference to the drawings forming part of this specification wherein:

FIG. 1 is a diagrammatic cross-sectional view of a fluid-dynamic engine embodying the present invention;

FIG. 2 is an end view of the engine of FIG. 1 taken along the lines 2–2 and 2–A, and

FIGS. 3 and 4 show two positions of the power output device corresponding respectively to maximum power output and zero power output.

The structure and operation of the basic fluid-dynamic engine is described in the aforementioned copending patent application and the disclosure thereof is incorporated herein by reference and a more detailed appreciation of the operation of the type of fluid-dynamic engine under consideration may be had by reference to said copending application. Briefly, for the purposes of the present invention, the fluid-dynamic engine E comprises a duct D for conveying a fluid stream thereto. The duct D has a convergent section **10** where the fluid or gas is accelerated to Mach 1 by an adiabatic expansion. The gas then enters the sonic section **11** of the duct D wherein heat is delivered to the fluid while it is moving at sonic speed. The fluid travelling at sonic speed then enters the diffuser section **15**. At this point the gas has a pressure P equal to the critical pressure, conventionally indicated by P^*, related to the stagnation pressure P_o by

$$P^*=[2/(k+1)]^{k/(k-1)}P_o$$

where k is the ratio of specific heats in the gas. This is a general relationship and applies for all gases moving at the speed of sound. For air, $k=1.40$ and $P^*=0.528\,P_o$. The moving gas contains kinetic energy, of which a part is used to compress the gas in a subsonic diffuser to a pressure closer to P_o, and a part remains available for utilization. It is therefore important that recompression be effected with a minimum of losses, in particular frictional losses. Since frictional losses occur predominantly in the boundary layer, where the moving gas is in contact with a solid wall, and are proportional to the velocity of the gas (which varies in a prescribed manner between two fixed predetermined limits, i.e., between the speed of sound at the exit of the sonic section and a preselected lower velocity at the end of the diffuser), the total frictional loss is essentially proportional to what is called the "wetted area," or the wall area of the duct in contact with the boundary layer. This, for a given diameter, is in turn proportional to the length of the diffuser. It is therefore important to build the shortest diffuser possible compatible with the requirement of effective recompression.

On the other hand, the adverse pressure gradient present in the diffuser tends to increase the thickness of the boundary layer as the gas progresses along the diffuser. If the pressure gradient is too steep the boundary layer becomes detached, the diffuser stalls, and no recompression is possible beyond the point of detachment. Hence effective diffusers cannot be built shorter than a minimum length imposed by the requirement that the pressure gradient be less steep than that which causes boundary layer detachment. Within these limitations, the most effective diffuser is the one which has the least wall area in proportion to the cross-sectional area of the duct.

An optimal diffuser, in this sense, can be built by taking advantage of the fluid-dynamic properties of a free jet impinging on a perpendicular flat plate. A free fluid stream, issuing at high velocity from an orifice (assumed of circular shape) and impinging on a flat plate perpendicular to the axis of the stream, is deflected radially away from the center of impingement, which is a point of stagnation. At this point the fluid is momentarily at rest and full stagnation pressure P_o is developed without any frictional losses, because the moving gas arriving there is nowhere in contact with any walls. The stagnation point is surrounded by a region in which most of the original kinetic energy of the gas arriving there is converted in pressure, whereby the gas comes arbitrarily close to stagnation, again without being in contact with any walls. This region is in turn surrounded by another in which the pressure is lower and more unconverted kinetic energy remains available and so on. The stream can be ideally subdivided in concentric stream tubes, separated by ideal stream surfaces, which are concentric surfaces of revolution each generated by a typical streamline rotated about the axis of symmetry. Each concentric stream tube is characterized by a maximum value of recovered pressure, which occurs at the point where the streamline in question is tangent to a surface of equal pressure in the fluid. Each streamline (and each corresponding stream tube) can be labeled with a characteristic value of the pressure ratio P_e/P_o, where P_e is the maximum value of pressure recovered, and P_o is the stagnation pressure (which is fully recovered at the center of the plate). If the wall of the diffuser is shaped to coincide with that stream surface which corresponds to some desired value of P_e/P_o, and extended to the point where said pressure ratio is developed in the fluid in contact with the wall (i.e. to the line where said wall is tangent to the surface of equal pressure $P=P_e$), then the flow inside is unaffected by the wall and the same pressure ratio P_3/P_o appears at all points of the surface of equal pressure, imbedded in the fluid, for which $P=P_e$. This surface intersects said plate along a circular line concentric to the stagnation point: the plate must extend at least to this line.

Under the simplifying assumption that the gas density does not depend on pressure (which is valid in the neighborhood of the stagnation point), the properties of the flow are best described in terms of the velocity potential (see, for example, L. Prandtl and O. G. Tietjens, "Fundamentals of hydro- and aeromechanics," Dover, N.Y. 1957, p. 143). It is easily found that the flow is axially symmetrical and that all streamlines in each plane passing through the axis are cubic hyperbolas described by the equation

$$R^2Z=C \quad (1)$$

where R is the radius, Z is the distance from said plate, and, C an arbitrary constant related to the pressure ratio P_e/P_0. One such streamline is shown as arrow **101** in FIG. **1**. All streamlines are geometrically similar and any one of them can be used to generate a surface of revolution which, together with a plate **102**, defines a possible and acceptable geometry for the diffuser. The coordinates for the diffuser wall **103** are computed by choosing an appropriate value for the arbitrary constant C in equation 1. Within this envelope the gas recompresses to full stagnation pressure P_o at the central point **104** of plate **102**. The surfaces of equal pressure are portions of oblated ellipsoids, each intersecting the plane of the paper of FIG. **1** along a ellipse, for example as represented by the dotted lines **105** and **106**, each identified by a particular value of the pressure ratio P_e/P_o. The line **106** is tangent to the diffuser wall **102** at point **107** and defines the exhaust compression ratio P_e/P_o for which the diffuser is designed. Wall **103** is extended to point **107** and needs go no further to provide recompression to a pressure ratio equal to P_e/P_o. Point **107** as shown in FIG. **1** is, of course, the intersection with the plane of the paper of a circular boundary concentric with the axis of the machine.

The above-described profile is rigorously correct only if the fluid density is constant. This is very nearly true at the end of the recompression process, but not so where the gas pressure P is still close to the critical pressure P^*, for which $M=1$. There is, in fact, a substantial variation of density associated with the very large pressure ratio P_e/P^* for which the diffuser must be designed. This effect, called "compressibility effect," is negligible in the neighborhood of plate **102**, but is predominant in the region immediately downstream of sonic section **11**, where $M=1$ and $P=P^*$.

Although it is possible to develop differential equations which describe the stagnation of a compressible free jet against a perpendicular flat plate, their solution is difficult and can be best handled by a digital computer. The result is a profile quite similar to that described by equation 1, but generally somewhat wider and less tapered near the exit of sonic section **11**. A very good approximation of this rigorous profile can be obtained by noting that compressibility effects are negligible by the time the fluid is slowed down to a Mach number smaller than about 0.3. The diffuser can be divided in two parts, of which part **108** is designed on the basis of a constant pressure-ratio gradient, and part **109** which is designed in accordance with equation 1. The two parts are joined at a point **111** where the Mach number M is approximately equal to 0.3, and the duct profile is computed by imposing the condition that all flow properties and their derivatives be continuous across this point. A small correction is then applied to the profile (whether rigorously generated by digital computer, or approximated as described above), to allow for the growth of the boundary layer: the correction consists of a displacement of the wall outwards by an amount equivalent to the boundary layer thickness at each point of the diffuser.

A diffuser designed in accordance with this disclosure has two important properties which are needed in the implementation of this invention:

a. It can recompress a gas from a pressure ratio P/P_o equal to the critical pressure ratio P^*/P_o (where $M=1$) to an exhaust pressure ratio P_e/P_o arbitrarily close to unity (for which the Mach number would be equal to zero), without suffering boundary layer separation and with very low frictional losses of kinetic energy.

b. It deflects the momentum vector associated with the moving fluid from an axial direction (at the end of sonic section **11**, where the fluid is moving in an essentially parallel stream) to a symmetrically radial direction at the exhaust of the diffuser between plate **102** and the boundary **107** of the sidewall **103**, as indicated by the tip of the streamline arrow **101**. The turning of the momentum vector is causes by the radial component of the pressure gradient between equal pressure surfaces (e.g. **105** and **106**), and does not require any turning vanes in the stream, with associated frictional and turbulent losses of kinetic energy.

It should be noted, parenthetically, that the diffuser here described can be used in applications other than fluid-dynamic engines, of the type under consideration, whenever a fluid must be recompressed by converting part of its initial kinetic energy over a large pressure ratio: if the fluid is incompressible (e.g. a liquid), equation 1 may be used without compressibility corrections.

The radially moving gas at the exhaust of the diffuser is imparted angular momentum by a ring of curved vanes **120** mounted between plate **102** and a flangelike extension **112** of the diffuser wall **103**. The curved vanes **120** are designed to have a radial extension such that they do not protrude beyond an elliptical contour coincident with the constant-pressure ellipsoid **106**, which defines the exhaust of the diffuser proper, and is also a surface of constant gas velocity; and such that they terminate on a straight cylindrical surface **113** on the outside. The curvature of the vanes is computed to provide channels of constant area **121** between them, so as to turn the momentum vector of the gas without changing its magnitude, and to eject the gas at a constant angle α with the tangent plane **126** to the cylindrical surface **113** at every point of the surface. The gas impinges then on a ring of turbine blades **122** designed to utilize the angular momentum of the gas and having the customary inlet and outlet angles appropriate to the velocity of the gas and the tangential angle α. Since the gas emerges uniformly from surface **113** and has everywhere the same velocity and the same tangential angle α, the turbine blades **122** have a constant profile and no twist, and can be cheaply fabricated from simple extruded bars of appropriate cross section. The blades are oriented parallel to the axis of the machine and parallel to the cylindrical surface **113**. They are mounted between a support ring **114** and a wheel **115**, carried by a hub **116**.

The torque collected by the turbine is transmitted to the output shaft **117** by means of a spline comprising an outer element **123** attached to wheel **115** and an inner element **124** attached to shaft **117**. The spline permits the entire wheel, hub and blades assembly to move axially while normally rotating and transmitting torque to shaft **117**. The axial position of the turbine assembly is controlled, for example, by a control lever **118** engaged in a groove **119** cut around hub **116**.

The extent of the axial movement of wheel **115** is sufficient to place the turbine blades **122** across the entire width of the exhaust of the diffuser (surface **113**), or to retract them entirely clear of the emerging stream of gas. FIG. **1** shows the turbine in an intermediate position. FIG. **3** shows the blades fully engaged, and intercepting the entire efflux from the diffuser. FIG. **4** shows the blades fully retracted, and not touched by the emerging gas stream **125**. It is clear that the device can pass from developing full torque to zero, and vice versa, simply by translating axially wheel **115** from one extreme to the other of the travel on spline **123–124** and back. Intermediate values of the torque can be selected by stopping the axial translation in an intermediate position.

When the wheel is in an intermediate position, as shown in FIG. **1** it extracts from the gas stream only the kinetic energy of that fraction of the entire gas stream which engages blades **122**. The remainder, passing between flange **112** and ring **114**, is not utilized: the kinetic energy is eventually dissipated in turbulence and is lost to the machine. For this reason the device here described is best used when the loss of some kinetic energy is of scarce importance compared with the need for quick changes of power output. Since the power delivered by a rotating shaft is equal to the product of the torque times the angular speed, and the torque developed is proportional to the fraction of exhaust stream **125** engaged by blades **122**, the power output can be changed without changing the angular speed of the shaft, which is an advantage in certain applications. This can be done by varying the torque, simply by translating the wheel axially on spline **123–124**. This is contrasted with the slow change of power output which can

be obtained from other turbines in which the angular momentum of the entire spinning assembly, sometimes very large, must be changed to change the output power.

When quick changes of power output are not desired, a loss of kinetic energy is avoided by leaving blades **122** fully engaged with the exhaust gas stream as shown in FIG. **3**. This is the normal operating position for the device when the desired power output is expected to remain constant or to change but slowly, and time is available for adjusting the power level of the device by other means.

What I claim is:

1. A fluid-dynamic engine for the transformation of heat energy into mechanical energy comprising

a fluid-conveying duct having a sonic duct section interposed between a convergent section and a divergent section,

means for conveying fluid through each section of said duct so that it moves in a fluid stream at the speed of sound in said fluid through the sonic section,

means for heating the fluid stream while it is moving at said sonic speed,

drive means including a rotatable shaft coupled to the downstream side of the divergent section for deriving power from the fluid-dynamic engine in the form of rotations of the shaft,

said fluid stream impinging on said drive means for imparting the energy of the fluid stream to said drive means,

and means for controlling the coupling between said fluid stream and said drive means for controlling the power extracted from the engine and thereby the rotations of the shaft.

2. A fluid-dynamic engine as defined in claim **1** wherein the divergent duct section is constructed and defined for recompressing the fluid stream and redirecting the fluid stream into a path for impingement onto said drive means.

3. A fluid-dynamic engine as defined in claim **2** wherein the divergent duct section includes a solid terminal portion upon which the fluid stream impinges and recompresses a portion of the fluid stream to stagnation pressure and includes a ring of curved vanes for redirecting the fluid stream.

4. A fluid-dynamic engine as defined in claim **3** wherein the drive means carries turbine blades adapted for receiving the fluid stream emerging from the curved vanes when coupled thereto.

5. A fluid-dynamic engine as defined in claim **4** including means for controlling the coupling of the drive means to vary the extent of impingement of the fluid stream with the turbine blades and thereby the power derived from the engine.

6. A method of operating a fluid-dynamic engine including the steps of

accelerating a fluid stream through an engine at the speed of sound at the sonic speed of the fluid,

imparting energy to the fluid while maintaining it at said sonic speed,

recompressing the fluid stream to stagnation pressure, and

deriving torque from the energy of the fluid stream.

7. A method for deriving energy from a fluid-dynamic engine including the steps of

accelerating a fluid stream through an engine at the speed of sound at sonic speed in the fluid,

imparting energy to the fluid stream while it is moving at said sonic speed,

passing the fluid through a diffuser at subsonic speeds for maintaining the fluid moving at sonic speed while energy is being imparted to it,

coupling a fluid-responsive element to the stream for deriving power therefrom, and

recompressing the fluid of the fluid stream passing through the diffuser and deriving power from the fluid stream.

8. A method as defined in claim **7** including the step of redirecting the path of the fluid stream in the diffuser prior to coupling to the fluid-responsive element.

9. A method as defined in claim **7** wherein said diffuser is represented by a solid surface shaped to coincide with the shape of a selected streamline surface such as would naturally result in the impingement of an otherwise unrestricted fluid stream of the same characteristics stagnating against said solid terminal portion.