A centrifugal air compressor (11) has an annular air flow path (14) of progressively increasing diameter and progressively diminishing cross-sectional area in which a plurality of internal compression-diffusion stages (30a to 30f) are provided. The stages of progressively greater diameter are defined by sets (29a to 29h) of impeller blades (12) which alternate with sets (32a to 32h) of stator blades (13) along the air flow path (14) and each such stage (30a to 30f) has a diffusion factor below about 0.55. The multi-stage construction combines higher isentropic efficiency with reduced size and may be used as a component of a gas turbine engine (18) or a turbocharger (57) or for other purposes requiring gas compression.
This invention relates to compressors for air or other gases and more particularly to centrifugal or radial flow compressors in which rotating blades are situated in a flow passage that increases in diameter towards its outlet end.

Compressors which employ rotating vanes or blades may be divided into two broad categories on the basis of the configuration of the air flow passage. Centrifugal or radial flow compressors, which constitute the first category, have a flow passage which increases in diameter in the direction of the air flow. Axial flow compressors form the second category and have a flow passage of constant or almost constant diameter.

Centrifugal compressors are basically simpler, more compact and less costly than those of axial flow form. These characteristics are highly desirable in many compressor usages such as in gas turbine engines and engine turbo chargers, but heretofore, it has been necessary to tolerate a relatively low isentropic efficiency in order to gain the benefits of these advantages. The relatively low efficiency typical of prior centrifugal compressors results from the fact that such mechanisms are single stage devices except where a lengthy and complex construction, involving what amounts to a series of such compressors coupled in tandem, is resorted to.
A single compressor stage consists of a set of revolving compressor blades followed by at least one set of diffuser blades which may be stationary or contra-rotating. A set of diffuser blades in addition to a set of compressor blades is required to form a compressor stage since a sizable proportion of the energy imparted to incoming air by the revolving set of compressor blades is initially tangential energy of motion of the air flow. To complete the compression process the air flow must then pass through at least one set of diffuser blades oriented at a different angle than that of the compressor blades to convert the tangential velocity energy into static pressure head.

The degree of compression accomplished in a rotary compressor is expressed by the pressure ratio which is the ratio of pressure at the outlet to that at the inlet. A high pressure ratio across a single compressor stage requires a high loading on the compressor blades, the blade loading being quantitatively expressed by the diffusion factor at a calculated design point. Where sizable pressure ratios are to be achieved in a single stage device. The design point diffusion factor must necessarily be high. Isentropic efficiency, which is an inverse function of the diffusion factor, is therefore, necessarily, low.

The high efficiency of many axial flow compressors largely results from the common practice of employing a series of stages in such compressors. An axial flow compressor may typically have several sets of compressor blades alternating with sets of diffuser blades along the axial flow path, each set of compressor blades in conjunction with a following set of diffuser blades constituting an individual stage. In such a device, the overall pressure ratio of the compressor is the product of the lower pressure ratios of the individual stages of the series. As each stage individually has a low
pressure ratio, each stage operates at high efficiency and the efficiency of the compressor as a whole is also high.

Prior applications of the multiple staging principle to centrifugal compressors have gained efficiency at the cost of sacrificing much of the inherent advantages of the centrifugal compressor configuration. More particularly, most prior multi-staged centrifugal compressors have as a practical matter included a series of essentially separate single stage centrifugal compressors connected together in tandem through bulky and complex air ducting for channeling the outlet flow from one stage radially inward to the smaller diameter inlet of the next subsequent stage. This results in a lengthy, complex and costly construction such as is found in axial flow compressors.

Prior centrifugal compressors also exhibit other problems apart from a relatively low efficiency where a high pressure ratio is to be realized. Designing for a high pressure ratio in a single staged compressor results in extremely high tangential air velocity behind the single long set of compressor blades. This in turn dictates that a bulky and heavy diffuser structure, including lengthly diffuser blades and a voluminous diffusion chamber, be provided at the outlet.

A further practical problem encountered in prior centrifugal compressors arises from the fact that different compressor usages require different pressure ratios and flow capacities. If each compressor model in a family of compressors of different pressure ratio and capacities must be manufactured with a large number of distinct parts usable only in the one particular model, the cost of manufacture of the line of compressors as a whole is increased.

According to the present invention a multi-stage centrifugal compressor having relatively rotatable
inner and outer elements defining an annular flow path of progressively increasing diameter from an inlet end to an outlet end of the compressor, is characterized in that a plurality of blading means are provided in

the flow path to define a plurality of compression-diffusion stages, each of the stages having a design point diffusion factor below about 0.55.

By providing a series of sets of compressor blades alternated with sets of diffuser blades within

a radial flow path of progressively increasing diameter, the efficiency of a centrifugal compressor is greatly increased while preserving the compactness of prior centrifugal compressors and while preserving much of the structural simplicity as well. For a given

pressure ratio and flow capacity, gains in efficiency may in fact exceed those realized in multiple stage axial flow compressors having a similar number of stages as centrifugal effects aid the blade action in generating a pressure rise. Consequently, fewer intern-
al stages may be needed to achieve a given pressure ratio with a given energy input.

As diffusion is largely accomplished internally in stages, a large diffuser is not necessarily needed at the outlet of the compressor. Thus the invention may be

relatively compact in the diametrical direction relative to prior centrifugal compressors. In addition, the multi-staged construction enables the manufacture of a family of centrifugal compressors of different pressure ratio and/or flow capacity without requiring a large

number of different structural elements for each model.

In instances where the multi-stage centrifugal compressor is a component of a gas turbine engine, an engine turbocharger, or the like, the high efficiency of the multi-stage compressor significantly reduces

power losses in such apparatus as a whole.

Various examples of compressors according to the
invention will now be described with reference to the
accompanying drawings in which:

Figure 1 is a broken out side elevation view of
a first embodiment of a centrifugal compressor
constituting the air intake component of a gas turbine
engine.

Figure 2 is an enlarged axial section view of a
portion of the air compressor of Figure 1.

Figure 3 is a broken out perspective view of the
impeller and stator portions of the air compressor
of Figures 1 and 2 further illustrating blading
structure within the compressor.

Figure 4 is a view taken along curved line
IV-IV of Figure 2 illustrating the configurations and
relative inclinations of the blades of successive stages
within the air compressor of the preceding figures.

Figure 5 is a graph depicting input power losses
as a function of blade loading or diffusion factor in
the present invention and in typical prior air compress-
ors.

Figure 6 is an axial section view of a portion of
an air compressor basically similar to that of Figure 2 but with modifications of the blading structure to
vary the air flow capacity.

Figures 7A to 7G are diagrammatic views illustrat-
ing further modifications of the compressor of Figure 1
which enable realization of any of a series of different
pressure ratios and/or flow capacities utilizing much
of the same basic structural components.

Figure 8 is an axial view of an engine turbocharger
having a compressor section in accordance with an
embodiment of the invention.
Referring initially to Figure 1 of the drawings a radial flow or centrifugal compressor 11 has an impeller 12 disposed for rotation within an annular stator member 13 respectively constituting relatively rotatable inner and outer elements that jointly define an annular flow path 14.

Impeller 12 is of progressively increasing diameter from the air inlet end 16 of the flow path 14 towards the air outlet end 17. Stator member 13 has an inner diameter which also progressively increases along the flow path 14 but at a lesser rate so that the spacing of the impeller from the stator diminishes towards the outlet end 17 of the flow path. The diminishing spacing of the impeller 12 from stator member 13 along flow path 14 compensates for the progressively increasing diameter of the flow path towards the outlet end which would otherwise cause the flow path to have a progressively increasing cross-sectional area. The decrease in spacing also compensates for the air compression that occurs along the flow path 14 and which progressively reduces the volume occupied by unit mass of air as it travels along the path.

In the embodiment of the invention depicted in Figure 1, compressor 11 constitutes the air intake component of a gas turbine engine 18 and certain structural features of this particular compressor 11 are specialized for this context. For example, the impeller 12 is supported on and driven by a forward extension of the main shaft 15 of the gas turbine engine 18 and the inner stator member 13 is secured to an outer stator member 19 which is itself secured to the main housing 21 of the gas turbine engine and supported thereby.

Aside from the air intake section defined
by the compressor 11, the gas turbine engine 18 may be of a known design such as that described in prior United States Patent 4,030,288 and therefore will not be further described except for certain components which directly coact with elements of the compressor. It should be understood that usage of a compressor embodying the invention is not limited to the context of gas turbine engines. When the invention is employed in other contexts or for other purposes, the impeller 12 may be journaled within the stator members 13 and 19 by suitable bearing structures known to the art and may be driven by any of a variety of known external motors. Similarly, the stator may be provided with appropriate support means of any of various known forms.

The inner and outer stator members 13 and 19 jointly form an annular diffusion chamber 22 which receives air from the outlet end 17 of the compressor flow path 14. In order to minimize the size of the compressor in the radial direction, outer stator member 19 is shaped to situate most of the volume of diffusion chamber 22 adjacent the smaller diameter forward portion of inner stator member 13. This is a practical configuration in that the lengthy, radially extending diffuser vanes required at the outlet end of the flow path in many conventional single stage centrifugal compressors are not necessarily required in the present invention.

Diffusion chamber 22 is communicated with a compressed air outlet tube 23 which in the present example supplies the compressed air to the combustor 24 of the gas turbine engine 18 through a heat exchanger module 26 which transfers heat from the exhaust of the engine to the incoming compressed air. In instances where the compressor 11 is used for purposes other than in a gas turbine engine, the outlet tube 23 may be replaced with a hose or
other conduit means suitable for connection with the compressed air utilizing device.

Compression of air within the flow path 14 is accomplished by blading means 27 depicted on a larger scale in Figure 2, which form a plurality of internal compression-diffusion stages 30a to 30f of low blade loading or diffusion factor within the flow path 14.

Referring now to Figures 2 and 3 in conjunction, a plurality of spaced apart sets of compressor blades 28 extend radially from impeller 12 into the flow path 14, there being six such sets 29a, 29b, 29c, 29d, 29e, 29f of compressor blades, proceeding from the air inlet end 16 to the air outlet end 17, in this example. The individual compressor blades 28 of each set 29a to 29f are equiangularly spaced around the rotational axis of the impeller 12 and owing to the progressively diminishing thickness of the flow path 14, the blades of each successive set extend progressively smaller distances from the impeller.

A plurality of spaced apart stationary sets of diffuser blades 31 extend into flow path 14 from the inner stator member 13, there being seven sets 32a, 32b, 32c, 32d, 32e, 32f, 32g of diffuser blades 31 in this example. The sets 32a to 32g of diffuser blades are alternated with the sets 29a to 29f of compressor blades 28 except that the two final sets of diffuser blades 32f and 32g are both behind the final set 29f of compressor blades. Individual blades 31 of each set 32a to 32g of diffuser blades are also equiangularly spaced apart with respect to the rotational axis of the compressor and the blades of each successive set 32a to 32g extend progressively shorter distances from the stator member to accommodate to the progressively diminishing thickness of the flow path 14.
Each set 29a to 29f of compressor blades in conjunction with the following set of diffuser blades 31 constitutes one of the plurality of compression-diffusion stages 30a to 30f situated in the flow path 14. Thus in the present example compressor blade set 29a and diffuser blade set 31a form a first compression-diffusion stage 30a and compressor blade set 29b in conjunction with diffuser blade set 32b form a second compression-diffusion stage 30b, there being six such stages in this example.

Referring now to Figure 4, the individual compressor blades 28 of each set 29a to 29g are inclined relative to the rotational axis 18' of the impeller to impart an increment of flow velocity to intercepted air as the blades turn in the direction indicated by arrows 33 in the drawing. The compressor blades 28 of each successive set 29a to 29g have a progressively increasing angulation relative to axis 18' to accommodate to the progressive increase of free stream velocity which occurs along the flow path. The blades 31 of the successive sets 32a to 32g of diffuser blades have an opposite angulation relative to axis 18', which also becomes progressively greater for each successive set of diffuser blades, in order to convert tangential velocity energy imparted to air by the preceding set of compressor blades into static pressure head energy.

Thus, with reference to Figures 1 and 2, the compression achieved by compressor 11 as a whole is accomplished in six distinct compression-diffusion stages 30a to 30f along the flow path 14. The pressure ratio of each individual stage 30a to 30f may therefore be low relative to a conventional centrifugal compressor having a single long set of compressor blades followed by a single long set of diffuser blades, designed to accomplish the same degree of
compression. Since each component stage 30a to 30f of compressor 11 operates at a low pressure ratio and therefore a high level of efficiency, the aggregate efficiency of the several stages in combination is itself high in comparison with conventional single staged devices.

In order to fully realize the gains in efficiency inherent in the multiple stage construction, each compression-diffusion stage 30a to 30f is designed to have a free stream flow velocity which is below supersonic throughout the region of blading means 27 and to have a diffusion factor below about 0.55 at each stage. As is known in the art, the diffusion factor of a single compression-diffusion stage may be selected, within limits, by an appropriate fixing of the shape, angulation and number of compressor blades and diffuser blades in relation to the configuration of the flow path and the rotational velocity of the compressor blades. More particularly, in a compressor stage wherein the free stream air velocity is subsonic throughout the region of the blading as is the case in the compressors of the present invention, diffusion factor (D.F.) is given by the expression:

\[
(D.F.) = 1 - \frac{V_2}{V_1} + \frac{\Delta V_e}{2 \sigma V_1}
\]

where:  
\( V_1 \) = inlet flow velocity relative to blade row  
\( V_2 \) = outlet flow velocity relative to blade row  
\( V_e \) = tangential flow velocity relative to blade row  
\( \sigma \) = blade row solidity (proportion of open flow space to total cross-sectional area of flow path in blade region)  

The benefit of establishing a design point diffusion factor below about 0.55 at each of the several compression-diffusion stages 30a to 30f may be seen
by referring to Figure 5 which is a graph depicting measured input energy losses, that is energy which does not become available as pressure energy at the outlet of the compressor, as a function of diffusion factor for three different types of rotary compressor all of which achieve the same overall pressure ratio or degree of compression. Rectangles 34 designate measured losses for a conventional single stage centrifugal compressor which necessarily must have a relatively high diffusion factor to accomplish the desired degree of compression in the single stage. Circles 36 indicate the relatively low measured losses in a conventional multiple stage axial flow compressor in which the diffusion factor for each individual stage may be much lower and therefore more efficient. Triangles 37 indicate the measured losses in a multiple stage centrifugal compressor embodying the present invention. It should be observed that the compression is accomplished in the present invention with a diffusion factor 37 per stage which is substantially lower than that 36 of the lengthier and more complex axial flow compressor. The reason for this greater efficiency of the present invention as indicated by triangles 37 is believed to be that centrifugal force supplements the direct effect of the blading in achieving compression. This effect does not occur in the non-radial flow path of an axial flow compressor.

The significance of a design point diffusion factor value of about 0.55 as an upper limit for the individual stages of the present invention is also evident in Figure 5. It may be seen that there is not a linear relationship between power loss and diffusion factor. Instead, as the diffusion factor is increased from a very low value, losses rise at a relatively moderate rate, indicated by lines 38, until a value of about 0.55 is reached. Thereafter losses increase much
more sharply with increasing diffusion factor as indicated by lines 39. Efficiency is an inverse function of power losses and thus it may be seen that efficiency drops off relatively sharply after the diffusion factor value of about 0.55 is passed.

Returning to Figure 1, the high efficiency of the compressor 11 in turn increases efficiency of the gas turbine engine 18 itself as power losses in the compressor section of the engine are reduced. As compared with a gas turbine engine utilizing an axial flow compressor configuration for the purpose of realizing somewhat comparable efficiencies, the engine 18 of this example is much more compact and the compressor section is simpler and less costly.

While the compressor 11 described above is provided with six internal compression-diffusion stages 30a to 30f, varying numbers of stages may be provided by changing the number of sets of compressor blades 28 and diffuser blades 31. Moreover, the construction readily lends itself to manufacture of a family of compressors of different pressure ratio and/or flow capacities by varying only the number and disposition of the sets of blades 28 and 31 within the flow path 14 while otherwise utilizing identical components for the several compressor models. Referring to Figure 6 for example, a compressor 11' having a lower pressure ratio but a smaller air mass flow rate and therefore a smaller driving power requirement may be produced simply by removing the first set 29a of compressor blades and the first set 32a of diffuser blades, shown in phantom in Figure 6, while otherwise utilizing components, such as impeller 12 inner stator member 13 and outer stator member 19 identical to those of the previously described embodiment. In general, the elimination of compression-diffusion stage blading means 27 from the air inlet 16 region
of the flow path 14 has an effect of reducing both air mass flow and pressure ratio while the elimination of stages of blades from the region nearest the air outlet end 17 has the predominate effect of reducing pressure ratio. Adding of stages at the inlet end increases mass flow and pressure ratio while additional stages near the outlet end predominately raise pressure ratio.

Thus while a limited number of specific blading modifications will be described with reference to Figures 7A to 7G and specific parameters will be given, such examples are not exhaustive of the possible modification. In accordance with the above discussed relationships, other modifications may be made to provide other mass flows and pressure ratios.

Figures 7A to 7C diagramatically illustrate how a series of compressors lla, llb, llc respectively of different pressure ratio and/or air flow capacity may be configured by simply varying the numbers of sets of blades in the air flow path while otherwise utilizing identical components. Where the compressors are embodied in gas turbine engines as previously described, this enables production of a family of engines 18a, 18b, 18c of different output power rating and fuel consumption requirements simply by varying the blading in the compressor section.

While the gas turbine engines 18a, 18b and 18c may be of known construction apart from the compressors lla, llb, llc, the coaction of the compressor sections with the other portions of the engines may best be understood by briefly reviewing certain basic structure of such engines. Referring specifically to Figure 7A, for example, such engines 18a have a fuel burning combustor 24a receiving compressed air from compressor lla through heat exchanger 26a. Output gasses from the combustor 24a drive a gasifier
turbine 42a that turns the impeller 12a of the compressor 11a. Nozzle vanes 43a direct the gas flow from combustor 24a and gasifier turbine 42a to a power turbine 44a which turns the engine output shaft 46a, the exhaust gas from the power turbine being discharged through the heat exchanger 26a to preheat the compressed air which is delivered to the combustor.

The modified compressor 11a of Figure 7A is similar to that previously described with reference to Figure 2 except that the first two compression-diffusion stages 30a, 30b have been eliminated by removing the first two sets 29a and 29b of compressor blades 28 and the first two sets 32a and 32b of diffuser blades 31. As a result of this simple modification, the compressor 11a of Figure 7A has a lower air flow and a lower pressure ratio of about 4.5. The output power rating of the gas turbine engine 18a is then typically about 894 kW, realized with a fuel efficiency of less than about 0.4 brake specific fuel consumption (BSFC).

Figure 7B illustrates a gas turbine engine 18b of substantially greater output power rating but which may be structurally identical to that of Figure 7A except for another modification of the blading structure within the compressor 11b. Compressor 11b is similar to the compressor 11 of Figure 2 except that the first and final sets 29a and 29g of compressor blades of Figure 2 and the first and final two sets 32a, 32f, 32g of diffuser blades 31 have been eliminated. The pressure ratio achieved by the compressor 11b of Figure 7B remains approximately the same as that of Figure 7A but the volume of air passing through the compressor 11b of Figure 7B and on to the combustor 24b is increased to the extent that the power output of the turbine engine 18b is now about 149 kW.
Figure 7C depicts another modification, confined to the blading means 27c of the compressor, by which a similar basic gas turbine engine 18c including similar impeller 12c and stator member 13c elements in the compressor may be used to produce an engine of still higher rated output power. The compressor 11c of gas turbine engine 18c is identical to that of the first described embodiment of Figure 2 except that the final set 29f of compressor blades of Figure 2 have been removed and the final two sets 32f and 32g of diffuser blades are now situated more forwardly in the flow passage and configured for that changed location. This makes the pressure ratio of the compressor 11c of Figure 7C about 6.5 and provides an increase of volumetric air flow relative to the Figure 7B embodiment. The rated power output of the gas turbine engine 18c of Figure 7C is typically about 2609 kW.

If the modifications of the compressor blading arrangements are accompanied by modifications of other components as well, the family of gas turbine engines may be extended to still higher output power ratings, examples of which are depicted in Figures 7D, 7E, and 7F. Referring initially to Figure 7D, by forming the impeller 12d and inner stator member 13d to be relatively elongated at the front end 16d, additional compression-diffusion stages, such as stage 30g, may be provided at the air inlet end of the compressor 11d to further increase rated power output of the engine 18d. Thus the compressor 11d of Figure 7D has an additional set of compressor blades 29g followed by an additional set 32h of diffuser blades at the front end of the air flow path 14d. The final two sets of compressor blades 29e and 29f of the embodiment of Figure 2 and the intermediate set of diffuser blades 32e have been removed. The final
two sets of diffuser blades 32f and 32g are again situated more forwardly in the flow passage and have configurations appropriate to that portion of the passage. With these modifications, the pressure ratio of the modified compressor 11d of Figure 7D remains at about 6.5 but air mass flow is sizably increased causing the rated power output of the gas turbine engine 18d to be increased to about 3728 kW.

By making somewhat more extensive modifications, still greater power output ratings may be obtained. For example as depicted in Figure 7E an auxiliary compressor section 47e may be added between the primary compressor 11e and the heat exchanger 26e. The auxiliary compressor section 47e may for example have two spaced apart sets 48e and 49e of compressor blades on an auxiliary impeller 50e each being followed by a set, 51e and 52e respectively of diffuser blades. An annular air duct 53e is provided to receive the output flow from the primary compressor section 11e and to return the flow radially inward for delivery to the air inlet end of the auxiliary compressor section 47e. Primary compressor section 11e is itself identical to the compressor 11d of the previous Figure 7D. To best realize the advantages of the compressor modification of Figure 7E, other elements of the gas turbine engine 18e are modified to the extent of providing an additional gasifier turbine stage 54e to drive the impeller 50e of the auxiliary compressor stage 47e. The modifications depicted in Figure 7E produce an overall compressor pressure ratio of about 12 and raise the rated power output of the gas turbine engine 18e to about 4100 kW.

Figure 7F illustrates still a further modification of the gas turbine engine 18f in which
the structure remains similar to that described above with reference to Figure 7E except that in the embodiment of Figure 7F the annular air duct 53f which communicates the primary compressor section 11f with the auxiliary compressor section 47f includes an intercooler or heat exchanger 55f which acts to cool the compressed air in passage between the two compressor sections. Intercooling reduces the amount of power required to drive a compressor and this power reduction is reflected in an increased power output at the output shaft 46f of the gas turbine engine 18f. By this further modification, the gas turbine engine 18f is made to deliver about 4846 kW.

Referring now to Figure 7G the power output and therefore the fuel consumption rate of any of the gas turbine engines described above may be adjusted downwardly as desired by disposing a set of air flow reducing stator vanes 56g in the inlet end of the air flow path 14g in front of the initial set 29g of compressor blades 28g. Stator blades 56g are angled relative to the air flow path 14g in order to constrict the air flow path and thereby reduce air mass flow to any desired extent.

As previously pointed out, the invention is not limited to compressors which function as an air intake component of gas turbine engines, but may also advantageously be utilized in free standing compressors for supplying compressed air to various pneumatic systems or to other mechanisms which include a compressor as one component. Figure 8 illustrates an example of the latter category in which a compressor 11h embodying the invention constitutes an air intake component of a turbocharger 57 for an internal combustion engine 58.

A turbocharger 57 increases the fuel
efficiency of the engine 58 by boosting intake manifold pressure and uses energy recovered from the exhaust gas of the engine for this purpose. More specifically, the turbocharger includes a turbine 59 driven by the engine exhaust flow and which drives the compressor 11h that supplies compressed air to the engine 58 intake manifold. Centrifugal compressors, preferably in combination with a centripetal turbine, are advantageous in turbochargers in view of the basic compactness and structural simplicity of such compressors but if a conventional single stage centrifugal compressor is used, the adiabatic efficiency of the turbocharger is undesirably limited. This adversely affects the power output of the associated engine 58 per unit of fuel consumed. Very high efficiency together with simplicity and compactness in both the axial and radial direction can be realized by embodying a multi-stage radial flow compressor 11h in accordance with present invention in a turbocharger 57.

The compressor 11h and turbine 59 are secured to opposite ends of a housing 61 which journals a drive shaft 62 that defines the rotational axis of both the compressor and turbine.

Compressor 11h has an annular outer stator member 19h secured to the front end of housing 61 in coaxial relationship with the drive shaft 62 and which defines a broad air intake passage 64. Stator member 19h also forms a volute or annular collection chamber 66 which is communicated with intake manifold 67 of engine 58, the collection chamber being coaxial with intake passage 64 and being of greater diameter. A rotatable impeller 12h is supported on the forward end of drive shaft 62 within stator member 19h and in conjunction with an inner stator member 13h forms an annular air flow path 14h leading from
air intake passage 64 to collection chamber 66. Impeller 12h and inner stator member 13h have configurations which cause the air flow path 14h to be of progressively increasing diameter in the direction of air flow while being of progressively diminishing thickness towards the air outlet end.

Multi-stage blading means 27h of the type previously described is situated within the flow path 14h to provide a plurality of sub-sonic internal compression-diffusion stages 30j, 30k, 30L each having a design point diffusion factor below about 0.55. In this example, the blading means 27h includes three spaced apart sets of compressor blades 28h secured to impeller 12h and alternated with three spaced apart sets of diffuser blades 31h secured to stator member 13h. Thus three compression-diffusion stages 30k, 30j, 30L are provided in this embodiment each being defined by a set of compressor blades 28h and the immediately following set of diffuser blades 31h.

While the multi-staged compressor 11h is advantageous in a turbocharger employing any of a variety of different types of turbine 59, very high efficiency is best realized by using a centripetal turbine 59 which is also of a multi-staged construction.

The turbine 59 of this example has an annular stator 76 secured to the back end of housing 61 and forming an exhaust gas outlet passage 77. A turbine rotor 78 is secured to the back end of drive shaft 62 in coaxial relationship with the shaft and in conjunction with an annular inner stator member 79 forms a gas flow path 81 which is of progressively less diameter but progressively increasing thickness from a gas inlet end 82 to a gas discharge end 83.

Stator 76 also forms an annular volute or gas receiving chamber 84 which is communicated with the inlet end 82 of gas flow path 81 and which is
also communicated with the exhaust gas manifold 86 of engine 58. To cause the exhaust gas flow to drive the turbocharger 57, three spaced apart sets 87a, 87b, 87c of rotor vanes are secured to rotor 78 and extend into the flow path 81, the rotor vanes being angled with respect to the direction of gas flow. To maximize the reaction forces of the gas flow on the rotor vanes 87a, 87b and 87c, one of three sets 88a, 88b and 88c of stator vanes precedes each set of rotor vanes 87a, 87b and 87c respectively along the gas flow path. As the pressure drop at each individual set of rotor vanes 87a, 87b and 87c is substantially lower than the total pressure drop through the turbine 59 as a whole, each set of vanes operates at a relatively high efficiency in comparison with a single stage centripetal turbine having a single long set of rotor vanes.

The above described turbocharger 57 construction enables the impeller 12h and rotor 78 to be situated on the same shaft 62 to turn at the same speed and in most cases the two elements need not have any large disimilarity in diameters. With the rotational speeds and diameters of both the impeller 12h and rotor 78 closely matched, centrifugal stresses are also closely balanced at a high but tolerable level to optimize air and gas throughput in relation to the size and weight of the turbocharger.

In the operation of the embodiment of the invention depicted in Figures 1 to 3, impeller 12 of the compressor 11 is turned by the gas turbine engine main shaft 18. The resulting rotary motion of the several sets 29a to 29f of compressor blades causes air to be drawn into inlet end 16 and to be forced along flow path 14 to the diffuser chamber 22.
from which it is transmitted to the fuel combustor 24 of the engine 18 through tube 23 and heat exchanger module 26.

Air is compressed in stages during passage through flow path 14 as each set of compressor blades 29a to 29f imparts additional energy to the air flow. At each successive set 29a to 29f of compressor blades the added energy appears in part as a rise of static pressure, in part as tangential velocity energy of motion and to some extent as heat. The set 32a to 32g of diffuser blades 31 situated behind each set 29a to 29f of compressor blades converts a substantial portion of the velocity energy into additional static pressure. This process of compression followed by diffusion is repeated at each successive compression-diffusion stage 30a to 30f and since the pressure ratio at each successive stage is substantially less than the pressure ratio of the compressor as a whole, each individual stage operates at high efficiency and the overall compression process is therefore highly efficient.

Where the compressor 11 is an air intake component of a gas turbine engine 18 as in this example, the gains in efficiency in the operation of the compressor translate into increased efficiency of the gas turbine engine itself. To the extent that power losses in the compressor 11 are reduced, the deliverable power output of the gas turbine engine 18 is increased. Moreover the compressor 11 is very compact in both the axial and radial direction enabling the gas turbine engine 18 as a whole to also exhibit a very desirable degree of compactness.

Significant aspects of the operation of the compressors 11a to 11g of the gas turbine engines 18a to 18g of Figures 7A to 7G are essentially similar except insofar as different pressure ratios and air
mass flows and therefore different output power ratings for the gas turbine engines are realized in the manner hereinbefore described.

In the operation of the turbocharger 57 of Figure 8, the exhaust gasses from engine 58 drive turbine 59 which in turn drives the compressor 11k through drive shaft 62. The blading means 27h of the compressor 11k draws air into flow path 14h and delivers such air to the intake manifold 67 of the engine 58 through diffusion chamber 66. Again, the multiple staged blading means 27h of the compressor 11h enables the compression and diffusion process to be accomplished in stages each of which individually exhibits a small pressure ratio and low diffusion factor thereby providing for high efficiency in the operation of the compressor 11h and thus in the operation of the turbocharger 57 as a whole.
1. A multi-stage centrifugal compressor (11) having relatively rotatable inner (12) and outer (13) elements defining an annular flow path (14) of progressively increasing diameter from an inlet end (16) to an outlet end (17) of the compressor, characterized in that a plurality of blading means (27) are provided in the flow path to define a plurality of compression-diffusion stages (30 a-f), each of the stages having a design point diffusion factor below about 0.55.

2. A compressor (11) according to claim 1, having a free stream flow velocity, relative to the blading means (27) which is below supersonic through the blading means (27).

3. A compressor (11) according to claim 1 or claim 2, wherein the blading means (27) includes a plurality of sets (29a to 29b) of compressor blades (28) extending into the flow path (14) from a first of the elements (12, 13) and a plurality of sets (32a to 32b) of diffuser blades (31) extending into the flow path (14) from the second of the elements (12, 13), the sets of diffuser blades being alternated with the sets of compressor blades along the flow path (14).

4. A compressor (11) according to claim 3, wherein the first element is the inner element (12) and the second element is the outer element (13).
5. A compressor (11) according to claim 4, wherein the inner element (12) is a rotatable impeller of progressively increasing outside diameter from the inlet end (16) to the outlet end (17) of the flow path (14), and wherein the outer element (18) is a non-rotatable stator member of progressively increasing inside diameter from the inlet end (16) to the outlet end (17) of the flow path (14).

6. A compressor (11) according to any of claims 3 to 5, wherein the outward curvature of the annular flow path (14) relative to the rotational axis of the compressor becomes progressively greater from the inlet end (16) to the outlet end (17), and wherein the flow path (14) is of progressively diminishing cross-sectional area from the inlet end (16) to the outlet end (17), successive ones of the sets (29a to 29h) of compressor blades and sets (32a to 32h) of diffuser blades have blades (28, 31) of progressively decreasing size.

7. A compressor (11) according to any of claims 3 to 6, wherein successive ones of the sets (29a to 29h) of compressor blades (28) and the sets (32a to 32h) of diffuser blades (31) each have a progressively increasing number of individual blades from the inlet end (16) to the outlet end (17) of the flow path.

8. A compressor (11) according to any of claims 1 to 7, further comprising vane means (56) for restricting the rate of air flow into the compressor (11) at the air inlet end (16) of the flow path (14).

9. A compressor (11) according to any of claims 1 to 8, in combination with a gas turbine engine (18), wherein the compressor (11) constitutes the air intake element of the gas turbine engine and wherein the
impeller (12) is driven by the engine (18).

10. A compressor (llh) according to any of claims 1 to 8, in combination with an engine turbocharger (57) turbine (59), wherein the compressor (llh) constitutes the air intake element of the turbocharger (57) and is driven by the turbine (59).
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The present search report has been drawn up for all claims

The Hague 01-10-1980 DE SCHEPPER