

[54] **CENTRIFUGAL COMPRESSOR BOUNDARY LAYER CONTROL**

963,540 1/1950 France 415/213 R
1,057,137 5/1959 Germany..... 415/213 R

[75] Inventor: **Burton A. Jones**, North Palm Beach, Fla.

Primary Examiner—Henry F. Raduazo
Attorney, Agent, or Firm—Jack N. McCarthy

[73] Assignee: **United Aircraft Corporation**, East Hartford, Conn.

[22] Filed: **Mar. 14, 1974**

[21] Appl. No.: **451,257**

[57] **ABSTRACT**

[52] U.S. Cl. **415/213 R; 415/DIG. 1; 415/215**
[51] Int. Cl.² **F04D 7/02**
[58] Field of Search..... **415/213, 215, DIG. 1;**
416/183, 186, 185

A centrifugal compressor is shown with an inlet directing a fluid thereto and a diffuser into which the compressor directs its output. A group or array of circumferential grooves are located in the surface of a stationary shroud which covers the blades of the compressor.

[56] **References Cited**

FOREIGN PATENTS OR APPLICATIONS

503,332 5/1954 Canada 415/213 R

3 Claims, 3 Drawing Figures

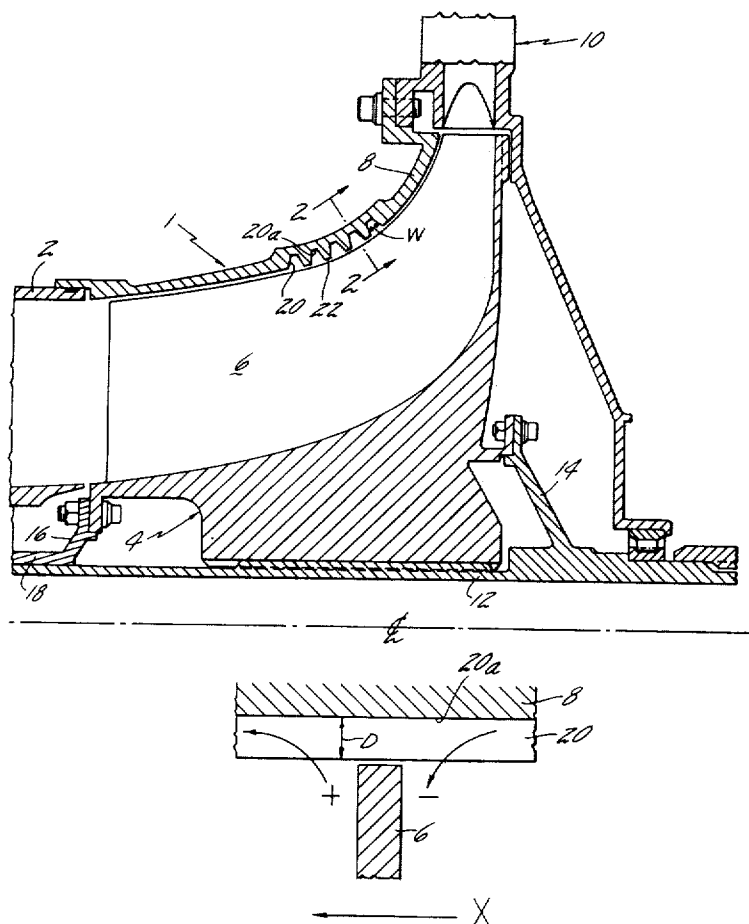
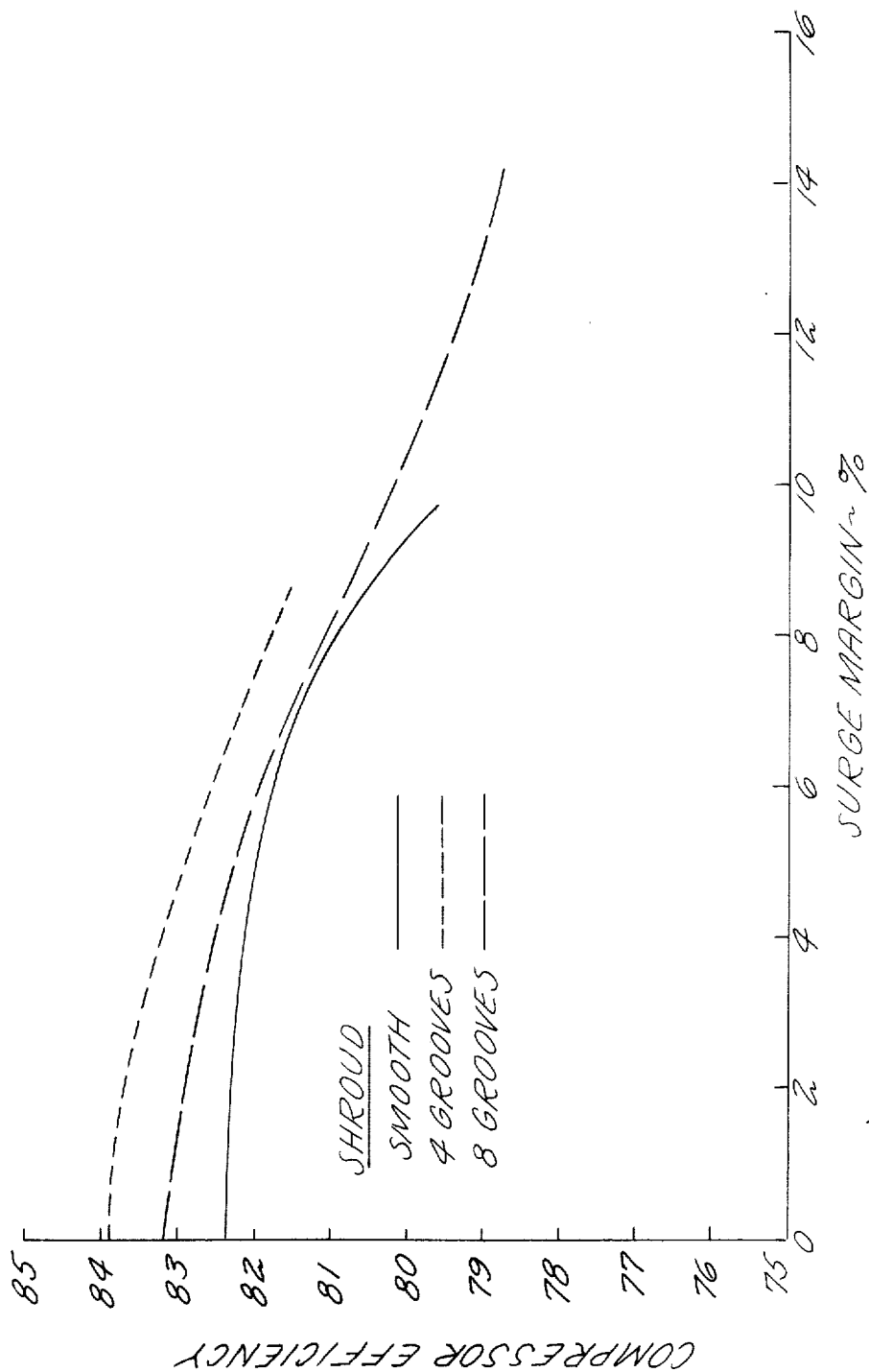


Fig. 3



CENTRIFUGAL COMPRESSOR BOUNDARY LAYER CONTROL

BACKGROUND OF THE INVENTION

This invention relates to means for increasing compressor performance and one means previously used in the prior art has been to incorporate bleed openings for obtaining a desired flow characteristic.

SUMMARY OF THE INVENTION

A primary object of this invention is to provide a high-work centrifugal compressor of good efficiency which will minimize flow energy losses due to shroud friction heating.

It is an object of this invention to reduce local adverse pressure gradients caused in a flowpath when the flow turns from an axial direction to a radial direction.

Another object of the invention is to produce a shroud-side flow profile at the impeller exit to provide good diffuser performance.

A further object of this invention is to relieve the local shroud-side adverse pressure gradient with a particular form of boundary layer control that stabilizes the shroud-side flow and improves the impeller exit flow profile and thereby increases the efficiency of the impeller/diffuser system.

Another object of the present invention is to equilibrate blade-to-blade pressure differences (circumferential flow nonuniformity) that tend to produce local time-dependent high adverse pressure gradients with respect to the tangential direction.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a longitudinal cross section of a centrifugal compressor.

FIG. 2 is an enlarged cross section taken along the line 2—2 of FIG. 1.

FIG. 3 is a chart showing a comparison of compressor efficiency with surge margin for a 6:1 pressure ratio impeller.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to FIG. 1 a centrifugal compressor 1 having an impeller 4 with blades 6 is shown with an inlet section 2, a stationary shroud 8 adjacent the blades 6, and a diffuser 10. The impeller 4 is splined to a shaft 12, which may form part of a gas turbine engine and is fixed to the shaft by being bolted at its rearward side to an annular flange 14 which is shown integral with the shaft. The forward end of the impeller 4 is bolted to another annular flange 16 which is fixed to a sleeve 18 mounted around the shaft 12. Sleeve 18 can be fixed with respect to shaft 12 by any means desired. A compressor of this type is shown and described in U.S. Pat. No. 3,420,435.

An array of circumferential grooves 20 are located in the surface of the stationary shroud 8 exposed to the flow path between the blades 6 and through the impeller 4. FIG. 2 shows one of the grooves 20. Each groove 20 has a bottom surface 20a and a depth D, while adjacent grooves are separated by a land 22 which is the portion of the shroud wall which joins two grooves. The side of the blade 6 facing the indicated direction X of impeller rotation is labeled with a plus (+) to indicate high pressure; and the side of the blade 6 facing away from the direction of rotation is labeled with a minus

sign (—) to indicate low pressure. The volume provided by the circumferential grooves 20 accommodates low energy flow adjacent the wall of the shroud 8. Low energy air adjacent the wall of the shroud 8 is pumped into each groove 20 by the high pressure side of the blades 6. In the cavity the meridional and normal velocity components are reduced to zero, resulting in increased static pressure and tangential velocity of the air in the cavity. This accommodation of low-energy air reduces the size of the boundary layer in the vicinity of the grooves. An equilibrium flow of air into and out of the grooves is established by the relative high and low pressure sides of the blade. The partially stagnated air in the groove is pumped out of the groove as a result of the pressure difference between the high pressure in the groove and the low pressure on the low pressure side of the blade, whereupon it mixes with the higher energy through flow and is carried downstream. As a result of the circumferential communication provided by the plurality of grooves, circumferential flow nonuniformities are washed out thus attenuating time non-steady pressure gradients.

For maximum effectiveness, grooves 20 are located near the point of incipient boundary layer separation on the surface of the stationary shroud 8. Conventional boundary layer calculation techniques are used to estimate this point. It will usually be found to occur where the surface of the shroud adjacent the flowpath turns abruptly toward the radial direction. The depth D and width W of each groove 20 should approximate the boundary layer displacement thickness calculated at the point of incipient separation. If the depth or width of a groove 20 exceeds the boundary layer displacement thickness by a value greater than 2, the groove volume will accommodate higher energy air and the effectiveness will diminish. This observation has been established from systematic experimentation with groove geometry. The optimum number of grooves and their spacing has also been established on the basis of systematic tests of a variety of grooved shroud configurations. The optimum number of grooves was found to be no less than 4 and no more than 8. The grooves were spaced one-half groove width apart, this means that the width of the land 22 is one-half of the width of the grooves. Where the side walls of the grooves are tapered the average groove width is used.

FIG. 3 is a comparison of net efficiency with surge margin for a 6:1 pressure ratio impeller with various shroud configurations. The shroud configurations which were used involved a shroud having a smooth inner surface, a grooved inner surface having 4 grooves, and a grooved inner surface having 8 grooves.

I claim:

1. A centrifugal compressor being mounted for rotation having an axial inlet for delivering a fluid thereto, and a radial outlet for receiving a compressed fluid therefrom, said compressor having blades thereon each blade having a continuous outer free edge from said inlet to said outlet, said blades having an abrupt flow path section from axial flow to radial flow, a fixed shroud located between said inlet and outlet spaced from the edges of said blades, said shroud having annular grooves located therearound facing said blades, said circumferential grooves being located on the surface of the shroud opposite the abrupt flow path section where the flowpath from said inlet turns abruptly toward a radial direction into said outlet.

2. A combination as set forth in claim 1 wherein the number of grooves range from 4 to 8.

3. A combination as set forth in claim 1 wherein the grooves are spaced one-half of a groove width apart.

* * * * *