# United States Patent [19]

**Downing**

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[54] **VANE ACTUATION SYSTEM**


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[73] Assignee: **General Motors Corporation,** Detroit, Mich.



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[52] **U.S. Cl.** ..... **415/151;** 415/149

[51] **Int. Cl.²** ..... **F04D 27/00**

[58] **Field of Search** ..... 415/147, 149, 150, 151, 415/18



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*Primary Examiner*—Henry F. Raduazo

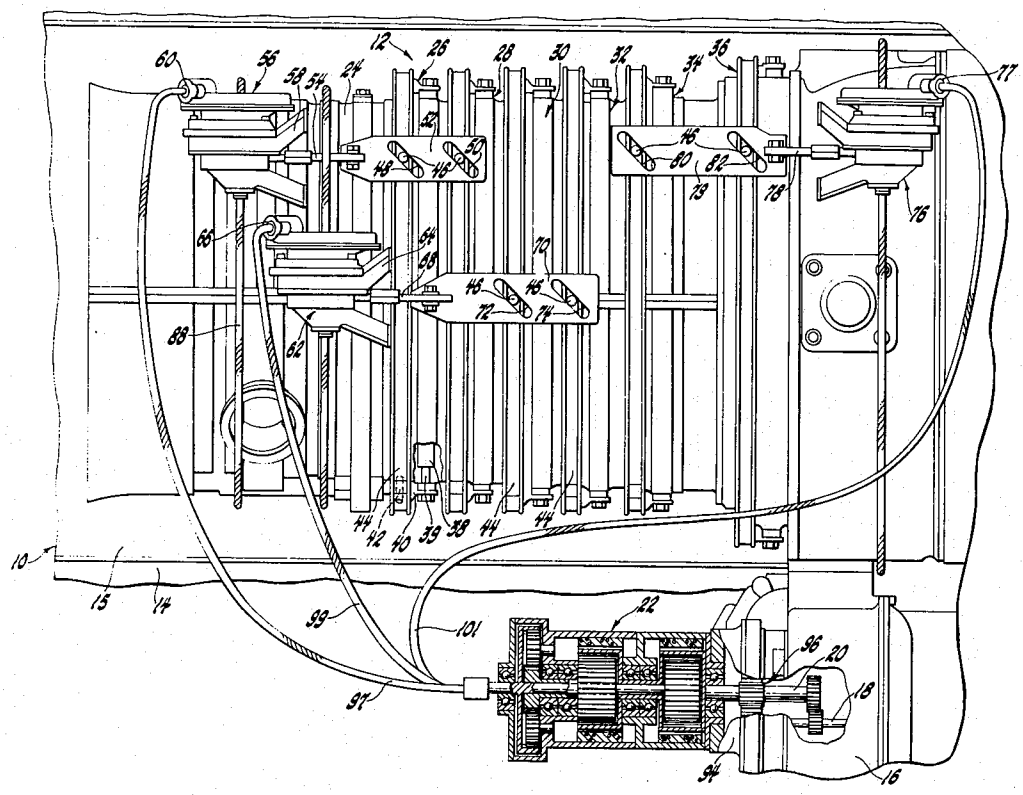
*Attorney, Agent, or Firm*—J. C. Evans



[57] **ABSTRACT**

A turbojet engine compressor has a plurality of variable stator vane stages, each stage operated by rotary to linear actuators driven by a high speed drive cable connected to a single drive unit driven by the turbojet engine at high input speeds. Each drive cable is connected to two back-to-back hysteresis clutches which are selectively activated by one or the other of two field coils energized to determine the direction of cable rotation and stator vane adjustment.


**2 Claims, 8 Drawing Figures**



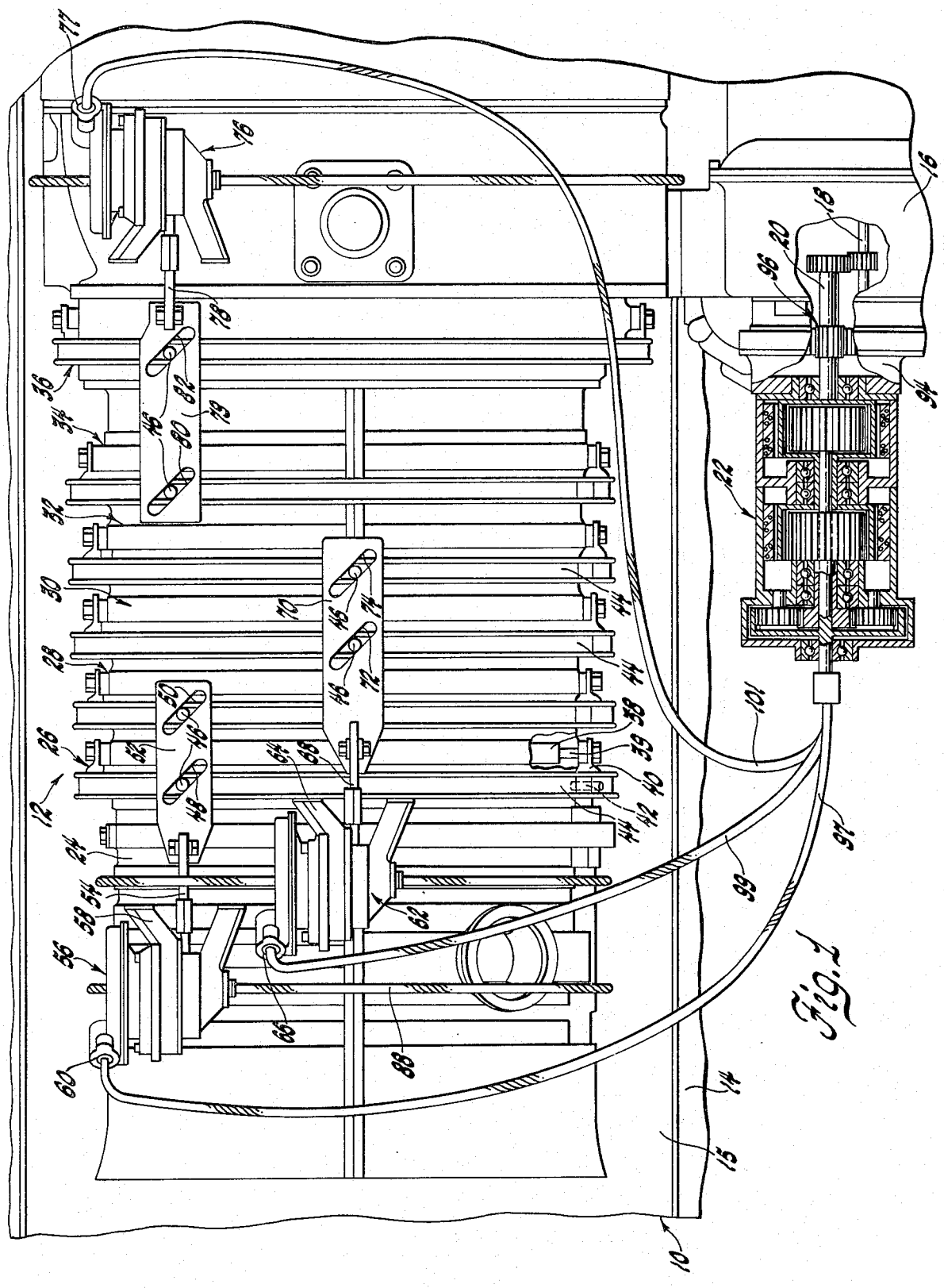

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Fig. 1

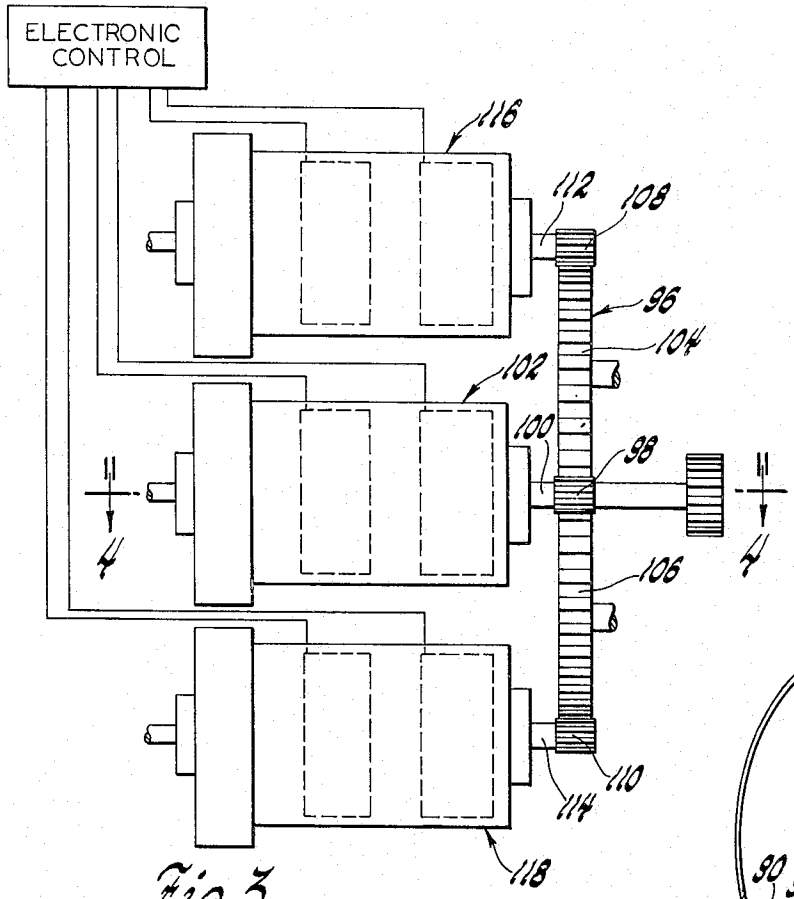



Fig. 3

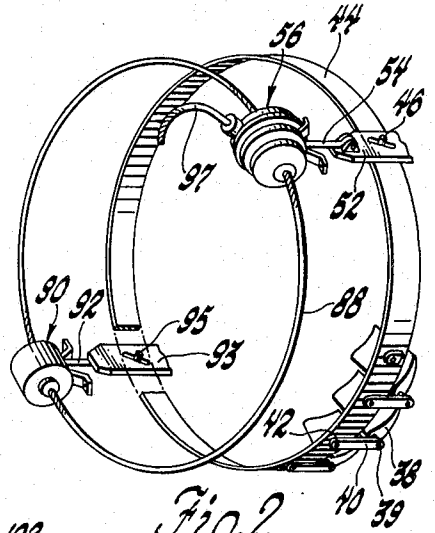



Fig. 2

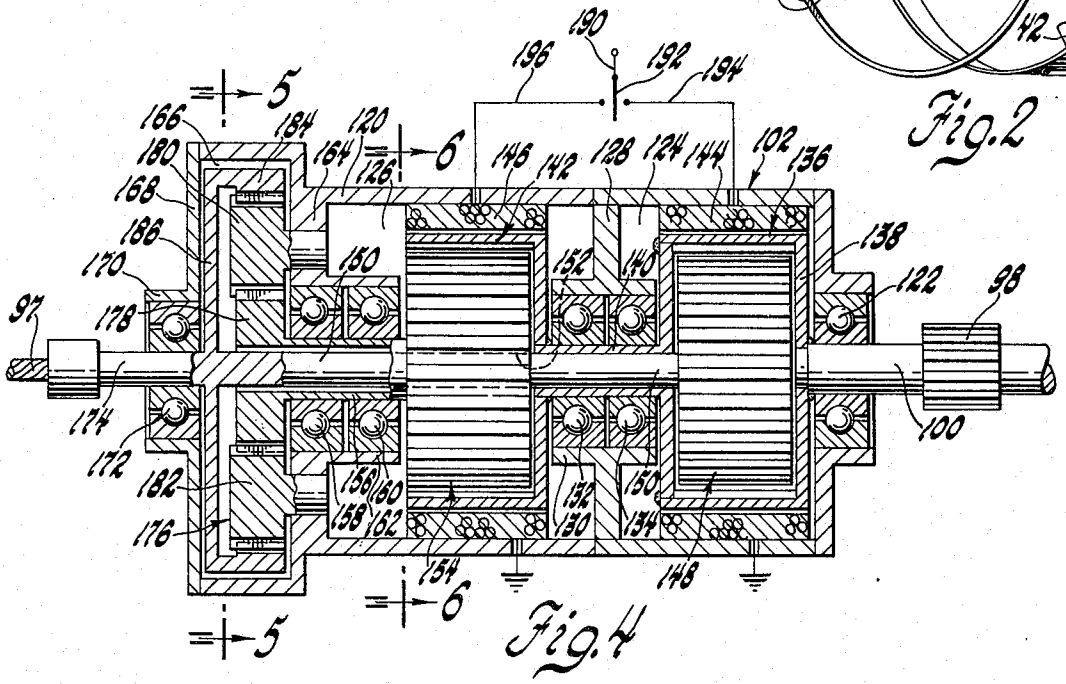



Fig. 4

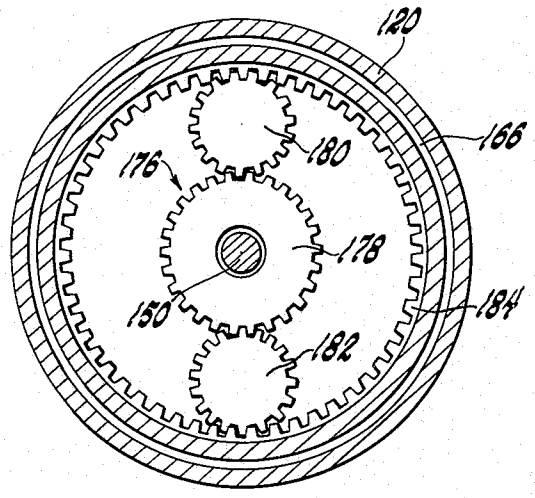



Fig. 5

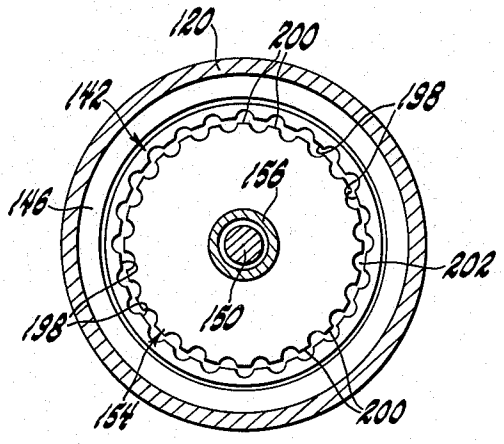



Fig. 6

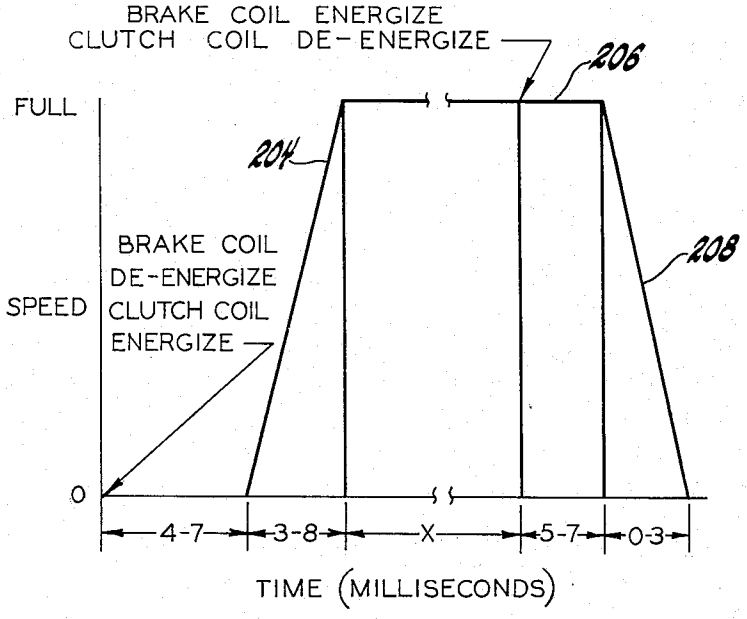



Fig. 7

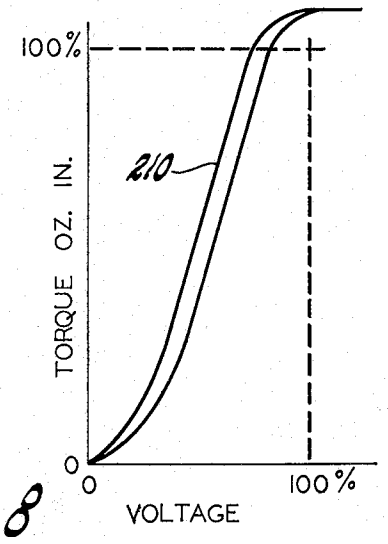



Fig. 8

# VANE ACTUATION SYSTEM

This invention relates to systems for actuating variable vanes in a turbine engine to control the operating condition of the engine and more particularly to an actuation system for operating each vane stage of such an engine individually.

Actuation of variable stator vanes in a multi-stage compressor has been accomplished by means including a movable cam plate which ties a number of variable stator vane stages together in a fixed relationship. The cam plate is moved by a fuel operated piston type actuator which uses fuel from the main engine fuel system as the power source. An example of a system of this type is more specifically disclosed in U.S. Pat. No. 2,931,168 issued Apr. 5, 1960, to W. C. Alexander et al.

In certain applications it is desirable to operate each variable vane stage individually. In such cases, multiple cam plate operators are required and it is necessary to include individual actuators with separate servo controls of the type set forth in the above-mentioned Alexander et al. patent to each variable stage. Such actuator packages, including the servo mechanism components therein, are relatively large so as to preclude locating a large number of such actuators between a surrounding bypass duct and the compressor case of an engine. Furthermore, the array of tubing for fluid operated primary and slave actuators requires elaborate plumbing and multiple sealed joints and couplings.

Additionally, the performance of such systems is difficult to control by use of electronic control signals applied to hydraulic servo components to control the high pressure fuel supply which is the power source for the fluid actuators to individual variable stator vane stages on the compressor.

Accordingly, it is an object of the present invention to improve the control of multiple stage variable stator vanes of a compressor in a jet engine by the provision therein of a single drive unit that is responsive to an electronic control signal to operate electrically energized fluid gap clutch means to directly couple a high speed drive from the engine through flexible drive cables to individual actuators on the engine compressor for individually operating each of a plurality of variable stator vane stages therein.

Yet another object of the present invention is to provide a low torque, high speed actuator system for quickly responding to electrical control signals to produce a fast adjustment of variable vanes in a turbine engine and to produce good resolution of vane position by the provision of a control signal responsive drive unit having fluid gap clutch means therein and flexible drive output cables activated to produce forward and reverse control of individual rotary to linear actuators coupled to the variable vanes.

Still another object of the present invention is to provide a direct drive geometry for actuating individual variable stator vane stages of a compressor by separate actuators thereon including means for converting rotary movement to linear movement and wherein a rotary input component of the vane actuators is connected to a high speed drive cable in turn coupled through an electrically energized fluid gap clutch to a turbine powered high speed output shaft and wherein the clutch includes a pair of components selectively energizable to produce reverse rotation of the flexible drive cable with a resultant reversal of control of the vanes in response to an electrical signal applied to field coils in the clutch.



Still another object of the present invention is to improve the control of variable vane stages in a turbine engine by the provision of a single drive unit having its input connected to the accessory drive of a turbine unit in the engine and a plurality of high speed output drive cables each driven from the input shaft under the control of a pair of back-to-back fluid gap clutches; each clutch having a field coil under the control of signal means responsive to engine operating conditions and wherein each of the high speed drive cables is in turn connected to a rotary to linear drive converter having a slotted cam plate connected thereto drivingly coupled to radially outwardly directed drive pins on the periphery of a control ring arcuately adjusted for positioning a plurality of drive cranks for adjusting individual vanes in a variable vane stage of the engine.

Further objects and advantages of the present invention will be apparent from the following description, reference being had to the accompanying drawings wherein a preferred embodiment of the present invention is clearly shown.

FIG. 1 is a diagrammatic view of a direct drive system for operating a plurality of variable vane stages in the compressor of a turbine engine from a single drive unit in accordance with the present invention;

FIG. 2 is a diagrammatically shown cable drive network at stages of the engine in FIG. 1;

FIG. 3 is a top elevational view of the single drive unit in FIG. 1;

FIG. 4 is an enlarged, longitudinal sectional view taken along the line 4—4 of FIG. 3 looking in the direction of the arrows;

FIG. 5 is a vertical sectional view taken along the line 5—5 of FIG. 4 looking in the direction of the arrows;

FIG. 6 is a vertical sectional view taken along the line 6—6 of FIG. 4 looking in the direction of the arrows;

FIG. 7 is a graph showing the typical response time for the clutch, brake action of the actuator in FIG. 4, and

FIG. 8 is a graph of a typical torque versus field current curve for the clutch components of the actuator of FIG. 4.

Referring now to the drawings, in FIG. 1 a turbojet engine 10 is illustrated including a multiple stage compressor 12 with a surrounding bypass duct 14 in spaced relationship to the outer periphery of compressor 12 to define an annular space 15 therearound. An accessory drive box 16 on the engine 10 has an output shaft 18 connected to an input shaft 20 of a power transfer unit 22 constructed in accordance with the present invention.

In the illustrated arrangement, the compressor 12 includes an outer case 24 having a plurality of variable vane stages 26, 28, 30, 32, 34 and 36 formed therearound. Each of the vane stages is made up of a plurality of vanes formed in a ring at each stage. One ring vane 38 is shown in stage 26 at a broken section in FIG. 1 and a complete ring of vanes 38 is shown diagrammatically in FIG. 2. Each such vane 38 has a radially outwardly directed shaft 39 pivotally supported with respect to the outer case 24. It in turn is connected to one end of a crank arm 40 having the opposite end thereof connected by means of a pivot pin 42 to a control ring 44 arranged radially outwardly of and circumferentially of case 24. There is a like control ring for

each of the vane stages 26 through 36 coupled to like vanes. Each control ring 44 includes a radially outward directed drive pin 46. The drive pins 46 at the vane stages 26, 28 are located in inclined slots 48, 50, respectively, in a reciprocating cam plate 52 connected to a reciprocating linear output shaft 54 of a rotary to a primary linear converter/actuator 56 having a connector flange 58 thereon fixedly secured to the outer case 24 and a rotary input shaft 60. A suitable actuator 56 for practicing the invention is available from The Bendix Corporation, part number 440489.

In accordance with certain principles of the present invention, given pairs of the vane stages 26 through 36 are operated individually. Hence a second rotary to linear converter actuator 62, like actuator 56, is connected by means of a flange 64 thereon to the outer case at a point axially spaced from the actuator 56. It has a rotary input shaft 66 and a linear output shaft 68 coupled to one end of a cam plate 70 with inclined slots 72, 74 therein that fit, respectively, over drive pins 46 on the control rings 44 of the vane stages 30, 32. Additional variability in the compressor stages is provided by a third rotary to linear motion converter actuator 76 having a rotary input shaft 77 and a linear output shaft 78 like previously described actuators 56, 62 coupled to a cam plate 79 with inclined slots 80, 82 therein into which the drive pin 46 of the vane stages 34 and a drive pin 46 of a larger diameter vane stage 36 on the outlet of the compressor fit, respectively. As seen in FIG. 3, actuator 56 has a cable loop 88 which transfers input drive to a second actuator 90 of the rotary to linear type whose linear output arm 92 is connected to rings 44 of stages 26, 28 through a slotted cam plate 93 like plate 52 which fits over pins 95 on each control ring 44 diametrically opposed to pins 46 to balance the ring input reaction. Actuators 62, 76 have like cable loops, each of which produces drive redundancy from the primary actuators 56, 62, 76 to the actuators 90 which constitute slave actuators.

Each of the actuators 56, 62, 76 are driven by high speed drive cables 97, 99, 101 coupled to rotary input shafts 60, 66, 77, respectively.

The power transfer unit 22 is illustrated in FIG. 2 as including a housing 94 that encloses a gear set 96 including a central pinion gear 98 coupled by a shaft 100 to a first fluid gap clutch unit 102. The gear set 96 further includes a pair of idle gears 104, 106 for transferring power to pinion gears 108, 110 respectively. The pinion gears 108, 110 are connected by shafts 112, 114, respectively, to second and third fluid gap clutch units 116, 118. The following description of clutch unit 102 applies equally to the actuators 116, 118 which have corresponding components.

As more particularly set forth in FIG. 4, hysteresis clutch unit 102 includes an outer housing 120 having input shaft bearing 122 at one end thereof and a pair of spaced cavities 124, 126 separated by a wall 128 having an annular flange 130 thereon to support a pair of bearings 132, 134 between the cavities 124, 126. An input rotor 136 has an end wall 138 connected to the input shaft 100 to the hysteresis clutch actuator 102. The input rotor is in the form of a hollow drum including an axial extension 140 thereon extending through bearings 132, 134 in supported relationship therewith. It is connected to a second input rotor 142 in the form of an open ended drum that is radially outwardly located within the cavity 126. The rotors 136, 142 are associated with direct current clutch field coil 144, 146, respectively, located in the cavities 124, 126 in surrounding relationship to the outer periphery of the rotors 136, 142.

An output rotor 148 is located within the hollow interior of the input rotor 136 and includes an axial shaft 150 extending therefrom through the axial extension 140 on the input rotor 136. The shaft 150 also extends through a center bore 152 in a second output rotor 154 located within the hollow input rotor 142. The output rotor 154 includes a tubular extension 156 extending axially from the output rotor 154 in telescoped relationship with the shaft 150. The tubular extension 156 is rotatably supported by a pair of bearings 158, 160 carried by a support ring 162 formed as part of a divider wall 164 between the cavity 126 and a gear cavity 166 at the outboard end of the outer housing 120. The gear cavity 166 is also formed by an end wall 168 on the housing 120 including an annular outboard flange 170 in which is supported a bearing 172 to rotatably support the outboard end 174 of the shaft 150.

A reversing gear set 176 is located in the cavity 166. It includes a sun gear 178 formed on the end of the tubular extension 156 and a pair of planetary gears 180, 182 that are in meshed relationship with a ring gear 184 connected to a gear carriage 186 that is connected to the outboard shaft end 174. The flexible drive cable 97 has its input end connected to shaft end 174. Cables 99, 101 are likewise connected to the outputs of actuators 118, 116, respectively.

The clutch unit 102 thus includes two back-to-back hysteresis clutches which are activated whenever one or the other of the two field coils 144, 146 are energized. The hysteresis clutches are available from American Precision Industries, Inc., Delevan Division as Model FH4C–30–24.

In the illustrated arrangement a simplified control configuration is illustrated including an input line 190 connected to a suitable signal source having a double throw, double pole switch 192 therein for selectively connecting the input line to a first conductor 194 that is connected across the field coil 144 to ground and to a conductor 196 connected across the field coil 146 to ground. When the switch 192 is in a vane advance control position, it will complete a circuit from the power source through conductor 194 so as to energize the coil 144. As best seen in FIG. 6, such energization will produce an eddy current flow between a plurality of radially outwardly directed poles 198 spaced circumferentially around the inner periphery of each of the input rotors 136, 148 and a like set of circumferentially spaced radially inwardly formed poles 200 in the outer periphery of the output rotors 148, 154. An annular air gap 202 is defined between poles 198, 200.

The energization of the coil 144 will produce a magnetic coupling of the input rotor 136 to the output rotor 148 so as to drivingly connect the drive input shaft 100 or like shafts of adjacent one of the actuators to produce a speed increase from zero to full speed of operation within a period of from seven to fifteen milliseconds as reflected by a representatively illustrated acceleration curve 204 in FIG. 7. This full speed drive will be transferred by the cable drives 97, 99, 101 to the actuators 56, 62, 76, and loop connected slave actuators 90, respectively, to quickly position the individual stator vanes in the compressor 12 to a desired control position following which time the switch 192 is opened to

produce a brake action which is initiated at point **206** in the curve of FIG. **7** with a retardation of the output shaft speed from the actuators **102, 116, 118** occurring from 5 to 10 milliseconds later as reflected by the deceleration curve **208** in FIG. **6.**

As shown in FIG. **8** each of the hysteresis clutches has a typical torque versus field current curve that is illustrated by the hysteresis curve **210** showing that 100% torque is attained at each of the clutches when the switch **192** is applied thereacross to produce an application of 100% of voltage to the coil. Modulation of control speed is produced by modulation of voltage applied across the field coils **144, 146.**

In order to obtain reverse operation of the control mechanism the switch **192** is positioned to energize the coil **146** to produce a rapid acceleration of the output rotor **154** which is connected through the reversing gear set **176** to the output shaft ends **174** so as to produce reverse rotation of the shaft with a resultant return of the adjustable stator vanes **38** to an originally set control position.

The clutch rotors and air gap **202** are in essence one form of a fluid gap clutch which finds a counterpart in magnetic particle clutches of the type having an input rotor and output rotor spaced apart and coupled by a fluid gap filled with a viscous fluid filled with magnetic particles. When a field coil is energized the magnetic particles align themselves in the rotor gap to produce a torque transfer couple. Use of such back-to-back fluid gap, electrically energized clutches produces a low torque, high speed system for fast response and good resolution of vane positioning. The illustrated compressor stages are merely representative of turbine engine stages which required variability, it being understood that the invention has equal application to a turbine stage with variable stator vanes controlling flow to a turbine wheel.

While a simple switching control is illustrated it is understood that the actuator system is suitable for use with electronic control systems sensing a suitable engine parameters to produce a selective energization of the field coils of each of the hysteresis clutch actuators in order to produce an engine control that fully reflects changes in engine operating conditions such as rotor speed, compressor inlet temperature, discharge pressure and turbine discharge temperature.

By virtue of the use of back-to-back fluid gap clutches in the aforesaid cable drive system, a transfer of torque to the rotary to linear converter actuator units **56, 62** and **76** is maintained virtually independent of speed. The torque characteristics of the drive system are infinitely adjustable and repeatable so as to produce a smooth control operation which is unusually stable.

The transfer of power from the turbine of the aforedescribed system also represents a low power consumption from the engine.

The system is further characterized by the elimination of physical contact between the moving input rotor and output rotor parts of the clutch actuator which eliminates the generation of metallic dust during operation of the drive transfer unit **22.** Furthermore, the use of a hysteresis clutch produces noiseless operation.

The system further is characterized by being easily interfaced with an electronic control. It defines an actuator locatable within a limited space. It eliminates the need for direct fuel supply operated actuators. It does not interfere with other control loop characteristics as might exist where the control is dependent on operating characteristics of an engine fuel supply.

While the embodiments of the present invention, as herein disclosed, constitute a preferred form, it is to be understood that other forms might be adopted.

What is claimed is:

**1.** A high speed actuator system for controlling a row of individually rotatable crank operated vanes coupled to a control ring on the exterior of a turbine engine comprising; a rotary to linear drive actuator having a rotary input shaft and a reciprocating output shaft, means for coupling said reciprocating output shaft to the control ring for producing a predetermined arcuate movement of the control ring in response to reciprocation of the output shaft of the actuator, a drive transfer unit including a pair of electrically energized fluid gap clutch means, each of said clutch means including an input rotor and an output rotor, an energizable field coil for coupling said input and output rotors in accordance with energization thereof, an input shaft to said transfer unit connected to both of said input rotors, a drive cable having one end thereof connected to the rotary input shaft of said actuator and the other end thereof to one of said output rotors, reversing gear means connecting the other of said output rotors to said drive cable, means for selectively energizing one or the other of said energizable field coils to produce reverse rotation of said drive cable thereby to produce reverse reciprocation of said actuator output shaft, thereby to control arcuate movement of the control ring to produce a controlled adjustment of the angular position of each of the crank operated vanes.

**2.** In a variable stator turbo jet engine of a type having a plurality of stages of compressor stator vanes each operated by means of a control ring having crank means thereon coupled to each of the compressor stator vanes and a drive pin on the ring, the improvement comprising a turbine output shaft, a plurality of individual rotary to linear actuator means, means for coupling the output of each of said actuator means to selected ones of the control rings for controlling stages of compressor stator vanes individually, a power transfer unit having an input drive shaft thereto coupled to said turbine output shaft, a plurality of pairs of back-to-back fluid gap clutches each having an input shaft thereto, gear means for transferring power from the input shaft of said power transfer unit to each of said clutch input shafts, each of said fluid gap back-to-back clutches having an input rotor and an output rotor with a fluid gap therebetween, means connecting each of said input rotors to the input shaft of each clutch, energizable coil means for magnetically coupling said input rotors to said output rotors, an output shaft from each of said pair of clutches coupled directly to one of said output rotors, reversing gear means coupling the other of said output rotors to said output drive shaft, a flexible drive cable having one end thereof to said output shaft and the opposite end thereof connected to the input of one of said rotary to linear converters, and means for selectively energizing each of said field coils of the plurality of pairs of back-to-back fluid gap clutches to produce individual control of a plurality of individual compressor stator vane stages.

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