

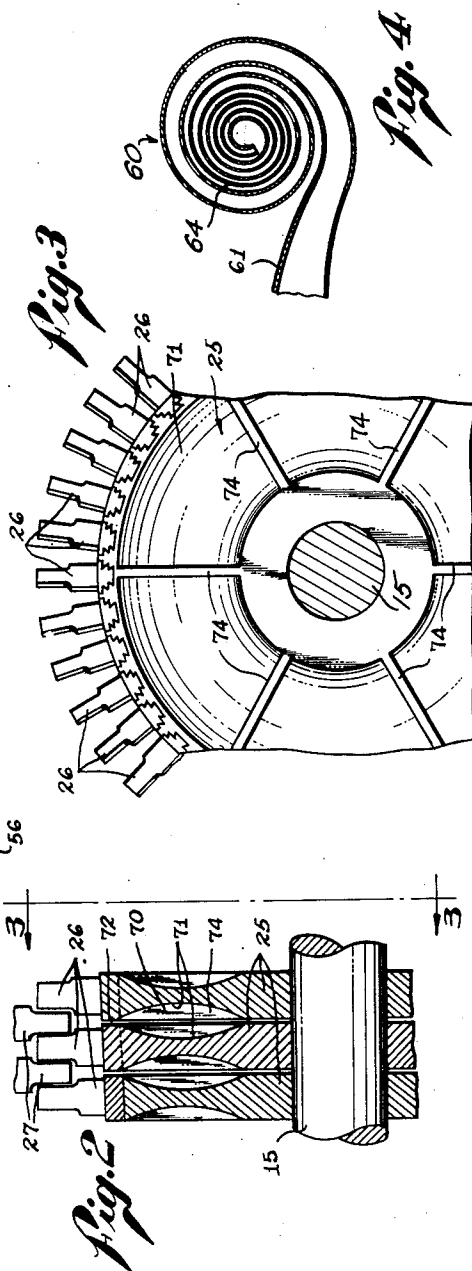
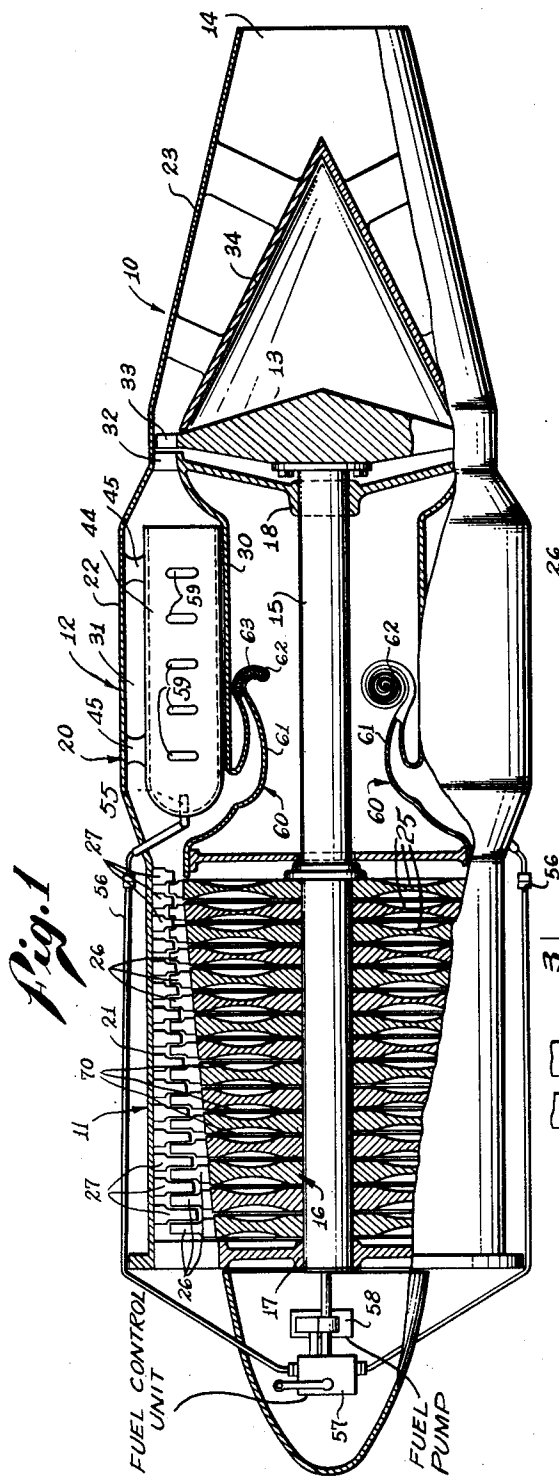
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GAS TURBINE WITH ACOUSTIC SURGE CONTROL

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GAS TURBINE WITH ACOUSTIC SURGE CONTROL

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This invention relates generally to the compressors of gas turbines, particularly, though not limited to, the compressors of aircraft turbo-jet engines. It will be evident, for example, that the invention is equally applicable to turbines for generating shaft horsepower.

Both centrifugal and axial flow compressors for modern turbo-jets suffer from "surging," which is an unstable flow condition exhibiting pulsations of flow which are very detrimental to operation. Ways have been found for avoiding the "surge region" with turbo-jets operating at comparatively low compression ratios, but with the present design trend toward higher compression ratios, the problem of surging has become most serious, and no satisfactory solution has to my knowledge been heretofore proposed.

The general object of the present invention is accordingly the provision of means for controlling or suppressing the phenomena of surging in turbo-jet compressors.

The problem of surging has been investigated by others, and certain significant facts have been authoritatively reported. For example, R. O. Bullock and H. B. Finger, of Lewis Flight Propulsion Laboratory, NACA, presented a paper at the SAE National Aeronautic Meeting, New York, April 16, 1951, in which was reported their finding that the pulsations in centrifugal superchargers occur at a frequency which is high for small volume, and low for large volume, and their further finding that, in general, the results of the supercharger investigations were apparently applicable to axial flow compressors. See SAE Quarterly Transactions, April 1952.

These investigators did not, however, draw the conclusion that their experimental data evidenced an acoustic resonant phenomenon, with the compressor enclosure functioning as a resonant acoustic cavity. Neither did they point out that such "surging" might be controlled by use of acoustic remedies for cavity resonance, which is the basic concept of the present invention.

The present invention therefore starts with the assumption that a centrifugal or axial flow compressor is capable of behaving as a resonant acoustic cavity. An acoustic cavity may be defined for present purposes as a space, chamber or conduit having reflective boundaries of such nature that an acoustic standing wave may be set up therein at one or more resonant frequencies, or such that it behaves as a Helmholtz resonator. It is known generally that gas driven through a resonant cavity is capable of resonating the cavity. I have found that the flow becomes unsteady, and surges at a resonant frequency of the cavity, the surge frequency varying inversely with the size of the cavity. Now, in a compressor, the rate of energy delivery of a compressor vane or blade to the air stream is a function of air stream velocity. If air stream velocity pulsates, the energy delivery from blades to air stream must pulsate accordingly. The back reaction of the air stream on the blades is thus a fluctuating factor, whose magnitude is proportional to the periodic deviation of flow velocity from its median

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value. The energy delivery from the blades accordingly has a fluctuating component, of periodicity equal to the acoustic resonant frequency of the compressor cavity. The effect is regenerative, in that the periodic flow pulsations are initiated in the first instance by the mere gas flow through the cavity, but, upon inception of periodic reaction with the blades, are greatly augmented and then maintained at high amplitude by the periodic character of the blade-to-air stream energy delivery. The discovery of this phenomenon is a very important part of my invention. The phenomenon is not unlike the drive of a violin string by a bow, where the vibration frequency of the string is set by the resonant frequency of the driven member (string) while the driver (bow) delivers its energy with a fluctuating energy flow, in step with the vibrations of the string. The string vibrates at its own resonant frequency as a result of being disturbed, the energy delivery from the constantly moving bow to the string is periodic because of the vibratory motion of the string, and the amplitude of the vibrations varies with the power of the bow. In similar fashion, the air surges at the resonant frequency of the cavity as a result of some disturbance; and because of this surging, the energy flow from blades to air stream is periodic, and the surge amplitude depends upon the power with which the blades are driven.

Surging, therefore, is a phenomenon initiated by resonant characteristics of the compressor cavity, and carried to high amplitude by the resultant periodic character of the blade-to-airstream energy flow.

Such periodic surging is not only detrimental to the thermodynamic cycle, and the operational characteristics of the turbo-jet as a whole, but is also detrimental to the blades themselves. It has been explained how the periodic energy flow from the blades to the air stream results in a periodic fluctuation of back reaction on the blades. This periodic fluctuation of reaction may be either at the fundamental resonant frequency of the compressor cavity, or at some higher harmonic. The periodic fluctuation of reaction tends toward vibration of the blades, and particularly if some harmonic frequency should coincide with a natural resonant frequency of the blade, the blade may be set into very serious vibration. Any vibrational tendency at the blades is of course highly undesirable, not only because it would be further conducive to flow instability, but because of blade fatigue.

The present invention provides acoustic means for causing the gas oscillations in the compressor cavity to subside. This is accomplished by using, in combination with the compressor cavity, a wave or gas vibration attenuator. This may take the form of a resonant absorber, tuned to the natural resonant frequency of the compressor cavity and having the ability to destroy or materially suppress the resonant peaking characteristics of the chamber. In the language of those versed in the acoustic art, my vibration attenuator materially reduces the "Q" of the compressor cavity, the factor Q being understood to denote the ratio of energy stored to energy dissipated per half-cycle of the vibration which is taking place. It does this by dissipating a large amount of the vibrational energy on each half-cycle, and in consequence, the amplitude of any gas vibration is kept within harmless limits.

An important object of the invention is to reduce the acoustic "Q" of the compressor cavity and thereby prevent or reduce the tendency of the D. C. flow compressor to convert a portion of its energy into A. C. flow.

It is accordingly not the intention merely to dissipate acoustic energy after it is generated. Rather, it is the intention to break the chain of mutual cooperation between cavity resonance and unsteady or periodic energy delivery by the blades, so that the cavity cannot induce the blades

to generate sound in the first place. This I accomplish by reducing the acoustic resonance of the cavity.

In the drawings,

Figure 1 is a longitudinal section of a turbo-jet engine embodying the invention;

Figure 2 is an enlarged detail of Figure 1;

Figure 3 is a view of the compressor rotor in the direction of arrows 3—3 in Fig. 2; and

Figure 4 is an enlarged detail section of an attenuator.

In the drawings, numeral 10 designates generally a somewhat diagrammatically illustrated turbo-jet engine, which is illustrative of the invention. This engine has an axial flow compressor 11, a burner section 12, a bladed turbine rotor 13, and a jet discharge orifice 14, turbine rotor 13 being on shaft 15 which carries and drives the bladed rotor 16 of compressor 11. The compressor and turbine rotors turn as a unit, supported by bearings 17 and 18.

The turbine has an external shell 20, including a forward cylindrical section 21 open at the forward end for air intake to compressor 11, an enlarged intermediate section 22 around the burner section, and a converging rearward section 23 terminating in orifice 14.

The compressor rotor is, in general form, of a conventional type having axially spaced disks 25 shrink-fitted onto shaft 15 and carrying at their peripheries suitably anchored compressor blades 26. These compressor blades 26 turn between stationary compressor blades 27 suitably supported inside the compressor section 21 of the shell. The annular air duct between the shell wall and the peripheries of the disks 25 converges toward the burner section, in the usual manner, as clearly illustrated. The spaces between the disks 25, excepting for certain presently described acoustic attenuator formations, are relatively small, e. g., $\frac{1}{16}$ " from blade to blade.

The burner section 12 includes an inner annular burner wall 30, forming an annular combustion chamber space or duct 31, and provided with any suitable means of support (not shown) to the outer shell. Within this annular space 31 are burners 44 comprising cylindrical tubes closed at their forward ends and open toward the rear, these being suitably supported, as by means of webs 45. It will be seen that the air discharged rearwardly from the compressor enters the forward end of duct 31, and that the airflow, including products of combustion from the burner tubes, leaves the rearward end of the duct 31 through stationary turbine blades 32 to impinge on turbine rotor blades 33; thence flowing rearwardly through the space between inner tail cone 34 and the convergent section of the shell, to be finally discharged at orifice 14.

A fuel nozzle 55 discharges fuel into the head end of each burner 44, the fuel being supplied by fuel line 56 from fuel control unit 57 and fuel pump 58. Air enters the burners 44 through intake ports 59. Combustion of the fuel and air mixture within burner 44 results in the discharge of high pressure combustion gases in a rearward direction to drive the turbine rotor, and eventually to be jet-discharged at 14.

Such a turbine is often subject to severe surging, as mentioned hereinabove. The surge frequency is relatively low, typically within the range of 10–1000 C. P. S.

My studies have shown that surging in such turbines can take place in various modes, either within the compressor as a discrete acoustic unit, or within the compressor and combustion sections acoustically coupled together, i. e., acting conjointly as a unitary acoustic device.

Considering the first mentioned mode, the shell portion 21 and disk and blade surfaces form an acoustic cavity, open at both ends, and this cavity possesses, among other probabilities, a fundamental resonant frequency for a longitudinal half-wave mode of gas oscillation. Such a mode is characterized by the presence of a half-wave length standing wave, with velocity antinodes (regions of maximum gas oscillation) at both ends, and a pressure antinode (region of maximum pressure amplitude oscilla-

tion) at some position intermediate the two ends. Along with this fundamental, harmonics may develop, giving additional pressure antinodes along the length of the compressor.

There are two possible principal modes with the compressor and burner sections coupled together or acting as a unitary acoustic device. The first of these is that characteristic of a Helmholtz cavity, where the neck of the cavity is formed by the compressor duct, and the body of the cavity is formed by the somewhat enlarged burner duct. In this case, the gas oscillates back and forth through the compressor duct, and the gas body within the burner duct experiences a substantial pressure cycle. The velocity and pressure oscillations occur at the resonant frequency of the compressor and burner ducts considered as the neck and body, respectively, of a Helmholtz resonator. Finally, as the second mode with the compressor and burner sections acting together as a single or composite resonant cavity, a low frequency mode can occur with the compressor and burner sections forming a long half-wave length conduit, with velocity antinodes at the two ends, and a pressure antinode near the junction of the compressor and burner sections.

The invention provides acoustic wave attenuator means in combination with these acoustic cavities or ducts formed by the compressor and by the compressor coupled with the burner section.

In general, where a distinct wave pattern is either ascertainable or probable, the most effective location for such acoustic means is in the region of pressure antinodes of the gas oscillation modes or patterns which are to be controlled. Accordingly, for the purpose of controlling the two modes which involves the compressor and burner sections acoustically coupled to one another, I here show, as illustrative of the invention, a plurality of attenuators 60, coupled to the system at the pressure antinode regions at or near the junction of the burner section with the compressor section. These attenuators, being fairly closely coupled to the compressor section, may also effect a degree of control over acoustic wave patterns set up in the compressor section as a unit.

In the illustrative embodiment, the attenuators comprise horns 61, preferably approximately of exponential taper, with attenuative terminations 62. The taper is calculated, according to well known design procedure, to accept wave frequencies of the wave or waves which are to be attenuated, and to convey such waves, without substantial reflection back, to the attenuative termination device 62, where the waves are dissipated by conversion of their wave energy into heat. The attenuative termination devices may consist of a packing 63 of some such fibrous sound wave absorptive material as fiber glass, located within the constricted throat of the horn. Preferably, however, and particularly for any cases in which the wave amplitudes are sufficiently high to be destructive of the packing material, I prefer to employ a long attenuator tube 64 connected to the throat of the horn, and preferably coiled, as shown in Fig. 4. This attenuator tube, if provided with certain rather critical ranges of dimensions, will possess strong attenuation characteristics without being of undue or unwieldy length. In general, the requirements are met if the horn has an area ratio between mouth and throat of at least 20 to 1, and of a length at least one-quarter wave length, and the tube has a lateral dimension of about .06 wave length and a length of two wave lengths.

For principal control of the sound wave pattern within the compressor section, I form resonant absorbers 70 between the disks 25 of the compressor. In the illustrative embodiment, these absorbers 70 are in the nature of Helmholtz resonators. To form them, opposite faces of the disks 25 are provided with annular channels or concavities 71. These concavities form the bodies or chambers of the Helmholtz resonant absorbers 70. The constricted necks of the absorbers 70 are supplied by the

narrow spaces 72 between the opposed lateral surfaces of the rim portions of the disks. Preferably, to reduce circular motion of the gases between the disks about the central axis, the disks are formed, within the concavities 71, with radial baffles 74. This formation reduces any tendency for setting up of circular modes of gas oscillation within the "chamber" of the Helmholtz resonator.

The dimensions of the Helmholtz resonators so formed are adjusted in accordance with well known acoustical practices to be resonant or responsive to the wave frequencies prevalent within the compressor section. If there are more than one wave frequency to be attenuated, some of the Helmholtz resonators can be adjusted to each frequency to be controlled.

It was previously mentioned that the most effective location for a sound wave attenuator to be coupled to an acoustic cavity or duct for suppression of a wave pattern therein is the region identified with a pressure antinode of the wave pattern. As also previously mentioned, the characteristic fundamental sound wave vibration mode or pattern has velocity antinodes at the two ends of the compressor duct, and a pressure antinode at an intermediate region of that duct. And if harmonics are present, additional pressure antinodes will appear at additional locations along the compressor. It will be seen that by use of the row of Helmholtz resonant absorbers provided by the invention, extending the full length of the axial flow compressor, there is certain to be one or more of such absorbers in the intermediate region of each pressure antinode within the compressor, even though the precise location of such pressure antinodes should not be exactly known.

The Helmholtz resonant absorbers as described function to very materially suppress or attenuate the resonant wave frequencies developed within the compressor section. In many cases, the vibration mode developed within the compressor section as an individual acoustic unit will be the chief or only offender. In this case, the resonant absorbers employed within the compressor section will fully suffice to correct the engine.

In cases in which, to any serious extent, other modes are present, particularly modes involving the burner section, additional acoustic attenuators, such as those indicated at 62, may be employed.

With the use of acoustic wave attenuators, as described, properly tuned or shaped to be responsive to the offending wave frequencies, the "surging" of a gas turbine is forced to subside, and desired stability thereby attained.

The present application is a continuation-in-part of my prior application entitled Control of Combustion Instability in Jet Engines, filed June 16, 1951, under Serial No. 231,954.

I claim:

1. A gas turbine having an axial flow compressor comprising a bladed stator and a bladed rotor between which is an annular axial flow duct within which gas surging tends to occur at a predetermined wave frequency, said rotor comprising an axially disposed shaft and a plurality of axially spaced disks mounted thereon, said disks fitted at their peripheries with blades for coaction with the blades of the stator, said disks having configurations in opposed side surfaces thereof forming Helmholtz resonators having enlarged chambers and constricted necks, said necks communicating with said axial flow duct, and said Helmholtz resonators having a resonant response to said predetermined wave frequency.

2. A gas turbine having an axial flow compressor comprising a bladed stator and a bladed rotor between which is an annular axial flow duct within which gas surging tends to occur at a predetermined wave frequency, said rotor comprising an axially disposed shaft and a plurality of axially spaced disks mounted thereon, said disks fitted at their peripheries with blades for coaction with the blades of the stator, opposed faces of adjacent disks having opposed annular channels sunk therein to form the bodies of Helmholtz resonant absorbers having an attenuative frequency response for said predetermined frequency, said disks having opposed close spaced annular surface portions outside said channels to form the constricted necks of said Helmholtz absorbers, and said Helmholtz absorbers having a resonant response to said predetermined wave frequency.

3. The subject matter of claim 2, including also a plurality of angularly spaced radially disposed baffles formed on the confronting surfaces of the disks within said annular channels.

4. A gas turbine having an air flow passage including a bladed compressor section, in which passage a periodic surge flow phenomena tends to occur at a characteristic sound wave frequency which sound wave frequency is of low frequency range relative to blade hum and which sound wave frequency is a function of cavity resonance gas vibrations reacting upon the compressor blade aerodynamic drive characteristics under conditions of high compression and relatively low flow such as occurs near aerodynamic stall of the blades during high load or accelerating conditions in gas turbines, and acoustic wave attenuation means so constructed and arranged as to have an attenuative frequency response for said sound wave stall surge frequency, which attenuation means is acoustically coupled to said air flow passage.

5. The subject matter of claim 4, wherein said attenuation means is coupled to said air flow passage in the region of said bladed compressor section.

6. The subject matter of claim 4, wherein said air flow passage has a burner section following said compressor section, and wherein said attenuation means is coupled to said passage in the region of the junction between said compressor and burner sections.

7. The subject matter of claim 4, wherein said air flow passage has a burner section following said compressor section, and wherein said attenuation means is coupled to said passage within said burner section, but in acoustic coupling relationship to said compressor section via the junction between said compressor and burner sections.

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