

July 13, 1965

J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 1

FIG. 1

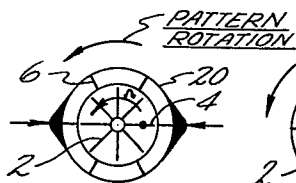


FIG. 2

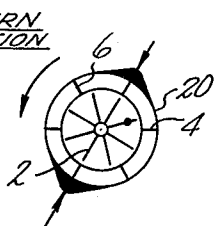


FIG. 3

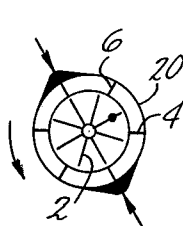


FIG. 4

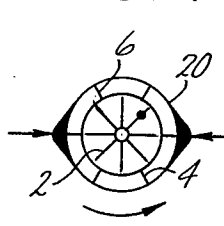


FIG. 5

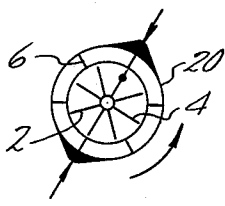


FIG. 6

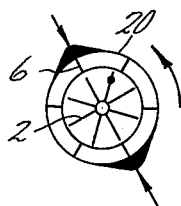


FIG. 7

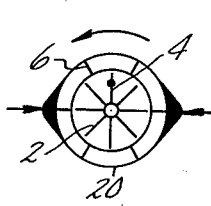


FIG. 8

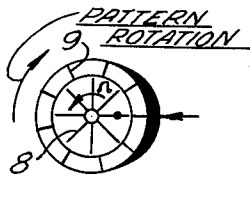


FIG. 9

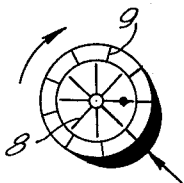


FIG. 10

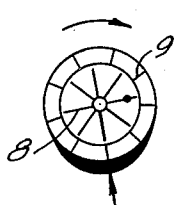


FIG. 11

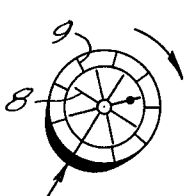


FIG. 12

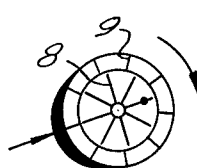


FIG. 13

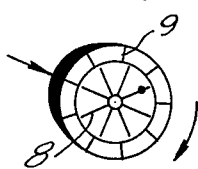


FIG. 14

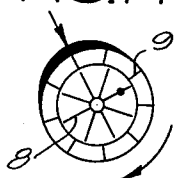


FIG. 15

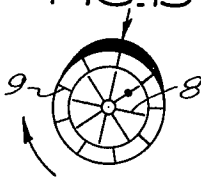


FIG. 16

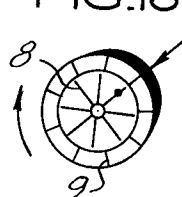
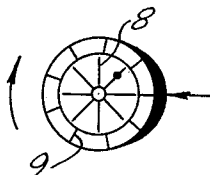


FIG. 17



INVENTORS
JOHN M. TYLER
THOMAS G. SOFRIN
BY *Vernon F. Hauschild*
ATTORNEY

July 13, 1965

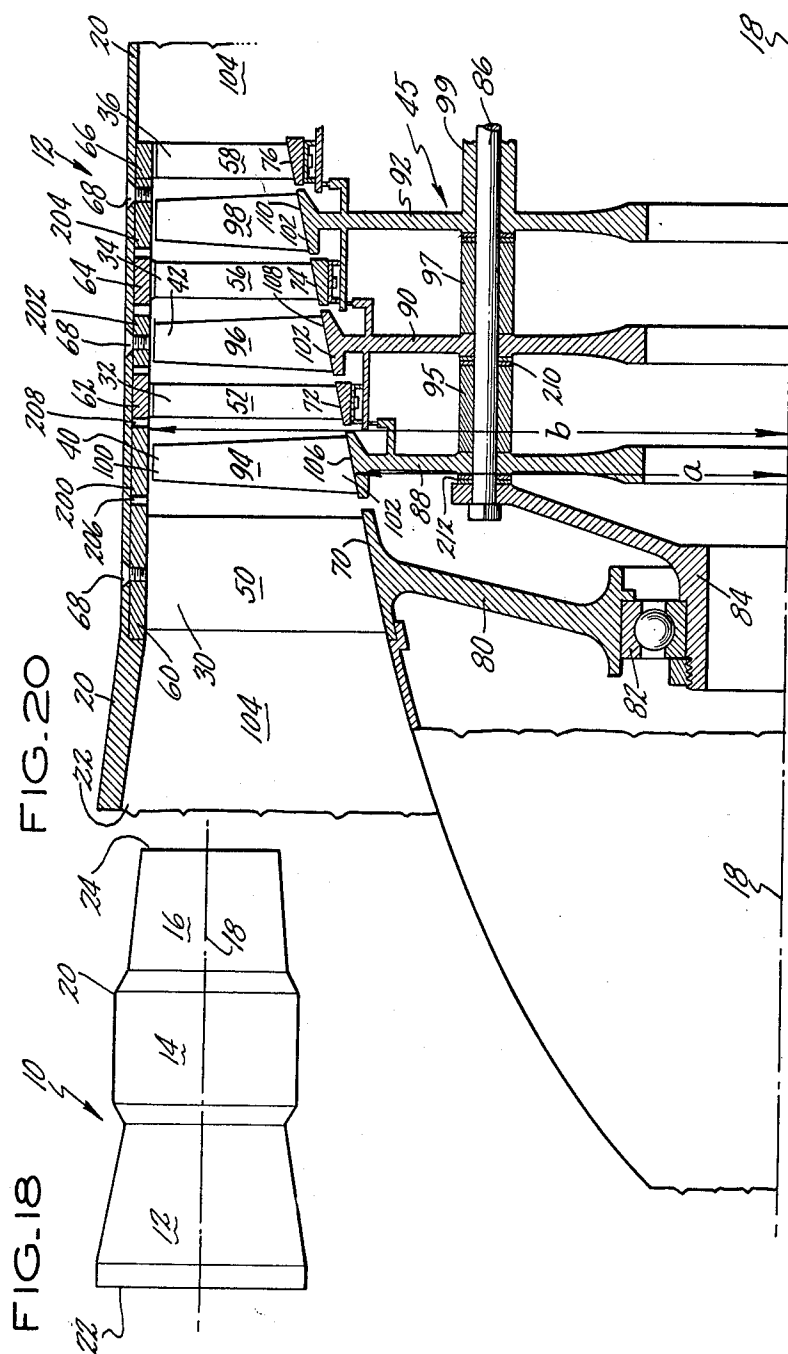
J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 2



July 13, 1965

J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 3

FIG. 26A

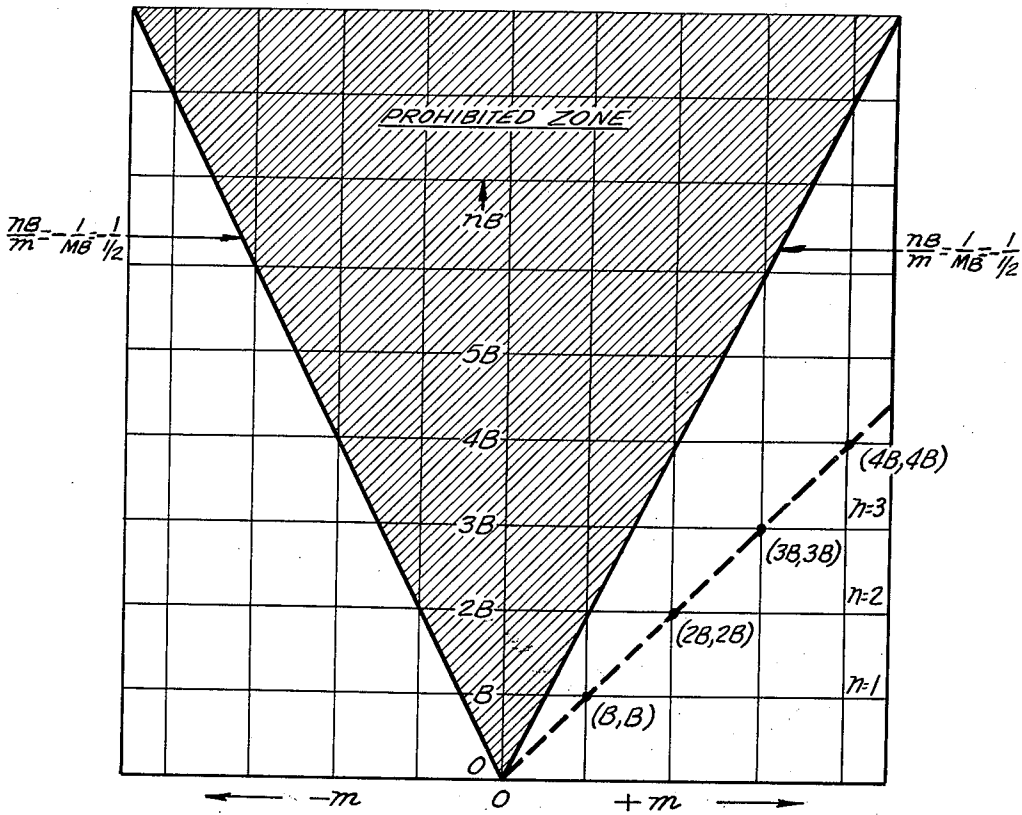
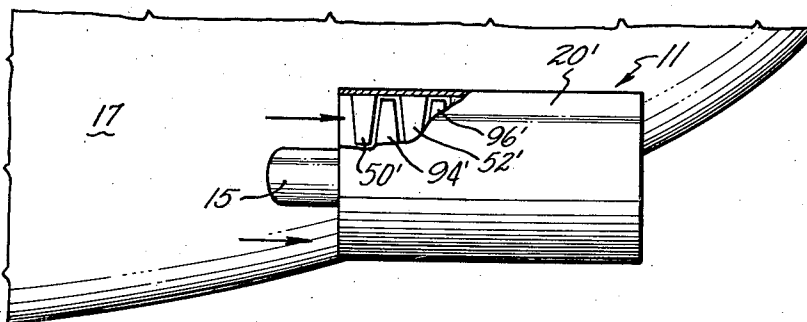


FIG. 19



July 13, 1965

J. M. TYLER ETAL

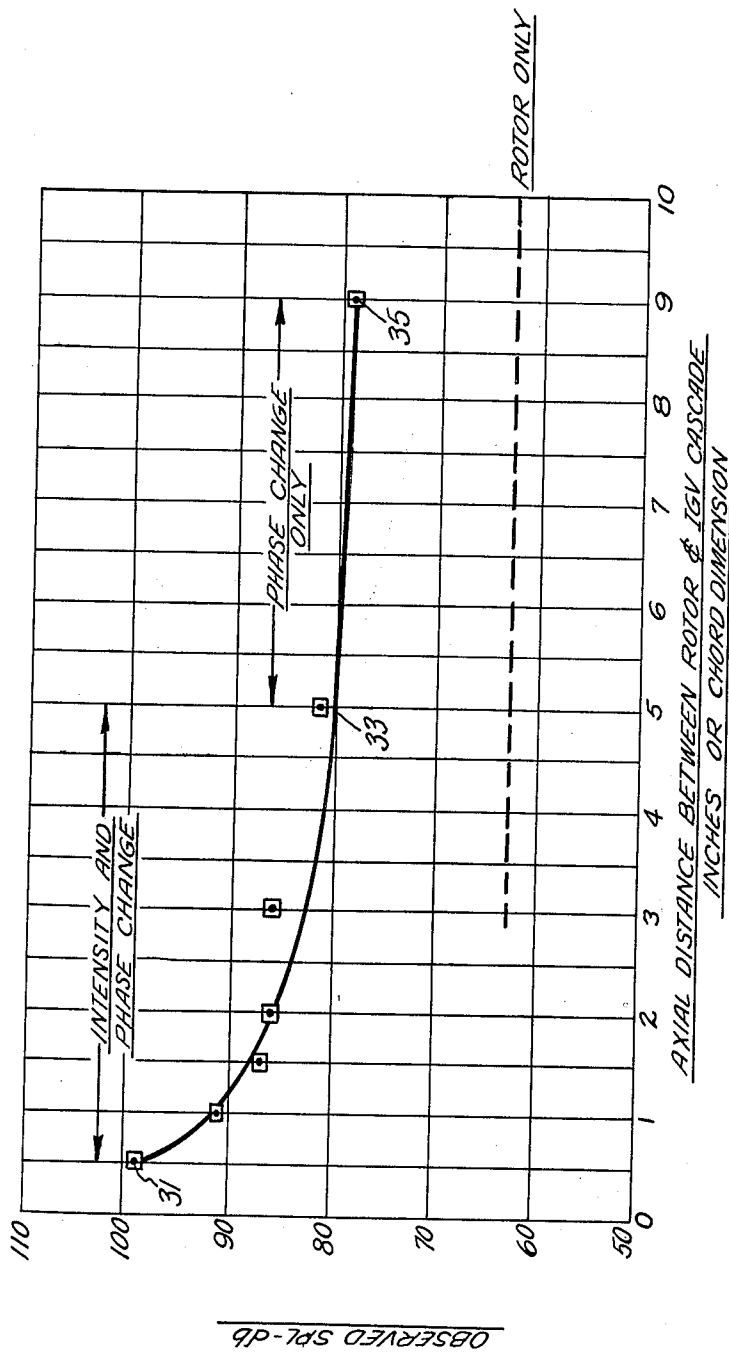
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 4

FIG-21



July 13, 1965

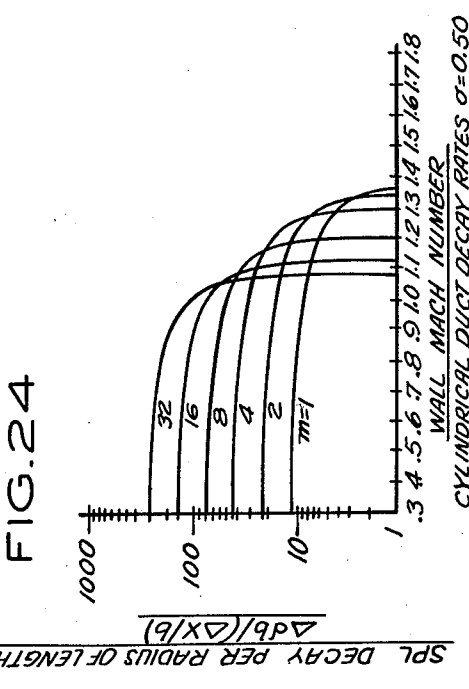
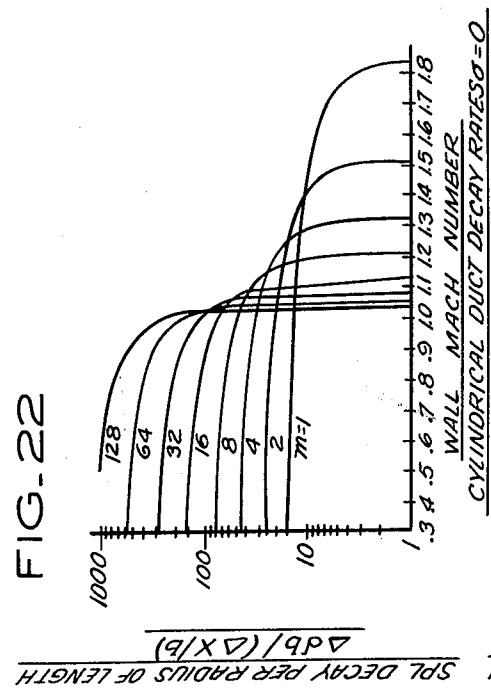
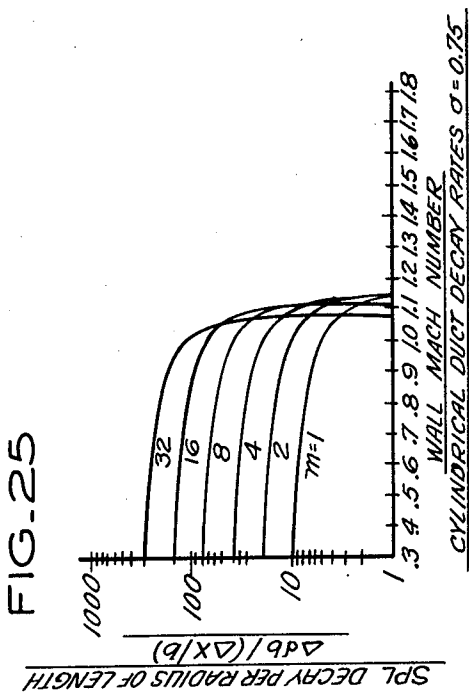
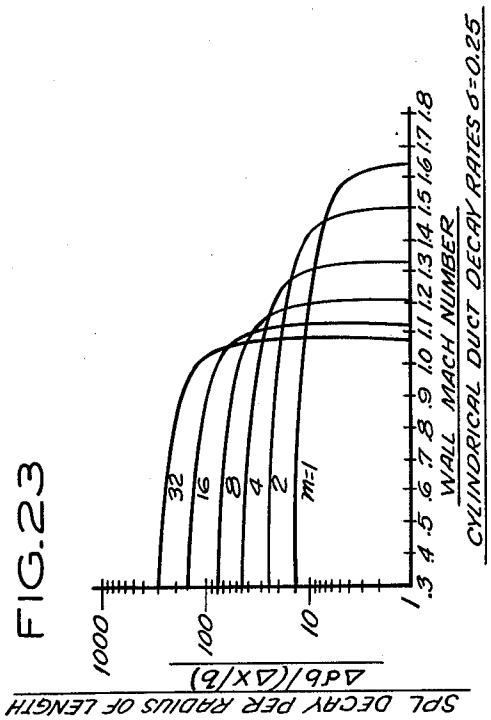
J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 5



July 13, 1965

J. M. TYLER ETAL

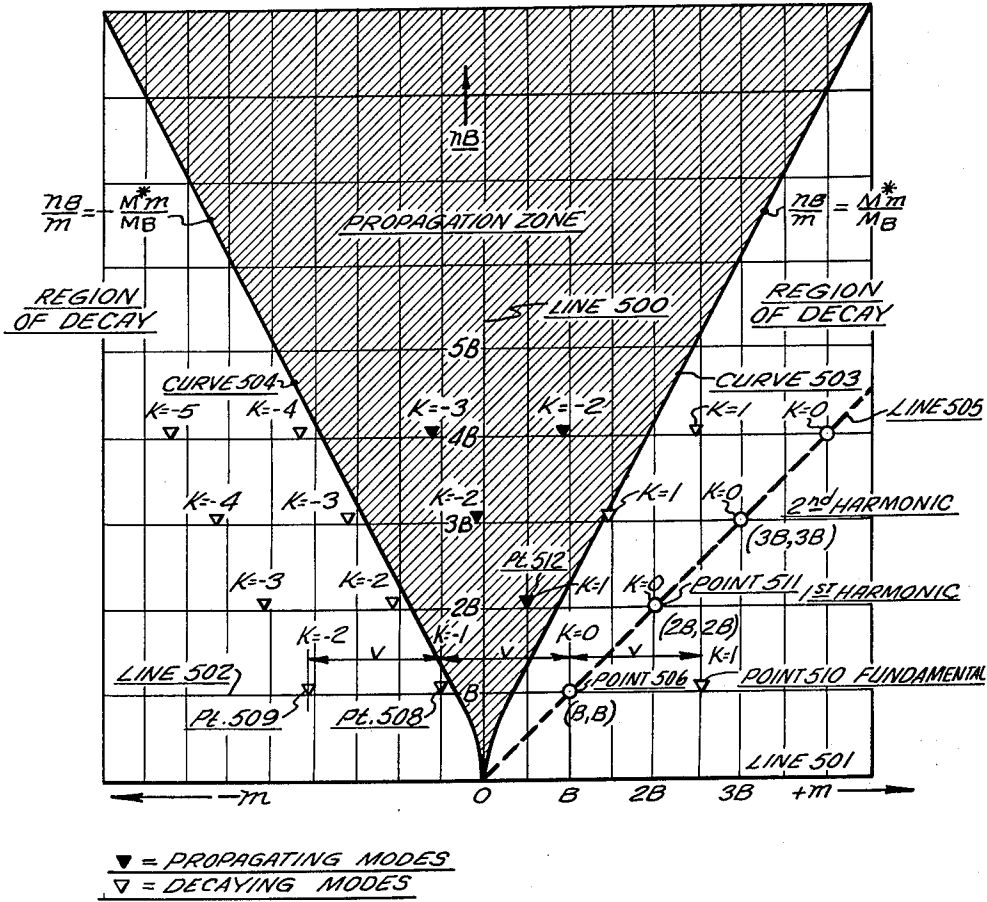
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 6

FIG. 26



July 13, 1965

J. M. TYLER ETAL

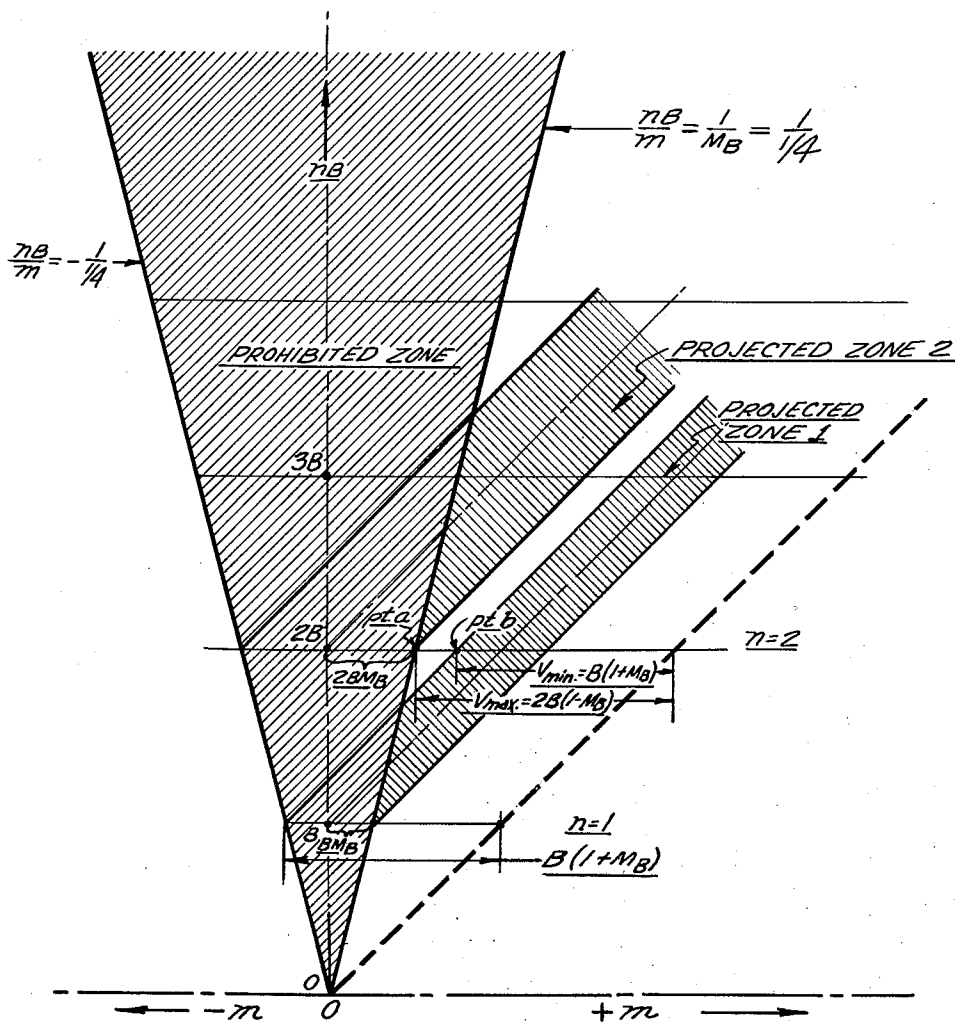
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 7

FIG. 26B



July 13, 1965

J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 8

FIG. 26C

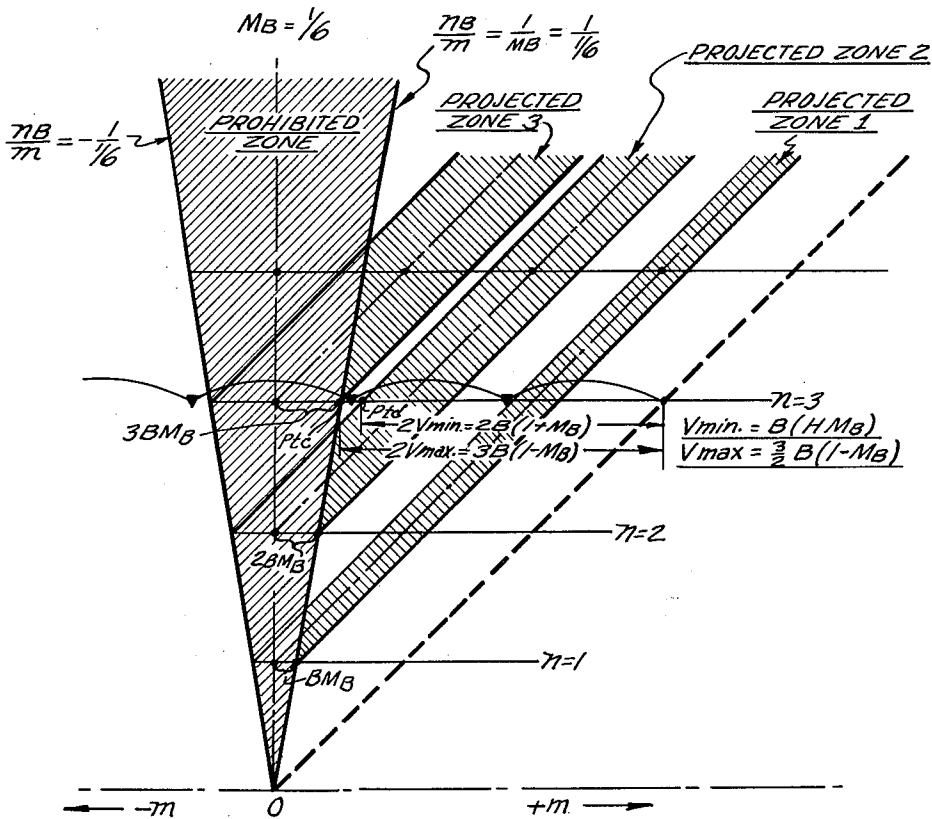
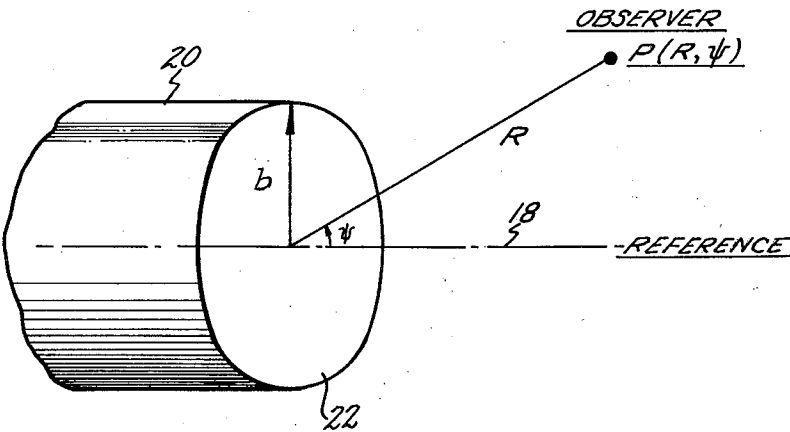


FIG. 36



July 13, 1965

J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 9

FIG. 27

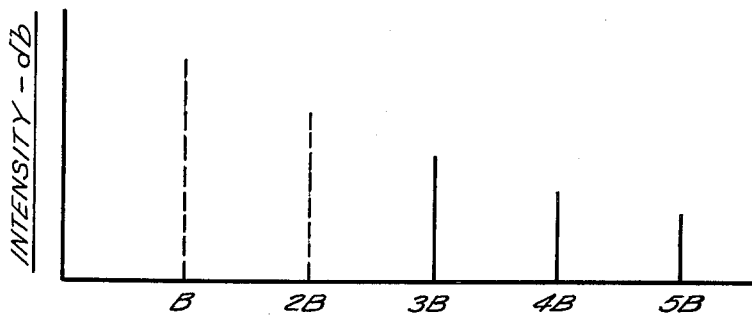


FIG. 28

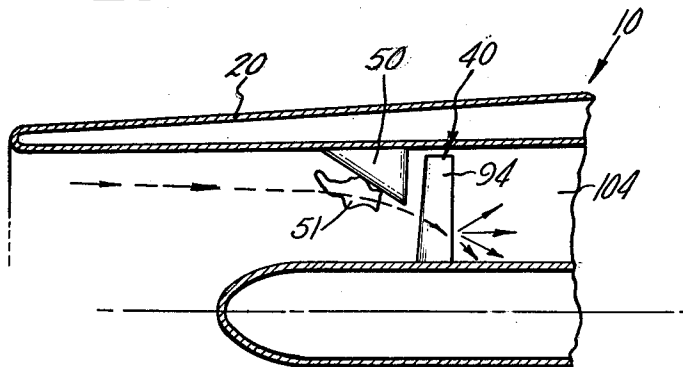
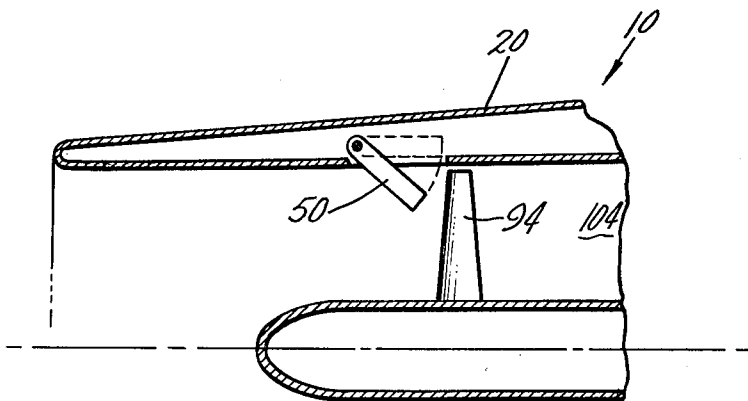


FIG. 29



July 13, 1965

J. M. TYLER ET AL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 10

FIG. 30

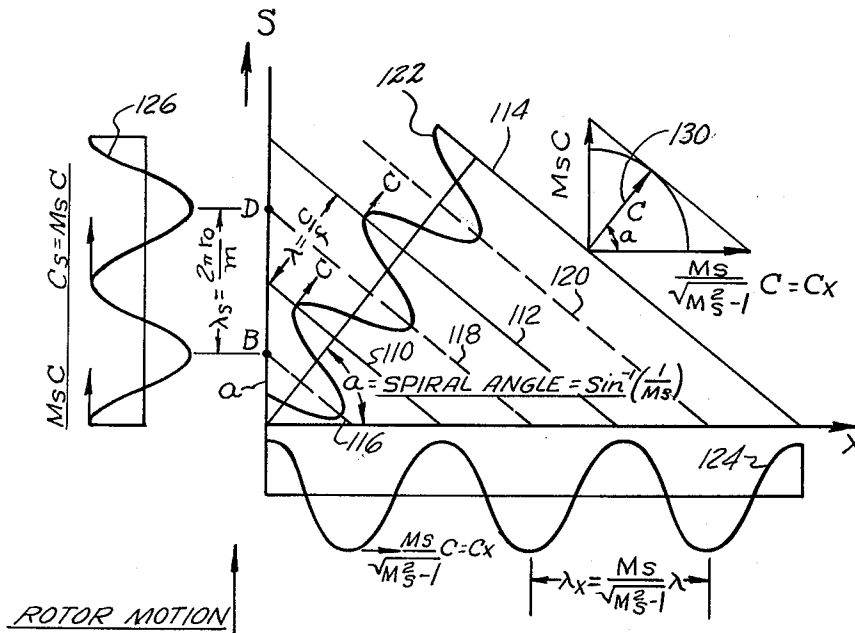


FIG. 46

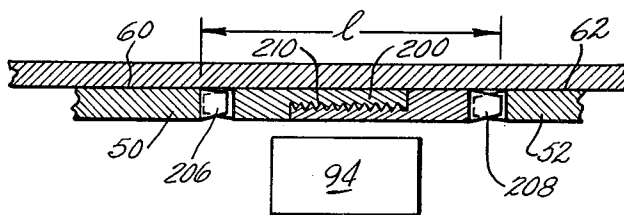
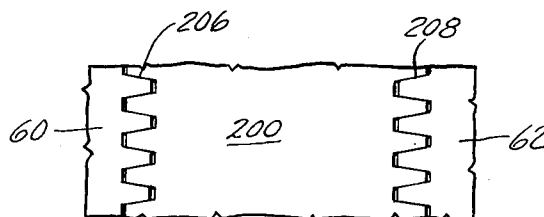


FIG. 47



July 13, 1965

J. M. TYLER ET AL

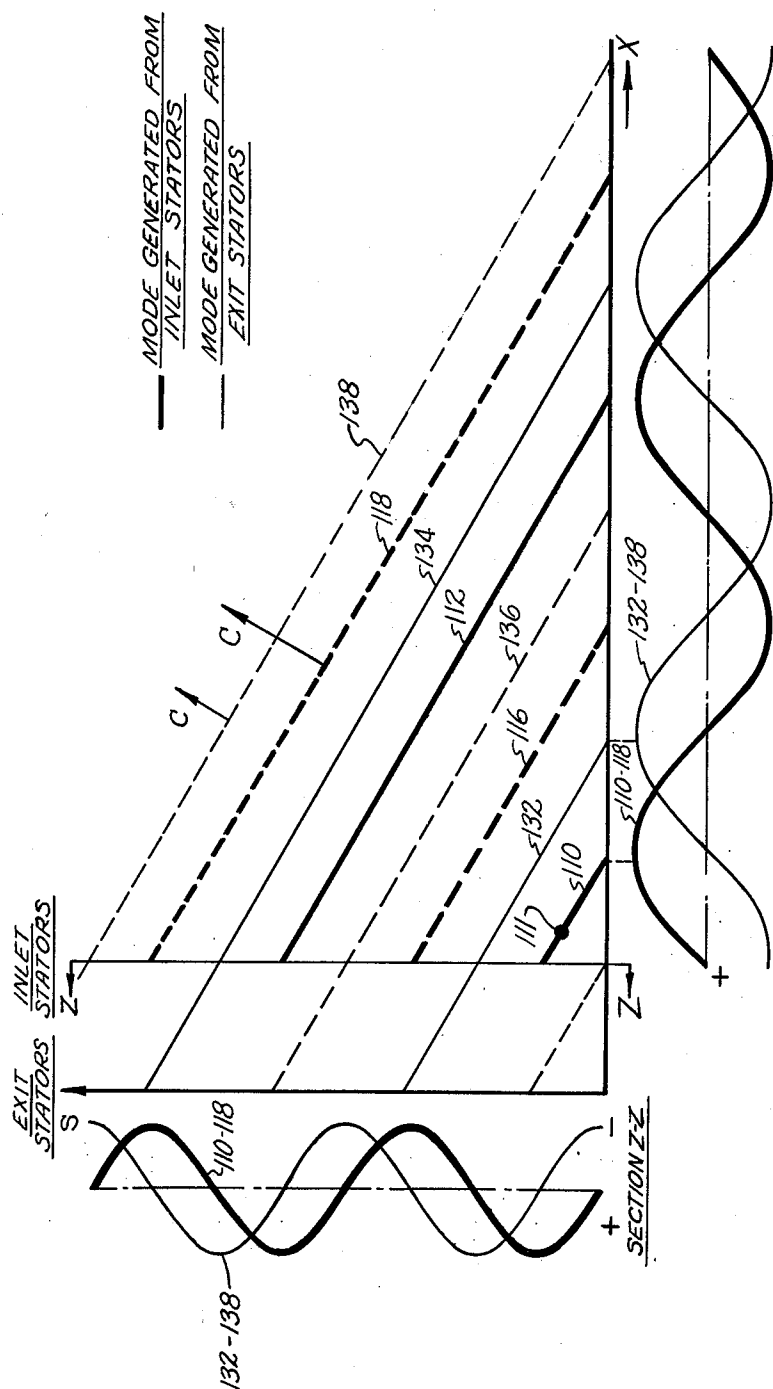
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 11

உ.உ.பி.



July 13, 1965

J. M. TYLER ETAL

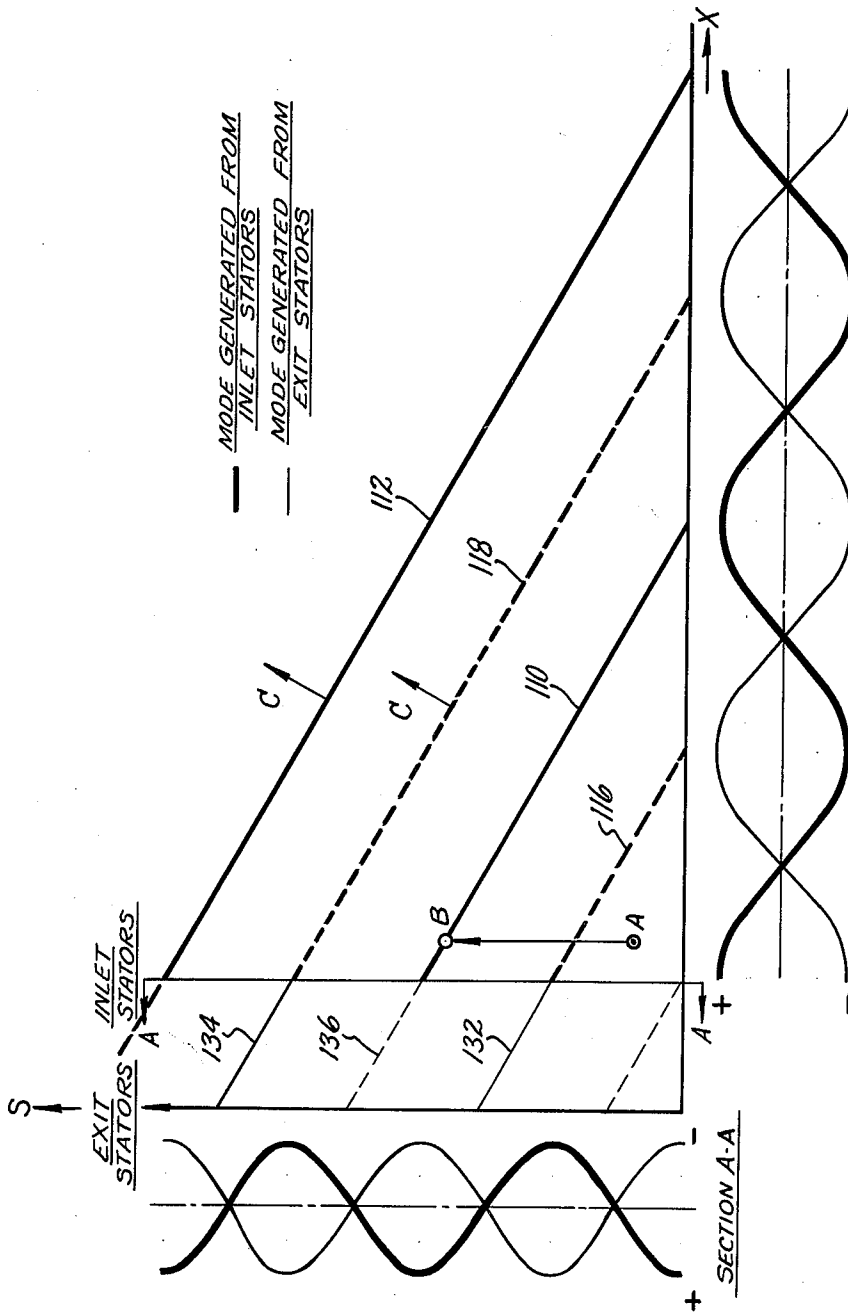
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 12

F/G.32



July 13, 1965

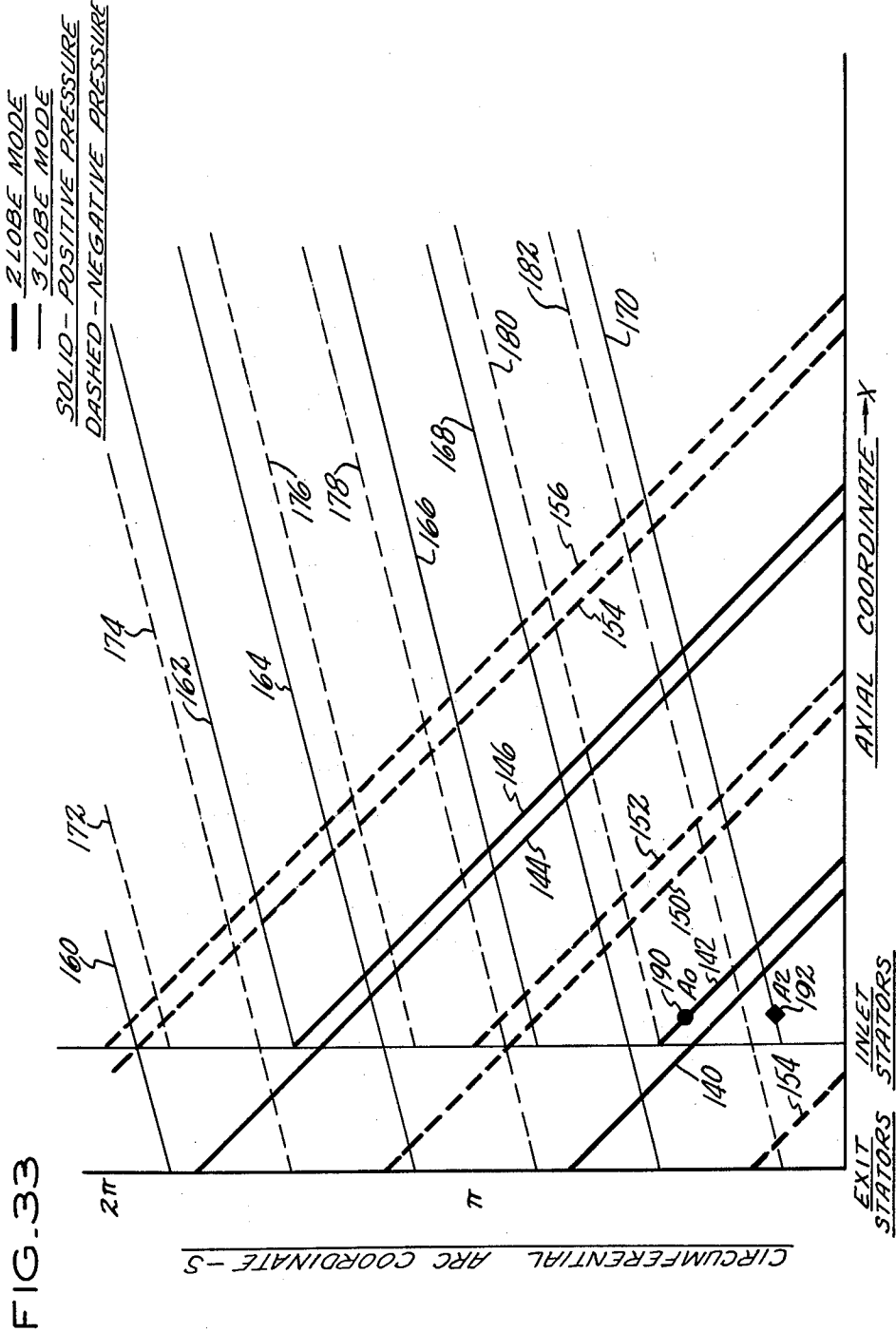
J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 13



July 13, 1965

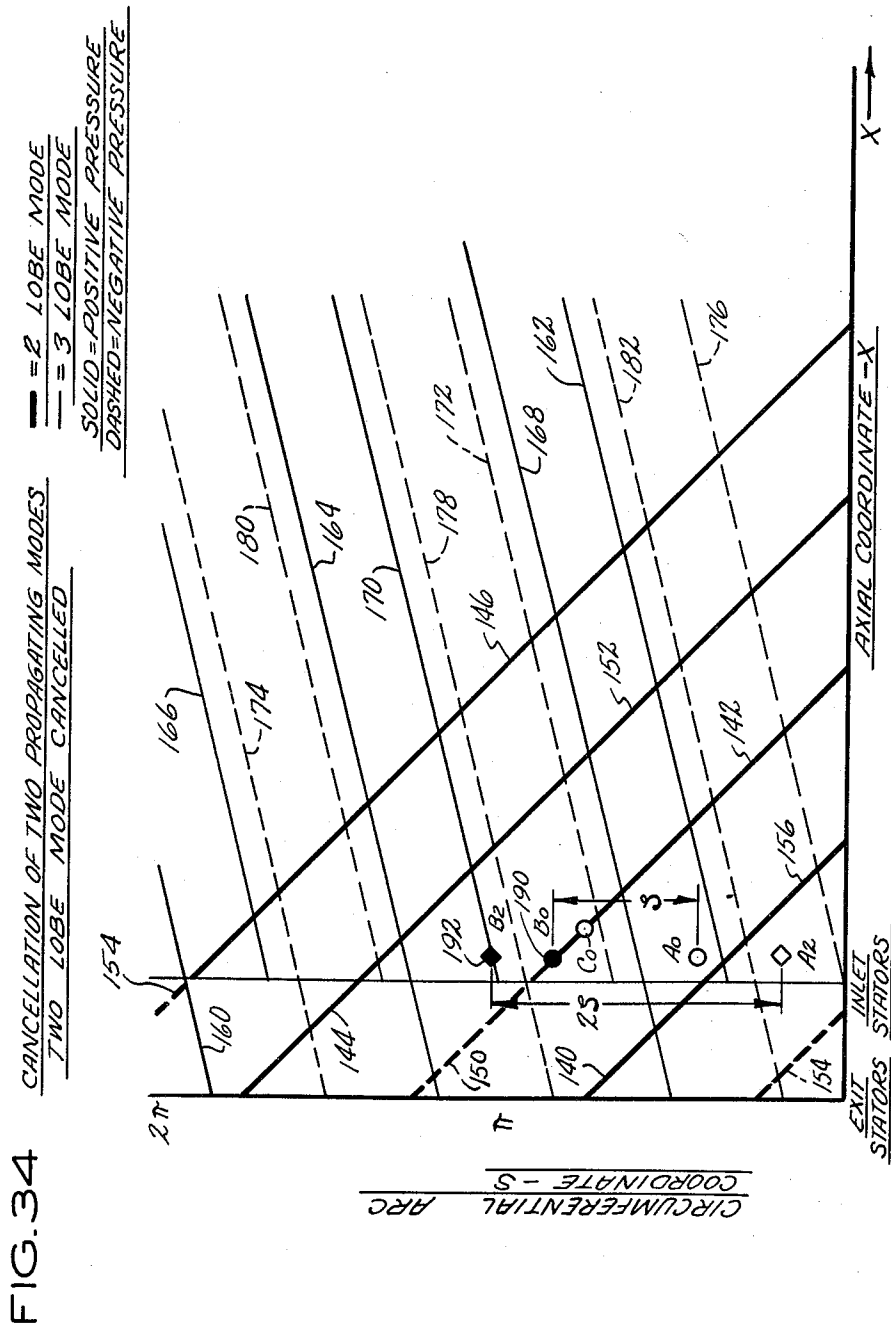
J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 14



July 13, 1965

J. M. TYLER ET AL

3,194,487

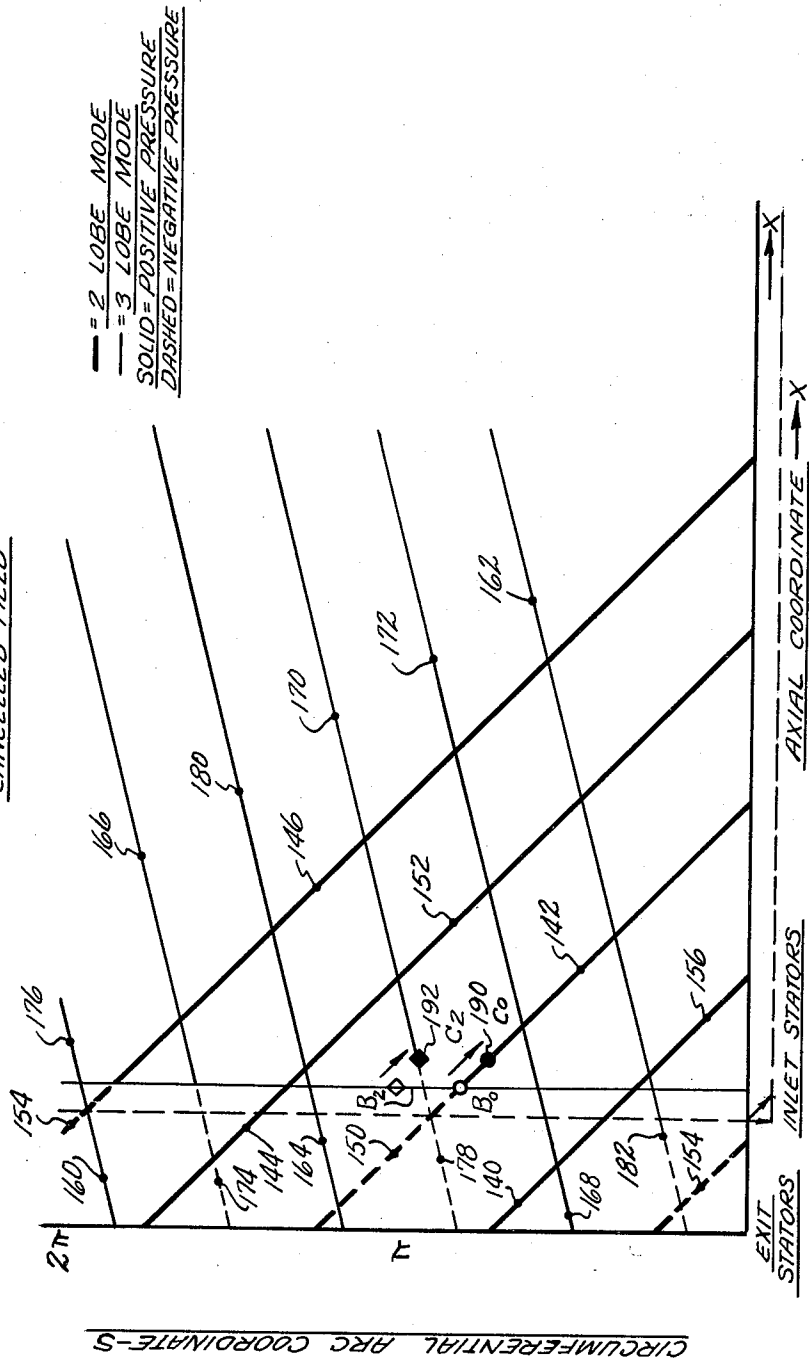
NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 15

FIG. 35

CANCELLATION OF TWO PROPAGATING MODES
CANCELLED FIELD



July 13, 1965

J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 16

FIG. 37

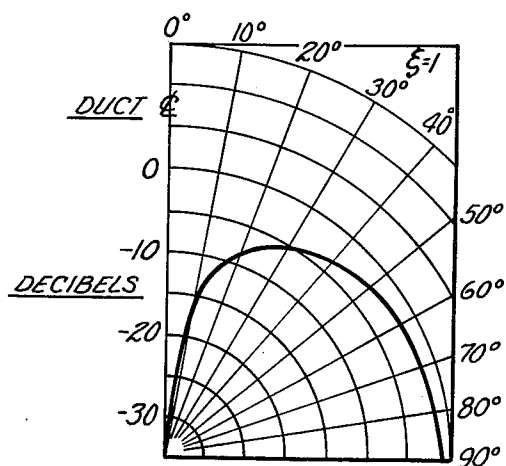


FIG. 39

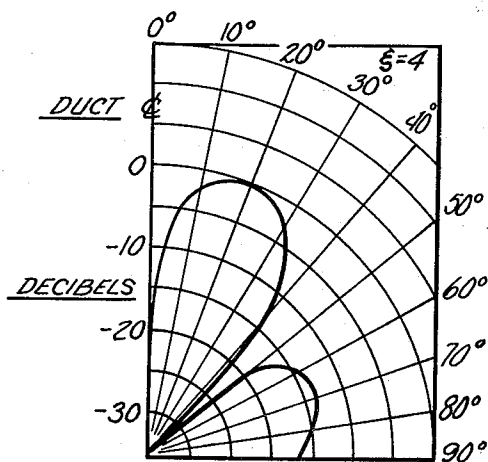


FIG. 38

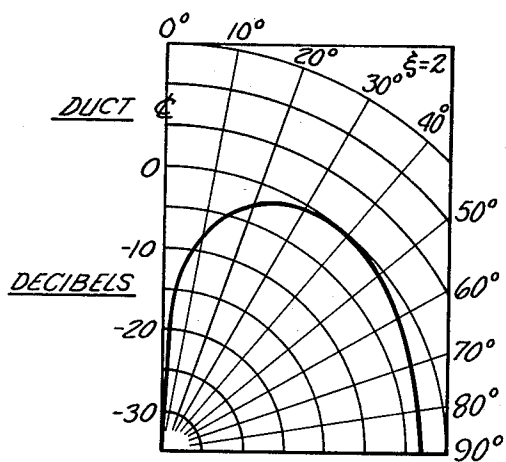
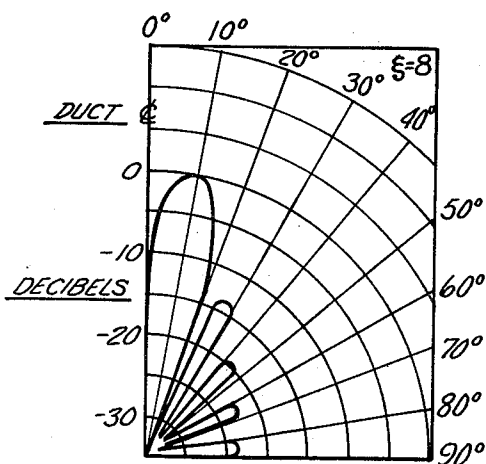


FIG. 40



July 13, 1965

J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 17

FIG. 41

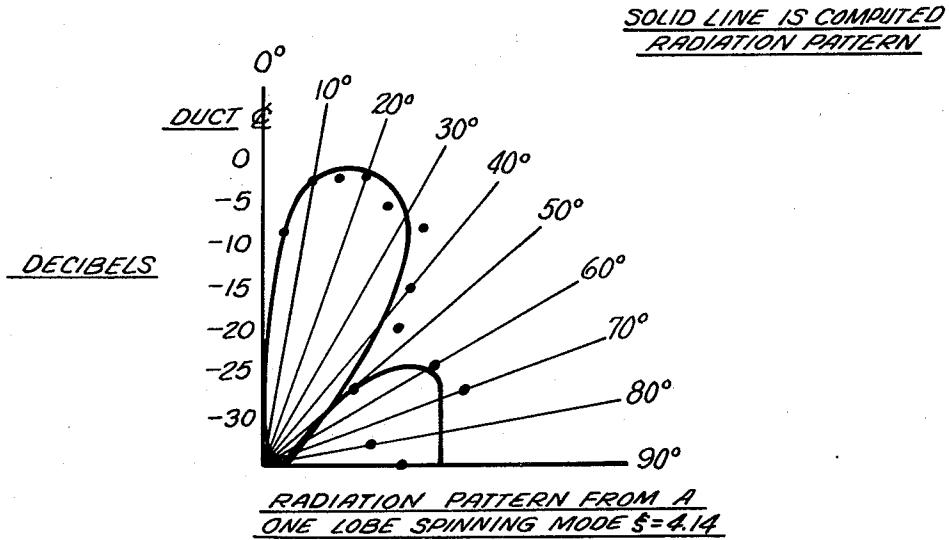
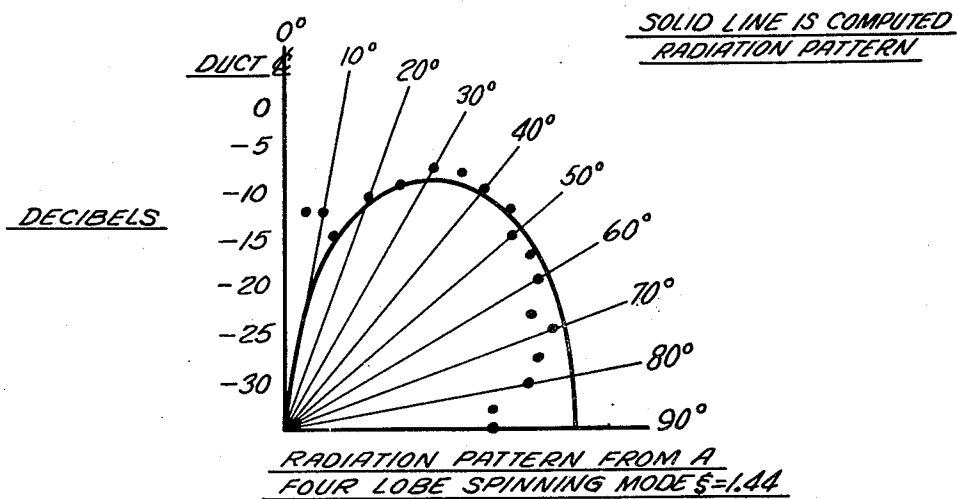


FIG. 42



July 13, 1965

J. M. TYLER ETAL

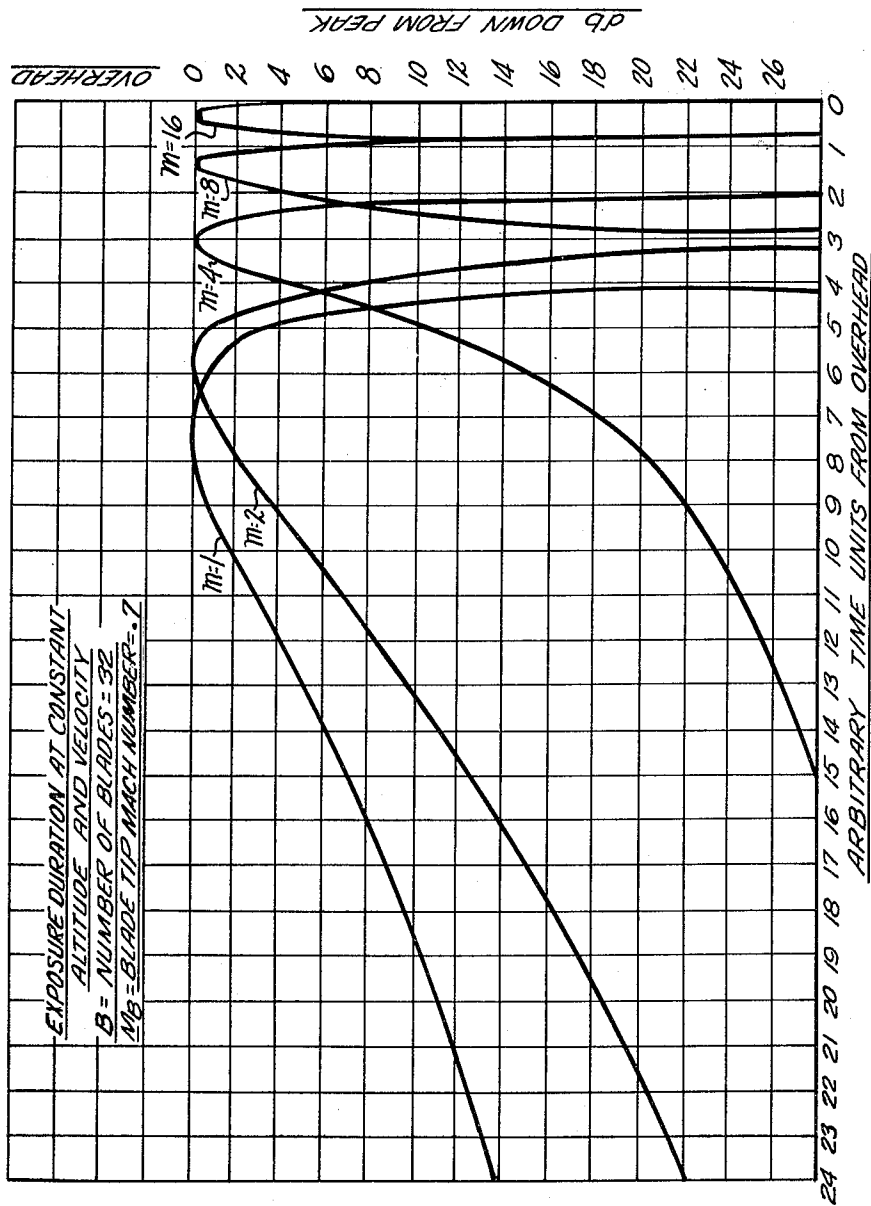
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 18

FIG. 43



July 13, 1965

J. M. TYLER ETAL

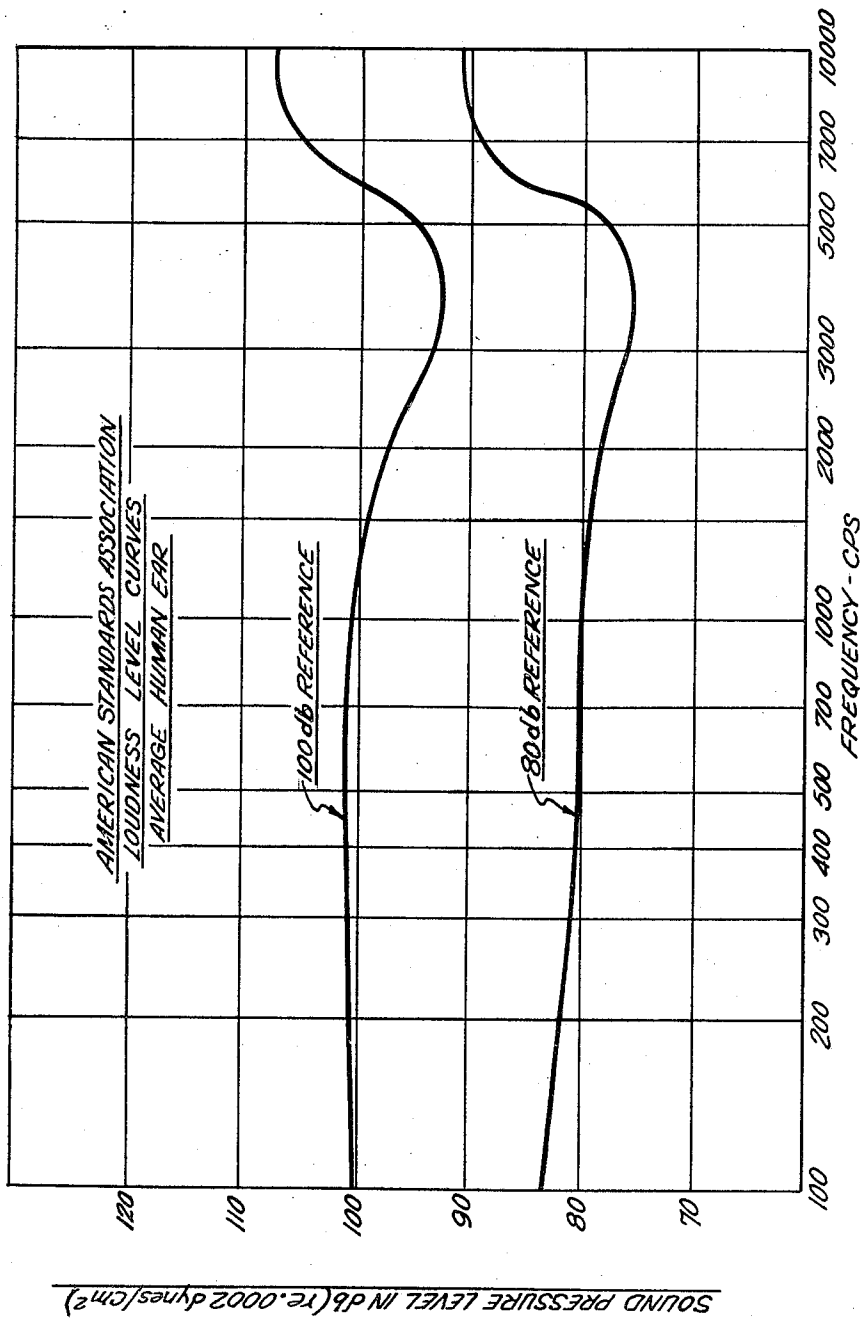
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 19

FIG. 44



July 13, 1965

J. M. TYLER ETAL

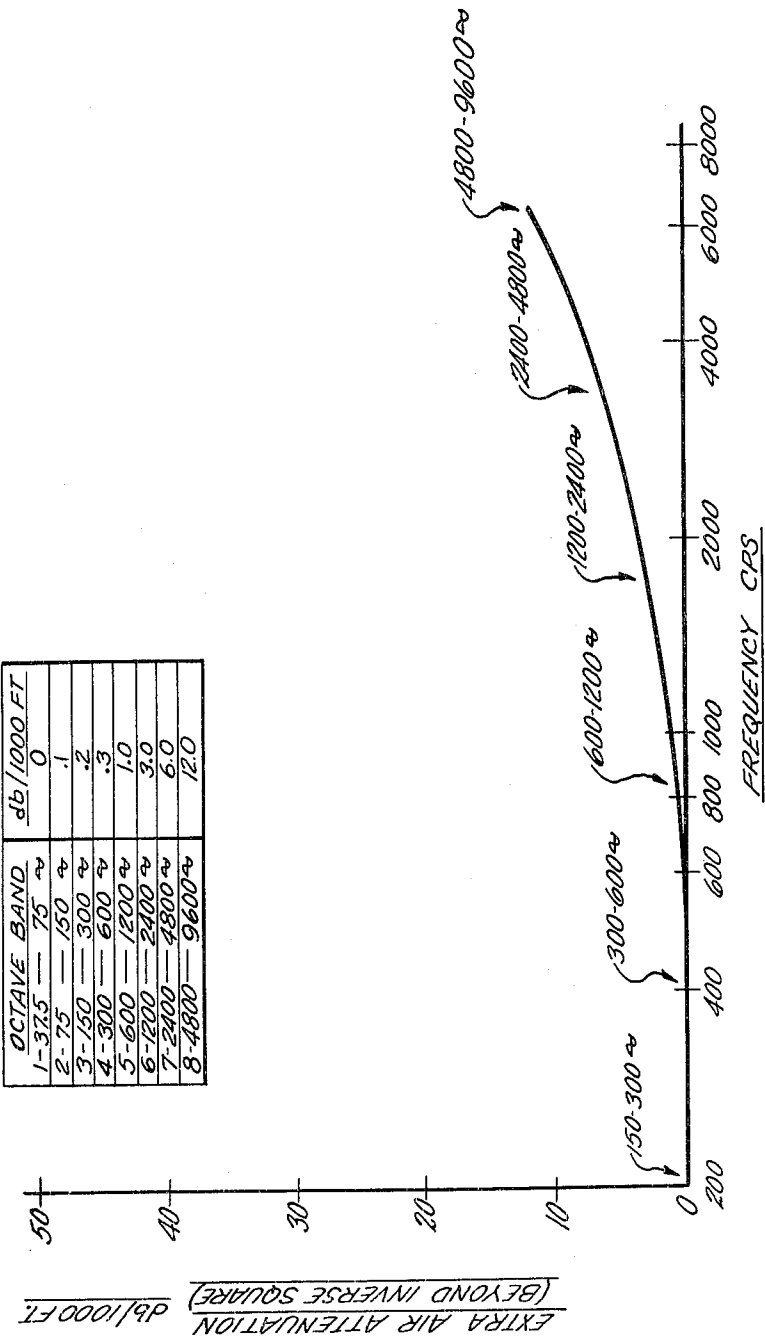
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 20

FIG. 45



July 13, 1965

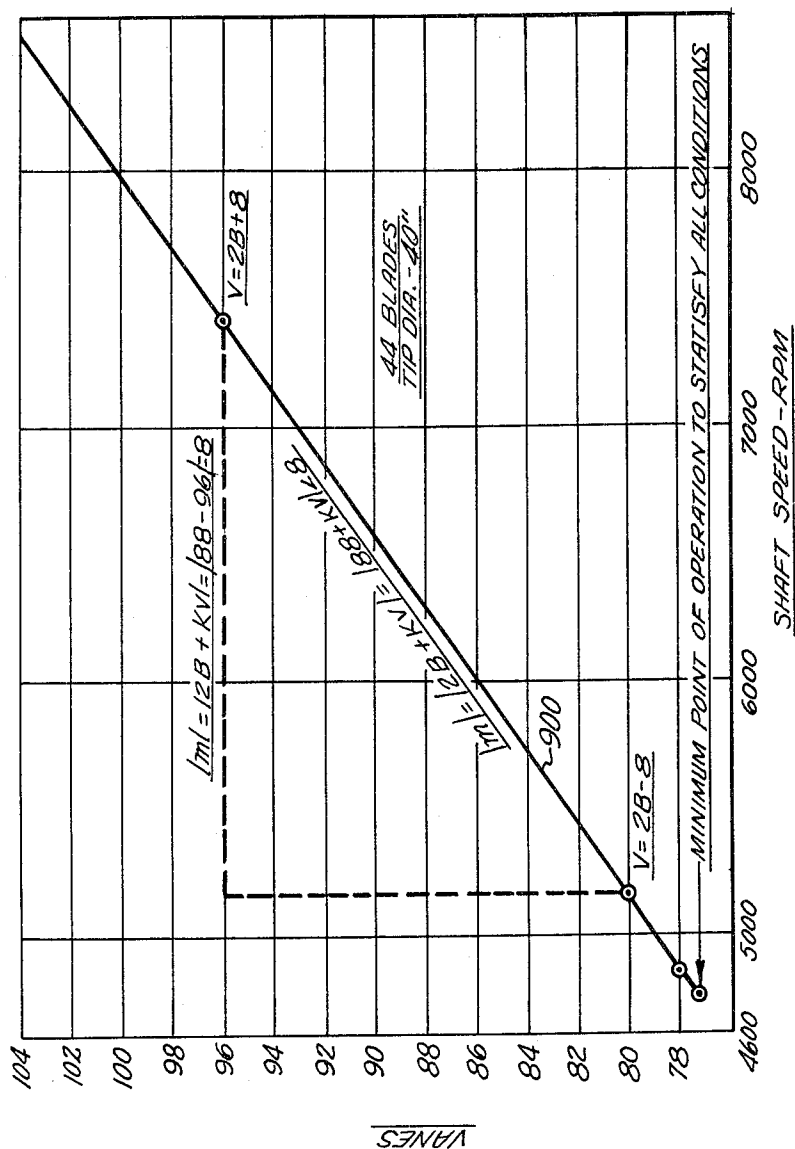
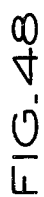
J. M. TYLER ETAL

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 21



July 13, 1965

J. M. TYLER ETAL

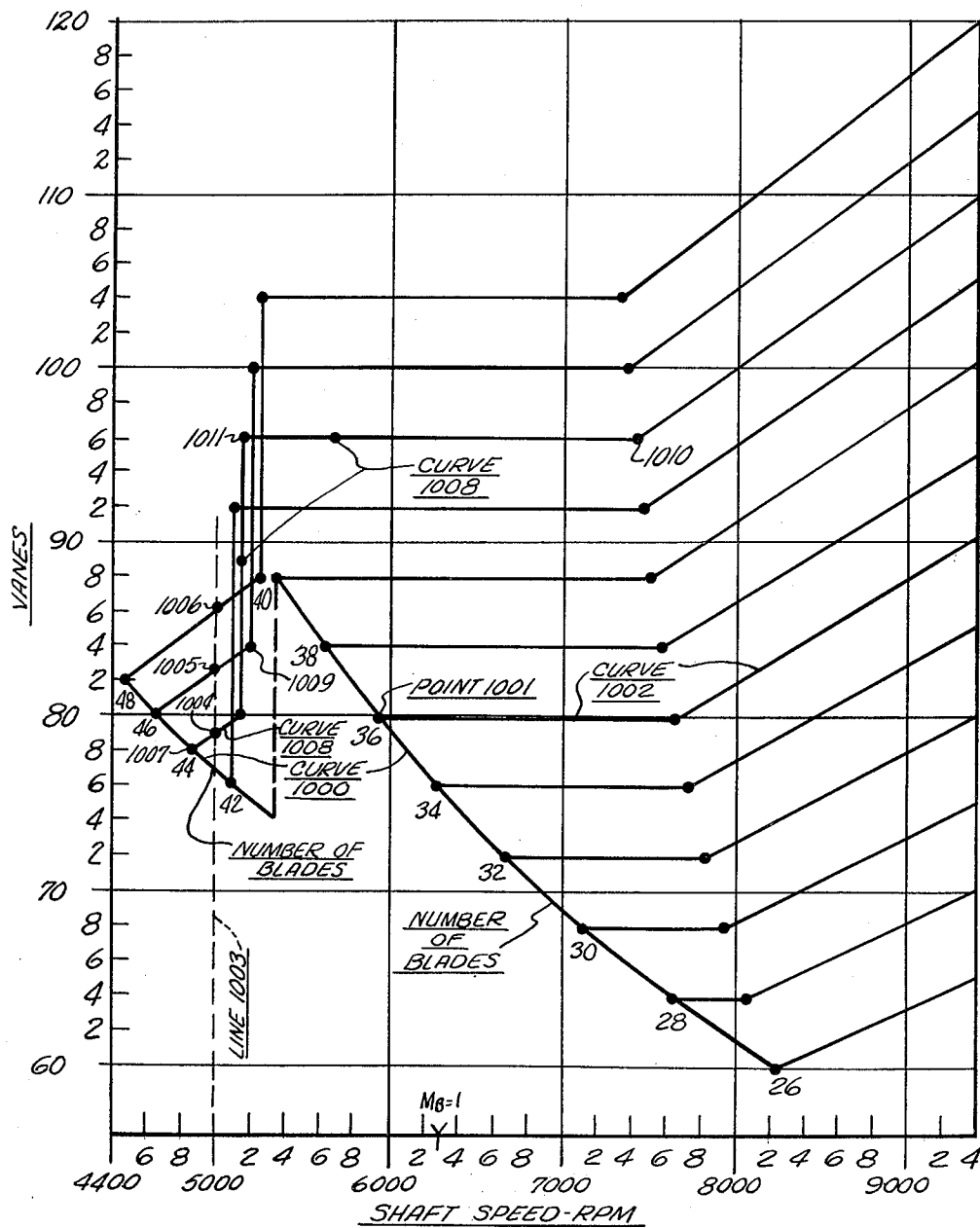
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 22

FIG. 49



July 13, 1965

J. M. TYLER ETAL

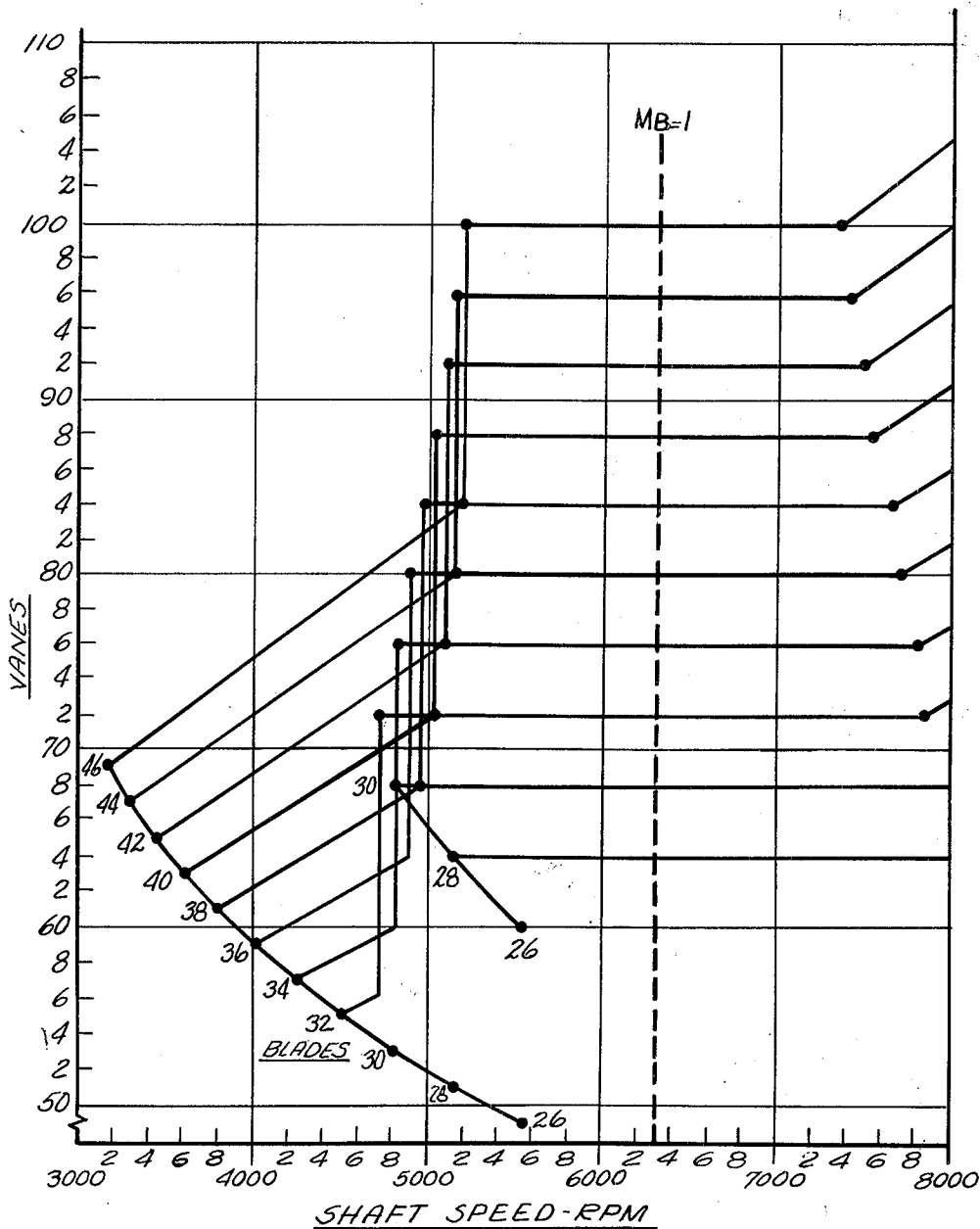
3,194,487

NOISE ABATEMENT METHOD AND APPARATUS

Filed June 4, 1963

23 Sheets-Sheet 23

FIG. 50



1

3,194,487

NOISE ABATEMENT METHOD AND APPARATUS
John M. Tyler, Glastonbury, and Thomas G. Sofrin, West
Hartford, Conn., assignors to United Aircraft Corpora-
tion, East Hartford, Conn., a corporation of Delaware
Filed June 4, 1963, Ser. No. 285,361
48 Claims. (Cl. 230-122)

This invention relates to noise suppression and more particularly to the abatement of noises of discrete frequency emanating from rotating machinery utilizing rotating blades and stationary vanes and in particular from an inlet of an axial flow compressor of the type commonly used in aircraft gas turbine powerplants and from shrouded marine propellers.

With the advent of jet powered aircraft and, in particular, commercial jet aircraft entering and leaving municipal airports, the question of how to suppress the noise created by a jet engine has been given extensive consideration. The first problem attacked by engineers in this field was the reduction or elimination of exhaust gas wake noise. Both the use of the exhaust gas wake noise suppressors developed during this early period, for example of the type taught in our U.S. Patent Nos. 2,845,775, 2,930,185, 2,931,171, 2,936,846, 2,944,392, 3,027,713 and 3,033,494, and the advent of the turbofan type power plant with its reduced jet exhaust noise, brought more sharply to the public's attention the discrete frequency noise, sometimes called "whine," generated by the compressor and being propagated groundwardly, mainly from the compressor inlet. This compressor inlet whine is most objectionable during municipal airport approach when the jet engines of the aircraft are operating at reduced power and hence the jet exhaust noise is greatly reduced. The "whine" is not so objectionable at take-off for the jet exhaust noise then conceals it. It had been thought that the compressor inlet discrete frequency noise was akin to the type of noise generated by an aircraft propeller. When the existing knowledge regarding propeller noise was applied to solve the compressor inlet discrete frequency noise problem, predicted results were at great variance with actual measurements. These discrepancies were left with the status of unresolved paradoxes and it has become clear that aircraft propeller generated noise and compressor generated discrete frequency noise could not be similarly explained.

Equally important is the problem of suppressing the noise made by marine propellers on naval vessels. During an attack upon a submarine, the surface vessel can detect the position of the target submarine from the noise made by its propellers and if the target submarine should have to discontinue the use of its propellers in order to avoid or eliminate the noise made thereby, it would be impossible for the target submarine to escape from the attack area. In addition, attacking submarines are aided in detecting the whereabouts of surface vessels by the surface vessel propeller noise. It is, therefore, important for both surface and submarine naval vessels that their propeller noises be eliminated or drastically reduced. We have found that since both air and water are fluids, they have similar wave transmission characteristics. The propagation of acoustic energy through air is similar in considerable detail to the propagation of hydrodynamic pressure waves through water.

While the description of our invention will be made using an axial flow compressor of an aircraft turbojet engine as the primary illustration, it should be borne in mind that the principles taught herein are equally applicable to shrouded marine propellers and other types of rotary machinery using rotating blades and stationary vanes or struts.

We have made extensive studies and experimentations

2

regarding axial flow compressor inlet discrete frequency noises and have found, as more fully described hereinafter, that they comprise spinning modes or rotating pressure fields caused both by the revolving compressor rotor blades by themselves and by the interaction established between adjacent rows of compressor rotor blades and compressor stator vanes.

It is an object of this invention to provide method and apparatus for reducing and eliminating discrete frequency noises, both fundamental and harmonic, generated in an axial flow compressor and which emanate from the compressor inlet and/or discharge.

It is a further object of this invention to teach method and apparatus for reducing the discrete frequency noise emanating from an axial flow compressor comprising selecting blade and vane combinations which are of such number and so positioned relative to one another that either a reduced strength spinning mode is formed, or a spinning mode is formed of such a nature that it will not be propagated through the compressor duct and out through the compressor inlet and/or discharge but, in fact, will decay within the compressor duct, or such that spinning modes of equal intensity and opposite phases are generated or established so they will cancel each other within the compressor duct, or such that the radiation directivity pattern of the discrete frequency noise emanating from the compressor inlet produces reduced exposure time to those on the ground as the airplane flies by overhead, or such that the modes which propagate from the compressor inlet are established at a sufficiently high frequency to be above the range of high human hearing sensitivity; namely, 7,000 c.p.s. (cycles per second).

It is still a further object of this invention to select the number of blades in rotors of an axial flow compressor and to select the number of vanes in stators of an axial flow compressor and to axially position the rotors and stators such that the spinning modes created by rotor-stator interaction are substantially equal, then indexing the rotors and stators circumferentially to cause the equal amplitude spinning modes to be out of phase and hence cancel.

It is still a further object of this invention to select blade-vane combinations in an axial flow compressor such that the number of vanes is approximately equal to or greater than twice the number of blades times the number of the harmonic index of the discrete frequency noise being considered.

It is still a further object of this invention to teach a method and means of reducing discrete frequency compressor inlet noises comprising choosing blade-vane combinations such that the spinning modes created have a minimum of approximately 8 lobes.

It is a further object of this invention to teach a method and means of reducing the discrete frequency compressor inlet noise in an axial flow compressor comprising selecting a blade-vane combination to produce a spinning mode having m number of lobes wherein m equals the harmonic index n , times the number of blades, B , plus an independent positive or negative integer, k , times the number of vanes, V , and so that the number of lobes, m , is equal to or greater than the number given by the formula

$$nB(M_B - M_m^*)$$

where M_B is the rotor circumferential tip Mach number and M_m^* is a cutoff or critical Mach number.

It is a further object of this invention to teach means and apparatus for producing a quiet compressor which means and apparatus comprise a system which is flexible and comprehensive and includes five basic courses of action which may be used, singularly or in combination, for controlling compressor noise.

Other objects and advantages will be apparent from the

specification and claims and from the accompanying drawings which illustrate an embodiment of the invention.

FIGS. 1-7 are schematic representations of the relative motion which takes place between the blades of an eight blade rotor, the vanes of a six vane stator and the two lobed spinning mode created thereby, which mode is rotating in a counterclockwise direction.

FIGS. 8-17 are schematic representations of the relative motion which takes place between the blades of an eight blade rotor, the vanes of a nine vane stator and the single lobed spinning mode created thereby, which single mode is rotating in a clockwise direction.

FIG. 18 is an external side view of a typical turbo-jet engine of the type used in a modern aircraft and which will utilize our invention.

FIG. 19 is a side view of a shrouded marine propeller, partially broken away to illustrate our invention.

FIG. 20 is an enlarged, cross-sectional view showing of the forward end of an axial flow compressor illustrating a preferred embodiment of our invention.

FIG. 21 is a graph illustrating the noise reduction which can be expected by axially spacing adjacent stators and rotors various distances. The graph scale chosen is 1 inch=1 chord length.

FIGS. 22, 23, 24 and 25 illustrate the decay rates of spinning modes having various lobe numbers m and which rotate at various wall speeds or Mach numbers. These figures illustrate the effects of decay when the hub-tip ratio is 0, 0.25, 0.50 and 0.75, respectively.

FIGS. 26, 26a, 26b and 26c are graphic representations of various solutions of Formula 1 used to illustrate how blade and vane combinations are selected to avoid spinning mode propagation and insure spinning mode decay within the compressor duct for various Mach numbers.

FIG. 27 is a typical far field spectrum of an axial flow compressor designed to decay the fundamental (B) and the first harmonic (2B), with the higher harmonic (3B etc.) propagating.

FIG. 28 is a schematic representation of cantilevered inlet guide vanes 59 projecting radially inwardly from the compressor outer case and extending throughout the blade tip region only so as to satisfy both the requirement of high vane number and object ingestion.

FIG. 29 is a compressor having retractable inlet guide vanes.

FIG. 30 is a developed view projected, both axially and circumferentially, illustrating a spinning mode and indicating the ridges of high and low pressure.

FIG. 31 is a graphic representation of a developed view of two spinning modes created by different sources in an axial flow compressor, which modes are illustrated precisely as generated.

FIG. 32 is similar to FIG. 31 but shows the modes in a corrected condition 180° out-of-phase and hence cancelling.

FIG. 33 is a developed view of a two-lobe mode and a three-lobe mode as generated in an axial flow compressor.

FIG. 34 corresponds to FIG. 33 but shows the two-lobe mode cancelled.

FIG. 35 corresponds to FIGS. 33 and 34 and shows both the two-lobe mode and the three-lobe mode cancelled.

FIG. 36 is an illustration of the inlet end of a compressor and the parameters used in considering the directivity pattern created by the spinning modes propagating therefrom.

FIGS. 37-40 illustrate the directivity pattern for a single lobed spinning mode as engine speed is increased.

FIGS. 41 and 42 illustrate a comparison of test data and theoretical data.

FIG. 43 is a graphic representation of the ground exposure time of directivity patterns established by spinning modes having various numbers of lobes m .

FIG. 44 is the Fletcher-Munson curve. This shows the

variation of ear sensitivity to noise as a function of frequency.

FIG. 45 is a graphic illustration of the extra air attenuation for noise generated at various frequencies.

FIG. 46 is a partial enlarged showing of a portion of the compressor shown in FIG. 20 and illustrating adjacent stators and rotors and their connection to permit axial, circumferential and helical indexing.

FIG. 47 is a developed view of the face spline connections of FIG. 46.

FIG. 48 is a showing of the relation between the number of vanes and shaft speed for an illustrative engine to meet the requirements of the indicated noise criteria.

FIGS. 49 and 50 show families of curves illustrating blade-vane combinations suitable for various operating speeds so as to satisfy the designated acoustic requirements.

In view of the complexity of the theory of the derivation and behavior of discrete frequency compressor noises, it will be necessary to use mathematical formulae and symbols throughout this description. According, Appendix A is provided at the end of the description to provide a table of definitions of the various symbols to be used herein.

THE NATURE OF COMPRESSOR NOISE

Since our initial study of the noise problem was directed toward solving the noises in turbojet aircraft engines, our description will be directed primarily toward the noise problems as they apply to a turbojet aircraft engine, but it should be borne in mind that there are other equally important applications which may be found in all types of rotating machinery of the vane-blade type, such as the marine propeller application referred to herein.

The nature of compressor noise is described in substantial particularity in an article by J. M. Tyler and T. G. Sofrin entitled "Axial Flow Compressor Noise Studies" appearing in SAE Transactions, Reprint, 1962, pages 309 through 332, which is hereby incorporated by reference and to which particular reference may be had.

As more fully developed in the aforementioned SAE article, there are two sources of the discrete frequency compressor noise, consisting of the fundamental blade-passage frequency and the harmonics of the first one or two stages. These sources are (1) the rotation of the periodic pressure field of the blades operating in uniform flow, and (2) the periodic cutting by the rotor of flow irregularities produced by stator vanes or the periodic impingement of rotating blade wakes on stator vanes.

Both sources produce what have been called spinning modes which are periodic pressure patterns that spin about the shaft axis. However, patterns produced by the two sources differ in an important way. In the case of the first source; a rotor operating in uniform flow, the pattern associated with the order nB consists of nB lobes, or cycles of simple harmonic variation, rotating at exactly rotor speed N , where B is the number of blades and n is the harmonic being considered, with $n=1$ for the fundamental, $n=2$ for the first harmonic and so forth. Thus, the fundamental blade-passage frequency, BN , is generated by a B -lobe pattern turning at rotor speed, the first harmonic, $2BN$, is associated with a pattern having $2B$ lobes turning at rotor speed, and so forth. As fully explained in the aforementioned SAE article, rotor-created spinning modes can be caused to decay rapidly by operating the rotor below cut-off speed according to the formula:

$$\frac{k'(\sigma)}{m\mu} > \frac{n}{B}$$

where $m=nB$, n =harmonic index, B equals number of blades, and

$$k'(\sigma)$$

is a characteristic number.

The patterns produced by the second noise source, that is, cutting of stationary wakes by rotating blades or impingement of rotating wakes on stationary vanes are quite different. If there are V stationary vanes and B rotating blades, the noise consists of a superposition of patterns associated with orders nB , having m lobes given by the relation: $m = nB + kV$ (1), where k is an index that takes on all positive and negative integers. In order to generate the order nB , each of these m -lobe patterns turns at a speed different from that of the rotor speed N , namely:

$$\frac{nB}{m} \times N \quad (2)$$

The understanding of some of these patterns may be made more plausible by means of a simple demonstration involving reasonably few blades and vanes. FIGS. 1 through 7 illustrate in diagrammatic form the case of an 8-bladed rotor interacting with a 6-vaned stator. Eqs. 1 and 2 predict the existence of an $8-6=2$ -lobe pattern, turning at $8/2=4$ times shaft speed for the $8N$ frequency. The rotor blades are represented by eight spokes, such as 2, one of which 4 is marked with a large dot for identification of rotor position. The six stator vanes, such as 6, are schematically shown as short radial lines extending inward from the duct wall 20. In FIG. 1, the marked blade 4 coincides with the right-hand stator vane, and a similar coincidence exists on the opposite side. The coincidences and directions of motion are identified by exterior arrows, and the positive portions of a 2-lobe pattern are shown by the shaded areas. FIGS. 2 through 7 show intermediate positions of the rotor (follow identifying mark) and the coincidences as the rotor makes $1/4$ of a turn. Notice that in this time interval, the 2-lobe pattern has completed one whole revolution, and is therefore turning at a rate of 4 times the rotor speed.

Similarly, FIGS. 8 through 17 represent the interaction of an 8-bladed rotor, such as 8, with a 9-vaned stator, such as 9. Here, using Equation 1, the expression for the smallest value of m gives $m = 8-9 = -1$. This result is interpreted as a 1-lobe pattern rotating in a direction opposite to that of the rotor. Using Equation 2, we find that its speed is eight times shaft speed. A glance at the diagrams in FIGS. 8 through 17 shows that a complete backward revolution of the 1-lobe pattern (identified by a single coincidence arrow) is accomplished in the time required for the rotor to make but $1/8$ of a turn.

Now it is a further basic result that the noise associated with each of these patterns, whether produced by the rotor only or by rotor-stator interaction, behaves in one of two distinctly different ways in its passage to the compressor duct entrance or discharge. If the pattern sweeps the duct wall at a tangential velocity such that its wall Mach number, M_m , is less than a critical value, M_m^* , the intensity of the pattern will decay exponentially with distance from the source. Thus, the pressure variations at the face of the duct will be extremely small and will not radiate significant noise away from the compressor. If, on the other hand, the tangential Mach number of the pattern is equal to or exceeds the critical Mach number, M_m^* , the pattern will propagate through the length of the duct with undiminished intensity and will radiate from the duct face into free air. In noise parlance, the point of "cut-off" is said to separate the "decay region" from the "propagation region" and "cut-off" occurs when the tangential Mach number of the lobed spinning mode is equal to the critical Mach number M_m^* . Actually, propagation would exist at "cut-off" and hence, to have decay it is essential to maintain lobe tangential mode Mach number below "cut-off." Appendix B, which will be found immediately following the description, provides a convenient table for the critical tip-Mach number M_m^* .

It is an immediate consequence of this result, as has

been experimentally confirmed, that the noise produced by a ducted rotor operating at subsonic tangential tip speed in uniform flow, decays exponentially with distance from the rotor and is not heard outside of the duct. This is completely different from the case of a conventional propeller, the noise characteristics of which are well known, and accounts for the unsuccessful attempts to explain compressor noise with the principles set forth in propeller theory.

Therefore, since from the preceding it will be seen that the spinning modes generated by the compressor rotor can be controlled so as to decay by the control of rotor speed, the only important noise sources in subsonic compressor operation are those associated with rotor-stator interaction patterns. Through knowledge of the behavior of these patterns it is possible to design a compressor so as to exploit the properties of rotor-stator interaction noise with respect to minimizing its effects on an observer located on the ground as the airplane flies by overhead.

We have found four areas of the compressor noise problem over which control may be exercised by proper design procedures. Each of the four areas will be discussed in greater particularity hereinafter but will now be mentioned in chronological order:

(1) The noise we are concerned with is first generated by the rotor cutting through flow nonuniformities such as wakes of upstream guide vanes or stators or by the impingement of rotor wakes on downstream stators.

(2) The next stage is the transmission of the generated pattern through the duct to the entrance and discharge planes.

(3) The pressure field at the duct face will radiate sound away from the compressor in a directivity pattern which can be computed from the results of our analysis. It will be apparent that the first three mentioned areas of noise control are equally applicable to the marine propeller noise problem.

(4) The final phase is the reaction of the typical human observer to the radiated compressor noise as the airplane flies by overhead, wherein such factors of the reaction as intensity level, frequency, quality and duration must be considered.

It is our objective to teach how to design or reoperate a compressor to control noise by affecting one or more of the above areas. There are five means by which such control may be exercised and these means may be used singly or in any combination as circumstances may direct. After summarizing these five means, detailed description of each will be presented. Finally, sample design or reoperation procedures will be described.

The five means for controlling compressor noise are:

(1) Reduce the strength of the source of rotor-stator interaction by axially separating the stator from the rotor so that the wakes are appreciably diffused.

(2) Exploit the phenomena of decay by selecting the number of rotor blades and stator vanes such that the modes associated with one or more harmonics of interaction noise are spinning at less than their critical tip Mach number, M_m^* . Conservatively, for high Mach number conditions as encountered in aircraft turbo-jet engines, it can be said that the number of vanes, V , should be equal to or greater than twice the number of blades, B , times the harmonic index n . For low Mach number conditions, such as are encountered in marine applications, it will be shown how this result may be achieved with a much smaller number of vanes.

(3) Where there are two or more sources of propagating interaction noise at the same frequency, select the blades and vanes such that these modes have the same number of lobes and comparable radial intensity distributions. Then index the sources by adjusting angular and axial positions of the blades and/or the vanes so that the relative phase relations of the spinning modes yield a zero

or minimum sum. This procedure is called cancellation by indexing.

(4) Select blade-vane combinations so that a propagating mode is obtained with a radiation directivity pattern that reduces the exposure time on the ground as the airplane flies by overhead.

(5) Select blade-vane combinations so that propagating modes are associated with frequencies above the drop in ear sensitivity indicated on the Fletcher-Munson curve between 4000 and 7000 c.p.s. An added advantage to increasing the noise frequency is that the high frequency noise attenuates in air more rapidly than noise of low frequency.

It will be evident that the five above-mentioned noise control means are fully applicable to the marine propeller noise problem as well as to the aircraft turbojet engine problem.

MEANS FOR CONTROLLING COMPRESSOR NOISE

Referring to FIG. 18 we see a typical turbo-jet engine 10 which consists of a compressor section 12, a burner section 14, and a turbine section 16 in axial alignment along central axis 18. The compressor, burner and turbine sections are preferably housed within engine outer case 20, which is usually of circular cross section and concentric about axis 18 and which includes compressor inlet aperture 22 and exhaust outlet aperture 24 at its opposite ends. Air enters compressor 12 and is compressed therein before being discharged into burner section 14, where it is heated. The heated air then passes through turbine 16, and has enough energy extracted therefrom so that turbine 16, can drive compressor 12, and is then discharged through exhaust outlet 24 to atmosphere to produce thrust. Engine 10 may be of the type described in greater particularity in U.S. Patent Nos. 2,711,631 and 2,747,367, which are hereby incorporated by reference.

FIG. 19 shows a side view, partially broken away, of a shrouded marine propeller 11 whose propeller shaft 15 extends from the hull 17 of the marine vessel. Shrouded propeller comprises a cylindrical shroud 20' having spaced and fixed stators including stationary vanes 50' and 52' and also having spaced, rotating rotors including rotatable blades 94' and 96'. The rotor and stator elements of the shrouded propeller 11 operate, in the same fashion as previously described for a compressor of a turbojet engine to drive a stream of water through the shroud 20' for marine vessel propulsion purposes. Primed reference numerals corresponding to those shown in FIG. 20 have been used in FIG. 19 to show the similarity between the shrouded marine propeller application of FIG. 19 and the axial flow compressor for a turbojet engine shown in FIGS. 18 and 20.

As previously mentioned, since both air and water are fluids, they have similar wave transmission characteristics and the propagation of acoustic energy through air is similar to the propagation of hydrodynamic waves through water. The radiated sound field of a free propeller consists of energy concentrated at the fundamental blade-passage frequency and its harmonics. There are two sources of excitation for this sound:

(1) The rotating thrust and drag forces of the blades in a uniform fluid flow.

(2) The intermittent cutting by the blades of any non-uniformities in the flow, such as hull wakes. This source is probably the more important from a practical standpoint.

These two sources always radiate sound away from the propeller.

When the propeller is enclosed in a duct, however a completely different phenomenon occurs. These two sources of excitation are still present, but before the sound energy can radiate away to large distances, it must first travel in the duct to the fluid entrance and exit planes. During the course of this duct travel, this sound may be caused to decay to comparatively negligible levels by the time it reaches the end of the duct. Only a small portion

of the original signal, therefore, will be radiated. The details of the conditions which produce this decay of sound in the duct are the same in a shrouded marine propeller as in a turbojet engine. The noise reduction problem in a shrouded marine propeller will be easier to solve in some respects because the marine propeller rotates at a much slower rate of speed, r.p.m., than does the rotor assembly of a turbojet aircraft engine, and because sound travels in water about five times as fast as it travels in air.

Returning to the consideration of engine 10, compressor section 12 is shown in greater particularity in FIG. 20 and comprises basically a plurality of stationary stators 30, 32, 34 and 36 and a plurality of rotatable rotors 40, 42 and 44 positioned alternately along axis 18 and concentrically thereabout within compressor or engine case 20. Rotors 40-44 are assembled in a fashion to be hereinafter described to form rotor unit or assembly 45. Each of the stators 30-36 includes a plurality of stationary vanes 50, 52, 56 and 58 which are radially extending and equally spaced in circumferential position about axis 18. Each of the vanes is supported between outer shrouds such as 60, 62, 64 and 66 which are supported in any competent fashion such as bolts 68 in case 20 and inlet shrouds such as 70, 72, 74 and 76, which may be supported either by the outer shrouds through the vanes or, as in the case of inner shroud 70 to support web 80 which serves to position forward compressor bearing 82 from the engine case 20. Rotors 40-44 are rotatably mounted as rotor unit 45 on shafts such as 84 and bearings 82, at the opposite ends thereof, and connected by thru-bolts 86 which extend through the discs 88, 90 and 92 of the rotors 40-44 and through spacer rings such as 95, 97 and 99. Each of the rotors 40-44 includes a plurality of radially directed and equally spaced circumferentially positioned blades, 94, 96 and 98, which include tip ends such as 100, which pass adjacent to the outer shrouds 60-66 and inner ends or root sections such as 102, which attach either integrally or by a conventional blade root attachment means to the rotor hubs 88-92.

The air which enters the compressor inlet 22 passes through annular gas passage 104, which is defined between the compressor or engine case 20 and the stator inner shrouds 70-76 and the rotor hub tips 106, 108, and 110, which are positioned on the periphery of hubs 88-92 and serve to support rotor blades 94-98. The air passing through annular passage 104 passes alternately through the inlet guide vanes or first stage stator vanes 50, the first stage rotor blades 94, and second stage stator vanes 52, the second stage rotor blades 96, the third stage stator vanes 56, the third stage rotor blades 98, the fourth stage stator vanes 58 and so on until it is eventually discharged into burner section 14.

With the foregoing compressor construction in mind, the control of compressor noise will now be discussed in detail.

The strength of the pressure pulse resulting from a blade cutting through the wake of a vane depends on the magnitude of the variation in angle of attack on the blade as it cuts the wake. It has been found possible to minimize the wake effects by separating the blades from the vanes axially to allow the vane wakes to diffuse to some extent before the blades cut through them. As the space between the blades and vanes is increased the noise level produced by the blades cutting vane wakes drops rapidly at first then almost levels out as shown in FIG. 21. Thus, there is an optimum spacing beyond which the noise reduction per unit increase in engine length is small. It will be noted by viewing FIG. 21 that very little noise reduction is attained by separating adjacent rotors and stators a greater distance than five chord lengths of the wake-creating airfoil and accordingly this distance is suggested for optimum yet practical spacing. FIG. 21 further illustrates that during the initial stages of stator-rotor separation, between points 31 and 33, both the intensity of the spinning mode pattern and the phase thereof

change. In the latter stages of stator-rotor separation, between points 33 and 35, intensity change is no longer encountered but phase change only is encountered and hence pattern phase change can be obtained in this latter region by axially spacing an adjacent stator and rotor.

The elimination of inlet guide vanes 50 results in a reduction in noise level beyond that achieved by moving them away from the first rotor blades 94. Thus, in engines where it is possible to support the rotor assembly 45 means of a bearing supported from case 20 through second or third stage stators 32 or 34, this will be found beneficial provided the noise produced by other rotor-stator interactions is not limiting.

Accordingly, it will be seen that by axially spacing adjacent stators and rotors, the strength of the noise source can be reduced and that, in instances where the inlet guide vane or first stage stator vanes 50 can be eliminated by supporting the compressor rotor 45 by other means or by retracting the vanes 50 during the approach condition, no spinning mode noise due to interaction upstream of the first stage rotor 40 can then be established.

We have also found that it is possible to design a compressor so that the sound field which is produced by the interaction of vane wakes with a rotor, or rotor wakes with a vane array, will be of such composition that it will decay in its passage in the compressor duct or case 20, instead of propagating to the face 22 of the duct and then radiating away from the compressor 12.

It has been found in our study that rotor-stator interaction noise consists of harmonics of blade-passage frequency. Acoustic energy is in the form of spinning modes consisting of m cycles of circumferential pressure variation, the calculation of values of m to be described hereinafter. Any one pattern with m lobes turns at Mach number:

$$M_m = \frac{nB}{m} M_B \quad (3)$$

where n =harmonic index=(1, 2, . . .), B is the number of rotor blades and M_B is the rotor tip Mach number. Now it can be shown (see Appendix B), that there exists a critical Mach number, M_m^* , such that if the m -lobe pattern sweeps the duct walls at a lower value of Mach number than M_m^* , the associated pressure level decays exponentially with distance in the duct as follows:

$$P(x) = A_m e^{-k_x x} \quad (4)$$

where

$$k_x = \frac{m}{b} \sqrt{M_m^{*2} - M_m^2}$$

b =duct outer wall radius

x =distance along duct axis

A_m =arbitrary amplitude coefficient

In decibel units this decay may be expressed as a decay rate:

$$\frac{db}{x/b} = 8.69m \sqrt{M_m^{*2} - M_m^2} \quad (\text{decibels per outer wall radius}) \quad (5)$$

Values of this decay rate for selected values of m and the hub-tip ratio of 0.25, 0.50 and 0.75 are given as an example in FIGS. 22-25. Values of M_m^* are given in Appendix B.

It will be noted that decay actually extends into the low supersonic region, and that all subsonic patterns decay at sufficiently high rates so that extremely large reductions are obtainable in reasonably short ducts.

It was previously pointed out that the field of a rotor in uniform flow produces a pressure field consisting of $B, 2B, \dots nB$ lobe patterns all travelling at exactly rotor speed. In the light of discussion of decay above, the field of a subsonic rotor in uniform, ducted flow will always decay and will not contribute to the overall radiation. In the case of interaction noise, however, in the light of Eq. 3, m -lobe patterns with sufficiently high supersonic

wall Mach number M_m and which thus will not decay can be created by a subsonic rotor.

An essential feature of our teaching concerns means for designing the compressor such that one or more harmonics of the interaction noise will be in a pattern that decays. This requires that the following relation hold:

$$|m| \geq nB \frac{M_B}{M_m^*} \quad (6)$$

Considering the landing approach condition, if the harmonic, nB is to decay, the rotor tip Mach number, M_B , during the landing approach multiplied by the harmonic index n and the number of rotor blades B , divided by the critical Mach number M_m^* must be less than the number of lobes, m .

As stated previously, we have found that the number of lobes m in the interaction pattern is given by the formula:

$$m = nB + kV \quad (1)$$

where n is the harmonic index, B the number of rotor blades, k is an index that ranges independently over all positive and negative integers, and V is the number of stator vanes. If the number of rotor blades is given, along with the rotor tip diameter and the approach r.p.m., the number of vanes can be selected so that for any desired harmonic the least absolute value of m in Formula 1 will be equal to or greater than the decay requirement.

These principles of blade-vane selection are illustrated in graphical form in FIGURE 26. A coordinate system is set up with units nB along the ordinate, line 500, and m along the abscissa, line 501, both to the same scale. The curve $nB = (M_m^*/M_B)m$ is then plotted, for a selected blade tip Mach number, in this coordinate system. In FIGURE 26 the curve labeled 503 is drawn for positive values of m and curve 504 is drawn for negative values of m with $M_B = .5$. The graph can be divided into two regions with the aid of the relation $|m| \geq nB (M_B/M_m^*)$, which must hold for decay of the harmonic nB . Regarding curve 503, for a particular value nB all points to the right of the curve represent values of m greater than $nB(M_B/M_m^*)$ or decaying modes and points which lie between curve 503 and the vertical axis, line 500 yield propagating m -lobe patterns since these values of m are less than $nB(M_B/M_m^*)$. Similar considerations apply to curve 504 and the area of the graph included between curves 503 and 504 is called the propagation zone while the area outside the curves is a region of decay. With the addition of the 45° line, $m = nB$, line 505, the relation $m = nB + kV$ may be solved graphically for each harmonic. Consider first, the fundamental blade passage frequency for which $n=1$. The m -lobe patterns generated are found by adding and subtracting multiples of the number of vanes V from the number of blades B . This is done graphically by starting from the point (B, B) , point 506, on line 505 and laying off, successively, values of the number of vanes V on the horizontal line 502. The resulting points are labeled 508, 509 and 510 and represent the m -lobe pattern generated as the index k in $m = B + kV$ takes on the values $-1, -2$, and $+1$ respectively. Notice that all of these points lie only in the region of decay, thus all of the m -lobe patterns generated by the fundamental will decay in this example. For the first harmonic ($n=2$) the procedure is similar but the starting point becomes $(2B, 2B)$, point 511. Here the first subtraction of V vanes from $nB=2B$, ($k=-1$), yields point 512 which lies within the propagation zone. Hence the first harmonic of blade passage frequency generates a propagating m -lobe pattern.

As the higher harmonics are considered notice that the propagation zone becomes wider and the number of propagating m -lobe patterns represented by points within the propagation zone increases. If it is desired that a given harmonic nB decay the number of vanes must be selected so that there are no resulting m -lobe patterns within the propagation zone or the value of V must be chosen to span the propagation zone.

The boundaries of the propagation zone in FIGURE 26 are defined by the relation

$$\frac{nB\bar{M}_B}{m} = M_m^*$$

so that a point just outside the zone represents a mode that is close to cutoff and may not be decaying rapidly enough to be appreciably attenuated. If a slightly narrower zone is selected by replacing M_m^* (which is always larger than one) with unity, then points which are even right on the new boundary represent modes that have substantial decay rates (see FIGURES 22-25 where $M_m=1$). The new relation $nB\bar{M}_B/m=1$, being linear, allows a simple diagram to be constructed for the operating blade tip Mach number, $\bar{M}_B$ so that vane selection may be made on the basis of certain direct rules which will be evolved from the following considerations.

The problem to be solved is the selection of the least number of vanes required to decay all harmonics up to and including a certain n of a rotor operating up to a maximum tip Mach number, $\bar{M}_B$. FIG. 26a is drawn for a rotor operating at $\bar{M}_B=1/2$, the same as in FIG. 26, and the prohibited zone is bounded by the lines $m=\pm nB\bar{M}_B=\pm 1/2 nB$. The term prohibited zone is used to denote that m must be outside thereof to insure effective decay. It is immediately obvious that to decay frequencies up to and including nB , V must be selected so that the value of m generated by $m=nB-V$, that is, the first m on the horizontal line nB to the left of the line $m=nB$ must lie on or to the left of the left-hand boundary of the prohibited zone. That is, V must exceed nB by at least the half-width of the zone or $V \geq nB(1+\bar{M}_B)$. If, for example, V is selected so that the first m given by $m=nB-V$ lies to the right of the zone, then it is clear that the next m , given by $m=nB-2V$ will land inside. This circumstance is a consequence of the width of the prohibited zone. If the zone is sufficiently narrow, however, it is equally obvious that, for some n , a much smaller V may be selected than implied by $nB(1+\bar{M}_B)$ so that successive values of m , which are separated by V , skip right across the prohibited zone. Since low blade Mach numbers exist in many applications, such as ventilating fans, and in ducted propellers or jet pumps for marine propulsion, where the shaft speed is lower than in aircraft and the speed of sound in water is about five times the speed of sound in air, it is an important objective of this invention to show how the least number of vanes may be selected.

FIG. 26b is drawn for a tip Mach number, $\bar{M}_B=1/4$. The boundaries of the prohibited zone for $n=1$ and 2 have been projected at 45° angles and are called projected zones. This permits viewing when a vane selection being made with respect to a particular value of n would fall into the prohibited zone for some lower value of n . Consider the problem of selecting V to decay modes associated with both $n=1$ and $n=2$. From the FIG. 26b it can be seen that V must be chosen to make $m=2B-V$ fall between the right-hand border of the prohibited zone, pt. a , and the left-hand border of projected zone 1, pt. b . Of course, V could be large enough to span the zones in one step, $V \geq 2B(1+\bar{M}_B)$, which has been already discussed, but a significant reduction in number may be had by making V lie between $B(1+\bar{M}_B)$ and $2B(1-\bar{M}_B)$ with the same result. Other values of V will produce propagating modes for $n=1$ or $n=2$ or both. It will be noticed that this procedure breaks down for n larger than 2, for projected zone 2 runs into the prohibited zone boundary on the $n=3$ line. It can also be seen from FIGURE 26b that if the Mach number is increased the prohibited zone and its projections widen with the consequence that pts. a and b move toward each other and the selection of V

becomes increasingly restricted. When $\bar{M}_B=1/2$ the points a and b coincide and only one value of V can be used:

$$V = \frac{4B}{3}$$

Hence $\bar{M}_B=1/2$ is the upper limit of Mach number which allows a reasonably small number of vanes to be used where both $n=1$ and $n=2$ interaction modes must decay.

As the Mach number is reduced it becomes possible to pick V so that additional harmonics decay without use of a large number of vanes. FIG. 26c is drawn for $\bar{M}_B=1/6$ and shows how the $n=3$ and lower orders may be caused to decay. A case as illustrated where V has been selected so that the first two values of m lie between projected zones and the third is to the left of the prohibited zone. The criterion may be formulated that the second value of $m=3B-2V$ for the $n=3$ order must lie between points c and d . That is: $2V$ must lie between $2B(1+\bar{M}_B)$ and $3B(1-\bar{M}_B)$. Hence V must satisfy

$$B(1+\bar{M}_B) \leq V \leq 3/2 B(1-\bar{M}_B)$$

The upper limit of $\bar{M}_B$ for which this procedure applies, that is when points c and d have moved toward each other and coincided may be found by equating the upper and lower limits of V to be equal to $1/5$.

By similar reasoning, it may be perceived that as the Mach number is sufficiently reduced, it will be possible to decay the $n=4$ and lower orders by picking V to that m makes 3 landings between projected zones before skipping across the prohibited zone on the $n=4$ line, and that this situation is possible up to a Mach number of $1/7$. These results may be summarized as follows:

If $\bar{M}_B > 1/3$ $V \geq nB(1+\bar{M}_B)$ is required to decay harmonic indices up through n . If $1/5 < \bar{M}_B \leq 1/3$ V must satisfy $B(1+\bar{M}_B) \leq V \leq 2B(1-\bar{M}_B)$ to decay $n=1,2$. If $1/7 < \bar{M}_B \leq 1/5$ V must satisfy $B(1+\bar{M}_B) \leq V \leq 3/2 B(1-\bar{M}_B)$ to decay $n=1,2,3$. If $1/9 < \bar{M}_B \leq 1/7$ V must satisfy $B(1+\bar{M}_B) \leq V \leq 5/4 B(1-\bar{M}_B)$ to decay $n=1,2,3,4$, and generally, if

$$\frac{1}{2n+1} < \bar{M}_B \leq \frac{1}{2n-1}$$

V must satisfy

$$B(1+\bar{M}_B) \leq V \leq \frac{n}{n-1} B(1-\bar{M}_B)$$

to decay all harmonics up thru n .

If a higher harmonic index n than can be accommodated by the above $\bar{M}_B$ limitations must be made to decay, then V must satisfy $V \geq nB(1+\bar{M}_B)$ in order to decay all harmonics indices up thru n .

For sufficiently low $\bar{M}_B$ it may not be possible to fit an integer V between the given limits. In this case it is clearly advantageous to adjust the value of B until an integral V can be found.

Returning to consideration of high Mach number application, examination of the FIG. 26 chart shows that a larger number of vanes is required than conventionally used. While the number of vanes required to decay high harmonics of blade passage frequency is too large to be practical, it is practical to design stators to decay the fundamental blade passage frequency in conventional rotors, and, in some cases, the first harmonic as well. It is important to note here that higher harmonics of blade passage frequency have lower noise levels, thus there is less need for these to decay. A typical far field spectrum which would result from a conventional compressor designed by these methods is shown in FIG. 27 where the dashed spikes indicate that these frequencies have decayed in the inlet duct.

It is not feasible, however, to design inlet guide vanes for aircraft jet engines in this manner. This is due to the conflicting requirement that the inlet guide vane assembly must allow birds to pass on through so that they may be ingested by the engine, i.e., be chopped up by the

first rotor 40 so that they do not restrict inlet airflow. An assembly of a large number of vanes 50 before the first rotor 40 would be a serious obstacle to bird ingestion and could not be tolerated. However, the swirl produced by the inlet guide vanes is needed only in the blade tip regions. A construction involving the required large number of vanes to yield decay, but cantilevered from the inlet duct wall extending part way to the hub would suit the plural requirement. The vanes 50 are slanted aft such that ingested objects 51 do not lodge on the vanes, but slide along them and enter the rotor 40 to be chopped up. This type of construction is illustrated in FIG. 28.

An alternative construction involves cantilevered, part-span guide vanes which protrude inward radially from the duct outer wall, the protrusion being only the fraction of hub-tip radial distance required to prevent rotor stall at the tip region. These vanes would be actuated to retract into the inlet cowl wall during the low power regime of the engine and would automatically be controlled to swing into the inlet air stream to turn the air during high power operation. This turning is not required by the rotor during the low power landing approach condition, which is the important phase of flight operation from the standpoint of minimizing compressor noise effects on the airport neighborhood. A typical construction of the retractable vanes is shown in FIG. 29.

As stated previously, the pressure field produced by rotor-stator interaction is composed of a superposition of m -lobe patterns rotating at various speeds. We have found that all but one or two of these modes, associated with the n th harmonic, decay as they travel through the compressor duct 20. It is then possible to eliminate the n th harmonic from the far field spectrum by indexing. As used herein, the term indexing means moving stators or rotors relative to one another either circumferentially, axially, or in some compound motion therebetween such as helically. The pressure field of the m th propagating mode is given by:

$$P(\theta, x, t) = a_m \cos [m\theta + k_x x - \omega t + \phi_m] \quad (7)$$

$$= a_m \cos [m(\theta - \Omega_m t) + k_x x + \phi_m] \quad (8)$$

where p is pressure, θ is the angular coordinate in radians, x is the coordinate in the direction of the compressor case axis, t is the time coordinate, a_m is the amplitude of the m th mode, m is the number of lobes or cycles of circumferential pressure variations, k_x is the wave number as used in equation:

$$k_x = \frac{2\pi}{c} \sqrt{f^2 - f_m^2} \quad (9)$$

ω is the circular frequency which is equal to 2π times the frequency in c.p.s., ϕ_m is the phase of the m th mode and Ω_m is the angular velocity of the spinning pattern, f is frequency in c.p.s. and f_m is cutoff frequency for an m -lobe pattern.

A developed view of this pressure field of the m th propagating mode is shown in FIG. 30 with the direction of the rotor as indicated by the arrow, and with the ridge lines of maximum pressure indicated as 110, 112, 114, while the ridge lines of minimum pressures are indicated by dotted lines 116, 118 and 120. The pressure field fluctuation as illustrated graphically by line 122, is plotted along the x or compressor axis 18 coordinate as represented by line 124 and along the S or in the circumferential direction as illustrated by line 126. Symbols are hereby provided to more completely illustrate FIG. 30 to those skilled in the art and include c , which is the speed of sound; λ , which is the wave length; f , which is the frequency in c.p.s.; M , which is Mach number; M_s , which is the Mach number of the pattern in the circumferential direction; r_0 , which is the wall radius; λ_s , which is the wave length in the circumferential direction.

With suitable pressure probes capable of measuring instantaneous pressure, it is possible to ascertain the position, the magnitude and the speed of rotation of each

of the spinning modes in the fashion more fully described in the aforementioned SAE article.

It will be noted that the maximum and minimum pressure ridge lines 110-120 and hence the spinning mode or wave train moves in a direction perpendicular to the ridge lines as indicated by arrow 130 with the speed of sound, c , the distance between crests being the wavelength, λ , associated with frequency, f , namely, $\lambda = c/f$. This wave train has traces in the x direction and in the circumferential s direction, the wavelengths and velocities thereof are shown by subscripts x and s respectively.

The field consists of a wave pattern moving in a spiral path in the compressor annulus passage 104. In the developed view shown, the field appears as a parallel wave train propagating in a direction inclined to the coordinate axis x . Sections of the field are taken along the axial and circumferential axes and in the propagation direction. It will be noted that in the propagation direction, the wave velocity and wave length are those obtained in free field.

The pressure field for the FIG. 26 developed view of the m th propagating mode may be expressed as follows:

$$p(s, x, t) = a_m \cos \left[\frac{2\pi}{\lambda_s} s + k_x x - \omega t + \phi_m \right] \quad (10)$$

where s is the circumferential arc coordinate, k_x is the wave number (a separation constant in the solution of the wave equation), and λ_s is the circumferential wave length.

It will be noted that the cosine term contains a phase angle, ϕ_m . Mathematically, the value of the phase angle is completely arbitrary. In actual compressors the value of ϕ_m is measured relative to some fixed point of reference on the shaft. This measured value of the phase angle depends on the position of the stator vanes relative to a fixed point in the duct. If the stators are rotated in the duct, the phase angle will change in a known way. We have found that as the stator is rotated one vane space, the phase angle ϕ_m will change by $2\pi k$ radians, where k is the index which generated the m -lobe pattern under consideration from Eq. (1). In terms of FIG. 30, rotation of the guide vanes 50 is equivalent to sliding the diagram along the vertical or s axis $2\pi k$ radians per vane space of rotation, for example, as best shown in FIG. 32, from point A to B.

If there are two sources of noise at the same frequency and both produce one propagating spinning mode having the same number of lobes, the resulting fields can be made to cancel each other. This process of cancellation is illustrated in FIGS. 31 and 32. In FIG. 31 the field of the two propagating modes of the same number of lobes and the same frequency are shown superimposed in a developed view. A field of this nature would be created from the interaction of rotor blades 94 with inlet stator vanes 50 and exit stator vanes 52, both having the same number of vanes.

It would also be possible to produce the same number of lobes in two interactions if the number of vanes in one stator were a multiple of the number of vanes in the other, or, if the number of stator vanes stood in the ratio of two small integers. For example, with 31 rotor blades and 30 vanes in one stator, a one-lobe propagating mode would be produced, and if the number of vanes in the second stator were 15, this interaction would also produce a one-lobe pattern according to the formula $m = 31 - 2 \cdot 15$. Also, if one stator had 10 vanes and the other had 15 then each interaction would again have a single lobe as given by $m = 31 - 3 \cdot 10$ in one case, and $m = 31 - 2 \cdot 15$ in the other.

The maximum pressure ridge lines for the inlet stator generated spinning mode are indicated as lines 110 and 112, as in FIG. 30, while the minimum or maximum negative pressure wave ridge lines are shown as 116 and 118, as in FIG. 30. Corresponding maximum pressure wave ridge lines for the exit stator generated spinning modes

are shown as lines 132 and 134 in FIG. 31, whereas the corresponding maximum negative pressure ridge lines are shown at 136 and 138. Reference point 111 on line 110 is provided to aid in following the wave movement in cancellation. FIG. 31 shows developed views of the similar two modes both along the x axis and along the s axis and the same reference numerals are used for identification purposes. As used herein, the term similar modes or patterns, is used to mean modes or patterns having the same number of lobes m , and which rotate in the same direction.

If the inlet guide vanes 50 are then rotated circumferentially relative to the fixed exit stator vanes 52 the two wave patterns change position relative to each other. When the inlet guide vanes 50 are positioned such that the waves are 180° out of phase they cancel to yield a zero or minimum sum. This cancellation by indexing is best illustrated in FIG. 32 wherein reference point 111 on line 110 has been moved from position A as in FIG. 31 to position B by rotating the inlet guide vane 50 and lines 110 and 112 have moved in direction s and are superimposed upon lines 136 and 138, and lines 116 and 118 have also so moved to become superimposed upon lines 132 and 134. This wave cancellation is best illustrated in the projection shown on the x and s axes.

Complete cancellation as shown in FIG. 32 is possible only when the modes from the two sources are exactly equal in strength at all radial positions. This is not always true in actual compressors. The radial distribution of pressure depends on the aerodynamic features of the interaction which are difficult to analyze theoretically. However, a minimum resultant pressure can always be obtained by indexing.

The field of four propagating modes from two sources is illustrated in FIG. 33. This field would be produced, for example, by a 7 blade rotor interacting with 5 inlet guide vanes and 5 exit stators to produce a two-lobe mode, and a three-lobe mode. Each of these two interaction sources produce both a forward rotating two-lobe pattern with $k=-1$ and a backward rotating three-lobe pattern with $k=-2$ for the fundamental blade passage frequency, $7N$. Equation 1 has been used to determine the number of lobes and the direction of the rotation thereof. The unrealistic numbers of vanes and blades have been chosen only to simplify explanation and illustration; the technique to be discussed is applicable to any physically realizable combination of blades and vanes which produces two propagating modes at the same frequency from each of the two sources.

In FIG. 33 the lines of constant phase for the two-lobe patterns are shown as the heavy lines, with the maximum pressure ridge lines shown as 140, 142, 144 and 146, while the minimum pressure ridge lines are shown as 148, 150, 152, 154 and 156, while the three-lobe patterns are represented by the finer lines, where the maximum pressure ridge lines are shown at 160, 162, 164, 166, 168 and 170, while the minimum pressure ridge lines are shown as 172, 174, 176, 178, 180 and 182. There is an obvious difference in the slope of the phase lines for the two and three-lobe patterns. This is because in addition to rotating in opposite directions, both patterns must produce the same frequency, $7N$, so that the two lobe mode must have a higher rotational speed or smaller spiral angle than the three-lobe mode.

It is possible to cancel either one of the two modes created by blade 94-vane 52 interaction by simply rotating the inlet stator vanes 50, as previously discussed, to give a 180° phase shift between the two waves. In FIG. 34, the inlet stator vanes 50 have been rotated so as to produce a change in phase of ξ radians, of the two-lobe mode and causes it to cancel with the two-lobe mode from the exit stator vanes 52. This is equivalent to sliding lines 142, 152, 146 and 156 ξ radians in the direction of increasing S until they become superimposed on the lines 150, 144, 154 and 140 respectively.

Reference points 190 and 192 are used on the two-lobe mode and the three lobe mode respectively, for ease in following the successive steps. In FIG. 33 reference point 190 on line 142 is at position A_0 and 192 on line 170 is at position A_2 . In FIG. 34 reference point 190 on line 142 has moved ξ radians to position B_0 and 192 on line 170 has moved 2ξ radians to position B_2 . The three-lobe mode generated by the inlet stator vanes 50 has undergone a change in phase of 2ξ radians, since k is -2 for this mode, as a result of the vane rotation. The net result of this first step has been only to cancel the two-lobe patterns while the three-lobe modes still produce a net resultant pressure as FIG. 34 illustrates. FIG. 34 illustrates this since lines 176, 162, 172, 170, 180 and 176, which are the positive and negative peak pressure lines of the three-lobe mode from the inlet stators 50, are not superimposed on the corresponding lines of peak pressures, 160, 182, 168, 178, 164 and 174 for the three-lobe mode from the exit stators 52.

In order to cause the three-lobe modes to cancel and to preserve the cancellation of the two-lobe patterns another technique must be introduced; that of axial indexing or axial positioning of the vanes in order to produce desired phase changes. We have found that shifting the axial position of the stators produces a change in the phase angle of $k_x x$, where k_x is the wave number. This is equivalent to sliding the diagram of phase lines for one source along the horizontal x -axis.

Returning to FIG. 34, if the axial position of the inlet guide vanes 50 is altered it is possible to cancel the three-lobe patterns, but in the operation the phase of the now cancelled two lobe pattern is not preserved. The only method by which it is possible to cancel the three lobe patterns and preserve cancellation of the two lobe patterns is to combine both axial and rotational indexing; i.e., index the inlet stator vanes 50 with a spiral movement, that is, in a motion having both an axial and a circumferential component. In FIG. 34 this is accomplished by sliding the diagrams diagonally along lines of constant phase for the two-lobe pattern, for example, by sliding point 190 along line 150 from position B_0 to position C_0 .

At the same time point 192 on line 170 moves from position B_2 to position C_2 and lines 176, 162, 172, 170, 180 and 176 associated with the inlet stators 50, become superimposed on the lines 160, 182, 168, 178, 164 and 174 associated with the exit stators 52, respectively, so that the three-lobe patterns are also 180° out of phase, solid and dotted lines become superimposed. The final result appears in FIG. 35 where all the modes have been cancelled to produce a zero or minimum net resultant pressure.

Again, complete cancellation is only possible if the radial distributions of the modes of each cancelled pair are equal. A minimum resultant can, of course, always be obtained.

Consideration will now be given to ways to reduce the effect of the discrete frequency noise radiated from an axial flow compressor so that the ground observer will be exposed thereto for a minimum time. FIG. 36 illustrates a cylindrical duct, 20 of radius b , representative of compressor case 20, with inlet 22. R is the distance from the center of the duct face, 22, to the observer and ψ is the polar angle measured from the duct centerline 18. As shown in appendix C of the aforementioned SAE article, we have found that it is possible to predict the radiation directivity for given compressor operating parameters by means of the formula:

$$P_{m\mu}(R, \psi, t) = \frac{B_{m\mu} e^{i(kR - \omega t)} J_m(k^1_{m\mu}) b k \sin \psi}{[J_{m-1}(kb \sin \psi) - J_{m+1}(kb \sin \psi)] R (k^2_{m\mu} - k^2 \sin^2 \psi)}$$

In terms of cutoff ratio,

$$\xi_{m\mu} = \frac{K}{K_{m\mu}}$$

The above equation can be written:

$$P_{m\mu}(R, \psi, t) = \frac{B_{m\mu} i (kR - \omega t) J_m(k_{m\mu}) b k \sin \psi [J_{m-1}(\xi_{m\mu} k_{m\mu} b \sin \psi) - J_{m+1}(\xi_{m\mu} k_{m\mu} b \sin \psi)]}{R k_{m\mu}^2 [1 - \xi_{m\mu}^2 \sin^2 \psi]}$$

The subscript, μ , denotes radial modes which will not be discussed in detail except to mention that associated with each m -lobe pattern is a superposition of μ -modes in the radial direction, the characteristics of which do not depend simply on the number of blades and vanes but on the aerodynamic details of the compressor geometry and the wall Mach number of the associated m -lobe pattern. The angular dependence of the radiation of an m -lobe mode is seen to depend only on parameters associated with compressor operation and the duct characteristic numbers, $k_{m\mu} b$, which are discussed and tabulated in the aforementioned SAE article by Tyler and Sofrin.

We have computed the constant radius directivity patterns for various m -lobe modes spinning at given Mach numbers, for a single stage, 32 blade compressor. The strong directionality of the field can be seen by examining FIGS. 37-40 where the field of the first radial mode $\mu=0$ is displayed in a sequence of diagrams for a one-lobe mode spinning at various Mach numbers. $\xi_{m\mu}$ is a useful parameter which is linearly proportional to the Mach number of the m -lobe mode, $\xi_{m\mu} = M_m / M_{m\mu}^*$. It will be noted that the radiation becomes focused toward the centerlines as the Mach number increases. Experimental data taken on a 10" diameter, single stage, 32 blade compressor model rig is shown in FIGS. 41 and 42. The experimental results, represented by dots, agree quite well with theoretical predictions, represented by solid lines, except in the region near 90° where some departure is evident and is the result of a simplifying assumption made early in the radiation analysis.

The constant radius results above afford a preliminary view of the radiation characteristics of axial flow compressors. In order to evaluate the noise received by ground observers as the airplane flies by overhead, theoretical exposure duration curves can be computed from the constant radius results. These curves are illustrated in FIG. 43. It will be noted that there is considerable decrease in the exposure duration time if the radiated energy is contained in a pattern having eight or more lobes as compared to, say, a two-lobe pattern.

It is therefore desirable to select blade-vane combinations such that propagating modes will be of eight or more lobes whenever possible.

This technique of reducing the effect of radiated noise on ground observers is extremely useful in modifying established compressor designs since it requires only small changes in the number of vanes to produce favorable results.

Consideration will now be given to the problem of selection of blade-vane combinations such that the compressor noise created will be at such a frequency that it will be less objectionable to the human ear. An interesting characteristic of the average human ear is its diminished sensitivity to frequencies above 7000 c.p.s. as shown in FIG. 44. Consequently, it would be a decided advantage from the standpoint of intensity if the lowest frequency heard from the compressor inlet 22 were in the range above 7000 c.p.s. In addition, such high frequencies attenuate rapidly in the air as shown in FIG. 45.

It is therefore desirable to select blade-vane combinations such that only frequencies above 7000 c.p.s. be allowed to radiate into free air.

Consideration will now be given to the manner of indexing stators and rotors with respect to one another both axially, circumferentially, and in some compound motion therebetween such as helically.

With respect to varying the axial position between adjacent stators or between adjacent rotors and stators it is suggested that as best shown in FIG. 20, adjacent vane supporting outer shrouds 60 through 66 have face splines

along their adjacent edges and are joined by spacer rings such as 200, 202, and 204, which have face spline rings around their forward and backward edges. Such a face spline connection between outer shroud 60 and outer shroud 62 through spacer ring 200 is shown at stations 206 and 208. As best shown in FIG. 46, face splines of shroud 60 interconnect with face splines of spacer ring 20. At the same time, face splines of spacer ring 20 interconnect with face splines 62. Face splines are a well-known mechanical device and are described in greater particularity in U.S. Patent Nos. 2,458,148 and 2,458,149. It will be obvious that by varying the length of spacer 200, either by substituting spacer rings of different axial dimension l or by fabricating spacer ring 200 such that it is actually made of two rings threadably connected as illustrated at station 210 in FIG. 46.

Such variation of spacer 200 is suitable to vary the axial distance between adjacent stator vanes 50 and 52 or between the rotor blades 94 with respect to either stator vanes 50 or stator vanes 52. It may be found necessary to use different length hub spacers such as 95 and 97 at the same time and to also use spacer washers such as 212 and 214 (FIG. 20) therewith to accomplish the desired axial spacing interrelation between the adjacent rotors and stators.

The use of face splines suits itself very well to circumferential indexing between adjacent stator vanes such as 50 and 52. The use of face splines, as will be evident by viewing the fragmentary developed view thereof shown in FIG. 47, permits circumferential repositioning of the adjacent outer shrouds such as 60 and 62 and hence the adjacent stator vanes such as 50 and 52. Such circumferential indexing of stators will not require rotor repositioning since there is no positive connection between the rotors and the stators.

Preferably, for minute circumferential indexing, it would be well to make face splines 206 and 208 of the differential variety. In such a construction, there are a substantial number of splines in both elements 206 and 208 but one of them includes an additional spline, for example, face spline 208 may include 100 teeth while face spline 206 may include 101 teeth. With such a construction, to obtain small increments of circumferential motion, stator 60 and spacer 200 would be moved forward one spline space in spline 208, then stator 60 would be rotated backwards one spline space in spline 206. In this fashion, stator 60 has been rotated a circumferential increment equal to one one-hundredth of a circle.

With respect to combined axial and circumferential indexing, the use of both the axial indexing and circumferential indexing methods described above, or other methods which will be evident to one having an ordinary skill in the art, will be required. For example, it might be desirable to join the O.D. of outer shroud rings 60-66, with or without spacer rings 200-204, to the I.D. of case 20 through mating splines or keyways extending in the direction of the spinning mode maximum and minimum pressure ridge lines 110-120 shown, for example, in FIG. 30.

Another possible construction would be to fasten the stator shrouds such as 60 and 62 to the outer case 20 by doweling or riveting and when indexing was desired, the stators could be moved to the desired new position and re-doweled or riveted to case 20.

EXAMPLES OF NOISE ABATEMENT TECHNIQUES

Two examples which illustrate the use of the methods previously described will now be provided. First, for a given number B of compressor blades, a shaft speed N r.p.s. can be chosen such that $2BN$, the first harmonic of blade passage frequency, is 7000 c.p.s. and hence is sufficiently high as to be unobjectionable to the human ear. It is then possible, in the manner to be described in greater particularity hereinafter, to choose a number of vanes that provides decay of the fundamental, $f=BN$.

The lowest frequency to be detected in the discrete frequency noise field of the compressor is then 7000 c.p.s.

A second example is analogous in some respects, but involves the additional technique of indexing. Again we consider a given number of blades B but now choose a shaft speed such that the second harmonic of blade passage frequency is 7000 c.p.s., so that, $f=3BN=7000$ c.p.s. and hence is less objectionable. The number of vanes is again chosen to provide decay of the fundamental. The remaining component of the discrete frequency noise below 7000 c.p.s., the first harmonic in which $f=2BN$, is then eliminated by indexing the inlet guide vanes, 50, relative to the exit stator vanes, 52, and hence the lowest frequency to be heard is again 7000 c.p.s.

It has been mentioned previously that propagating modes with a number of lobes greater than or equal to eight ($m \geq 8$), possess decided advantages with regard to duration time. While complete cancellation by indexing is theoretically possible, in practical applications imperfect speed control and differences in radial mode structure between the pressure patterns originating at the stators and the inlet guide vanes result in incomplete cancellation. In view of the radiation analysis, the residual acoustic energy due to incomplete cancellation should be contained in an eight or more lobed pattern to reduce its radiated effect. It would be beneficial if the order producing 7000 c.p.s. also satisfied this condition.

In the above suggested methods of choosing compressor components we have required only that the number of vanes be large enough to decay the fundamental. It is now of importance to investigate the m -lobe pattern obtained for the propagating mode of the higher orders of blade passage frequency. Should the m -lobe modes of selected higher orders be less than 8-lobe patterns, the difficulty can be resolved only at the expense of adding more vanes to satisfy the relation of Equation 1, that is, $m=(nB+kV)=8$ for $n=2, 3, 4, \dots$ and $k=\dots -1, 0, 1 \dots$

In order to investigate the possible combinations of blades, vanes and speeds under which all of the desired conditions are satisfied for either of the two suggested schemes a plot of vanes vs. speed for a given number of blades is helpful. An example of such a plot is shown in FIG. 48.

To permit us to more fully illustrate the first example given above, consider an engine whose first stage rotor has 44 blades and is 40 inches in diameter. The first condition of interest is the minimum shaft speed which will make the first harmonic of blade passage frequency 7000 c.p.s.

$$N = \frac{f}{nB} = \frac{7000}{2(44)} = 79.5 \text{ r.p.s.} = 4760 \text{ r.p.m.} \quad (18)$$

The speed of an m -lobe pattern is

$$\Omega_m = \frac{nB}{m} \Omega \quad (19)$$

or in terms of Mach numbers

$$M_m = \frac{nB}{m} M_B \quad (20)$$

A conservative requirement for decay of the m -lobe pattern is then:

$$\left| \frac{nB}{m} M_B \right| = 1, \text{ or } |m| = nBM_B \quad (21)$$

The vertical lines representing absolute value.

The pattern with the least number of lobes which will decay at a given speed and order can now be calculated. First obtain:

$$M_B = \frac{(\pi D/12)N \text{ r.p.s.}}{1100} = \frac{(\pi 40/12)79.5}{1100} = .756 \quad (22)$$

where 1100 represents the speed of sound in feet per second and the numerator represents the blade tip speed in feet per second.

For the fundamental:

$$m = nBM_B = (1)(44)(.756) = 33.2 \quad (23)$$

since m must be an integer but never less than nBM_B , m is chosen as 34.

The number of vanes required to produce this decaying m -lobe pattern is easily computed from Equation 1.

$$m = nB + kV \quad (1)$$

Now, with $n=1$ for the fundamental and $k=-1$

$$V = B + m = 44 + 34 = 78 \quad (1)$$

It is possible to allow a slight increase in speed with this number of vanes and still maintain the restriction that the fundamental blade passage noise decay. The allowable increase may be calculated as follows:

$$M_B = \frac{m}{nB} \quad (24)$$

and since m is 34,

$$M_B = \frac{34}{44} = .773 \quad (25)$$

$$N = (6.28 \times 10^3)(.773) = 4850 \text{ r.p.m.} \quad (26)$$

The next step is to investigate the number of lobes in the propagating mode of the first harmonic. Using Formula 1 with $n=2$ for the first harmonic, $B=44$, $V=78$

$$m = 2(44) + (-1)(78) = 10 \quad (27)$$

Hence, the radiation duration time criterion is satisfied.

The result of the calculation is then a combination of 44 blades and 78 vanes which satisfies all conditions in the speed range 4760 r.p.m. to 4850 r.p.m. The speed cannot be allowed to decrease below 4760 r.p.m. or the lowest frequency to propagate is below 7000 c.p.s., however, it is of interest to allow the speed to increase above 4850 and investigate the number of vanes then required. The number of vanes is governed by the relation

$$M_B \frac{nB}{|m|} = 1 \quad (21)$$

$$M_B nB = |m| = -(nB + kV) \quad (28)$$

$$V = nB(M_B + 1)k = -1 \quad (29)$$

Since B is fixed and n is one, V is just a linear function of speed, as represented by the line 900 in FIG. 48.

If in addition, it is required that the first harmonic ($n=2$) be contained in a mode of greater than or equal to eight lobes the line must be modified. Using the Formula 1, i.e.,

$$m = nB + kV \quad (1)$$

then

$$8 \leq |88 + kV|$$

Notice that through the portion of the curve shown in FIG. 48 where $V=80$ to $V=96$ the m -lobe mode of the first harmonic is less than eight. The only alternative by which to avoid the difficulty is to add a step to the line; i.e., allow V to increase with speed until $V=2B-8$ then increase V to $V=2B+8$ as shown by the dotted portion of FIG. 48. When other numbers of blades are considered for the same tip diameter, a family of curves is generated as shown in FIG. 49 for a 40 inch tip diameter.

A diagram of this type is useful in engine design or more typically engine redesign where a rotor tip diameter has been selected.

A final selection of blades, vanes, and approach speed for a particular engine is necessarily a compromise between acoustic requirements and compressor design requirements.

FIGURE 49 is intended to show a range of selections for the number of blades, vanes, and engine r.p.m. which satisfy acoustic considerations and allow flexibility in acoustic criteria when considered along with design limitations to achieve a satisfactory design from both stand-

75 points.

Preliminary attention should be focused along the lower boundary of the figure, labeled curve 1000. On this curve, points are designated by a number of rotor blades ranging from 26 to 48 blades. These points represent the minimum speed and minimum number of vanes to be used with a particular number of blades in order that B c.p.s. decays and $2B$ c.p.s. = 7000 c.p.s. with $m=8$. If, for example, 36 is initially chosen as the number of rotor blades then point 1001 gives 5960 r.p.m. and 80 vanes as the minimum speed and minimum number of stator vanes, respectively, to be used with this number of rotor blades. To investigate how the number of vanes must be adjusted if the speed is increased all that is necessary is to follow along curve 1002. This curve gives the minimum number of vanes to be used with 36 blades at any particular speed greater than 5960 r.p.m. A number of vanes larger than that specified by curve 1002 for 36 blades also satisfies the requirement that the fundamental blade passage frequency decay and that the number of lobes in the m -lobe pattern of the first harmonic is greater than 8.

Returning to the initial choice of 36 blades (point 1001), design considerations may require that the approach r.p.m. be less than the 5960 r.p.m. required if 36 blades are used. Suppose that 5000 r.p.m. is the recommended approach speed for the engine in question. Drawing the vertical line 1003 at 5000 r.p.m. it is found that there are three points labeled 1004, 1005 and 1006 which will satisfy the acoustic conditions at 5000 r.p.m. and these require 44 blades with 79 vanes (point 1004), 46 blades with 83 vanes (point 1005) or 48 blades and 87 vanes (point 1006). The final choice among these three possibilities would be point 1004 with 44 blades and 79 vanes since it is usually desirable to have as few stator vanes as possible.

Having selected the combination of blades and vanes from point 1004, it is of interest to relax the 5000 r.p.m. requirement and examine the effect of changing speed. Starting from point 1004, as speed is decreased proceed along curve 1003 toward point 1007. Point 1007 gives the minimum allowable speed to be used with 44 blades; namely, 4850 r.p.m. Recalling that 79 vanes were chosen from point 1004, it is important to note that this number of vanes also satisfies all requirements for point 1007. Parenthetically, the number of vanes to be used with 44 blades operating at a given speed must be equal to or greater than the value given by curve 1003.

Returning to point 1004, if the engine speed is to be increased above 5000 r.p.m., follow along curve 1003 to point 1009. The higher speed at point 1009, 5140 r.p.m., requires that the number of vanes be increased from 79 at point 1004 to 80 at point 1009. Now if speed is increased still further, one could proceed along a straight line connecting points 1009 and 1010 if it was only necessary that the fundamental decay. However, since it is required that $M_{2B} = |nB + kV| = |88 - V| \geq 8$ the number of vanes must always be less than 80 or greater than 96 when B is 44 and V must be increased from 80 at point 1009 to 96 at point 1011 to allow the speed to increase. Now a speed increase means no further increase in the number of vanes required as one moves toward point 1010 on curve 1003. A still larger increase in speed from 7420 r.p.m. at point 1010 again requires a linear increase in the number of vanes.

Returning to the second example given above, a similar curve, FIG. 50, is useful. The method of calculation is analogous except it is now required that $3BN \geq 7000$ and that the m -lobe pattern of the first harmonic, which is to be cancelled by indexing, be a minimum of 8 lobes.

While but two examples have been given utilizing the combination of approaches including reducing the noise source, noise abatement by decay, noise cancellation by indexing, control of directivity pattern, and control of the frequency of the noise generated, numerable other combinations of these methods will be apparent to those skilled in the art on the basis of the index included therein, such

that the two examples given should not be construed as limiting but purely as explanatory.

It will be obvious to those skilled in the art that for compressors used in engines having a normal operating condition it is possible, when designing the compressor, to select blade-vane combinations in the manner described herein so that the engine can be designed to experience compressor inlet discrete frequency noise abatement during the condition of normal operation, and this can be done without the requirement of indexability of parts.

It is to be understood that the invention is not limited to the specific embodiment herein illustrated and described but may be used in other ways without departure from its spirit as defined by the following claims.

Appendix A

LIST OF SYMBOLS

- B = Number of rotor blades
- N = Rotor shaft speed, r.p.s.
- Ω = Rotor shaft speed, radians/sec.
- θ = Angular coordinate, radians
- t = Time coordinate
- p = Pressure
- n = Harmonic number ($n=1$ for fundamental)
- a_n = Amplitude coefficient
- ϕ_n = Phase angle
- r = Radial coordinate
- f = Frequency, c.p.s.
- ω = Circular frequency ($\omega=2\pi f$)
- x = Coordinate in direction of duct axis
- y = Rectangular duct coordinate
- d = Rectangular duct width
- q = Index number
- a_q = Pressure amplitude in cross mode q
- λ = Wavelength
- λ_y = Wavelength in the y -direction
- c = Speed of sound in free-field
- f_q = Cutoff frequency for q th cross mode
- k_x = Wave number
- r_0 = Annular duct radius
- Ω = Angular velocity of spinning pattern
- s = Circumferential arc coordinate
- c_s = Wave velocity in s -direction
- m = Number of lobes or cycles of circumferential pressure variation
- a_m = Amplitude of m -th mode
- ϕ_m = Phase of m th mode
- M_s = Mach number of pattern in circumferential direction
- f_m = Cutoff frequency for m -lobe pattern
- k_x = Wave number
- Δx = Change in x position
- Δdb = Sound pressure level drop
- λ_s = Wavelength in circumferential direction
- Ω_m = Angular velocity of m -lobe spinning pattern
- σ = Hub-tip ratio ($\sigma=a/b$)
- a = Inner wall radius
- b = Outer wall radius
- μ = Radial mode index
- $E_{m\mu}^\sigma$ = Characteristic E-function
- $f_{m\mu}$ = Cutoff frequency of (m,μ) mode
- $k_{x\mu}$ = Wave number
- $k_{m\mu}^\sigma$ = Characteristic number
- M_m = Circumferential Mach number for m -lobe pattern
- M_m^* = Cutoff Mach number for (m,o) mode
- V = Number of stator vanes
- q = Index ($q=1,2,\dots,V-1$) (used differently in 4.1)
- k = Index ($k=\dots,-1,0,1,2,\dots$)
- R = Radial coordinate
- ψ = Angle to observer
- $P_{m,\mu}$ = Radiated pressure
- $\xi_{m,\mu}$ = Cutoff ratio

m	$\sigma=0$	$\sigma=1/4$	$\sigma=1/2$	$\sigma=3/4$
1	1.84	1.64	1.35	1.15
2	1.53	1.50	1.34	1.15
4	1.33	1.33	1.29	1.14
8	1.21	1.21	1.20	1.14
16	1.13	1.13	1.13	1.12
32	1.08	1.08	1.08	1.08

$$\sigma = \text{hub tip ratio} = \frac{\text{Blade Tip Radius}}{\text{Blade Root Radius}}$$

We claim:

1. In an axial flow compressor located within a fluid flow confining duct and which develops noise creating spinning modes due to interaction between stator vanes and rotor blades, the method of reducing the strength of the noise creating spinning modes comprising positioning a fixed stator within the duct which stator has a preselected number of stationary vanes around the periphery thereof, supporting a rotor for rotation within the duct which rotor has a prescribed number of blades around the periphery thereof and with the blades of the rotor being of different number from the vanes of the stator thereby developing noise creating spinning modes due to interaction between the stationary vanes and the rotating blades, and positioning the rotor to about five chord lengths from the stator to reduce the strength of the interaction between spinning modes so developed as to reduce the noise.

2. In an axial flow compressor comprising a rotor mounted for rotation within a fluid flow confining duct and having a stator adjacent thereto, the method of eliminating noise formed by spinning modes due to the interaction between blades carried by the rotor and stationary vanes carried by the stator on the rotation of said rotor, the method comprising selecting a rotor having a fixed number of blades thereon, selecting a stator having a fixed number of vanes thereon that is different in number from that of the blades of the rotor, rotating said rotor to produce the aforesaid spinning modes, retracting said stator vanes from the fluid flow confining duct whereby at least one spinning mode due to the interaction between rotor blades and stator vanes is eliminated and the noise from the aforesaid interaction is reduced.

3. In an axial flow compressor within a fluid flow confining duct, the method of reducing compressor inlet noise of fundamental blade passage frequency and noise harmonics up to and including the noise harmonic of index n created by noise creating spinning modes formed by compressor rotor-stator interaction comprising selecting the number of stator vanes and rotor blades according to the formula:

$$V \geq 2nB$$

where V =number of vanes, B =number of blades and n =noise harmonic index, thereby producing spinning modes which decay rapidly.

4. In rotating blade-vane machinery within a fluid flow confining duct where maximum rotor tip Mach number encountered is $1/3$ or less, in particular greater than

$$\frac{1}{2n+1}$$

and equal to or less than

$$\frac{1}{2n-1}$$

the method of selecting blade-vane combinations according to the formula:

$$B(1+M_B) \leq V \leq \frac{n}{n-1}B(1-M_B)$$

such that all components of blade passage noise up to and including that of noise harmonic index n decay,

where V equals the number of vanes, B equals the number of blades, n equals the noise harmonic index, and M_B equals the blade tip Mach number.

5. In an axial flow compressor located within a fluid flow confining duct, the method of reducing noise created by spinning modes formed by rotor-stator interaction by producing noise creating spinning modes which decay rapidly comprising selecting the number of stator vanes V and rotor blades B to produce a noise creating spinning mode with m -lobes so that the least number in absolute value of m , given by equation $m=nB+kV$, also satisfies $m \geq nBM_B/M^*_{tm}$, where n equals the noise harmonic index, k is an index that ranges independently for all positive and negative integers, M_B is the rotor tip Mach number, and M^*_{tm} is the critical or cutoff Mach number for the m -lobe pattern.

6. An axial flow compressor within a fluid flow confining duct having an axis, a rotor with blades about the periphery thereof mounted for rotation within said duct, a stator axially upstream of said rotor and having a number of substantially flat stationary vanes about the periphery thereof and with the number of vanes in said stator being different from the number of blades in said rotor and being sufficient in number to form noise creating spinning modes due to blade and vane interaction, said vanes extending radially inwardly from said duct for a part blade span while presenting a leading edge which extends inwardly and downstream and forms an acute angle with the duct axis to cause the noise creating spinning modes to decay.

7. In an axial flow compressor within a fluid flow confining duct having an axis and which develops noise creating spinning modes of particular frequency and lobe number m due to interaction between blades and vanes, the method of cancelling noise creating spinning modes comprising positioning a pair of axially spaced, equal vaned stators within the duct, mounting a bladed rotor for rotation between the stators and selecting the blade number to be different from the vane number to produce a first noise creating spinning mode due to interaction of the rotor blades and the first of the pair of stators and a second noise creating spinning mode due to interaction of the rotor blades and the second of the pair of stators, and positioning the rotor with respect to each stator so that the spinning modes will be of equal intensity, frequency and lobe number m and opposite phase to cancel each other.

8. The method according to claim 7 wherein noise creating spinning modes of identical frequency and lobe number from two separate blade-vane interactions are caused to cancel by altering the axial spacing between the interactions.

9. The method as in claim 7 wherein noise creating spinning modes of identical frequency and lobe number from two separate blade-vane interactions are caused to cancel by circumferentially indexing an element of the first interaction with respect to an element of the second interaction.

10. The method according to claim 7 wherein said cancellation is caused by altering the axial spacing and circumferential relative positions between an element of the first interaction with respect to an element of the second interaction.

11. The method according to claim 7 wherein the number of lobes m are equal to or greater than eight.

12. In an axial flow compressor located within a fluid flow confining duct and which produces noise creating spinning modes due to interaction between blades and vanes, the method of reducing ground exposure duration to noise radiated from the inlet of the axial flow compressor comprising mounting a stator with fixed vanes about the periphery thereof within said duct, mounting a rotor adjacent the stator within said duct for rotation, which rotor has a plurality of blades positioned about the periphery thereof which coact with the stationary vanes

to produce noise creating spinning modes having a number of lobes m , selecting the number of blades B and the number of vanes V so that the least number in absolute value of m given by the equation:

$$m = nB + kV$$

satisfies the condition $m=8$ where n is the harmonic index of noise radiated, B is the number of blades, V is the number of vanes and k is an index that ranges independently over all positive and negative integers.

13. In an axial flow compressor located within a fluid flow confining duct and which produces noise creating spinning modes having a number of lobes m due to interaction between rotating blades and stationary vanes, the method of controlling the noise created in the compressor comprising mounting a stator within the duct having a number of stationary vanes V positioned about the periphery thereof, mounting a rotor for rotation adjacent the stator within the duct and having a selected number of blades B about the periphery thereof so that noise creating spinning modes are developed due to blade and vane interaction, selecting the number of rotor blades B such that at the operating speed N of interest, the frequency of the first harmonic noise is above 7000 c.p.s. using the formula:

$$B \geq \frac{7000}{2N}$$

where 2 is the harmonic index, causing the fundamental frequency noise to decay by selecting the number of vanes V in the stator so that the least number in absolute value of m given by the equation:

$$m = B + kV$$

also satisfies the equation:

$$m \geq BM_B / M_m^*$$

where k is an index that ranges independently over all positive and negative integers, M_B is rotor tip Mach number, and M_m^* is the critical Mach number for the m -lobe pattern.

14. In an axial flow compressor located within a fluid flow confining case wherein said compressor develops noise creating spinning modes due to rotating blade and stationary vane interaction, which noise creating spinning modes have a number of lobes m , the method of controlling the noise created by the compressor comprising mounting a first fixed stator within the case having a selected number of vanes V positioned about the periphery thereof, mounting a rotor for rotation adjacent the first stator having a particular number of blades B about the periphery thereof so that a first noise creating spinning mode is established due to blade and vane interaction, mounting a second stator adjacent the rotor so that a second noise creating spinning mode is created due to blade and vane interaction, selecting the number of rotor blades B such that at the operating speed N of interest the frequency of the second harmonic noise is above 7000 c.p.s. using the formula:

$$B \geq \frac{7000}{3N}$$

where 3 is the harmonic index, causing the fundamental frequency noise to decay by selecting the number of vanes V in the stators so that the least number in absolute value of m given by the equation:

$$m = B + kV$$

also satisfies the equation:

$$m \geq BM_B / M_m^*$$

where k is an index that ranges independently over all positive and negative integers, M_B is rotor tip Mach number, and M_m^* is the critical Mach number of the m -lobe pattern, indexing an element of the first blade and vane interaction with respect to an element of the second blade

and vane interaction to cause the spinning modes so created thereby to be of equal intensity and opposite phase to cancel the first harmonic noise.

15. In an axial flow compressor located within a fluid flow confining duct and which develops noise creating spinning modes by interaction between blades on rotating rotors and vanes on stationary stators with the number of vanes being different from the number of blades, the method of cancelling noise creating spinning modes formed by a plurality of interactions comprising selecting the number of blades and the number of vanes such that the propagating modes have the same number of lobes and are of comparable radial intensity distribution, indexing the noise sources by adjusting the relative position between blades and vanes so that the amplitude and phase relations of the noise creating spinning modes yield a zero sum and thereby dispensing the noise.

16. The method of reducing ducted compressor inlet noise in an axial flow compressor within a fluid confining duct comprising assembling a first rotor between first and second equal vaned stators in said flow confining duct, said rotor having a different number of blades than the number of the vanes, then rotating said first stage rotor to establish two noise creating interacting spinning modes at blade-vane passing frequency including a first noise creating interaction spinning mode at blade passing frequency caused by interaction between the vanes of said first stator and the blades of said first rotor and a second noise creating interaction spinning mode at blade passing frequency produced by the interaction of the blades of said first stage rotor and the vanes of said second stage stator, adjustably positioning said first and second stators with respect to one another so that said first and second noise creating spinning modes are of substantially equal magnitude and out of phase so as to cancel and thereby dispense the noise.

17. The method of reducing ducted compressor inlet noise in an axial flow compressor within a fluid flow confining duct comprising assembling a first bladed rotor between first and second equal vaned stators in the fluid flow confining duct, said rotor having a different number of blades than the number of the vanes, causing said first stage rotor to rotate to establish two noise creating interacting spinning modes at blade frequency including a first noise creating interaction spinning mode at blade-vane passing frequency caused by interaction between the vanes of said first stator and the blades of said first rotor and a second noise creating interaction spinning mode at blade-vane passing frequency produced by the interaction of the blades of said first stage rotor and the vanes of said second stage stator, rotating one of said first and second stators with respect to the other said first and second stators to cause said first and second noise creating interaction spinning modes to be out of phase and cancel.

18. The method according to claim 17 wherein the adjustable positioning of the stators relative to each other is substantially a helical movement relative to the axis of the rotor.

19. The method of reducing compressor inlet noise in an axial flow compressor within a fluid flow confining duct comprising assembling a first bladed rotor between first and second vaned stators within the duct, then causing said first stage rotor to rotate to establish two noise creating interacting spinning modes at blade frequency including a first noise creating interaction spinning mode at blade passing frequency caused by interaction between the vanes of said first stator and the blades of said first rotor and a second noise creating interaction spinning mode at blade passing frequency produced by the interaction of the blades of said first stage rotor and the vanes of said second stage stator, axially positioning said first and second stators with respect to said first rotor so that said first and second noise creating interaction modes are of substantially equal magnitude, circumferentially rotating one of said first and second stators with respect to the other

of said first and second stators to cause said first and second substantially equal magnitude noise creating interaction spinning modes to be out of phase and cancel, adding a second stage rotor having an equal number of blades as said first stage rotor and a third stage stator having an equal number of vanes with said first and second stage stators such that said second stage rotor is positioned between said second and third stage stators, rotating said second stage rotor with respect to said second and third stage stators to establish third and fourth noise creating interaction spinning modes at blade passing frequency with said third noise creating interaction spinning mode at blade passing frequency being produced by interaction between the vanes of said second stator and the blades of said second rotor, and said fourth noise creating interaction spinning mode at blade passing frequency being produced by interaction between the blades of said second rotor and the vanes of said third stator, axially positioning said second rotor and third stator with respect to said second stator so positioned so that said third and fourth noise creating modes are of equal magnitude, circumferentially indexing said third stage stator with respect to said second stage stator to cause said third and fourth noise creating modes to be out of phase and cancel out, thereby cancelling all of said noise creating interaction spinning modes at blade passing frequency, circumferentially rotating said first rotor with respect to said second rotor until the resultant of the first harmonics noise is a minimum.

20. In the assembly of an axial flow compressor within a fluid flow confining duct to reduce noise resulting from operation in which the compressor has a pair of rows of stator vanes on opposite sides of a row of rotor blades, the steps of adjusting the axial spacing between the vanes and the blades so that the magnitudes of the rotating noise creating pressure fields generated on opposite sides of the rotor row are substantially equal, and adjusting the angular relation of the vanes of the pair of rows one with respect to the other so that the rotating noise creating pressure fields generated on opposite sides of the rotor row are substantially 180° out of phase.

21. In the assembly of an axial flow compressor within a fluid flow confining duct having a pair of rows of stator vanes and a row of rotor blades therebetween within the duct, in which a fundamental first noise creating rotating pressure field having pressure ridges is produced by the interaction of the wakes of the first row of stator vanes and the row of rotor blades, and a second noise creating fundamental rotating pressure field having pressure ridges is produced by the interaction of the wakes of the row of rotor blades and the second row of stator vanes, said rotating fields being dependent on the number of vanes in the stator rows and the number of blades in the rotor row, each row of vanes having the same number of vanes and the number of rotor blades being different from the number of stator vanes, and in which the second row of vanes is positioned so that the first fundamental rotating noise creating pressure field is substantially 180° out of phase with the second noise creating fundamental rotating pressure field, said vanes and blades also producing first harmonic noise creating rotating pressure fields comparable to the fundamental fields and having a different rate of rotation therefrom, means for positioning the second row of vanes helically relative to the axis of the rotor with respect to the first row of vanes until the first noise creating harmonic pressure fields are 180° out of phase thereby balancing these harmonic noise creating fields, said helix being such that the fundamental noise creating pressure fields move along their pressure ridges as they propagate noise so that the cancellation previously established is maintained.

22. In the assembly of an axial flow compressor within a fluid flow confining duct having a pair of rows of stator vanes and a row of rotor blades therebetween within the duct, in which two fundamental rotating noise

creating pressure fields having pressure ridges and having different lobe numbers are produced by the interaction between the wakes of the first row of stator vanes and the row of rotor blades, and in which two fundamental rotating noise creating pressure fields having pressure ridges and having different lobe numbers are produced by the interaction of the wakes of the rotor blades and the second row of stator vanes, said noise creating rotating fields being dependent on the number of vanes in the stator rows and the number of blades in the rotor row, each row of vanes having the same number of vanes and the number of rotor blades being different from the number of stator vanes, and in which the second row of vanes is positioned so that one of the two said fundamental rotating noise creating pressure fields produced by the first vane row interaction is substantially 180° out of phase with the corresponding rotating noise creating pressure field of the second vane row interaction to cancel, means for positioning the second row of vanes helically with respect to the first row of vanes until the other of the two said fundamental rotating noise creating pressure fields produced by the first vane row interaction is substantially 180° out of phase with the corresponding rotating noise creating pressure field of the second vane row interaction to cancel, this helix being such that the fundamental pressure fields move along their pressure ridges as they propagate noise so that the cancellation previously established is maintained.

23. In an axial flow compressor located within a fluid flow confining duct and which develops noise creating spinning modes having lobes due to blade and vane interaction, a rotor positioned between a first and second stator in said duct, said rotor having a number of blades different from the number of vanes on each of said stators, said rotor having a selected number of blades to satisfy the equation:

$$n_1 MB \geq 7000 \text{ c.p.s.}$$

where N is rotor speed, n_1 is the noise harmonic index and equals two for the first harmonic and B is the number of blades, and also where each of said stators has a number of stationary vanes greater than the product of the number of blades B, times the harmonic index n_2 , plus the number of lobes m , in the rotor-stator interaction noise creating spinning mode pattern, where m is selected to provide decay according to the formula:

$$m \geq \frac{n_2 B M_B}{M_m^*}$$

where m is the number of lobes, n_2 is the harmonic index next smaller than n_1 , M_B is the blade-tip Mach number, and M_m^* is the cutoff Mach number for the m -lobe pattern.

24. In an axial flow compressor located within a fluid flow confining duct and which develops noise creating spinning modes having lobes due to blade and vane interaction, a rotor positioned between a first and second stator within said duct, said rotor having a number of blades different from the number of vanes on each of said stators, said rotor having a selected number of blades to satisfy the equation:

$$n_1 NB \geq 7000 \text{ c.p.s.}$$

where N is rotor speed, n_1 is the noise harmonic index and equals two for the first noise harmonic and B is the number of blades, and also where each of said stators has a number of stationary vanes greater than the product of the number of blades B, times the noise harmonic index n_2 , plus the number of lobes m , in the rotor-stator blade-vane interaction noise creating spinning mode pattern where m is selected to provide decay according to the formula:

$$m \geq n_2 B M_B$$

where n_2 is the noise harmonic index next smaller than n_1 , and M_B is the blade tip Mach number.

25. In an axial flow compressor located within a fluid

flow confining duct and which develops noise creating spinning modes having lobes due to blade and vane interaction, a rotor positioned between a first and second stator in said duct, said rotor having a number of blades different from the number of vanes on each of said stators, said rotor having a selected number of blades to satisfy the equation:

$$n_1NB \geq 7000 \text{ c.p.s.}$$

where N is rotor speed, n_1 is the noise harmonic index and equals two for the first noise harmonic and B is the number of blades, and also where each of said stators has a number of stationary vanes greater than the product of the number of blades B, times the noise harmonic index, n_2 , plus the number of lobes m , in the rotor-stator interaction noise creating spinning mode pattern where m is selected to provide decay according to the formula:

$$m \geq n_2BM_B$$

where n_2 is a noise harmonic index smaller than n_1 , and M_B is the blade tip Mach number and wherein the vanes of said stators are positioned with respect to one another such that the stator-rotor interaction noise creating spinning modes having a different frequency of noise harmonic index n_3 are of substantially equal intensity and out of phase so as to cancel the noise.

26. In an axial flow compressor within a fluid flow confining duct, a rotor positioned between two stators in said duct, said stators having an equal number of vanes positioned about the periphery thereof, said rotor having a plurality of blades different in number from the number of said vanes in each of said stators and positioned about the periphery thereof and coacting with said stator vanes to form two similar noise creating spinning mode patterns, means to position the vanes of each stator relative to one another and said rotor whereby said noise creating spinning mode patterns are of substantially equal intensity and out of phase so as to cancel the noise.

27. In an axial flow compressor within a fluid flow defining duct comprising a rotor positioned between two stators within the duct and having the same number of stationary vanes around the periphery thereof, said rotor having a plurality of blades positioned about the periphery thereof to coact with said stator vanes to form two similar pairs of propagating noise creating spinning mode pressure patterns, means for cancelling said similar pairs of noise creating patterns including means for indexing said stators relative to one another so that said similar pairs of noise creating patterns are of substantially equal intensity and out of phase to dispense the noise.

28. Apparatus according to claim 27 wherein said noise creating patterns are two-lobed and three-lobed respectively and wherein said indexing means includes means to circumferentially index said stators so that said two-lobed noise creating patterns are out of phase and for additional indexing said stators in the direction of constant phase lines of said two-lobed noise creating patterns until said three-lobed noise creating patterns are also out of phase to dispense the noise.

29. In an axial flow compressor located within a fluid flow confining duct and which develops noise creating spinning modes having lobes due to blade and vane interaction, a rotor positioned between a first and second stator to create noise creating spinning modes having a highest objectionable frequency of noise harmonic index n due to interaction between said blades and vanes, said stators having a number of stationary vanes V, according to the equation:

$$V = nB(1 + M_B)$$

where n is the noise harmonic index of the highest frequency being suppressed, B is the number of blades, and M_B is the blade tip Mach number, thereby causing all spinning modes having harmonic indices up to and including n to decay.

30. In an axial flow compressor having a fluid flow confining case with an open inlet end, a rotor of selected blade number located within said case, two stators of selected vane number located within said case on opposite axial sides of said rotor, said stators having a number of vanes different from the number of blades in said rotor, and means to position said stators with respect to each other and to said rotor so as to reduce the intensity of the noise creating spinning modes created by the rotation of said rotor between said stators and thereby reduce the noise intensity external of said case, wherein said noise reduction is caused by noise field cancellation and wherein said cancellation is caused by selecting vanes and blades of different numbers and relative positions to produce substantially equal intensity spinning wave patterns which are out of phase.

31. Apparatus according to claim 26 wherein said means axially displaces said stators relative to one another and said rotor.

32. Apparatus according to claim 26 wherein said means circumferentially displaces said stators relative to one another and said rotor.

33. Apparatus according to claim 26 wherein said means helically displaces said stators relative to one another and said rotor.

34. Apparatus according to claim 31 wherein said axial displacement is to about five chord lengths of the wake creating member.

35. In an axial flow compressor located within a fluid flow confining duct and which produces noise creating spinning modes having a number of lobes m due to interaction between blades and vanes, a stator fixed within the duct having a number of stationary vanes V positioned about the periphery thereof, a rotor mounted for rotation adjacent the stator within the duct and having a selected number of blades B about the periphery thereof different from the number of vanes V so that noise creating spinning modes are developed due to blade and vane interaction, the number of rotor blades B being selected such that at the operating speed N of interest, the frequency of the first harmonic noise is above 7000 c.p.s. using the formula:

$$B \geq \frac{7000}{2N}$$

where 2 is the harmonic noise index, the number of vanes V in the stator being selected so that the least number in absolute value of m given by the equation:

$$m = B + kV$$

also satisfies the equation:

$$m \geq BM_B/M_m^*$$

thereby causing the fundamental frequency noise to decay where K is an index that ranges independently over all positive and negative integers, M_B is rotor tip Mach number, and M_m^* is the critical Mach number for the m -lobe pattern.

36. In an axial flow compressor located within a fluid flow confining case and which develops noise creating spinning modes due to blade and vane interaction, which noise creating spinning modes have a number of lobes m , a first stator fixed within the case having a selected number of vanes V positioned about the periphery thereof, a rotor mounted for rotation adjacent the stator and having a particular number of blades B about the periphery thereof different from the number of vanes V so that a first noise creating spinning mode is established due to blade and vane interaction, a second fixed stator having a number of vanes V positioned about the periphery thereof and positioned adjacent the rotor so that a second noise creating spinning mode is created due to blade and vane interaction, the number of rotor blades B being selected such that at the operating speed N of interest the fre-

quency of the second harmonic noise is above 7000 c.p.s. using the formula:

$$B \geq \frac{7000}{3N}$$

where 3 is the harmonic noise index, the number of vanes V in the stators being selected so that the least number in absolute value of m given by the equation:

$$m = B + kV$$

also satisfies the equation:

$$m \geq BM_B / M_m^*$$

thereby causing the fundamental frequency noise to decay, where k is an index that ranges independently over all positive and negative integers, M_B is rotor tip Mach number, and M_m^* is the critical Mach number of the m -lobe pattern, said first stator being positioned with respect to said second stator to cause the noise creating spinning modes so created thereby to be equal intensity and opposite phase to cancel the first noise harmonic.

37. An axial flow compressor within a fluid flow defining duct and having a pair of rows of stator vanes and a row of rotor blades positioned within the duct therebetween and with each of said rows of vanes having the same number of vanes, means to support said row of rotor blades and said rows of stator vanes such that said blades rotate with respect to said vanes thereby establishing a first fundamental rotating noise creating pressure field due to the interaction of the pressure fields of the wakes of the first row of stator vanes and the pressure fields at the inlet end of rotor blades and a second fundamental rotating noise creating pressure field caused by the interaction of the pressure fields of the wakes of the row of rotor blades and the second row of stator vanes and also establishing first harmonic rotating noise creating pressure fields comparable to the fundamental fields but having a different rate of rotation, means for angularly positioning said second row of vanes such that the first fundamental rotating noise creating pressure field is substantially 180° out of phase with the second fundamental rotating noise creating pressure field, and means for adjusting the position of said second row of vanes in a direction along the pressure ridges of the fundamental noise creating pressure fields until the harmonic noise creating pressure fields are substantially 180° out of phase, thereby balancing the noise creating harmonic fields while maintaining a balance between the noise creating fundamental fields.

38. An axial flow compressor within a fluid flow defining duct and having a pair of spaced rows of stator vanes and a row of rotor blades positioned therebetween within the duct, each of said rows of vanes having the same number of vanes, means to support said rows of vanes stationary in spaced relation and to support said row of blades therebetween for rotation with respect thereto to establish two identical pairs of fundamental rotating noise creating pressure fields having different lobe numbers produced by the interaction between the wakes of the first row of stator vanes and the row of rotor blades and the interaction between the wakes of the rotor blades and the second row of stator vanes, means for positioning said second row of vanes which respect to said first row of vanes so that one pair of said fundamental rotating noise creating pressure fields are substantially 180° out of phase, and means to adjust the position of said second row of vanes further along the high pressure ridges of the first pair of noise creating pressure fields, thereby maintaining the relationship between the first pair of fundamental noise creating pressure fields, until the second pair of fundamental rotating noise creating pressure fields are substantially 180° out of phase.

39. An axial flow compressor within a fluid flow defining duct and having first and second rows of stator vanes with equal number of vanes in both rows and a row of

rotor blades therebetween within the duct, in which a first rotating noise creating pressure field is produced by the interaction of the wakes of the first stator vanes and the first rotor, and another noise creating pressure field is produced by the interaction of the wakes of the rotor blades and the second stator vanes, and in which the rotating noise creating pressure fields, having the same number of lobes, are rotating at the same r.p.m. and in the same direction, the vanes of the first row and the vanes of the second row being so positioned relative to each other that the first noise creating pressure field is out of phase with the second noise creating pressure field and the two fields are substantially cancelled out.

40. The apparatus as in claim 39 in which the axial spacing of the rotor blades to the stator vanes is adjusted so that the magnitudes of the two fields are substantially equal.

41. The apparatus as in claim 39 in which a second row of rotor stage blades is located downstream of the second stator vanes and a third row of stator vanes having the same number of vanes as the first and second stator rows, is located downstream of the second row of rotor blades, and in which the angular relation of the vanes in the third row of stator vanes is such with respect to the vanes in the second row as to have the rotating noise creating pressure field produced by the interaction between the second rotor blades and the third stator row out of phase with the rotating noise creating pressure field produced by the interaction between the second stator row and second rotor row.

42. The apparatus as in claim 41 in which the first and second blade rows have an equal number of blades and in which the angular relation of the blades of the first rotor row is such, with respect to the blades of the second rotor row, that the first harmonic noise of the resultant of the interactions on opposite sides of the first rotor row are out of phase with the first harmonic noise of the resultant of the interactions on opposite sides of the second rotor row.

43. A marine propeller within a fluid flow defining duct and including a rotor and a first and a second stator, means to support said rotor for rotation between said two stators to establish stator-rotor interaction noise creating spinning modes between said first stator and said rotor and between said rotor and said second stator, means to cause said rotor to rotate so that said spinning modes are established with the highest troublesome harmonic noise equal to n and so that rotor blade tip Mach number M_B is $\frac{1}{3}$ or less, in particular greater than

$$\frac{1}{2n+1}$$

and equal to or less than

$$\frac{1}{2n-1}$$

and so that the number of blades in said rotor B and number of vanes in said stators V is selected according to the formula:

$$B(1+M_B) \leq V \leq \frac{n}{n-1} B(1-M_B)$$

44. In an axial flow compressor within a fluid flow confining duct, a stator positioned between two rotors in said duct, said rotors having an equal number of blades positioned about the periphery thereof, said stator having a plurality of vanes different in number from the number of said blades in each of said rotors and positioned about the periphery thereof and coacting with said rotor blades to form two similar noise creating spinning mode patterns, means to position the blades of each rotor relative to one another and said stator whereby said noise creating spinning mode patterns are of substantially equal intensity and out of phase so as to cancel the noise.

33

45. In axial flow machinery within a fluid flow defining duct and having a first multi-bladed rotor positioned between a first and second stator having an equal number of stationary vanes, a second multi-bladed rotor havinng a different number of blades from the first rotor and positioned between the second stator and a third stator having an equal number of vanes with the first and second stators so that the first rotor blades coact with the vanes of the first and second stator to produce first and second noise creating spinning modes of a fundamental noise frequency equal to the speed of the first rotor times the number of blades in the first rotor and so that the blades of the second rotor interact with the vanes of the second and third stator to form third and fourth noise creating spinning modes having a different number of lobes than the first and second noise creating spinning modes and having a fundamental noise frequency equal to the speed of the second rotor times the number of blades in the second rotor, the method of cancelling said noise creating spinning modes comprising indexing the first stator with respect to the second stator to cancel said first and second noise creating spinning modes, and indexing the third stator with respect to the second stator to cancel said third and fourth noise creating spinning modes.

46. An axial flow spaced machinery within a fluid flow defining duct and comprising first, second, and third stators having an equal number of stationary vanes, a first rotor positioned between said first and second stators, a second rotor positioned between said second and third stators, said first rotor and said second rotor having a different number of blades on the periphery thereof, means to cause said first rotor to rotate with respect to said first and second stators so that the blades of said first rotor interact with the vanes of said first and second stators to establish first and second noise creating spinning modes, means to cause said second rotor to rotate with respect to the vanes of said second and third stators so that the blades of said second rotor interact with the vanes of said second and third stators to produce third and fourth noise creating spinning modes, means to index said first stator with respect to said second stator to cancel said first and second noise creating spinning modes, means to index said third stator with respect to said second stator to cancel said third and fourth noise creating spinning modes.

47. In an axial flow compressor having a fluid flow confining case with an open inlet end, a rotor of selected blade number located within said case, two stators of selected vane number located within said case on opposite sides of said rotor, and means to position said stators with respect to each other and to said rotor so as to reduce the intensity of the noise creating spinning modes

34

created by the rotation of said rotor between said stators and thereby reduce the noise intensity external of and beneath said case wherein said noise reduction is caused by selecting said vane numbers and said blade numbers so that the velocity of said spinning mode is below cutoff and wherein the number of said vanes in each of said stators is greater than twice the number of said blades multiplied by the harmonic index of the noise source being reduced.

48. In an axial flow compressor having a fluid flow confining case with an open inlet and, a rotor of selected blade number located within said case, two stators of selected vane number located within said case on opposite sides of said rotor, and means to position said stators with respect to each other and to said rotor so as to reduce the intensity of the noise creating spinning modes created by the rotation of said rotor between said stators and thereby reduce the noise intensity external of and beneath said case wherein said noise reduction is caused by selecting said vane numbers and said blade numbers so that the velocity of said spinning mode is below cutoff and wherein the number of said vanes in each of said stators and the number of said blades produce spinning modes in a wave pattern having at least eight lobes.

References Cited by the Examiner

UNITED STATES PATENTS

1,502,903	7/24	Campbell	253—39
1,525,814	2/25	Lasche	253—39
1,585,218	5/26	Watanabe.	
1,688,809	10/28	Gill	103—89 X
1,788,307	1/31	Lack	253—148
1,868,008	7/32	Gardner.	
2,191,341	2/40	Curley	230—120
2,870,956	1/59	Dhonan et al.	230—114
2,893,508	7/59	Baruch et al.	230—232 X
2,976,011	3/61	Stalker	230—122 X
3,006,602	10/61	Caruso et al.	253—39
3,035,759	5/62	Paulson et al.	230—122
3,096,073	7/63	Homola et al.	253—78 X

FOREIGN PATENTS

139,785	10/20	Great Britain.
226,203	7/25	Great Britain.
300,002	11/28	Great Britain.
348,032	5/31	Great Britain.
668,434	3/52	Great Britain.

KARL J. ALBRECHT, *Primary Examiner.*JOSEPH H. BRANSON, Jr., *Examiner.*