Oct. 31, 1961 W. J. CARUSO ET AL 3,006,603

TURBO-MACHINE BLADE SPACING WITH MODULATED PITCH

Filed Aug. 25, 1954 4 Sheets-Sheet 1

Fig. 1.

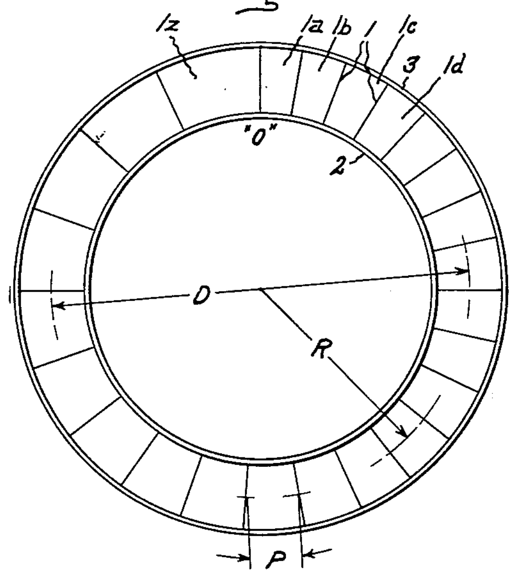

Fig. 1a.

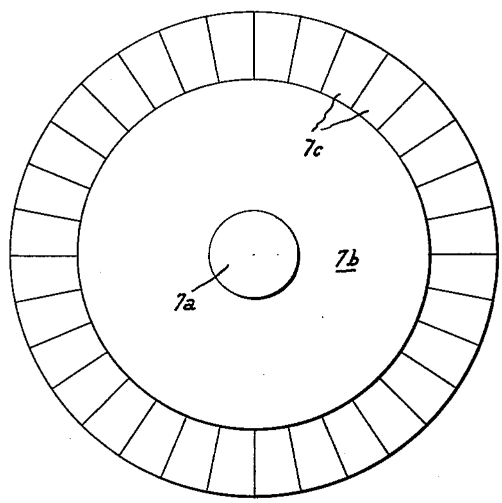

Fig. 2.

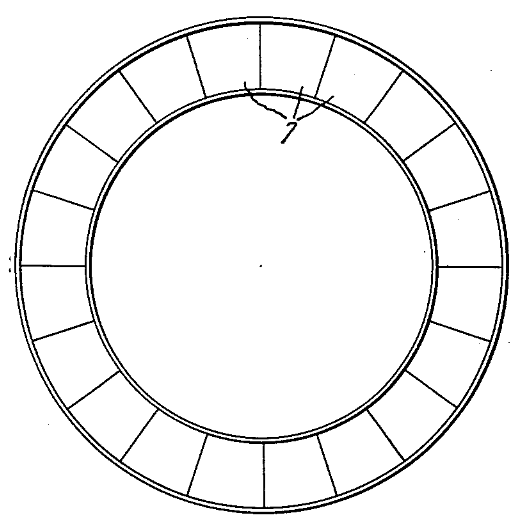

Fig. 2a.

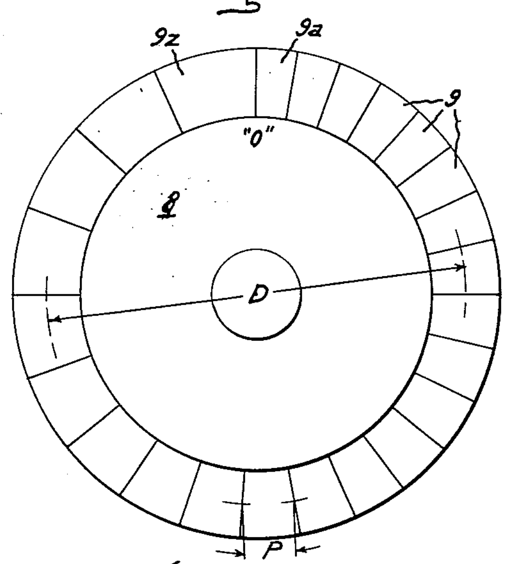

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Fig. 3.

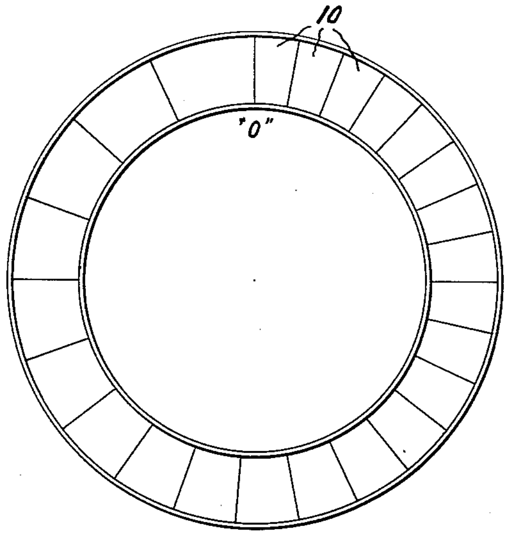

Fig. 3a.

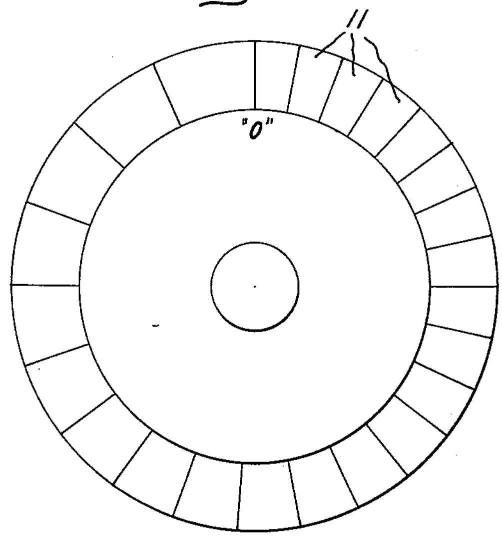

Fig. 4.

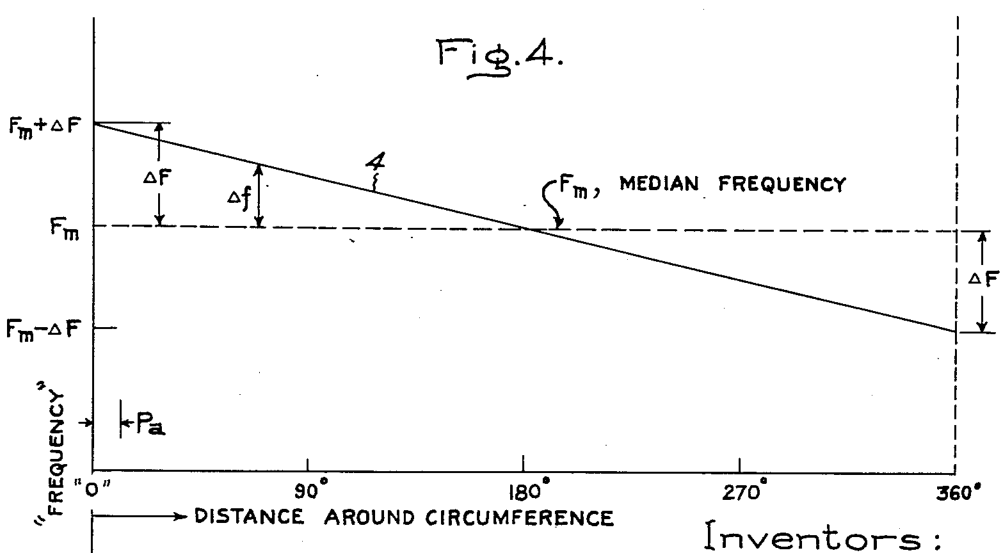

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Fig. 5.

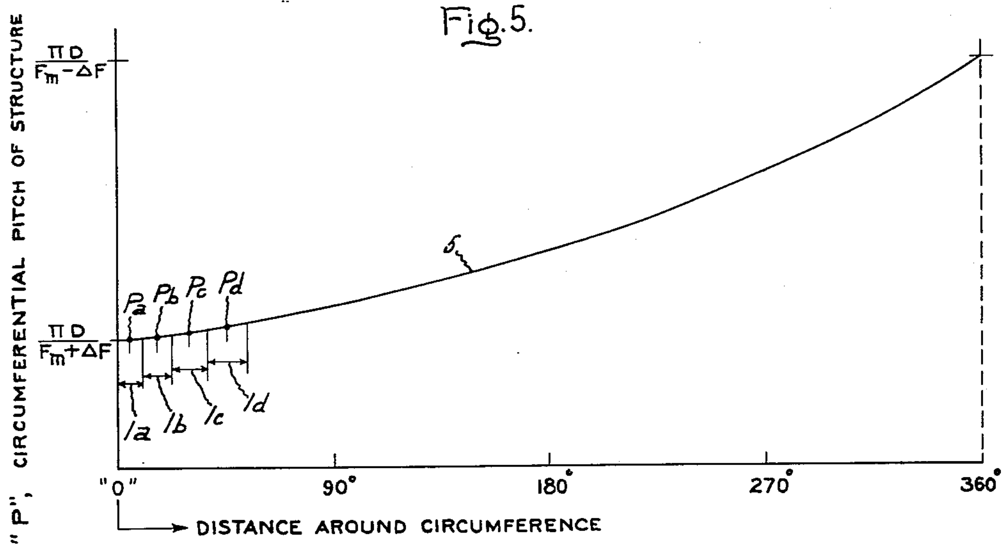
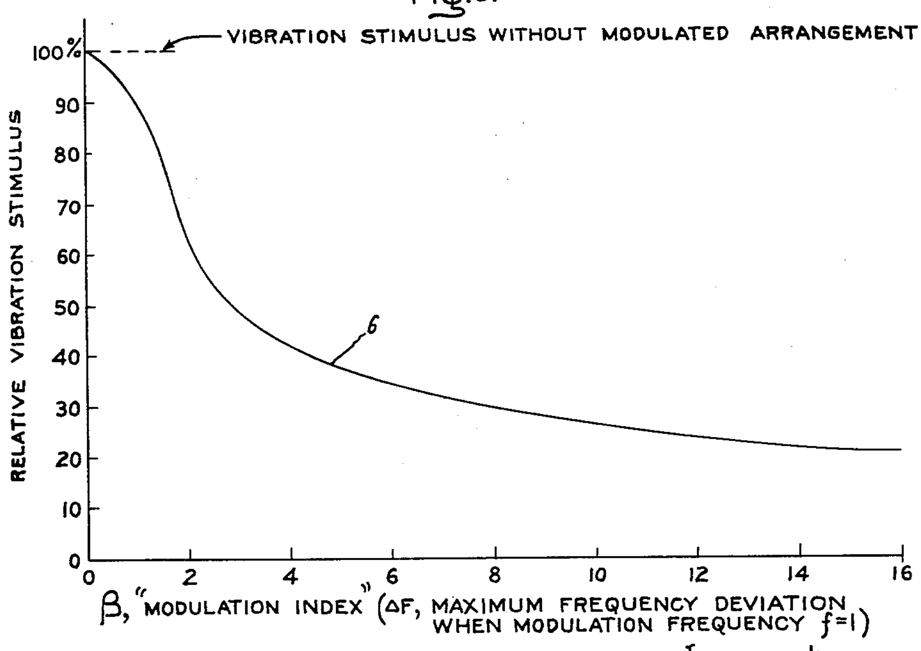


Fig. 6.



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TURBO-MACHINE BLADE SPACING WITH MODULATED PITCH

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Fig. 7.

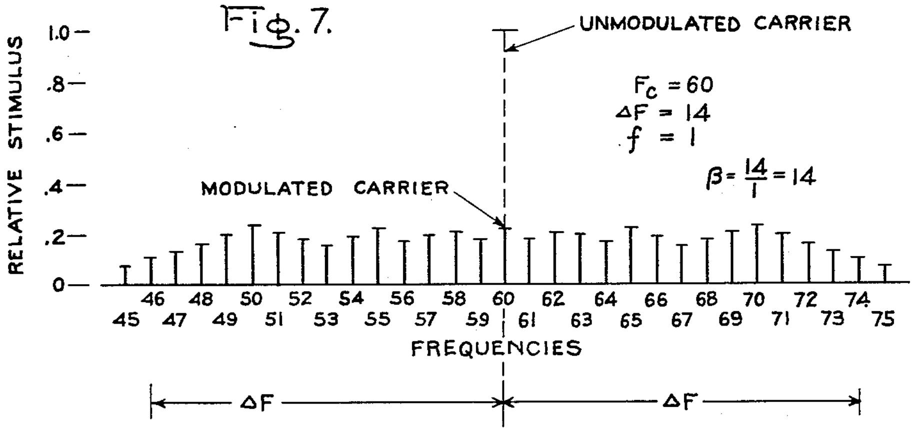


Fig. 8.

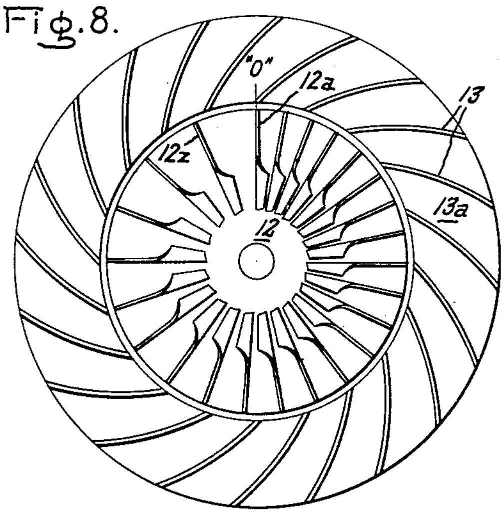


Fig. 9.

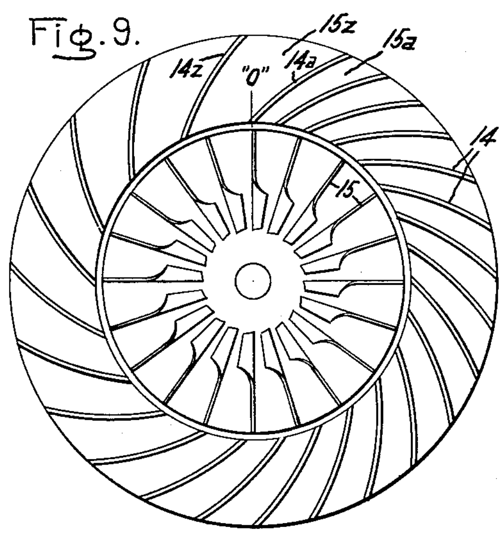
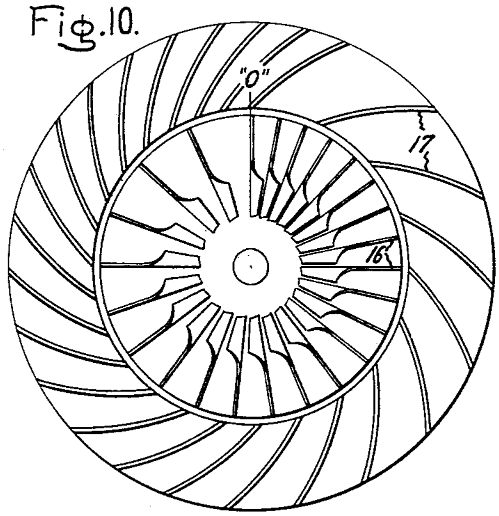


Fig. 10.



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TURBO-MACHINE BLADE SPACING WITH MODULATED PITCH

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13 Claims. (Cl. 253—39)

This invention relates to a novel theory and method for reducing the maximum value of the stimulus applied to mechanical systems subject to vibration, particularly as applied to the problem of reducing destructive vibrations in rotating machinery such as fluid-dynamic apparatus like axial or radial flow turbo-machines having a circumferential row of nozzles delivering motive fluid to a relatively rotating "bucket-wheel."

In such a machine, each nozzle imparts a discrete impulse to each bucket passing by the nozzle. If the nozzles are of equal circumferential pitch and equally spaced, as is usually the case in prior art turbo-machines, a given bucket will receive such impulses at a frequency depending on the speed of rotation of the bucket-wheel relative to the stationary nozzles, and on the number of nozzles comprising the complete nozzle circle. It is always necessary to correlate the many factors involved (which are further complicated by the fact that the turbine wheel will have to operate over a range of speeds) so that the exciting impulses imparted to the buckets will not result in destructive vibration amplitudes within the normal operating range of the machine. This is difficult and sometimes even impossible to achieve with the prior art turbo-machine designs. Once the turbine designer ascertains the range of exciting frequencies which will be imparted to a given bucket over the intended operating range of the machine, it is his problem to so design the blade structures that they will have no natural frequency of vibration coinciding with any of the exciting frequencies likely to be impressed on the bucket.

In the past, the enormous complexity of this problem has resulted in numerous "rule-of-thumb" design methods based on long years of practical experience, by which turbine designers have determined the material and shape, and the number and disposition of rotating and stationary blades, in order that buckets designed and manufactured by available techniques can withstand the vibration stimuli encountered.

A study of the basic theory of vibration phenomena associated with turbo-machine nozzles has led us to the discovery that there are certain striking similarities in the mathematical analysis of such phenomena and the analysis of cyclical electrical phenomena, namely certain mathematical theories used in radio-communication. Analytical studies along such lines have led to the discovery that the principles of so-called "frequency modulation" broadcasting can be adapted to the design of the arrangement of buckets in the rotating turbine bucket-wheel and to the arrangement of nozzles in the stationary nozzle ring, and to various other analogous structures, in such a manner as to achieve a very great reduction in the stimulus which is applied to that member subject to destructive vibration, such as the stationary nozzle blades, and/or the buckets on the rotating wheel, etc.

Accordingly, an object of the present invention is to provide an improved arrangement of adjacent moving and stationary members disposed in a fluid stream, whereby the stimulus imparted to the vibrating members is minimized.

A further object is to provide an improved arrangement for reducing vibration stresses in the associated nozzles and blades of a turbo-machine such as an axial

or radial flow turbine, or an axial or radial flow compressor, or an analogous machine.

A still further object is to provide an improved design method for the arrangement of turbo-machine nozzles and blades (or "buckets") which lends itself readily to mathematical analysis by methods developed in the electrical communication arts, without resort to the empirical or "rule-of-thumb" design methods which previously had to be relied upon.

Another object is to provide a turbo-machine design method which effects substantial reductions in the noise level of the machine.

Other objects and advantages will become apparent, and the improved design method will be understood from a consideration of the following description taken in connection with the accompanying drawings, in which FIG. 1 is a diagrammatic plan view of a turbine nozzle ring having a complete circumferential row of nozzles designed in accordance with the invention; FIG. 1*a* is a diagrammatic representation of a bucket-wheel intended to be used with the nozzle ring of FIGURE 1; FIGS. 2, 2*a* show a conventional nozzle ring and a bucket wheel designed in accordance with the invention; FIGS. 3, 3*a* represent a nozzle ring and bucket wheel both incorporating the characteristic feature of this invention; FIGS. 4, 5, 6 and 7 are graphical representations of certain design characteristics and results achieved with a turbo-machine nozzle ring or bucket wheel designed in accordance with FIGS. 1, 2, and 3; and FIGS. 8, 9 and 10 illustrate the application of the invention to centrifugal compressors.

Generally stated, the invention is practiced by designing one or both of the cooperating members of a turbo-machine stage so that the "frequency" at which successive impulses are imparted by fluid-dynamic forces to the associated member changes continuously and progressively, throughout one complete revolution of the rotating member. A single pattern of frequency changes occurring only once in each revolution, results in optimum reduction of stimulus.

In describing this novel design method, the following definitions and notations will be employed, in explaining one example of a particular application.

$D=2R$ pitch diameter, the mean or nominal diameter of the fluid-delivering nozzle member (see FIG. 1).

P circumferential distance between similar points on adjacent fluid-delivering nozzles, the required variation of which is determined in accordance with the "modulated pitch" method described below.

F frequency, the integral number of fluid-delivering nozzles which would be included in a complete circumferential row of the modulated pitch member if all the nozzles had the same circumferential pitch P as that of a given nozzle. As evidenced by the formula

$$F=\frac{\pi D}{P}$$

"F" varies inversely with "P."

F_m median frequency, a particular frequency "F" representing an average frequency which occurs at some point around the circumference from the "starting point" "0," as defined below.

Δf frequency deviation at a given location along the circumference from median frequency "F_m" (see FIG. 4).

ΔF maximum frequency deviation from the median frequency "F_m"; in other words, maximum value of "Δf" (see FIG. 4).

0 "starting point," the location in the circumferential row of modulated pitch nozzles at which the single

pattern of frequency changes begins and ends (see FIG. 1).

The significance and interrelation of these factors will become apparent from the following description of the theory and practice of this "modulated pitch" turbomachine design.

FIG. 1 is a purely diagrammatic representation of the nozzle ring of an axial flow turbine in which the nozzle blades, one edge of each of which is identified by the radial lines **1**, are spaced circumferentially in accordance with our modulated pitch theory. It will, of course, be appreciated by those familiar with turbine design that this nozzle ring comprises annular concentric wall members **2, 3** which with each pair of adjacent radial partitions **1** defines a fluid delivering nozzle. The complete ring of nozzles does not form a homogeneous annular stream of fluid, but instead produces a series of disturbances in the fluid stream, by reason of the finite thickness of the nozzle partitions **1**.

The pitch diameter D of the nozzle ring and the circumferential pitch P of a single nozzle are identified on FIG. 1. The starting position "0" is at "12 o'clock."

In accordance with one embodiment of the invention, each nozzle (except at "0") has a pitch P which is slightly larger than the preceding nozzle, and slightly smaller than the succeeding nozzle. That is, the first nozzle **1***a* has the smallest circumferential pitch P of any nozzle in the ring, and the last nozzle **1***z* has a larger circumferential pitch P than any other. The intermediate nozzles increase in pitch progressively in the clockwise direction by a specified increment. The manner in which this increase in pitch of the nozzles is determined will be seen in more detail from the following discussion of FIGS. 4, 5, 6 and 7.

It is to be assumed that a special nozzle ring in accordance with FIG. 1 will have an associated bucket-wheel, as illustrated in FIG. 1*a*, with a shaft **7***a*, disk **7***b*, and a plurality of radially disposed vanes (or "buckets") **7***c*, which will ordinarily be equally spaced around the circumference of the wheel. If the nozzle ring consists of 24 nozzles, as in FIG. 1, then a given bucket **7***c* rotating 360° clockwise from the "0" position will be acted upon by 24 discrete impulses from the nozzles **1***a*, **1***b*, **1***c*, **1***d* . . . **1***z*. If these 24 nozzles were of identical pitch and uniform distribution around the nozzle ring, then a given bucket would experience 24 evenly spaced impulses during each revolution of the bucket-wheel. Because the nozzles, in accordance with the invention, are not of equal circumferential pitch, it is necessary to introduce and define the concept of the "frequency," identified F in the above notation, which corresponds to the "pitch" P of a given nozzle.

Accordingly, the "frequency" of a given nozzle is defined as the integral number of nozzles which would constitute a complete 360° ring if all were equal in pitch to the nozzle in question and uniformly distributed throughout the circle. Thus, for example, the minimum pitch nozzle **1***a* may have a "frequency" F of 34, since it would require this many nozzles identical to **1***a* to make a complete 360° ring of nozzles of pitch diameter D. Similarly, the frequency of the last nozzle **1***z* may be 14, since that many identical nozzles would be required to make a complete ring of the same pitch diameter D. Thus, for the purpose of this discussion, the "frequency" of a given fluid-delivering nozzle member is defined as the number of identically spaced nozzles required to form a complete ring. And, to each nozzle in the ring can be assigned a frequency F, which decreases continuously in the clockwise direction, as the circumferential pitch P of the nozzles increase. Thus, the frequency of the nozzles changes, or is "modulated," throughout the nozzle ring circumference. Therefore, we have provided a "frequency modulation" method of varying the frequency of the impulses applied by the fluid disturbances to the moving buckets.

The precise manner in which the frequency of the nozzle varies around the circumference of the nozzle ring, and the manner in which the change in circumferential pitch P of the nozzles is determined, will now be described.

In FIG. 4 the median frequency F_m is represented by the broken horizontal line. The initial frequency, that of nozzle **1***a*, is indicated by the maximum ordinate of the curve **4**. The minimum ordinate of the curve, at the 360° point, represents the frequency of the last nozzle **1***z*. It is to be observed that the deviation of the frequency of a given nozzle from the median frequency F_m is denoted by the symbol Δf. The maximum deviation of the frequency of any nozzle from the median frequency F_m is denoted by ΔF. It is to be particularly noted that the maximum positive deviation ΔF of the bucket **1***a* from the median F_m is equal in magnitude to the maximum negative deviation of the frequency of bucket **1***z*. In this example, curve **4** is actually drawn as a straight line. In other words, the frequency is modulated as a straight-line function of distance from "0" around the circumference of the nozzle ring. To be completely rigorous, the curve **4** would have to be represented by a series of discontinuous points, each point being on the curve at a location along the abscissa corresponding to the middle of the nozzle which the point represents, since there is only one frequency for each nozzle and each nozzle has a finite pitch P. However, for design purposes, the curve **4** is drawn as a continuous line and then used to ascertain the precise pitch required for each nozzle as follows.

In using this design method, it will be first necessary to ascertain the nominal pitch diameter D of the nozzle ring and the approximate number of nozzles desired in 360°. This will be done in accordance with the turbine designer's experience concerning the optimum height-to-width or "aspect ratio" for a turbine nozzle, and other well-known factors. It is also necessary to select the "median frequency," which will ordinarily be the number of nozzles in a complete ring of the same nominal diameter "D" that would be used in accordance with previously known turbine design methods. In addition, the integral number representing the maximum deviation ΔF from this median frequency must be selected. For instance, in the example represented by FIGS. 1 and 4, assume that the selected median frequency F_m is 24 and that the selected maximum deviation in frequency ΔF is 10. It follows that the frequency of nozzle **1***a* will be 34 and the frequency of nozzle **1***z* will be 14. This means that the circumferential pitch P of the nozzle **1***a* will occupy $\frac{1}{34}$ of the circumference of the nozzle ring. The "frequency" of nozzle **1***a* is represented in FIG. 4 on curve **4** by the ordinate $(F_m+\Delta F)$ at the "0" point on the abscissa.

FIG. 5 is a plot of the resulting circumferential pitch P of succeeding nozzles, and is derived from FIG. 4 by the formula

$$F=\frac{\pi D}{P}$$

Here again, the abscissa represents the distance around the "pitch circle" of the nozzle ring from the starting point "0"; and the ordinate represents the circumferential pitch P. It will be apparent that the nozzle pitch curve **5** is an inverse function of the frequency curve **4** in FIG. 4. Specifically, the nozzle pitch P is represented by the general expression

$$\left(\frac{\pi D}{F_m \pm \Delta f}\right)$$

The pitch "P_a" corresponding to the "frequency" of nozzle **1**a is represented in FIG. 5 on curve **5** by the ordinate

$$\left(\frac{\pi D}{F_m+\Delta f}\right)$$

at the "0" point on the abscissa. The specified pitch "**1**a" is now laid off along the abscissa of FIG. 5 starting at "0," and the ordinate of curve **5** at this point on the abscissa defines the second nozzle pitch "**1**b." Accordingly, the pitch "**1**b" is laid off next to "**1**a"; and then the ordinate of curve **5** at this point on the abscissa defines the third nozzle pitch "**1**c." Repetition of this process will determine the circumferential pitches of succeeding nozzles around the periphery of the nozzle ring.

It will, of course, be understood by turbine designers that it is necessary to take into consideration the finite thickness of the nozzle partitions in determining the pitch of the nozzles. It is also necessary to limit the maximum variation in nozzle pitch, defined by the maximum frequency deviation ΔF, to a value which will be acceptable from the standpoint of aerodynamic performance. It may even be necessary to make the shape of the nozzle passages of different aerodynamic designs, in accordance with the variation in pitch, in order to achieve as large a maximum frequency deviation as possible, while retaining good aerodynamic performance from all nozzles.

Analysis of a nozzle ring pitch modulated as in FIG. 1 by the mathematical theories applicable to radio "frequency modulation," shows that the above discussed arrangement reduces the stimulus to a practical minimum for a specified frequency deviation ΔF. Although the complexities of such mathematical theory make it impractical to go into details here, some fundamental definitions and relationships will be discussed briefly in order to better understand the advantages, theory, and design philosophy behind this invention.

If the nozzles were distributed around the circumference of the nozzle ring (of pitch radius R) with a uniform pitch P, to which corresponds a constant frequency F, then the distribution of the magnitude of discrete impulses caused by the nozzle partitions may be described conventionally as

$$Y(x)=Y\sin\phi_x=Y\sin\left(\frac{2\pi}{S}X\right) \qquad (a)$$

where Y is a constant amplitude of the sinusoidal disturbance, x is the variable central angle in radians, and $S=P/R$ is the central angle which corresponds to pitch P at radius R. Phase

$$\phi_x=\frac{2\pi}{S}x$$

and

$$\frac{2\pi}{S}$$

is a constant along the circumference and equal to F. However, if

$$\frac{2\pi}{S}$$

is not constant but varies along the circumference, as in a "pitch-modulated" nozzle ring, then

$$\frac{2\pi}{S}$$

becomes

$$\frac{2\pi}{S_x}$$

which is an instantaneous value of the pitch angle and instantaneous phase

$$\phi_x=\int\frac{2\pi}{S_x}dx \qquad (b)$$

and Equation a becomes more general, thus

$$Y(x)=Y\sin\left[\int\frac{2\pi}{S_x}dx\right] \qquad (c)$$

It is convenient to replace angular pitches S_x by the corresponding frequencies

$$F_x=\frac{2\pi}{S_x}$$

Equation c becomes then, in terms of frequencies,

$$Y(x)=Y\sin\left[\int F_x dx\right] \qquad (d)$$

Consider a nozzle ring, as above, with nozzles of equal pitch and frequency F_m, the median frequency, which will be designated from now on as "carrier frequency F_c." This carrier frequency F_c is varied or modulated along the pitch circle of the nozzle ring. Theoretically, the "cycle of modulation" may repeat itself an integral number of times around the nozzle ring, in which case the number of such repetitions is called the "modulation frequency f." The carrier frequency F_c may be modulated in any desired manner. For the purpose of clarity we assume that the variation of frequency F_c within a modulation cycle takes place in a sinusoidal manner, as in present day FM radio transmission. The amplitude of modulating sinusoid we designate as ΔF and define as "maximum frequency deviation."

The modulating frequency is then

$$\Delta F\sin fx$$

and the modulated carrier frequency F_c becomes

$$F_x=F_c+\Delta F\sin fx \qquad (e)$$

Amplitude (Equation c) will become

$$Y(x)=Y\sin\left[\int(F_c+\Delta F\sin fx)dx\right]$$

or after integration

$$Y(x)=Y\sin\left[F_cX-\frac{\Delta F}{f}\cos fx\right]$$

or

$$Y(x)=Y\sin\left[F_cX-\beta\cos fx\right] \qquad (f)$$

and phase becomes

$$\phi_x=\int F_x dx=F_cX-\beta\cos fx \qquad (g)$$

where:

X is the variable central angle
F_c is carrier frequency
f is modulation frequency
ΔF is maximum frequency deviation

$$\beta=\frac{\Delta F}{f}\text{ is the "Modulation Index"}$$

Y is the amplitude of the sinusoidal disturbance
Y_x is the local magnitude of disturbance corresponding to angle X.

The physical meaning of Equation e becomes clear after it is expanded into a Fourier Series:

Fourier expansion indicates that a sinusoidal modulation with frequency deviation ΔF, modulation frequency f and modulation index β of a carrier frequency F_c with amplitude Y results in the following:

(a) In addition to the carrier frequency F_c an infinite number of harmonic oscillations with other frequencies occurs. These frequencies are F_c+f, F_c-f, F_c+2f, F_c-2f, F_c+3f, F_c-3f, F_c+nf, F_c-nf, etc. They are spaced apart by the modulation frequency f. The higher frequency components above the carrier frequency are called "upper sideband" frequencies and the lower frequencies below the carrier frequency "lower sideband" frequencies. However, for practical purposes all components with significant amplitudes are confined within a frequency band equal to twice the maximum frequency deviation $2\Delta F$. In other words the frequency modulation creates a whole band of other frequencies on both sides of the carrier frequency F_c.

(*b*) The new amplitude of the carrier frequency F_c and the amplitudes of the new sideband frequencies are equal to the unmodulated amplitude of the carrier frequency Y multiplied by certain coefficients, all less than 1. The coefficients are values of Bessel functions (first kind) whose order is equal to the order of the particular sideband frequency and whose argument is modulation index β. The coefficient for the new amplitude of the carrier frequency F_c is equal to the value of Bessel function of zeroth order for the same index β.

(*c*) The energy content of each harmonic is proportional to the square of its amplitude, therefore, the whole energy content of the unmodulated carrier of amplitude Y and frequency F_c is dispersed throughout the upper and lower sidebands in a manner proportional to the square of the amplitude of the respective harmonic. It appears then that the most effective way to disperse the energy of the carrier is to distribute it uniformly throughout the band width by making all harmonic amplitudes equal which also results in minimum harmonic amplitude. With a sinusoidal modulation this is not possible to accomplish, as for practical values of β (modulation index) the amplitudes of harmonics differ considerably and some are not a small fraction of the unmodulated carrier amplitude. Only when β reaches very large values (approaching infinity) is this theoretically possible for sinusoidal modulation.

(*d*) It has been discovered that, if the number of modulation cycles f was limited to only one in a complete nozzle ring (i.e. $f=1$) and if the manner of modulation was confined to a gradual and preferably uniform or almost uniform increase of frequency, from a minimum value to a maximum value once in 360°, then the energy was effectively and almost uniformly dispersed throughout the band width, also for practically acceptable values of β.

(*e*) The previously discussed linear or saw-tooth modulation of carrier frequency F_c shown in FIGURE 4, very efficiently disperses energy throughout band width $2\Delta F$.

FIGURE 7 shows the amplitude frequency spectrum for a nozzle ring with $F_c=60$, maximum frequency deviation $\Delta F=14$, modulation frequency $f=1$; and

$$\beta=\frac{\Delta F}{f}=14$$

It can be seen that the maximum amplitude of any harmonic in the spectrum does not exceed 22% of the unmodulated amplitude of the carrier. The reduction of stimulus achieved by modulating with such a single saw-tooth (linear) cycle is shown in FIGURE 6. Here the abscissa of curve **6** is the modulation index β, which for this case, when $f=1$, is identical with maximum frequency deviation ΔF. The ordinate of curve **6** represents the maximum sideband amplitude as a percentage of the amplitude of the unmodulated carrier. It is seen that for a maximum frequency deviation of 4, the maximum amplitude is 42% of the unmodulated carrier; and a ΔF of 16 reduces the amplitude to only 21% of its original magnitude.

In the example discussed in detail above, where $F_c=24$ and $\Delta F=34-24=24-14=10$, the curve 6 indicates that amplitude of any sideband does not exceed 27% of the amplitude of the unmodulated carrier.

It is believed that for most purposes a maximum frequency deviation of between 4 and 12 will be practical. This will result in a reduction of stimulus by about 60% to 75% in a modulated nozzle ring, as compared with the stimulus obtained with a conventional unmodulated nozzle ring.

While so far the invention has been described as applied to the stationary nozzle ring of an axial flow turbine, it may also be applicable to the rotating wheel. In this case the stationary nozzles might be of uniform pitch and distribution around the nozzle ring; and the modulation theory would be applied to the arrangement of the buckets on the moving wheel. Such an arrangement is illustrated in FIGS. 2 and 2*a*, in which the stationary nozzle ring has identical nozzles **7** designed in accordance with conventional turbine practice, while the rotating bucket wheel **8** has a plurality of buckets **9** spaced in accordance with the modulated pitch theory. That is, the initial bucket **9*a*** is of a minimum pitch and maximum frequency, while the last bucket **9*z*** has a maximum pitch and minimum frequency.

Furthermore, the modulated pitch design may be applied to both the stationary nozzle ring and the moving bucket wheel, as illustrated in FIGS. 3, 3*a*. Here, the stationary nozzles **10** are spaced similarly to the nozzles **1** of FIG. 1, while the moving buckets **11** are spaced as described in connection with the buckets **9** of FIG. 2*a*.

The invention may also be applicable to analogous turbo-machinery, for instance centrifugal compressors having a rotor with radial vanes defining rotating fluid-delivering "nozzles," the disturbances created by which affect the stationary blades or "vanes" in the diffuser of the compressor. FIG. 8 illustrates such an application, in which the centrifugal impeller **12** has radial blades **12*a*** . . . **12*z*** of modulated pitch, while the diffuser vanes **13** define identical diffusing passages **13*a*** evenly spaced around the circumference of the impeller.

FIG. 9 illustrates how the modulation theory may be applied to stationary diffuser vanes **14** associated with equi-spaced and identical impeller blades **15**.

Or, the modulated spacing may be applied to both the stationary and moving members of a centrifugal machine, as illustrated in FIG. 10. Here the rotating wheel has blades **16** with a modulated spacing, as shown in connection with the blades **12** of FIG. 8; and the stationary member has blades **17** modulated as shown in connection with the diffuser vanes **14** of FIG. 9.

FIGS. 8, 9, or 10 may also be taken to represent "radial in-flow" turbines, in which the stationary diffuser vanes form the fluid-delivering nozzles, and the fluid jets delivered inwardly therefrom drive the central rotor. Likewise, the axial flow structures of FIGS. 1, 1*a*, 2, 2*a*, 3, 3*a* may be considered to be stages of axial flow compressors.

With respect to FIGS. 1, 1*a*, it is comparatively easy to see how, as described above in connection with the design method and data presented in FIGS. 4, 5, 6, 7, the invention works when the "modulated" stationary nozzles of FIG. 1 are delivering jets of varying "frequency" to the rotating buckets **7*c*** of FIG. 1*a*. It is apparent that a given bucket **7*c*** will experience impulses of progressively changing "frequency" as it rotates 360° from the initial position "0" past the variably spaced nozzles of FIG. 1. However, it is not quite so obvious why the modulated spacing of the buckets in FIG. 2*a* has a beneficial effect on the vibration characteristics of the structures.

In this connection, it is to be noted that the radially extending nozzle partitions which form the uniformly pitched nozzles **7** of FIG. 2 are also subject to destructive vibration forces. Even though the fluid is flowing from nozzles **7** to the bucket wheel **8**, a given nozzle blade will experience an impulse each time a bucket **9** passes. This is because passage of the closely adjacent bucket **9** produces a small local disturbance in the fluid pressure distribution adjacent the exit edges of the stationary nozzle blades, and the passage of this small pressure disturbance results in the application of a vibration-inducing impulse to the radially extending wall partition of the stationary nozzle member. Thus, modulating the pitch of the moving buckets **9** as in FIG. 2*a* has a beneficial effect in reducing the vibration forces applied to the radially extending nozzle partitions or blades of the stationary nozzle ring member **7**.

Likewise, with respect to the centrifugal compressor of FIG. 8, the fluid is flowing from the radially extending impeller passages defined between the modulated blades **12** into the stationary passages defined between the diffuser vanes **13**. The rotor **12** is in effect a "rotating nozzle" member delivering discrete fluid jets into the passages **13**a defined between the stationary vanes **13**. The vibration characteristics would be substantially the same if rotor **12** were held stationary and the annular diffuser member carrying the vanes **13** were permitted to rotate as a radial flow turbine bucket-wheel. In either case, the modulation of the spacing of the fluid-delivering members **12**a prevents destructive vibration forces being built up in the cooperating annular row of blades **13**. Furthermore, if the structure of FIG. 9 is considered a radial in-flow turbine, then it will be obvious that the motive fluid delivered inwardly through the passages **15**a . . . **15**z so as to drive the rotor **15** would have quite similar vibration characteristics to the modulated pitch nozzle of FIG. 1 cooperating with the uniformly spaced buckets of FIG. 1a. But it will also be appreciated by those familiar with the centrifugal compressor art that each of the uniformly spaced rotor blades **15** of FIG. 9 experience a slight impulse due to changes in local pressure distribution around the adjacent edges of the respective diffuser vanes **14**, caused by passage of the rotor blades, so that the modulated pitch of vanes **14** prevents destructive vibrations being set up in the tip portions of the compressor blades **15**.

It may also be noted that it is not necessary that the number of the "fluid-passing nozzles" be equal to the number of the associated relatively rotating structures. Thus in FIG. 1 the stationary nozzle comprises 24 modulated pitch nozzles, while the bucket wheel of FIG. 1a has 32 blades or buckets 7c. In FIG. 2 the stationary nozzles are shown with 20 uniformly pitched members, while the bucket wheel in FIG. 2a has 24 modulated pitch blades. By the same token, the modulated pitch nozzle ring of FIG. 3 need not have the same number of elements as the modulated pitch bucket wheel of FIG. 3a.

Nor is it necessary that the change in frequency should occur in the same direction on each member. In FIG. 3 and 3a the pitch of the nozzles **10** and the pitch of the buckets **11** both increase in the clockwise direction. This is not necessary, and in FIG. 10 the rotor blades are spaced with an increasing pitch in the clockwise direction, while the stationary diffuser blades have an increasing pitch in the counterclockwise direction.

The above discussion of the various possible permutations and combinations will show that the important criterion in the practice of the invention is that, if a given one of a pair of adjacent relatively rotatable members has a portion subject to destructive vibration forces, then any adjacent discontinuities in the other member may advantageously be spaced in accordance with the modulated pitch theory of the invention in order to prevent such destructive vibration. It is immaterial which member is stationary and which is moving, and it is likewise immaterial whether the fluid is flowing from the modulated pitch member towards the member subject to vibration, or in the reverse direction, or if the adjacent relatively rotatable members are not actually "delivering" a continuous flow of fluid from one to the other but are merely rotating adjacent each other so that a pressure distribution is created by circumferentially spaced discontinuities on the rotating bodies, so that the small local pressure irregularities tend to set up vibrations in the vibration-responsive portions of the other member.

The invention is also applicable to other rotating structures, such as fans, propellers, etc. The intensity of the aerodynamic noise level of such apparatus may be greatly reduced by appropriately "modulating" the spacing of the fluid-delivering or fluid-passing nozzle members. In such apparatus, the advantages of noise level reduction will be obtained even though there is no stationary member associated with the rotor subject to mechanical vibrations. In other words, the noise is caused by vibration or discontinuity of the fluid stream itself. Thus it will be apparent that an axial flow fan or propeller having no associated stationary diffuser member exerts somewhat of a "siren effect" on the column of air moving through it. That is, the rotating impeller serves to interrupt the uniform column of fluid moving through the rotor and these periodic interruptions generate a frequency which shows up as "noise." If the interruptions are uniform, then a definite pitch of sound will be produced. By appropriately modulating the fluid-passing members, the net average noise level can be greatly reduced.

Thus it will be seen that the invention provides a basically new approach to the theoretical design of rotating fluid-dynamic machinery in which a "fluid-passing member" produces a cycle of disturbances in the fluid flow which are likely to set up sound waves and destructive vibrations in associated members. By use of the principles of frequency modulation radio transmission, as outlined herein, it appears possible to reduce the vibration stimulus applied to the associated vibrating member to a degree wholly impossible with the design methods known to the prior art.

While only a few applications of the theory have been described specifically herein, it will be obvious to those skilled in the art that many other applications may be made; and it is of course intended to cover by the appended claims all such modifications as fall within the true spirit and scope of the invention.

What we claim as new and desire to secure by Letters Patent in the United States is:

1. In a fluid-dynamic machine, the combination of first and second relatively rotatable members, one of which defines an annular row of fluid-passing nozzle members, the other of which has a circumferential row of members subject to vibration, said nozzles having a frequency of spacing which changes substantially uniformly and progressively in the same direction, the pattern of frequency changes occurring only once throughout each 360° of relative rotation.

2. In a fluid-dynamic machine, the combination of first and second relatively rotatable members one of which defines an annular series of fluid-delivering nozzle members and the other of which has at least one member receiving impulses from the jets delivered by said nozzles and having a tendency to vibrate at an inherent natural frequency, means causing the frequency of the vibration impulses applied to the fluid-receiving member to vary in discrete increments progressively and substantially uniformly throughout each 360° of relative rotation of the fluid-delivering and fluid-receiving members, the single pattern of such frequency changes occupying the full 360° of such relative rotation, whereby the net effective vibration stimulus applied to the fluid-receiving member is reduced.

3. In a fluid-dynamic machine, the combination of a first stationary member and a second associated relatively rotatable member, the stationary member comprising an annular series of fluid-passing nozzles and the rotating member having a plurality of uniformly spaced structures subject to vibration induced by the fluid discharged from the stationary member, said nozzles having a fresquency of spacing which increases progressively and substantially uniformly from a minimum frequency to a maximum frequency, the pattern of said frequency changes occurring only once around the complete periphery of the stationary nozzle member.

4. In a fluid-dynamic machine, the combination of a first stationary member and a second relatively rotatable member, the rotating member having a circumferential row of discrete structures, the stationary member having an annular row of uniformly spaced fluid-passing nozzles with wall portions subject to vibration from impulses caused by the passage of the structures on the associated rotating member, said rotating structures having a frequency of spacing which changes progressively and sub-

stantially uniformly from a minimum frequency to a maximum frequency, the pattern of said frequency changes occurring only once around the complete periphery of the rotating member.

5. In a fluid-dynamic machine, the combination of first and second relatively rotatable members one of which defines an annular row of fluid-passing nozzle members and the other of which has at least one member receiving fluid impulses derived from said relative rotation, the frequency of the spacing of said nozzle members varying progressively and substantially uniformly, the single pattern of said frequency changes occupying the full 360° of the nozzle member, whereby the net effective vibration stimulus imposed on said other member is reduced.

6. In a fluid-dynamic machine, the combination of a first member defining an annular row of fluid-passing structures, and at least one relatively rotatable second member adjacent said first member and subject to vibration, said fluid-passing structures having a fraquency of spacing which changes progressively, in discrete increments and substantially uniformly from a minimum frequency to a maximum frequency, the pattern of said frequency changes occurring only once around the complete periphery of the first member.

7. In a fluid-dynamic machine, the combination of a first member and a second adjacent and relatively rotatable member, said first member having an annular series of fluid-passing nozzles including wall members subject to vibration due to fluid impulses derived from the passage of discrete portions of the adjacent second member, said second member having an annular row of fluid-passing structures with portions also subject to vibration due to fluid impulses derived from relative motion of the nozzles of said first member, said relatively rotatable structures each having a frequency of spacing which changes progressively and substantially uniformly, the respective patterns of said frequency changes occurring only once around the periphery of the respective members.

8. In an axial flow turbine, the combination of a stationary member defining an annular row of fluid-delivering nozzle members, a bucket wheel supported for rotation concentric with said nozzle ring and having a circumferential row of equally spaced bucket members having a tendency to vibrate due to the impulses from the fluid jets from said nozzles, the nozzles being of a circumferential width which changes progressively and substantially uniformly throughout the full 360° of the nozzle ring, whereby the frequency of spacing of the nozzles is modulated according to a single pattern of changes progressively and substantially uniformly in the same direction throughout each complete revolution of the bucket wheel relative to the nozzle ring.

9. In a fluid-dynamic turbo-machine, the combination of a stationary member defining an annular row of fluid-delivering nozzles, a relatively rotatable wheel adjacent said stationary member and having an annular row of equally spaced bucket members adapted to receive motive fluid from said nozzles, the nozzles of said stationary member having a frequency of spacing which changes substantially uniformly and progressively in the same direction, the pattern of frequency changes occurring only once around the periphery of the nozzle ring member, whereby the vibration stimulus applied to the moving buckets by the jets issuing from said nozzles is reduced.

10. In a fluid-dynamic turbo-machine, the combination of a first nozzle ring member having an annular row of uniformly spaced nozzles having wall portions subject to vibration, a relatively rotatable bucket wheel having a circumferential row of bucket members adapted to receive motive fluid from said stationary nozzle ring, the frequency of spacing of said buckets changing progressively in the same direction and substantially uniformly, the pattern of frequency changes occurring only once around the complete periphery of the bucket wheel.

11. In a fluid-dynamic turbo-machine, the combination of a first nozzle member with an annular row of fluid delivering nozzles, a relatively rotatable wheel having a circumferential row of blades adapted to receive motive fluid from said nozzles, both said nozzles and said blades having a frequency of spacing which changes progressively and substantially uniformly, the pattern of said frequency changes occurring only once around the complete periphery of the respective nozzle and wheel members.

12. In a radial flow turbo-machine, the combination of an inner member having an annular row of radially extending vane members defining fluid passages, and an outer annular member concentric with said inner member and having an annular row of spaced vane members defining radially extending fluid passages, at least one of said annular rows of vanes being unequally spaced to provide a frequency of spacing which changes progressively and substantially uniformly, the pattern of said frequency changes occurring only once around the complete periphery of the member, whereby vibration resulting from the fluid impulses derived from the relative rotation of the variably spaced anular row of vanes and imposed on the vanes of the other annular row are reduced.

13. A fluid-dynamic machine having a rotating member with a circumferential row of fluid-passing structures, the frequency of spacing of said structures changing progressively and substantially uniformly around the periphery of the member, the single pattern of said frequency changes occupying the entire 360° of the periphery of the member, whereby the net average noise level of the discrete fluid jets passing said structures is reduced.

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