Nov. 9, 1954 A. D. S. CARTER 2,693,905

ELASTIC FLUID COMPRESSOR

Filed March 13, 1952 3 Sheets-Sheet 1

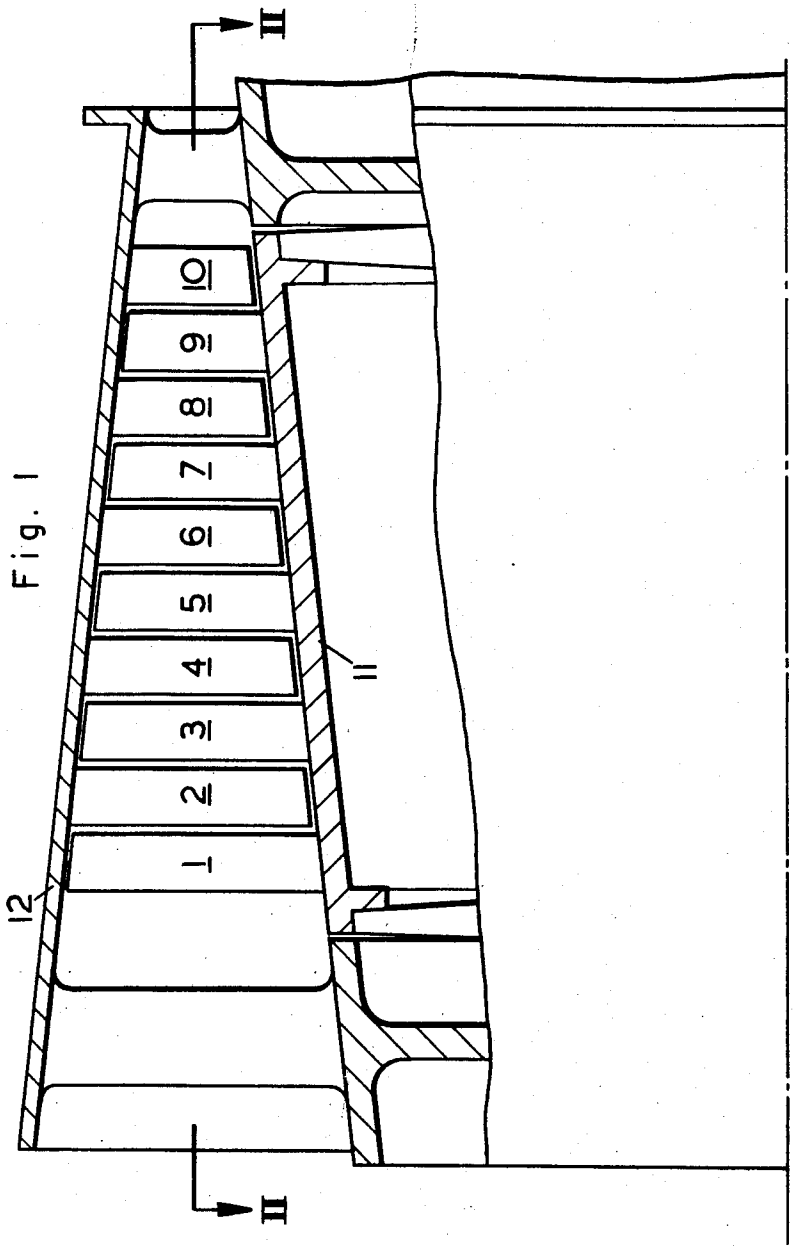


Fig. 1

Inventor
Alfred Denis Snowdon Carter
By
Stevens, Davis, Miller & Mosher
Its Attorneys

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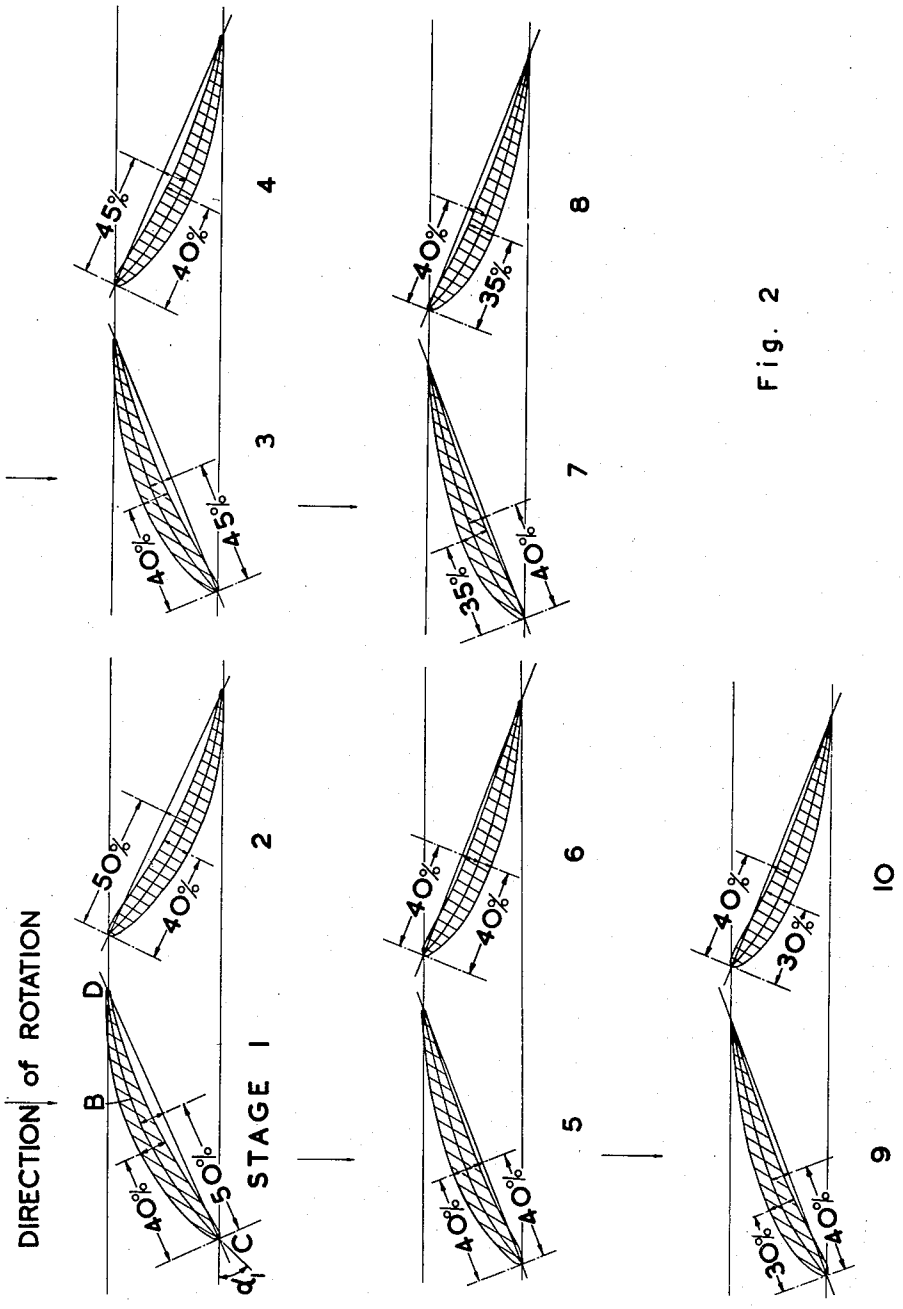


Fig. 2

Alfred Davis Swandon Carter Inventor
By
Stevens, Davis, Miller & Mosher
his Attorneys

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ELASTIC FLUID COMPRESSOR

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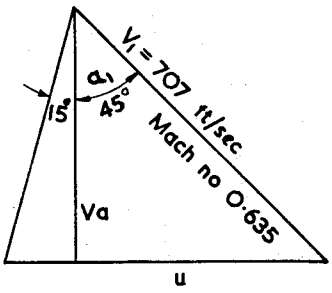


Fig. 3a

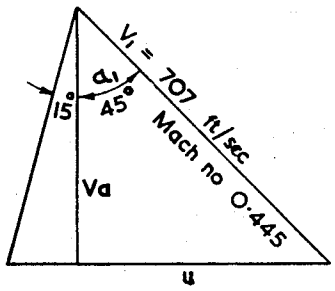


Fig. 3b

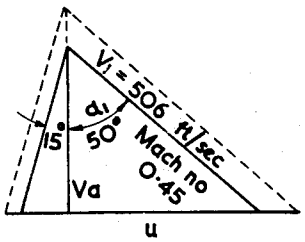


Fig. 4a

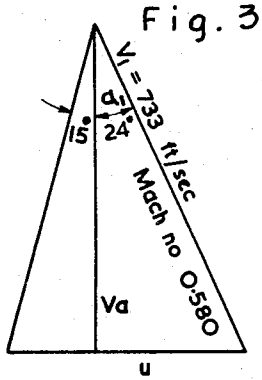


Fig. 4b

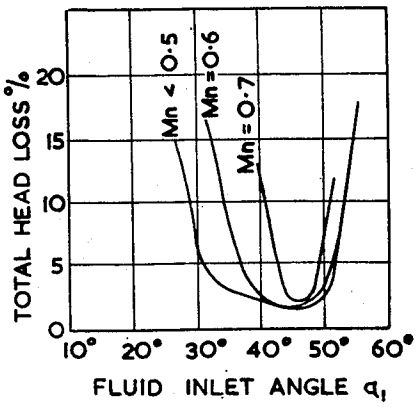


Fig. 5a

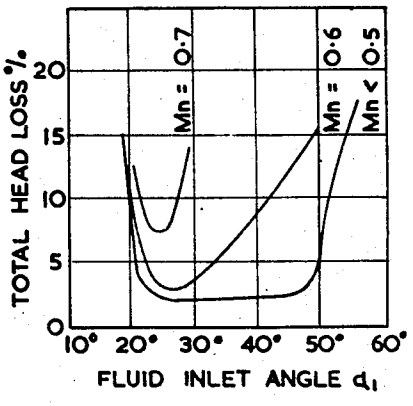


Fig. 5b

Inventor
Alfred Denis Swanton Carter
By
Stevens, Davis, Miller & Mosher
Attorneys

United States Patent Office 2,693,905 Patented Nov. 9, 1954

2,693,905

ELASTIC FLUID COMPRESSOR

Alfred Denis Snowdon Carter, Farnborough, England, assignor to Power Jets (Research and Development) Limited, London, England, a British company

Application March 13, 1952, Serial No. 276,340

Claims priority, application Great Britain March 22, 1951

7 Claims. (Cl. 230—122)

This invention relates to elastic fluid compressor of the dynamic as distinct from the positive displacement type, and in particular to rotary bladed multi-stage axial flow compressors, and is concerned with the design of the blading thereof.

In general, profiled blades in cascades as used in such machines may operate efficiently only over a limited range of the angle of approach or incidence of the impinging fluid stream, there being an extremely rapid increase in the losses associated with the fluid flow through the blades beyond this incidence range. Consequently, while such a compressor may be provided with blading which operates at particular designed conditions of fluid flow at an efficient incidence, it is always prone to a severe loss of efficiency should changing flow conditions lead to an adverse change in the incidence of any part of the blading. The ability of a compressor to avoid such a possibility is affected by its structural form. For example, by mounting each or some of the compressor stages on rotationally independent shafts, the rotational speed of each (upon which, in each case, the approach angle of the fluid is dependent) may be adjusted to suit changing conditions, or alternatively by mounting all or some of the blades so that they are each capable of angular adjustment about a longitudinal axis, their angular disposition may be adjusted in accordance with changes in the approach angle of the fluid. However, these and similarly flexible multi-stage compressor arrangements involve considerable complexities in regard to constructions and method of control as compared with the more conventional arrangement, in which all stages have fixed blades and are mounted on a common shaft, and are consequently desirably to be avoided. On the other hand, the convenience and simplicity of this more conventional arrangement of multi-stage compressor is negated to some extent by the fact that there is an inherent tendency in such a compressor toward the promotion of adverse operating conditions when it is required to perform at rotational speeds and loads below the designed values, with the result that the effective load range of the compressor may be undesirably limited. This tendency is due to the fact that a reduction in speed and load is accompanied, at each stage through the compressor, by a departure from the designed values of the pressure ratio and so of density ratio of compressed to uncompressed fluid, leading to a redistribution of the axial velocity of the fluid throughout the compressor. A variation in axial velocity at reduced load is not, in itself, undesirable since, at the reduced rotor blade speed, the axial fluid velocity should be modified in order to ensure the correct incidence of the fluid with respect to each blade. The variation which occurs however, does not produce such a result; in general, the mean axial velocity of fluid at compressor inlet tends to fall below and at outlet tends to rise above the most desirable value, with the consequent tendency for the incidence range of the blading at the inlet and outlet stages to be respectively positively and negatively exceeded. The ensuing phenomena, indicative of a rapid loss of compressor performance, are known respectively as stalling (or surging if the instability is great) and choking.

Heretofore this situation has been accepted as inescapable because it is not possible merely by appropriate design of the compressive elements, i. e. the blades, to avoid an adverse redistribution of fluid axial velocities at reduced loads and multi-stage axial flow compressor design has been based in general on the conception of selecting a basic blade profile having, ideally, a wide incidence range at all relative fluid velocities and of modifying this basic profile in respect of its inlet and outlet angles, amount of camber (camber angle), chord and maximum thickness to afford profiles appropriate to the various compressor stages. Obvious practical limitations are imposed on the design conception, however, by the fact that the ideal basic blade profile does not exist.

In fact, although many basic profiles may be selected each having a wide incidence range at low relative fluid velocities, the incidence range is always diminished as the relative fluid velocity is increased. The difficulty of selecting a basic blade profile is aggravated further by the fact that at the inlet stages of a compressor, although as previously described the mean axial velocity of fluid is reduced at reduced compressor loads, the axial velocities at various radii are disproportionately modified (i. e. the axial velocity profile is changed). For instance, when a compressor is operating at the designed load conditions, the tip of an inlet stage blade may be subjected to high fluid approach velocities at high incidences. At reduced loads, however, the fluid approach velocities at radially intermediate parts of the blade will be reduced and the incidences increased, but at the blade root and/or tip (depending on the type of design) the relative fluid velocities may still be high. The inlet stages, therefore, are always operating at high incidences and often at high fluid approach velocities. On the other hand, in the outlet stages of the compressor the velocity profile is substantially unaltered. All fluid approach velocities are increased relatively to the blade speed and consequently the incidences are reduced until choking occurs. Moreover, the incidence range of any particular blade is influenced not only by the relative fluid velocity but in addition by the temperature of the fluid; it is convenient, therefore, to compare performances of various blade profiles in terms of incidence range at varying Mach numbers of fluid flow, the Mach number of a given fluid flow being dependent on its relative velocity and its temperature. This method of comparison is necessitated by the fact that the temperature of the fluid in a multi-stage compressor tends to rise through successive stages so that the Mach number of the flow is greater at inlet than at outlet for a given relative fluid velocity.

The present invention has for an object to provide an improved basis for the design of the blading of multi-stage axial-flow rotary elastic fluid compressors, more especially but not exclusively of the single rotor, fixed blade type with a view to improving the characteristics thereof at varying loads. The invention has its origin in the realisation that the difficulty outlined in the previous paragraph would be mitigated to some extent if blades were used for each of the compressor stages whose characteristics as regards the relationship between their incidence range and Mach number were related to the fluid flow conditions obtaining in the particular stage at various loads, and the invention accordingly proposes the use of such blading.

As mentioned above basic blade profiles in general have the characteristics of a reduced incidence range with rising relative fluid velocities or, more conveniently expressed, with rising Mach number. However, between individual profiles the mode of such reduction of incidence range changes; thus, for various individual blades, the particular incidence which may be departed from substantially equally either positively or negatively without substantially increased losses, that is the mean incidence, may, with progressively increased Mach numbers, be either reduced, or increased, or remain constant, depending on the profiles of the various blades. That is to say, the "centre of gravity" of the incidence range varies with varying Mach numbers. In compressors based on the known design conception outlined in the foregoing, this characteristic was substantially constant for the blading of all stages because the characteristic of a basic blade profile is not materially affected by changes in its inlet and outlet angles, amount of camber (camber angle), chord and maximum thickness.

The invention accordingly proposes, in a multi-stage

axial flow rotary elastic fluid compressor, that blading be used for the inlet stages having a rising mean incidence for rising Mach numbers and for the outlet stages having a falling mean incidence for rising Mach numbers, the blades at intermediate stages having mean incidence to Mach number characteristics graduated between these extremes. Thus in a compressor according to the invention, a reduction in load from the designed value will involve at the inlet stages two conditions of flow, namely high Mach numbers at high approach angles toward the outer region of the blades and low Mach numbers at high approach angles over the remainder of the blades, both corresponding to the incidence range characteristic of the inlet blades, and at the outlet stage simply increased Mach numbers with falling approach angles similarly corresponding to the incidence range characteristic of the outlet blades.

With this prospect in view, investigations have been carried out which show that the incidence range characteristics of a blade profile may be controlled, in a manner acceptable from more general fluid dynamic considerations, by varying the position (expressed as a percentage of chord from the profile leading edge) of the point either of maximum thickness or of maximum camber.

It is found, for a given basic blade profile, that the effect of decreasing or increasing the distance from the leading edge of the point of maximum camber or of maximum thickness (other factors being constant) is in each case to increase or decrease respectively the mean incidence (i. e. the centre of the incidence range) at reduced Mach numbers. In each case also however there is an adverse effect in that the critical Mach number (that is, the Mach number above which losses become excessive at any incidence) is somewhat reduced, but this occurs to a lesser extent in the case of decreasing distance from the leading edge of the point of maximum thickness, the reduction in critical Mach number being then comparatively slight. On the other hand, the reduction of the distance from the leading edge of the point of maximum camber is accompanied by an increase in the choking mass flow at negative incidence.

Accordingly the invention proposes, more particularly in a compressor in which the fluid velocities are of a higher order, that the distance from the leading edge of the point of maximum thickness of the blades of successive rows be progressively reduced between the inlet and outlet blade rows. The invention further proposes, more particularly in a compressor in which the fluid velocities are of a lower order, that the distance from the leading edge of the point of maximum camber of the blades of successive rows be progressively reduced between the inlet and outlet blade rows. Preferably the positions of the points of maximum camber and maximum thickness are both varied throughout the blade rows of a compressor. It follows that a preferred basis of design of a compressor according to the invention is that of using for the inlet stages where, in general, high Mach numbers may be expected both at the designed and reduced loads blading which has its points of maximum thickness and maximum camber both relatively remote from the leading edge, so ensuring a falling mean incidence at falling Mach numbers, but with the point of maximum thickness advanced toward the leading edge as compared with that of maximum camber, so avoiding any substantial reduction at reduced loads of the critical Mach number. For the blading of the outlet stages, where the Mach number is low at the design load but where a tendency to choking is to be expected at reduced loads the points of both maximum thickness and maximum camber are advanced as compared with the inlet stage blading to such an extent that the mean incidence rises with falling Mach numbers. The characteristics of the inlet and outlet stage blading and also that of intermediate stages should, of course, be selected within the ambit of the foregoing requirements with regard also to the matching of the stages to ensure that all stages are similarly effective at various operating conditions.

It is generally desirable that the reduction in distance from the leading edge of the point of maximum thickness should occur more rapidly in the later than in the earlier blade stages of a compressor, but that the reduction in distance from the leading edge of the point of maximum camber should occur more rapidly in the earlier blade stages than in the later stages. Thus the distance for the blades of the middle row, or the mean distance (i. e. mean percentage of blade chord from the leading edge) of the blades of the middle two rows is in the case of the point of maximum thickness desirably greater, and in the case of the point of maximum camber less, than the mean of the corresponding values for the blades of the inlet and outlet rows.

In general, the requirements as discussed in the foregoing are satisfied to an acceptable degree by a compressor having inlet blading with its maximum thickness and camber respectively 40 and 50 per cent of the chord from the leading edge, and outlet blading having corresponding values of 30 and 40 per cent. The corresponding values for the blading of intermediate stages, having regard to the matching of the stages, should preferably be such that while the rate of advancement toward the blade leading edge of the point of maximum thickness increases with successive stages from inlet to outlet the compressor, the rate of advancement of the point of maximum camber decreases with successive stages. Thus, in a ten stage axial flow compressor according to the invention the preferred values are tabulated below:

Stage of compression from inlet to outlet	Percentage of chord from leading edge of—	
	Maximum thickness	Maximum camber
1	40	50
2	40	50
3	40	45
4	40	45
5	40	40
6	40	40
7	35	40
8	35	40
9	30	40
10	30	40

It will be understood that these values may be departed from in moderation without serious adverse effects. For simplicity, in the above case, the variation of the two parameters proceeds in successive pairs of stages, a rotor stage and the stator stage following it having similar values.

In order that the invention may be more readily understood some design considerations of the preferred embodiment of compressor above described will be briefly summarised with reference to the accompanying drawings. In particular, considerations concerning the first and last rotor blade stages will be compared and contrasted.

In the drawings:

Figure 1 represents a half elevation to one side of the axis of a multistage rotary axial flow compressor partly sectioned to show successive blade stages;

Figure 2 represents a transverse section through a blade of each of the successive stages of the compressor of Figure 1 at the mean blade diameter (II—II in Figure 1);

Figures 3(*a*) and (*b*) represent velocity triangles at design conditions corresponding respectively to the first and last stages of rotor blading of Figure 2;

Figures 4(*a*) and (*b*) represent estimated velocity triangles at part load conditions corresponding respectively to the first and last stages of rotor blading of Figure 2;

Figures 5(*a*) and (*b*) are graphical representations of the loss characteristics at various conditions of the first and last stages of rotor blading of Figure 2.

The multi-stage axial flow compressor of Figure 1 comprises a rotor **11** mounted for rotation with respect to stator structure **12** with which it defines an axial flow passage of annular cross-section for the fluid under compression. The rotor carries five rotor blade stages **1, 3, 5, 7** and **9** extending radially into the flow passage which are arranged in interdigital relationship with five stator blade stages **2, 4, 6, 8** and **10** carried on the stator structure **12.**

It is supposed that the compressor is to have a designed pressure ratio of 5:1, that the fluid outlet angle from any rotor (odd numbered) stage is to be 15°, that the mean diameter at each stage of the blade annulus is constant and that the fluid temperature rise is the same in each stage.

In Figure 2, for convenience of drawing, the blade sections at mean diameter of the ten successive stages are divided into three groups. Each blade section is constructed about a camber line indicated at B in the section of the first stage blading. The straight line passing through the points of intersection of the camber line and

the profile of the section at its leading and trailing edges, CD in the first stage, in the chord line the distance CD being the chord of the section. The positions of the points of maximum camber of the camber line relative to the chord line and maximum thickness of the profile of the section are indicated in each section as percentages of the chord and agree in each case with the values tabulated above. The inlet angle of a blade section indicated as α_1 in the first stage is the angle between the tangent to the camber line at the leading edge and the normal to the plane of rotation of the rotating blades. The outlet angle is the corresponding angle at the trailing edge of the blade section.

Figures 3(*a*) and (*b*) show the velocity triangles at the means diameter corresponding respectively to first and last rotor stages **1** and **9**. It will be noted that these triangles are assumed to be identical for each stage at the designed operating conditions, and thus that the fluid inlet angle α_1 and velocity V_1 are equal at 45° and 707 feet per second respectively. The Mach numbers, however, are substantially different due to the rising temperatures of the fluid as it is compressed, being 0.635 and 0.445 respectively at first and last stages, corresponding to fluid temperatures of 288° K. and 480° K. U represents the blade speed and V_a the axial component of velocity of the fluid through the stage.

Figures 4(*a*) and (*b*) show in full line the estimated velocity triangles of the two stages at the mean blade diameter when the rotational or blade speed U is reduced to three quarters of the design value. V_a varies in first and last stages disproportionately as compared with the reduced value of U, and in opposite senses, i. e. low at the inlet stage (Figure 4(*a*)) giving a higher inlet angle α_1 of 50° and high at the outlet stage (Figure 4(*b*)) giving a lower inlet angle α_1 of 24°.

At the mean diameter at the inlet stage the fluid velocity V_1 is low, namely 506 feet per second and, the fluid temperature being the same as at the design condition, the Mach number is comparatively low, i. e. 0.45. At diameters of the inlet stage blading radially outward of the mean diameter, however, the fluid velocity V_1, its axial component V_a and the blade speed U are higher than at the mean diameter, as represented by the broken lines in Figure 4(*a*) and the Mach numbers are comparatively high. It will be noted that the inlet angle of the fluid in the broken line velocity triangle compares with that of the full lines velocity triangle being high relative to the design value of Figure 3(*a*). At the mean diameter at the outlet stage (Figure 4(*b*)) the fluid inlet velocity is high, i. e. 733 feet per second, due to the high axial component V_a, but the temperature of the fluid is still high (although not as high as at the design condition) so tending to reduce the Mach number, which, in consequence, is not excessive, being 0.580. There is no substantial variation at other radii from the fluid velocity conditions of the mean radius in the outlet stage.

Figures 5(*a*) and (*b*) show graphs of the loss characteristics of the blading of stages **1** and **9** respectively. In the graphs the percentage loss in total head of fluid passing through the blading at various values of Mach number Mn (ordinates) are plotted against fluid inlet angle α_1 (abscissae).

It is seen from Figure 5(*a*) that for the inlet stage blading the range of fluid inlet angles corresponding to a loss of, say, 5 per cent or less has a progressively higher mean value at higher Mach numbers, so that at the higher Mach numbers, i. e. 0.6 to 0.7, at which the range is very limited the mean value is relatively high at about 45° as is required by the velocity triangle of Figure 3(*a*). At the same time the inlet angle range at lower Mach numbers permits the satisfaction of the various conditions of the full and dotted line velocity triangles of Figure 4(*a*). It will be seen that there is no appreciable variation in the minimum value of the losses at any Mach number within the working range and thus the critical Mach number at which the minimum value of the losses become excessive (say over 5 per cent) is considerably above the highest Mach number approached in the compressor inlet stage in opeartion.

From Figure 5(*b*) it is seen that for the outlet stage blading the range of fluid inlet angle corresponding to a loss of 5 per cent or less has a progressively lower mean value at higher Mach numbers so that the various requirements of the velocity triangles of Figures 3(*b*) and 4(*b*) are satisfied. It will be noted that the critical Mach

number is much lower in this case than in Figure 5(*a*), the minimum losses rising above 5 per cent at a Mach number 0.6 and 0.7. However, as the working range of Mach numbers in the outlet stage blading of the compressor lies below that of the inlet stage, the outlet stage critical Mach number is not approached in operation.

It will be noted that, in the embodiment of the invention above described with reference to the drawings, the total variation throughout the compressor stages of the percentage chord from the leading edge of the point of maximum camber and thickness, in each case 10%, is accommodated similarly in each case in two relatively large transitions of 5% each. This is not particularly desirable from a theoretical viewpoint, but is adopted merely because it is extremely difficult in present practice to manufacture blades sufficiently accurately to warrant designing the compressor with a smaller transition between stages. However, with improved blade manufacturing techniques, it may be feasible to reduce such transitions to the order of 2½% from stage to stage.

I claim:

1. A rotary bladed axial flow elastic fluid compressor having several rows of blades arranged, and through which the fluid flows, in axial succession, the blades of any one of said rows being similar and of cambered aerofoil form, wherein the blades of the first of said rows traversed by the fluid have a transverse section in the region of their mid-length of which the points of maximum camber and maximum thickness are disposed at percentages of the chord of the section from its leading edge which are both greater than the corresponding values for the points of maximum camber and maximum thickness respectively of a transverse section in the region of their mid-length of the blades of the last of said rows traversed by the fluid.

2. A rotary bladed axial flow elastic fluid compressor having several rows of blades arranged and through which the fluid flows, in axial succession, the blades of any one of said rows being similar and of cambered aerofoil form, wherein the blades of the first of said rows traversed by the fluid have a transverse section in the region of their mid-length of which the point of maximum camber is disposed at a percentage of the chord of the section from its leading edge which is greater than the corresponding value for the point of maximum camber of a transverse section in the region of their mid-length of the blades of the last of said rows traversed by the fluid.

3. A rotary bladed compressor acording to claim 2, wherein blades of any one of said rows intermediate between said first and last rows have a transverse section in the region of their mid-length of which the point of maximum camber is disposed at a percentage of the chord of the section from its leading edge whose value is neither greater than the corresponding value for the blades of the row immediately preceding said intermediate row in the direction of fluid flow nor less than the corresponding value for the blades of the row immediately succeeding said intermediate row.

4. A rotary bladed compressor according to claim 2, wherein the blades of one of said rows situated substantially mid-way between said first and last rows have a transverse section in the region of their mid-length of which the point of maximum camber is disposed at a percentage of the chord of the section from its leading edge whose value is less than the arithmetic mean of the respective corresponding values for the blades of said first and last rows.

5. A rotary bladed axial flow compressor having several rows of blades arranged and through which the fluid flows, in axial succession the blades of any one of said rows being similar and of cambered aerofoil form, wherein the blades of the first of said rows traversed by the fluid have a transverse section in the region of their mid-length of which the point of maximum thickness is disposed at a percentage of the chord of the section from its leading edge which is greater than the corresponding value for the point of maximum thickness of a transverse section in the region of their mid-length of the blades of the last of said rows traversed by the fluid.

6. A rotary bladed compressor according to claim 5, wherein the blades of any one of said rows intermediate between said first and last rows have a transverse section in the region of their mid-length of which the point of maximum thickness is disposed at a percentage of the chord of the section from its leading edge whose value

is neither greater than the corresponding value for the blades of the row immediately preceding said intermediate row in the direction of fluid flow nor less than the corresponding value for the blades of the row immediately succeeding said intermediate row.

7. A rotary bladed compressor according to claim 5, wherein the blades of one of said rows situated substantially mid-way between said first and last rows have a transverse section in the region of their mid-length of which the point of maximum thickness is disposed at a percentage of the chord of the section from its leading edge whose value is greater than the arithmetic mean of the respective corresponding values for the blades of said first and last rows.

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