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.

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 rotation speed of each in turn, which, in each case, the approach angle of the fluid is dependent) may be adjusted to a 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 desirable to be avoided. On the other hand, the convenience and simplicity of this more conventional arrangement of multi-stage compressor is required to some extent by the fact that there is an inherent tendency in such a compressor towards the production 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 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. This is in contrast to a compressor in which the incidence range of the blades at the inlet and outlet stages may be respectively positively and negatively exceeded. The ensuing phenomenon, indicative of a rapid loss of compressor performance, is known respectively as stalling (or surging if the instability is great) and choking.

Hereafter 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 de-
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 blading of intermediate stages in between being mean incidecnes. 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 the inlet stage in 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 toward the remaining portion 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 the leading edge 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 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 of the blades at the outlet stage, whereas the Mach number is low at the design load but where a tendency to choking is to be expected at reduced loads the point of 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 stage and also that of the 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 will be understood that these values may be departed from in moderation without serious adverse effects. For simplicity, the characteristic 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 shown at 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.

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, 2, 3, 4, 5, 7 and 9 extending radially into the flow stage and 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 compression 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 every case with the values tabulated above. The inlet angle of a blade section indicated as $\alpha_1$ in the first stage is the angle between the tangent to the camber section at the leading edge and the normal to the plane of rotation of the horizontal blade. 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 $\alpha_1$ and velocity $V_1$ are equal at 45° and 707 feet 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 450° 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 speed and at last stages proportionately as compared with the reduced value of $U$, and in opposite senses, i.e. low at the mean diameter of Figure 4(a) giving a higher inlet angle $\alpha_1$ at 50° and high at the outlet stage (Figure 4(b)) giving a lower inlet angle $\alpha_1$ of 24°.

If the mean diameter at the inlet stage the fluid velocity $V_1$ is low, namely 506 feet per second and, and the fluid temperature being the same as at the design condition, the Mach number is comparatively low, i.e. 0.45. At the center of the inlet stage blading radially outward of the mean diameter of Figure 4(a) giving a higher inlet angle $\alpha_1$ at 50° and high at the outlet stage (Figure 4(b)) giving a lower inlet angle $\alpha_1$ of 24°.

The Mach number $M_n$ (ordinates) are plotted against fluid inlet angle $\alpha_1$ (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 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 same variations in the full and dotted line velocity triangles of Figure 4(a). It will be seen that there is no appreciable variation in the maximum value of the losses at any Mach number within the working range at a percentage 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 operation.

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 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 or three 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 is feasible to reduce such transitions to the order of 1% or 2% 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 point of maximum camber and maximum thickness are disposed at 25% of the chord of the section from its leading edge which is 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 and maximum thickness are disposed at 25% of the chord of the section from its leading edge which is 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.

3. A rotary bladed compressor according to claim 2, 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 camber is disposed at a percentage of the chord of the section from its leading edge whose value is less than the corresponding value for the blades of said intermediate rows.

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 corresponding value for the blades of said intermediate row.

5. A rotary bladed 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 whose value is less than the corresponding value for the blades of said intermediate rows.
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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