PUBLISHED	G. JENDRASSIK	Serial No.
APRIL 27, 1943.	APPARATUS FOR GAS TURBINES	217,505
BY A. P. C.	Filed July 5, 1938	2 Sheets-Sheet 1

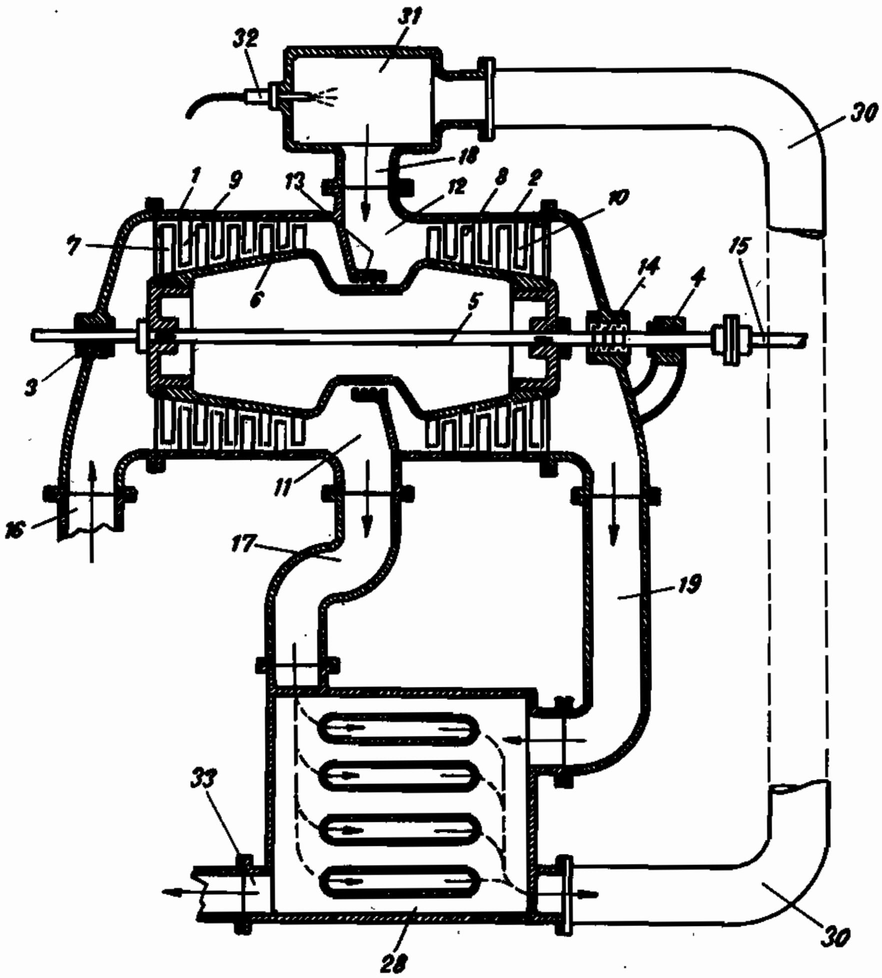

Fig. 1.

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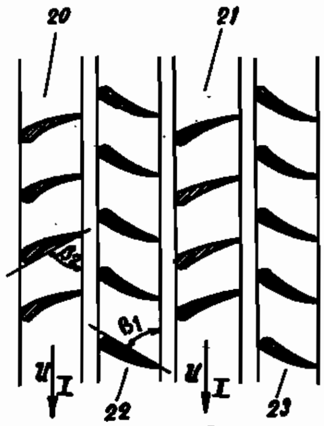

Fig. 2.

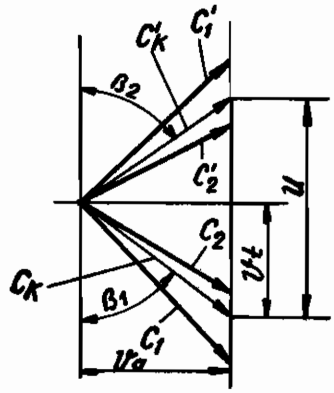

Fig. 3.

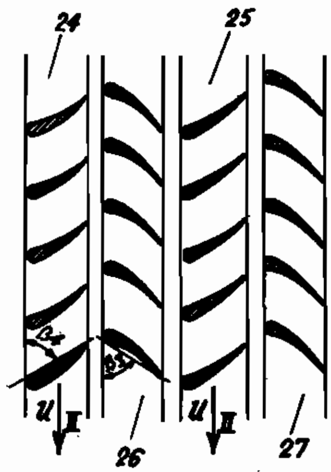

Fig. 4.

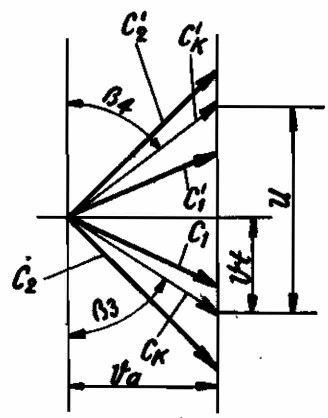

Fig. 5.

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ALIEN PROPERTY CUSTODIAN

APPARATUS FOR GAS TURBINES

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Application filed July 5, 1938

In gas turbines, where the working medium is being compressed to a pressure higher than the admission pressure of the compressor, the efficiency of the compressor plays a very essential and in fact decisive part, seeing that the work supplied by the equipment really consists in the difference as between the work transmitted by the turbine to the shaft and the work absorbed from the shaft by the compressor. Particularly suitable for enabling high efficiency to be achieved are the compressors of the type in which the mean diameter of any stationary or rotating blade ring is at least approximately equal to the average of the mean diameters of the two adjacent blade rings and in which the blades have a cross-section resembling the profiles of aeroplane wings as usual in aerodynamical practice, or a thin sheet-like cross-section. Compressors of such a type are operating at a high efficiency as long as the velocity of the working medium relatively to the blades does not approximate the velocity of the propagation of sound vibrations in the working medium. For this reason it is advisable to fix the upper limit of the velocity of the working medium relatively to the blades at 0.6–0.7 times the figure of the velocity of propagation of sound vibrations. In view of the fact that the velocity of propagation of sound vibrations in air amounts at a temperature of 15° C to about 330 m/sec, the relative velocity in question should preferably not exceed 230 m/sec. If hereto it is added that, on the other hand, in view of obtaining a high efficiency the meridian velocity of the medium (its velocity of throughflow in the axial section) should preferably be assumed at 0.3 to 0.8 times the peripheral velocity, there will result the disadvantageous situation that if the relative velocity of 230 m/sec mentioned is adhered to, it would not be possible for the peripheral speed to exceed 185 m/sec. On the other hand in order that such compressors whilst keeping their weight and dimensions low, should be able to supply an as large output of work as possible, i. e. a sufficiently high pressure, it is essential that the peripheral speed of the blade rings should be as high as possible and should approximate the figure which is still permissible from the point of view of mechanical resistance in the blades or in the rotor carrying the blades. For this reason the equipment will in case of the low peripheral speed mentioned above become, for many cases of employment, too heavy and too expensive.

Owing to the limitation prescribed regarding the relative velocity, a disadvantageous position will result also owing to the fact as in the turbine it is in any case possible to permit a larger relative velocity than in the compressor and thus if the compressor is direct-coupled with the turbine it would only be possible for it to be of smaller diameter than the turbine. In the case of such a type of design, however, the compressor will not be able to draw-in and supply the necessary quantity of gas. Accordingly, in this case, if in accordance with the proposals as made up to now it is desired to utilise in a full extent in the turbine as well as in the compressor the peripheral velocities of permissible magnitude, it would be necessary for the compressor and for the turbine to run at different members of revolutions per minute, which would necessitate an expensive and complicated transmission gear.

The apparatus according to the invention eliminates these drawbacks in such a manner, that it diminishes the relative velocity between the blades and the working medium in the compressor, by setting the working medium into rotation, in the sense of rotation of the compressor rotor. The working medium rotates in the compressor in the direction of rotation at least on one diameter of the blades at a velocity amounting on the average approximately to one-half the peripheral velocity of the rotor, it is, however, also possible for the velocity of rotation, to deviate therefrom so as to possess a higher or a lower figure than the one defined above. If the average peripheral velocity of the working medium is sufficiently high relatively to the peripheral velocity of the rotor e. g. if it amounts to at least ¼ of the latter, a substantial diminution of the relative velocity will already be obtained. On the other hand if the velocity of rotation of the medium is greater than ¾ of the peripheral velocity of the rotor, it is relatively to the stationary blades that the relative velocity will be high. For this reason it is necessary that the average velocity of rotation, at least on one diameter of the blades, should remain between ¼ and ¾ of the peripheral velocity of the rotor. Thus a much higher peripheral velocity than the one permissible up to now can be permitted in the rotor, in consequence whereof, on the one hand, the efficiency of the compressor will not deteriorate, whilst on the other hand its capacity of performance will be increased or with a given capacity of performance it will be possible to reduce its dimensions and direct coupling of the compressor with the turbine will become possible. The average rotation at a considerable velocity of the working medium in the compressor is obtained by the suitable adjustment of the blading.

In case the average rotation of the working medium is equal to one-half the peripheral velocity of the rotor, the relative velocities relatively to the rotating and to the stationary blades will be approximately equal.

In case the turbine likewise is fitted with bladings possessing a cross-section resembling the aeroplane wing profiles usually employed in aerodynamical practice, and in case the mean diameter of any stationary or rotating blade ring of the turbine is at least approximately equal to the average of the mean diameters of the two adjacent blade rings, it is possible to obtain a more advantageous efficiency in the turbine also, if the relative velocity as between the working medium and the blades is kept below the figure of the velocity of the propagation of sound vibrations. The velocity of the propagation of sound vibrations in the turbine is in general higher than in the compressor, owing to the fact that the temperature of the working medium is also higher. On the other hand the higher temperature of the working medium will increase the volume of the latter also and thus it will usually be necessary to permit higher throughflow velocities in the turbine, whereby the relative velocity as between the working medium and the blades will be increased. For this reason it will, in view of not approximating the velocity of propagation of sound vibrations too closely, be advantageous to set the working medium into substantial rotation in the direction of the peripheral velocity of the rotor in the turbine also, preferably in such a manner that the working medium should at least on one diameter of the blades circulate at approximately, on the average, one-half the peripheral speed of the blades, which aim can likewise be assured by the suitable adjustment of the blading.

In order to enable the invention to be more readily understood, Fig. 1 shows the diagrammatical section of a compressor and turbine direct-coupled or built integral. Fig. 2 is a development into a plane of a section taken through the blades of the compressor. Fig. 3 represents the velocity triangles relating to the stationary and rotating blades of the compressor, Fig. 4 is a development into a plane of a section taken through the blades of the turbine, whilst Fig. 5 shows the velocity triangles relating to the stationary and rotating blades of the turbine.

On Fig. 1 the rotor 6 keyed on the shaft 5 journalled in the bearings 3 and 4 is arranged in the compressor or turbine casing 1; this rotor carries on the one hand the rotating compressor blades 7 and on the other hand the rotating turbine blades 8. It is into the compressor casing 1 that the stationary compressor blades 8 are mounted whilst the stationary turbine blades 10 are mounted into the turbine casing 2. In the embodiment shown by way of example the high pressure space 11 of the compressor is closed off from the admission space 12 of the turbine by the labyrinth packing 13. On the discharge end of the turbine the shaft is rendered tight by means of the labyrinth packing 14. The work obtained can be taken off on the shaft 15. The method of operation of this apparatus is the following:

The working medium enters the compressor through the inlet duct 16 and leaves the compression space 11 in a compressed condition through the duct 17. Heat is introduced by means of the combustion of fuel in a manner and with the aid of apparatus not shown on the drawing into the working medium discharged through the duct 17, following which the working medium thus heated is led through the inlet duct 18 into the turbine in which it expands, performs work and finally leaves the turbine through the duct 19.

It is also possible—according to known proposals—to effect, before the introduction of the heat of the fuel, the pre-heating, in a suitable heat exchange device, by means of the heat of the spent gases leaving the turbine, of the compressed working medium discharged through the duct 17. According to other proposals it is also possible to introduce the heat of the fuel during the passage of the working medium through the turbine, or, entirely or partly even after the working medium has performed its throughflow through the turbine, and in this latter case to make provision for the heat generated being transferred to the fresh quantity of working medium by means of a heat exchange device.

On Fig. 2 the moving compressor blade rings 20 and 21 are rotating with the peripheral velocity u in the direction of the arrow I, whereas the stationary blade rings 22 and 23 are immovable. The base line of the moving blade profiles (which is approximately identical with the aerodynamically neutral direction, i. e. with the direction defined by the fact that a current of air attacking in this direction will not cause any force of buoyancy on the blade) forms with the peripheral direction, at the point of leaving the blade ring, the angle β_2, whilst the base line of the stationary blades is, at the point of leaving the blade ring likewise, forming with the peripheral direction the angle β_1.

In the velocity diagrams shown on Fig. 3, v_a denotes the throughflow (meridian) velocity of the working medium when flowing through the compressor, u denotes the peripheral velocity of the rotor, whilst v_t denotes the mean velocity in the peripheral direction of the working medium. Into any stationary blade ring the working medium enters with the absolute velocity c_1 and leaves it with the absolute velocity c_2. The mean value c_k of the two velocities represents the mean absolute velocity of the working medium. The component in the peripheral direction of this last named velocity is v_t, the average figure of which, taken at least on one diameter of the blade, is preferably approximately equal to one half the peripheral velocity, but may also be greater or smaller than this figure (what is important from the point of view of the invention being that the average peripheral velocity of the medium should be sufficiently high relatively to the peripheral velocity of the rotor). The relative inlet velocity relatively to the rotating blades is obtained by adding-up the outlet velocity relatively to the stationary blades and the velocity u. The velocity thus obtained is c_1'; the relative outlet velocity is c_2', whilst the relative mean velocity is c_k'.

The aerodynamically neutral direction of the blades is, as has been mentioned, the direction defined by the fact that in case of a relative flow in this direction the blade force perpendicular to the direction of the flow is zero, and accordingly this direction is approximately identical with the direction of the tangent on the compressed (concave) side of the blade profile, i. e. with the direction of the base line of the blade profile. The angle formed by the direction defined in this manner and by the peripheral direction is according to Fig. 3 also in the case of the stationary blades β_1, and in the case of rotary blades β_2.

In order to ensure that the relative velocity as between the working medium and the blades should be as small as possible, the mean velocity of rotation of the medium should preferably be made equal to one-half of the peripheral velocity. This condition is approximately satisfied in that case when the angles formed by the peripheral direction with the base lines of the profiles of the stationary and of the rotary blades are, at least on one diameter of the blades, approximately mutually identical.

In the case of the turbine blades shown on Fig. 4 the moving blade rings **24** and **25** are rotating in the direction of the arrow II with the peripheral velocity u and the base lines of their blade profiles (representing approximately the aerodynamically neutral direction) are at the point of inlet into the blade ring forming with the peripheral direction the angle β_4, whilst the base lines of the blade profiles of the stationary blade rings **26** and **27** are at the point of inlet into the blade ring likewise forming with the peripheral direction the angle β_3.

In the velocity triangles shown on Fig. 5 the denotations are, suitably interpreted, identical with the denotations shown on Fig. 3. With a given peripheral and given meridian velocity the lowest relative velocity between the blades and the working medium is obtained in case the average velocity of rotation of the working medium is, at least on one diameter of the blades, approximately equal to one-half the peripheral velocity of the rotor. This can be assured if the blade angles are adjusted in such a manner that the base lines (i. e. the aerodynamically neutral directions) of the profiles of the stationary and of the moving blades should, at least on one diameter of each of these two kinds of blade rings, form approximately equal angles with the peripheral direction.

In view of the fact the temperature of the working medium varies in the compressor as well as in the turbine, it is not necessary that the condition of the lowest relative velocity should be fulfilled in each stage. A very large advantage is obtained already in case a substantial average rotation is given to the working medium in the direction of the peripheral velocity of the rotor. This can be achieved in the case of the compressor, as well as in the case of the turbine, by deflecting the base line of the stationary blades from the meridian plane, in the direction—viewed in the direction of the flow—of the peripheral velocity of the rotor.

The constructional condition for the average velocity of rotation of the medium being situated between ¼ and ¾ of the peripheral velocity of the rotor consists in that the value of the fraction composed of the tangent of the angle β_1 formed by the base line of the stationary blades with the peripheral direction as of a numerator and of the tangent of the angle β_2 formed by the base line of the rotating blades with the peripheral direction, as denominator, should at least on one blade diameter be situated between ⅓ and 3.

Instead of the form of construction shown on the drawing and described in the specification by way of example it is possible to employ a great many other kinds of constructional forms also, whereby the substance of the apparatus according to the invention and the range of protection of the invention are not modified in any way. Thus the compressor or the turbine may, for instance, also be arranged for radial throughflow, or throughflow may also take place along a cone surface, or in general around a rotational surface. From an aerodynamical point of view it is advantageous if the average velocity of rotation of the working medium stands in inverse proportion to the distance from the axis of rotation. This can be ensured by the suitable torsional deflection of the blades. The blades may also, as mentioned in the introductory part of this specification, be of thin sheet-like shape, which shape can be considered to represent the "limit" case of aerodynamical profiles.

Finally, it is worth mentioning that the invention modified in a logically corresponding manner, can also be employed on gas turbine equipments, in which the compressor or the gas turbine proper or both of these two main parts now mentioned are designed in such a manner that in addition to the rotor, the so-called "stator" is also made rotatable in a sense of rotation opposite to that of the rotor. If the peripheral velocity in any blade ring of the "stator" is denoted at a certain blade diameter by u_1, whilst in the adjacent blade ring of the rotor cooperating with the said blade ring of the "stator" the peripheral velocity is denoted by u_2, u_1 being $\leqq u_2$, it is the difference of velocities u_2-u_1 which will represent the figure of velocity which is decisive regarding the peripheral velocity in the direction of rotation of the rotor of the working medium flowing in the compressor, so that it will not be possible in the cases according to the invention for the peripheral velocity of the working medium to be lower than ¼ of this velocity or to be greater than ¾ of this velocity. Employing the same denotations it is possible to indicate regarding the angles β_1 and β_2 of the stationary and rotating blades of the compressor and the turbine the following mathematical relation according to the invention:

$$\frac{1-\frac{u_1}{u_2}}{4} < \frac{1-\frac{u_1}{u_2}\cdot\frac{tg\beta_1}{tg\beta_2}}{1+\frac{tg\beta_1}{tg\beta_2}} < \frac{3}{4}\left(1-\frac{u_1}{u_2}\right)$$

which relation will in case of $u_1=0$ (i.e. in case of the compressor or turbine casing being stationary) become converted into the condition:

$$\frac{1}{3} < \frac{tg\beta_1}{tg\beta_2} < 3$$

already referred to above.

In case of blade rings rotating in mutually opposite directions the arrangement will be practically the most advantageous for the compressor if $u_1=u_2$, as in such a case the working medium will only rotate relatively to the rotor and to the casing of the compressor, but will not rotate actually.

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