

Feb. 15, 1949.

D. MacL. SMITH ET AL  
MULTISTAGE COMPRESSOR

2,461,931

Filed April 6, 1944

3 Sheets-Sheet 1

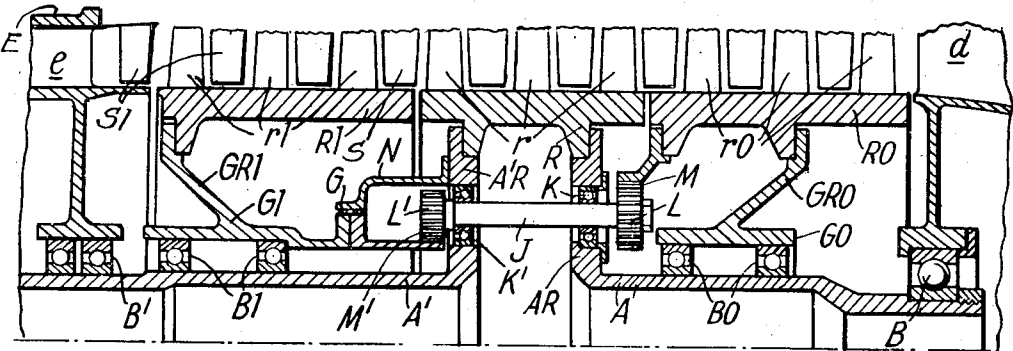


FIG. 1.

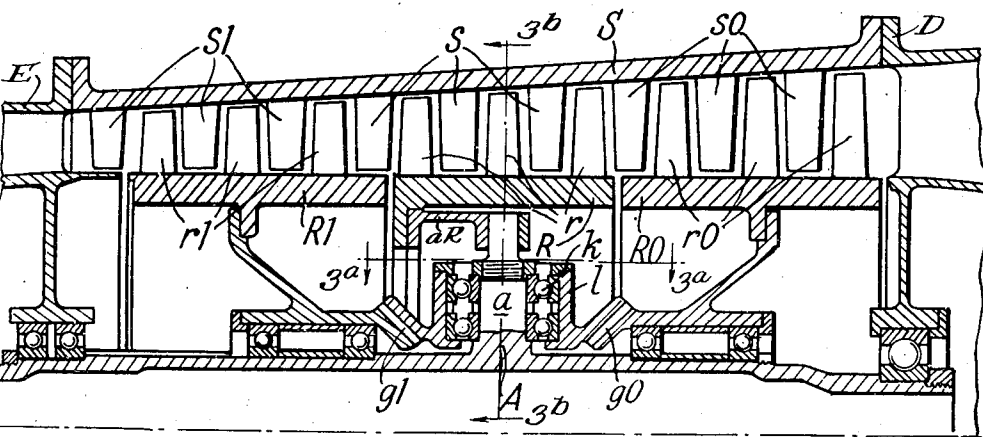


FIG. 3.

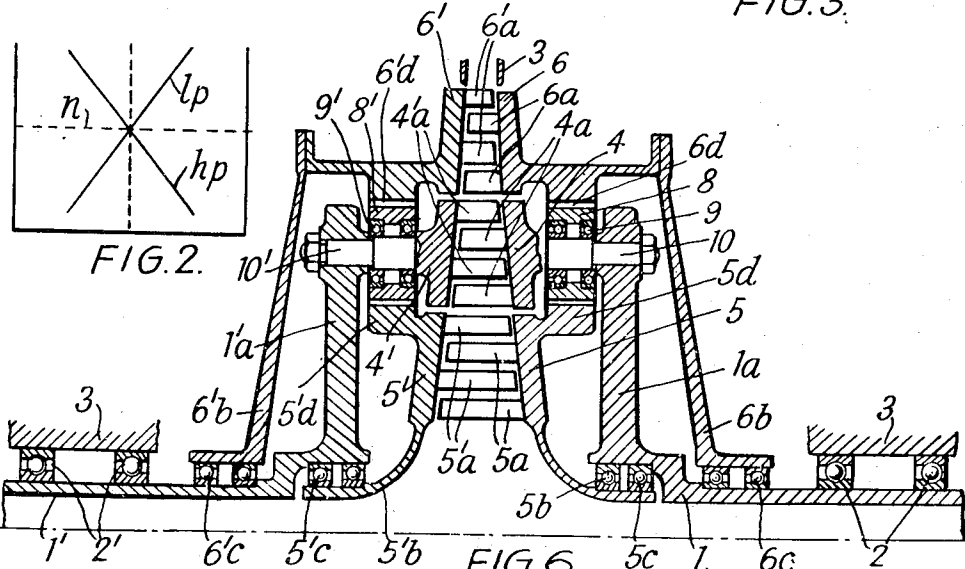


FIG. 6.

INVENTORS  
David Macdonald Smith  
Charles Graham Lloyd  
BY Lloyd Hall Sutton  
ATTORNEY

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3 Sheets-Sheet 2

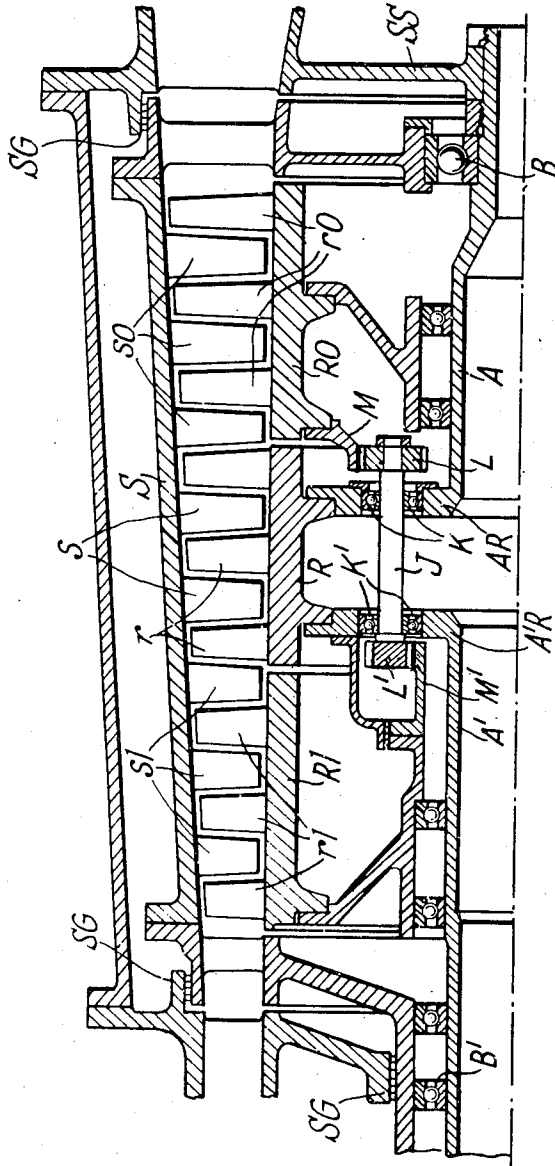


FIG. 4

INVENTORS  
David MacL. Smith  
Charles Graham Lloyd  
BY Lloyd Hall Sutton  
ATTORNEY

Feb. 15, 1949.

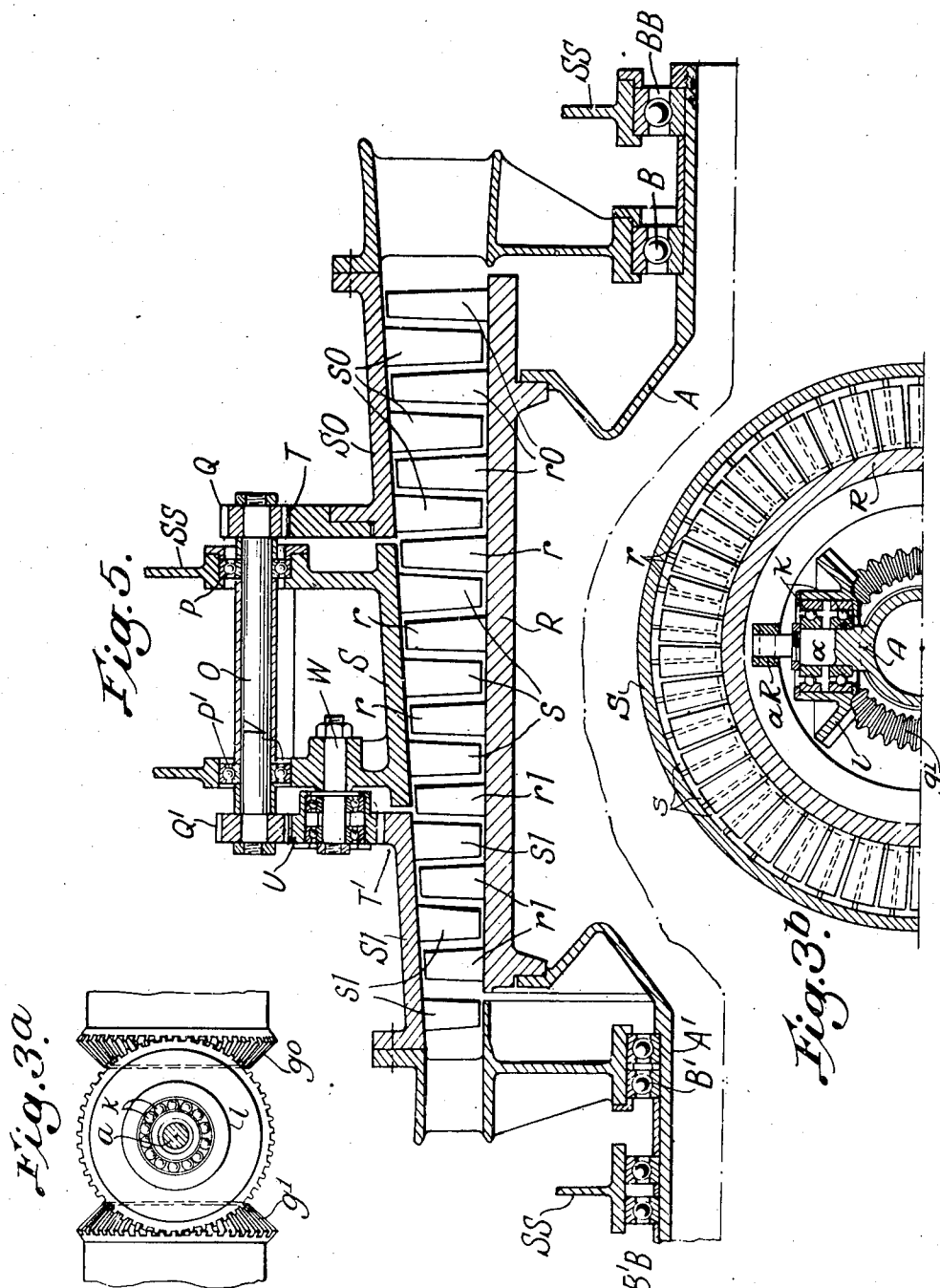
D. MacL. SMITH ET AL

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MULTISTAGE COMPRESSOR

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3 Sheets-Sheet 3



INVENTORS  
David MacL. Smith  
Charles Graham Lloyd  
BY Lloyd Hall Sutton  
ATTORNEY

## UNITED STATES PATENT OFFICE

2,461,931

## MULTISTAGE COMPRESSOR

David MacLeish Smith, Stretford, Manchester,  
and Charles Graham Lloyd, Baguley, England,  
assignors to Metropolitan-Vickers Electrical  
Company Limited, London, England, a com-  
pany of Great Britain

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Patent expires January 4, 1963

8 Claims. (Cl. 230-114)

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This invention relates to multi-stage turbo-compressors. In order that such type of compressor has good efficiency or performance characteristics it must be designed for a predetermined ratio of absolute discharge pressure to absolute inlet pressure: the blading throughout the several stages of compression must employ the correct inlet and outlet angles for a definite pressure difference at each compression stage, with a corresponding resultant velocity, and these compressors must be operated at constant speed, or at the designed relative speeds of the co-operating bladed elements when the compressor is of the contrarotational type. Changes in the pressure ratio may be caused by speed variations such as are likely to occur in the case of compressors employed in internal combustion turbine plant (namely due to variation of combusted fuel for variable load) and, in the case of compressors employed in such or other plant, to a lesser degree by variations in the temperature of the inlet air. Where the pressure ratio differs considerably from that for which the compressor is designed (hereinafter called the design ratio) the change in the specific volume of the air in the compression stages is such that velocity ratios of air to blade become unsuitable in at least some of the compression stages, causing the angle of incidence of air flow to the blades to be unsuitable for the inlet edge angle of the blades.

Where the ratio of axial air velocity to blade velocity for a compression stage is too high, the pressure rise and efficiency in that stage are low, and in extreme cases an actual drop in pressure may take place in the stage, particularly at the high-pressure end of the compressor. When the ratio of the axial air velocity to blade velocity in any stage is too low the compressor tends to stall.

At values of pressure ratio lower than that of the design ratio for the compressor, the characteristics may be improved by decreasing the relative blade velocity for the low-pressure blades and by increasing that for the high-pressure blades, and conversely, for pressure ratios higher than that of the design ratio, the characteristics may be improved by increasing the relative blade velocity for the low-pressure blades and decreasing that for the high-pressure blades. These effects are achieved according to the present invention by associating blades of a low-pressure blade group and blades of a high-pressure blade group with one another and with blades of an intermediate-pressure blade group

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through the intermediary of differential torque balancing gear.

Broadly, two methods may be used in carrying out the invention. According to the first method, the differentially-connected low-pressure and high-pressure blades may be on the rotor, or on one of two contra-rotational rotors for a compressor of the contra-rotational type, in which case the intermediate-pressure blades with which they are associated will run at the speed of the driving turbine or other motor, and at pressure ratios different from that of the design ratio for the compressor the differentially-connected blades will be automatically rotated in opposite directions relatively to the intermediate-pressure blades with which they are associated, by reason of the unbalance in the torques acting respectively on the low-pressure and high-pressure blade groups; thus, the relative blade velocity for co-operating blades of one group, that is, low-pressure or high-pressure, will be increased with respect to that of the intermediate-pressure group while the relative blade velocity for the other group will be decreased with respect to that of the intermediate-pressure group. The direction of the respective rotations of the interconnected blades of the blade groups in question is automatically determined according to whether the pressure ratio is higher or lower than that of the design ratio. According to the second method, the differentially-connected blades are on a stator in which case the associated intermediate-pressure blades remain stationary while at pressure ratios different from that of the design ratio the differentially-connected blades will be automatically rotated slowly in opposite directions by reason of the unbalance in the torques on the respective blade groups; the blades which are thus caused to run in the opposite direction to that of the main rotor will act to increase the relative blade velocity of the appertaining group while the blades caused to run in the same direction as that of the main rotor will act to decrease the relative blade velocity of the appertaining group.

The design may be such that, when the compressor is operating near the rated condition, there is little differential motion between the interconnected blades of the low-pressure and high-pressure groups so that, with the differential gear in the form of a train of meshing gear wheels, the relative tooth speed of the gearing is low and the gear losses small.

The differential motion of the interconnected blades being produced automatically by the

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change in torque at the respective low-pressure and high-pressure ends of the compressor, the relative speeds at which the rotating blades will run for maintaining a particular designed ratio between the respective torques, will be determined by the choice of the gear ratio.

The torque balancing gear ensures that the loaded mean speed of the two blade groups driven through the differential gear is equal to the speed of the positively driven member on the one hand or to the relative blade speed of the intermediate-pressure blade group on the other hand.

The invention may be applied to compressors either of the axial flow type or of the radial flow type, and moreover may be applied to compressors of the uni-rotational type or of the contra-rotational type.

In order that the invention may be fully understood and carried into effect, various embodiments thereof will now be described, by way of example, with reference to the accompanying diagrammatic drawings, in which:

Fig. 1 is a partly broken axial sectional view of the upper half of one arrangement of uni-rotational axial flow compressor according to the invention, and

Fig. 2 is an explanatory diagram illustrating its mode of operation;

Fig. 3 is a partly broken axial sectional view of the upper half of a uni-rotational axial flow compressor having an alternative form of differential torque balancing gear according to the invention; Figs. 3a and 3b are detail transverse sections on the lines 3a—3a and 3b—3b respectively in Fig. 3.

Fig. 4 is a partly broken axial section of the upper half of another arrangement of axial flow compressor according to the invention; and

Fig. 5 is a partly broken axial section of the upper half of a still further arrangement of axial flow compressor according to the invention, while

Fig. 6 is a partly broken axial section of the upper half of a radial flow type compressor arranged in accordance with the invention and in which the mutually-reacting bladed elements are contra-rotational.

In the drawings, like parts in the several figures are indicated by like reference characters.

In the compressor illustrated in Fig. 1, three coaxial bladed rotors, namely, rotor cylinder R equipped with rows of blades  $r$ , and rotor cylinders R0 and R1 equipped with blade rows  $r0$  and  $r1$  respectively, disposed one on either side of the rotor R, are arranged for mutual reaction with a coaxial bladed stator comprising external casing S equipped with blade rows  $s0$  for cooperating with the blade rows  $r0$ , blade rows  $s$  for cooperating with the rotor blade rows  $r$  and blade rows  $s1$  for co-operating with the rotor blade rows  $r1$ . Said mutually-reacting bladed elements function to increase the pressure of working fluid, such as air, admitted to the compressor through diffuser means formed by one or more passages  $d$  in casing D fast with the stator S at the nose or inlet end of the compressor and discharged through one or more passages  $e$  in casing E fast with the rear end of the stator S; thus, the blades  $r0$  and  $s0$  jointly constitute a low-pressure, or inlet, blade group, blades  $r$  and  $s$  an intermediate-pressure blade group, and blades  $r1$  and  $s1$  a high-pressure, or outlet, blade group.

The rotor cylinder R for the intermediate-pressure blade group is attached at one end to a sleeve shaft A through a disc or radial flange

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AR, and at the other end to a coaxial sleeve shaft A' through disc or flange A'R. Said shafts A and A', thus mechanically connected together through rotor R, jointly constitute the main rotor shaft of the compressor and are journaled in bearings B and B' respectively mounted the one, B, in the nose casing D and the other, B', in outlet casing E; in use, said rotor shaft A, A' will be rotated by any suitable driving means, such for example as an internal combustion turbine employed in conjunction with the compressor for propulsion or other purposes, which driving means, though not shown in the drawing, may be attached to said shaft say at the discharge end of the compressor.

The rotor cylinder R0 carrying the rotor blades  $r0$  for the low-pressure blade group is attached to shaft G0 which is coaxial with the rotor shaft A and is mounted thereon through bearing B0, so that the low-pressure rotor blades  $r0$  are rotatable relatively to the intermediate-pressure rotor blades  $r$ . Likewise, the rotor cylinder R1 carrying rotor blades  $r1$  for the high-pressure blade group is attached through one or more webs GR1 to sleeve shaft G1 mounted, through bearings B1, on the rotor shaft A', so that the high-pressure rotor blades  $r1$  are rotatable relatively to the intermediate-pressure rotor blades  $r$ .

A layshaft J which is journaled, with its axis parallel to that of the rotor shaft A, A', in bearings K and K' mounted in the rotor discs AR and A'R respectively, has fast with it at one end a gear wheel L meshing with an internally-toothed gear wheel M coaxial, and fast, with rotor R0, whilst at its other end said layshaft J has fast with it a gear wheel L' meshing with an externally-toothed gear wheel M' coaxial, and fast, with the sleeve G1 and thus rotor R1; in this embodiment, the gear L', M' is enclosed by casing N attached at one end to the rotor disc A'R and having rotational clearance with flanges attached respectively to the hub of gear wheel M' and the sleeve G1 through a sealing gland G.

In operation, with the rotor shaft A, A' rotated at a speed (revolutions per minute), and thus a determinate relative velocity between the co-operating blades  $r$  and  $s$  of the intermediate-pressure group, corresponding to the design ratio (as hereinbefore defined) of the compressor, the low-pressure bladed rotor R0,  $r0$  and the high-pressure bladed rotor R1,  $r1$  will be constrained, through the gears LM and L'M' and layshaft J, to rotate with the rotor shaft A, A' at determinate speeds, and thus relative blade velocities for the respective blade groups, such that the torque acting through the high-pressure blades  $r1$  on the rotor R1,  $r1$  is balanced by the torque acting through the low-pressure blades  $r0$  on rotor R0,  $r0$ .

However, when the pressure ratio varies from the design ratio for the compressor, as for instance due to change of speed of the rotor shaft A, A', the torque acting through the high-pressure blades  $r1$  on rotor R1,  $r1$  will be greater or less than the torque acting through the low-pressure blades  $r0$  on rotor R0,  $r0$ , according as the pressure ratio is respectively greater or less than the design ratio. The resultant unbalanced torque acts through gears L, M and L', M' differentially on the rotors R0,  $r0$  and R1,  $r1$  accordingly to cause the speed of one of them, say R0,  $r0$  automatically to be increased and that of the other, say R1,  $r1$  to be decreased, or vice versa, until the condition of balanced torque is restored.

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In this manner, and by appropriate design, the arrangement will function so that, when the pressure ratio of the compressor falls below the design ratio for the compressor, the relative blade velocity for the low-pressure blade group  $r0$ ,  $s0$  will be correspondingly reduced while that for the high-pressure blade group  $r1$ ,  $s1$  will be correspondingly increased with reference to the relative blade velocity for the intermediate-pressure group  $r$ ,  $s$ ; conversely, when the pressure ratio rises above the design ratio, the relative blade velocity for the low-pressure blade group  $r0$ ,  $s0$  will correspondingly be increased and that for the high-pressure blade group  $r1$ ,  $s1$  reduced, with reference to the relative blade velocity for the intermediate pressure group  $r$ ,  $s$ . In this manner, the efficiency of the compressor may be maintained over a desired range of pressure ratio.

The general sense of the changes in relative blade velocities for low-pressure and high-pressure blade groups effected automatically in accordance with the invention can be seen from Fig. 2, which relates to a preferred practical example of the compressor arrangement according to Fig. 1. In the diagram, wherein abscissae denote values of relative blade velocity and ordinates denote values of overall pressure ratio for the compressor, the curve  $lp$  indicates the relation between the relative blade velocity for the low-pressure blade group and the pressure ratio of the compressor, while curve  $hp$  indicates the relation for the high-pressure blade group. As indicated in the diagram, the relative blade velocities for the low-pressure and high-pressure blade groups are equal when the compressor is operating at a pressure ratio given by the ordinate of line  $n$ . This value of pressure ratio will preferably be equal to the design ratio for the compressor; in these circumstances, there being little or no differential motion between the low-pressure and high-pressure bladed rotors at and near to the normal rated condition for the compressor, the relative tooth speed for gears  $M$ ,  $L$ , and  $M'$ ,  $L'$  is low and the gear losses correspondingly small.

The arrangement illustrated in Fig. 3 is similar to that of Fig. 1 inasmuch as the compressor includes an external stator  $S$  having an inlet, or low-pressure, blade group  $s0$ , an intermediate-pressure blade group  $s$  and an outlet, or high-pressure, blade group  $s1$ , together with coaxial rotors  $R0$ ,  $R$  and  $R1$  equipped with groups of blades  $r0$ ,  $r$  and  $r1$  respectively to cooperate with the blade groups  $s0$ ,  $s$ ,  $s1$ ; moreover, the rotor  $R$  for the intermediate-pressure blades  $r$  is fast with a main rotor shaft  $A$  which will be coupled to the turbine or other driving means for the compressor, while the rotors  $R0$  and  $R1$  for the low-pressure blade group  $r0$  and high-pressure blade group  $r1$  respectively are rotatably mounted on said rotor shaft  $A$  and are associated with the rotor  $R$  for the intermediate-pressure blades  $r$  through differential torque balancing gear.

In the differential torque balancing gear of Fig. 3 a bevel gear wheel  $l$  meshing with bevel gear wheels  $g0$  and  $g1$  respectively attached to, so as to be rotatable with, the rotors  $R0$  and  $R1$  for the low-pressure and high-pressure blade groups  $r0$  and  $r1$  respectively, is mounted rotatably through bearings  $k$  on a layshaft  $a$  which is fixed to and extends radially of the rotor shaft  $A$  and connects said shaft with a flange  $aR$  fast with the rotor  $R$  for the intermediate-pressure blade group  $r$ ; the gear wheel  $l$  carried by the rotor shaft  $A$  thus providing a differential drive

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between the gears  $g0$  and  $g1$  connected respectively to the rotors  $R0$  and  $R1$ .

The differential torque balancing gear of Fig. 3 acts in similar manner to that described in respect of the arrangement of Fig. 1, namely, automatically to cause increase and decrease of the relative blade velocity between the blades  $r0$  and  $s0$  of the low-pressure, or inlet, blade group, and decrease and increase of the relative blade velocity between the blades  $r1$  and  $s1$  of the high-pressure, or outlet, blade group, with respect to the relative blade velocity between the blades  $r$  and  $s$  of the intermediate-pressure or central blade group, so as to compensate for increase and decrease respectively of the pressure ratio from the design ratio for the compressor.

The arrangement illustrated in Fig. 4 is generally similar to that of Fig. 1 in that the compressor includes an outer bladed element  $S$  having an inlet or low-pressure group of blades  $s0$ , an intermediate group of blades  $s$  and an outlet, or high-pressure, group of blades  $s1$ , together with three coaxial bladed elements, namely, an inlet end element  $R0$  carrying blades  $r0$  for co-operating with the low-pressure blades  $s0$ , a central element  $R$  equipped with blades  $r$  for co-operating with intermediate-pressure blades  $s$ , and an outlet end element  $R1$  equipped with blades  $r1$  for co-operating with high-pressure blades  $s1$ ; moreover, the elements  $R0$  and  $R1$  for the low-pressure and high-pressure blades  $r0$  and  $r1$  respectively are mounted on, so as to be rotatable relatively to, respective shafts  $A$  and  $A'$  to which the bladed element  $R$  is attached by discs or flanges  $AR$  and  $A'R$  respectively; furthermore, said bladed elements  $R0$  and  $R1$  are associated with the bladed element  $R$  through differential torque balancing gear.

However, in the arrangement of Fig. 4, the outer bladed element  $S$  is mounted through bearings  $B$  and  $B'$  on the shafts  $A$  and  $A'$  respectively and is adapted to be coupled to the turbine or other driving means for the compressor so as to be rotated at full speed, as by connecting its hub at the extreme left-hand end in this figure to the driving turbine, whilst the shafts  $A$  and  $A'$  are immobile, the shafts  $A$  and  $A'$  being attached at their outer ends to stationary casing  $SS$  with which the outer bladed element  $S$  has rotational clearance through sealing glands  $SG$ .

The differential torque balancing gear for this embodiment is of similar construction to that shown in Fig. 1 in that it includes gear wheels  $L$  and  $L'$  which are carried on a layshaft  $J$  mounted through bearings  $K$  and  $K'$  in the shaft attachments  $AR$  and  $A'R$ , and are respectively in mesh with gear wheels  $M$  and  $M'$  fast with rotors  $R0$  and  $R1$  respectively.

In operation, the outer bladed element  $S$  and therewith the blades  $s0$  of the low-pressure blade group, blades  $s$  of the intermediate-pressure blade group, and blades  $s1$  of the high-pressure blade group, will be rotated at full speed while the inner bladed element  $R$  and therewith the blades  $r$  of the intermediate-pressure blade group, will be stationary; moreover, the blades  $r0$  of the low-pressure group and the blades  $r1$  of the high-pressure group will remain stationary so long as the torque acting on the low-pressure blade  $r0$  is equal to, and balanced by, the torque acting on the high-pressure blades  $r1$ , namely, the conditions corresponding to the compressor operating normally, that is, at its rated speed, to give an overall pressure ratio having a value equal to that of the design ratio,

7 However, when the pressure ratio exceeds the design ratio the torque acting on the blades  $r1$  of the high-pressure blade group will exceed the torque acting on the blades  $r0$  of the low-pressure group and this unbalanced torque will act through the differential gear L, M and L', M' and layshaft J to cause the low-pressure blades  $r0$  to rotate slowly in the opposite direction to that in which the co-operating low-pressure blades  $s0$  are rotating and to cause the high-pressure blades  $r1$  to rotate in the same direction as that in which rotate the co-operating high-pressure blades  $s1$ , so that the relative blade velocity between co-operating blades of the low-pressure blade group  $r0, s0$  will be automatically increased while the relative blade velocity between co-operating blades of the high-pressure blade group  $r1, s1$  will be automatically reduced, with respect to the relative blade velocity between co-operating blades of the intermediate-pressure blade group  $r, s$ , until the condition of balanced torques is restored. Conversely, when the pressure ratio becomes less than the design ratio, the torque acting on the low-pressure blades  $r0$  will exceed the torque acting on the high-pressure blades  $r1$  and the out-of-balance torque will act through the differential gear M, L and M', L' to cause the high-pressure blades  $r1$  to rotate in the opposite direction to that in which rotate the co-operating blades  $s1$  and to cause the low-pressure blades  $r0$  to rotate in the same direction as that in which rotate the co-operating blades  $s0$ ; thus, the relative blade velocity between the co-operating blades of the low-pressure group  $r0, s0$  will be reduced while the relative blade velocity between co-operating blades of the high-pressure blade group  $r1, s1$  will be increased, with respect to the relative blade velocity between the co-operating blades of the intermediate-pressure group  $r, s$ .

In some instances, the arrangement may be such that, during normal operation of the compressor with overall pressure ratio substantially at the value for the design ratio, the low-pressure blades  $r0$  rotate slowly in the same direction as that of the co-operating blades  $s0$ , while the high-pressure blades  $r1$  rotate slowly in the opposite direction to that of the co-operating blades  $s1$ , so that the relative blade velocity between co-operating blades increases progressively in three steps from the inlet end to the outlet end of the compressor. Evidently, with this arrangement, the differential torque balance gear will function in similar manner to that already described to adjust the relative blade velocities in the low-pressure, or inlet, and high-pressure, or outlet, blade groups, with respect to the relative blade velocity in the intermediate-pressure, or central, blade group to compensate for variation of pressure ratio from the design ratio.

Still referring to Fig. 4, it will readily be appreciated that the shaft A, A' may be arranged for contra-rotation relatively to the outer bladed element S; in this case, the low-pressure, intermediate-pressure and high-pressure blades  $r0, r$  and  $r1$  respectively will be rotated at full speed contra-rotationally with their co-operating blades  $s0, s$  and  $s1$  respectively, during such time as the overall pressure ratio of the compressor is at, or near to, the design ratio, and the differential gear L, M and L', M' and layshaft J will act to cause appropriate adjustment of the relative blade velocities in the low-pressure and high-pressure blade groups respectively, with respect to the relative blade velocity in the intermediate-

pressure group, as and when the overall pressure ratio departs from the design ratio.

In the arrangement illustrated in Fig. 5, the compressor comprises a bladed main rotor and an outer and coaxial subdivided bladed element co-operating therewith. As shown in the drawing, the main rotor R attached to shafts A, A' which are mutually journaled in bearings BB and B'B respectively, carried by a stationary casing SS and which shafts will be coupled to the turbine or other driving means for the compressor, is equipped with a centrally disposed group of blades  $r$  and end groups of blades  $r0$  and  $r1$  respectively. The outer subdivided bladed element comprises a central portion S equipped with blades  $s$  for co-operating with the said rotor blades  $r$  and constituting therewith the intermediate-pressure blade group, and two end portions  $S0$  and  $S1$  respectively equipped with blades  $s0$  and  $s1$  for co-operating, the blades  $s0$  with the rotor blades  $r0$  and constituting therewith the low-pressure blade group, and the blades  $s1$  with the rotor blades  $r1$  and constituting therewith the high-pressure blade group.

The low-pressure and high-pressure bladed elements  $S0$  and  $S1$  respectively are mutually mounted on, so as to be rotatable relatively to, the rotor shaft A, A' through bearings B and B' respectively while the intermediate-pressure bladed element S is fast with stationary casing SS.

In this embodiment the rotor  $S0$  for the low-pressure blades  $s0$  is interconnected with the rotor  $S1$  for the high-pressure blades  $s1$  through a differential gear wheel train in which layshaft O, mounted through bearings P and P' respectively on stationary casing SS so as to be rotatable about an axis parallel to that of the rotor shaft A, A', has fast with it gear wheels Q and Q'. Gear wheel Q is in mesh with gear wheel T which is fast with the rotor  $S0$  for the low-pressure blades  $s0$ . The gear wheel Q', however, is in mesh with a gear wheel U mounted through bearings V on a stub shaft W which is attached to the casing SS with its axis parallel with that of the layshaft O. In turn, gear wheel U is in mesh with gear wheel T' which is fast with the rotor  $S1$  for the high-pressure blades  $s1$ .

In operation, the low-pressure and high-pressure blades  $r0$  and  $r1$  respectively will be rotated with the intermediate-pressure blades  $r$  at full speed while the co-operating blades  $s0$  and  $s1$  will remain substantially stationary with blades  $s$  so long as the overall pressure ratio of the compressor is at, or near to, the value for the design ratio, owing to the fact that in these circumstances the torque acting on the low-pressure blades  $s0$  is substantially equal to, and balanced by, the torque acting on the high-pressure blades  $s1$ .

However, when the pressure ratio exceeds the design ratio the torque acting on the high-pressure blades  $s1$  will exceed the torque acting on the low-pressure blades  $s0$  and the unbalanced torque will act through the differential gear Q, T and Q', U, T' to cause the rotor  $S0$  for the low-pressure blades  $s0$  to rotate in opposite direction to that in which the co-operating low-pressure blades  $r0$  are rotating, and to cause the rotor  $S1$  for the high-pressure blades  $s1$  to rotate in the same direction as that in which the co-operating high-pressure blades  $r1$  are rotating, the intermediate-pressure blades  $s$  of course remaining stationary; conversely, when the pressure ratio becomes less than the design ratio the

torque acting on the low-pressure blades  $s_0$  will exceed the torque acting on the high-pressure blades  $s_1$  and the unbalanced torque will act through the differential gear to cause the rotor  $S_1$  for the high-pressure blades  $s_1$  to rotate in opposite direction to that in which the high-pressure blades  $r_1$  are rotating and to cause the low-pressure blades  $s_0$  to rotate in the same direction as that in which the co-operating low-pressure blades  $r_0$  are rotating. As will readily be appreciated the blades of the high- or low-pressure group which are caused by the differential gear to rotate in opposite direction to that in which the co-operating blades of the group are rotated will provide an increase in relative blade velocity for the appertaining blade group while the blades of the other group which are caused by the differential gear to rotate in the same direction as that in which the co-operating blades of the group are rotated will provide a reduction in the relative blade velocity for the appertaining group.

In the arrangement illustrated in Fig. 6 for a compressor of the radial flow type, shafts  $1$  and  $1'$  which are journaled in axial alignment with one another in bearings  $2$  and  $2'$  carried by a stationary casing  $3$  and which shafts will be coupled to the turbine or other driving motor for the compressor so as to be rotated in opposite directions, are attached by radial arms or discs  $1a$ ,  $1'a$  to co-operating bladed elements in the form of discs  $4$ ,  $4'$  extending radially of the axis of the shafts  $1$ ,  $1'$  and respectively equipped with blades  $4a$  and  $4'a$ . Disposed at opposite ends of the group of co-operating blades  $4a$  and  $4'a$  are two groups of co-operating blades  $5a$ ,  $5'a$  and  $6a$ ,  $6'a$  respectively, the blades  $5a$  and  $5'a$  being at the low-pressure, or inlet, end of the blades  $4a$ ,  $4'a$  and the blades  $6a$ ,  $6'a$  being at the high-pressure, or outlet, end thereof.

The low-pressure co-operating blades  $5a$ ,  $5'a$  are attached to respective arms or discs  $5$ ,  $5'$  extending radially of the axis of shafts  $2$ ,  $2'$  and having attached sleeves  $5b$ ,  $5'b$  between which and the shafts  $2$ ,  $2'$  are interposed bearings  $5c$ ,  $5'c$  respectively.

The high-pressure co-operating blades  $6a$ ,  $6'a$  are attached to respective discs  $6$ ,  $6'$  extending radially of the axis of shafts  $2$ ,  $2'$  and having attached sleeves  $6b$  and  $6'b$  mounted through bearings  $6c$  and  $6'c$  on shafts  $2$ ,  $2'$  respectively.

It will be appreciated that air or other working fluid will be admitted radially to the compressor through diffuser means (not shown) and will pass through the group of low-pressure co-operating blades  $5a$ ,  $5'a$ , thence through the group of intermediate-pressure blades  $4a$ ,  $4'a$ , and after passing through the group of high-pressure blades  $6a$ ,  $6'a$  will be discharged through an outlet in casing  $3$ .

In this embodiment the rotors  $5$  and  $6$  for the low-pressure blades and high-pressure blades  $5a$  and  $6a$  respectively are interlinked with one another and associated with the rotor  $4$  for the intermediate-pressure blades  $4a$  through differential gear comprising gear wheel  $8$  mounted through bearings  $9$  on stub shaft  $10$  which is attached to the rotor  $4$  with its axis parallel with the axis of shafts  $2$ ,  $2'$ , and conveniently serves in the attachment of said rotor  $4$  with the radial arm or disc  $1a$  fast with rotor shaft  $1$ , said gear wheel  $8$  meshing with an externally toothed gear wheel  $5d$  fast with the rotor  $5$  and also in mesh with an internally toothed gear wheel  $6d$  fast with the rotor  $6$ . Likewise, the rotor  $5'$  and the

rotor  $6'$  are interconnected with one another and associated with the rotor  $4'$  through differential gear comprising gear wheel  $8'$  mounted through bearings  $9'$  on stub shaft  $10'$  fast with rotor  $4'$  and conveniently acting to attach said rotor with the radial arm or disc  $1'a$  fast with the rotor shaft  $2'$ .

In operation, with the rotor shafts  $1$ ,  $1'$  driven contra-rotationally, the co-operating blades of the intermediate-pressure group  $4a$ ,  $4'a$  will be correspondingly rotated at full speed and, so long as the pressure ratio of the compressor remains at the value for the design ratio, the low-pressure blades  $5a$  and the high-pressure blades  $6a$  will be rotated substantially in unison with said intermediate-pressure blades  $4a$ , while the low-pressure blades  $5'a$  and the high-pressure blades  $6'a$  will rotate substantially in unison with the intermediate-pressure blades  $4'a$ , the torque acting on the low-pressure blade group  $5a$ ,  $5'a$  being equal to, and balanced by, that acting on the high-pressure blade group  $6a$ ,  $6'a$ .

However, when the pressure ratio of the compressor departs from the design ratio, the torque acting on the co-operating blades of the low-pressure blade group  $5a$ ,  $5'a$  will become greater or less than the torque acting on the co-operating blades of the high-pressure blade group  $6a$ ,  $6'a$ , the sense of the unbalanced torque being determined by the sense of the departure of the pressure ratio from the design ratio. This unbalanced torque will act through the differential gear  $5d$ ,  $8$ ,  $6d$  to cause the low-pressure blades  $5a$  and the high-pressure blades  $6a$  to rotate in opposite directions relatively to the intermediate-pressure blades  $4a$ , and likewise through gear  $5'd$ ,  $8'$ ,  $6'd$  to cause the low-pressure blades  $5'a$  and the high-pressure blades  $6'a$  to rotate in opposite directions relatively to the intermediate-pressure blades  $4'a$  and in the same sense as that of the co-operating blades  $5a$ ,  $6a$ , respectively, until the condition of balanced torques is restored; in this manner, the relative blade velocity for the co-operating blades of the high-pressure blade group and the relative blade velocity for the co-operating blades of the low-pressure blade group will be automatically adjusted with respect to the relative velocity of the co-operating blades of the intermediate-pressure group.

It will be appreciated that in all the arrangements illustrated in the drawings, the number of rows of blades in the respective low-pressure, high-pressure and intermediate-pressure blade groups may be varied according to requirements; the design of the blades in each row or rows for the respective groups may be carried out according to the well-known principles of blade design for both axial and radial flow compressors. Moreover, the design of differential torque balancing gear may be varied and such gear may be arranged to secure different relative speeds of the rotors for the respective blade groups when the compressor is operating at the rated design ratio.

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

1. A multi-stage compressor comprising two relatively rotatable elements, a plurality of rows of blades attached to one of said elements, including one row disposed intermediately of two end rows; a row of blades fixed to the other element to co-operate with said intermediately-disposed row of blades in providing an intermediate stage of compression, a row of blades having means for rotatably mounting them relatively

to said other element to co-operate with one of said end rows of blades in providing a low-pressure stage of compression, and a row of blades having means for rotatably mounting them relatively to said other element to co-operate with the other said end row of blades in providing a high-pressure stage of compression, and means including gearing having means fixed to the blades which are fixed to said other element for supporting said gearing for rotation on an axis eccentric to the axis of relative rotation of said elements, and gears connected respectively to said rotatably mounted blades of the low-pressure and high-pressure stages and meshing with said gearing whereby blades of the low-pressure and high-pressure stages are differentially interconnected blades of the intermediate-pressure stage for inversely changing the relative blade velocities of the low-pressure and high-pressure stages relatively to that of the intermediate-pressure stage.

2. A multi-stage compressor comprising a casing, three groups of blades attached thereto one group intermediately of the two other groups, a driving shaft journaled in the casing, a group of blades fixed to said shaft for direct drive therefrom and for cooperation with blades of the intermediate group of casing blades, two sleeves rotatably mounted on said shaft, two groups of blades respectively attached to said sleeves, one group of sleeve-attached blades to co-operate with the group of blades attached to the casing at one end of the intermediate blade group and the other group of sleeve-attached blades to co-operate with blades attached to the casing at the other end of the intermediate blade group, and means for rotating said sleeve-attached blades including gearing secured to said driving shaft for rotation on an axis other than the axis of the shaft and in fixed relation with said blading fixed to the shaft and for bodily rotation about the axis of said shaft, and gears meshing with said gearing and attached respectively to each group of sleeve-attached blades.

3. A multi-stage compressor comprising a casing, three groups of blades attached thereto, one intermediately of two-end groups, a shaft journaled in the casing and having attached thereto for rotation in invariable speed relation therewith blades to co-operate with blades of the intermediate group of casing blades, two sleeves rotatably mounted on said shaft, two groups of blades respectively attached to said sleeves, one group of sleeve-attached blades to co-operate with one of said end groups of casing blades and the other group of sleeve-attached blades to co-operate with the other end group of casing-attached blades, a gear wheel coaxially attached to one sleeve, a gear wheel coaxially attached to the other sleeve, a radial projection on said shaft in fixed relation with said shaft-attached blades, and a gear wheel journaled on said projection and in mesh with both said sleeve-attached gear wheels.

4. A multi-stage turbo-compressor comprising a casing, a plurality of rows of blades attached thereto, a main rotor housed within the casing, a row of blades fixed to said main rotor to co-operate with blades attached to the casing to increase the pressure of fluid passing through the casing transversely of the rows of blades, a row of blades having means for mounting them coaxially of the main rotor for free rotation relatively thereto and for cooperation with the row of blades attached to the casing at the

fluid-inlet end of the rows of blades, a row of blades having means for mounting them coaxially of the main rotor for free rotation relatively thereto and for cooperation with the row of blades attached to the casing at the fluid-outlet end of the rows of blades, and means including a differential-gear mechanism mounted for rotation on an axis which is rotatable about the axis of the main rotor and is in fixed relation with the blades fixed to the main rotor for bodily rotation therewith and with the blades fixed thereto, and gears meshing with said gear mechanism and connected respectively to the freely rotatable inlet blade row and the freely rotatable outlet blade row whereby said rotatable inlet and outlet blade rows are interconnected for rotation in opposite directions relatively to the blades fixed to the main rotor.

5. A multi-stage axial-flow turbo-compressor comprising a casing, three axially-spaced rows of blades attached thereto, a driving shaft journaled in the casing, a row of blades having means for rotatably mounting them on said shaft for co-operating with the row of blades attached to the casing at the fluid-inlet end of the casing, a row of blades having means for rotatably mounting them on said shaft for co-operating with the row of blades attached to the casing at the fluid-outlet end of the casing, a row of blades fast with the shaft for rotation therewith and co-operating with the row of blades attached to the casing intermediately of the inlet and outlet rows of casing blades, and means including differential gear mechanism mounted for rotation on an axis which is rotatable about the axis of the driving shaft and is in fixed relation with the blades fast with said shaft for rotation therewith, and gears connected respectively to the rotatable inlet blades and the rotatable outlet blades and meshing with said gear mechanism whereby the rotatable inlet row of blades and the rotatable row of outlet blades are interconnected for rotation at different speeds one greater, and the other less, than that of the row of blades fast with the shaft.

6. A multi-stage axial-flow turbo-compressor comprising a casing, a shaft journaled in the casing, a group of relatively rotatable blades including a row of blades attached to the casing at the fluid-inlet end thereof, and a co-operating row of blades having means for rotatably mounting them on the shaft, a group of relatively rotatable blades including a row of blades attached to the casing at the fluid-outlet end thereof, and a co-operating row of blades having means for rotatably mounting them on the shaft, a group of relatively rotatable blades including a row of blades attached to the casing intermediately of the inlet and outlet rows of blades, and a co-operating row of blades fixed to the shaft intermediately of the inlet and outlet rows of blades rotatably mounted thereon, and rotatable in invariable speed relation with the shaft, and means including a differential gear means mounted for rotation on an axis which is rotatable about the axis of the shaft and is in fixed relation with the blades fixed to the shaft and is rotatable bodily with the shaft and the blades fixed thereto, and gears connected respectively to the rotatable row of inlet blades and the rotatable row of outlet blades and meshing with said gear means whereby the rotatable row of inlet blades and the rotatable row of outlet blades are interconnected with the row of intermediate blades fixed to the shaft for increasing the speed of

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relative rotation of the blades of the inlet blade group and decreasing the speed of relative rotation of the blades of the outlet blade group and vice versa relatively to the speed of relative rotation of blades of the intermediate blade group.

7. A multi-stage turbo-compressor comprising a casing, a plurality of rows of blades attached thereto, a main rotor housed within the casing, a row of blades attached to the main rotor to co-operate with blades attached to the casing to increase the pressure of fluid passing through the casing transversely of the rows of blades, a row of blades having means for mounting them coaxially on the main rotor for free rotation relatively thereto and to co-operate with the row of blades attached to the casing at the fluid inlet end of the row of blades, a row of blades having means for mounting them coaxially on the main rotor for free rotation relatively thereto and to co-operate with the row of blades attached to the casing at the fluid-outlet end of the rows of blades, a gear wheel coaxially attached to the freely rotatable fluid-inlet row of blades, a gear wheel coaxially attached to the freely rotatable fluid-outlet row of blades, a lay shaft journaled on the main rotor for rotation about an axis parallel to and spaced radially from that of said main rotor, and two gear wheels coaxially mounted on said lay shaft respectively to mesh with the gear wheels attached to the freely rotatable fluid-inlet and fluid-outlet rows of blades.

8. In a multi-stage compressor the combination of a group of blades having means for mounting them for relative rotation and co-operating to provide a low-pressure stage of com-

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pression, a group of blades having means for mounting them for relative rotation to provide a high-pressure stage of compression, and a group of blades having means for mounting them for relative rotation between said groups of blades and co-operating to provide an intermediate-pressure stage of compression, and means including gearing having means fixed to blades of the intermediate-pressure stage for supporting said gearing for rotation on an axis other than the axis of relative rotation of the blades of the intermediate-pressure stage, and gears connected respectively to blades of the low-pressure stage and blades of the high-pressure stage and meshing with said gearing whereby blades of the low-pressure stage and blades of the high-pressure stage are interconnected with blades of the intermediate-pressure stage to increase the relative blade velocity of the low-pressure stage and reduce the relative blade velocity of the high-pressure stage and vice versa correspondingly with variation of the relative blade velocity of the intermediate-pressure stage.

DAVID MACLEISH SMITH.

CHARLES GRAHAM LLOYD.

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