(No Model.)

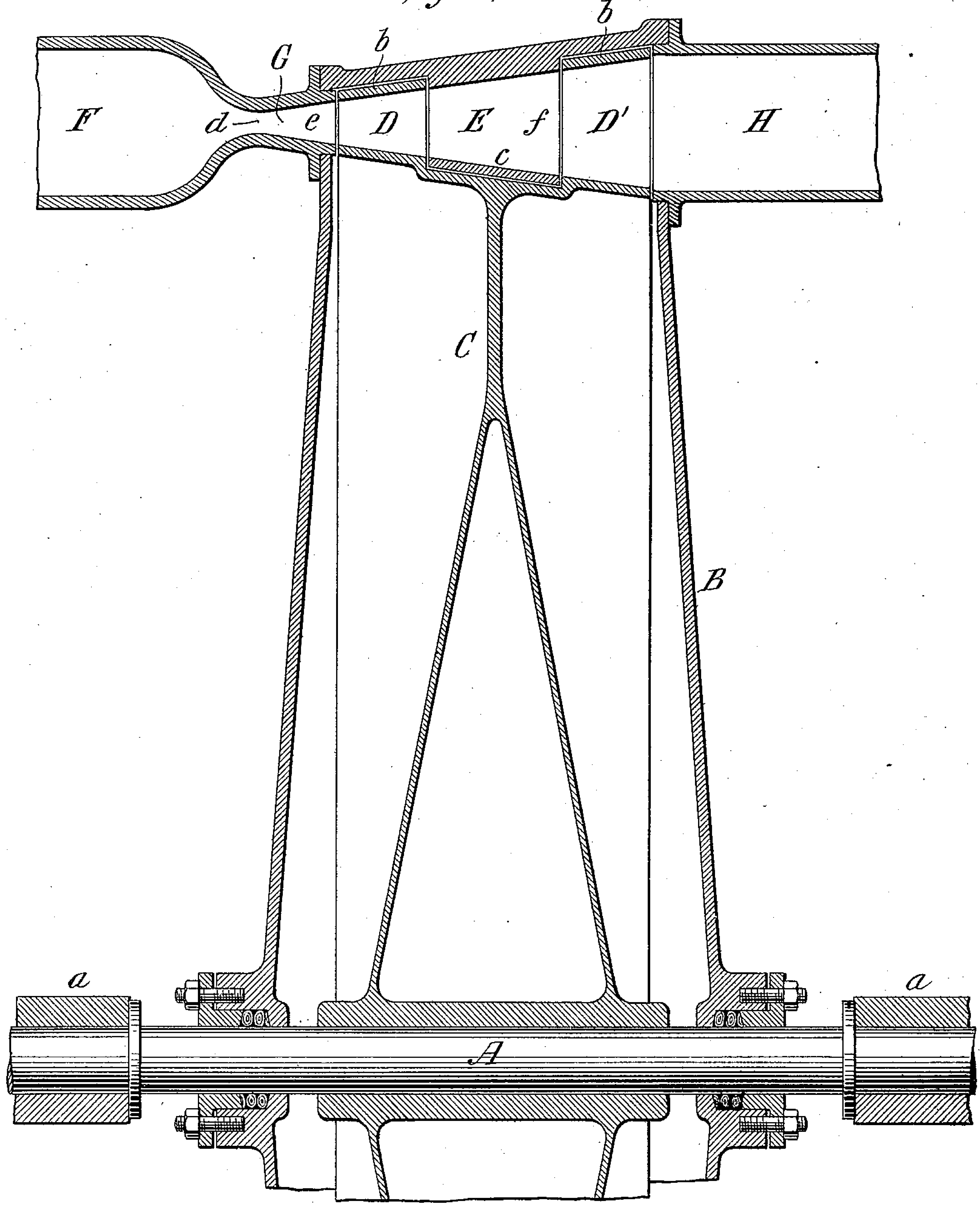
9 Sheets—Sheet 1.

C. G. CURTIS.

ELASTIC FLUID TURBINE.

No. 566,968. Patented Sept. 1, 1896.

Fig. 1,



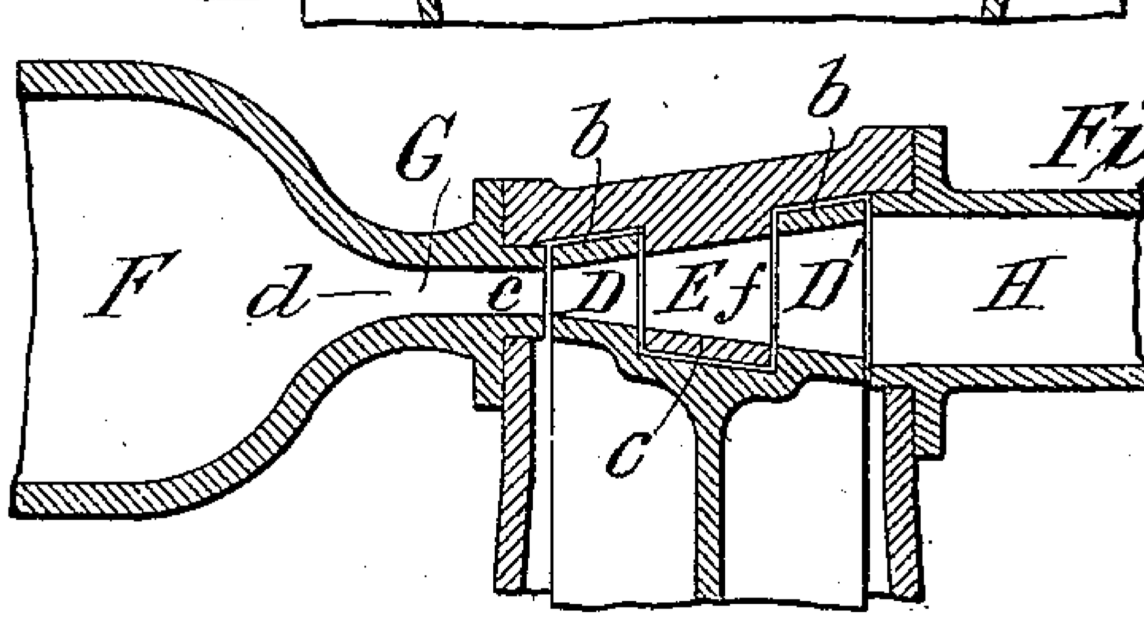


Fig. 2,

WITNESSES:

C. E. Ashley

H. W. Lloyd.

INVENTOR:

Charles G. Curtis

By his Attorneys

Dyer & Driscoll.

THE NORRIS PETERS CO., PHOTO-LITHO., WASHINGTON, D. C.

(No Model.)

9 Sheets—Sheet 2.

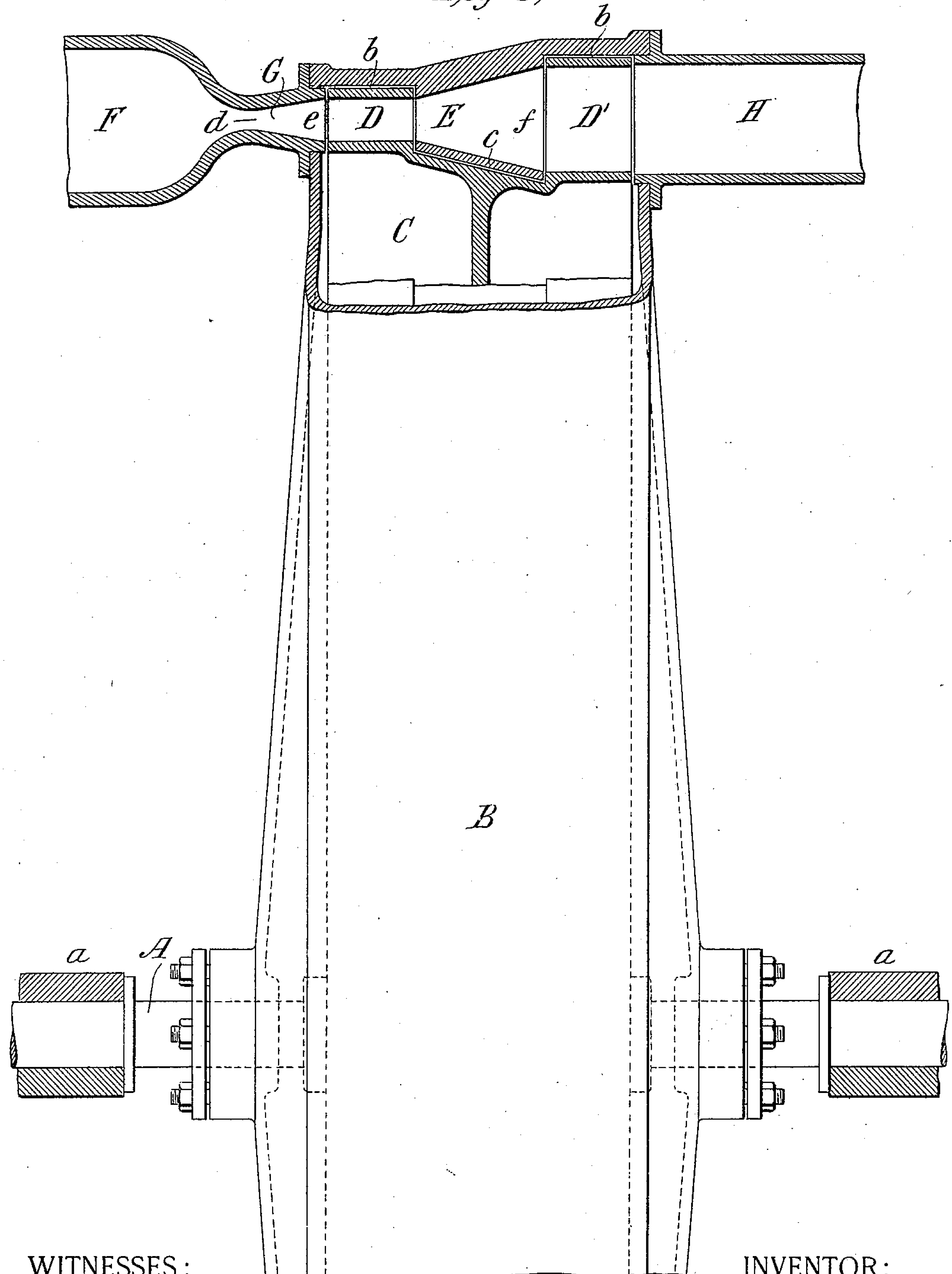
C. G. CURTIS.

ELASTIC FLUID TURBINE.

No. 566,968.

Patented Sept. 1, 1896.

Fig. 3,



WITNESSES:

C. E. Ashley

H. W. Lloyd.

INVENTOR:

Charles G. Curtis,

By his Attorneys

Dyer & Driscoll.

(No Model.)

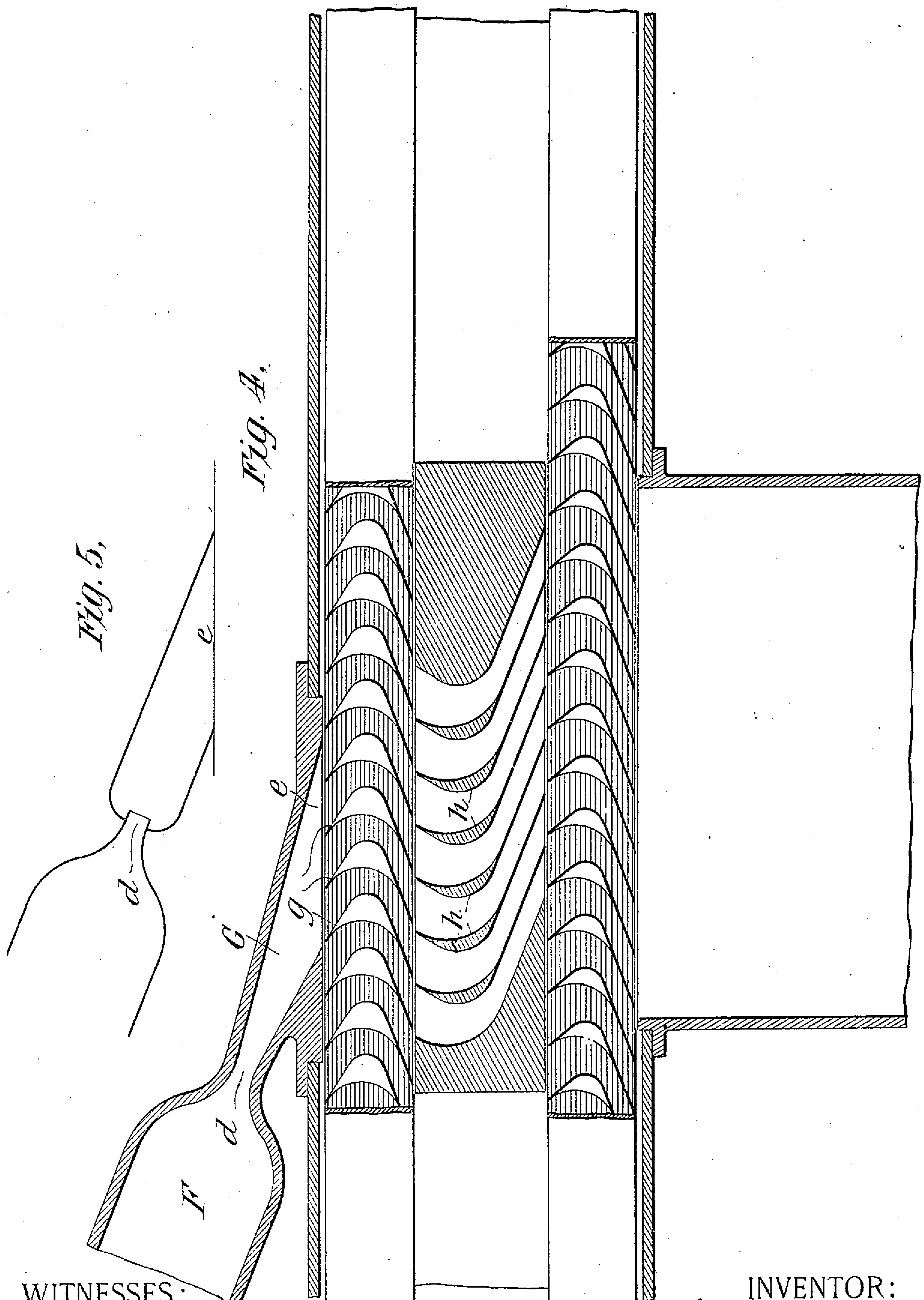
9 Sheets—Sheet 3.

C. G. CURTIS.

ELASTIC FLUID TURBINE.

No. 566,968.

Patented Sept. 1, 1896.



WITNESSES:

C. E. Ashley

H. W. Lloyd.

INVENTOR:

Charles G. Curtis

By his Attorneys

Dyer & Driscoll.

(No Model.)

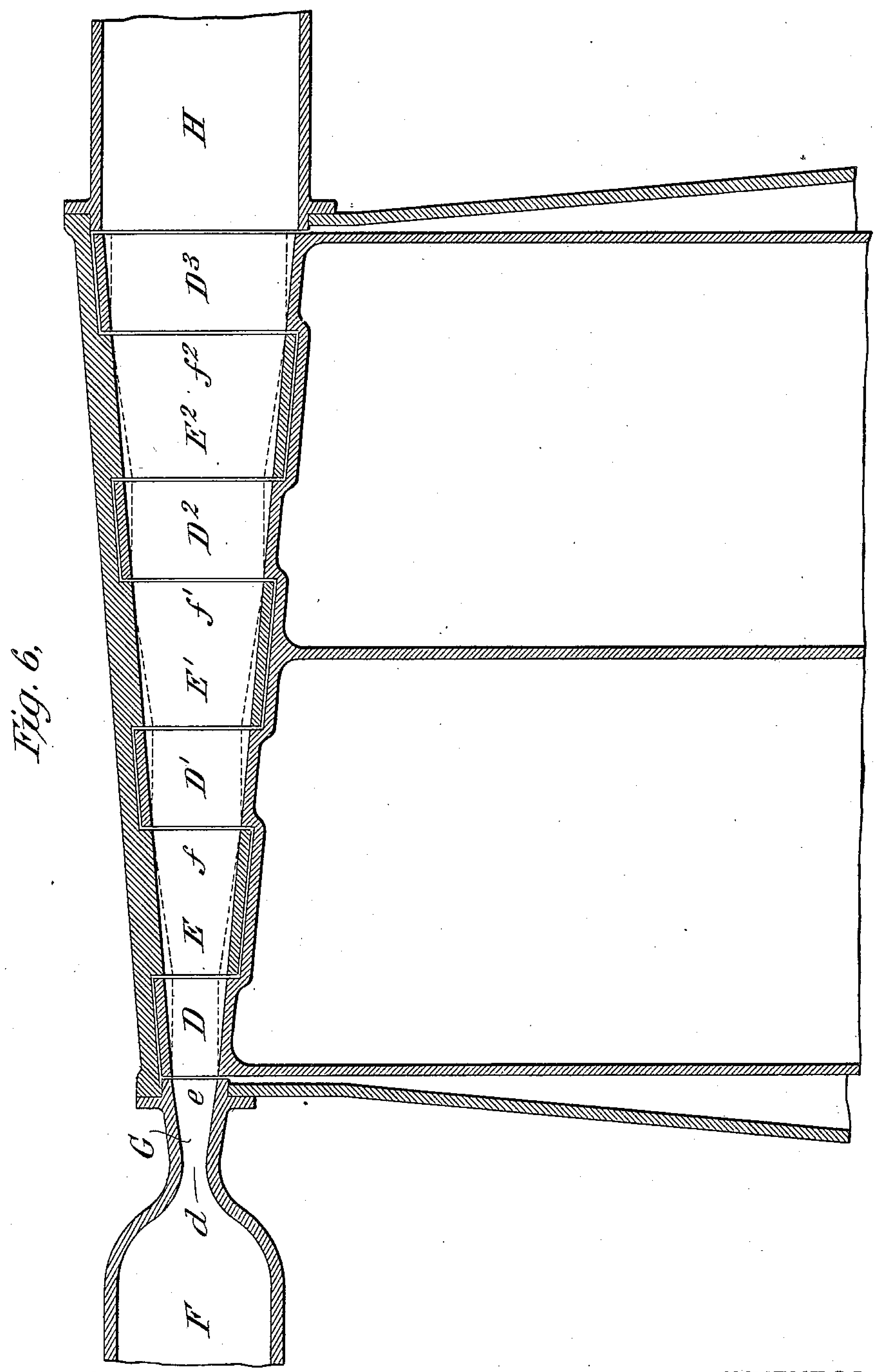
9 Sheets—Sheet 4.

C. G. CURTIS.

ELASTIC FLUID TURBINE.

No. 566,968.

Patented Sept. 1, 1896.



WITNESSES:

C. E. Ashley

H. W. Lloyd

INVENTOR:

Charles G. Curtis,

By his Attorneys

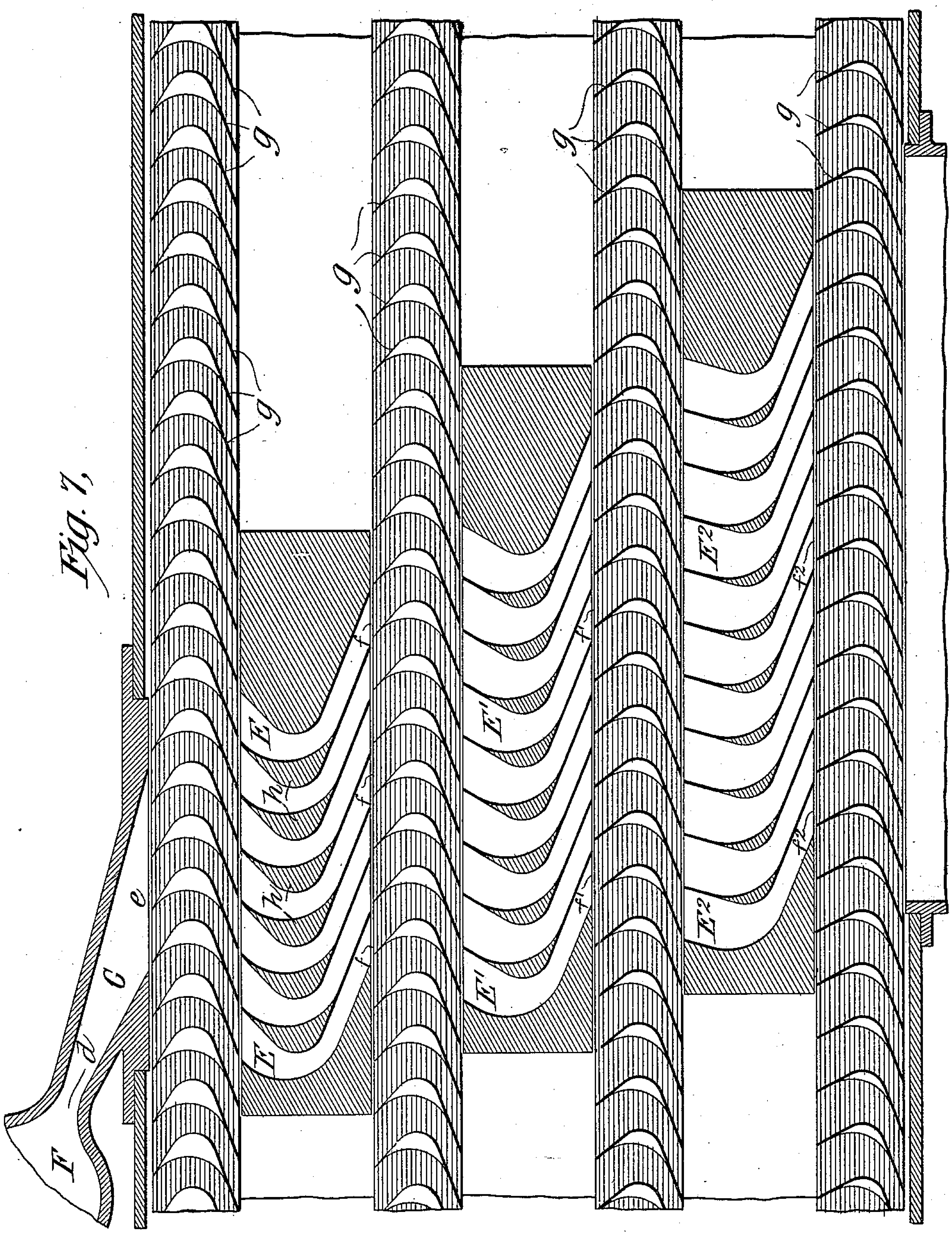
Dyer & Driscoll.

(No Model.) 9 Sheets—Sheet 5.

C. G. CURTIS.

ELASTIC FLUID TURBINE.

No. 566,968. Patented Sept. 1, 1896.



WITNESSES:

C. E. Ashley

H. W. Lloyd

INVENTOR:

Charles G. Curtis,

By his Attorneys

Dyer & Driscoll.

(No Model.)

9 Sheets—Sheet 6.

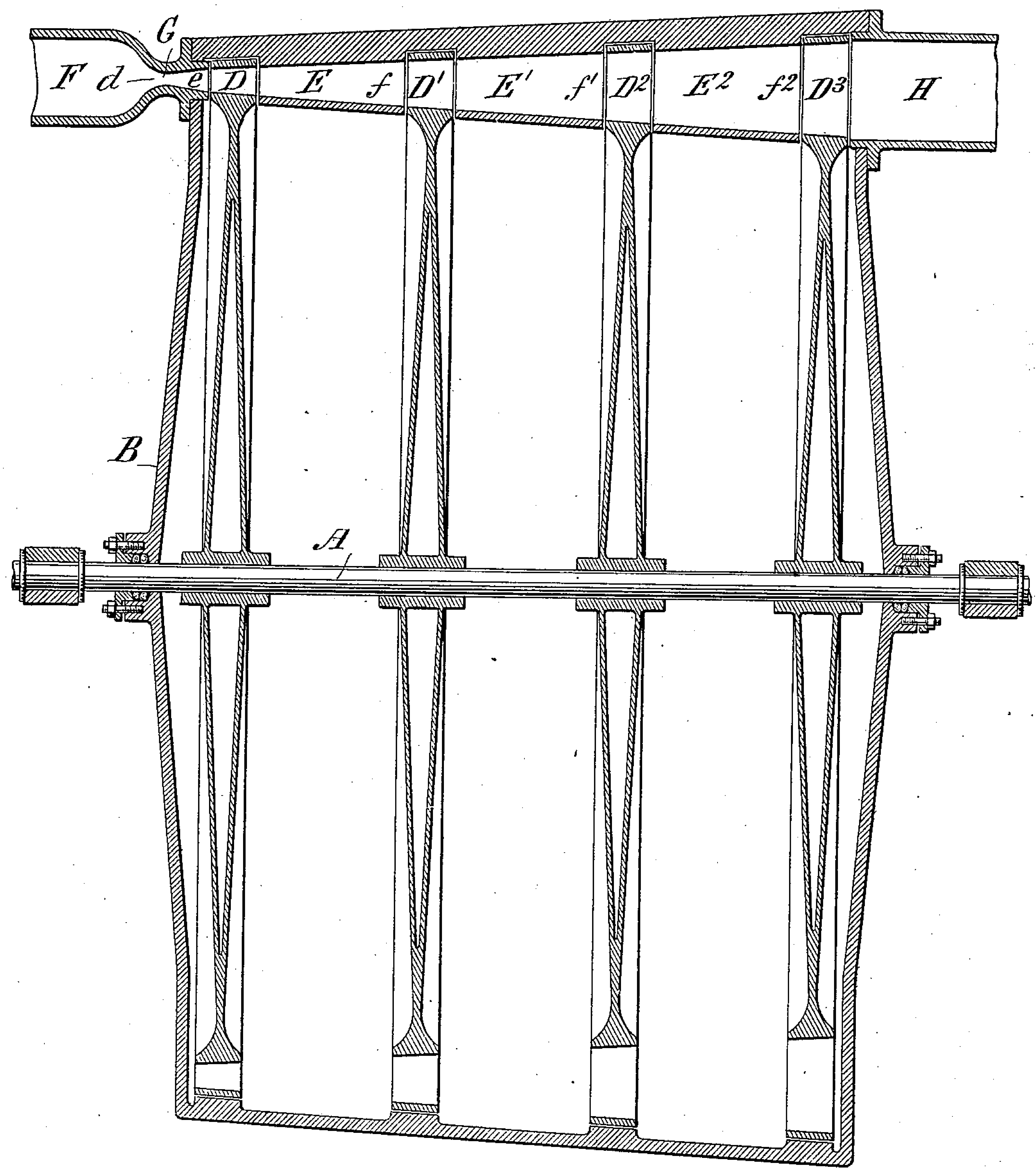
C. G. CURTIS.

ELASTIC FLUID TURBINE.

No. 566,968.

Patented Sept. 1, 1896.

Fig. 8.



WITNESSES:

C. E. Ashley

H. W. Lloyd.

INVENTOR:

Charles G. Curtis,

By his Attorneys

Dyer & Driscoll.

THE NORRIS PETERS CO., PHOTO-LITHO., WASHINGTON, D. C.

(No Model.)

9 Sheets—Sheet 7.

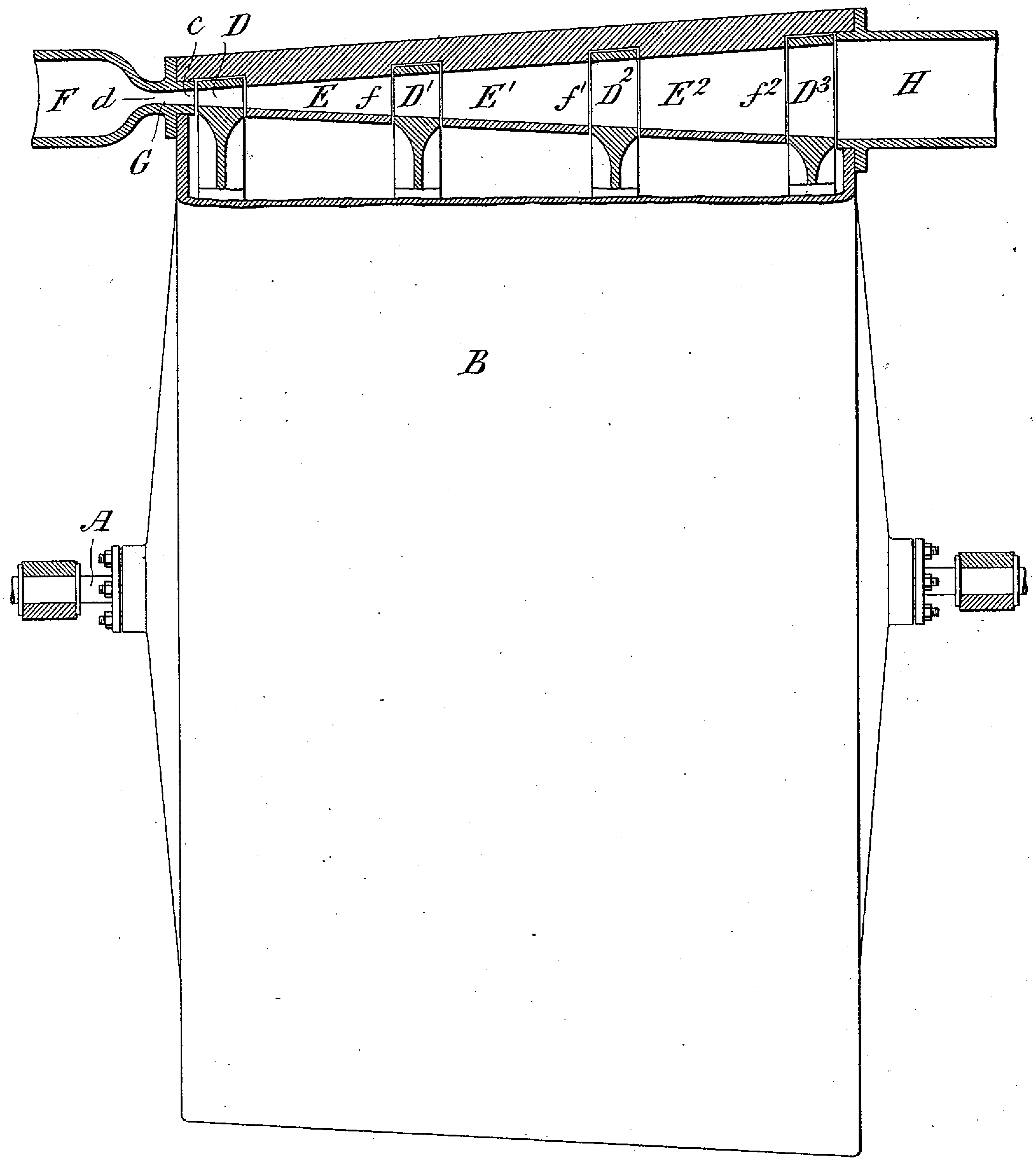
C. G. CURTIS.

ELASTIC FLUID TURBINE.

No. 566,968.

Patented Sept. 1, 1896.

Fig. 9.



WITNESSES:

C. E. Ashley

H. W. Lloyd

INVENTOR:

Charles G. Curtis,

By his Attorneys

Dyer & Driscoll.

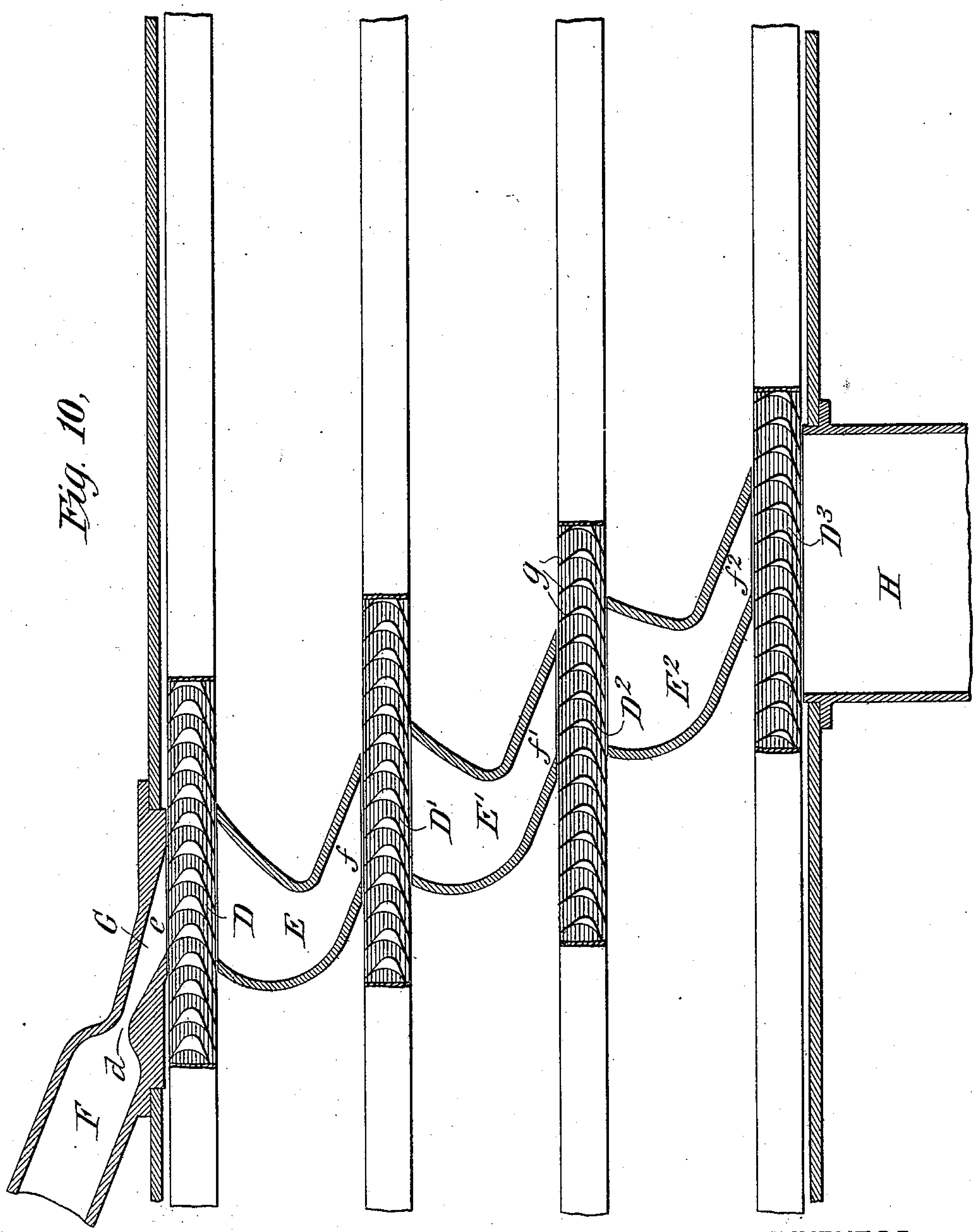
(No Model.) 9 Sheets—Sheet 8.

C. G. CURTIS.

ELASTIC FLUID TURBINE.

No. 566,968. Patented Sept. 1, 1896.

Fig. 10,



WITNESSES:

C. E. Ashley

H. W. Lloyd.

INVENTOR:

Charles G. Curtis,

By his Attorneys

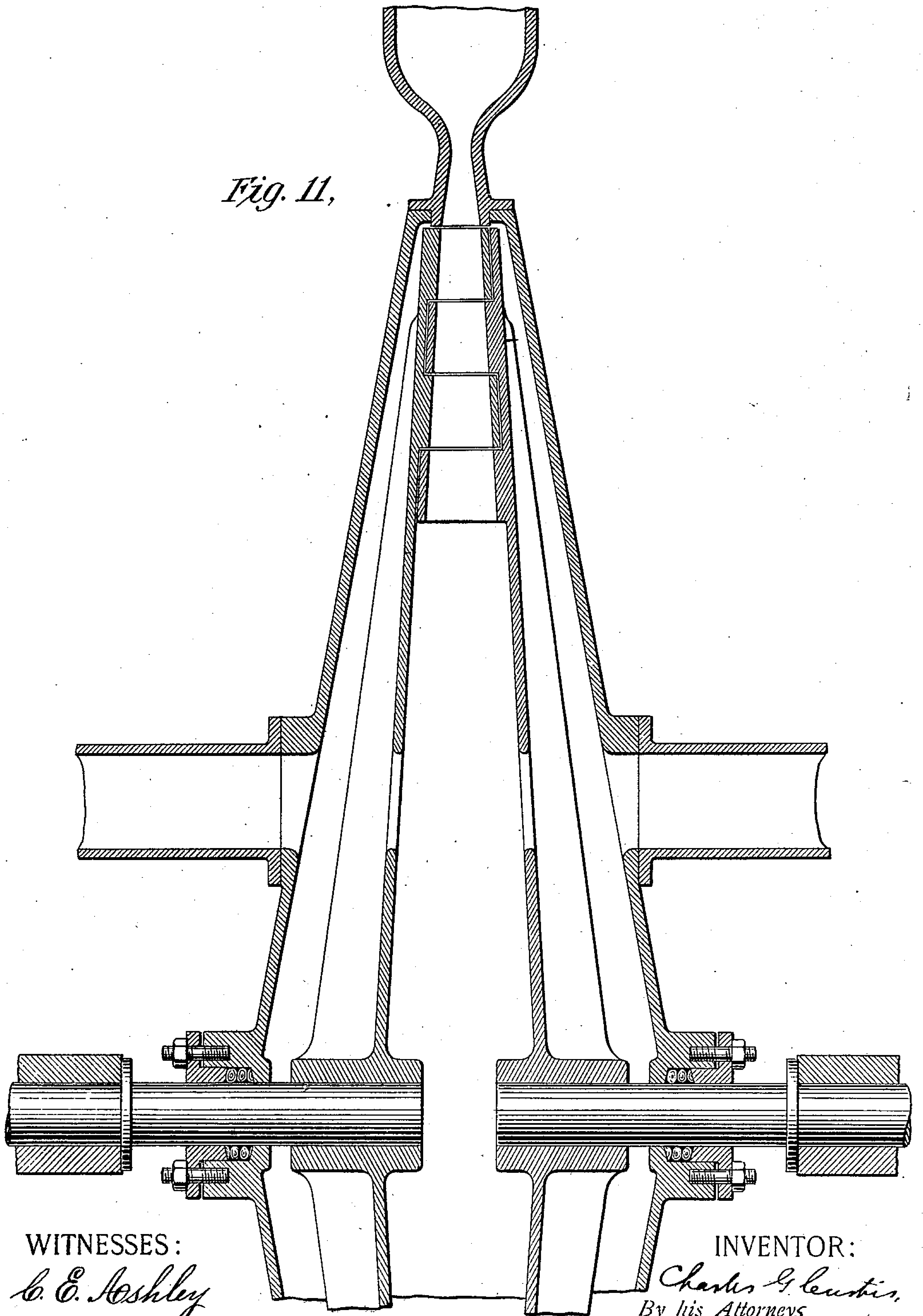
Dyer & Driscoll.

(No Model.) 9 Sheets—Sheet 9.

C. G. CURTIS.
ELASTIC FLUID TURBINE.

No. 566,968. Patented Sept. 1, 1896.

Fig. 11,



WITNESSES:
C. E. Ashley
H. W. Lloyd.

INVENTOR:
Charles G. Curtis,
By his Attorneys
Dyer & Driscoll.

UNITED STATES PATENT OFFICE.

CHARLES G. CURTIS, OF NEW YORK, N. Y., ASSIGNOR TO THE CURTIS COMPANY, OF SAME PLACE.

ELASTIC-FLUID TURBINE.

SPECIFICATION forming part of Letters Patent No. 566,968, dated September 1, 1896.

Application filed January 13, 1896. Serial No. 575,244. (No model.)

To all whom it may concern:

Be it known that I, CHARLES G. CURTIS, a citizen of the United States, residing at New York city, in the county and State of New York, have invented a certain new and useful Improvement in Elastic-Fluid Turbines, of which the following is a specification.

The object I have in view is to convert the energy of steam or other elastic fluid under pressure into mechanical power by utilizing its *vis viva* or velocity in a turbine in such a manner as to secure not only a higher efficiency, but also a much lower speed of revolution than has been obtained from any turbine heretofore, as well as increased simplicity of the apparatus.

In carrying out my invention I construct my turbine so as to first secure the highest velocity attainable in the fluid consistent with obtaining the most economical result. This I do by passing the elastic fluid through an expansion-nozzle, which converts all the pressure into *vis viva* except what may be required to cause the flow of the fluid in the most efficient manner through the vane-passages, and I abstract this *vis viva* fractionally and convert it into mechanical rotation by directing the fluid two or more times in succession against the moving vanes of the turbine, which vanes have only a fraction of the initial speed of the fluid. In this way, and if the conditions which will be hereinafter described are provided, the *vis viva* is substantially all utilized and a moderate speed of the turbine is obtained. The turbine is also one of marked simplicity compared with compound turbines heretofore employed in practical work. It has been heretofore attempted to convert the entire pressure of the steam into velocity by means of an expansion-nozzle and to utilize the high velocity thus obtained upon one set of rotating vanes. This attempt has only met with partial success for various reasons. Unless the boiler-pressure is exceedingly low the *vis viva* of the steam when fully developed involves so high a velocity that, in order to utilize it most efficiently, the speed of a single set of vanes would have to be so great as to disrupt the apparatus by centrifugal force. Consequently a lower speed than that giving the highest efficiency has been employed, and a reducing-gear has even then been necessary to bring the speed down to the point where it becomes commercially available. With my compound turbine, however, the speed of the vanes need not approach the maximum velocity of the fluid-jet in order to secure high efficiency, because the velocity which remains in the fluid-jet after leaving the first set of vanes is utilized in one or more succeeding sets of vanes. The energy extracted from the fluid and converted into mechanical rotation by each set of vanes is the difference between the *vis viva* admitted to and the *vis viva* flowing out of the vane-passages, and, other things being equal, may be roughly expressed with reference to each set of vanes as the difference between the squares of the velocities at entrance and at issue. For this reason the energy extracted by a set of vanes is increased without increasing the speed of their rotation by increasing the velocity of the fluid at the point of admission to the vane-passage. It will therefore be seen that by obtaining the maximum available velocity of the fluid-jet before delivering it to the first set of vanes a greater percentage of the total energy will be abstracted in the first set of vanes and the number of subsequent sets of vanes required to abstract the remaining energy will be lessened, thus greatly simplifying the apparatus.

In order to secure the most efficient results, other conditions in the construction and operation of the turbine should be provided for. These will now be referred to.

The velocity or *vis viva* of the fluid-jet being abstracted fractionally by the two or more sets of rotating vanes, it follows that the velocity of the fluid-jet will be reduced by each passage through the rotating vanes. To accommodate the flow of the fluid-jet at the reduced velocities, so that its flow will not be impeded and no choking will occur, the working passage through the turbine must be enlarged in the direction of flow directly as the velocity is decreased. It is not necessary for this purpose that the passage through any set of the movable vanes should be expanded between the receiving and discharging ends of the passage, since the speed of the jet through a movable passage relative to the walls of the passage is maintained, though the actual velocity of the jet is reduced by the movement of the passage itself; but in discharging into a stationary passage a larger cross-sectional area should be provided, and this cross-sectional area should be maintained throughout the length of the stationary passage, so that the fluid-jet in being presented to the second set of movable vanes shall occupy a cross-section which is as many times greater than the cross-section of the delivery end of the nozzle as the velocity is times less. It also follows that the second set of movable vanes must afford this additional cross-section. The retardation of the fluid-jet by surface friction and a similar effect produced by eddy currents, which tend to reconvert the velocity into pressure, are also important factors in the construction of my turbine. Both of these actions are referred to hereinafter as "frictional retardation." Unless the former of these actions is compensated for and the latter is prevented a considerable loss in efficiency will occur in the operation of the turbine. To provide against this I maintain a difference in pressure between the delivery end of the nozzle and the exhaust, which pressure is converted into velocity in the working passage of the turbine by a further expansion of that passage in the direction of flow beyond that required to take care of the reduced velocity of the fluid-jet due to the mechanical action of the moving vanes. This expansion or enlargement of the working passage is preferably made throughout the length of the passage in both its movable and stationary portions, so as to produce a gradual expansion and hence a constant draft or pull upon the fluid-jet in the direction of the exhaust. The amount of initial pressure in the jet that is necessary to overcome this frictional retardation is best ascertained empirically, depending as it does upon the shape and size of the passage and the character of the surface exposed to the fluid-jet. Free flow through the passage without choking and with the highest attainable velocity at the entrance to each movable part is the desideratum in practice. With reference to all conditions, the passages through which the fluid flows are enlarged in proportion to the decrease of the fluid's velocity, whether caused by the velocity-extracting vanes or by the retardation of flow caused by frictional consumption of energy or by the redevelopment of heterogeneous vibration in the fluid. I prefer to convert into *vis viva* before the fluid is delivered to the first set of vanes all the energy contained in the fluid, except the amount which will compensate for the frictional loss and to counteract the tendency of the *vis viva* to be reconverted into pressure. I prefer to construct the expanding-nozzle so as to convert all the pressure into velocity, except sufficient pressure to maintain the flow of the fluid-jet in the most efficient manner; but a higher pressure than this may be employed at the entrance to the working passage of the turbine, in which case the discharging end of the delivery-nozzle will not be enlarged relatively to so great an extent, and a portion of the useful or available pressure will be converted into velocity after the delivery of the fluid-jet to the first set of vanes and while it is passing through the working passage of the apparatus, and the working passage beyond the delivery-nozzle will be expanded with reference to the conversion of this additional pressure into *vis viva* beyond the increased cross-section required to compensate for the decreased velocity produced by the movable vanes and beyond that required by frictional retardation. I prefer to construct the working passage of my compound turbine, *i. e.*, the passages beyond the delivery-nozzle, with two or more complete circular ranges of curved vanes, mounted upon one or more drums, wheels, or disks, the succeeding sets of vanes being connected by stationary passages, which are likewise curved and change the direction of flow of the jet, so as to deliver it to the succeeding sets of vanes at the same angle as it is delivered to the first set of vanes by the delivery-nozzle; but a single set of rotating vanes may be employed and the intermediate stationary passages may serve to deliver the jet successively to different portions of the vanes of this single set, as will be understood, or the passages beyond the delivery-nozzle may be composed entirely of movable vanes mounted upon oppositely-rotating bodies. The movable vanes are preferably constructed on a curve from their receiving ends to a point beyond their centers, from which point they extend at a lesser angle to their discharging ends in order to obtain as much reactive effect as possible. This form of vane also permits the fluid-jet, which is somewhat compressed by the impact against the curved faces of the vanes, to expand to its normal volume in passing along the extended discharging ends of the vanes and before reaching the points of clearance at the discharging ends of the vanes. The vanes are also preferably given a progressively greater angle at their receiving ends in successive sets of vanes to compensate for the decreased velocity of the jet relative to the speed of the vanes. The intermediate stationary passages which are employed in the preferred form of my apparatus are wide enough to receive the jet from all the vanes from which it is discharged, including those into which the jet may "spill" at the preceding point of delivery, and where the stationary passages are single or undivided they are preferably spread out sidewise at their receiving ends in the direction of the movement of the vanes to collect the divided portions of the fluid-jet, and these sides converge toward the center of the passage, so as to reduce the jet to its normal volume. The successive passages are likewise set progressively farther forward at their forward sides, so as to compensate for the additional "lead" in the discharge from the movable vanes produced by the increased ratio of the speed of the vanes to that of the jet. I prefer, however, to make the stationary passages wide enough for all purposes, and to divide them into a number of parts by means of vanes which have a shape similar to the movable vanes. In this way eddy-currents, due to a wide passage, are prevented, the fluid-jet occupying only such parts of the divided stationary passages as it may be delivered to by the preceding set of movable vanes. In the case of the divided stationary passages the succeeding passages may be made progressively of additional width.

It is the design of my turbine to employ at the delivery end of the nozzle and in the working passage a "jet" of steam or other elastic fluid, *i. e.*, a solid stream of the fluid having an oblong form in cross-section, whose thickness bears a considerable proportion to its width, so that its cross-sectional area will be large compared with its perimeter, as distinguished from an annular film of elastic fluid whose cross-sectional area is small compared with its perimeter. By this means the surface friction is greatly reduced and the efficiency is largely increased.

In the accompanying drawings, forming part hereof, Figure 1 is a vertical section taken through the central line of one set of passages of my preferred form of apparatus. Fig. 2 is a similar section, on a smaller scale, of an apparatus of the same kind designed to work with a different exhaust-pressure. Fig. 3 is a section on the same scale as Fig. 1, in partial elevation, of an apparatus working under the same conditions as the apparatus of Fig. 1, but having the expansion take place in the working part of the apparatus wholly in the stationary intermediate passage. Fig. 4 is a horizontal section through the delivery-nozzle, stationary intermediate passage, and exhaust-opening of the apparatus of Figs. 1, 2, and 3, the rims surrounding the two circular ranges of vanes being partly broken away to disclose the vanes, which are developed in a horizontal plane. Fig. 5 is a view in diagram illustrating a modified form of the delivery-nozzle. Fig. 6 is a sectional view similar to Fig. 1 and on the same scale, showing a form of apparatus employing four sets of movable vanes. Fig. 7 is a sectional view similar to Fig. 4 of the apparatus of Fig. 6. Fig. 8 is a vertical section of an apparatus having four sets of movable vanes mounted on separate wheels and connected by single or undivided stationary passages, the view being on one-half the scale of Fig. 6. Fig. 9 is a sectional view, in partial elevation, of an apparatus corresponding to Fig. 8, but proportioned so that a part of the available pressure is converted into velocity within the working part of the apparatus. Fig. 10 is a sectional view taken horizontally through the passages of the apparatus of Figs. 8 and 9; and Fig. 11 is a vertical section on the same scale as Fig. 1, showing an apparatus with oppositely-rotating sets of movable vanes and without stationary intermediate passages.

Referring particularly to Figs. 1 to 4, A is a shaft mounted in suitable bearings *a* and passing centrally through a fluid-tight case B. Within the case B, the shaft A carries the wheel or drum C, upon the rim of which are mounted two sets D D′ of curved vanes, each set forming a complete circular range of such vanes extending entirely around the rim of the drum. The vanes of each set are covered at their outer sides by hoops or rings *b*. The vanes form horizontal passages parallel in one plane with the shaft A across the portions of the rim of the drum which they occupy, these passages being closed at their bottoms by the rim of the drum, and at their tops by the rings *b*.

E is the stationary intermediate passage, which is carried by the shell B and occupies the space between the sets of vanes D D′ in line with the delivery-nozzle and the exhaust-opening, this passage being closed at its bottom by the plate *c*.

F is a pipe extending from a steam-boiler or other source of elastic fluid under pressure and connected at its end with the expansion-nozzle G, which extends through the side of the shell B and approaches as closely as practicable the receiving ends of the first set of movable vanes D, leaving clearance enough, however, for free rotation. The receiving end or throat *d* of the expansion-nozzle G is smaller than its discharging end *e*, in order to cause an expansion and convert the pressure of the elastic fluid into *vis viva*.

H is the exhaust-opening, leading out of the shell B opposite the vanes D′, from which the fluid-jet is discharged. This exhaust-opening may be connected with a condenser or other means for producing less than atmospheric pressure, or it may open into the air.

To comprehend the design of this apparatus and its mode of operation, it is necessary to analyze what takes place at all points in the passage-way through which the jet of steam flows from the nozzle to the exhaust. The nozzle being set at the desired angle with reference to the plane of the wheel, the virtual angle at which the fluid enters the buckets or vanes depends upon the velocity of these vanes as compared with the actual velocity of the fluid and is the resultant of these two velocities. Since the velocity of the vanes bears a material ratio to that of the fluid, it follows that the virtual angle of entrance is somewhat greater than that of the nozzle, and therefore the receiving ends of the vanes should be set at this increased angle in order to avoid impingement of the jet on the rear faces of the vanes. The fluid having entered the movable passages, sufficient cross-section must by provided to convey it and discharge it freely on the other side. If we were dealing with a non-elastic fluid, such as water, no expansion or enlargement of the movable passages would be needed, because so long as the angle of the tails of the vanes or buckets is the same as or at least no less than the angle of the nozzle, the tails of the passages are necessarily mathematically capable of discharging the same quantity of fluid that is delivered by the nozzle, and hence no enlargement of these passages is necessary. With an elastic or expansion fluid the same is true merely as regards conducting the fluid; but in order to permit an expansion of the fluid in the movable passages themselves and thus maintain a drop in pressure between the ends of such passages, as I prefer to do, I provide an actual enlargement in these passages. This may be done either by diverging the top and bottom walls or by setting the tails of the buckets at a greater angle than that of the nozzle, thus providing a virtually-increased cross-section of passage at this end; but I prefer to obtain it in the former way, and the amount of such expansion will, of course, depend upon the respective pressures it is desirable to maintain at these points and the corresponding volumes. Owing to the fact that the vanes themselves have a velocity which is considerable compared with that of the fluid, the latter issues from the vanes at an angle which is materially greater than that of the tails, this angle being the resultant of the combined velocities of the fluid and vanes. In order therefore to receive the fluid as it enters the stationary passage at a reduced velocity, I set its receiving end E at an angle which is the same as that at which the fluid enters it, the opening being just sufficient to include all the fluid which escapes from the wheel, and the mere fact of setting the passage at this angle mathematically provides the increased cross-section necessary to convey the fluid at the reduced velocity. From this point to the delivery end of the stationary passage the cross-section measured at all points at right angles to the direction of flow should be at least preserved. In order to provide for friction and maintain the flow and prevent or diminish the formation of eddy-currents, I provide a gradually-increasing cross-section in the stationary passage, as well as in the movable passages, this increase in cross-section being adapted to permit the necessary expansion to maintain the desired drop in pressure between the terminals of the passage. This increase in cross-section may be secured in several ways. As shown in the drawings, it is obtained by increasing the depth or radial dimension of the passage; but it may, evidently, be obtained by increasing the width or circumferential dimension, or both. If the stationary passage be curved around so as to deliver the fluid to the second set of vanes at an angle equal to that of the nozzle, as is desirable, a considerable increase in the radial dimension is necessary to make up for the reduced width of the passage resulting from the reduced angle of its walls and to provide for the additional enlargement necessary to permit the desired expansion. The illustrations herein given in the drawings are not intended to show the exact relative proportions of the parts, but merely show the general plan of operation. In the second set of movable passages the conditions are similar to those existing in the first set, the angle of the receiving ends of the vanes in this case being, however, somewhat greater, owing to the fact that the velocity with which the jet strikes the vanes is less, while the velocity of the vanes remains the same, (if these two sets of vanes have the same diameter or are moving at the same rate of speed,) so that the virtual angle of entrance into these passages is somewhat greater than in the first set. For a like reason the angle of issue from the second set into the second stationary passage will be somewhat greater than in the first set, and hence I set the receiving end E' of this passage at an increased angle. The receiving ends of the subsequent vanes, both movable and stationary, are likewise set at progressively-increasing angles, provision being made in this way for the reduction in the jet's velocity, which takes place in its flow through each set of movable passages. In each of the subsequent movable and stationary passages, as in the first, I provide the proper expansion in each portion, so as to maintain the desired pressure difference, the pressure gradually declining through the system and issuing from the last set of vanes with little or no residual pressure above that of the exhaust. It is preferable that the desired expansion should take place in both the movable vanes D and the stationary passage E, as illustrated in Figs. 1 and 2, so as to obtain a gradual expansion throughout the system, whether composed of two or more sets of movable vanes, but the requisite expansion may entirely or largely take place in the stationary passages E, as illustrated in Fig. 3, in which case the passages formed by the vanes D have parallel top and bottom sides, the essential point being that the cross-sectional area of the stationary passage E shall be sufficiently larger than the cross-sectional area at the discharging end *e* of the nozzle G to accomplish the result stated. The second set of vanes D' may also flare outwardly, or expand, as shown in Figs. 1 and 2, or they may have parallel sides, as shown in Fig. 3. I prefer, however, to have the expansion take place also in the movable passage, so as to provide sufficient expansion to take care of the frictional retardation in the movable passage itself.

In Fig. 5 is illustrated a construction in which the side walls of the nozzle are parallel; but the proper relation is maintained between the receiving and discharging ends *d e* to give the desired conversion of pressure into velocity. This form, while not so effective, gives approximate results with low ratios of expansion.

The discharging end e of the nozzle G may be sufficiently larger than the receiving end d to convert the pressure wholly into velocity, except an amount of pressure required to supply the frictional consumption of energy in the working part of the apparatus and to maintain the flow without developing eddy-currents or what may be termed "heterogeneous motion," or the ratio of expansion between the receiving and discharging ends of the nozzle may be lessened, so as to deliver the fluid to the first set of vanes at a pressure higher than that thus required. I prefer the former plan. It is desirable that the steam should strike the first set of vanes at as high a velocity as possible, because the greater the velocity at which it strikes the first set of buckets the greater the work done in the first set, and hence fewer successive passages are required to abstract all the energy it contains. By using an expansion-nozzle properly proportioned the maximum velocity may be obtained that is possible while still leaving sufficient pressure above that of the exhaust to cause the flow in the most efficient manner, thus securing the highest aggregate amount of *vis viva* at the various points of action and therefore the greatest effect possible from the fluid.

The difference in pressure that it is desirable to maintain between the terminals of the working passages of the machine depends upon the length, size, and general proportions of and the number of bends in the working passages and the character of the surfaces over which the fluid sweeps, and can only be determined by trial in each particular style or size of machine. In large machines this quantity will be less than in small machines, and in machines highly compounded it will be greater than in those compounded to a less degree. Let us assume, for illustration, the particular case where the boiler-pressure is one hundred and fifty pounds and the exhaust two pounds per square inch, (corresponding to about twenty-six inches vacuum,) these pressures being absolute and not by gage. Let us assume that a pressure of, say, five pounds above that of the exhaust is necessary to overcome friction and to cause the flow of the fluid through the working passage in the most efficient manner, that is, so as to obtain the highest aggregate velocities possible at the several points of action, this figure being used merely as an illustration. Under these circumstances the pressure at the delivery end of the nozzle should therefore be five pounds above that of the exhaust, or seven pounds, (absolute.) The true proportions of the nozzle can then be determined as follows: The small or receiving end is made just large enough to permit the required quantity of steam to flow to develop the power the turbine is designed for. This can be ascertained empirically, or it may be calculated approximately by taking the pressure at this point, which will be approximately fifty-eight per cent. of the boiler-pressure (measured above the exhaust) and calculating the velocity developed at this point due to the drop in pressure. This velocity can be calculated by taking the heat-units or equivalent work in foot-pounds performed by the steam in thus expanding and calculating the equivalent velocity of the fluid, assuming that all the energy in foot-pounds becomes converted into an equivalent amount of *vis viva*. Under the conditions assumed the pressure will be about ninety pounds and the velocity between thirteen hundred and fourteen hundred feet per second at the receiving or small end of the nozzle. Knowing the velocity and the density of the fluid, the size or cross-section of the receiving end of the nozzle necessary to conduct a given amount is easily determined. In a similar way the size or cross-section of the large or delivery end of the nozzle can be computed by first ascertaining the velocity of flow developed by dropping to the required pressure at this point, and then, knowing the density from the pressure, furnishing just sufficient cross-section at this point to conduct the same quantity of fluid as passes through the small end, allowance being made in the case of a saturated vapor, such as steam, for the condensation and consequent diminution in volume, which takes place during the given expansion. In the case assumed as an example a drop in pressure from one hundred and fifty pounds to seven pounds would develop a velocity of about three thousand one hundred feet per second, and the condensation would be about fifteen per cent., so that the volume would be less by fifteen per cent. than what it would be were there no condensation. In the case assumed the cross-section of the larger end of the nozzle should be about five times as great as that of the smaller end, and this is the general proportion which it is intended to illustrate in Figs. 1 and 3, taken in connection with Fig. 4.

The apparatus illustrated in Fig. 2, taken in connection with Fig. 4, is intended to represent a machine designed to work between a boiler-pressure of one hundred and fifty pounds and atmospheric pressure; in other words, a non-condensing machine. In this case if the loss by friction represents, say, five pounds of the whole available pressure, as before, the pressure at the delivery end of the nozzle will be about twenty pounds, the speed of the jet will be about two thousand six hundred feet per second, and the discharging end of the nozzle will be about twice as large in cross-section as the receiving end. The excess of pressure at the discharging end of the nozzle over the atmospheric pressure at the exhaust will be that required to overcome friction in the working passages of the apparatus and maintain the flow therethrough in the most effective manner, as before. It is highly important that the large end of the nozzle be no larger than just sufficient to conduct the required volume at the pressure and velocity designed to exist at that point, for otherwise the passage would be too large, and there would be a less velocity at this point than should exist there, resulting in a loss of power. It must be borne in mind that the effect obtained from the fluid is as the square of its velocity at the time it is acting on the vanes, and that therefore a slight change in velocity causes a very considerable loss of power. For this reason it is highly important that the correct and highest available velocities be obtained at all points where the fluid acts upon the vanes. It is better to ascertain the exact proportions between the receiving and discharging ends of the delivery-nozzle by careful trial, the result being to some extent influenced by friction in the nozzle itself, the best plan being to vary the size of the large end of the nozzle, keeping the small end constant until a pressure is obtained at this point which is only just sufficient to overcome friction and tendency to reconversion into pressure in the working passage beyond, and at the same time the cross-section is only just sufficient to conduct the quantity of steam flowing at the particular pressure and velocity existing at this point. After passing through the first set of vanes or movable passages the steam or fluid emerges with a reduced velocity. In the case of two movable passages let us assume that forty per cent. of the velocity of the fluid is abstracted at each passage, leaving only twenty per cent. of the original velocity as it enters the exhaust, corresponding to four per cent. of the original energy. Since the energy is proportional to the square of the velocity, it is evident that the energy extracted by the first movable passage will be sixty-four per cent., while that extracted by the second movable passage will be thirty-two per cent., of the total original energy. On emerging from the first set of movable vanes and as the steam-jet enters the second set of movable vanes from the stationary passage its velocity will therefore have been reduced by forty per cent., and it will now have sixty per cent. of its original velocity. Since the stationary passage is increased in size inversely as the velocity of the steam-jet, it follows that under the assumed conditions the stationary passage E will, from this cause, be required to be made 100/60 or 1.66 times larger in cross-section than the delivery end *e* of the nozzle G. At the same time provision should be made for expanding the passages to a further extent, as already explained, to cause the desired decline in pressure to overcome surface friction and the tendency to reconvert velocity into pressure by eddy-currents, this decline in pressure being, under certain conditions, a considerable portion of the total drop in pressure in the working passage of the turbine. The entire increase in the cross-sectional area at the point *f* over the point *e* is therefore the product of these two ratios, and this is the proportion intended to be illustrated in Figs. 1, 2, and 3. By observing this relation of the parts an economical conversion of the pressure into *vis viva* and of the *vis viva* into mechanical power will be obtained, while the speed of the movable vanes will be only a fraction of the speed of the fluid-jet.

The proper decline in pressure of the steam in passing through the turbine depends upon so many conditions that it is best in every case to determine it by experiment. It will be greater in traversing the first set of vanes than in subsequent ones, owing to its higher velocity, and likewise it will be greater in traversing the first set of stationary vanes than in subsequent ones for the same reason. Allowance should also be made for the fact that the steam in sweeping against the concave vanes is compressed to a greater or less degree by its own centrifugal force, so that unless when it leaves each set of vanes it has reëxpanded to its normal volume its density at these points will be less than would have been the case had there been no centrifugal compression. This compression may to a large extent be reduced by means of the straight or slightly-curved tails, (which I have shown especially in the case of the intermediate fixed vanes,) which allow the steam to flow for a certain length in a straight or slightly-curved course, thus permitting it to expand again and assume more nearly its normal volume before delivery to the next set of vanes. By these means I am also enabled to reconvert the energy or work expended in centrifugal compression into velocity or partially into velocity again, thus recovering a large part of this energy and causing the fluid to impinge upon the second or successive sets of vanes with additional velocity.

Owing to the various conditions which affect the actual volume at the various points in the apparatus, the best plan is to ascertain the proper proportion of the various points in the passages by trial, the essential features of economical working being that the successive cross-sections of the passages should be just large enough to conduct the fluid without choking it or backing up the pressure at that point, and so as to obtain the highest velocities practically obtainable at the various points of action upon the blades, (that is, so as to obtain the greatest sum-total of all the actions upon the successive vanes,) and yet no larger than is necessary for this purpose; otherwise there will be too low a velocity and a corresponding loss of energy. If more than two sets of movable vanes be employed, the principle of operation and construction is the same, the delivery-nozzle having its receiving end of the precise size required to pass the desired quantity of the fluid and its discharging end of a size as many times larger as may be required to expand the fluid down to the pressure required to overcome the frictional retardation in the working passages of the turbine and give the best result, while the delivery ends of the succeeding stationary intermediate passages will have their size increased in proportion as the velocity of the fluid-jet has been reduced and as its volume has been increased by the decline in pressure.

In Figs. 6 and 7 is represented a turbine having four sets of movable vanes D D′ D^2 D^3 and three stationary intermediate passages E E′ E^2 constructed and operating on this principle. In this case, and assuming that the apparatus is designed to work between the terminal pressures before taken for illustration, *i. e.*, a boiler-pressure of one hundred and fifty pounds and an exhaust-pressure of two pounds per square inch, (absolute,) the delivery-nozzle G will be proportioned in the manner described, so as to expand the steam down to a given pressure, say ten pounds at its discharging end, the remaining pressure above the pressure of the exhaust being that assumed to maintain the flow in the working passages. If the steam-jet enters the exhaust with twenty per cent. of the original velocity, each of the four sets of movable vanes will abstract twenty per cent. of the velocity and will in their order absorb, respectively, thirty-six, twenty-eight, twenty, and twelve per cent. of the energy represented by the original velocity, while four per cent. of the energy will pass off as residual velocity in the exhaust. The decline in pressure in the working passages will be such that at the discharge ends f, f', and f^2 of the stationary intermediate passages the pressures will be less and less, declining at a certain rate, which should be found by experiment in each particular type and size of machine, and owing both to reduced velocity and to increased volume of the fluid-jet arising from reduced density the cross-sectional areas at the points f, f', and f^2 will be correspondingly increased over the cross-sectional area of the discharge end e of the nozzle and over each other successively. The expansion in the working passages is shown in full lines as taking place in both the movable and stationary passages, while by the dotted lines this expansion is shown as taking place wholly in the stationary passages. I prefer the former plan.

In Fig. 8 (taken in connection with Fig. 10) is shown an apparatus designed to have the same proportions of the expanding-nozzle and the working passages and to operate under the same conditions as the apparatus of Figs. 6 and 7. It has the peculiarity in construction of having the different sets of movable vanes mounted upon separate wheels. It also differs in the construction of the stationary intermediate passages, which will be presently described.

In Fig. 9 (also taken in connection with Fig. 10) is shown an apparatus in which the two operations of converting the pressure into *vis viva* and *vis viva* into mechanical power overlap to a greater extent, although the useful or available pressure is still largely, although not wholly, converted into *vis viva* by the expansion delivery-nozzle, which delivers the fluid-jet to the first set of movable vanes at a pressure greater than is required to overcome frictional retardation in the working passages of the apparatus. In other words, the apparatus of Fig. 9 is designed to convert into *vis viva* in its working passages a part of the pressure which, in the constructions before described, is converted into *vis viva* by the expansion-nozzle. Taking the illustration of terminal pressures of one hundred and fifty pounds and two pounds per square inch (absolute) and assuming, as we did for the apparatus of Fig. 8, that a pressure of eight pounds above the exhaust-pressure is required to overcome frictional retardation in the working passages, the apparatus of Fig. 9 is designed to have a pressure of, say, twenty pounds at the delivery end e of the nozzle G, and the remaining expansion required to convert the balance of the useful pressure (ten pounds) into *vis viva* takes place between the delivery end e of the nozzle G and the delivery end f^2 of the stationary passage E^2. To accomplish this, the delivery end e of the nozzle G and the delivery ends f, f', and f^2 of the stationary passages E, E′, and E^2 are given cross-sectional areas, increasing in size from one end of the apparatus to the other, the increase at each point being just sufficient to take care of the increased volume of the fluid at each point and also to allow for the diminished velocity resulting from its passage through the previous set of vanes. The expansion for converting the balance of the useful pressure into *vis viva*, which takes place in the working passage of the apparatus of Fig. 9, may take place uniformly in both the movable and stationary passages, as illustrated, or it may take place wholly in either the stationary or movable passages or in a part of the working passages, either movable or stationary, or both, but I prefer to enlarge both the movable and stationary passages uniformly. At the same time that the expansion of the working passages to convert the balance of the useful pressure into *vis viva* is provided, those passages are expanded to compensate for reduced velocity and for frictional retardation. The proper proportions will accomplish this combined result. The proportions and pressures I have given are merely for purposes of general illustration to show the principles involved in the construction and operation of my apparatus.

Having thus described the principle of operation and construction of my compound turbine, I will now refer to some details of construction which have not been as yet specifically described. The vanes g of each set of movable vanes are curved from their receiving ends to beyond their centers, and from thence to their discharging ends they are straight or slightly curved or set at a lesser angle, as shown in Figs. 4, 7, and 10, so as to obtain as much effect as possible from the fluid before it leaves the vanes. In Fig. 10 single or undivided stationary passages are shown. These are made wide enough at their receiving ends to receive the fluid-jet from all the movable vanes from which it may be discharged, (including the additional width made necessary by the spill of the fluid-jet,) and the receiving end of each stationary passage is given a lead, that is, set with a certain advance, if necessary, over the discharging end of the nozzle or previous stationary passage to properly receive the discharge from the movable vanes, it being remembered that, owing to the speed of the movable vanes themselves, the jet issues from them at a point slightly ahead of the point at which it is received. The amount of this lead should increase at each succeeding set of vanes, owing to the progressively-diminishing velocity of the fluid compared with that of the buckets, as indicated in the drawings. The diverging top and bottom walls of these passages serve not only to maintain the cross-sectional area, which would be contracted toward the discharging ends of the passages if such top and bottom walls were parallel, but also to give the necessary expansion to maintain the desired drop in pressure between the ends of the passages. Instead of using single or undivided stationary passages, as shown in Fig. 10, I prefer, under certain conditions, to use divided stationary passages, as shown in Figs. 4 and 7. These passages are made of sufficient width to receive the entire discharge from the movable vanes, when running at the normal speed, and may be made of additional width in succeeding passages to give the necessary lead to provide for any rate of speed. These passages are divided each into a number of narrower passages by vanes *h*, which have curved receiving ends and straight discharging ends, as have the movable vanes, and also have their receiving ends arranged progressively at a greater angle in the vanes of succeeding passages, as have the movable vanes in successive sets, in order to receive the jet properly and furnish the required cross-section. As in the case of single fixed passages, the angles at which their receiving ends are progressively set will determine their respective cross-sections, and by making these angles the same as the angles of issues from the moving vanes the requisite cross-section will be provided.

Besides the forms of apparatus which have been described, many features of my invention are involved in the construction and operation of such an apparatus as that shown in Fig. 11, in which only movable vanes are employed in the working part of the apparatus, the sets of vanes being alternately mounted upon oppositely-rotating disks and delivering the fluid-jet from one set of movable vanes directly to another set without the interposition of stationary passages. It will be understood that the movable vanes have all the characteristics of the movable vanes already described, but that those on one disk are set or curved oppositely to those on the other disk. In the apparatus illustrated in Fig. 11 the expanding-nozzle is designed to convert all or the larger portion of the useful pressure into *vis viva*, although the overlapping of the operations of converting pressure into *vis viva* and *vis viva* into mechanical power can likewise be employed in an apparatus of this kind.

It will be observed that in all forms of the apparatus the steam or other elastic fluid is delivered to the working passages practically in the form of a solid stream or jet whose cross-sectional area is large compared with its perimeter, and that this jet acts at one time only on a small section of the vanes of a circular range. This arrangement has a great practical advantage over one wherein the fluid is delivered in the form of an annular film simultaneously to all the vanes of a complete circular range. In the latter arrangement, in order to reduce the total cross-section of the passage to that required and still have the proper velocity, the depth of the vanes has to be exceedingly small and the surface exposed to the flowing fluid is very large compared with the cross-sectional area of the passages. The result is great loss by surface friction and churning action, and in addition the clearance space is so extended that it becomes large in proportion to the area of the passages.

By the expression "frictional retardation" I mean to include not only the actual loss of energy by friction in the passages, but also the loss of energy in available form due to the reconversion of *vis viva* into pressure or heterogeneous vibration.

By the expression "expansion-nozzle" I mean any nozzle or delivery passage having an enlarged cross-sectional area toward its delivery end considered with reference to its narrowest part.

By the expression "working passage" I mean that portion of the apparatus through which the steam flows, beginning at the delivery end of the expansion-nozzle and ending at the exhaust, whether composed of vane passages or of vane passages and fixed passages.

It should be understood that the use of two or more sets of circular ranges of vanes through which the fluid passes in succession and of one or more circular ranges of vanes through which the fluid passes two or more times in succession are equivalent constructions and are intended to be included herein.

What I claim is—

1. In an elastic-fluid turbine, the combination with an expansion-nozzle, of movable vanes, and means for causing the fluid to act upon such vanes two or more times in suc-

cession, whereby the high velocity developed in the fluid by expansion in the nozzle is fractionally abstracted by the movable vanes, substantially as set forth.

2. In an elastic-fluid turbine, the combination with an expansion-nozzle, of two or more sets of rotating vanes, and one or more stationary intermediate passages connecting the sets of vanes, for causing the fluid to act on the vanes successively, whereby a high velocity is developed in the fluid by expansion in the nozzle and is fractionally abstracted by the rotating vanes, substantially as set forth.

3. In an elastic-fluid turbine, the combination with an expansion-nozzle, of a working passage expanding in the direction of the flow of the fluid, substantially as set forth.

4. In an elastic-fluid turbine, the combination with an expansion-nozzle, of a working passage comprising movable vanes and means for causing the fluid to act upon such vanes two or more times in succession, said working passage expanding in the direction of the flow of the fluid, substantially as set forth.

5. In an elastic-fluid turbine comprising a nozzle and a working passage comprising movable vanes to which the fluid is delivered two or more times in succession, a gradually enlarging or expanding passage-way for the fluid, part of the enlargement taking place in the nozzle and part in the working passage, substantially as set forth.

6. In an elastic-fluid turbine, the combination with an expansion-nozzle delivering the fluid at a pressure above that of the exhaust, of a working passage expanding to such an extent that the pressure at the end of the working passage is the same as the pressure in the exhaust, substantially as set forth.

7. In an elastic-fluid turbine, the combination with an expansion-nozzle adapted to convert the pressure of the fluid into velocity while retaining a pressure in the fluid above that of the exhaust, of an expanding working passage leading through a succession of revolving vanes, such passage having increasing cross-sectional areas sufficient only to conduct the required volume of the fluid at the maximum attainable velocity and yet not so contracted as to impede the flow, whereby the fluid is delivered to each successive set of vanes at as high a velocity as practicable, substantially as set forth.

8. In an elastic-fluid turbine, the combination with an expansion-nozzle adapted to convert the pressure of the fluid into velocity while retaining a pressure in the fluid above that of the exhaust, of a working passage enlarged or expanded in the direction of flow of the fluid, such enlargement being sufficient to convert the remaining available pressure of the fluid into velocity before delivering it to the exhaust, substantially as set forth.

9. In an elastic-fluid turbine, the combination with an expansion-nozzle adapted to convert the pressure of the fluid into velocity while retaining a pressure in the fluid above that of the exhaust, of a working passage comprising two or more sets of rotating vanes and one or more intermediate stationary passages, such working passage expanding in the direction of the flow of the fluid so as to deliver the fluid to each set of rotating vanes at the maximum velocities obtainable at each passage therethrough, substantially as set forth.

10. In an elastic-fluid turbine, the combination of an expansion delivery-nozzle placed obliquely, with two or more sets of rotating passages and one or more intermediate stationary passages, each of said passages having curved side walls having a less angle at the discharging end than at the receiving end of the passage, and with top and bottom walls diverging toward the discharging end of the passage to such an extent as to provide at the discharging end of each movable and stationary passage a cross-section greater than that at the discharging end of the preceding stationary or movable passage, substantially as set forth.

11. In an elastic-fluid-jet turbine, the combination of two or more sets of rotating vanes and one or more connecting stationary intermediate passages for delivering the fluid-jet to the successive sets of rotating vanes after the first, such stationary intermediate passages being set at their receiving ends at the same angle as that at which the fluid-jet is discharged from the rotating-vane passage, whereby the cross-sectional area provided for the jet in the stationary passage at its receiving end is substantially the same as the cross-sectional area of the fluid-jet as it is discharged from the respective rotating passages, substantially as set forth.

12. In an elastic-fluid turbine, the combination with two or more sets of rotating vanes, of a nozzle delivering a fluid-jet continuously to a portion of the vanes of one set, and one or more intermediate stationary passages connecting the rotating vanes, said intermediate stationary passage or passages having a lead or leads proportional to the diminished velocity of the fluid-jet, substantially as set forth.

This specification signed and witnessed this 8th day of January, 1896.

CHARLES G. CURTIS.

Witnesses:

EUGENE CONRAN,
JOHN R. TAYLOR.