April 19, 1960 A. M. CADDELL 2,933,287

MULTIPLE STAGE TURBINE UNIT

Filed May 28, 1956

FIG. 1

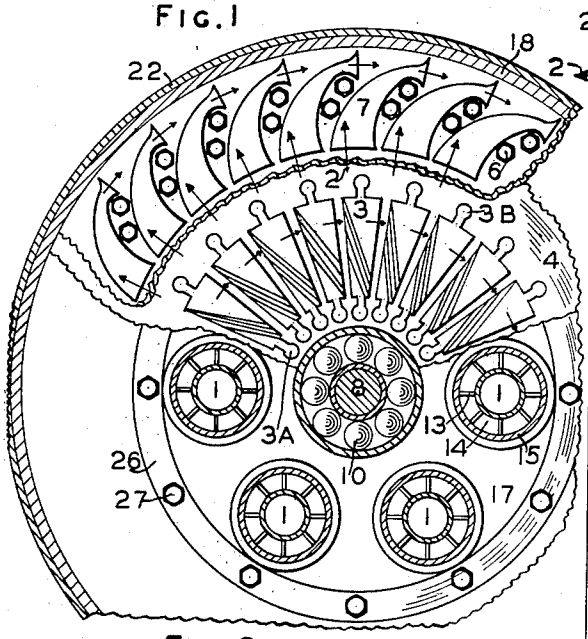


FIG. 2

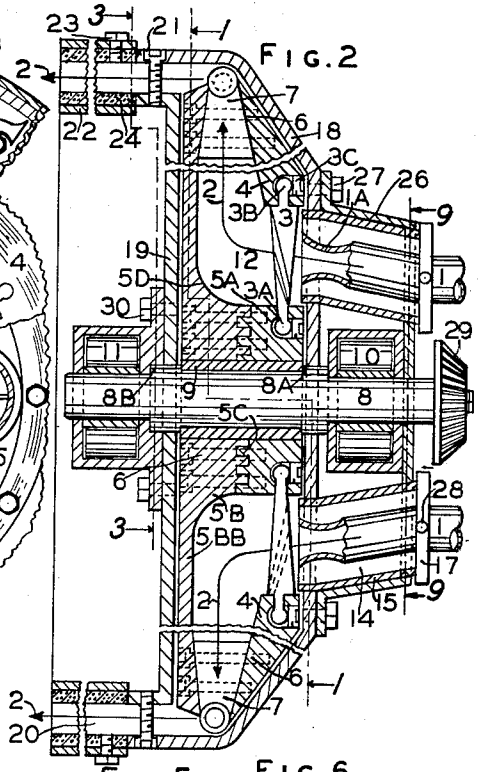


FIG. 3

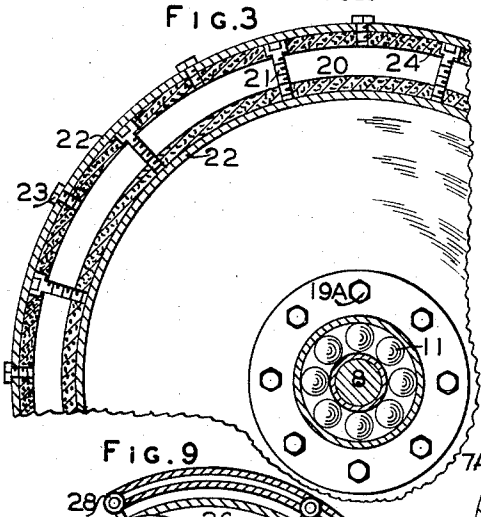


FIG. 4

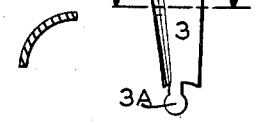


FIG. 5

FIG. 6

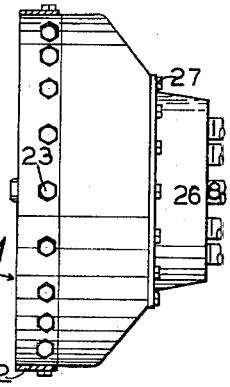


FIG. 7

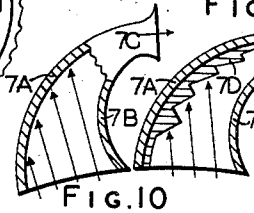


FIG. 8

FIG. 9

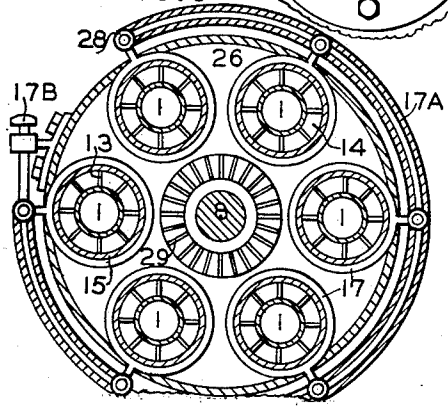


FIG. 10

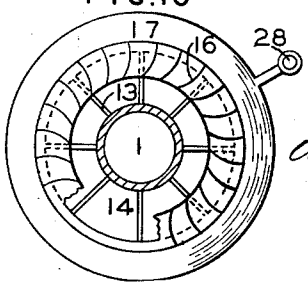


FIG. 11



INVENTOR.

Alfred M. Caddell

United States Patent Office 2,933,287 Patented Apr. 19, 1960

2,933,287

MULTIPLE STAGE TURBINE UNIT

Alfred M. Caddell, Philadelphia, Pa.

Application May 28, 1956, Serial No. 587,865

5 Claims. (Cl. 253—87)

This application is in part a continuation of applications entitled Mutually Assisting Powerplants in Combination, Serial No. 219,232, filed April 4, 1951, now abandoned, and Exhaust System With Power-Driven Evacuator, Serial No. 300,170, filed July 22, 1952, and now abandoned.

The turbine unit described herein differs in practically every respect from any other known engine. It, a single engine, is doubly driven by turbines—initially by expansion of a working fluid against turbine blades in its central intake area and reactively by subsequent expansion of the same fluid against turbine buckets at its peripheral discharge end. Action and reaction being equal and opposite, the sequential reactive drive effect, added to that of the impulse drive against the centrally located turbines, practically doubles the power output of the unit for the single passage of a high temperature fluid therethrough. Such compound energy application obviates the necessity of having to pass the fluid through a several-stage turbine engine to obtain an equivalent power output, thus reducing the bulk and weight otherwise necessary in a powerplant.

After an expansible fluid impinges against turbine blades positioned in the central section of the rotor's forward disc, it enters a radially disposed chamber formed between the forward and the rear discs and is thrown into reactive-drive-causing buckets mounted in the periphery of the rotor. Constant impingement against said blades and simultaneous impingement against said buckets causes, first, a powerful turning effort of the rotor; second, a strong reactive turning effort as the fluid strikes and is deflected off the inner ends of the buckets and, third, a still stronger turning effort as the fluid is discharged through a converging-diverging orifice of the buckets in a direction opposite to that of rotation to add to the initial drive effort instituted by the central turbine in the direction of rotation.

As will undoubtedly be understood, the working fluid, after its impingement against the blades, loses the amount of heat or expansion pressure that is converted into work. But inasmuch as the reaction buckets are mounted in the periphery of the rotor, the fluid acquires centrifugal force as it is being thrown radially within the chamber, and this centrifugal force, added to the prevailing expansion pressure of the fluid, compensates to a large extent for the loss of heat incurred in doing work.

Further, due to the thus revitalized expansion pressure striking and being deflected from the buckets' inner surfaces and then discharged at a high velocity at the extreme radius of the rotor where maximum leverage obtains, the advantage of said reactive effects will be more apparent. Obviously, the radius of the rotor of this invention has an important bearing upon the degree of energy conversion into power, for the greater the radius the greater the leverage and the greater the centrifugal pump effect imparted to the fluid stream.

As will undoubtedly be apparent, the object of employing this direct and reactive drive is to convert into work more of the energy than has been possible to accomplish by means of conventional impulse turbines, and do it at minimum cost in back pressure against the source of the fluid; which minimum cost is greatly influenced by the centrifugal pump effect brought into being by centrifugal force within the fluid within the rotor.

The source of the fluid to be employed in this engine may be the end product of any combustible process or mixture, the exhaust gas from a free piston engine or that from either a compression-ignition or a spark-ignition powerplant. Inasmuch as internal combustion engines are playing an ever-increasing role in the economy of the world, the exhaust energy from this latter source is suggested for conversion into power by the herein described multiple stage turbine unit.

Efficient use of combustion temperatures

Combustion of a hydro-carbon fuel in air, as in the combustion chamber of a piston engine, results in a flame temperature of between 4,000 and 4,500 degrees F., the actual temperature depending upon whether or not supercharging is used, or upon the compression ratio of the fuel-air mixture, or upon the volumetric efficiency in the cylinders. Regardless of the operational phase governing the situation, the greater the amount of air (oxygen) that can within safety limits be packed into a cylinder, the hotter will be the combustion temperature and the greater the expansion potential of the working fluid. In a piston engine the intake manifold pressure rarely exceeds 3½ atmospheres, or 51 inches of mercury.

Limited employment of temperatures

The utility of any heat engine depends almost entirely upon the range of temperature it can usefully employ. Due to the intermittency of combustion and the lack of centrifugal stresses, the piston engine can successfully use the upper ranges of this high temperature but, unlike a steam engine cannot, in a practical sense, use the exhaust of high-pressure cylinders to drive larger pistons in low-pressure cylinders.

On the other hand, the conventional turbine engine, due to the constancy of high temperature application and the centrifugal stress limitations imposed on the turbine blades under adverse (white hot) operating conditions, is restricted by present metallurgical limitations to approximately the 1500–1525 degree F. range. The logical step to take, therefore, is to combine the two forms of engine, safeguarded by adequate inlet temperature control at entry to the turbine, into one powerplant to convert into work the fall of heat from a high temperature source down to as near prevailing atmospheric temperature and pressure, as possible.

In industrial circles, considerable development work along this line has already been undertaken, but largely toward directing piston engine exhaust gas against a turbine to turn an impeller coupled to a blower. In the compression-ignition field, by employing turbo-supercharging, the power output of engines has been increased from 50 to 100 percent. But not much success has been attained by coupling a turbine to a gas engine's crankshaft for converting energy in the engine's exhaust into power. The great drawback has been the back pressure that is built up in the cylinders, which back pressure not only retards movement of the piston but creates excessive heat retention that destroys volumetric efficiency and results in piston seizing, valve warping and the like. Nevertheless, as a principle, mounting a turbine on the shaft of a piston engine is eminently sound practice, for the 50 percent of energy that now goes out

with the exhaust of a piston engine constitutes a challenge which, it is felt, may be overcome by the improved features offered by the herein described multiple turbine unit.

In the drawings:

Fig. 1 is a composite, frontal view of the multiple turbine unit taken on the line 1—1, Fig. 2, wherein is exposed the reaction-drive buckets in the rotor's periphery, the centrally positioned ring of turbine blades and the fabricated duct assemblies that convey the working fluid and as much cooling air as may be necessary to safeguard the blades from high fluid temperatures.

Fig. 2 is a side view showing in cross section the peripherally mounted reaction buckets, the centrally positioned turbine blades and the means for controlling the temperature of the fluid striking the blades.

Fig. 3 is a partial rear view of the engine taken on the line 3—3, Fig. 2.

Fig. 4 is a view representative of the curvature of the turbine blades, taken on the line 4—4, Fig. 5.

Fig. 5 is a frontal view of a single turbine blade showing at each end thereof the means employed to secure the blades in the forward wall of the rotor.

Fig. 6 is a half-size exterior side view of the engine.

Fig. 7 is a cross-sectional view of any one of the reaction-drive-causing buckets identified as 7, Figs. 1 and 2.

Fig. 8 is another view of the reaction-drive-causing bucket showing the installation of a staggered-step liner against the inner wall of the end that causes reaction drive.

Fig. 9 is a frontal view taken on the line 9—9, Fig. 2, wherein is depicted a means for controlling the extent of opening of the diaphragm employed to regulate the flow of air over the surface of the high-temperature fluid-conveying ducts.

Fig. 10 shows in enlarged form a fabricating duct, depicting in detail the diaphragm, or shutter, arrangement for regulating the flow of cooling air over the surface of the central fluid-conveying duct. In this view, the diaphragm is shown in a half-closed position and is broken away entirely at the lower end to expose a cooling air passageway.

Fig. 11 shows two views of the terminus of a reaction-drive-causing bucket, indicating that the diverging discharge end may have any degree of angle desired.

A high-temperature, high-pressure fluid is conveyed through the forward wall of the casing by ducts 1 from a source distant from the multiple turbine unit. It flows slantingly radial to strike a ring of turbine blades 3 which, besides serving as deflecting, driving surfaces, also serve as a means to secure the rotor's forward plate 4 to forward section 5A of the hub. When inserted in slots formed in said hub section and in slots formed in the innermost ends of forward plate 4, extrustions 3A and 3B respectively accomplish that purpose. small screw inserts 3C may be used to lock these extrusions in position to prevent misalignment of the blades receiving the attack of the working fluid.

Plate 4 inclines inwardly from its juncture with turbine blades 3 and also tapers radially to lessen the weight of the rotor. Rear disc 5BB which is a continuation of hub section 5Bfi similarly inclines inwardly as it approaches its periphery and has a progressively increasing cross section near said periphery. As will be noted, these discs are formed to create a wedge-shaped area therebetween to secure the reactive-drive-causing buckets, and are held together by a plurality of screwbolts 6, shown in dotted outline. Buckets 7 are formed to fit this wedge-shaped area and when installed between said discs become an integral part of the rotor capable of withstanding high centrifugal stresses.

Hub sections 5A and 5B are formed to mate with each other as per spline formation 5C, after which the integrated hub is keyed to shaft 8 by keys 9, shaft 8 having a greater diametered section where it passes through said hub to accommodate said keys.

Shaft 8 is mounted between radial thrust roller bearing assembly 10 at its forward end and 11 at its rear. The housing of assembly 10 abuts shoulder 8A of said shaft and the housing of assembly 11 abuts shoulder 8B, thus maintaining the rotor in its prescribed functioning alignment. Bearing assembly 10 may be shrunk-fitted on shaft 8 while the housing of bearing 11 is bolted to rear wall 19 of the casing by means of screw bolts 30.

The fabricated duct assemblies are comprised of duct 1, spacer ribs 13 which form a plurality of air passageways 14 which, in turn, are encompassed by housing 15. This housing is encompassed by an annular casing 17 which is a repository for a camera-type diaphragm, or shutter, 16 as shown in Fig. 10. This shutter—there is one for each fabricated duct assembly—may be opened to any desired extent to allow cooling air from atmosphere to flow convectively over the surface of the fluid-conveying duct. Opening of this shutter means may be manually controlled, or it may be actuated by a thermal switch. A manual control means is indicated by 17B, Fig. 9, which shows a flexible push-pull cable operating within a sheath 17A.

Fluid-conveying ducts 1 have a reduced area 1A formed adjacent their discharge ends to increase the velocity of the fluid just before it strikes turbine blades 3. This reduced area also produces a venturi action which entrains cooling air through passageways 14 formed between ducts 1, spacer ribs 13 and housing 15; which air, after absorbing heat from the fluid, mixes therewith, thus adding volume to it and, together, the fluid and air strike blades 3 of the centrally positioned turbine. These fabricated ducts are mounted at progressively increasing radii to assist in the flow of the fluid and air therethrough.

Upon deflecting inwardly off turbine blades 3, thereby causing primary rotation of the rotor, the combination of fluid and air is thrown axially from the blades and is also deflected radially upon striking sweeping radius 5D at the base of the rear disc. Continuing radially through chamber 12, the mixture of fluid and air strikes the underside of buckets 7 which comprise, in association with the rotor's forward and rear discs, a second turbine stage. Obviously, the greater the radius of this chamber the greater will be the centrifugal throw and consequently the pull effect on the fluid which, if the radius is great enough, may extend all the way back to the piston engine's cylinders and relieve back pressure therein. Obviously, too, as the fluid's heat is converted into work and as it expands further through the chamber it loses its expansion pressure progressively. On the other hand, by being thrown radially within the rotor the fluid acquires centrifugal force so that when it strikes end wall 7A of buckets 7, or step-liner 7D, shown in Figs. 7 and 8, which end walls are advancing in the direction of rotation, the fluid exerts powerful pressure against said walls, thereby causing internal reactive drive.

As will be seen by referring to their mounting in Fig. 2, the sides of these buckets converge toward their longitudinal centers. Also, by referring to Figs. 7 and 8, it will be observed that the end walls thereof are curved to attain certain objectives: One, end wall 7A curves openly at its bottom to allow unrestricted entrance of the centrifugally thrown fluid and air into the interior of the buckets; two, end wall 7B curves from its base throughout its radial length to its periphery, thereby bringing about concentration of the fluid and air at a point about three-quarters the radial length of the buckets. Together, the converging sides and the converging end walls convert compression pressure in the fluid into velocity pressure as it passes through converging-diverging orifice 7C of the buckets; after which the orifices

diverge to cause a change from velocity pressure back into compression pressure, the gas discharging in a direction opposite to that of rotation, thus generating further reactive drive.

In Fig. 8 a form of step-liner 7D is shown as an alternative to the smooth wall depicted in Fig. 7. The purpose of the step-liner is to create as much reactive drive effect as possible within the buckets, the step formation yielding more surface than the smoother wall and imparting to the bucket an intermittent impulse effect for translating more of the energy in the fluid into reactive drive.

In this connection, it is well known that intermittent application of a fluid under pressure results in more forceful impact against a surface than if the same fluid is uninterruptedly applied. Intermittent application of water under pressure against a scale will register greater poundage than will the non-interrupted application under the same pressure. This principle is used for flushing and dirt-dislodging purposes in mining operations. Intermittency is therefore simulated in the advancing wall of the buckets by interrupting the flow of the fluid, thereby intermittently increasing reactive pressure against the wall of the bucket advancing in the direction of rotation.

At this juncture, it will bear repeating that both the internal and external reactive drive effects brought into being by means of the construction and mounting of these buckets occur at the maximum leverage position of the rotor; which means maximum reactive drive efficiency.

The casing that surrounds the rotor is comprised of wall **18** in the forward side and wall **19** in the rear. Cross sectional views of each wall may be observed in Fig. 2. The forward casing wall conforms to the shape of the rotor, which it spatially encompasses, and turns at its periphery to form a right angular continuation that extends rearwardly any desired distance. Rear wall **19** also turns at a right angle at its periphery and continues rearwardly to form, in conjunction with the forward wall's angular continuation, an annular fluid discharge channel **20**, shown in Figs. 2 and 3. These angular continuations are held in spaced relation to each other by means of screwbolts **21** spaced circumferentially therearound.

Housing **26**, shown in cross section, Fig. 2, encompasses the several fabricated duct assemblies, a frontal view of which is shown in Fig. 9. This housing is removably secured to forward casing wall **18** by means of screw bolts **27**. Also, this housing supports shutter assembly **17** and push-pull control means **17A** which means, in turn, actuates toggle arms **28** to operate shutters **16**. Hence, the volume of cooling air allowed to enter spaces **14** of the duct assembly may be controlled at all times.

Means for reducing engine discharge noise

Coincident with the conversion of fluid energy into power by means of the herein described multi-stage turbine lies the positive certainty of being able to prevent the occurrence of noise. The exhaust noise of an engine is the result of high-temperature, high-pressure gas puncturing and violently displacing air. Being an elastic substance and under heavy static pressure itself, air immediately rushes into the pockets created by such displacement at a speed equal to that at which it was displaced. From a practical standpoint, this reaction takes place instantly and the result is a collision of air molecules, which collision translates into noise. Hence, exhaust noise is a reaction phenomenon. Clearly, then, the more thorough the conversion of temperature and pressure into work, the less the energy available for displacing air and, of course, the less the resulting reaction which we identify as noise.

Should the source of the working fluid be the exhaust of an internal combustion engine, it will of course be in the form of intermittent impulses. Upon passing between the impulse turbine blades, the impulses will become thoroughly decimated and smoothed out within the chamber of the rotor. Also, the impulses will be further minimized upon passing through the reaction buckets in the rotor's periphery. Nevertheless, should any acoustical frequencies still remain in the working fluid, they may be absorbed by passing the fluid through discharge channel extension **22**, which encompasses the peripheral right angular formation of casing walls **18** and **19**. This channel extension is secured to said angular wall formation by a plurality of screw bolts **23**.

Channel extension **22** is lined with sound absorbing media **24** so that any acoustical frequencies resident in the gas will become absorbed in the interstices of said media. As indicated, this extension may be of any length desired.

Bevel gear **29** is shown mounted at the forward end of shaft **8** to suggest a power-take-off means or for coupling the turbine shaft with the shaft of, for example, an internal combustion engine.

Having described my invention, I claim:

1. A multiple stage turbine unit including; a shaft having power connection means thereon, a two stage rotor which is impulse and reactively driven by a single passage of high temperature, high pressure motive fluid therethrough, and a casing having forward and rearward portions encompassing and supporting said rotor for rotation therein, said casing having an annular assembly of controllable air-cooled inlet ducts connected to said forward portion thereof for conveying air and said motive fluid through said casing, said two stage rotor including a forward disc and a rearward disc, each of said discs being mounted on said shaft for rotation therewith, said forward disc having an annular opening therethrough and a plurality of centrally positioned and radially extending impulse blades mounted in said annular opening to comprise the first turbine stage of said unit, said blades being removably secured in said opening in said forward disc and being in radial alignment with said inlet ducts to receive the discharge of said fluid and air therefrom, a second turbine stage including a radially disposed chamber formed between said forward and rearward discs and communicating with said opening for the centrifugal throw therethrough of said fluid and air, a plurality of buckets mounted in said chamber and secured to said forward and rearward discs, said buckets curving radially and converging right angularly in a direction opposite to that of rotation and having a converging-diverging orifice in their peripheral ends for the discharge therethrough of said fluid in a direction opposite to that of rotation.

2. In a multiple stage turbine unit; a shaft, a two stage rotor mounted on said shaft and comprised of forward and rearward discs spaced from each other to form a chamber throughout a major portion of their radial lengths, means securing said discs to each other and means securing said discs to said shaft, said forward disc comprised of a hub and an annular radially spaced plate concentric with said hub, a plurality of impulse blades mounted in the space between said hub and said plate for receiving thereagainst the impingement of a high-temperature, high-pressure fluid, said rearward disc having a rounded contour formed on the forward face thereof opposite said impulse blades for deflecting said fluid radially through said chamber, reaction drive buckets mounted in said chamber between said discs at the peripheries thereof, the facing walls of said discs inclining toward each other to form a wedge-shaped rim at their peripheries.

3. The two stage rotor according to claim 2 wherein said reaction drive buckets have sides that conform to the wedge-shape of said rim for being secured therebetween, said buckets having open, wide diametered bases for establishing open communication with said fluid and having radial ends which open to atmosphere, the ends of said buckets advancing in the direction of rotation and having a wall that commences radial and curves gradually to form, relative to its base, a right angle at its periphery, the end trailing in the direction of rotation hav-

ing a shorter radial length from its base to its periphery and, commencing at said base, curving to form a right angle at its periphery, said sides converging inwardly toward each other and said ends converging to form in conjunction with said sides a converging-diverging orifice for the discharge of said fluid therefrom in a direction opposite to that of said rotor's rotation.

4. The two stage rotor according to claim 2 wherein said reaction buckets have a converging-diverging orifice formed at the buckets' discharge ends, the converging part of said orifice facing in a direction opposite to that of rotation, the diverging part thereof extending on its side facing said forward disc a distance greater in said opposite-to-rotation direction than that of its opposite side, said greater distance forming an angular terminus at said bucket ends for directing fluid discharging therethrough in a partially rearward direction relative to the mounting of said buckets.

5. The two stage rotor according to claim 2 wherein said reaction buckets have an inner liner conforming in shape to the advancing wall of the buckets and being secured thereto, said liner having on its innermost surface a plurality of stepped formations extending inwardly therefrom for promoting intermittent impulse effects upon the fluid striking against said advancing wall.

References Cited in the file of this patent

UNITED STATES PATENTS

883,910	Pierce	Apr. 7, 1908
1,009,534	Lamberti	Nov. 21, 1911
1,115,853	Maxim	Nov. 3, 1914
1,298,564	Rice	Mar. 25, 1919
1,681,607	Bowen	Aug. 21, 1928
1,717,203	Gottschalk	June 11, 1929
2,020,793	Meininghaus	Nov. 12, 1935
2,039,800	Jack	May 5, 1936
2,081,150	Meininghaus	May 25, 1937
2,099,699	Meininghaus	Nov. 23, 1937
2,111,136	Bauer	Mar. 15, 1938
2,428,330	Heppner	Sept. 30, 1947
2,428,999	Smith et al.	Oct. 14, 1947
2,484,774	Yates	Oct. 11, 1949
2,636,344	Heath	Apr. 28, 1953
2,648,519	Campini	Aug. 11, 1953
2,651,172	Kennedy	Sept. 8, 1953
2,652,899	Walton et al.	Sept. 22, 1953