MAIN DEVELOPMENT TRENDS IN THE DESIGN OF GEAR PUMPS

(SURVEY OF FOREIGN PATENTS)

V. V. Burenin, V. P. Dronov, and A. I. Yakovlev

UDC 621.664"71"(047)

Gear pumps are one of the most widely applied rotary pumps and are used in many branches of industry to pump gasoline, diesel fuel, various oils, petroleum, and other liquids. Gear pumps have a number of advantages over other types of rotary pumps: they are simple in construction, reliable and have a long service life, are easy to operate, can operate at relatively high speeds, and are small and light in weight.

Gear pumps are available with outside and inside meshing gears. Single stage two rotor gear pumps with outside meshing have received the widest acceptance since they have the simplest construction and are cheapest to fabricate. However inside meshing gear pumps are more compact.

Development of gear pump design is proceeding along the following lines: increased reliability and service life of the pumps as a whole and the individual assemblies; use of new materials of construction; development of high head, high speed pumps, controlled throughput, and reversible direction of pumping; improving fabrication technology and reducing the costs of production and operation; and increasing the cavitation margin, the volumetric and mechanical efficiency.

A gear pump with a special lubricating system for the gear journals [1] has been developed. The ends of three short journals enter three corresponding dead-end chambers. A fourth chamber is provided around the neck of the drive shaft. All four chambers are interconnected by passages in the shafts and chamber and with the suction line through a common passage. A valve is built into the common passage which maintains a certain pressure in the chambers. During pressurized operation of the pump the journals are lubricated by liquid flowing through the clearances. A valve is provided connecting the lubricating chambers with the discharge line for lubricating the journals when the pump operates under no load and leakage is small. When pump discharge pressure is low this valve is kept open by a spring and liquid flows through it into the lubricating chambers. The valve closes on increasing pressure and lubrication occurs only due to leakage.

The construction of a reversible gear pump with inside meshing is shown in Fig. 1 [2]. A cylindrical floating ring 2 is located in the casing 1. This ring has an eccentric internal bore and a radial slot. Lever 3 is located in this slot. Outer rotor 4 is mounted in the eccentric ring with inner rotor 5 connected to the drive shaft, located inside 4. Suction and discharge openings, connected with the corresponding suction and discharge lines, are provided in the pump housing which also has two pins 6, located 180° from each other, which limit the rotation of the floating ring. Liquid in the spaces between the teeth of the rotors is moved from the suction to the discharge line when the rotors turn. When the direction of shaft rotation is reversed the floating ring rotates in the same direction since it tightly encloses the outer rotor. The ring turns until the lever presses on the pin and opens up the split floating ring. The rotation of the floating ring changes the location of the eccentric ring relative to the suction and discharge openings. This preserves the direction of liquid flow.

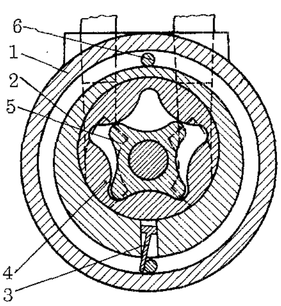

Fig. 1. Reversible gear pump with inside meshing.

An original reversible gear pump [3] is shown in Fig. 2. The two halves of the casing 3 and 8 and gears 4 and 10 are fabricated of Derlin plastic by

Translated from Khimicheskoe i Neftyanoe Mashinostroenie, No. 5, pp. 44-48, May, 1971.

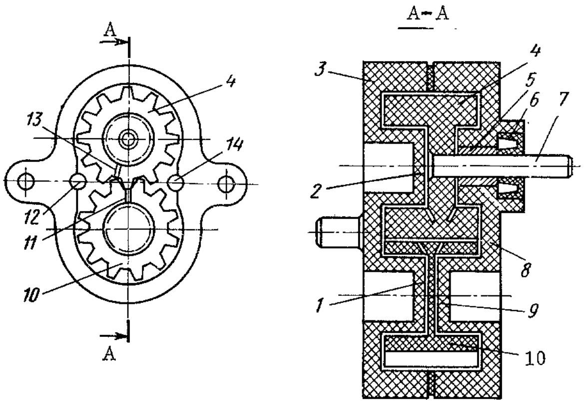

Fig. 2. Plastic gear pump.

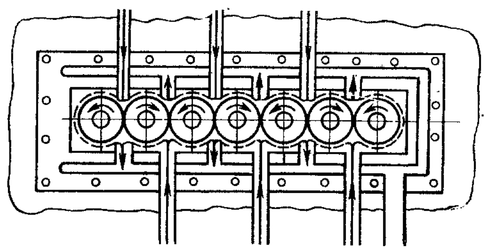

Fig. 3. Multigear pump.

pressing on a casting machine. The diameter of journals 1, 2, and 9 is 0.3–0.4 of the diameter of the average circumference of the gears. The large bearing surface allows the casing to have equal wall thickness throughout which results in uniform stress distribution and effective heat removal from the rubbing surfaces. Lead gear 4 is rotated by the driver through shaft 7 which rotates in sliding bearing 5 and is sealed in the casing by sleeve 6. Liquid is supplied and discharged from the casing of the pump along channels 12 and 14. Openings 11 and 13 are provided between the gear teeth which supply pumped liquid to the support surfaces under pressure. The construction of the pump is simple and compact.

A gear pump with double walled casing construction [4], designed for pumping corrosive liquids, is noteworthy. The internals of the casing in contact with the liquid are fabricated of corrosion resistant metals (stainless steel, titanium, tantalum) or porcelain and are simple in shape. The inner parts are pressed tightly together by the outer casing parts which are bolted. The suction and discharge nozzles are cemented to the outer casing. This cement solution is poured into the annular spaces between the nozzles and the outer casing parts through special openings. Pump construction is simple and well engineered.

The relative simplicity of construction and operating reliability of two rotor single stage gear pumps is used as a basis for development of various designs of multistage, multirotor pumps which are compact, simple, and develop higher heads and have greater capacity.

A patented gear pump for liquid oxygen [5] consists of two sections separated by a plate. Straight tooth drive and driven gears are located in the upper section while the lower section has herringbone gears. The upper section supplies liquid from its discharge nozzle along a passage to the inlet chamber of the lower section which delivers the liquid to the discharge nozzle. The upper section does relatively little work, functioning as a prepump, so that the liquid is not heated up significantly reducing the possibility of evaporation and cavitation in the bottom section. The high capacity of the bottom section and the small hold-up volume of the connecting passage also help reduce liquid evaporation and cavitation. The capacity of this pump, transferring liquid oxygen at –182°C from a tank at 0.35 atm gage pressure to a chamber at 35 atm gage pressure, is 14 liters/min at a drive shaft speed of 600 rpm.

A gear pump with a large solar gear and six small satellite gears located in the casing deserves mention [6]. The point of contact of each satellite with the solar gear is in effect a separate pump. A special feature of this design is floating teeth in the solar gear alternating with fixed teeth, all of which are fabricated as part of the gear. The floating teeth of the solar gear improve the contact between the gear and the casing and reduce leakage improving efficiency and the operating pressure developed by the pump.

Figure 3 shows the construction of an oil-filled gear pump with several intermeshing gears [7]. Each pair of meshing gears represents a small pump. This design considerably reduces the number of gears required to create an equal number of pumping units compared to individual pumps with two gears each.

A multigear pump for delivery and mixing of viscous liquid [8] is worth mentioning. The casing of this pump is formed by three plates. The middle plate is bored for gear installation. A central gear is mounted on the pump shaft. This gear meshes with four driven gears located on the axes of the pump. Each driven gear meshes with an auxiliary gear whose outer circumference is in contact with the periphery of the adjacent driven gear. Liquid is fed to one of the driven gears and is discharged from the adjacent driven

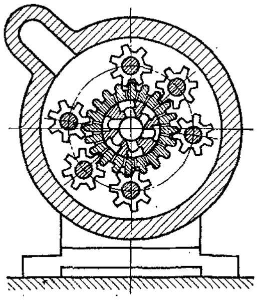

Fig. 4

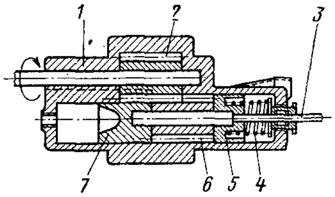

Fig. 5

Fig. 4. Planetary gear pump with liquid distribution by slide valves.

Fig. 5. Gear pump with automatically controlled capacity.

gear. The cavities formed by the drive, two driven, and auxiliary gears are connected by side passages in the cover with pockets between the drive and drive gears. The liquid thus passes in series through all the cavities and pockets of the pump insuring good mixing.

A patented design of planetary gear pump is shown Fig. 4 [9]. Four or six satellite gears and a sun gear are mounted in the casing of this pump. The pole is stationary and there is no crown wheel. The sun gear is driven by a shaft connected to the driver. The cylindrical collector cavity inside the sun gear is connected to the troughs of this gear by radial passages sealed from within by ball check valves, which are kept closed by a common ring shaped spring, or slide valves. During rotation of the sun gear the volume of liquid in the trough is compressed when both surfaces of a tooth of the satellite touch the circumference of the sun gear. Due to the decrease in geometric volume the excess liquid is forced out along the radial passage through the check valve or gate valve into the collector cavity of the sun gear and into the discharge chamber of the pump.

Figure 5 shows the construction of a gear pump with outside gear meshing. The capacity of this pump is continuously variable [10]. Drive 2 and driven 6 gears are installed in casing 1. Pistons 5 and 7 are tightly pressed to the drive gear. Piston 5 is pressed to the drive gear by spring 4 while the pumped liquid pressure presses on piston 7. The block, consisting of the drive gear and pistons, can be displaced parallel to the axis of the drive shaft. This movement varies the length of gear meshing. Increased system pressure reduces pump capacity by means of piston 7. Piston 5 is displaced when pressure decreases increasing the length of meshing and, thereby, pump capacity. Driver power remains constant. Stem 3 is used for manual control of the pump.

Another variable capacity pump design has a casing consisting of two identically shaped halves with three gears mounted one above the other [11]. The casing surrounds the circumference of the upper and lower gears. A chamber for letting down internal leakage is provided close to the middle gear whose axis is eccentrically located relative to the other two gears. Capacity is varied by changing the degree of eccentricity of the middle gear relative to the other two. A bolt connecting the two halves of the casing passes through the axis of the middle gear. The force exerted in tightening this bolt determines the side clearance between the gears and the housing. This bolt is also used to attach the level which is used to vary the position of the middle gear.

There is another variable capacity three gear pump [12] which is of interest. The three gears are located in a row in machined cavities in the casing. The middle gear is installed on an eccentric bushing with large gaps relative to the casing and can be moved from one gear to the other within the limits of meshing. The suction and discharge openings of the pump are located between the side and middle gears. The pump capacity is zero when the regulating gear is in its central position since the delivery of the pumps formed by the middle and side gears are equal and opposed in direction, i.e., liquid flows from one gear to another passing around the middle gear. When the middle gear is moved in any direction its meshing is increased with one of the side gears and decreased with the other one. This either increases or decreases the pump throughput.

A gear pump [13] is available which is designed to maintain constant composition of two component mixtures at the suction end and constant discharge flow. The pump consists of a driver, stand, mounting plate

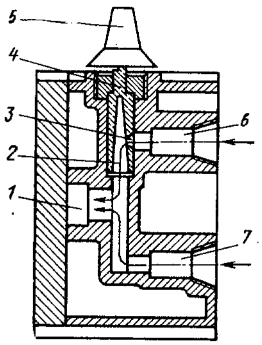

Fig. 6

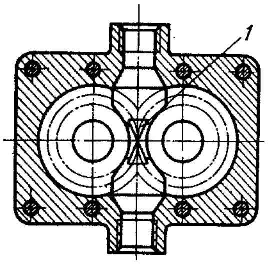

Fig. 7

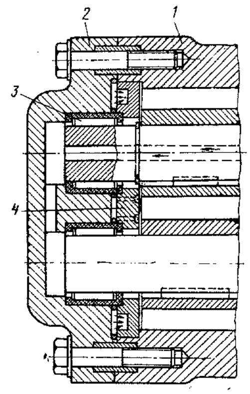

Fig. 8

Fig. 6. Distributor of a gear pump designed for pumping two component mixture of a preset composition.

Fig. 7. Gear pump with a device to prevent compression of liquid trapped in gear teeth troughs.

Fig. 8. Gear pump with automatic end clearance compensation.

with two built-in gears and distributor with inlet valves, mixing chamber, and regulating mechanism in one of the intake passages. All these elements are connected by bolts.

In the distributor (Fig. 6) one of the components of the mixture entering the pump is directed to mixing chamber 1 along directional passage 7 while the other flows along passage 6 through opening 3 and passage 2 of the control mechanism 4. The upper part of the control mechanism is screwed into the distributor housing. Valve 6 can be partially or completely closed by turning lever 5 which is rigidly mounted on the extension of the control mechanism. This will vary the quantitative ratio of the components of the mixture in the mixing chamber. Liquid flows to the operating gears from the mixing chamber.

The liquid is sealed in the trough of the gear teeth by the tight (without gaps) meshing of the teeth as well as the simultaneous meshing of two or several pairs of teeth. The volume of liquid confined in the trough between teeth decreases during gear rotation. Since liquid is relatively incompressible this causes a sharp increase in pressure causing liquid to be pressured through the end clearances to the suction line. Increased pressure results in high loads on the teeth and the support surface of the bearings.

When liquid volume in the trough between teeth is decreased the pressure drops with evolution of vapors and air from the liquid. Pump capacity is reduced when vapors and air enter the suction chamber. Vapors and air are rapidly compressed when they enter the discharge resulting in shock waves against the teeth of the gears which often results in damage to the pump. Special unloading passages in the side covers of the casing, radial borings in the troughs, axial passages in the shaft etc., are provided to prevent trapping of liquid in the troughs of the teeth.

A gear pump [14] has been developed with two narrow grooves cut in the side walls of the casing. One of these grooves extends to the opening under the bearing and serves as the supply route of lubricating liquid. The second groove is dead ended. It provides surge volume at the moment when liquid, which is confined in the troughs between teeth, is isolated from the suction and discharge chambers. The confined volume between the teeth is connected to both grooves at this moment reducing the degree of compression of the confined liquid.

Another interesting gear pump design reduces the noise level during pump operation and the wear rate of the gears [15]. An additional chamber is provided in the casing cover at the point of meshing of the gears which rotates together with the liquid volume confined between the teeth.

This chamber acts as a dampener of pressure peaks when liquid volume decreases and prevents pressure from dropping too rapidly when the confined volume increases. By eliminating pressure surges at the point of gear meshing shock generation, and, consequently, tooth wear and noise level are reduced.

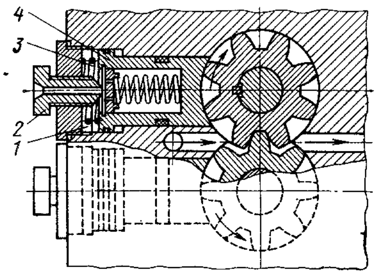

Fig. 9

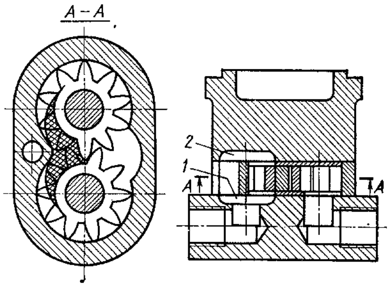

Fig. 10

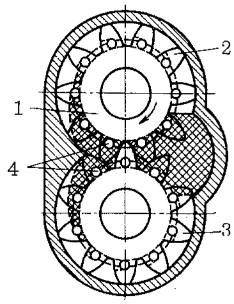

Fig. 11

Fig. 9. Mechanism for radial clearance compensation in a gear pump.

Fig. 10. Gear pump with improved cavitational performance.

Fig. 11. Gear pump with improved suction flooding of troughs of the gear (suction and discharge openings are shaded).

Figure 7 shows the construction of a gear pump which eliminates the sharp increases or decreases in liquid pressure in the confined spaces between meshing teeth [16]. Recesses 1 are provided in the side walls of the end discs. These recesses are contoured to follow meshing lines of the gears. The confined space between the teeth is connected, at the moment of meshing, through these recesses with adjacent spaces which are open and are connected to the suction or discharge chambers.

Power losses occur during the operation of a gear pump which are caused by liquid leaking from the discharge chamber to the suction chamber through the radial clearance between the pump casing and the tips of the gear teeth and through the end clearance between the side walls of the pump casing and gear faces, as well as along the line of contact of meshing teeth which are incorrectly shaped. Leakage is reduced by attempts to minimize the radial clearance. The minimum radial clearance is determined primarily by the clearance in the bearings, their axial misalignment, and the eccentricity of the gears in the casing. The magnitude of leakage is proportional to the cube of the clearance and depends on the liquid viscosity, the velocity of movement of the tips of the gear teeth relative to the plane of the pump casing and the pressure gradient. Liquid leakage is practically nil along the line of meshing contact if gears are fabricated in accordance with high quality standards.

Liquid leakage through end clearances is several times larger than through radial clearances. This is due to resistance to liquid flow which is present in the radial clearances between the surface of the teeth and the casing during gear rotation, while gear rotation contributes to liquid leakage through the end clearances in the direction of rotation. Therefore to achieve a high volumetric efficiency for a gear pump it is necessary to minimize end clearances by designing for high pressures with flexible end walls which are pressed toward the face of the gears by the liquid pressure (hydraulic compensation for end clearances).

Figure 8 shows a proposed pump design with automatic end sealing of the gears. The shafts of the gears in this pump are installed in casing 1 and cover 2 on knife-edge bearings 3 [17]. End discs 4 are located in machined cavities in the casing whose diameter is somewhat larger than that for installation of the gears. There is a small clearance between the walls of the gears and the discs. A deep curved groove is provided in the rear wall of each disc. During pump operation the liquid is fed from the discharge zone to the chamber between the cover and the discs through four holes.

The discs are deflected and touch the walls of the gears under liquid pressure acting on the bottom of the curved passages. This achieves hydraulic compensation of wall clearances. The discs return to their initial position when the pump is unloaded or stopped due to the elasticity of the disc material.

A proposed pump design has the gears attached to shafts installed in thin bushings in the covers of the casing [18]. Hydraulic compensation of wall clearances is effected by opening up a composite oval disc which is fabricated of three stamped plates. These plates are brazed together with hard solder along their outer periphery and are clamped between the casing and cover. Eight contoured openings are stamped in the middle plate. This creates a chamber into which the operating liquid is fed.

One of the central triangular chambers is connected to the discharge zone while the second is connected to the suction zone by holes in the plate which is adjacent to the walls of the gears. Pumped liquid is supplied, under pressure, to each of the six curved chambers from the gap between the teeth of the drive or driven gear. Distribution of liquid pressure which presses the inner plate to the gears is effected by appropriate location of the chambers and selection of diameters of the supply holes. This distribution corresponds to the pressure distribution on the plate which is caused from the gear side by pressure variation in the troughs of the gears. This construction automatically achieves the required magnitude and distribution of forces adjusting the plates for any variation in pressure gradient in the troughs between the teeth which can occur as a result of changes in the operating conditions of the pump or the presence of a large amount of air.

One version of the composite oval disc design has two inner chambers, one of which is connected to the suction zone by an opening while the second is connected to the discharge zone and a number of troughs between the teeth of the gears. In another version the disc has only one common inner chamber for hydraulic compensation of wall clearances.

Hydraulic compensation of wall clearances can also be achieved by the use of two floating bushings which are pressed to the faces of the gears by liquid pressure supplied from the discharge side of the pump. When the pump operates with zero pressure the bushings are pressed to the gears by springs. Since the pressures in the zones of the wall clearances from the side of the gears adjacent to the discharge and suction chambers are different, misalignment and uneven wear, and, sometimes, seizing of the bushings and gears can occur with symmetrical pressing of the floating bushings to the gears. Therefore the area of adjustment of the bushing must be displaced from the center of the bushing in the direction of the discharge chamber of the pump.

Floating bushings with circular projections located eccentrically with respect to the axes of the gears have been proposed [19]. The ends of these projections are under suction pressure while the remaining wall surface of the bushings are under discharge pressure. The eccentricity and dimensions of the projections are selected so that liquid pressure on the bushings from the gear side is balanced.

A gear pump has been patented which has a device built into the casing which limits the increase in radial clearances under operating pressure. The middle section of the casing, in which the gears rotate, consists of two (inner and outer) parts which are close fitted. A narrow annular gap is provided in the outer part. Pressure from the discharge line is applied to this gap compressing the inner part during operation. Another version has a single piece casing; narrow crescent-shaped grooves are provided under the gears in the machined cavity along the periphery to the suction port. These grooves are connected to the discharge line. The pressure applied to the grooves compresses the middle part of the casing preventing its expansion and reducing radial clearances [20].

A device has been developed to regulate the radial clearance in gear pumps [21]. The pump casing (Fig. 9) has two cylindrical openings in which pistons 4 are inserted. The diameters of these pistons are larger than the width of the gears. The concave surface of the piston is under pump discharge pressure. Liquid enters chamber 1 through an opening in the piston and then enters an overflow line through a throttling opening. The size of the opening is controlled by screw 2. Discharge pressure acting on the concave surface tries to move the piston to the left. This force is counteracted by spring 3 and the effect of lower pressure in chamber 1 on the left end of the piston. The clearance is controlled automatically: if the piston moves to the left of the established position the area of the throttling opening is reduced increasing pressure in chamber 3 and the piston moves to the right; when the piston is displaced to the right the area of the throttling opening is increased reducing pressure acting on the left face of the piston and the piston moves to the left.

It is necessary to reduce suction losses and to ensure that the troughs of the gears are flooded with liquid for a gear pump to operate without cavitation. Centrifugal forces, acting on the liquid in the troughs of the gears, impede the flooding of the troughs since liquid enters the rotating gear at a reduced pressure moving from the periphery to the center, i.e., in the opposite direction to the effect of centrifugal force. This limits the allowable rotational velocity of the gears of the pump which is undesirable since higher speeds increase the capacity reducing the size and weight of the pump. A minimum gear rotational speed of 300 rpm is recommended since volumetric efficiency drops off very rapidly below this speed.

Flooding of the troughs between the teeth is improved if a converging inlet nozzle and a diffuser type discharge nozzle are used. The contact angle between the gears and the casing cavity surface increases if liquid is supplied to the gears and withdrawn from them along passages in the form of narrow grooves. By using a converging inlet nozzle the shape of the stream at the gear inlet is improved, the flooding of the gear troughs is increased, air contained in the pumped liquid is uniformly distributed between the two gears, and the power loss due to shock on entry into the recesses of the gears is reduced. These features, described in a US patent [22], increased the absolute pressure at the pump inlet from 100 to 700 mm Hg.

It has been proposed to improve cavitational performance of gear pumps by building special shaped suction chambers 1 and 2 (Fig. 10) in the side covers of the casing (shown shaded in Fig. 10). The cross-sectional area of the chamber decreased in the direction of gear rotation. Flooding of the troughs between the teeth is improved by the action of centrifugal force on the liquid [23].

Gear pumps which can utilize centrifugal force are of great interest. Liquid is supplied in these pumps in a direction from the axis to the periphery, i.e., in the direction of centrifugal forces. Special nozzles are cut in the teeth on the side of the liquid inlet to the gears. The meshing edges of the teeth are not cut and are shaped like vanes. The vanes increase the head and improve suction capabilities of the pump during gear rotation. Therefore centrifugal forces help, rather than impede, the flooding of the troughs between the teeth [24].

The cavitational performance of a high speed gear pump can be improved by making the suction chambers in the side covers of the casing in the shape of a bent wedge. The wide section of the wedge ensures free passage of liquid to the zone where the teeth are just beginning to separate while the vertex, located on the radius of the troughs of the teeth, serves for replenishing liquid which leaves the spaces between the teeth under centrigugal force [25].

Figure 11 shows the construction of a gear pump with casing constructed so that the suction opening is located close to the point of meshing of the gears while the outlet opening is located on the opposite side. Passages 4 are provided in the teeth of one or both gears 2 and 3. These channels are inclined in the direction of the suction opening 1 located in the end cover of the casing. One end of passage 4 is located in the middle at the base of the tooth while the other is at the side surface of the tooth. These passages are connected to the liquid inlet opening before the space between the teeth of the gear which contributes to complete flooding of this volume removing the possibility of cavitation [26].

LITERATURE CITED

1. Federal German Republic (FGR) Patent No. 1160735, Class 59e, 3/01 (1964).
2. US Patent No. 3118387, Class 103-117 (1964).
3. US Patent No. 3105464, Class 103-126 (1963).
4. Japanese Patent No. 7219, Class 63B212 (1967).
5. US Patent No. 3146717, Class 103-5 (1964).
6. US Patent No. 3123012, Class 103-126 (1964).
7. UK Patent No. 1108897, Class F1F (1968).
8. US Patent No. 3266430, Class 103-126 (1966).
9. US Patent No. 3259073, Class 103-126 (1966).
10. UK Patent No. 1152188, Class F1F (1969).
11. UK Patent No. 968998, Class F1F (1964).
12. US Patent No. 3168043, Class 103-120 (1965).
13. US Patent No. 3390638, Class 103-2 (1968).
14. French Patent No. 1385191, Class F05g (1964).
15. US Patent No. 3145661, Class 103-126 (1964).
16. US Patent No. 3303792, Class 103-126 (1967).
17. UK Patent No. 1003844, Class F1F (1965).
18. US Patent No. 3174435, Class 103-126 (1965).
19. French Patent No. 1433169, Class F05g (1966).
20. French Patent No. 1466275, Class F05g (1966).
21. French Patent No. 1553041, Class F04c (1968).
22. US Patent No. 3280756, Class 103-126 (1966).
23. French Patent No. 1405724, Class F05g (1965).
24. UK Patent No. 983898, Class F1F (1965).
25. UK Patent No. 1035529, Class F1F (1966).
26. French Patent No. 1491968, Class F04c (1967).