Oct. 25, 1938. P. F. KIPFER 2,134,286

TWO-STROKE INTERNAL COMBUSTION ENGINE

Filed May 20, 1936 3 Sheets-Sheet 1

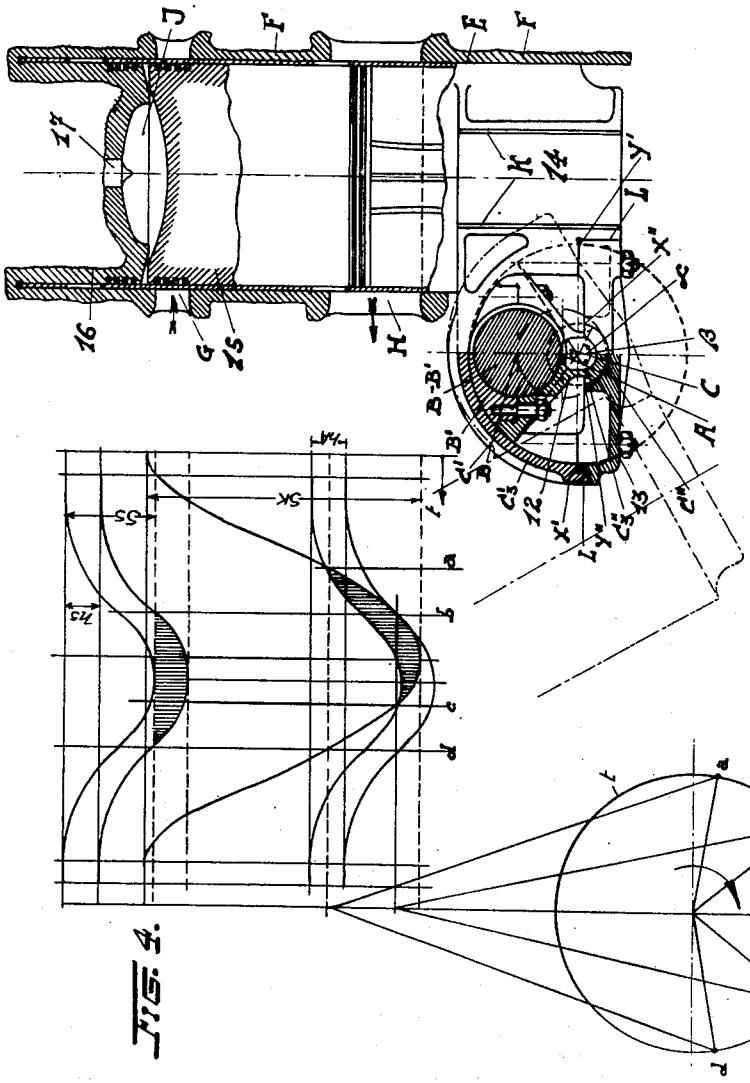


FIG. 1.

FIG. 4.

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TWO-STROKE INTERNAL COMBUSTION ENGINE

Filed May 20, 1936 3 Sheets-Sheet 2

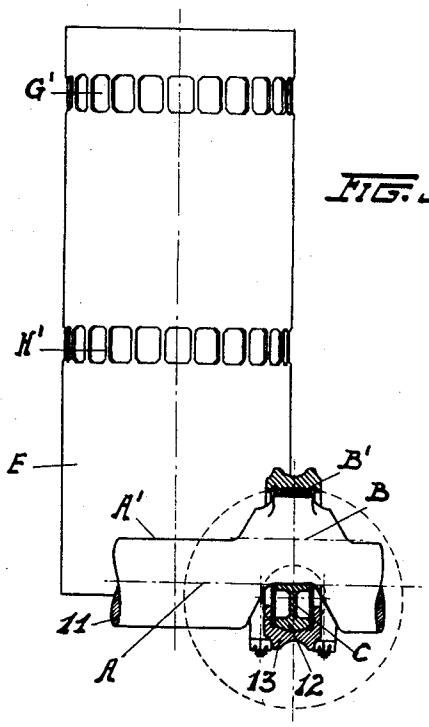
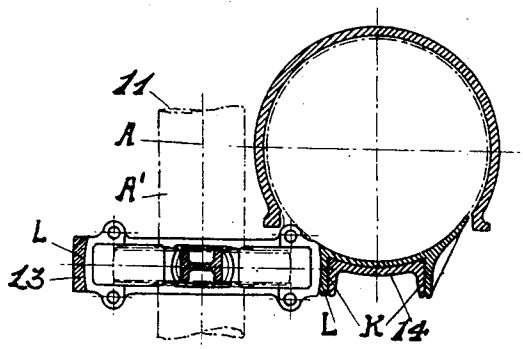


FIG. 3.

FIG. 2.



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TWO-STROKE INTERNAL COMBUSTION ENGINE

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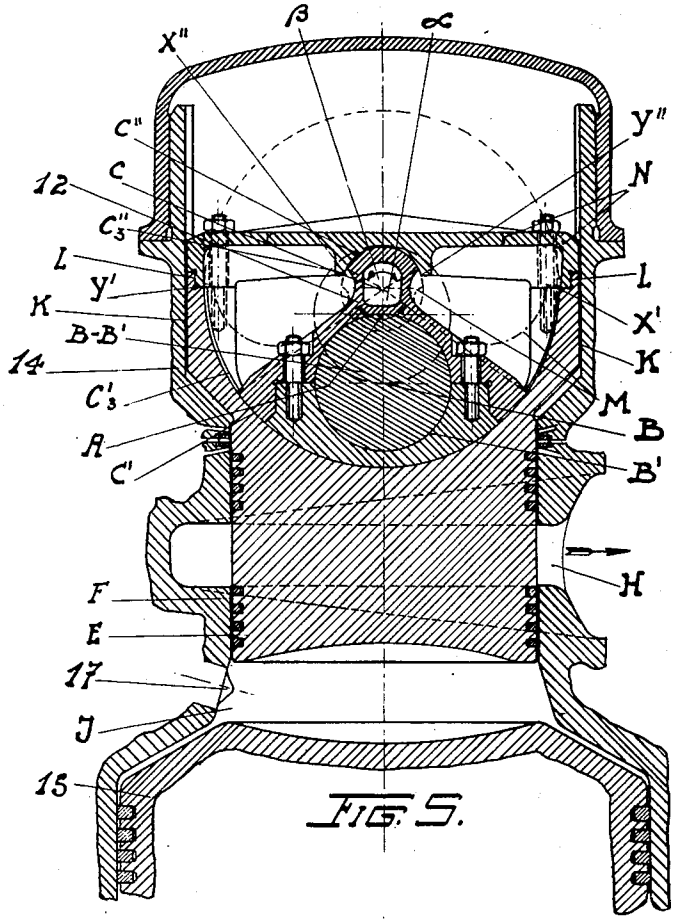


FIG. 5.

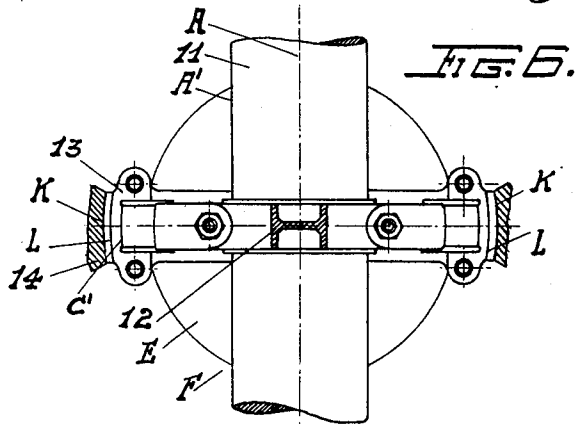


FIG. 6.

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UNITED STATES PATENT OFFICE

2,134,286

TWO STROKE INTERNAL COMBUSTION ENGINE

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Application May 20, 1936, Serial No. 80,660
In Germany and France May 24, 1935

9 Claims. (Cl. 74—50)

The invention relates to a two-stroke internal combustion engine in which the inlet and outlet ports are controlled by reciprocating or reciprocating and oscillating valves.

The load which can be placed on such a machine and its maximum speed of rotation i. e. its maximum performance per litre, depends primarily on the load which can be placed on the mechanism driving the control members, because the magnitude of the acceleration and resultant speed imparted to the control members by this mechanism during the opening and closing of the outlet and inlet ports determines the magnitude of the cross-sectional areas exposed in the given short time and thus determines whether or not the working chamber can be scavenged from the old combustion gases and charged with fresh air during the short time. Now as in two-stroke machines the opening and closing of the outlet and inlet members must be effected once during each rotation of the crank shaft and not once during each second rotation as in four-stroke engines, the control shaft in two-stroke engines must also rotate at the same speed as the crank shaft, i. e. twice as fast as the control shaft of a four-stroke engine. The acceleration of the control mechanism increases as the square of the speed of rotation of the control shaft and thus in the above case is four times as large as in a four-stroke engine. Also the inlet and outlet ports of the two-stroke engine, which are considerably larger than those of a four-stroke engine, involve correspondingly large and heavy closure members. The resulting momenta which have to be taken up by the driving mechanism and the load which can be imposed on it, determine however the maximum speed of rotation and thus the litre-performance of the machine.

In many machines, particularly those in which the outlet and inlet ports are controlled by small reciprocating pistons, the driving mechanism for these control pistons must also withstand the combustion pressures, wholly or in part.

The totality of the said difficulties is of so serious a nature that today, apart from the Junker counter-piston engine there is no high efficiency two-stroke engine which even approximately compares with a high efficiency four-stroke engine in respect of litre-performance. This is due solely to the facts mentioned above.

The invention resides in the use in such machines and in machines as initially set forth for driving the inlet and outlet members of a mechanism for converting the rotary motion of a shaft into a reciprocating motion, which as driving mechanism for the inlet and outlet members of high efficiency two-stroke internal combustion engines affords in the construction and combination according to the invention important new advantages in such machines. These advantages can briefly be summarized as of a mechanical nature in so far as they permit maximum working load under momenta, combustion pressures and momenta without erosion or fracture or dangerous torsional oscillation arising in the control shafts, particularly when use is made of pendulum bearing surfaces in the driving mechanism, and secondly of a kinematic nature in so far as the driving mechanism appropriately combined with the movement of the working piston affords maximum speeds of opening and closing and maximum apertures for scavenging and charging the working chamber towards the lower dead-centre position and minimum speeds and maximum overlap of the sealing surfaces and closure members which are subjected to the highest combustion pressure towards the upper dead-centre position of the working piston, and thirdly of a constructional nature in so far as with a given maximum throw and a given maximum performance the drive occupies a minimum space. All these three factors which determine the success of the engine occur however only in the construction and use in accordance with the invention of the new mechanism for driving the inlet and outlet members. The high capacity of the new control on the one hand and the inventive combination with high-efficiency two-stroke internal combustion engines on the other hand, afford the new result that high efficiency two-stroke internal combustion engines can be constructed with success in which with the same speed of rotation can be charged with the same fuel-air weight per working stroke as high-efficiency four-stroke engines and thus these new two-stroke engines attain practically completely the theoretical advantage of 75% over four-stroke engines.

The novelty thus resides in that for driving the known reciprocating or oscillating inlet and outlet members of high efficiency two-stroke internal combustion engines, use is made of a crank gear of special construction, the characteristic feature of which, in comparison with the first application, resides in that the crank pin or eccentric bearing surface B^1 is embraced by the diametrically opposed concentric spherical or cylindrical bearing surface sectors C′ and C″ or C_3' and C_3'' of different radii of curvature as well as by the connecting lines $y'—x''$ and $y''—x'$ to the reciprocating or oscillating centre, or by the pro-

jection parallel to the axis of the eccentric or crank pin of the above bearing surface sectors and their connecting lines, this reciprocatory motion for actuating the outlet member or inlet member or both preferably being synchronized with the reciprocatory motion of the working piston in such manner that the dead-centre position of maximum working and combustion pressure of thhe working piston corresponds with the dead-centre position of most uniform speed of the outlet and inlet members.

From the constructional and kinematic character of the actuating mechanism on the one hand and its combination with the inlet and outlet members of high efficiency two-stroke internal combustion engines with corresponding appropriate synchronization of the movement of the outlet and inlet members with the movement of the working piston on the other hand, all the advantages fundamental to practical success are obtained for the above class of internal combustion engine.

Fig. 1 is a longitudinal section of the cylinder of an internal combustion engine, showing a sleeve valve together with the control mechanism therefor;

Fig. 2 is a transverse section of the same;

Fig. 3 is an elevation of the sleeve valve and the control therefor, the control mechanism being shown in section;

Fig. 4 is a diagram illustrating the operation of the valve under the action of the control mechanism;

Fig. 5 is an enlarged sectional elevation of a cylinder of a two-stroke internal combustion engine showing a piston valve and control mechanism therefor above the cylinder; and

Fig. 6 is a plan of the same with the cover removed.

As follows from Figs. 1, 2, 3, 5 and 6, such a drive occupies only about ⅔ of the structural height and has about ½ the weight of the drive dealt with in the preceding application, while apart from the new advantages the new drive also exhibits all the advantages of the old drive.

The entire control arrangement comprises four rigid co-operating parts namely the eccentric or crank shaft 11 with centre A and bearing surfaces A' and the eccentric disc or crank pin B—B' with centre B and bearing surface B', the connecting rod disc 12 with the reciprocating centre C and the associated bearing surfaces C' and C'', which is mounted directly in the bearing surfaces C_3' and C_3'' of the reciprocating control carriage 13. The control carriage 13 is preferably wrought integrally with the control member E or is bolted thereto rigidly or by suitable movable members. The carriage 13 is guided in the corresponding guide frame 14 which takes up lateral thrust. In the same way as the control carriage and control member, the frame 14 is preferably wrought integrally with the rigid cylinder sheath F which is directly cooled by the cooling agent and in which the inlet and outlet ports are provided.

The reciprocating control carriage can with advantage be replaced by a control housing which embraces the connecting rod disc 12 and oscillates about a fixed pin, the free oscillating end being connected by means of a link to the control valve which is to be actuated.

The concentric bearing surfaces C_3' and C_3'' ground on the carriage 13 or angle lever have the important characteristic that they are formed as sectors of different radii of curvature the effective bearing surface sectors C_3' commencing at x' and terminating at y' whereas the bearing surface sector C_3'' continuing in the same direction commences at x'' and terminates at y''. The function of the broken line bearing surface sector y'—x' which is cut away is fulfilled by the bearing surface sector C_3'' whereas the broken line bearing surface sector x''—y'' which is cut away is replaced by the bearing surface sector C_3'.

This construction has the outstanding feature that with the minimum distance B—C and given forces, the bearing surfaces B', C' and C'' or C_3' and C_3'' can be made of any desired size and radius of curvature without becoming constructionally impossible. In particular, it is a material advantage that on the basis of the invention the mounting of the connecting rod disc 2 is no longer restricted to a closed bearing surface the minimum diameter of which is determined by the diameter of the crank pin and its distance from the reciprocating centre (see dotted supplementary curves y'—x' in Fig. 1) but this bearing surface can be assembled from two sectors one of which can have a radius of curvature almost arbitrarily small and accordingly the entire gear becomes very compact and light. The two bearing sectors ground on the connecting rod disc, the lengths of which are determined by their angular extents α and β, are preferably chosen of such length that their angular lengths α and β are not smaller than the maximum angular throw of the centre line of the connecting rod disc during one rotation of the crank shaft. The angular extents α and β could however, if desired, be considerably enlarged by a shoe-like configuration of the connecting rod disc. This step however involves corresponding apertures in the cover of the control carriage which however can readily be shaped accordingly without increasing either the weight or the space occupied by the mechanism. The shoe-like configuration of the connecting rod disc and the apertures cut in the cover of the control carriage are shown by the dotted contours M and N in Fig. 6.

In all embodiments, the bearing surfaces C' and C'', and C_3' and C_3'' are spherical. This has three important advantages:—

1. The connecting rod disc is itself held against axial displacement.

2. Angular inaccuracies in manufacture or assembly between the journal A of the control shaft 1 and the axis of the crank or eccentric pin B or relatively to the plane of motion of the reciprocating slide, are unimportant because due to the self adjustment of the connecting rod disc, no wedging can occur. The surface pressures on the spherical bearing surfaces C' and C'' or C_3' and C_3'' are always uniformly distributed and consequently may attain maximum values even in permanent operation.

3. If with the divided control carriage or angle lever the division is effected in a medial plane through the centre C of the spherical surface and if the corresponding connection L (Figs. 1, 2, 5 and 6) is constructed as a cylindrical surface with its axis passing through the centre C and at right angles to the plane of division, and if in the case of the carriage the two carriage slides K are cylindrical and have the same cylinder axis, then the entire element with its spherical surfaces, its plane of division and the associated cylindrical connecting surfaces can be produced with mathematical accuracy by machine tools and all bearing surfaces of the two parts of the carriage or angle lever are centred with mathematical accu-

racy. Figs. 5 and 6 show a piston valve mechanism which unites all these features.

4. In many cases the bearing surfaces C_3' and C_3'' in the carriage 13 or in the angle lever do not need to be divided at all for the connecting rod disc 12 to be fitted. The connecting rod disc can be introduced in a position pivoted out of the working plane and mounted in the working position by subsequent pivotation.

All bearing surfaces of the control carriage are either case hardened or nitrided and ground, lapped and highly polished. The bearing surfaces of the connecting rod disc are treated similarly. In the case of connecting rod discs of unhardened steel and friction mounting, the bearing surfaces may be coated with a layer of white metal or copper-lead alloy 0.25 to 0.5 mm. thick. Preferably the divided or undivided connecting rod disc is made of wrought bronze or wrought duralumin or electron.

In the case described above as in the case of the earlier main application, in order completely to eliminate lateral thrust on the carriage guide, two small drives or three such drives carried on two shafts which rotate in opposite directions, can be incorporated in the carriage. Consequently the relationship of the rod to the crank radius of the drive can be made still smaller and thus the speed of opening and closing of the control gear made materially higher, or else two or more parallel control pistons can be driven by two control shafts rotating in opposite directions.

The displacement-time diagram of Fig. 4 is obtained by plotting horizontally the time for a complete rotation of the crank shaft and plotting vertically the corresponding positions at any time of the upper edge of the working piston and the control edges of the sleeve valve. This diagram shows that all kinematic requirements are completely fulfilled by the drive. At the point *a* at the lower end of the working chamber, after the working piston 15 has travelled two thirds of its entire stroke S_K, its upper edge passes with maximum speed over the upper edge of the exhaust port H' of the sleeve valve E which has only travelled one fifth of its complete stroke Ss and the exhaust ports H' of which slide down simultaneously over the upper edge of the outlet port H in the fixed cylinder sleeve F. It is apparent that during the entire high pressure period and during almost the entire expansion period the control valve has scarcely moved out of its dead-centre position. Somewhat later, at the point *b*, when the full height *h* of the exhaust ports H' is exposed, the lower edge of the inlet ports G' in the sleeve valve E at the upper end of the working chamber slide at maximum speed over the upper edge of the inlet ports G in the fixed cylinder sleeve F. The exhaust is closed by the upper edge of the working piston sliding over the upper edge of the inlet ports H' in the sleeve valve E at the point *c*. At this moment the inlet ports G' have scarcely passed beyond their maximum opening h_s. Somewhat later, at the point *d*, the scavenging ports G' of the sleeve valve are closed by their lower edge travelling at maximum speed over the upper edge of the scavenging ports in the fixed cylinder sleeve F. Thus the opening of the exhaust, as well as the opening and closing of the inlet, is effected at maximum speed and the closing of the exhaust occurs at an instant of time at which the inlet has scarcely passed beyond its full opening. In this way satisfactory charging of the working chamber up to the scavenging pressure is obtained.

Figs. 1 to 6 show clearly the elegant manner in which the present combination of the known control means on the one hand and the actuating mechanism according to the invention on the other hand co-operate to give in the control of high efficiency two-stroke engines the advantages of all the mechanical and kinematic properties of the said means (which are not available in any other solution to the problem) and render such high efficiency two-stroke engines possible.

The constructional property of the driving mechanism of the control means, that the bearing surfaces C' and C'' or C_3' and C_3'' as well as their connecting lines to the reciprocating centre C of the disc 12 embrace the bearing surface B of the crank pin B—B', it is possible to make the ratio

$$\frac{A-B}{C-B}$$

of the crank radius A—B to the distance C—B of the rotary centre from the reciprocatory centre of the connecting rod disc as large as possible, approximating to the limiting value of unity and preferably lying between 0.6 and 0.9.

The result of this is that as in Fig. 4 the speed curve of the points of reversal of the reciprocating motion of the control means tends very rapidly on both sides of one dead-centre position from a maximum value to zero at the dead-centre position whereas the speed on both sides of the other dead-centre position only approaches zero very gradually as it has assumed a very low value long before this dead-centre position. The speed curve is very clearly shown on the diagram. If, as is a fundamental requirement for the technical success of the control, this motion of the control means is synchronized with the motion of the working piston in such manner that the dead-centre position of gradual speed alteration of the control means coincides with the period of maximum combustion pressure in the working chamber then two main conditions for the possibility of a very high speed high efficiency two-stroke engine are satisfied, because there is not only smallest speed and largest overlap of the sealing surfaces of the control means and smallest speeds and torques in the driving mechanism far beyond the duration of the maximum pressure in the working chamber, but the desired maximum speeds of opening and closing of the control members are automatically obtained for the moments of communication of the exhaust and inlet ports with the working chamber, and thus maximum apertures are obtained in a given time between opening and closing. This also is seen from the control diagram, Fig. 4. One of the most important stipulations for the technical success is that which was stated initially concerning the large overlap of long duration and the small speed of the sealing surfaces of the control means, particularly for high speed Diesel engines. In such engines such difficulty is encountered in providing a seal against the high combustion pressure of 60–80 atmospheres and the piston rings bear so heavily against the bearing surface under the abrupt large increase in pressure in the working chamber, that the wear on the rings, the risk of fracture and the distortion of the grooves accommodating the rings are very large and smallest possible speeds of the closure members are desirable during the

high pressure period. During the combustion period, at least three rings must overlap until the pressure in the working chamber has dropped to 35 atmospheres, as otherwise there will inevitably be blowing past the rings and rapid destruction of the closure members so that the machine becomes inoperable.

It could be added that it should be possible to increase the overlap of the closure members by enlarging their stroke. In the case of sleeve valve engines that is impossible and in engines with piston valves as in Figs. 5 and 6, it is very undesirable for reasons of weight and space.

Exhaustive experiments show that a high speed two-stroke engine with scavenging blower operates best when the maximum openings of the inlet and outlet ports are approximately the same. From Fig. 4 which shows the control diagram for a sleeve valve engine it is apparent that on increasing the stroke of the sleeve valve the overlap at the upper end of the valve is increased with the same maximum port opening but the maximum port opening at the lower end of the working chamber is decreased by exactly the same amount. Thus from the standpoint of the driving mechanism, from considerations of volumetric charging and with reference to the overlap there is an optimum stroke of the sleeve valve and this optimum is present if on the one hand the stroke is between ¼ to ⅓ of the stroke of the working piston and, on the other hand the ratio

$$\frac{A-B}{C-B}$$

lies between 0.7 and 0.9. When these stipulations are satisfied, favourable constructional conditions are also obtained for the driving mechanism and the opening and closure members.

Fig. 1 also shows (in chain-dotted lines) the simple and elegant manner in which in sleeve valve engines having two rows of cylinders in V-formation, all the cylinders can be controlled with a single strong control shaft. It also clearly follows that in such an engine with 2 x 6 cylinders or 2 x 8 cylinders the control shaft must have a high torsional rigidity and thus as large a diameter as possible.

This applies more particularly and to a greater extent to sleeve valves E' on which the combustion pressures operate in the axial direction or to a series engine according to Figs. 5 and 6 where the control shaft has not only to withstand momenta and torques but also the combustion forces and torques.

From this and from Figs. 1 to 6 however, it is also apparent that on the one hand the ratio

$$\frac{A-B}{C-B}$$

in the driving mechanism must approximate to the limiting value of unity and preferably be between 0.7 and 0.9, but on the other hand this is quite impossible even with an excessively thin control shaft and excessively thin cross-head pin these would overlap and be mutually impossible, whereas on the basis of the inventive construction and combination of known inlet and exhaust members with the new actuating mechanism, a new control is provided for high efficiency two-stroke internal combustion engines, the greatest diameter of the control shaft and smallest diameter of the connecting rod disc not being limited by given kinematic relationships but the control consists of a few components which are simple, small and light, are subjected to smallest possible deformation and can be produced with greatest accuracy, and which at high speeds and surface pressures afford absolute silence, reliability and smallest wear and the co-operation of which in accordance with the invention affords an almost ideal combination of all kinematic and mechanical requirements for the control of high-speed high-efficiency two-stroke internal combustion engines.

Apart from the significant saving in weight and space and the corresponding constructional freedom involved by the sector-like form of the mounting of the connecting rod disc in the control carriage or angle lever, all the above advantages, possibilities and characteristic properties of the control are also rendered possible by the old drive and the combination thereof in accordance with the invention.

What I claim is:

1. In a driving mechanism for the control valve of an internal combustion engine, the combination comprising a crank shaft, a strap surrounding the crank of the crank shaft, the strap having diametrically opposite concentric bearing sectors of different radii, and a movably mounted carriage connected to the valve, embracing said strap and having bearing surfaces engaging the bearing sectors of the strap.

2. In a driving mechanism for the control valve of an internal combustion engine, the combination comprising a crank shaft, a strap surrounding the crank of the crank shaft, the strap having diametrically opposite concentric, spherical bearing sectors of different radii, and a movably mounted carriage connected to the valve embracing said strap and having bearing surfaces engaging the bearing sectors of the strap.

3. In a driving mechanism for reciprocating and oscillating control valves for two stroke internal combustion engines in combination an eccentric rotating control pin whereby the motion of the valve is produced, a connecting rod knuckle the outer concentric bearing surfaces of which embrace the said control pin and are eccentric therewith, a control member connected with the control valve and on which the connecting rod knuckle moves, the connecting rod knuckle moving in such a way on the eccentric rotating control pin and on the control member that the moving path of the valve is constrained during the control of the ports by that part of the connecting rod knuckle and its outer concentric bearing surface which holds the eccentric rotating control pin.

4. In a driving mechanism for reciprocating and oscillating control valves for two stroke internal combustion engines in combination an eccentric rotating control pin whereby the motion of the valve is produced, a connecting rod knuckle the outer concentric bearing surfaces of which embrace the said control pin and are eccentric therewith, a control member connected with the control valve and on which the connecting rod knuckle moves, the connecting rod knuckle moving in such a way on the eccentric rotating control pin and on the control member that the moving path of the valve is constrained during the control of the ports by that part of the connecting rod knuckle and its outer concentric bearing surface which holds the eccentric rotating control pin the two outer concentric bearing surface segments of the connecting rod knuckle having the same radii of curvature.

5. In a driving mechanism for reciprocating and oscillating control valves for two stroke internal combustion engines in combination an ec-

centric rotating control pin whereby the motion of the valve is produced, a connecting rod knuckle the outer concentric bearing surfaces of which embrace the said control pin and are eccentric therewith, a control member connected with the control valve and on which the connecting rod knuckle moves, the connecting rod knuckle moving in such a way on the eccentric rotating control pin and on the control member that the moving path of the valve is constrained during the control of the ports by that part of the connecting rod knuckle and its outer concentric bearing surface which holds the eccentric rotating control pin, the two outer concentric bearing surface segments of the connecting rod knuckle having radii of curvature differing from one another and the connecting rod knuckle segment which holds the eccentric rotating control pin having the larger radius of curvature.

6. In a driving mechanism for reciprocating and oscillating control valves for two stroke internal combustion engines in combination an eccentric rotating control pin whereby the motion of the valve is produced, a connecting rod knuckle the outer concentric bearing surfaces of which embrace the said control pin and are eccentric therewith, a control member connected with the control valve and on which the connecting rod knuckle moves, the connecting rod knuckle moving in such a way on the eccentric rotating control pin and on the control member that the moving path of the valve is constrained during the control of the ports by that part of the connecting rod knuckle and its outer concentric bearing surface which holds the eccentric rotating control pin, the two outer bearing surface segments of the connecting rod knuckle forming at least approximately semi-circular segments.

7. In a driving mechanism for reciprocating and oscillating control valves for two stroke internal combustion engines in combination an eccentric rotating control pin whereby the motion of the valve is produced, a connecting rod knuckle the outer concentric bearing surfaces of which embrace the said control pin and are eccentric therewith, a control member connected with the control valve and on which the connecting rod knuckle moves, the connecting rod knuckle moving in such a way on the eccentric rotating control pin and on the control member that the moving path of the valve is constrained during the control of the ports by that part of the connecting rod knuckle and its outer concentric bearing surface which holds the eccentric rotating control pin, the control member on which the connecting rod knuckle moves forming a housing closely fitting the said connecting rod knuckle and having cut-away portions through which the free ends of the larger bearing segment of the connecting rod knuckle pass in their position of maximum displacement.

8. In a driving mechanism of the control valve of an internal combustion engine the combination comprising an eccentric rotating control pin, a connecting rod knuckle surrounding the said control pin, the said connecting rod knuckle having diametrically opposite concentric bearing sectors of different radii and a movably mounted control member connected to the valve, embracing said connecting rod knuckle and having bearing surfaces engaging the said concentric bearing sectors of the said connecting rod knuckle.

9. In a driving mechanism for the control valve of an internal combustion engine, the combination comprising an eccentric rotating control pin, a connecting rod knuckle surrounding the said control pin, the said connecting rod knuckle having diametrically opposite concentric, spherical bearing sectors of different radii and a movably mounted control member connected to the valve embracing said connecting rod knuckle and having bearing surfaces engaging the said concentric bearing sectors of the said connecting rod knuckle.

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