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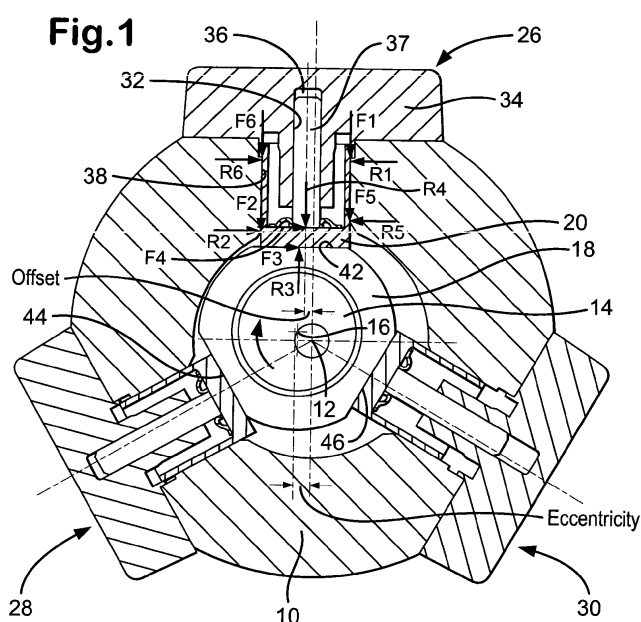
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(54) Fuel pump

(57) A fuel pump for use in an internal combustion engine has a pumping plunger (37) for pressurising fuel within a pump chamber (36) during a plunger pumping stroke and a drive member (20), such as a tappet, for imparting axial drive to the pumping plunger (37) to perform its plunger pumping stroke. The drive member (20) is movable within a drive member bore (38) having a drive bore axis and a shaft-driven cam (14) co-operates with the drive member (20). The cam (14) has a first axis of rotation (16) and the drive shaft has a second axis of

rotation (12) displaced from the first axis of rotation, the first and second axes of rotation (12, 16) being offset from one another to define a pump eccentricity. The drive bore axis (38) is displaced relative to the second axis of rotation (12) by an offset amount, the offset amount being selected to be that amount at which the difference in energy absorbed at regions (120b, 220b, 120c, 220c) of contact between the drive member (20) and the drive member bore (38) as the drive member experiences torque, in use, is substantially minimised.



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Description

[0001] The invention relates to a fuel pump of the type suitable for use in a common rail fuel injection system of an internal combustion engine. In particular, but not exclusively, the invention relates to a common rail fuel pump of the type having at least one pumping plunger that is driven by means of a drive member cooperating with an engine-driven cam.

[0002] In a known common rail fuel pump of radial pump design, for example as described in EP 1 184 568A, three pumping plungers are arranged at equi-angularly spaced locations around an engine driven cam. Each plunger is mounted within a respective plunger bore provided in a pump head housing and, as the cam is driven, each of the plungers is caused to reciprocate. As the plungers reciprocate, each causes pressurisation of fuel within an associated pump chamber. The delivery of fuel from the pump chambers to a common high pressure supply line is controlled by means of respective delivery valves associated with each of the pumps. The high pressure line supplies fuel to a common rail, or other accumulator volume, for delivery to the downstream injectors of the common rail fuel system.

[0003] In one known fuel pump of the aforementioned type, each of the plungers is coupled to a drive member in the form of a tappet. The cam carries a cam ring, or cam rider, which travels over the surface of the cam as it is driven by the engine. Each tappet is located within a tappet bore provided in a main pump housing and is arranged so that, as the cam is driven, each tappet is caused to reciprocate within its respective bore, resulting in reciprocating motion to the plungers.

[0004] As the rider rides over the cam surface to impart drive to the tappet in an axial direction, a base surface of the tappet is caused to translate laterally over a cooperating region of the rider surface. As the tappet is driven radially outward from the shaft, its respective plunger is driven to reduce the volume of the pump chamber. This part of the pumping cycle is referred to as the pumping stroke of the plunger, during which fuel within the associated pumping chamber is pressurised to a relatively high level.

[0005] During the pumping stroke, the axis of the driving force applied to each tappet, in turn, must pass approximately through the centre of the rider. As a consequence, the driving force is generally misaligned with the axis of each tappet bore. This misalignment causes turning moments to be applied to the tappets, which are resisted by reaction forces in the tappet bore. The amount of misalignment varies sinusoidally throughout the revolution, with maxima occurring at 90 degrees and 270 degrees after bottom-dead-centre (BDC).

[0006] In order to reduce the turning moments applied to the tappets, the axes of the plungers and tappets are commonly offset relative to the centre of the drive shaft. It has been found that for a given applied load, a tappet/plunger bore offset equal to 50% of the pump's radial

eccentricity (25% of the pumping stroke) results in minimum peak turning moments.

[0007] However, it has now been found that for a pump configuration of this type there is an imbalance in the work done in overcoming friction at each contact region between the tappet and its bore. It is with a view to addressing this problem that the present invention provides an improved common rail fuel pump.

[0008] According to the present invention, there is provided a common rail fuel pump for use in an internal combustion engine, comprising a pumping plunger for pressurising fuel within a pump chamber during a plunger pumping stroke and a drive member for imparting axial drive to the pumping plunger to perform its plunger pumping stroke. The drive member is movable within a drive member bore having a drive bore axis. A shaft-driven cam co-operates with the drive member and has a first axis of rotation. The drive shaft has a second axis of rotation, offset from the first axis of the cam, so that rotation of the drive shaft results in a motion of the cam which drives the drive member. The offset between the first and second axes of rotation defines a pump eccentricity. It is a characterising feature of the invention that the drive bore axis is displaced radially relative to the second centre of rotation by an offset amount selected to be that amount at which the difference in energy absorbed, or work done, at regions of contact between the drive member and the drive member bore (which occur as torque is applied to the drive member, in use) is substantially minimised.

[0009] In other words, the regions of contact are those regions of the drive member which are caused to make contact with the mating face of the drive member bore as the drive member experiences torque through the pumping cycle.

[0010] In a preferred embodiment of the invention, the drive member takes the form of a tappet, such as a bucket tappet, which rides within a tappet bore provided in a main pump housing.

[0011] In one embodiment, the plunger rides within a plunger bore provided in a pump head housing which is secured to the main pump housing. In a radial pump, several pump units (typically three) may be mounted around the drive shaft, each of which includes a plunger and an associated tappet.

[0012] In another embodiment, the pump housing may be a mono-block, so that the or each pump head housing and the main pump housing together form an integral housing part.

[0013] The invention recognises that reduced wear of the tappet bore is achieved by minimising the difference in energy absorbed (work done) at regions of contact between the tappets and their bores which occur due to turning moments, or torque, being applied to the tappets as they are driven, rather than by minimising the turning moments applied to the tappets. This is a surprising result and provides the benefit that wear of the pump is reduced and, thus, pump service life can be prolonged.

[0014] In a preferred embodiment of the invention, the regions of contact between the tappet drive member and the drive member bore include upper right, upper left, lower right and lower left regions of the tappet drive member.

[0015] More preferably, the upper right and lower right regions of the tappet drive member are regions of a right side wall of the tappet drive member and the upper left and lower left regions of the tappet drive member are regions of a left side wall of the tappet drive member.

[0016] In a preferred embodiment, the drive bore axis is displaced radially relative to the second axis of rotation by an offset amount falling within a range of between 70% and 80 % of the pump radial eccentricity. More preferably, the range is between 72% and 78% of the pump radial eccentricity, more preferably still between 73% and 77% of the pump radial eccentricity, and more preferably still between 74% and 76% of the pump radial eccentricity.

[0017] In a particularly preferred embodiment the drive bore axis is displaced radially relative to the second axis of rotation by substantially 75% of the pump radial eccentricity.

[0018] The invention will now be described, by way of example only, with reference to the following drawings in which:

Figure 1 is a sectional view of a common rail fuel pump of the present invention;

Figure 2 is a schematic view of a tappet drive member forming part of the fuel pump in Figure 1;

Figure 3 shows, for a known fuel pump, the locus of the cam rider centre as it moves around the pump drive shaft;

Figure 4 shows, by way of comparison with Figure 3, the locus of the cam rider centre for the fuel pump of the present invention;

Figure 5 is a graph to illustrate the peak reaction force acting on four regions (right upper - R1, right lower - R5, left upper - R6, left lower - R2) of the tappet drive member, for a known fuel pump compared with the fuel pump of the present invention;

Figure 6 is a graph to illustrate the total work done per revolution at four regions (right upper - R1, right lower - R5, left upper - R6, left lower - R2) of the tappet drive member, for a known fuel pump compared with the fuel pump of the present invention;

Figure 7 is a graph to illustrate peak reaction force as a function of tappet bore offset for four regions (right upper - R1, right lower - R5, left upper - R6, left lower - R2) of the tappet drive member; and

Figure 8 is a graph to illustrate work done as a function of tappet bore offset for four regions (right upper - R1, right lower - R5, left upper - R6, left lower - R2) of the tappet drive member.

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[0019] Referring to Figure 1, a fuel pump of the present invention includes a first housing part in the form of a main pump housing 10 provided with an axially extending opening through which a pump drive shaft extends. In the section shown in Figure 1, the centre axis 12 of the drive shaft is identified, although the drive shaft itself is not visible. The drive shaft has an integrally formed cam 14 which has a centre axis 16 offset radially from the centre axis 12 of the drive shaft (i.e. the cam 14 is eccentric to the drive shaft axis 12). As is common in the field of fuel pumps, the radial offset between the centre axis 12 of the pump drive shaft and the centre axis 16 of the cam 14 is referred to as the 'pump radial eccentricity'. The pump radial eccentricity defines the length of stroke of the plunger 37, which is equal to half the pump radial eccentricity, as will be apparent from the following description.

[0020] Mounted around the cam 14 at equi-angularly spaced locations are first, second and third pump heads or units, generally referred to as 26, 28, 30 respectively. Each pump unit 26, 28, 30 includes an associated one of three drive members 20, 22, 24. A cam ring, or cam rider 18, is carried by the cam 14 and provides an interface between the cam 14 and the pump drive members 20, 22, 24. As each pump unit 26, 28, 30 is identical to the others only one will be described in detail below.

[0021] The first pump head 26 includes a pumping plunger 32 which is reciprocal within a plunger bore 32 provided in a housing 34 of the pump head 26 to cause pressurisation of fuel within a pump chamber 36 defined at a blind end of the plunger bore 32. The plunger bore 32 has a plunger bore axis which defines the axis of movement of the plunger 37. Fuel at relatively low pressure is supplied to the pump chamber 36, in use, and is pressurised to a high level suitable for injection as the plunger 37 is driven to perform a pumping stroke. The pumping stroke is that period of the pumping cycle for which the plunger 37 is moving between bottom-dead-centre (BDC) and top-dead-centre (TDC). A plunger return stroke is that period of the pumping cycle for which the plunger 37 is moving between top-dead-centre (TDC) and bottom-dead-centre (BDC). The plunger return stroke is also referred to as the filling stroke, as it represents that period of the pump's pumping cycle during which the pump chamber 36 is filled with low pressure fuel for pressurisation during the subsequent pumping stroke.

[0022] Each of the drive members 20, 22, 24 takes the form of a tappet which is movable within a tappet bore 38 provided in the main pump housing 10, with the tappet bore 38 having a second, tappet bore axis substantially in alignment with the axis of the associated plunger 37 and its plunger bore 32. Referring also to Figure 2, the

tappet 20 takes the form of a bucket tappet, of generally U-shaped or channelled cross section, including a base 20a and first (left) and second (right) side walls, 20b and 20c respectively, with each side wall having an outer surface, facing the tappet bore surface, of generally curved (convex) form. The left side wall 20b has a left upper region, referred to generally as 120b, and a left lower region, referred to generally as 220b, and the right side wall 20c has a right upper region, referred to generally as 120c, and a right lower region, referred to generally as 220c. The left and right regions 120b, 220b and 120c, 220c of the tappet side walls 20b, 20c are those regions of the outer surfaces of the tappet side walls 20b, 20c which are caused to react against, and make contact with, the tappet bore 38 as the tappet 20 experiences anti-clockwise and clockwise torque throughout the pumping cycle. Hereinafter, the regions 120c, 220c, 120b, 220b of the tappet side walls 20b, 20c will also be referred to as 'contact regions'.

[0023] The upper surface 20d of the tappet base 20a may be provided with a shallow platform 20e for locating the lower end of the plunger 37. Although not shown in Figures 1 and 2, the lower end of the plunger 37 carries a spring plate to define an abutment surface for one end of a plunger return spring which is mounted concentrically with the plunger 37 and serves to drive the plunger return stroke between TDC and BDC. Although the tappet 20 and the plunger 37 are not physically coupled to one another, so that relative axial movement is permitted between these parts, the spring tends to maintain tappet and plunger contact throughout the pumping cycle. Examples of how the tappet 20 and the plunger 37 may be coupled together are described in our co-pending European patent application EP 1431577.

[0024] A lower surface of the tappet base 20e is co-operable with an associated one of three flattened surface regions, or flats 42, 44, 46, provided on the cam rider 18. The three flats 42, 44, 46 of the cam rider 18 are equi-angularly spaced around the rider, which thus has a generally prismatic-type form, so that each is co-operable with a respective one of the three tappets 20, 22, 24.

[0025] In use, as the cam rider 18 is caused to ride over the surface of the cam 14 upon rotation of the drive shaft, an axial drive force is imparted to the tappet 20, and in turn to the plunger 37, causing the plunger 37 to reciprocate within the plunger bore 32. During the pumping stroke, the tappet 20 and the plunger 37 are driven radially outward from the shaft (i.e. vertically upwards for the pump head 26 in Figure 1) to reduce the volume of the pump chamber 36. During the plunger return stroke, which is effected by means of the return spring, the tappet 20 and the plunger 37 are urged in a radially inward direction (i.e. vertically downwards for the pump head 26 in Figure 1) to increase the volume of the pump chamber 36.

[0026] As the tappet 20 is driven in a radially outward direction, imparting drive to the plunger 37 along the

plunger bore axis, a degree of lateral or sliding movement of the tappet 20 occurs across the flattened rider surface 42, in a back and forth manner. The tappet 20 slides across the flattened rider surface 42 in a similar manner during the return stroke.

[0027] During the pumping stroke, the axis of the driving force applied to each tappet 20 passes through approximately the centre axis 16 of the cam 14 and cam rider 18 and, thus, is generally misaligned with the axis of the tappet bore 38. This offset is identified in Figure 1 and the driving force is indicated by arrow R3. The misalignment between the force, R3, and the axis of the tappet bore 38 varies sinusoidally throughout the pumping cycle between 0 and 360 degrees, with maxima in the misalignment occurring at 90 degrees and 270 degrees after BDC. The misalignment causes turning moments (torque) to be applied to the tappet 20 which are resisted by reaction forces applied to the outer surface of the tappet side walls 20b, 20c (indicated by R1, R2, R5 and R6 in Figure 1, as described further below). The turning moments lead to wear of the tappet 20 and/or its bore 38 due to the contact over the regions

[0028] It is a feature of known pumps generally of the aforementioned type that in order to minimise the turning moments applied to the tappet bore 38 the plunger bore axis, and hence the tappet bore axis, is displaced radially from the axis 12 of the drive shaft by an offset amount equal to 50% of the pump radial eccentricity. For the purpose of this specification, the offset between the tappet bore 38 and the centre axis 12 of the drive shaft shall be referred to as the "tappet bore offset".

[0029] Figure 3 illustrates, for a known pump in which a 50 % tappet bore offset is employed, the locus of the cam rider 18 relative to the centre axis 12 of the drive shaft. The turning moment (torque) applied to the tappet 20 through the pumping stroke, and their direction (clockwise or anti-clockwise) are also indicated. During an initial period of the pumping stroke of the plunger 37 (between about 0 and 30 degrees of shaft rotation), an anti-clockwise turning moment is applied to the tappet 20. At around 30 degrees of shaft rotation, the torque reverses direction so that a clockwise torque is applied to the tappet 20 throughout the middle period of the pumping stroke. After about 150 degrees of shaft rotation, during the final period of the pumping stroke, the torque reverses direction again and an anti-clockwise turning moment is applied to the tappet 20 as the plunger 37 approaches TDC.

[0030] Figure 4 illustrates the locus of the cam rider centre for an embodiment of the present invention in which the tappet bore offset is selected to be 75% of the pump radial eccentricity (a 75% tappet bore offset). In this case, an anti-clockwise turning moment is applied to the tappet bore 38 for an initial period of the pumping stroke (between about 0 and 50 degrees of engine rotation) occurring over a greater angular range compared with a fuel pump having a 50% tappet bore offset. The middle region of shaft rotation where the direction of

torque is reversed occurs over a shorter angular region of the pumping stroke (between about 50 degrees and 130 degrees) than for the pump having a 50% tappet bore offset. The final region of shaft rotation where the torque direction reverses again (i.e. anti-clockwise) occurs over a longer period of shaft rotation (between about 130 degrees and 180 degrees) compared with a pump having a 50% tappet bore offset.

[0031] The particular selection of a 50% tappet bore offset has been shown to minimise the turning moments (torque) applied the tappet bore 38 and is thus commonly employed. The turning moments are increased if a greater tappet bore offset is employed and heretofore this has been considered to be undesirable. However, the inventors have now made the surprising discovering that the energy absorbed at each contact region between the tappet 20 and the tappet bore 38 over the pumping cycle is minimised if a tappet bore offset equal to about 75% of the pump radial eccentricity is selected, compared with a tappet bore offset of 50% of the pump radial eccentricity. Minimising the energy absorbed at each contact region over the pumping stroke, as opposed to minimising the *turning moment* over the pumping stroke, has the beneficial effect that wear of the tappet bore 38 is reduced compared with that found in a pump having a 50% tappet bore offset.

[0032] In general, it is understood that the reason why the increased turning moments acting on the tappet bore 38 for a tappet bore offset of 75% do not cause an undue problem is that, over the angular region of drive shaft rotation for which these turning moments are a maximum (around BDC and TDC), the linear velocity and sliding distance of the tappet 20 is relatively low. Reaction force maxima occurring within over these regions of angular rotation therefore only result in minimal work done (the integral of friction force x tappet sliding distance). The benefits of selecting a tappet bore offset of around 75% have therefore been found to outweigh any advantages achieved using a 50% tappet bore offset.

[0033] The beneficial effect will now be explained in further detail with reference to Figures 5 to 7. Figure 5 shows the peak reaction forces, R1, R2, R5 and R6, which act on the tappet 20 over contact regions 120c, 220b, 220c and 120b respectively, as a consequence of torque being applied to the tappet bore 38 as the tappet 20 is driven, in use. In other words, a reaction force R1 is applied to the right upper region 120c of the tappet sidewall 20c, a reaction force R2 is applied to the left lower region 220b of the tappet sidewall 20b, a reaction force R5 is applied to the right lower region 220c of the right tappet sidewall 20c and a reaction force R6 is applied to the left upper region 120b of the left tappet sidewall 20b. Figure 6 shows the total work done per shaft rotation (complete revolution of the shaft) over the tappet side wall regions 120c, 220b, 220c and 120b.

[0034] In Figure 5, it can be seen that, for a pump having a tappet bore offset of 50%, the peak (maximum) reaction forces are approximately the same (i.e. bal-

anced) over the contact regions 120b, 220b, 120c, 220c of the tappet side walls. In other words, each of the right upper 120c, right lower 220c, left upper 120b and left lower 220b regions of the tappet side walls 20b, 20c will experience approximately the same peak reaction force. However, for a pump having a 75% tappet bore offset, the peak reaction forces, R5 and R6, for the right lower 220c and left upper 120b regions of the tappet side walls 20b, 20c are somewhat greater than for a pump having a 50% tappet bore offset. Whilst it has previously been considered desirable to balance the peak reaction forces R1, R2, R5 and R6 to minimise wear, it has now been discovered that an increase in the peak reaction forces R5 and R6 at the right lower 220c and left upper 120b regions does not lead to increased wear in these regions as may be expected. This is because the sliding distances of the tappet 20 over the period of shaft rotation for which the peak reaction forces R5 and R6 are experienced are at a minimum (hence relatively small), and so the energy generated as a result, which leads to wear, is minimised.

[0035] Figure 6 illustrates the aforementioned benefit most clearly. Here it can be seen that although the work done at the right lower 220c and left upper 120b regions of the tappet sidewalls is higher than for the same regions in a pump having a 50% tappet bore offset, the work done is approximately the same over for all four contact regions 120b, 220b, 120c, 220c (corresponding to R1, R2, R5 and R6). In other words, the work done is balanced over regions 120b, 220b, 120c, 220c.

[0036] By way of further illustration, Figures 7 and 8 compare the peak reaction force as a function of tappet bore offset (Figure 7) and the work done as a function of tappet bore offset (Figure 8) over the four contact regions of the tappet sidewalls, 120b, 220b, 120c, 220c. It will be appreciated from Figure 8 that although a tappet bore offset which is 75% of the pump radial eccentricity is selected as the optimum tappet bore offset (for which the work done over each of the four regions 120b, 120c, 220b, 220c is balanced), in practice a tappet bore offset falling within the range of between about 70% and 80% of the pump radial eccentricity will ensure wear is reduced adequately. A tappet bore offset falling within the range of about 72% to 78% is more preferable and a range of about 73% to 77% is more preferable still.

Claims

1. A fuel pump for use in an internal combustion engine, the fuel pump comprising;

a pumping plunger (37) for pressurising fuel within a pump chamber (36) during a plunger pumping stroke,
a drive member (20) for imparting axial drive to the pumping plunger (37) to perform its plunger pumping stroke,

- the drive member (20) being movable within a drive member bore (38) having a drive bore axis, a shaft-driven cam (14) with which the drive member (20) co-operates, the cam (14) having a first axis of rotation (16) and the drive shaft having a second axis of rotation (12) displaced from the first axis of rotation, the first and second axes of rotation (12, 16) being offset from one another to define a pump eccentricity, **characterised in that** the drive bore axis (38) is displaced relative to the second axis of rotation (12) by an offset amount, and whereby the offset amount is selected to be that amount at which the difference in energy absorbed at regions (120b, 220b, 120c, 220c) of contact between the drive member (20) and the drive member bore (38) as the drive member experiences torque, in use, is substantially minimised.
2. The fuel pump as claimed in claim 1, wherein the regions of contact between the drive member (20) and the drive member bore (38) include upper right (120c), upper left (120b), lower right (220c) and lower left (220b) regions of the drive member (20) which experience a torque as the drive member (20) is driven, in use.
 3. The fuel pump as claimed in claim 2, wherein the upper right (120c) and lower right (220c) regions of the drive member (20) are regions of a right side wall (20c) of the drive member (20) and the upper left (120b) and lower left (220b) regions of the drive member (20) are regions of a left side wall (20b) of the drive member (20).
 4. The fuel pump as claimed in claim 3, wherein the drive member takes the form of a tappet (20) having left and right tappet side walls (20b, 20c) defining the regions of contact (120b, 220b, 120c, 220c).
 5. The fuel pump as claimed in any one of claims 1 to 4, wherein the drive bore axis (38) is displaced relative to the second axis of rotation (12) by an amount falling within a range of between 70% and 80 % of the pump eccentricity.
 6. The fuel pump as claimed in claim 5, wherein the range is between 72% and 78% of the pump eccentricity.
 7. The fuel pump as claimed in claim 6, wherein the range is between 73% and 77% of the pump eccentricity.
 8. The fuel pump as claimed in claim 7, wherein the range is between 74% and 76% of the pump eccentricity.
 9. The fuel pump as claimed in claim 8, wherein the drive bore axis (38) is displaced relative to the second axis of rotation by substantially 75% of the pump eccentricity.
 10. The fuel pump as claimed in any one of claims 1 to 9, including three pumping plungers (37) arranged at equi-angularly spaced locations around the pump drive shaft, each of the pumping plungers (37) being co-operable with a respective one of three drive members (20, 22, 24).
 11. The fuel pump as claimed in claim 10, wherein the plunger bore (32) is defined within a pump head housing (34) and wherein the drive member bore (38) is provided within a main pump housing (10), wherein the pump head housing (34) is mounted upon the main pump housing (10).
 12. The fuel pump as claimed in claim 10, wherein the plunger bore (32) is defined within a main pump housing.

Fig.1

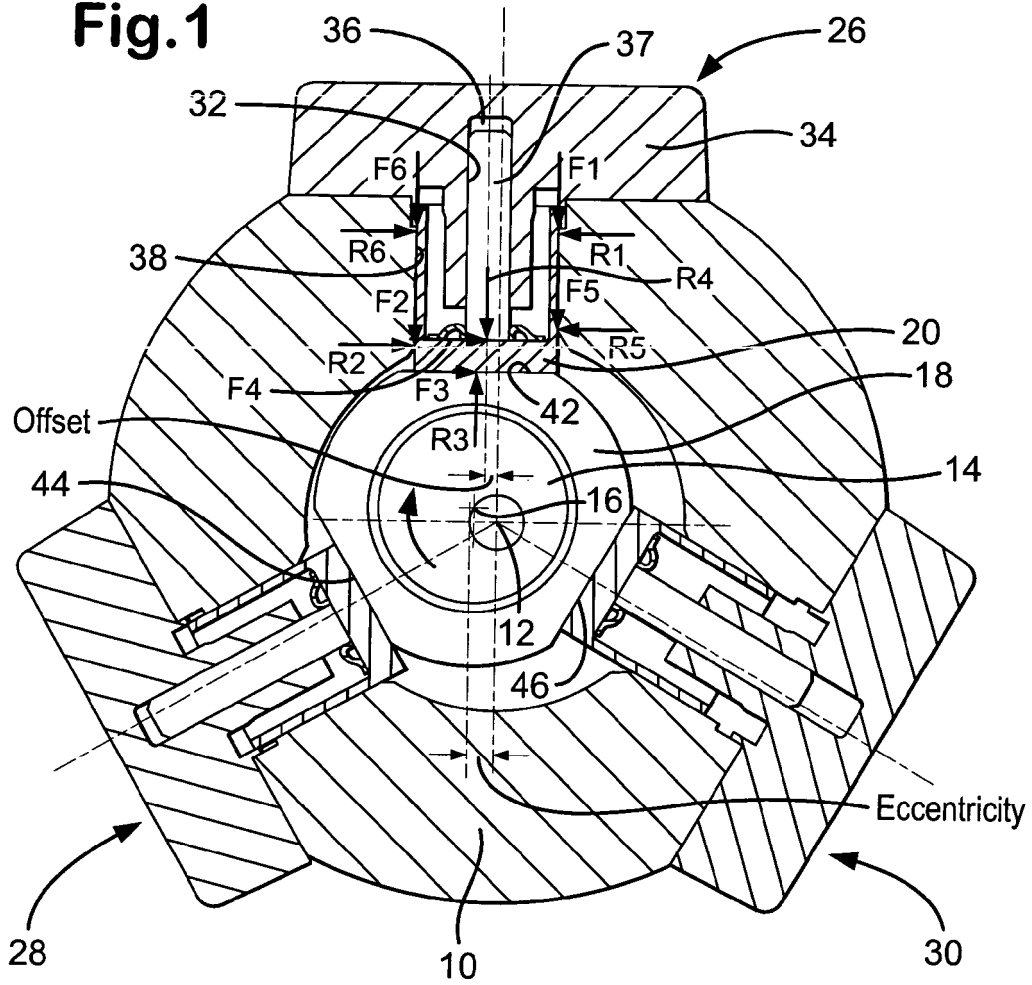


Fig.2

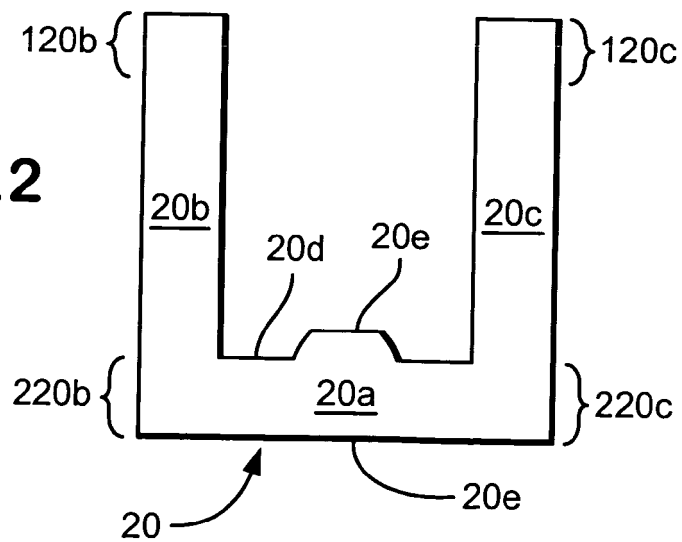


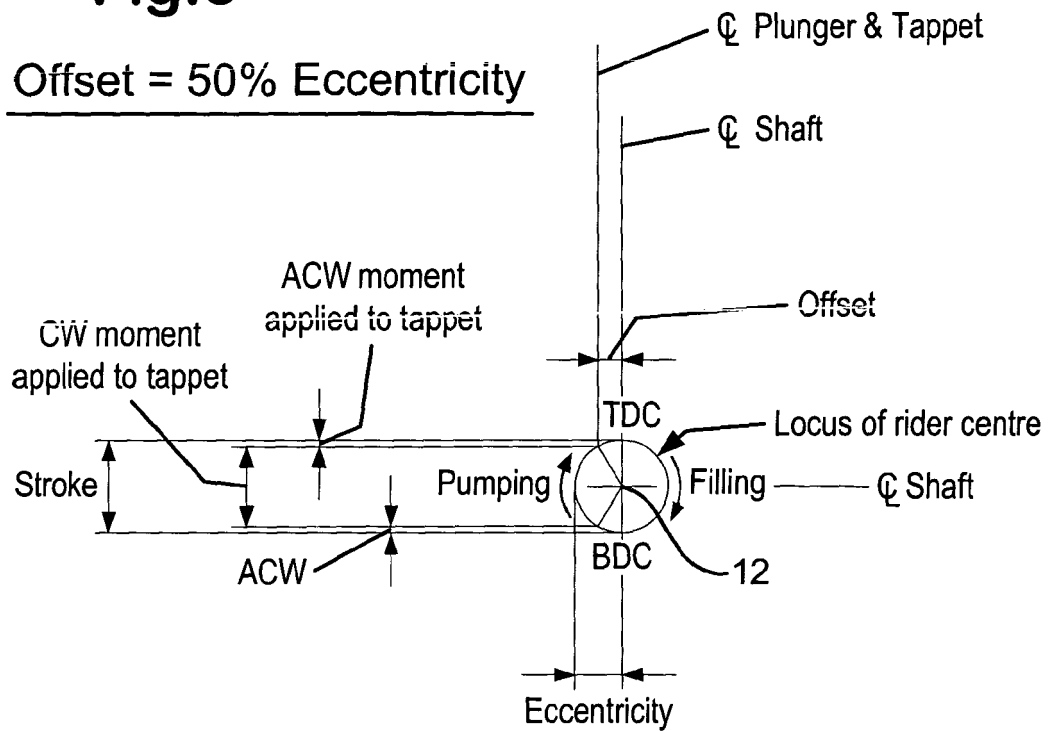
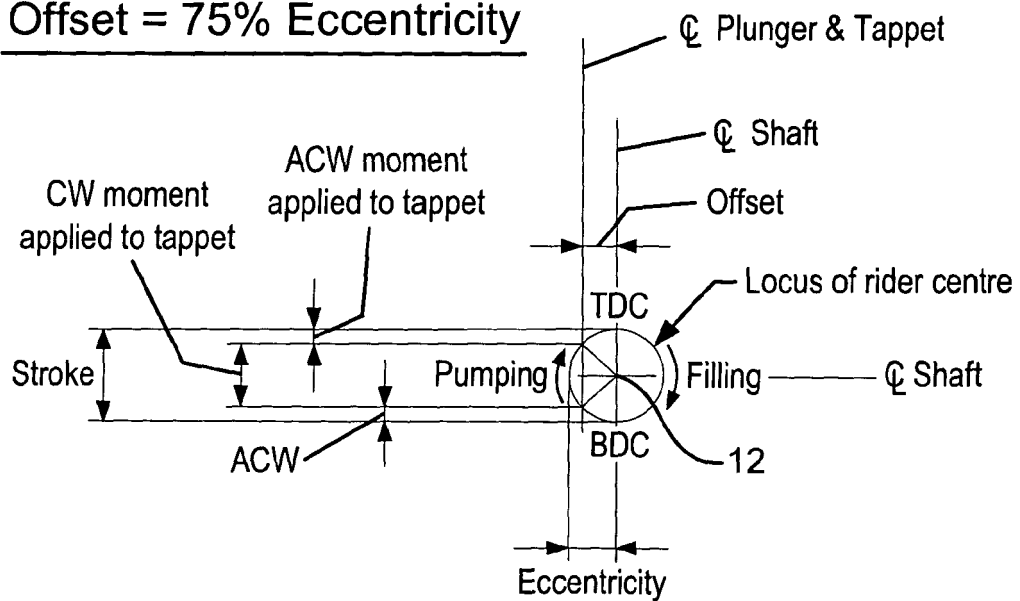
Fig.3Offset = 50% Eccentricity**Fig.4**Offset = 75% Eccentricity

Fig.5

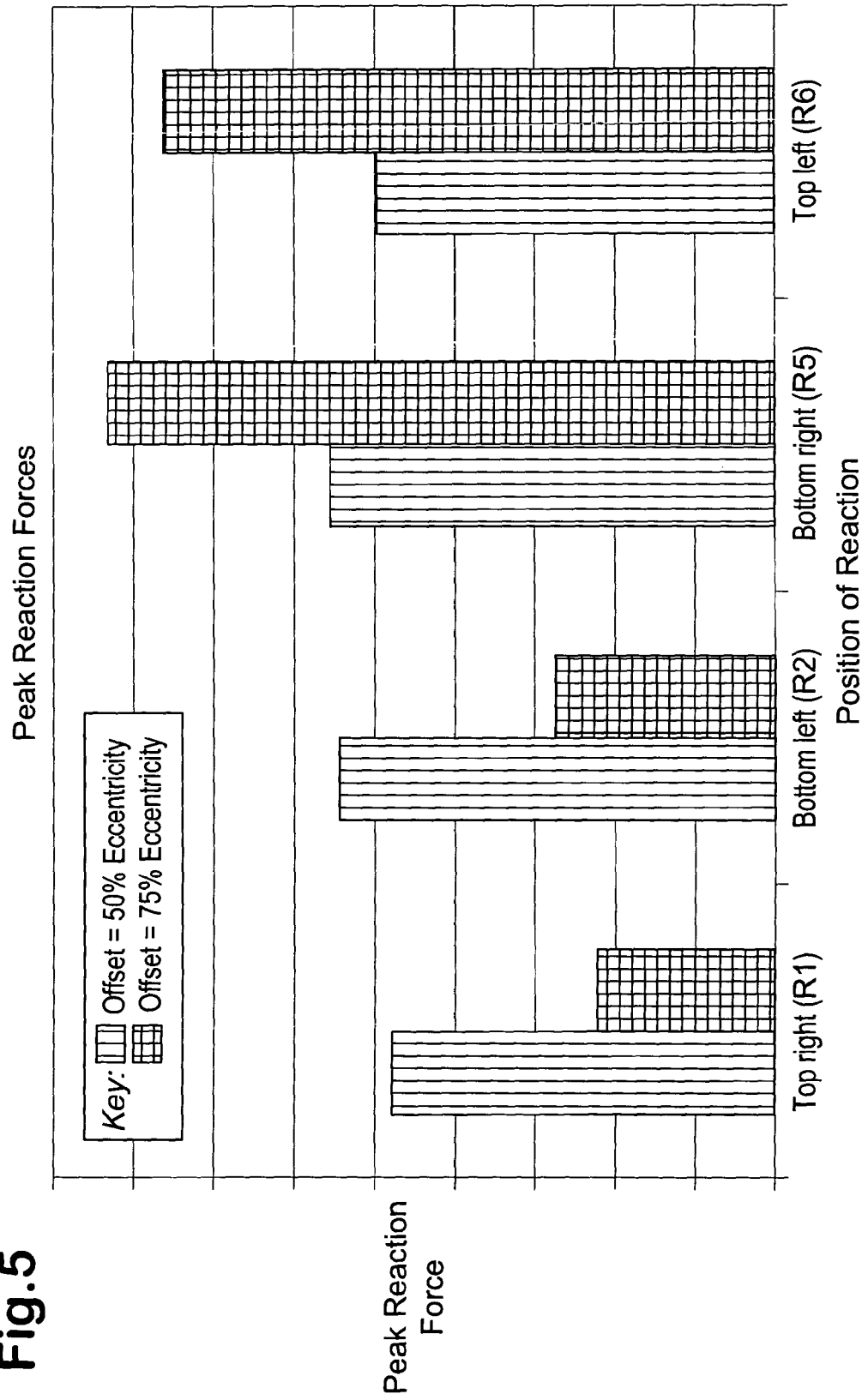


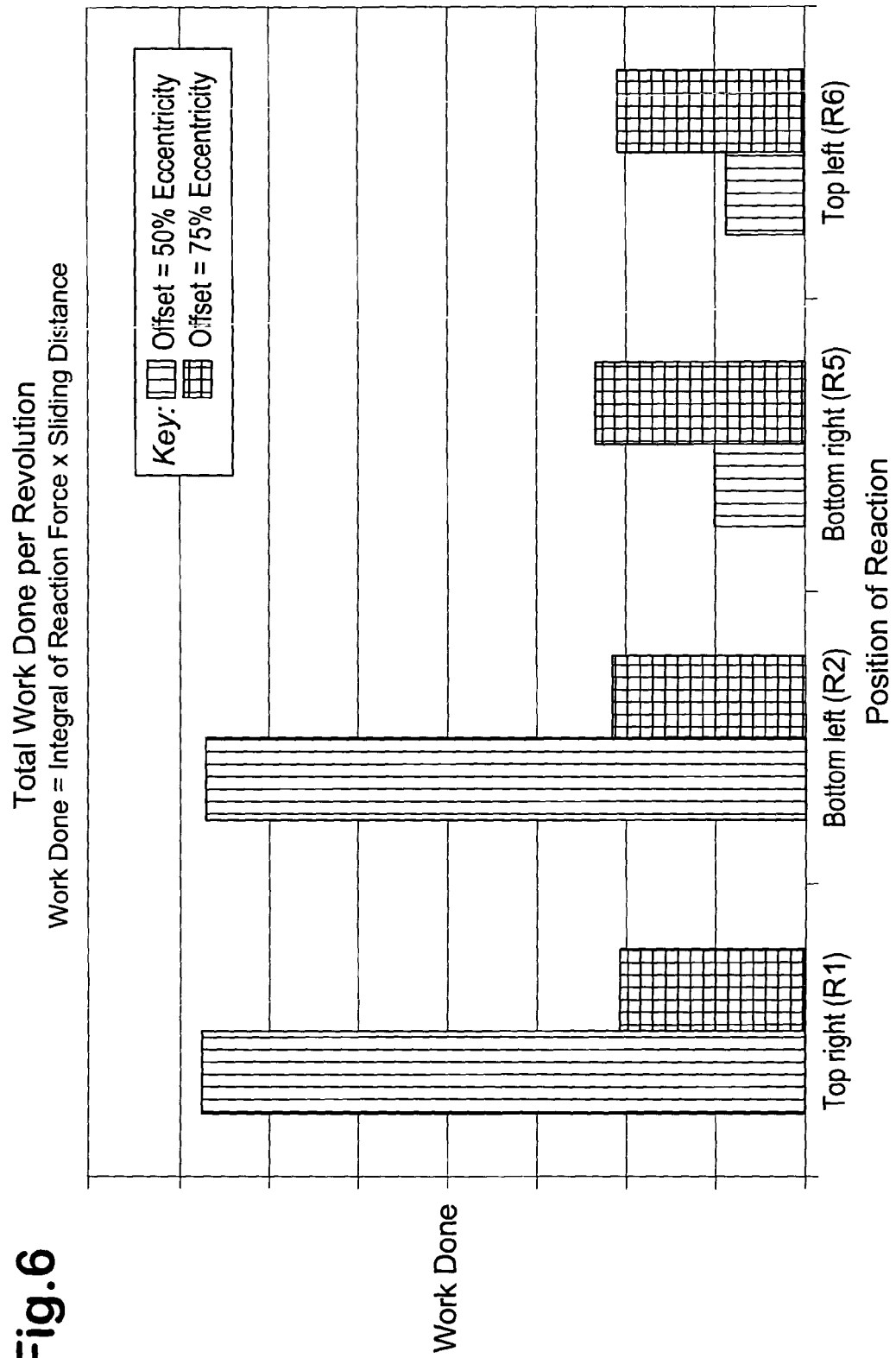
Fig. 6

Fig.7

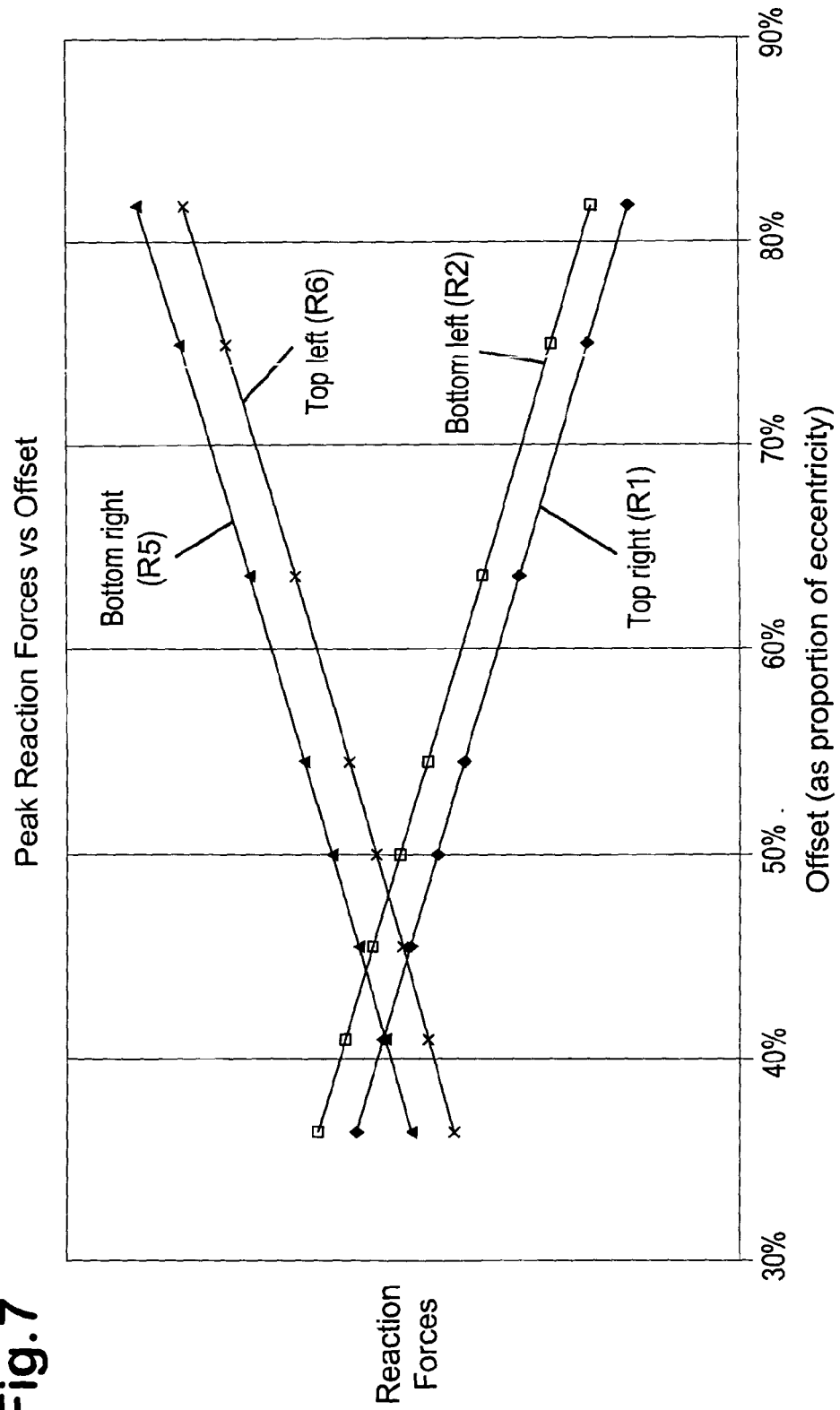
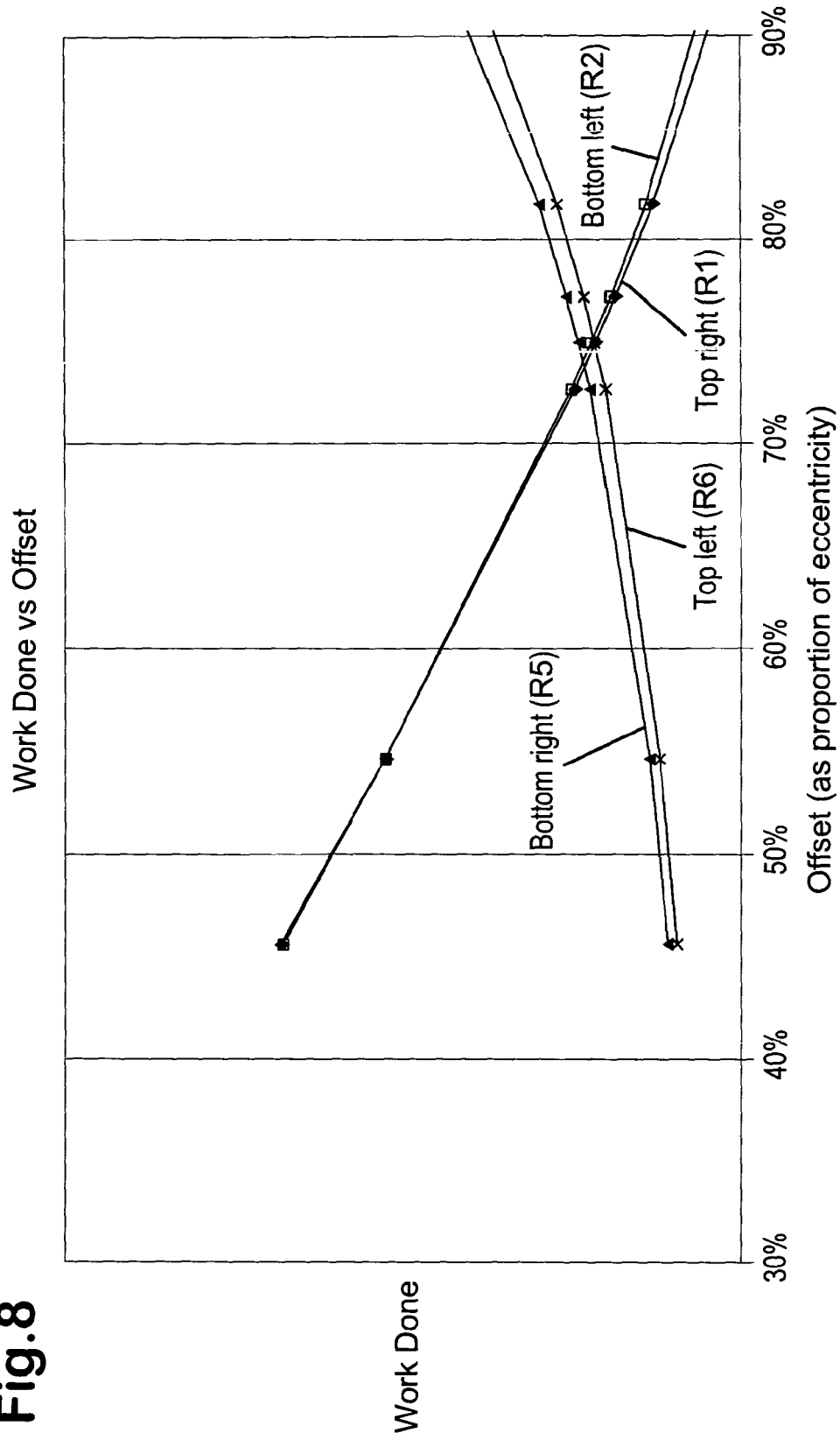


Fig.8





European Patent
Office

EUROPEAN SEARCH REPORT

Application Number
EP 05 25 1838

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.7)
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