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(54) **HYDRAULIC MACHINE COMPRISING DUAL GEROTORS**

ZWEISTUFIGE INNENLÄUFER-GEROTOR-MASCHINE

MACHINE HYDRAULIQUE COMPRENANT UN DOUBLE ROTOR DENTE

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(73) Proprietor: **Valeo Electrical Systems, Inc.**
Auburn Hills,
Michigan 48326-2356 (US)

(72) Inventors:
- **MACHESNEY, Kerry, A.**
  Centerville, OH 45458 (US)
- **HILL, John, S.**
  Dayton, OH 45459 (US)
- **MIENTUS, Michael, J.**
  Metamora, Illinois 61548 (US)

(74) Representative: **Lemaire, Marc et al**
Valeo Management Services,
Direction Propriété Industrielle,
43, rue Bayen
75017 Paris (FR)

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Description

[0001] The invention concerns a cooling system for an automotive vehicle comprising a dual-rotor gerotor machine as specified in the preamble of claim 1. Such a system is known from US-A-4189919. All rotors are placed in a single plane. This arrangement succeeds in placing the gerotors in a housing of small axial length, yet causing them to provide a large displacement of hydraulic fluid per revolution. This arrangement provides large horsepower in a small package.

BACKGROUND OF THE INVENTION

[0002] Figure 1 illustrates a hydraulic machine 2 of the gerotor type, found in the prior art. A shaft (not shown) engages splines 9, and rotates rotor 6. The machine 2 can operate either as a pump or motor, but since operation as a pump is perhaps easier to understand, the explanation will be framed in terms of a pump. Plates such as plate 18 in Figure 7 seal the chambers 3 and 12 in Figure 2, which are described below.

[0003] In Figure 2, rotor 6 rotates about center CA, as indicated by the arrow pointing to that center. Rotor R rotates about center CB, as indicated by the arrow. The distance between centers CA and CB is defined as the "eccentricity" of the two rotors.

[0004] Figures 3 - 6 illustrate these two rotations. Figure 3 illustrates the starting position. Dots D1 and D2 have been added for reference. In Figure 4, rotor 6 has been rotated counter-clockwise by the shaft (not shown) through about 20 degrees. The other rotor R is carried along, but not through a full 20 degrees (because the tooth ratio between the rotors is 6/7). Chamber CH1 has been reduced in volume, thereby causing fluid to become expelled through conduits which are not shown.

[0005] In Figure 5, rotor 6 has been further rotated another 20 degrees counter-clockwise. Rotor R is again carried along, but not the full 20 degrees, and chamber CH1 is further reduced in volume.

[0006] In Figure 6, rotor 6 has been further rotated another 20 degrees, for a total of 60 degrees, compared with Figure 3. Rotor R is carried along, but, again, not by the full 20 degrees. Now a visible separation SEP between dots D1 and D2 begins to appear, indicating the lag of rotor R behind rotor 6. Chamber CH1 is almost compressed to zero volume.

[0007] When the machine operates as a motor, the opposite sequence occurs: pressurized fluid delivered to chambers such as CH1 forces the chambers to expand, thereby inducing rotation of both rotors 6 and R about their respective centers CA and CB.

OBJECTS OF THE INVENTION

[0008] An object of the invention is to provide an improved cooling system for an automotive vehicle.

SUMMARY OF THE INVENTION

[0009] In one form of the invention, a cooling system for an automotive vehicle is as specified in claim 1.

BRIEF DESCRIPTION OF THE DRAWINGS

[0010]

Figures 1 and 2 illustrate a prior-art hydraulic machine 2, of the internal gear, one-tooth difference type.
Figures 3 - 6 illustrate a sequence of events occurring during rotation of motor 2 of Figure 1.
Figure 7 is a prior art figure illustrating a wall 18, which is part of a housing (not shown) containing the motor 2.
Figure 8 illustrates one form of the invention.
Figure 9 is a plan view of one form of the invention.
Figure 10 illustrates centers C1, C2, and C3, about which respective rotors OR, RR, and IR rotate.
Figures 11 - 18 illustrate a sequence of events occurring during rotation of the invention.
Figure 19 is a cross-sectional schematic view of the invention of Figure 10.
Figure 20 illustrates one embodiment of the invention, wherein block 58 represents a radiator in a motor vehicle.
Figure 21A illustrates another form of the invention;
Figure 21B is a view taken along the line 21B-21A in Figure 21B;
Figures 22 - 25 illustrate other forms of the invention; and
Figure 26 illustrates another form of the invention.

DETAILED DESCRIPTION OF THE INVENTION

[0011] Figure 8 illustrates one form of the invention, comprising an inner rotor IR, a ring rotor RR, and an outer rotor OR. Two gear sets, or sections, are present. The outer gear 33 of inner rotor IR and the inner gear 36 of ring rotor RR cooperate to form a first gear set S1, or first gerotor pair. The outer gear 27 of ring rotor RR and the inner gear 30 of outer rotor OR cooperate to form a second gear set S2, or second gerotor pair.

[0012] Both gear sets are shown as one-tooth difference type, but that type is not considered essential. Each gear set operates as a separate, though linked, hydraulic motor, or pump, depending on the mode of operation chosen.

[0013] A plate 37 contains ports HP1, HP2, LP1, and LP2, which deliver fluid to the two gear sets. Figure 9 illustrates the plate 37 in plan view. Two high-pressure ports, HP1 and HP2, deliver fluid to respective gear sets S1 and S2. Two low-pressure ports, LP1 and LP2, exhaust fluid from the respective gear sets S1 and S2.

[0014] In operation, outer rotor OR rotates about center C1 in Figure 10, as indicated by the arrow pointing to C1. Ring rotor RR rotates about center C2, and inner rotor IR rotates about center C3, both as indicated by arrows.

[0015] In actuality, the ring rotor RR would be sized so that point P1 would contact point P2, and the contact would act as a seal. Similarly, point P3 would contact point P4, for the same reason. However, for ease of generating drawings, in order to show the rotation which will now be discussed, these points P1 and P3 are shown separated from points P2 and P4.

[0016] Operation as a motor will now be explained. Figure 11 illustrates the starting position. Pressurized fluid is injected into chambers CH2 and CH3, through the ports HP1 and HP2 in plate 37 in Figures 8 and 9. In Figure 11, reference dots D3, D4, and D5 are added.

[0017] The pressurized fluid causes all rotors to rotate about their respective centers shown in Figure 10, as the sequence of Figures 11 through 18 indicates. The ratios of rotation are in proportion to the tooth ratios, and are 6/7 and 10/11. Thus, for every 7 revolutions of inner rotor IR, the ring rotor RR undergoes 6 revolutions with respect to the inner rotor IR. Similarly, for every 11 revolutions of ring rotor RR, the outer rotor RR undergoes 10 revolutions. Overall, a speed reduction occurs from inner rotor IR to outer rotor OR, in the ratio (6/7) x (10/11).

[0018] Figure 19 is a schematic cross-sectional view of the apparatus of Figure 8. Wall 37 is not a flat plate, but contains fluid conduits, and other apparatus. The motor operates under two speed conditions, using a single pressure source (not shown), applied to line 50. For high speed of shaft SH, displacement valve D is closed, thereby causing hydraulic fluid to be applied to port HP1 exclusively. Both rotors IR and OR rotate as shown in Figures 11 - 18, and at a relatively high speed and high pressure drop across the motor 2. This is called "single-displacement" mode.

[0019] A check valve CK is used during single-displacement mode. At this time, gear set S2 in Figure 9 is not used as a motor, so that set operates as a pump. The check valve CK allows oil being pumped by set S2 to flow in a continuous loop from outlet LP2 to inlet HP2, and at low pressure.

[0020] For relatively low speed of shaft SH, displacement valve D opens, based on a pressure differential sensed on lines L1 and L2 (or other measured parameter, such as engine speed, radiator fluid temperature, vehicle speed, and so on), and applies pressurized fluid to both ports HP1 and HP2. The same rotation occurs as shown in Figures 11 - 18, but now at a lower speed and with the same flow rate. That is, the same relative rotation of the three rotors IR, RR, and OR occurs, at the same ratio as before, namely, (6/7) and (10/11), but now at a lower speed, and lower pressure drop across the motor 2. This is called "dual-displacement" mode. Check valve CK is closed.

[0021] In one embodiment, the motor 2 in Figure 20 is used to drive a fan 55 to cool a radiator 58, used in an automotive vehicle 62. Pressure is applied by an engine-driven pump (not shown), and the pressure reaching the motor 2 is controlled by a regulator (also not shown). The regulator provides the desired pressure to the motor. Such pumps and regulators are known in the art.

[0022] At low engine speeds, as in slow traffic, large cooling from fan 55 is required, so single-displacement mode is used, to provide high-speed operation of motor 2, at relatively high fluid pressure. At high engine speeds, as in highway driving, incoming ram air is sufficient to cool the radiator 58, so that low-speed operation of motor 2 is desired. Dual-displacement mode is used, to provide low-speed operation of motor 2, at relatively low fluid pressure.

[0023] Other modes of operation are contemplated. For example, at engine idling speeds, the motor 2 can operate in either single or dual-displacement mode, depending on the cooling requirements. As another example, when the vehicle tows a trailer, a high fan speed and pressure during dual displacement may be required, such as 3500 rpm at 1400 psi.

[0024] The selection between low- and high-speed operation is, as explained above, determined by displacement valve D in Figure 19. That valve can be controlled by a signal on an input line IN. Alternately, the fluid supplied on line 50 can be provided by a hydraulic pump which is driven by the engine (not shown) of the vehicle 62. The flow on line 50 will be closely proportional to the speed of the engine.

[0025] Thus, at low engine speeds, the valve D is designed to remain closed, thereby providing high speed of motor 2. As engine speed increases, the pressure in line L1 will increase. When the differential reaches a threshold, the valve D opens, thereby providing low speed of motor 2.

[0026] It should be understood that the preceding discussion illustrates a specific embodiment of the invention, and that other modes of operation can be , implemented within the scope of the appended claims.

[0027] Two examples of the two modes of operation are the following. The motor 2 is designed such that, in dual-displacement mode, it displaces 9,83 l (0.6 cubic inch) per revolution, written as 9.83 l/rev (0.6 cu. in./rev). In single-displacement mode, it displaces 4.1 l/rev (0.25 cu. in./rev).

[0028] One gallon of fluid occupies 3,785 l (231 cubic inches). Thus, two gallons occupy 7,57 l (462 cubic inches). For the motor 2 to consume 7,57 l (two gallons) per minute in single-displacement mode, 1848 revolutions per minute (rpm) are required: 462/0.25 = 1848. For the motor 2 to consume the same 7,57 l (two gallons) in dual-displacement mode, 770 rpm are required: 462/0.60 = 770.

[0029] Thus, for a given flow rate, two speeds are possible, by selecting between single- and dual-displacement modes. Further, in each mode, modulation is possible, by modulating the pressure applied to the motor.

[0030] The ratio of these two speeds is roughly two: 1848/770 or 2.4 to 1. If a fixed, single-displacement pump, of the prior art type, were used, then, to accomplish this change in speed, a corresponding change in displacement would be required. That is, if rotation at 770 rpm required two gallons per minute, then rotation at 1848 would require 2.4 x 7.57 l (2 gallons) per minute. The invention eliminates this requirement.

[0031] It is a fact that, in motor 2, torque produced equals displacement*pressure/constant, where the constant is 75.4 (for units in lb.ft, cu.in/rev and psi). Adding units:

$$\text{torque (lb. ft.)} = \text{displacement (cu. in./rev)} * \text{pressure (psi)}.$$

For a pressure of 6,9MPa (1,000 psi), the torques produced by single- and dual-displacement modes are the following:

dual: 0.6 * 1,000/75.4 = 10,8 Nm (7.95 lb. ft).

single: 0.25 * 1,000/75.4 = 4,5 Nm (3.31 lb. ft).

Alternate Embodiments

[0032] The two gear sets S1 and S2 may be constructed of four distinct gears, as shown in Figures 21A and 21B. The gears 27 and 36 are not carried by a single ring rotor RR as in Figure 8, but take the form of separate gears RR2 and RR1 in Figures 21A and 21B, bottom. The axial thicknesses T1 and T2 of the two pairs are shown, and need not be the same.

[0033] For example, in Figure 22, the gear RR2 is physically separate from gear RR1, and rests upon RR1 as indicated by the dashed lines in Figure 22.
Alternately, gear RR2 may occupy two axial regions, as shown in Figure 23. When inserted into gear RR1, gear RR2 may occupy the axial thickness T1, and also extend beyond T1 by the difference (T2 - T1), as shown in Figure 25.

[0034] It may be desirable to make gear RR1 thicker than RR2, as shown in Figure 24.

[0035] Inner gear RR2 may be constructed in a single piece, reducing the number of gears from four to three.

Additional Considerations

[0036] 1. The volume between the pair of gears 27 and 30 in Figure 8, which is displaced per revolution of rotor RR (with respect to rotor OR), depends on the shapes of the gear teeth, and is controllable. Similarly, the volume between the pair of gears 33 and 36, which is displaced per revolution of rotor IR (with respect to rotor RR), depends on the shapes of the gear teeth, and is also controllable.

[0037] In one embodiment, these volumes are designed to be identical. In another embodiment, the volumes are 4.9 ml (0.3 cubic inch) between gears 27 and 30, and 3,28 ml (0.2 cubic inch) between gears 30 and 33.

[0038] In another embodiment, the volume between the inner gears 36 and 33 is larger than that between gears 27 and 30. The physically larger gerotor pair displaces a smaller volume.

[0039] 2. The invention of Figure 20 provides a significant savings in energy, compared with other approaches. For example, one set of calculations shows that, if motor 2 delivers about 5.2kW (7 horsepower), then about 10,4kW (14 horsepower) in hydraulic fluid is required to be delivered to motor 2. That is, the motor 2 consumes 14 horsepower, and delivers 5,2kW (7 horsepower), for an efficiency of 50 percent. The efficiency exceeds 40 percent.

[0040] In contrast, clutch fans driven by the engine (not shown) are in widespread use to perform the function of motor 2. Many of them consume about 22,4 kW (30 horsepower), in order to deliver the same engine cooling capability. The efficiency is less than 25 percent.

[0041] 3. The pressure ratio HP1/LP1 need not be the same as the ratio HP2/LP2; the pressure ratios may be different.

Further, the pressures at ports HP1 and HP2 may be different.

[0042] 4. The invention can be used either as a motor or a pump. In motor operation, fluid pressure is converted into torque. In pump operation, torque is converted into fluid pressure. In both cases, a transfer between pressure and torque occurs.

[0043] In addition, in some instances, dual operation can occur. For example, gear set S1 in Figure 5 can act as a motor, and gear set S2 can act as a pump. In this case, port HP2 becomes a low-pressure port, and port LP2 becomes a high-pressure port.

[0044] The invention should be distinguished from gear systems, such as planetary gear systems, which contain lubricants. Because of factors such as viscosity and other fluidic effects, the lubricant exerts some forces upon the gears, and the gears also exert forces upon the lubricant. It could be said that a transfer between pressure and torque occurs.

[0045] However, any transfer of this type is of minor significance. No significant conversion between torque and these pressures occurs. "Significant" refers to a conversion rate exceeding 25 percent, so that, for example, over 25 percent of the energy contained in a given volume of fluid is converted into torque.

[0046] 5. In Figure 8, the rotors IR, RR, and OR contain axial faces A, which face in the axial direction (as viewed in Figure 8), that is in the direction axis 51 extends. Plate 37, when assembled to the motor, has a face F which is parallel to, and adjacent, the axial faces A.

[0047] 6. Figure 10 shows two pairs of gears: pair 27 and 30, which have 10 and 11 teeth, respectively, and pair 33 and 36, which have 6 and 7 teeth respectively. The tooth difference in each pair is one.

[0048] 7. The rotors in Figure 8 are substantially coplanar, and rotate about centers which have eccentricity, with respect to each other.

[0049] 8. Gerotors are commercially available. The following U.S. patents, assigned to Sumatomo Electric Company of Japan, describe approaches to designing gerotors: 4,504,202, 4,673,342, 4,657,492, 4,518,332. In addition, Sumatomo Electric designs gerotor motors and pumps to meet specifications provided by a purchaser.

[0050] 9. The invention provides a "dual-displacement" hydraulic machine. One definition of "dual-displacement" is that, for a given machine speed, two selectable flow rates of fluid through the machine are available. Other definitions are possible.

[0051] 10. During both single and dual-displacement operation, the speed of motor 2 is infinitely variable between its minimum and maximum limits.

Claims

1. A cooling system for an automotive vehicle, comprising:

    a fan (55) and a hydraulic motor (2) driving the fan and comprising a first gerotor (33,36);

    characterised in that said hydraulic motor (2) driving the fan is a dual-displacement hydraulic motor comprising a displacement valve (D) for selecting said dual-displacement hydraulic motor between:

    i) a low-displacement mode in which a given flow rate causes a relatively high motor speed; and
    ii) a high-displacement mode in which the given flow rate causes a relatively low motor speed;

    said dual displacement hydraulic motor further comprising:

    a second gerotor (27,30) that is substantially coplanar with said first gerotor, said first and second gerotors being coupled to a common drive shaft (54) that is coupled to said fan (55):

    said first gerotor (33,36) rotating about a first axis and said second gerotor rotating about a second axis, wherein said first axis is offset from said second axis;
    herein said displacement valve (D) controls hydraulic fluid in said hydraulic motor to select between one of said first or second gerotors to both of said first and second gerotors when it is desired to drive said fan between said relatively high motor speed and said relatively low motor speed, respectively.

2. System according to claim 1, and further comprising means for selectively adjusting pressure or flow delivered to the motor in each mode.

3. The cooling system as recited in claim 1 wherein said system further comprises:

a regulator for modulating the pressure applied to said dual-displacement hydraulic motor.

4. The cooling system as recited in claim 1 wherein said first gerotor (33,36) comprises
a first thickness and said second gerotor (27,30) comprises a second thickness; said first and second thickness being the same.

5. The cooling system as recited in claim 1 wherein said first gerotor comprises
a first thickness and said second gerotor comprises a second thickness; said first and second thickness being different.

6. The cooling system as recited in claim 1 wherein said displacement valve (D) directs fluid to either one or both of said first and second gerotors when high or low cooling, respectively, by said fan is desired.

7. The cooling system as recited in claim 1 wherein said first and second gerotors comprise three distinct gears.

8. The cooling system as recited in claim 1 wherein said first and second gerotors comprise four distinct gears (27,30,33,36).

9. The cooling system as recited in claim 1 wherein said dual displacement hydraulic motor generates at least 10,8 Nm (7.95 lb. ft.) torque during said high-displacement mode.

10. The cooling system as recited in claim 1 wherein said dual displacement hydraulic motor generates at least 0,5 Nm (3.31 lb. ft.) torque during said low-displacement mode.

Patentansprüche

1. Kühlsystem für ein Kraftfahrzeug, mit :

ein Gebläse (55) und einen das Gebläse antreibenden hydraulischen Motor (2) und mit einem ersten Getriebe-Rotor (33, 36), **dadurch gekennzeichnet, dass** der genannte hydraulische Motor (2) ein hydraulischer Motor dualer Verschiebung mit einem Verschiebungsventil (D) zur Auswahl des genannten hydraulischen Motors dualer Verschiebung zwischen

i) einem niedrigen Verschiebungsmodus ist, in dem der bestimmte Durchsatz eine relativ hohe Motorgeschwindigkeit verursacht; und
ii) einem hohen Verschiebungsmodus ist, in dem der bestimmte Durchsatz eine relativ niedrige Motorgeschwindigkeit verursacht;

wobei der genannte hydraulische Motor dualer Verschiebung weiterhin Folgendes umfasst:

einen ersten Getriebe-Rotor (33, 36)
einen zweiten Getriebe-Rotor (27, 30), der im Wesentlichen mit dem ersten Getriebe-Rotor koplanar ist, wobei die genannten ersten und zweiten Getriebe-Rotoren an einer gemeinsamen Hinterachse (54) angekoppelt sind, die an dem genannten Gebläse (55) angekoppelt ist;

wobei der genannte erste Getriebe-Rotor (33, 36) um eine erste Achse rotiert und der genannte zweite Getriebe-Rotor um eine genannte zweite Achse rotiert,
wobei die genannte erste Achse zur genannten zweiten Achse versetzt ist;
hierbei kontrolliert das genannte Verschiebungsventil (D) den hydraulischen Fluss im genannten hydraulischen Motor zur Auswahl zwischen dem genannte ersten oder dem genannten zweiten Getriebe-Rotor zu beiden der genannten ersten und zweiten Getriebe-Rotoren, wenn gewünscht wird, das genannte Gebläse zwischen der relativ hohen Motorgeschwindigkeit und der genannten relativ niedrigen Motorgeschwindigkeit jeweils anzutreiben.

2. System gemäß Anspruch 1, weiterhin umfassend Mittel zur selektiven Anpassung des Drucks oder des Flusses, der in jedem Modus für den Motor geliefert wird.

3. Kühlsystem gemäß Anspruch 1, in dem das genannte System weiterhin Folgendes umfasst:

eine Reguliereinrichtung zum Modulieren des auf den genannten hydraulischen Motor dualer Verschiebung ausgeübten Drucks.

4. Kühlsystem gemäß Anspruch 1, in dem der genannte erste Getriebe-Rotor (33, 36) eine erste Dicke umfasst und der genannte zweite Getriebe-Rotor (27, 30) eine zweite Dicke umfasst; wobei die genannte erste und zweite Dicke identisch sind.

5. Kühlsystem gemäß Anspruch 1, in dem der genannte erste Getriebe-Rotor eine erste Dicke umfasst und der genannte zweite Getriebe-Rotor eine zweite Dicke umfasst; wobei die genannte erste und die zweite Dicke unterschiedlich sind.

6. Kühlsystem gemäß Anspruch 1, in dem das genannte Verschiebungsventil (D) den Fluss entweder zu einem oder zu beiden der genannten ersten und zweiten Getriebe-Rotoren leitet, wenn vom Gebläse jeweils eine starke oder schwache Kühlung gewünscht wird.

7. Kühlsystem gemäß Anspruch 1, in dem der genannte erste und zweite Getriebe-Rotor drei unterschiedliche Getriebe umfassen.

8. Kühlsystem gemäß Anspruch 1, in dem der genannte erste und zweite Getriebe-Rotor vier unterschiedliche Getriebe (27, 30, 33, 36) umfassen.

9. Kühlsystem gemäß Anspruch 1, in dem der genannte hydraulische Motor dualer Verschiebung während des genannten Modus mit starker Verschiebung wenigstens ein Drehmoment von 10,8 Nm - 7,95 libra Fuß - generiert.

10. Kühlsystem gemäß Anspruch 1, in dem der genannte hydraulische Motor dualer Verschiebung während des genannten Modus mit schwacher Verschiebung wenigstens ein Drehmoment von 4,5 Nm - 3,31 libra Fuß - generiert.

Revendications

1. Système de refroidissement d'un véhicule automobile, comprenant :

un ventilateur (55) et un moteur hydraulique (2) entraînant le ventilateur et comprenant une première pompe à rotor (33, 36), **caractérisé en ce que** ledit moteur hydraulique (2) entraînant le ventilateur est un moteur hydraulique à double cylindrée comprenant une soupape de cylindrée (D) pour choisir le fonctionnement dudit moteur hydraulique à double cylindrée entre :

i) un mode cylindrée faible dans lequel un débit donné entraîne une vitesse de moteur relativement élevée ; et
ii) un mode cylindrée élevée dans lequel le débit donné entraîne une vitesse de moteur relativement faible ;

ledit moteur hydraulique à double cylindrée comprenant en outre :

une première pompe à rotor (33, 36)
une seconde pompe à rotor (27, 30) qui est sensiblement coplanaire avec ladite première pompe à rotor, lesdites première et seconde pompes à rotor étant couplées à un arbre d'entraînement commun (54) qui est couplé audit ventilateur (55) ;
ladite première pompe à rotor (33, 36) tournant autour d'un premier axe et ladite seconde pompe à rotor tournant autour d'un second axe, ledit premier axe étant décalé par rapport au second axe ;

dans lequel ladite soupape de cylindrée (D) commande le fluide hydraulique dans ledit moteur hydraulique pour choisir l'une desdites première ou seconde pompes à rotor ou les deux des première et seconde pompes à rotor lorsqu'on souhaite entraîner ledit ventilateur entre ladite vitesse de moteur relativement élevée et ladite vitesse de moteur relativement faible, respectivement.

2. Système selon la revendication 1, comprenant en outre un moyen pour ajuster de façon sélective la pression ou le flux délivré au moteur pour chaque mode.

3. Système de refroidissement selon la revendication 1, dans lequel ledit système comprend en outre :

un régulateur pour moduler la pression appliquée audit moteur hydraulique à double cylindrée.

4. Système de refroidissement selon la revendication 1, dans lequel ladite première pompe à rotor (33, 36) comprend une première épaisseur et ladite seconde pompe à rotor (27, 30) comprend une seconde épaisseur ; lesdites première et seconde épaisseurs étant les mêmes.

5. Système de refroidissement selon la revendication 1, dans lequel la première pompe à rotor comprend une première épaisseur et ladite seconde pompe à rotor comprend une seconde épaisseur ; lesdites première et seconde épaisseurs étant différentes.

6. Système de refroidissement selon la revendication 1, dans lequel ladite soupape de cylindrée (D) dirige le fluide vers l'une ou les deux des première et seconde pompes à rotor lorsqu'on souhaite obtenir un refroidissement faible ou élevé, respectivement, par ledit ventilateur.

7. Système de refroidissement selon la revendication 1, dans lequel lesdites première et seconde pompes à rotor comprennent trois engrenages distincts.

8. Système de refroidissement selon la revendication 1, dans lequel lesdites première et seconde pompes à rotor comprennent quatre engrenages distincts (27, 30, 33, 36).

9. Système de refroidissement selon la revendication 1, dans lequel ledit moteur hydraulique à double cylindrée génère un couple d'au moins 10,8 Nm(7,95 lb-pi) pendant ledit mode cylindrée élevé.

10. Système de refroidissement selon la revendication 1, dans lequel ledit moteur hydraulique à double cylindrée génère un couple d'au moins 4,5 Nm (3,31 lb-pi) pendant ledit mode cylindrée faible.

FIG-1
(PRIOR ART)

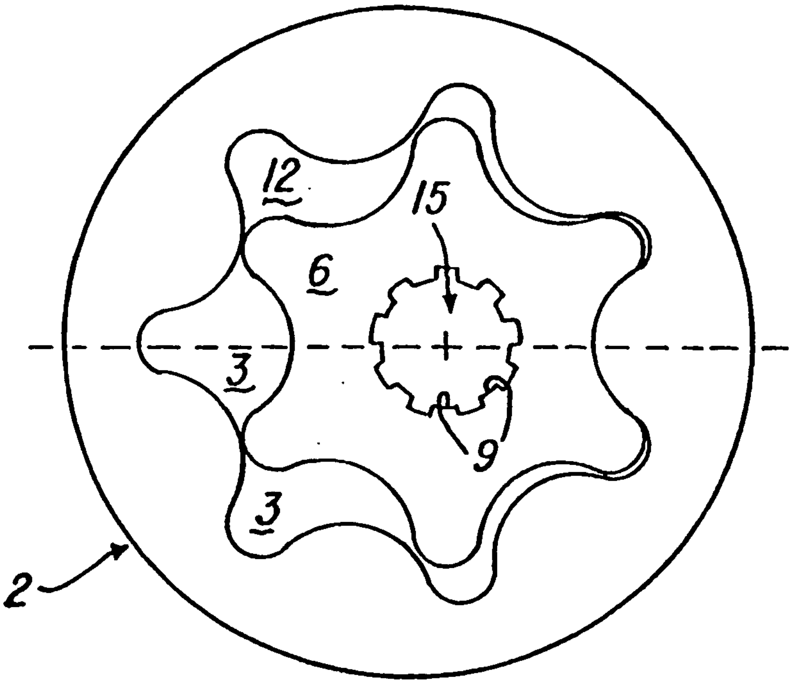

FIG-2
(PRIOR ART)

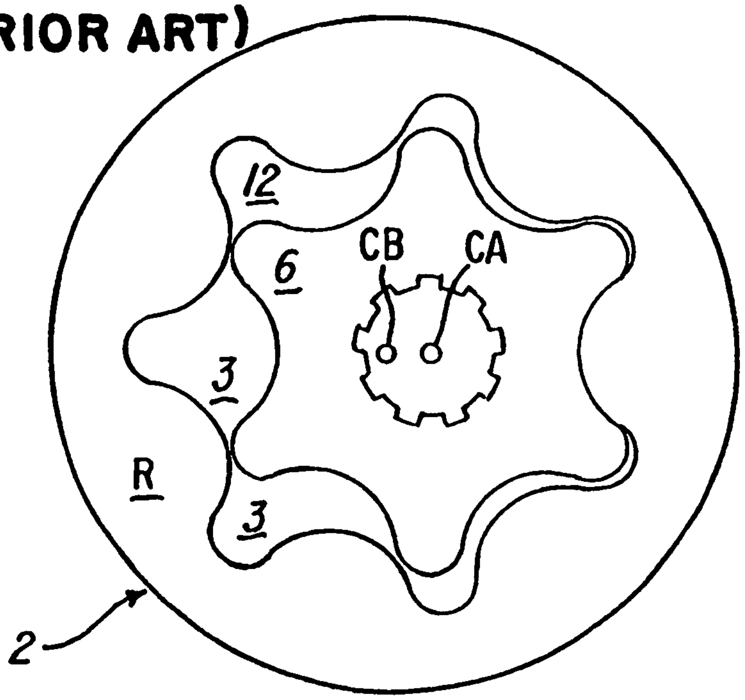

FIG-3
(PRIOR ART)

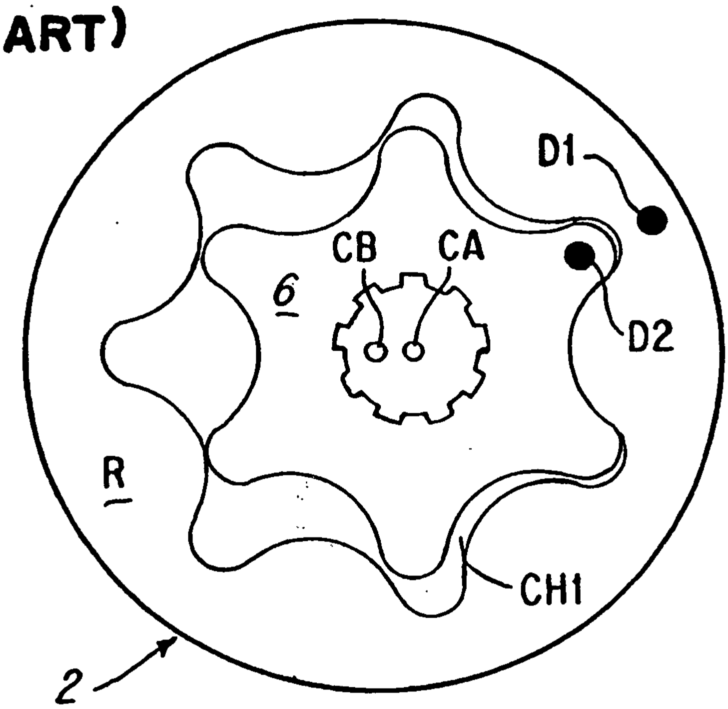

FIG-4
(PRIOR ART)

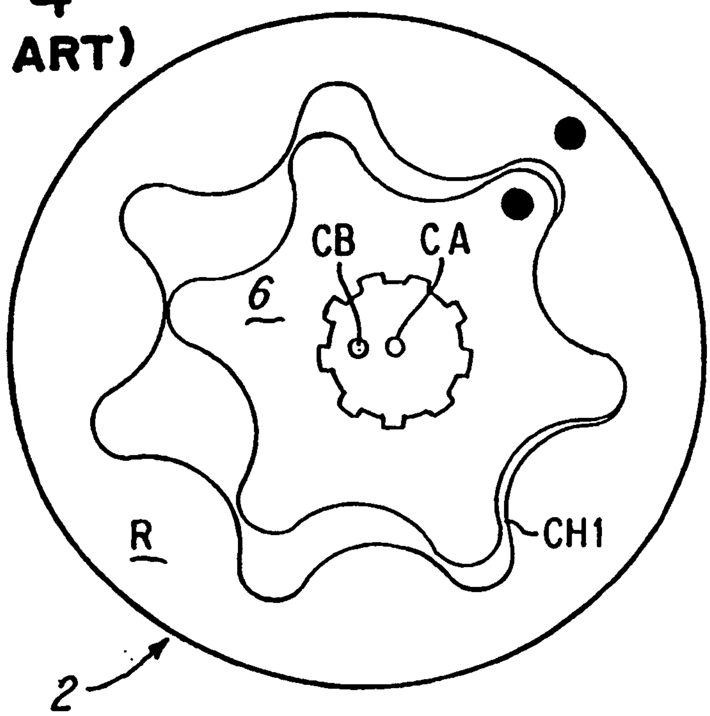

FIG-5
(PRIOR ART)

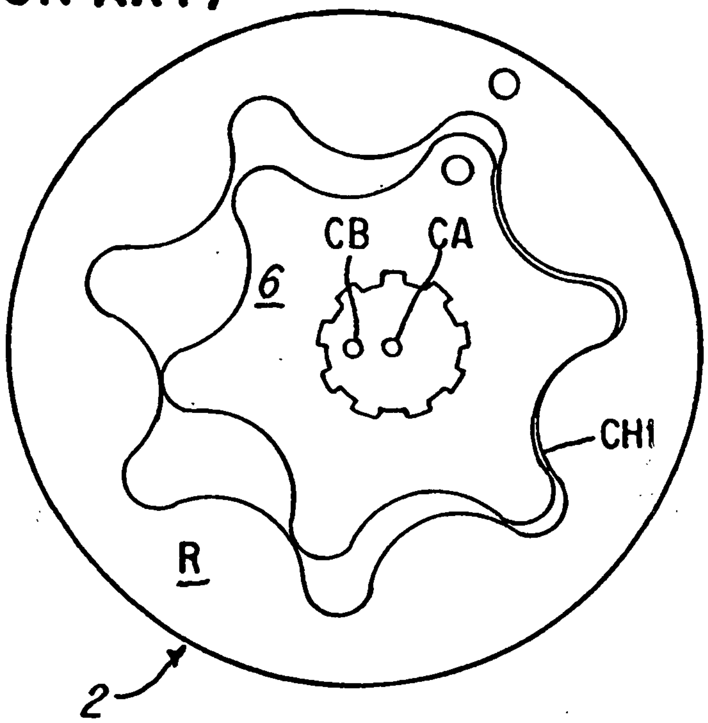

FIG-6
(PRIOR ART)

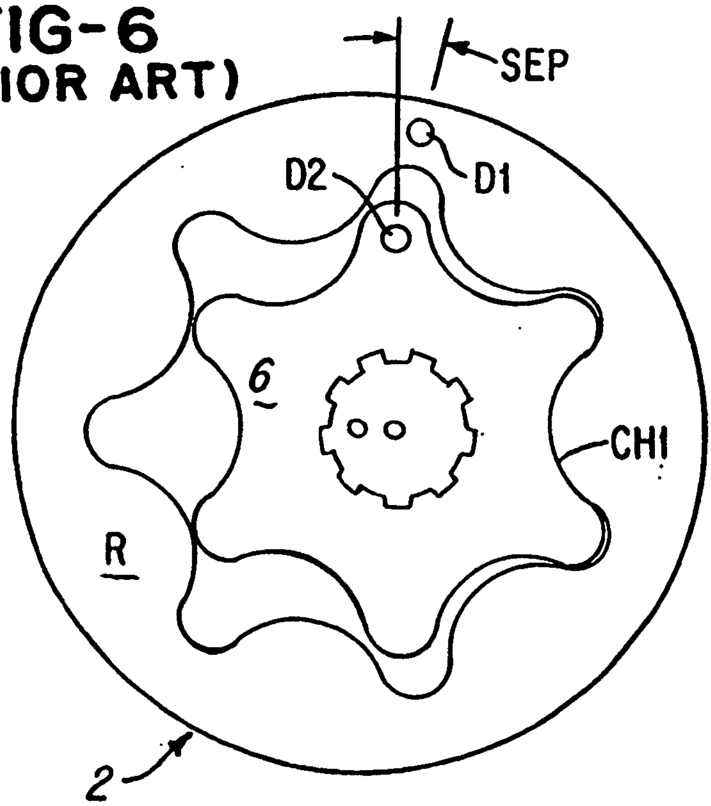

FIG-7
(PRIOR ART)

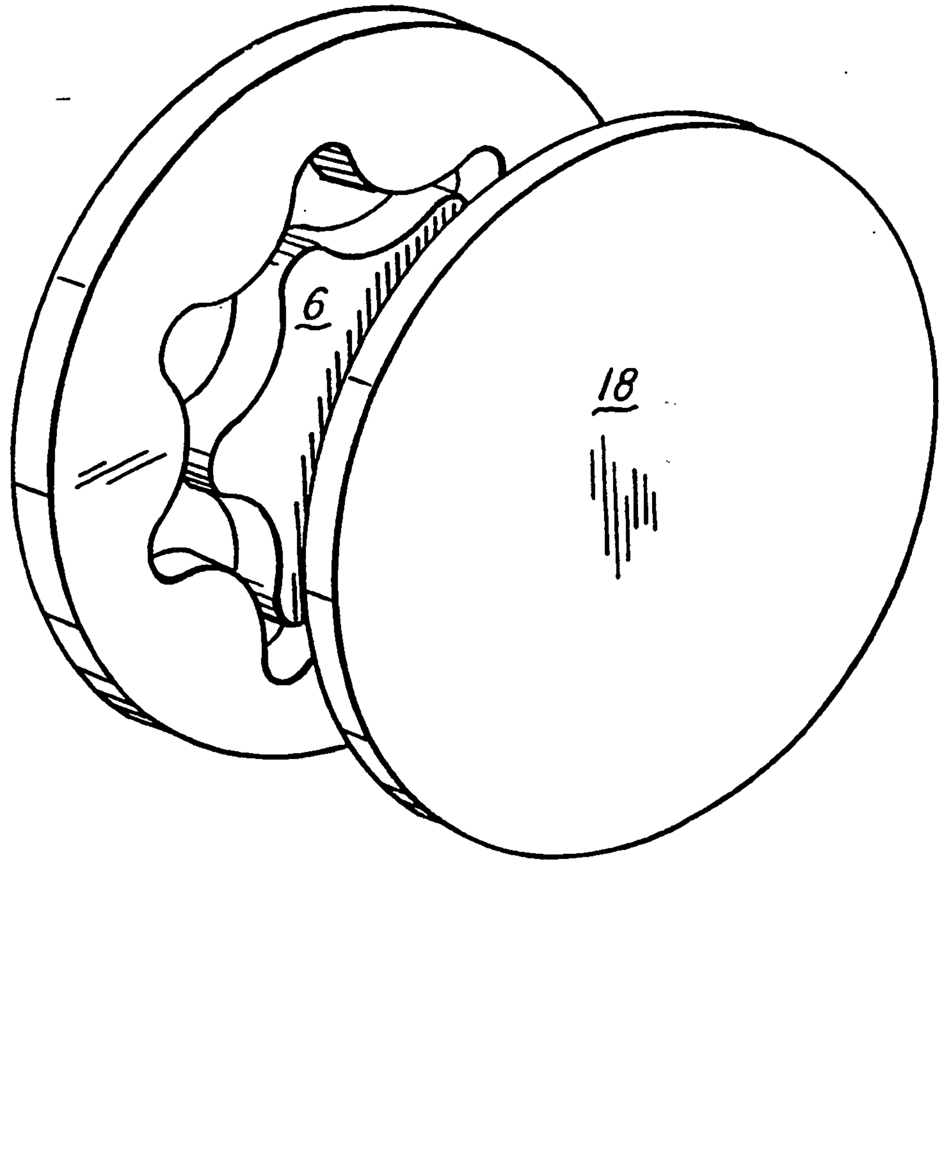

FIG-8

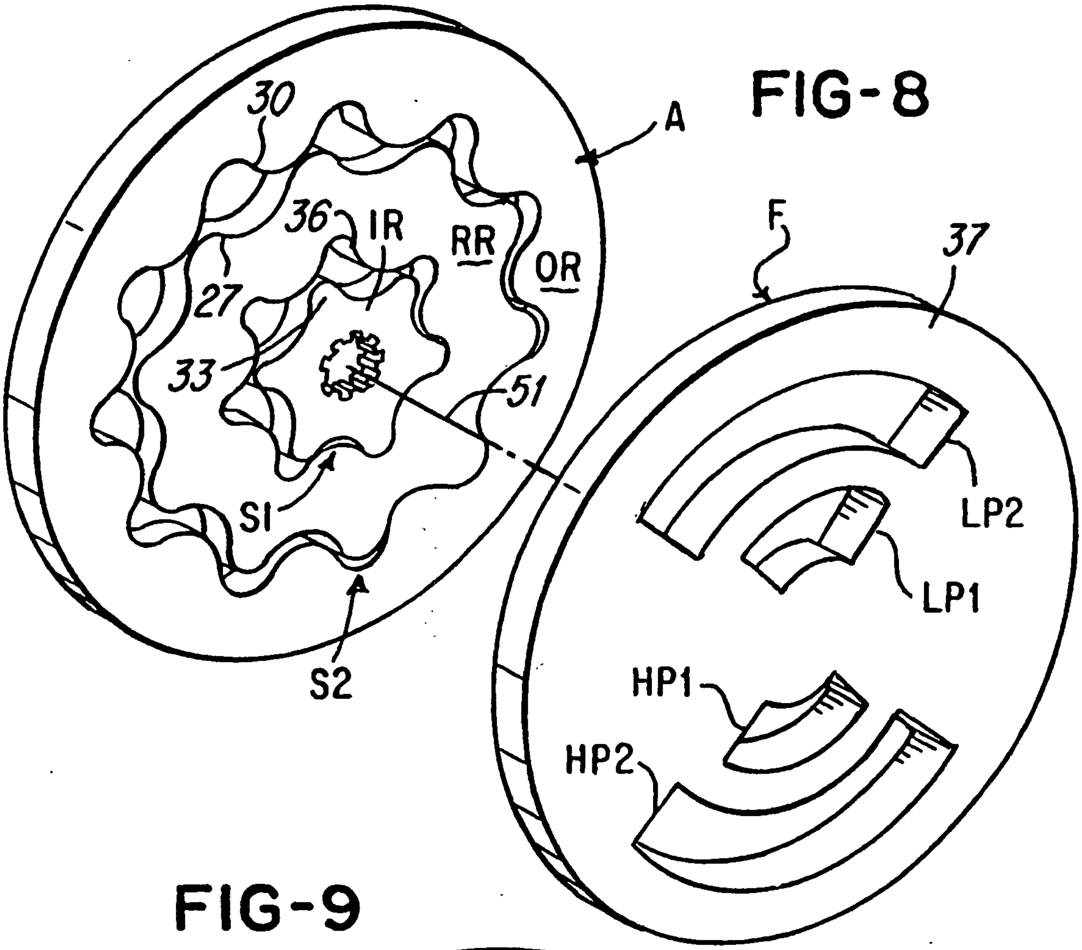
30
A
36
IR
RR
OR
27
33
51
S1
S2
F
37
LP2
LP1
HP1
HP2

FIG-9

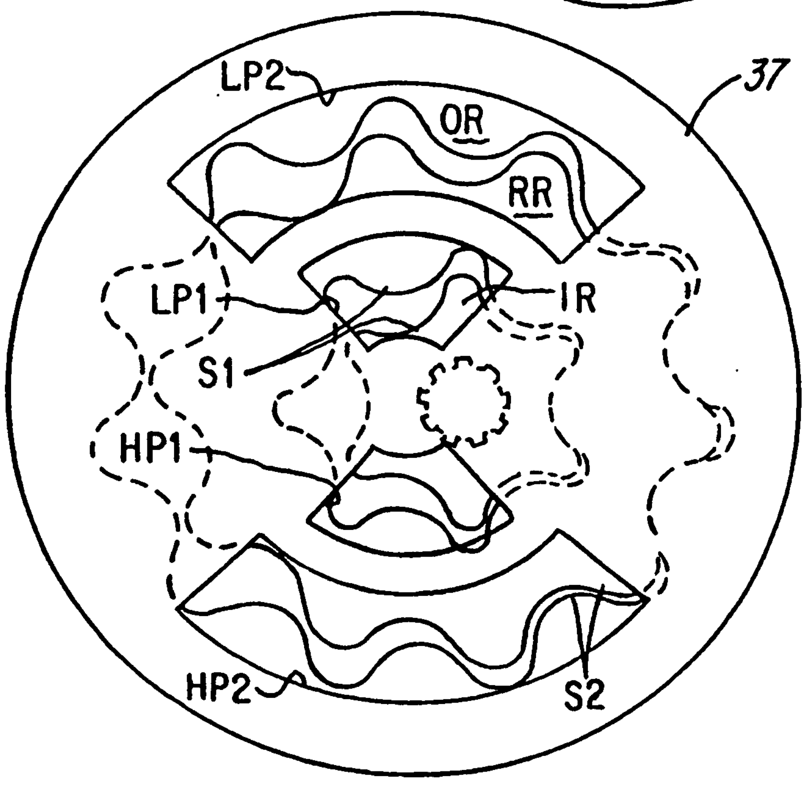
LP2
37
OR
RR
LP1
IR
S1
HP1
HP2
S2

FIG-10

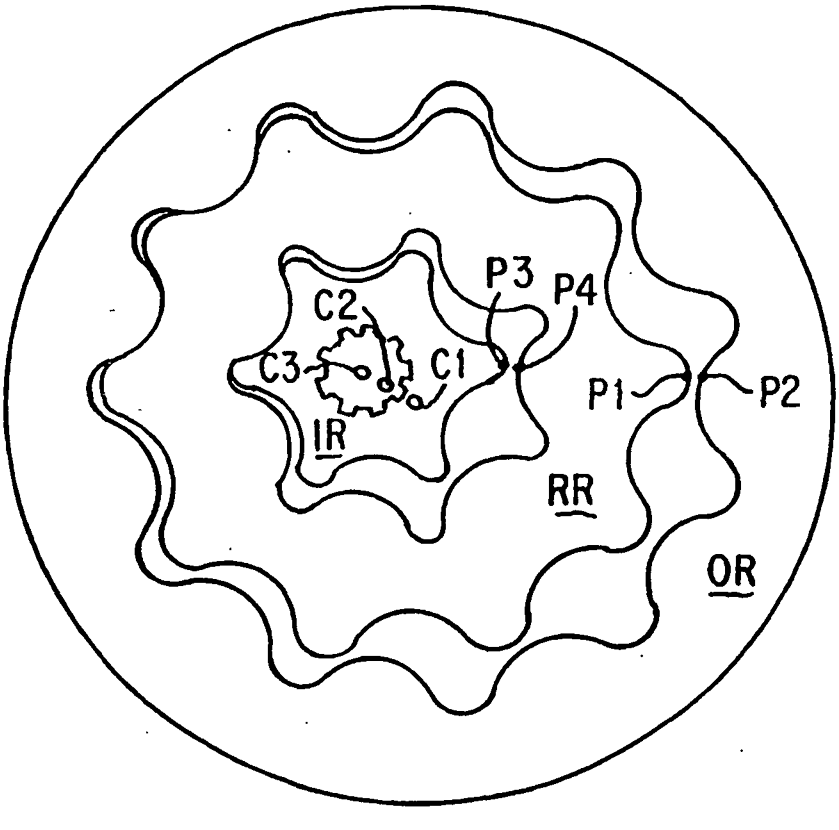

FIG-11

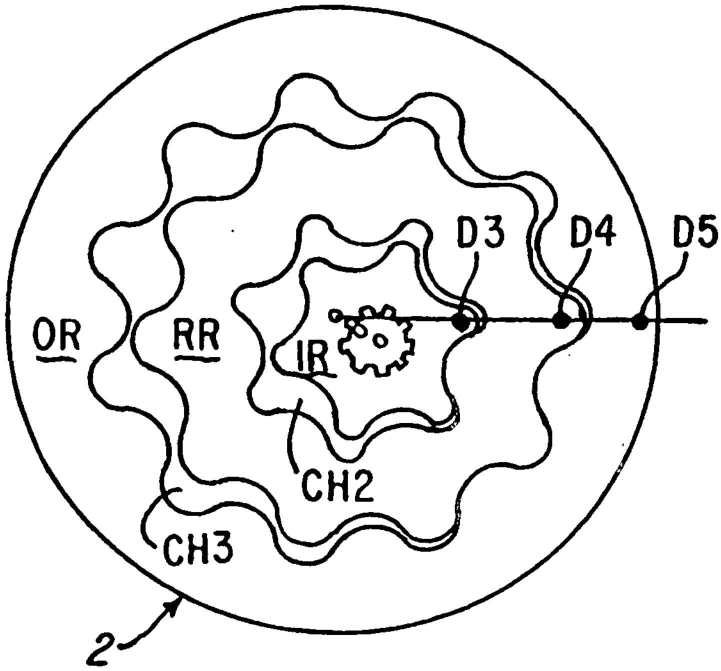

FIG-12

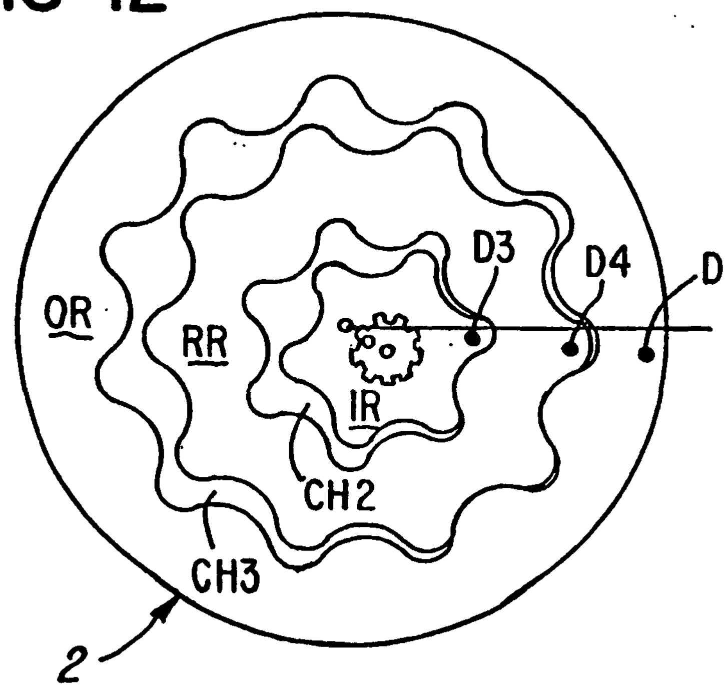

FIG-13

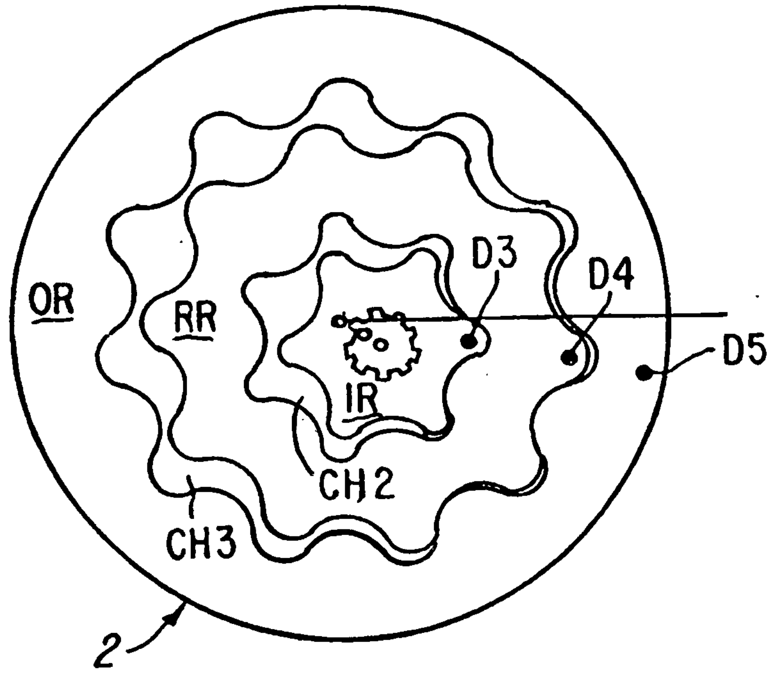

FIG-14

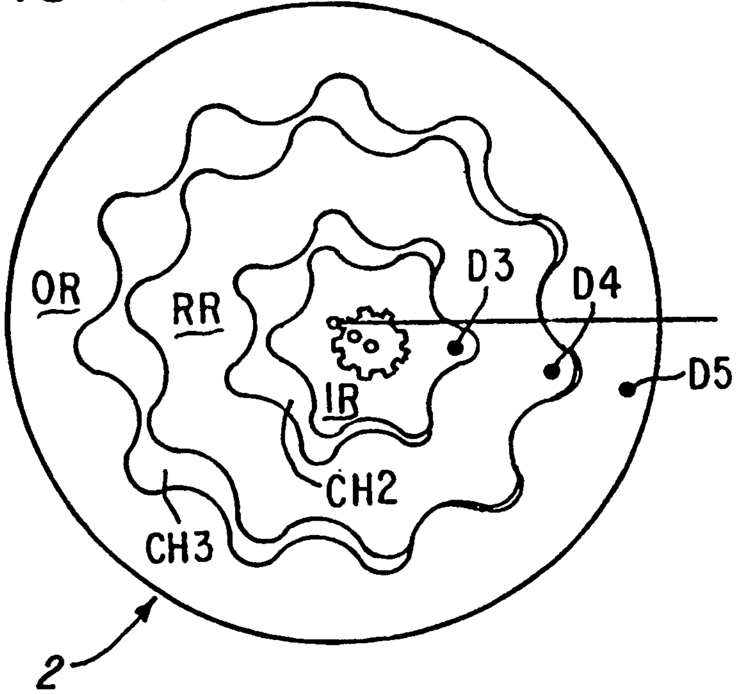

FIG-15

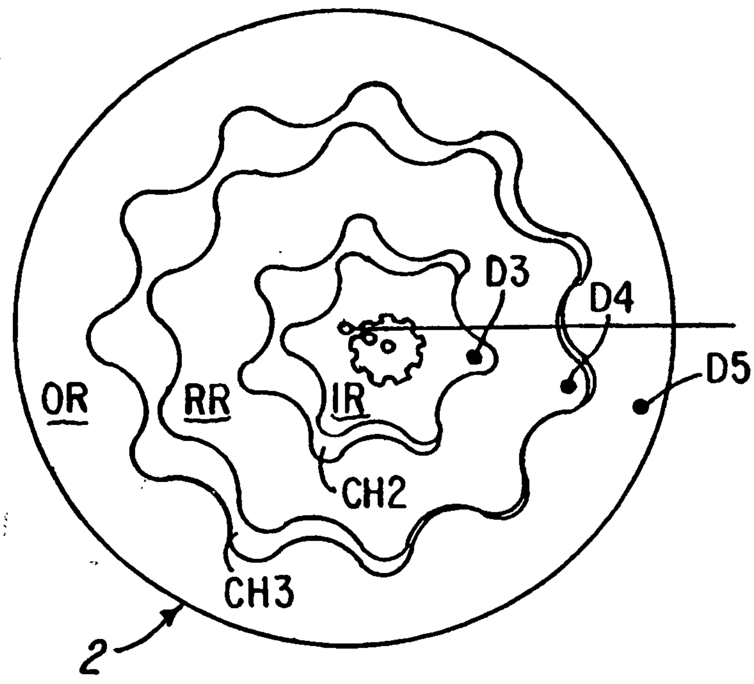

FIG-16

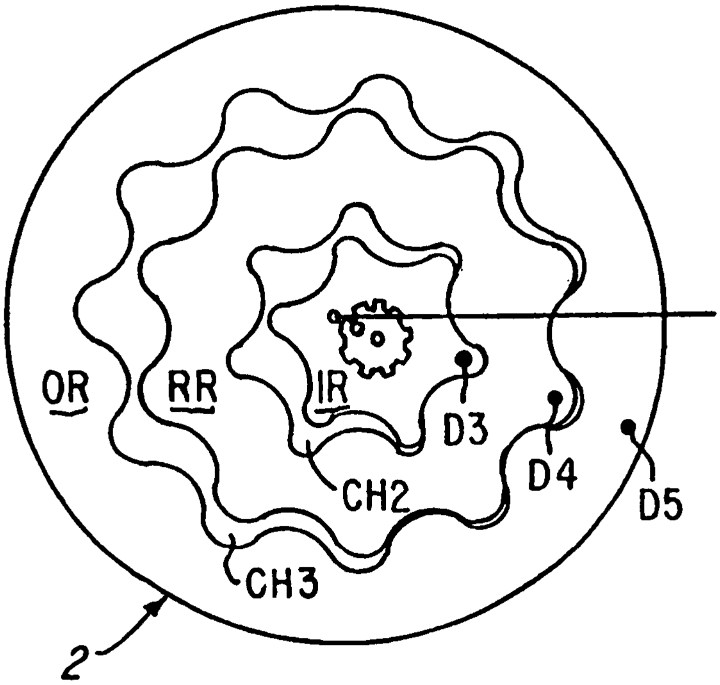

FIG-17

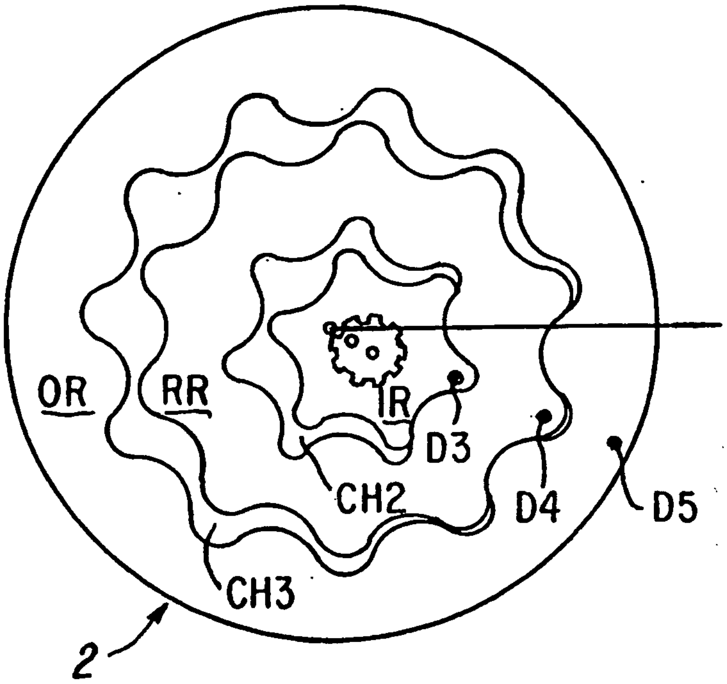

FIG-18

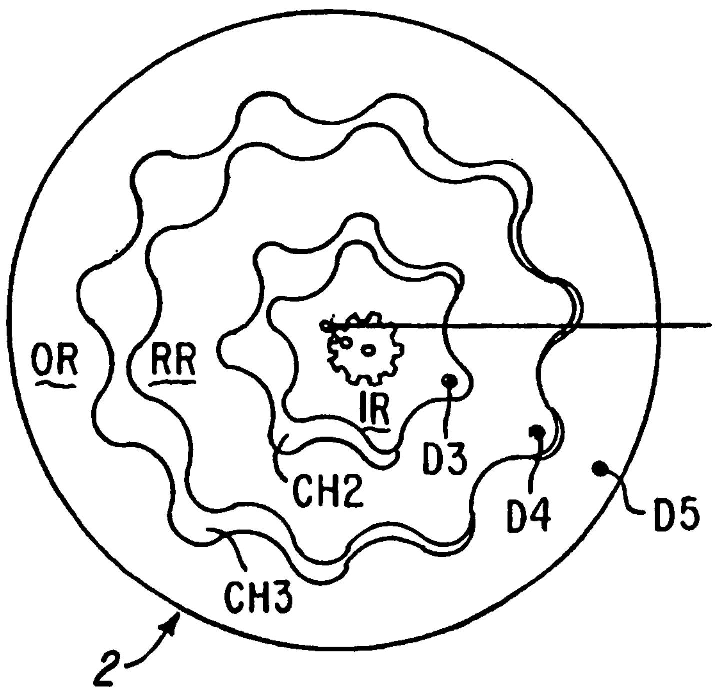

FIG-19

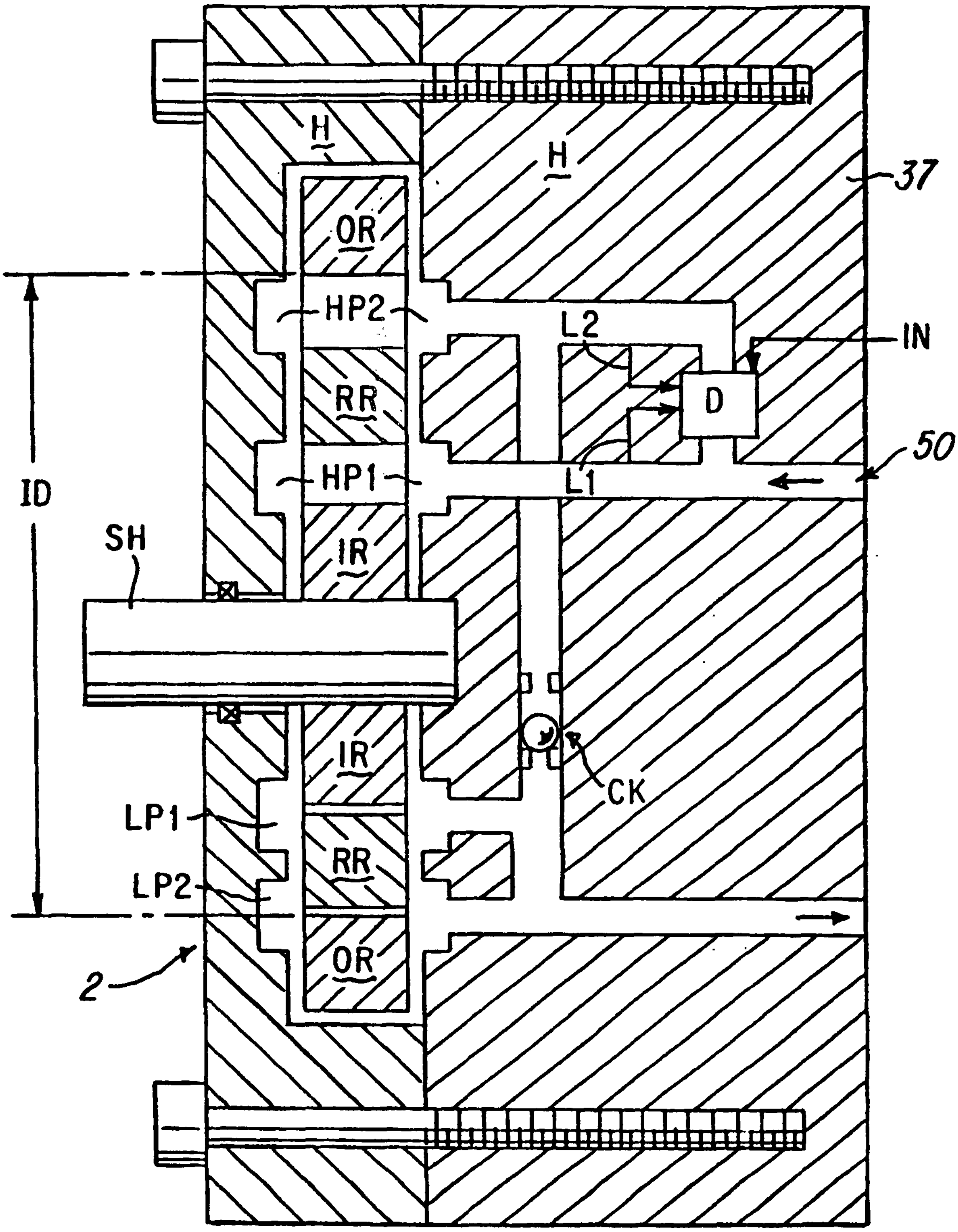

FIG-20

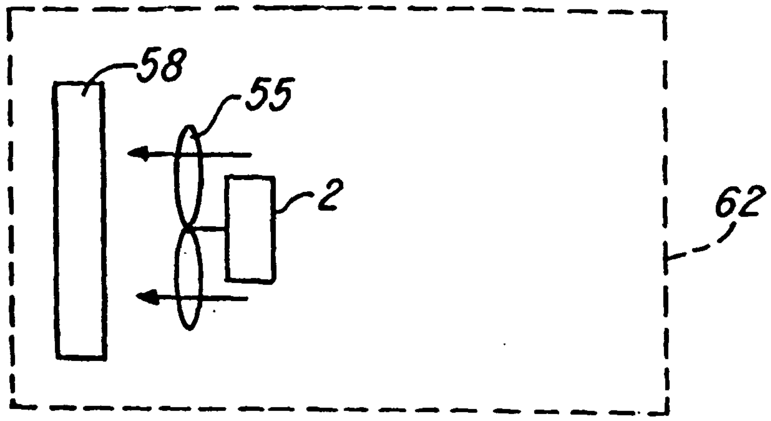

FIG-21A

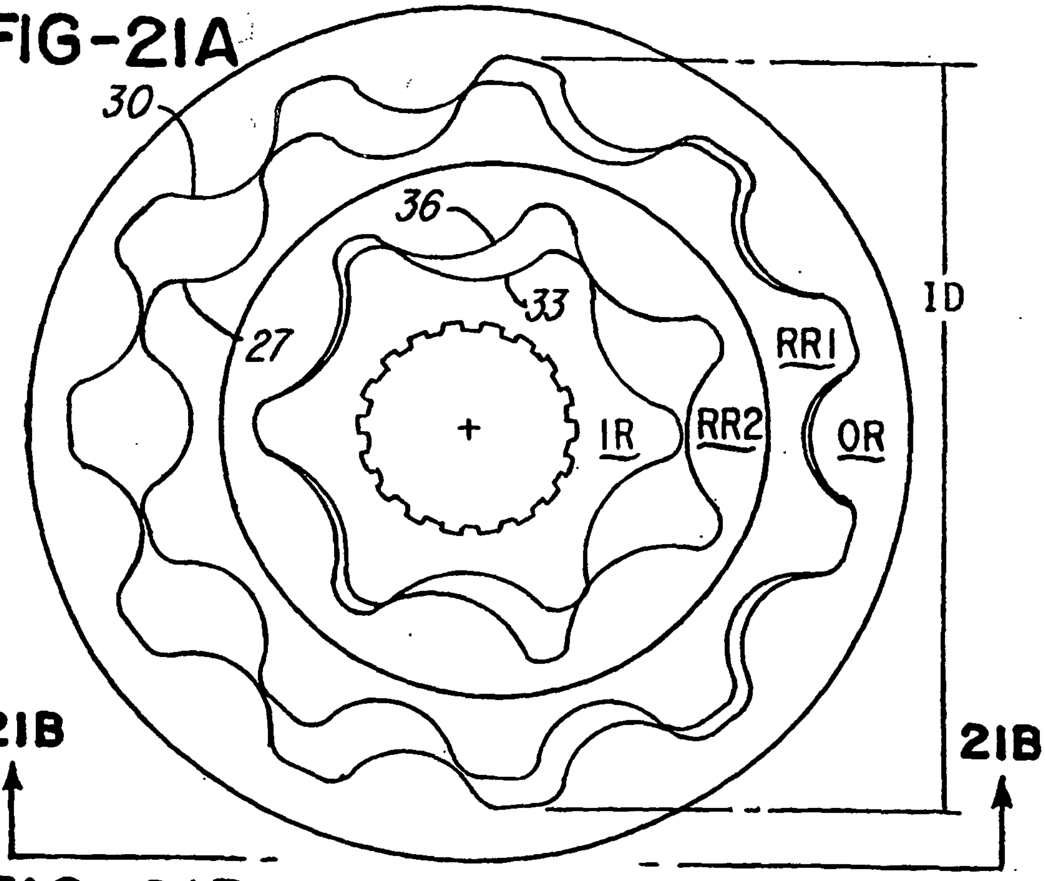

FIG-21B

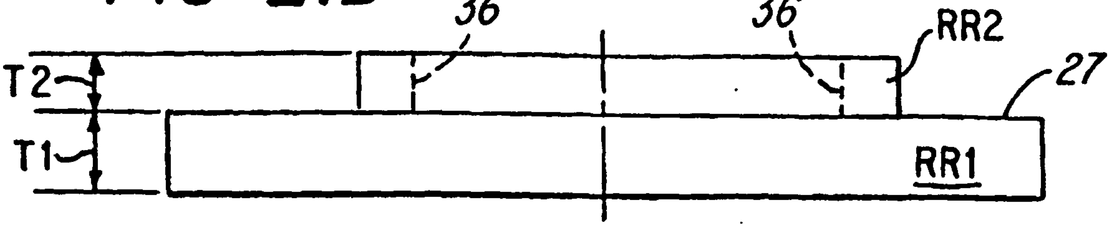

FIG-22

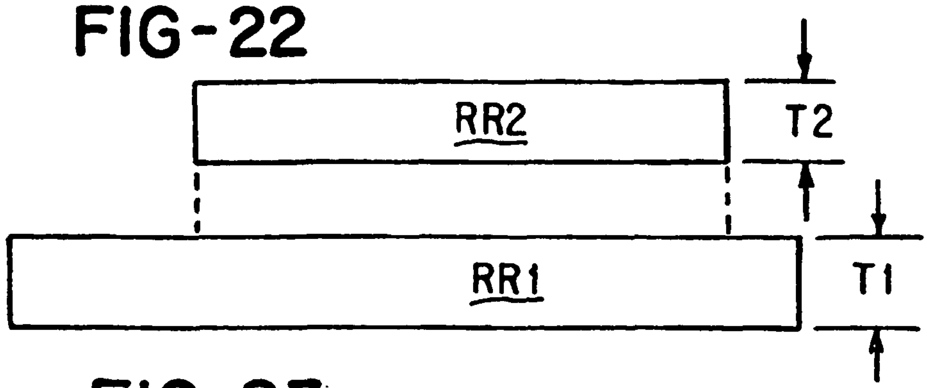

FIG-23

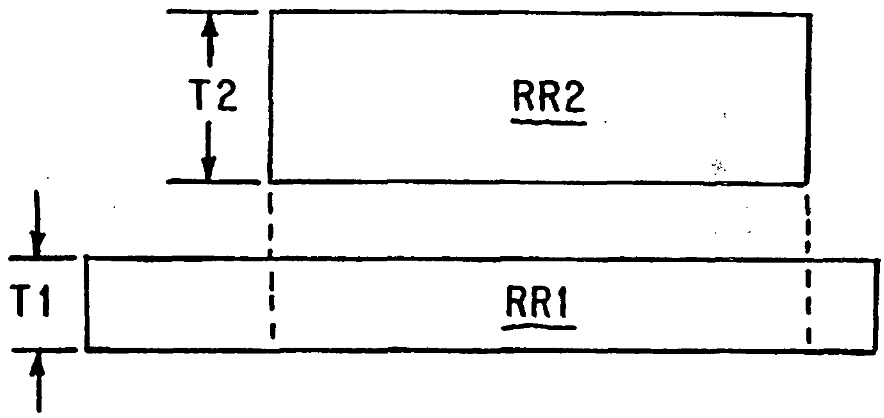

FIG-24

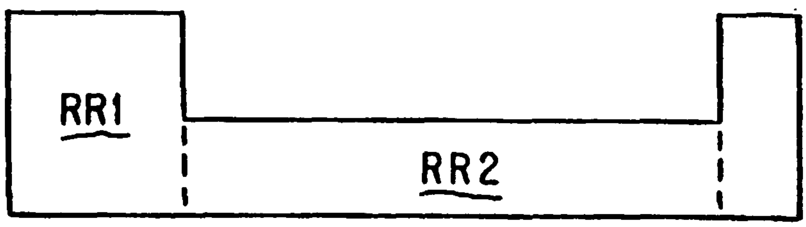

FIG-25

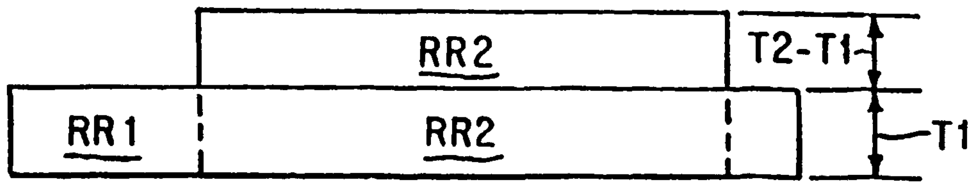

FIG-26